SARGENT & LUNDY
ENGINEERS
CHICAGO

SEISMIC SURVIVABILITY STUDY
FOR
MP-45 DIESEL GENERATORS

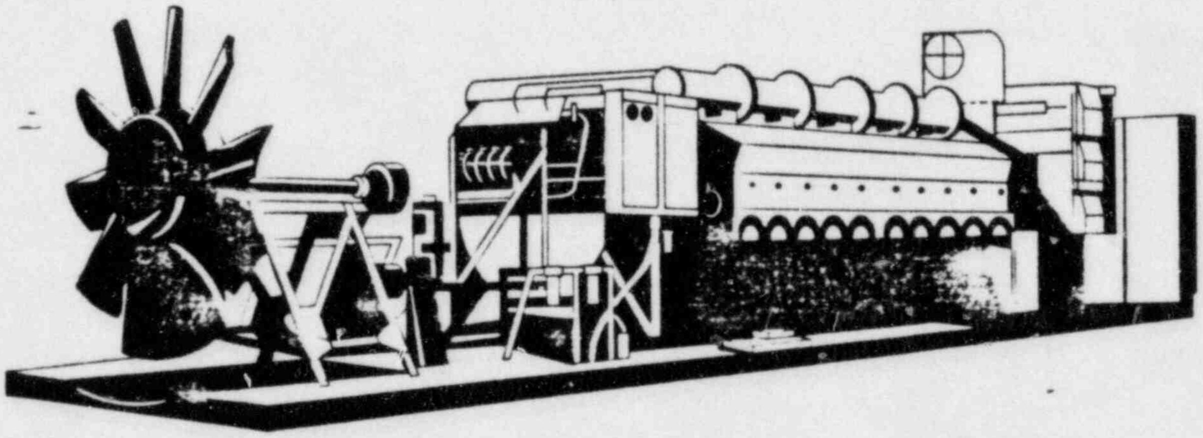
Prepared By
Component Qualification Division
For
Long Island Lighting Co.

Shoreham Nuclear Power Station
Project Number: 6995-00

Report No. CQD-014046
Rev. 00
June 12,1984

Shoreham
Nuclear Power Station

Seismic Survivability Study for MP-45 Diesel Generators



COMPONENT QUALIFICATION DIVISION

SARGENT & LUNDY
ENGINEERS

8001-1 05-84-272

ISSUE SUMMARY

COMPONENT QUALIFICATION DIVISION	SARGENT & LUNDY ENGINEERS	PROJ. NO.: 6995-00 CQD- 014046

REV	COMMENTS	RESPONSIBILITY	DATE
00	Original Issue	PREPARED BY: Dod Wright	6-12-84
		REVIEWED BY: Ismail KISISEL	6-12-84
		APPROVED BY: J Sinnappan	6-12-84
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		APPROVED BY:	

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PROJECT: SHOREHAM

PROJECT NO.: 6995-00

CALC. NO.: COD-014046 REVISION: 00

REVIEWED BY: Ismail Kinnez DATE: 6/12/84
(signature)

A. CONCLUSION OF REVIEW

☒ Accepted ☐ Rejected

Comments:

B. DESIGN INPUT DATA

B1. Has the input data been approved for use?

☒ Yes ☐ No

Comments: (SEE SECTION "E" OF THIS CHECK LIST)

B2. Is the input data applicable for this analysis?

☐ Yes ☐ No

Comments:

C. DOCUMENTATION

C1. Does the analysis include, as applicable, purpose, input data, assumptions, and references?

☒ Yes ☐ No

Comments:

C2. Is the analysis properly documented, in accordance with Quality Assurance Procedure GQ-3.08?

☐ Yes ☐ No

Comments:

D. TYPE OF ANALYSIS (Check one or both, as applicable)

☒ Hand-prepared design calculation

☒ Computer-aided design calculation

E. TECHNICAL ADEQUACY

E1. Assumptions. Are the assumptions used valid? ☒ Yes ☐ No

Comments:

MECHANICAL DEPARTMENT STANDARD

CHECKLIST FOR NONSTANDARD
NUCLEAR SAFETY-RELATED ANALYSES

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MAS-COD-2.1
Page 1 of 3

Form MAS-COD-2.1 Approved by Dept. Mgr.
Rev. Orig. (11-11-82)

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E2 Model. Is the analytical model(s) used adequate for this application?

☒ Yes ☐ No

Comments:

Complete E3 and/or E4, depending on the type of calculation.

E3. Hand-prepared calculation

E3.1 Method of review. The review was conducted using:

☒ A detailed review of the original calculation. (Detailed Dynamic Analyses)
☐ A review by an alternate, simplified or approximate method of calculation.
☒ A review of a representative sample of repetitive calculations. (Others)
☐ A review of the calculation against a similar calculation previously performed.

E3.2 Are the hand-prepared calculations technically adequate?

☒ Yes ☐ No

Comments:

E4 Computer-aided design calculation

E4.1 Program Acronym: SLSAP , RSG Program Number: 097130660, 097050560

E4.2 Run I.D.(s)/Date(s): See attachment to this checklist for the list (Page 1 of 1)

E4.3 Is the computer program applicable for this calculation?

☒ Yes ☐ No

E4.4 Is the program maintained in Computer Services Division-controlled files?

☒ Yes
☐ No-Has the program been validated in accordance with Appendix H of CSD Standards and Procedures Manual (GOP 4-1)?
  ☐ No
  ☐ Yes-a) Provide validation documentation file no.:
    b) Has it been documented that the program used is identical to the one validated?
    ☐ Yes ☐ No

E4.5 Is the computer program input correct?

☒ Yes ☐ No

E4.6 Does the program input contain sufficient accuracy to produce results within any numerical limitations of the program?

☒ Yes ☐ No

E4.7 Are the computer results consistent with the input?

☒ Yes ☐ No

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V

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E4.8 Are the results correct and within stated assumptions and limitations of the program?

☒ Yes ☐ No

E4.9 Are the computer-aided calculations technically adequate?

☒ Yes ☐ No

Comments:

F COMMENTS

Certain design input such as dimensions and weights, were obtained from the walkdown and vendor catalogs. In some instances, conservative estimates based on experience were used. Sources of input information is documented.

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MAS-CQD-2.1

Calc. No: CQD-04-046
Rev: 00 Date: 6-1-84
Proj. No: 6995-00
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Purpose

The purpose of this report is to present the findings of the seismic survivability study performed on the General Motors MP-45 diesel generators, at LILCo's Shoreham Station. Seismic survivability is defined as the ability of a mechanical or electrical component to remain structurally intact and operable following an SSE level earthquake. All of the essential components that are located in the system have been addressed in one of three sections of this report. Each section deals with a specific group or type of component.

Appendix B addresses all of the accessory components that are not an intricate part of the engine assembly. A list of all of the essential mechanical components that are located in the system can be found on pages 12 through 19. This list contains both the accessory components as well as the intricate engine components. Appendix D.1 addresses the engine assembly as a whole, as well as all of the components that are bolted directly to the block. A detailed list of these components is included on pages D.1.34 through 3.1.42.

The electrical switchgear and control instrumentation are addressed in Appendix C.2. A list of these components is also included in Appendix C.2 and can be found on pages C.2.3 through C.2.15.1. The equipment is separated and listed by panel.

Appendix C.2 contains a detailed finite element analysis of the "worst case" electrical panel. The analysis was used to develope elevated accelerations for the electrical devices mounted on the panels, as well as demonstrate that the panels will remain structurally intact when subjected to an SSE level loading. The results of the elevated response spectrum and base spectrum comparisons can be found in Appendix A.1.

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Conclusions

The findings from this study will be presented in two parts. The first part will address all of the mechanical equipment. This includes the engine, the intricately mounted engine components, and all of the remaining accessory components. The second part will deal with the electrical devices.

Mechanical Evaluation

The mechanical components were analyzed for the most part, in two sections of this report. Appendix B addresses all of the mechanical components that are not an intricate part of the engine assembly. This includes all of the components that are not directly bolted to the engine. A list of all of the accessory items can be found on pages 12 through 19. All of these items, with the exception of the radiator bracing, water jacket pipe support, weld repairs and panel supports; were found to be suitable to withstand an SSE level earthquake and remain operable following the event. These conclusions are based on an analysis or evaluation of each accessory item in the system. Modifications for those items listed above have been included in the recommendations section of this report. The equipment has been analyzed with the suggested modifications and found to be adequate.

The engine assembly and all of its intricate components were also found to be able to experience an SSE level earthquake and function properly following the event. This was demonstrated by a combination of test and analysis. All of the components that were considered to be part of the engine assembly are listed in Appendix D on pages D.1.34 to D.1.42.

Electrical Evaluation

The electrical components and devices that were found to have an existing seismic qualifications document that enveloped the required response spectrum for Shoreham, are qualified. A list of the devices and the method of analysis can be found on pages C.2.3 through C.2.15. The remaining electrical equipment was addressed using a study that was performed for the Nuclear Regulatory Commission entitled Subsystem Fragility. The study, NUREG/CR-2405, makes use of statistical techniques to obtain confidence levels in the equipment's ability to survive an earthquake on a generic basis. Using the study and applying it to the remaining components on the MP-45, we found that a 99% confidence level was reached. The study addresses both structural integrity and operability of the components. To further supplement the study, a detailed check of the mounting bolts on many of the instruments was performed. All of the bolts were found to be acceptable and would remain intact during an SSE event.

Recommendations

A - Radiator Support

The existing perforated channel sections running lengthwise on both sides of the center line were found to be overstressed for both OBE and SSE loads. Therefore, it is recommended that the addition of two 3 in. I-beams (S 3 x 7.5) be made. The additional I-beams should be attached to the side walls and placed in a position such that they support the perforated channels at their mid-spans. One of these new beams will be supporting the channel sections on the fan side span, and the other will be on the engine side span (see attached sketch). These additional supports will bring the stresses within the allowable limits for both OBE and SSE load cases.

B - Switchgear Cubicle 1A Door Panel

The auxiliary switchgear Cubicle 1A door panel is used to support a large number of electrical devices. As it stands, the panel was found to have predominant frequencies in the high energy range of the Shoreham spectrum. In order to control excessive amplification of ground motion it is recommended that the panel be stiffened. It was shown by analysis that the use of 2x2x1/4 in. angle sections will provide additional structural strength to the panel. These reinforcements can easily be mounted to the front of the panel. See attached sketch for the suggested locations.

C - Engine Jacket Water Pipe Supports

The 3-1/2" diameter engine jacket water pipe running diagonally with respect to the radiator units has two clamped supports; one near the fan side wall and the other near the middle beam. These items are corroded and should be replaced with a new and improved design.

D - Batteries

Gaps existing between the batteries and housing walls should be padded with elastic material preferably along the side ribs of the batteries. Complete filling of the gaps is not recommended, some allowance for air circulation should be made.

E - Radiator Compartment Walls

Excessive corrosion was observed at several locations on the radiator compartment walls and integral structural members. These corrosion products should be removed and the welds should be re-worked to assure structural continuity.

F - General Observations During Master Unit Walkdown

During the walkdown performed on the master unit (Engine No. 2), the aftercooler ducts were found to be cracked. The cracks had been welded to stop any propagation but had reopened. These cracks should be repaired again to ensure that further propagation will not occur. It should be noted that these cracks are in an air duct and would not in any way cause the failure of the generator

F - General Observations During Master Unit Walkdown (Cont'd.)

system to operate. Slight weld cracking was also noticed near the air turning box mounting brackets. These welds should also be repaired to ensure that structural integrity be maintained. It is recommended that a thorough inspection of all four of the engines be performed and that repairs be made as required.

G - Equipment Requiring Confirmatory Analysis

The equipment listed below is in need of further analysis to confirm the findings reached in this report. These are components that were determined to be acceptable based on sound engineering judgement. The items are as follows:

I) Generator Cooling System
II) Air Turning Box
III) Battery Charger
IV) Exciter
V) Shutter System
VI) Enclosure
VII) Electric Starter Motors
VIII) Cable Buss Assembly

It is also recommended that a detailed walkdown be performed on Units 1, 3 and 4 to ensure that the critical devices are similar to those found on the master unit.

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Calcs. For	
☐ Safety-Related	☐ Non-Safety-Related

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Client		Prepared by	Date
Project		Reviewed by	Date
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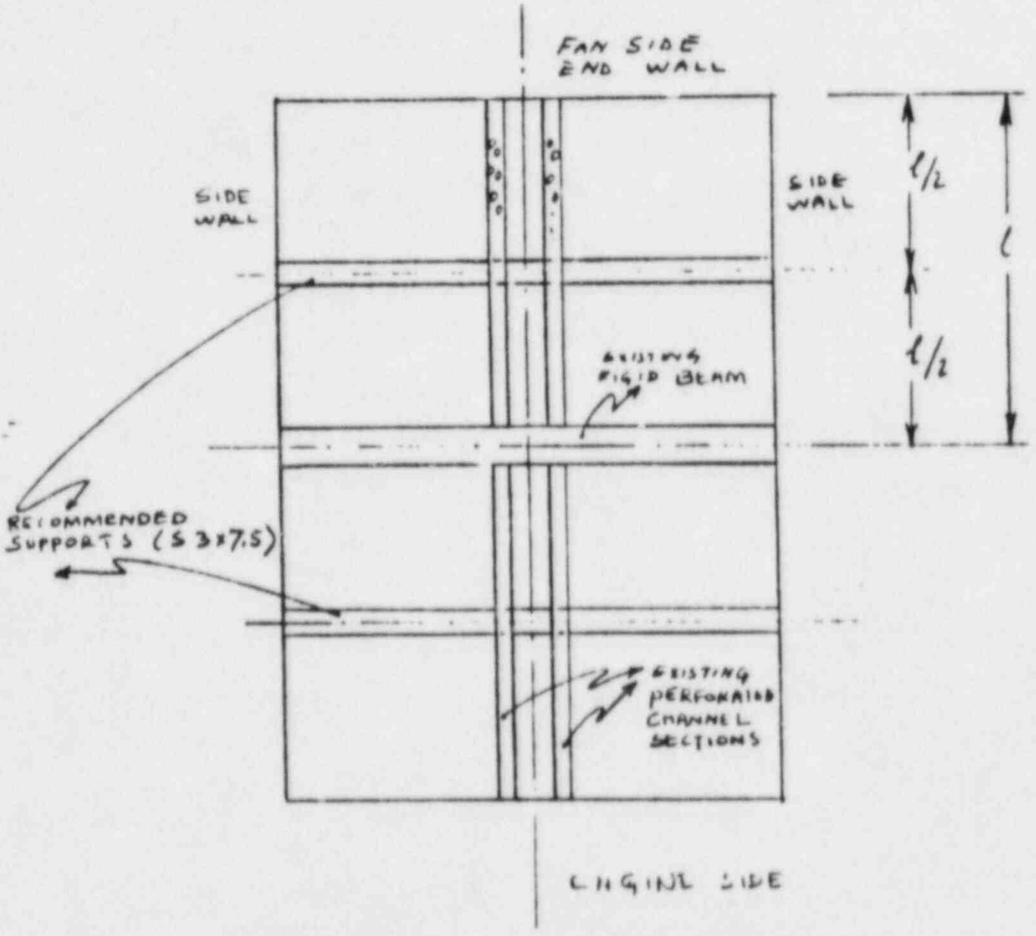
Radiator Support Modification



RADIATOR SUPPORT STRUCTURE

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Calcs. For

☐ Safety-Related ☐ Non-Safety-Related

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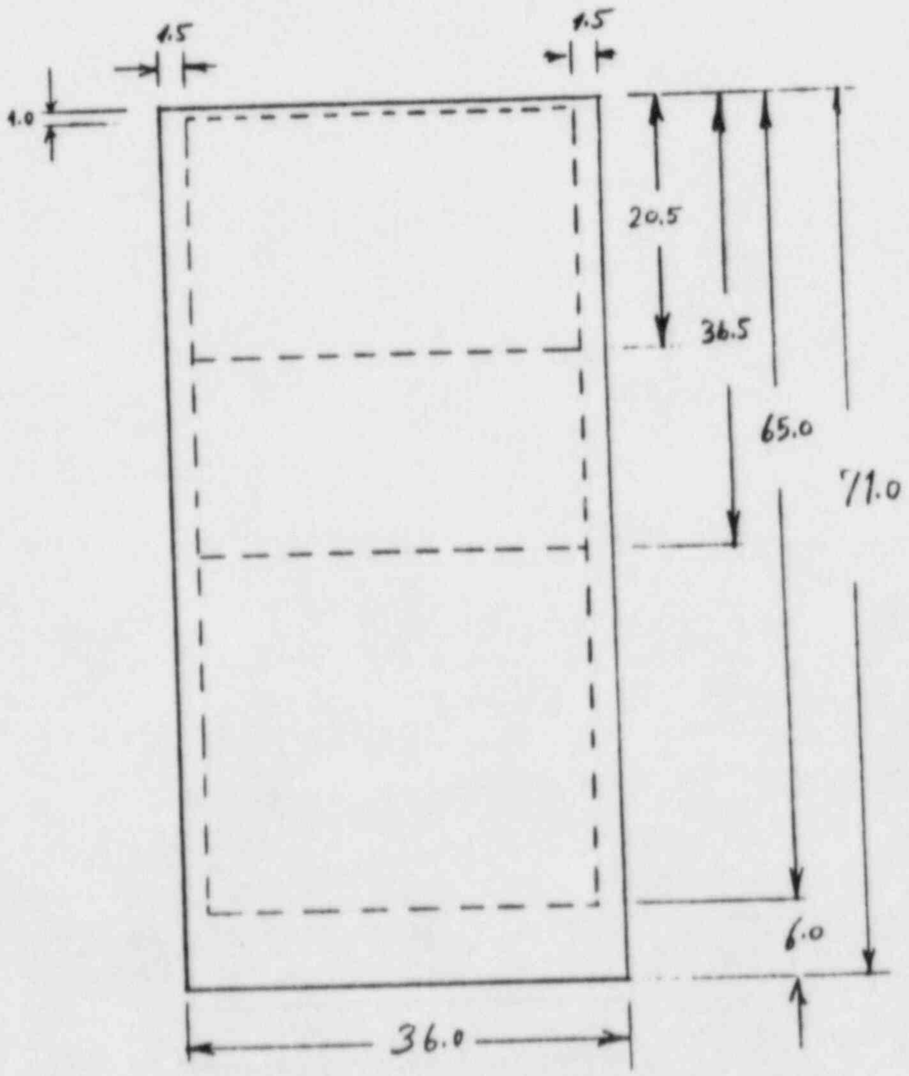
Project
Proj. No. 6995-00 Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

Electrical Panel Modification



---- Recommended Stiffeners
(L 2×2×1/4)
All dimensions are in inches

Form GO 3.08.1 Rev. 2

METHODOLOGY

- Develop floor response spectrum from the seismic time history provided by LILCO.
- Compare this floor response spectrum against floor response spectrum used in the seismic qualification of Emergency Diesel Generator System for LaSalle County Station. (It should be noted that the diesel engine used for LaSalle Station is same model as the one used in LILCO's Shoreham Station).
- Prove the seismic survivability of the GM-EMD engine by the methodology provided at the end of this section.
- Analyze all of the accessory components that are not an intricate part of the engine assembly by simple hand calculation using the upperbound response spectrum. Structural integrity and operability as required, were proven by this analysis.
- Perform detailed finite element analysis on the worst case electrical panel to prove the structural integrity of the panels. Also, obtain elevated response spectrum at device locations which will be used for the seismic evaluation of the electrical components.
- Compare and show these elevated response spectra are bounded by the elevated response spectra obtained for the panels in LaSalle County EDGS.
- Review the S&L data base to obtain seismic qualification documents for some of the electrical devices and reference this data.
- Provide a confidence levels for the seismic survivability electrical devices by use of the NUREG/CR2405 document.
- Provide recommendations on required modifications on the EDGS system.

EMD ENGINE EVALUATION

The engine evaluation will be performed using a combination of analysis and test. Use will be made of the existing work that has been performed by GM-EMD and outlined in their E77-1 and E77-1A reports.

I) Test

The engine block and all of its internal components will be qualified in conformance with IEEE-344 (1975), Section 7, paragraph 7.5 "Shock Testing." In this paragraph the standard states that shock testing performed in conformance with various military standards, "for example MIL-S-901C," that can be shown to have sufficiently high accelerations, "far higher than earthquake levels," and sufficient duration can be used for qualification purposes. The Navy shock tests referenced in GM's report E77-1 were carried out in conformance with MIL-S-901C. The acceleration levels measured during the test (at the control point) were 100 to 200 g's in the vertical and 45 to 120 g's in the horizontal. Required ZPA levels to cover the LaSalle, Clinton and Shoreham plants are 0.3 g's in the horizontal and 0.775 g's in the vertical. Along with the above-mentioned shock tests there is also an extensive amount of work done by SWRI*to determine resonant frequencies and operating loads for certain components. In this documentation it is shown that the engine is in fact a rigid structure, therefore, ZPA levels apply (this is true for the block and all internal components). Using both of the above-mentioned reports along with some simple analysis showing the magnitude of inertial loading during operation, a strong case can be made as to the adequacy of the engine block and internal components. An actual experience with running EMD engines during an earthquake of magnitude 6.2 on the Richter scale will also be utilized to further substantiate the adequacy of these engines under seismic loads.

II) Analysis

In order to supplement the shock test and address any external components attached to the engine which may have predominant frequencies below 33 Hz, the GM analysis outlined in E77-1A will be used. In this analysis detailed frequency calculations were performed and correlations made between the shock levels and given responses to an input of 3 g's in the horizontal and 1 g in the vertical. This is done for each item found to be in the flexible range.

It should be noted that in order for the engine and engine components to survive the normal operating loads of the engine, they have been designed to have frequencies higher than the engine vibration range and well above the seismic range. (Note - Previous finite element analysis has proven structural integrity of the engines).

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* Southwest Research Institute

L E G E N D

LOG NUMBER SCHEMA

WWXXYYZZZZ
EXAMPLE: LOG NUMBER 08 03 04 1400 REPRESENTS THE LILCO 20-CYLINDER ENGINE COOLING WATER EXPANSION TANK

WW - CLIENT IDENTIFICATION

08 - LONG ISLAND LIGHTING COMPANY
SHOREHAM PLANT

XX - MAJOR SYSTEM IDENTIFICATION

03 - 20 CYLINDER EMD-645 ENGINE
06 - GENERATOR
07 - AIR START

YY - SUBCOMPONENT AND/OR SYSTEM LEGEND

01 - SKID
02 - ENGINE
03 - ACCESSORY RACK
04 - COOLING WATER, RADIATOR SYSTEM
05 - NOT USED
06 - FUEL OIL SYSTEM
07 - GENERATOR
08 - EXCITER
09 - NOT USED

YY - SUBCOMPONENT AND/OR SYSTEM LEGEND (CONT'D)

10 - FAN
11 - RADIATOR
12 - RIGHT ANGLE GEAR DRIVE
13 - GEAR DRIVE COOLER
14 - INTAKE AIR SYSTEM
15 - EXHAUST GAS SYSTEM
16 - LUBE OIL SYSTEM
17 - BOOSTER PUMP SYSTEM
18 - CONTROL OR TRANSFORMER PANEL SYSTEMS
19 - DAY FUEL TANKS
20 - LOCAL INSTRUMENT OR ENGINE GAUGE PANEL
21 - COMMON SWITCHGEARS
22 - ELECTRIC START SYSTEM
23 - NOT USED
24 - BATTERIES
25 - BATTERY CHARGER
26 - JUNCTION BOX
27 - SWITCH BOX
28 - BRACING ASSEMBLY, PUMP DRIVE SYSTEM

ZZZZ - GENERIC COMPONENT INDENTIFICATION

SEE ITEMS LISTED INSIDE

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PROJECT NAME: LILCO 1
PROJECT NO. : 6995-00
CLIENT: LONG ISLAND LIGHTING CO.

SARGENT & LUNDY
COMPONENT QUALIFICATION DIVISION
STATUS OF EDGS MECHANICAL COMPONENTS
FOR LILCO MP-45 DIESEL GENERATORS

REVISION: 0
DATE: 12 JUN 84
PAGE 1 OF 7

LOG NUMBER	DESCRIPTION	PART NUMBER/ DRAWING NO.	VENDOR	LOC	PHOTOGRAPHIC REFEI NCE	PAGE REFERENCE	METHOD OF ANALYSIS
0803010000	SKID, 20 CYLINDER ENGINE		GM-EMD	ON		B.9.10	STATIC COEFFICIENT PER IEEE-344 (1975).
0803010100	ENCLOSURE,20 CYL.ENGINE		POWER SYS.	ON		B.8.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803020000	ENGINE	20-645E4	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803020100	GOVERNOR ASS'Y	EGB-10	WOODWARD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803020200	GOVERNOR DRIVE ASSEMBLY	8163123	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803020300	FLYWHEEL ASS'Y RINGGEAR & COUPLING DISCS		GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803020501	GUARD, FLYWHEEL, LHS	8361631	GM-EMD	ON		B.9.22	STATIC COEFFICIENT PER IEEE-344 (1975).
0803020502	GUARD, FLYWHEEL, RHS	8361632	GM-EMD	ON		B.9.22	STATIC COEFFICIENT PER IEEE-344 (1975).
0803030000	RACK, ACCESORY			ON	54	B.9.24	DETAILED DYNAMIC ANALYSIS.
0803040100	FLEX CONNECTION, ASST WATER JACKET PIPING	55381-3,4		ON		B.7.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040101	PIPING, SCH 40 ENGINE OUTLET TO RADIATOR	SKETCH	GM-EMD	ON	5,6,55,56	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040103	ADAPTER, WATER OUTLET ENGINE TO H-X PIPING	8363280	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803040104	PIPING, RADIATOR TO LUBE OIL COOLER		GM-EMD	ON	7	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040106	PIPING, COOLER TO WATER PUMPS (2) (TEE)	8367825 8359018	GM-EMD GM-EMD	ON	12	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040107	PIPING, DRAIN		GM-EMD	ON	8	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040108	PIPING, IMMERSION HEATER		GM-EMD	ON	9	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040109	PIPING, COOLING SYSTEM VENT		GM-EMD	ON	57	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040110	PIPING, EXPANSION TANK OVERFLOW		GM-EMD	ON	10,11	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040111	PIPING, EXPANSION TANK TO WATER PUMPS		GM-EMD	ON	12	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040401	SUPPORT,WATER JACKET PIPING IN RADIATOR ROOM		GM-EMD	ON			ENGINEERING EVALUATION- SEE RECOMMENDATIONS
0803040601	PUMP, WATER, LHS	8248341	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803040602	PUMP, WATER, RHS	8248341	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803040603	PIPING, WATER PUMP	8346691	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.

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PROJECT NAME: LILCO 1
PROJECT NO. : 6995-00
CLIENT: LONG ISLAND LIGHTING CO.

SARGENT & LUNDY
COMPONENT QUALIFICATION DIVISION
STATUS OF EDGS MECHANICAL COMPONENTS
FOR LILCO MP-45 DIESEL GENERATORS

REVISION: 0
DATE: 12 JUN 84
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LOG NUMBER	DESCRIPTION	PART NUMBER/ DRAWING NO.	VENDOR	LOC	PHOTOGRAPHIC REFERENCE	PAGE REFERENCE	METHOD OF ANALYSIS
	INLET ELBOW, LHS						
0803040604	PIPING, WATER PUMP INLET ELBOW, RHS	8346547	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803040701	AFTERCOOLER, LHS	8408491	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803040702	AFTERCOOLER, RHS	8408491	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803040703	PIPING, AFTERCOOLER WATER INLET, LHS	8352553	GM-EMD	ON		B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040704	PIPING, AFTERCOOLER WATER INLET, RHS	8352554	GM-EMD	ON		B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040705	PIPING, AFTERCOOLER WATER OUTLET, LHS	8352551	GM-EMD	ON		B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040706	PIPING, AFTERCOOLER WATER OUTLET, RHS	8352551	GM-EMD	ON		B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040707	FLANGE, AFTERCOOLER WATER OUTLET, 'Y'	8289021	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803040800	IMMERSION HEATER	73V1001	THERMOLINK	ON	58,9	B.3.16	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040900	TEMPERATURE CONTROL MANIFOLD	8314924	GM-EMD	ON	13,14,15	B.3.28	STATIC COEFFICIENT PER IEEE-344 (1975).
0803040901	BRACKET, TEMPERATURE CONTROL MANIFOLD	SKETCH	GM-EMD	ON		B.3.28	STATIC COEFFICIENT PER IEEE-344 (1975).
0803041003	FLEX CONNECTION, WATER PUMP INLET (2)	55381-4	AEROQUIP	ON		B.7.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803041004	FLEX CONNECTION, ENGINE OUTLET	55381-4	AEROQUIP	ON		B.7.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803041008	FLEX CONNECTION, EXPANSION TANK TO ASPIRATOR	8362588	GM-EMD	ON	12	B.7.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803041009	FLEX CONNECTION, OVERFLOW TEE	55381-1	AEROQUIP	ON	10	B.7.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803041201	VALVE, COOLING SYSTEM DRAIN	8354227	GM-EMD	ON	8	B.3.11	DETAILED DYNAMIC ANALYSIS.
0803041400	EXPANSION TANK	8366563 SKETCH	GM-EMD	ON	16,17	B.3.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803041401	EXPANSION TANK, SIGHT GLASS	8390299	GM-EMD (ESSEX BRASS)	ON		B.3.27	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060100	VALVE, FUEL CUT OFF	8175542 SKETCH	GM-EMD (HANCOCK)	ON		B.2.76	DETAILED DYNAMIC ANALYSIS.
0803060302	GAUGE & TUBING ASSEMBLY PUMP OUTLET PRESSURE		GM-EMD	ON		B.2.15	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060303	FUEL OIL TRANSFER PUMP, UPPER		VIKING	ON	20,21,22	B.2.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060304	FUEL OIL TRANSFER PUMP,		DELCO	ON	20,21,22	B.2.2	STATIC COEFFICIENT PER

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	UPPER						IEEE-344 (1975).
0803060309	FUEL OIL TRANSFER PUMP, LOWER		VIKING	ON	20,21,22	B.2.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060310	FUEL OIL TRANSFER PUMP, LOWER		DELCO	ON	20,21,22	B.2.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060311	FUEL OIL TRANSFER ASS'Y MOUNTING FRAME	SKETCH			20,21,22	B.2.9	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060500	STRAINER, FUEL SUCTION	152080	MARVEL ENGINE CO.	ON	24,25	B.2.55	OTHER-SEE REFERENCED SECTION
0803060501	STRAINER, FULE TRANSFER SYSTEM	152080	MARVEL ENGINE CO.		23	B.2.55	OTHER-SEE REFERENCED SECTION
0803060502	FILTER, FUEL TRANSFER SYSTEM (2)	1750-105A	COMMERCIAL FILTERS			B.2.46	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060600	PUMP, ENGINE DRIVEN FUEL	8410219	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803060701	PIPING, FUEL OIL PUMP TO MANIFOLD			ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803060702	PIPING, MANIFOLD TO MANIFOLD INLET			ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803060703	PIPING, MANIFOLD TO MANIFOLD OUTLET			ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803060708	PIPING, SUCTION STRAINER TO FUEL PUMP			ON	24,25	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060800	FUEL OIL MANIFOLD ASSEMBLY	8 9059 SKETCH	GM-EMD	ON		B.2.60	STATIC COEFFICIENT PER IEEE-344 (1975).
0803060900	FILTER, FUEL OIL ASSEMBLY	831121 SKETCH	GM-EMD	ON		B.2.60	STATIC COEFFICIENT PER IEEE-344 (1975).
0803061000	LINKAGE ASSEMBLY, INJECTOR CONTROL ROD		GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803061100	FUEL OIL CROSSOVER MANIFOLD ASSEMBLY					B.2.70	STATIC COEFFICIENT PER IEEE-344 (1975).
0803101000	FAN, RADIATOR COOLING	8-108-8AL	KOPPERS			B.3.50	STATIC COEFFICIENT PER IEEE-344 (1975).
0803101100	FAN SHAFT	11182-17				B.3.50	STATIC COEFFICIENT PER IEEE-344 (1975).
0803101110	DUNLAP SHUTTERS	9001	DUNLAP			B.3.63	ENGINEERING EVALUATION.
0803101120	SHUTTER CONTROL	MP-9750	BARBER COLEMAN			B.3.64	STATIC COEFFICIENT PER IEEE-344 (1975).
0803101130	TEMPERATURE CONTROLLER	TP-222	BARBER COLEMAN			C.2	ENGINEERING EVALUATION.
0803101140	REAR SUPPORTS, SHAFT	SKETCH				B.3.60	STATIC COEFFICIENT PER IEEE-344 (1975).
0803101150	FRONT SUPPORTS, SHAFT	SKETCH				B.3.50	STATIC COEFFICIENT PER IEEE-344 (1975).

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0803110000	RADIATOR	ESL-150-518	D & M			B.3.35	STATIC COEFFICIENT PER IEEE-344 (1975).
0803140100	FILTER, AIR CLEANER ASSEMBLY	SKETCH	FARR	ON	29,30	B.5.20	ENGINEERING EVALUATION.
0803140400	ENGINE AIR TURNING BOX	SKETCH		ON	29,30	B.5.23	ENGINEERING EVALUATION.
0803140601	TURBOCHARGER		GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803140701	LHS AFTER COOLER DUCT	8460334	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803140702	RHS AFTERCOOLER DUCT	846C309	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150100	EXHAUST, SILENCER	23-3902	UNIVERSAL	OFF		B.5.8	STATIC COEFFICIENT PER IEEE-344 (1975).
0803150200	EXHAUST, DIFFUSER ADAPTOR	8366591	GM-EMD	ON		B.5.17	STATIC COEFFICIENT PER IEEE-344 (1975).
0803150201	TURBOCHARGER EXHAUST DUCT		GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150300	EXHAUST STACK AND SNUBBER	6041F09007	POWER SYS.	ON		B.5.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803150501	EXPANSION JOINT, BETWEEN CHAMBERS (4)	8408985	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150502	SCREEN/TRAP ADAPTER	8482700	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150503	EXPANSION JOINT, TO TURBO CHARGER	8408987	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150511	EXHAUST, REAR CHAMBER	8424247	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150512	EXHAUST, INTERMEDIATE REAR CHAMBER	8424336	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150513	EXHAUST, CENTER CHAMBER	8424323	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150514	EXHAUST, INTERMEDIATE FRONT CHAMBER	8424336	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803150515	EXHAUST, FRONT CHAMBER	8422468	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803160100	STRAINER, LUBE OIL	8186637	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803160200	COOLER, LUBE OIL	SKETCH	GM-EMD	ON		B.1.62	DETAILED DYNAMIC ANALYSIS.
0803160300	FILTER, MAIN LUBE OIL	23800-86L	COMMERCIAL FILTERS	ON		B.1.92	DETAILED DYNAMIC ANALYSIS.
0803160500	PUMP, LUBE OIL (MOTOR) CIRCULATING (PUMP)	8[illegible]733-A	DELCO VIKING	ON		B.1.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803160700	FILTER, SOAK BACK	1750-105P	COMMERCIAL	ON		B.1.16	STATIC COEFFICIENT PER

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			FILTERS				IEEE-344 (1975).
0803160800	PUMP, SCAVENGING	8029579	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803160900	PUMP, MAIN & PISTON COOLING	8380573	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803151000	FILTER, TURBOCHARGER		GM-EMD	ON	31	D.1	REVIEW OF GM E77-1 & E77-1A.
0803161100	SEPARATOR, LUBE OIL	C-23656	FARR	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803161101	LUBE OIL SEPARATOR COVER	C-42525	FARR	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803161200	LUBE OIL EJECTOR ASSEMBLY		GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803161201	TUBING, EJECTOR		GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803161202	FLEX CONNECTION, CONVOLUTED		GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803161300	PISTON COOLING PUMP PIPE ELBOW	8360555	GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803161400	PIPING, STRAINER PIPE ELBOW	8360557		ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803161500	PIPING, STRAINER TO SCAVENGING PUMP		GM-EMD	ON		D.1	REVIEW OF GM E77-1 & E77-1A.
0803161601	VALVE, BALL	8354227	GM-EMD	ON		B.1.98	OTHER-SEE REFERENCED SECTION
0803161703	VALVE, CHECK ON SOAKBACK FILTER	8366533	GM-EMD	ON		B.1.90	OTHER-SEE REFERENCED SECTION
0803161704	VALVE, CHECK ON TURBO FILTER	8366577	GM-EMD	ON		B.1.87	OTHER-SEE REFERENCED SECTION
0803161705	VALVE, RELIEF, INLET TO PRIMARY FILTER	316C-1-30	KEPNER	ON		B.1.103	OTHER-SEE REFERENCED SECTION
0803161707	VALVE, RELIEF FILTER	8320705	EMD	ON		B.1.60	OTHER-SEE REFERENCED SECTION
0803161800	STRAINER, 'Y'		SARCO	ON	32	B.1.69	OTHER-SEE REFERENCED SECTION
0803161903	GAUGE & TUBING ASSEMBLY, MAIN LUBE PRESSURE		GM-EMD	ON		B.1.29	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162002	SWITCH & TUB'G ASSEMBLY, CRANKCASE PRESSURE	8362040	GM-EMD	ON		C.2	SEE ELECTRICAL SECTION
0803162005	SWITCH, CRANKING LOCKOUT	9012-ACW-21	SQUARE D	ON		C.2	SEE ELECTRICAL SECTION
0803162201	PIPING, Y STRAINER TO SOAK BACK FILTER		GM-EMD	ON	33,34,35	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).

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LOG NUMBER	DESCRIPTION	PART NUMBER/ DRAWING NO.	VENDOR	LOC	PHOTOGRAPHIC REFERENCE	PAGE REFERENCE	METHOD OF ANALYSIS
0803162202	PIPING, SOAKBACK FILTER TO TURBO FILTEC		GM-EMD	ON		B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162203	PIPING, PUMPS TO Y STRAINER		GM-EMD	ON	37,38	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162204	PIPING, ENGINE TO CIRC PUMP		GM-EMD	ON	39,40,41,42	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162205	PIPING, SCAVENGING PUMP TO OIL FILTER		GM-EMD	ON	43,44	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162206	PIPING, LUBE OIL FILTER TO OIL COOLER		GM-EMD	ON	45	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162207	PIPING, LUBE OIL COOLER TO STRAINER		GM-EMD	ON	46,47,48,49	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162208	PIPING, FILTER INLET FROM Y STRAINER		GM-EMD	ON	50	B.7.20	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162301	FLEX CONNECTION, NEAR CIRC PUMP	55381-1	AEROQUIP	ON	50	B.7.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803162302	FLEX CONNECTION, OIL COOLER TO STRAINER	55381-4	AEROQUIP	ON	49	B.7.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803180100	CABINET,ENGINE CONTROL	SKETCH		OFF		B.9.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0803180300	CABINET,GOVERNOR AND GENERATOR CONTROL	6041F11002	POWER SYS.	OFF		C.1	DETAILED DYNAMIC ANALYSIS.
0803190100	DIESEL FUEL DAY TANK			ON		B.2.56	ENGINEERING EVALUATION.
0803200000	PANEL, SEQUENCE CONTROL	6041F11001	POWER SYS.	ON		C.1	DETAILED DYNAMIC ANALYSIS.
0803200200	PANEL, DEADLINE CONTROL AND MASTER EXTRA	6041F11001	POWER SYS.	OFF		C.1	DETAILED DYNAMIC ANALYSIS.
0803210100	COMMON SWITCHGEAR ENCLOSURE AND PANELS					C.1	DETAILED DYNAMIC ANALYSIS.
0803220100	MOTORS, ELECTRIC START	1109758	DELCO	ON		B.4.3	STATIC COEFFICIENT PER IEEE-344 (1975).
0803220600	BRACKET, ELECTRIC STARTER MOTOR, LHS			ON		B.4.16	STATIC COEFFICIENT PER IEEE-344 (1975).
0803220700	BRACKET, ELECTRIC STARTER MOTOR, RHS			ON		B.4.16	STATIC COEFFICIENT PER IEEE-344 (1975).
0803220800	BATTERIES					B.4.2	ENGINEERING EVALUATION.
0806070000	GENERATOR	8323894	GM-EMD	ON		B.6.2	STATIC COEFFICIENT PER IEEE-344 (1975).
0806080000	EXCITER			ON		C.2	ENGINEERING EVALUATION.
0806090000	INERTIAL AIR CLEANER	29659	FARR	ON		B.6.16	ENGINEERING EVALUATION.
0806090100	BLOWER	8366082	GM-EMD	ON		B.6.16	ENGINEERING EVALUATION.

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0806090200	DUST BIN BLOWER DUCT		GM-EMD	ON		B.6.16	ENGINEERING EVALUATION.
0806090300	FLEXIBLE DUCT	8332267	GM-EMD	ON		B.6.16	ENGINEERING EVALUATION.
0806090400	AIR DISCHARGE DUCT	8426312	GM-EMD	ON		B.6.16	ENGINEERING EVALUATION.

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4.1

A.1 RESPONSE SPECTRA COMPARISON

APPENDIX A.1

RESPONSE SPECTRA COMPARISON

Purpose:

Objective of this calculation is to compare the base and elevated response spectra for OBE and SSE load cases obtained for the EDGS for Shoreham station with corresponding spectra used by S&L previously for similar equipment and components.

Conclusion:

Comparison results indicate that spectra used by S&L previously for similar equipment and components, conservatively envelope the spectra obtained for Shoreham EDGS in the frequency range of interest for all load cases and directions considered.

References:

1) Base Time History Record, (Mag. Tape Supplied by LILCO) S&L Tape Save Library No. 07542, Dated 5-7-84.
2) CQD File No. 011392, Rev. 00, LaSalle, Clinton OBE 2%, SSE 3% Horizontal & Vertical Base Response Spectra for EDGS, Date 5-31-83.
3) CQD File No. 009249, Rev. 00, Generic OBE 2%, SSE 3% Horizontal & Vertical Final Envelope, Elevated Response Spectra for EDGS, Date 8-10-83.
4) EMD File No. 008028, Rev. 00, Review of Seismic Qualification Report for Diesel Generators; Equipment No. 0,1,2DG01K dated 4/19/77.
5) Appendix "C.1(b)"Results of Time History Analysis of the Present Report.
(Applicable Computer run microfiche copies included)

Analysis - Base Response Spectra:

Base response spectra in horizontal and vertical directions for load cases of OBE and SSE are obtained through the execution of S&L computer program RSG (program number: RSG097050560) with Section 4.2 options. The acceleration time history record supplied by LILCO (Ref. 1) was modified as follows for various load cases and directions:

OBE - Horizontal: Multiply accelerations by 1.

OBE - Vertical: Multiply accelerations by 2/3.

SSE - Horizontal: Multiply accelerations by 2.

SSE - Vertical: Multiply accelerations by 4/3.

These time history records are individually input to RSG and response spectra curves are obtained. A total of 80 separate periods are used for the computations. These periods include the standard 76 distinct periods suggested by NRC. Plots of the individual spectra and the corresponding time history records are presented in the following pages. The corresponding base spectra curves used for similar equipment and components (Ref. 2 & 4) by S&L are also plotted on these graphs with dashed lines for comparison purposes. As seen from these plots, base spectra obtained from the present analysis for OBE-Horizontal, SSE-Horizontal and Vertical cases are enveloped by S&L spectra in the frequency range of 0.5 Hz to 500. Hz, and OBE-Vertical case is enveloped in the frequency range of 1. Hz-500. Hz with significant margin of conservatism. None of the equipment that was analyzed in this study was found to have any predominant frequencies in the range below 1Hz. For this reason, the frequency range below 1Hz will not affect this analysis. The following Table lists the Run ID and date of the applicable computer runs for the present response spectra computations (Base) and microfiche copies are included at the end of the report for reference purposes.

Computer Run List

Run ID	Date	Subject	Fiche Orig. No.
SIKS	5/9/84	OBE-Horizontal, Base Reponse Spect.	1
SIKA	5/10/84	OBE-Vertical, Base Response Spect.	1
SIKC	5/10/84	SSE-Horizontal, Base Response Spect.	1
SIKB	5/10/84	SSE-Vertical, Base Response Spect.	1

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☒ Safety-Related	☐ Non-Safety-Related

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Project		Reviewed by Ismail Kisser	Date 5-18-84
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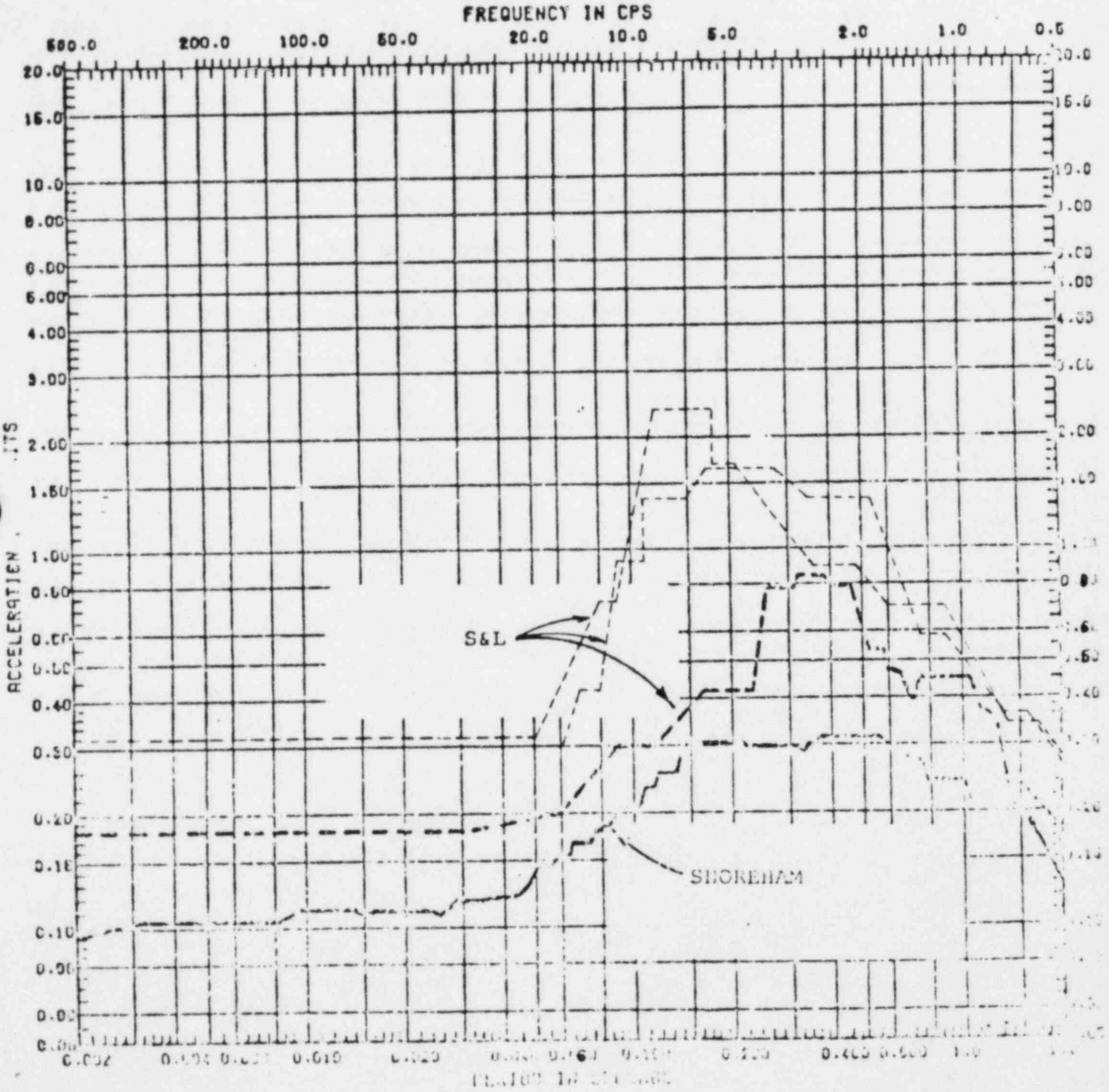
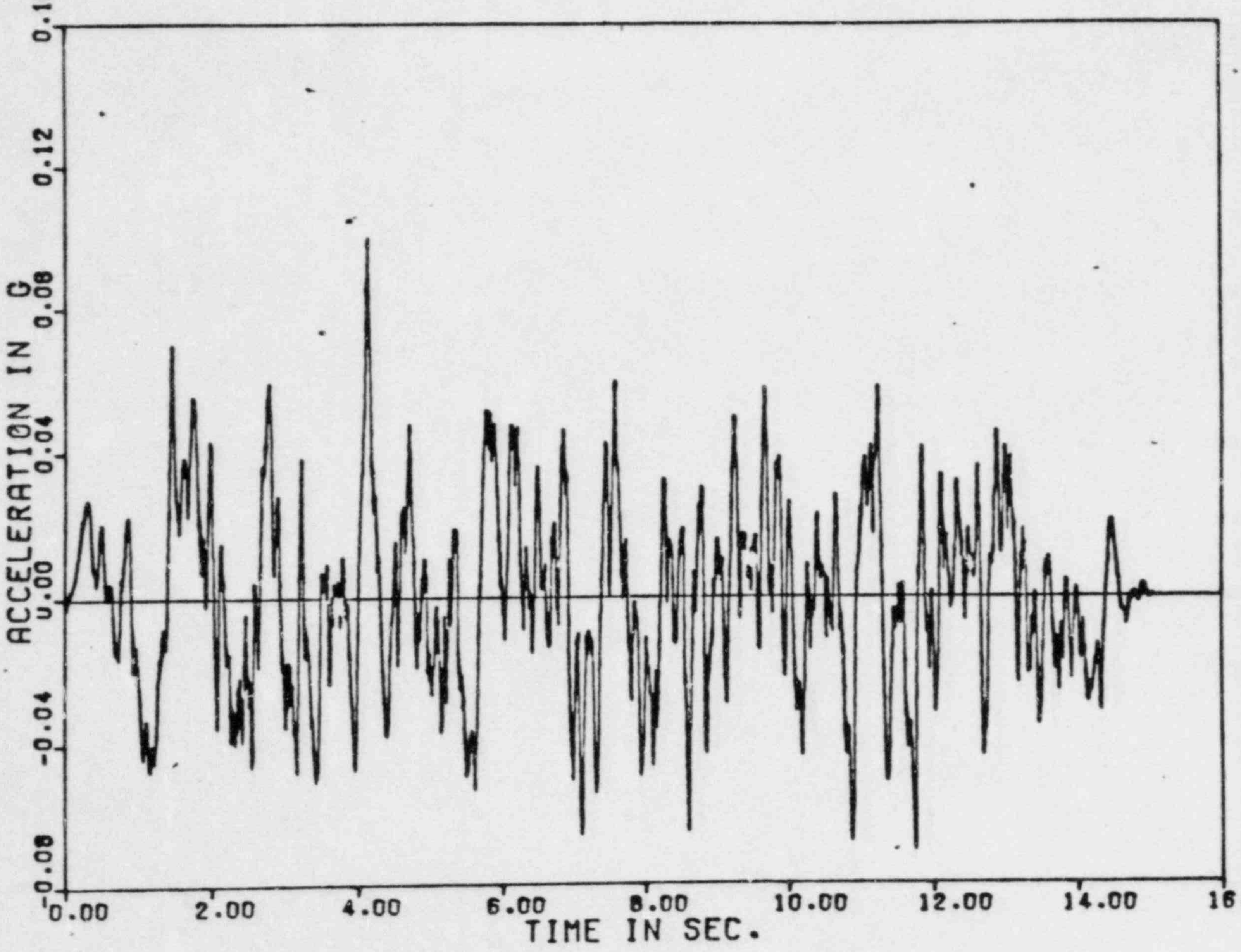


☑ HORIZONTAL SPECTRA - OBE
☐ VERTICAL SPECTRA
COMBINED BY ☐ ABS ☐ SRSS N/A

LOCATION: FDGS
ELEVATION: BASE
DAMPING: 2 %

S
09 MAY 8



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BASE OBE 2% (BASE) GROUND EDGS HOR N/A

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Calcs. For	
☒ Safety-Related	☐ Non-Safety-Related

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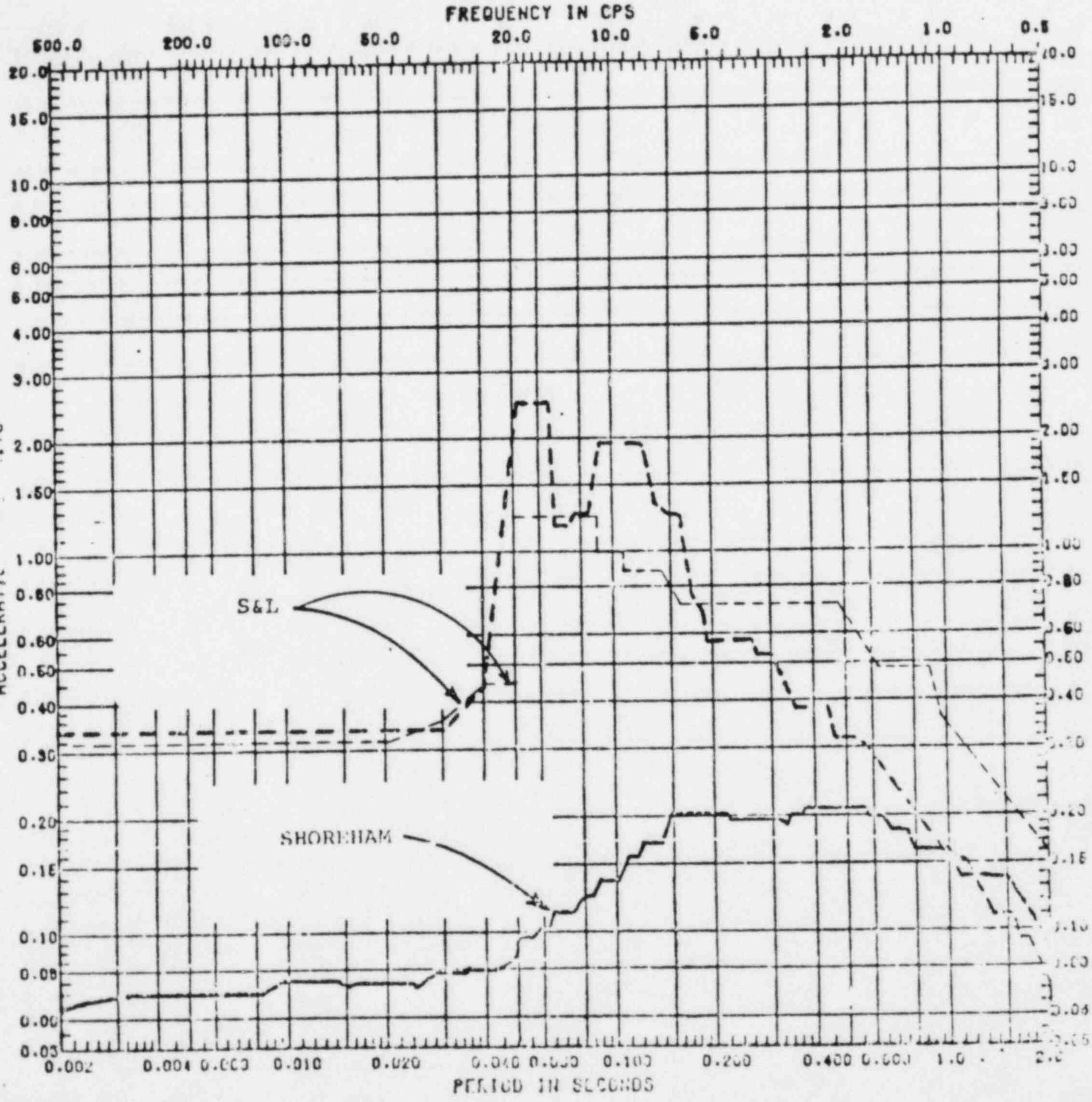
Project	
Proj. No. 6995-00	Equip No.

Prepared by M. Danez	Date 5-17-84
Reviewed by Ismail Kisisel	Date 5-18-84
Approved by Y. A. Patel	Date 5-19-84



☐ HORIZONTAL SPECTRA

☑ VERTICAL SPECTRA - OBE

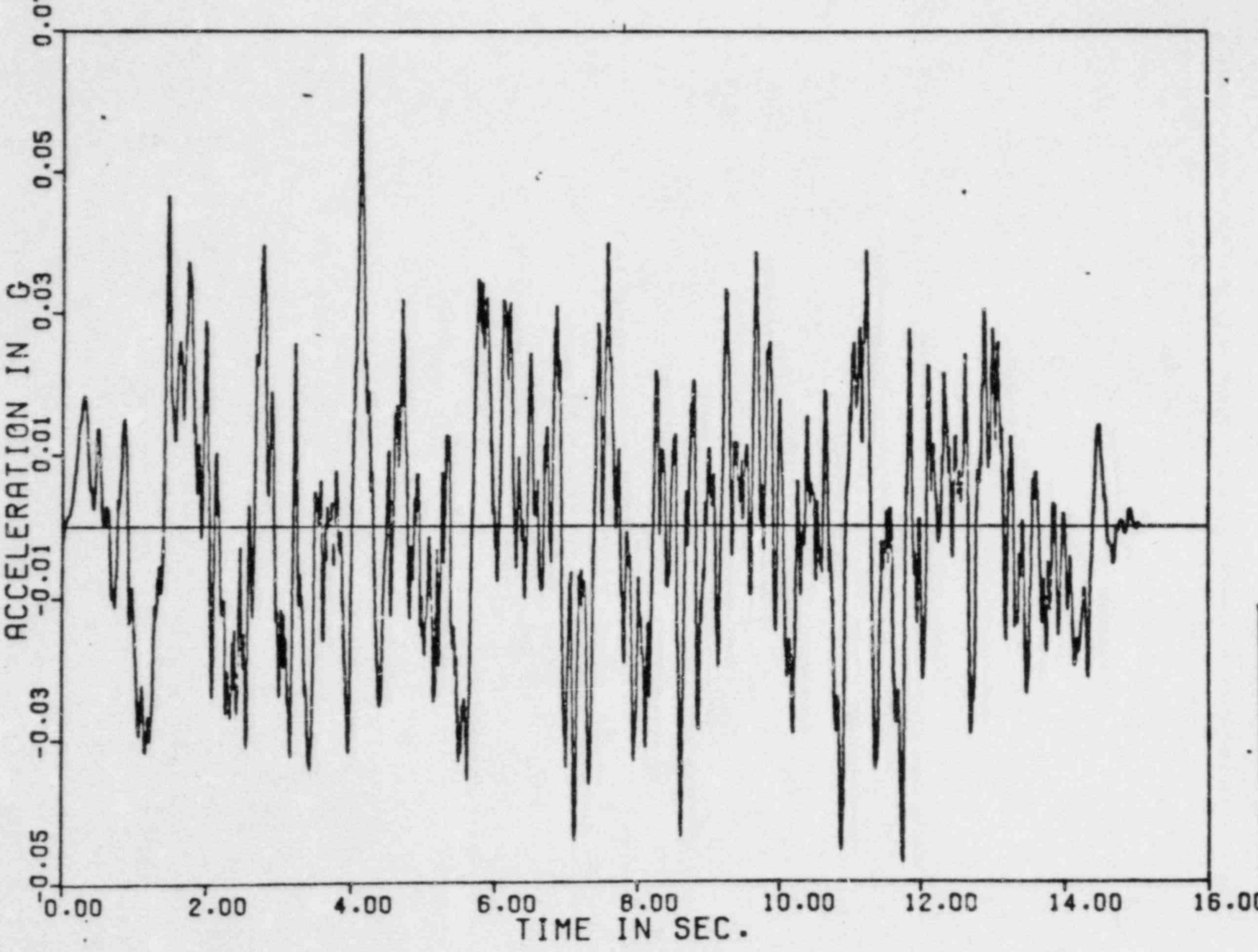
COMBINED BY ☐ ABS ☐ SRSS N/A

LOCATION: EDGS

ELEVATION: BASE

DAMPING: 2 %

KA
10 MAY 84



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BASE OBE 2% (BASE) GROUND EDGS VER N/A

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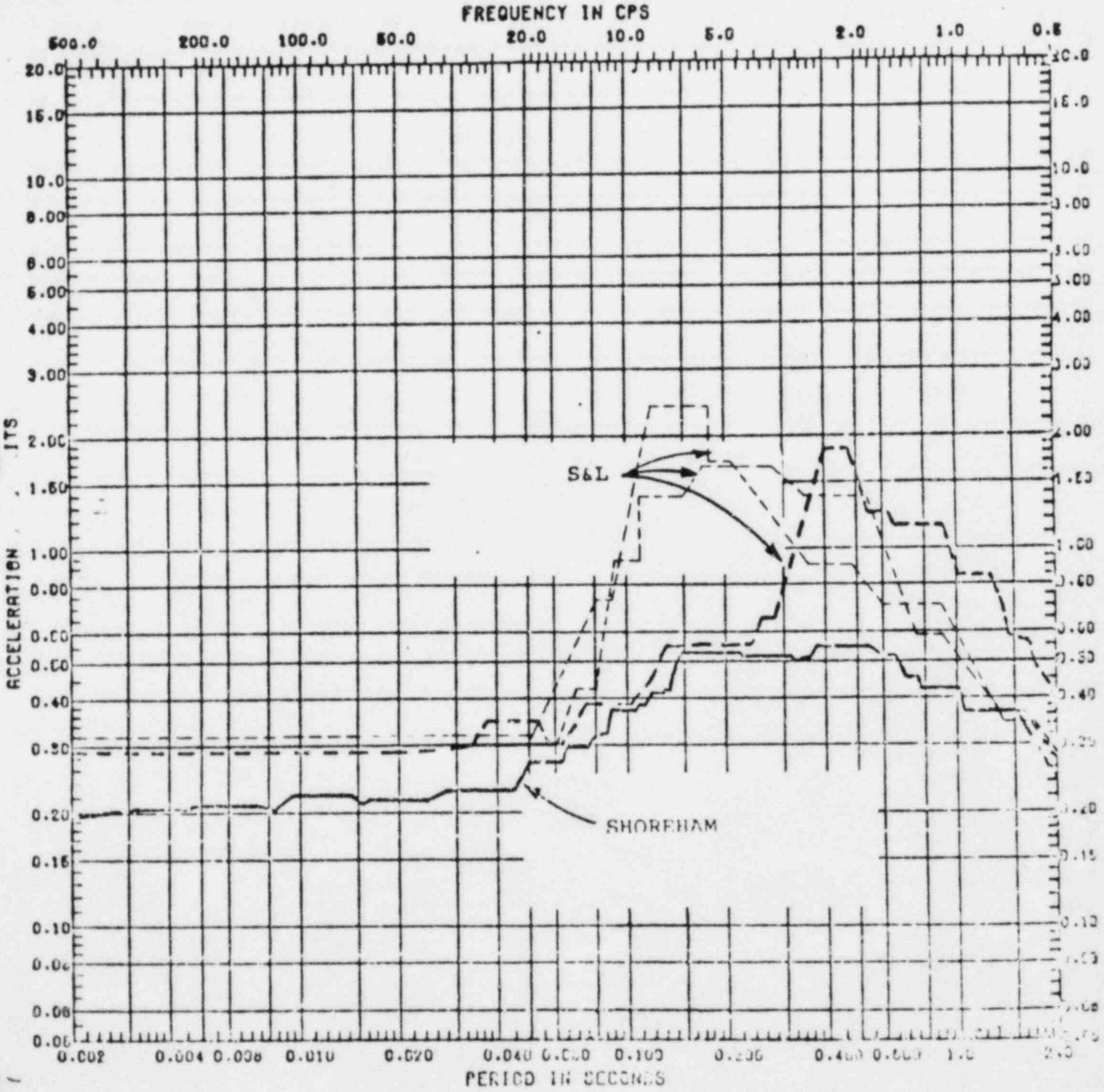
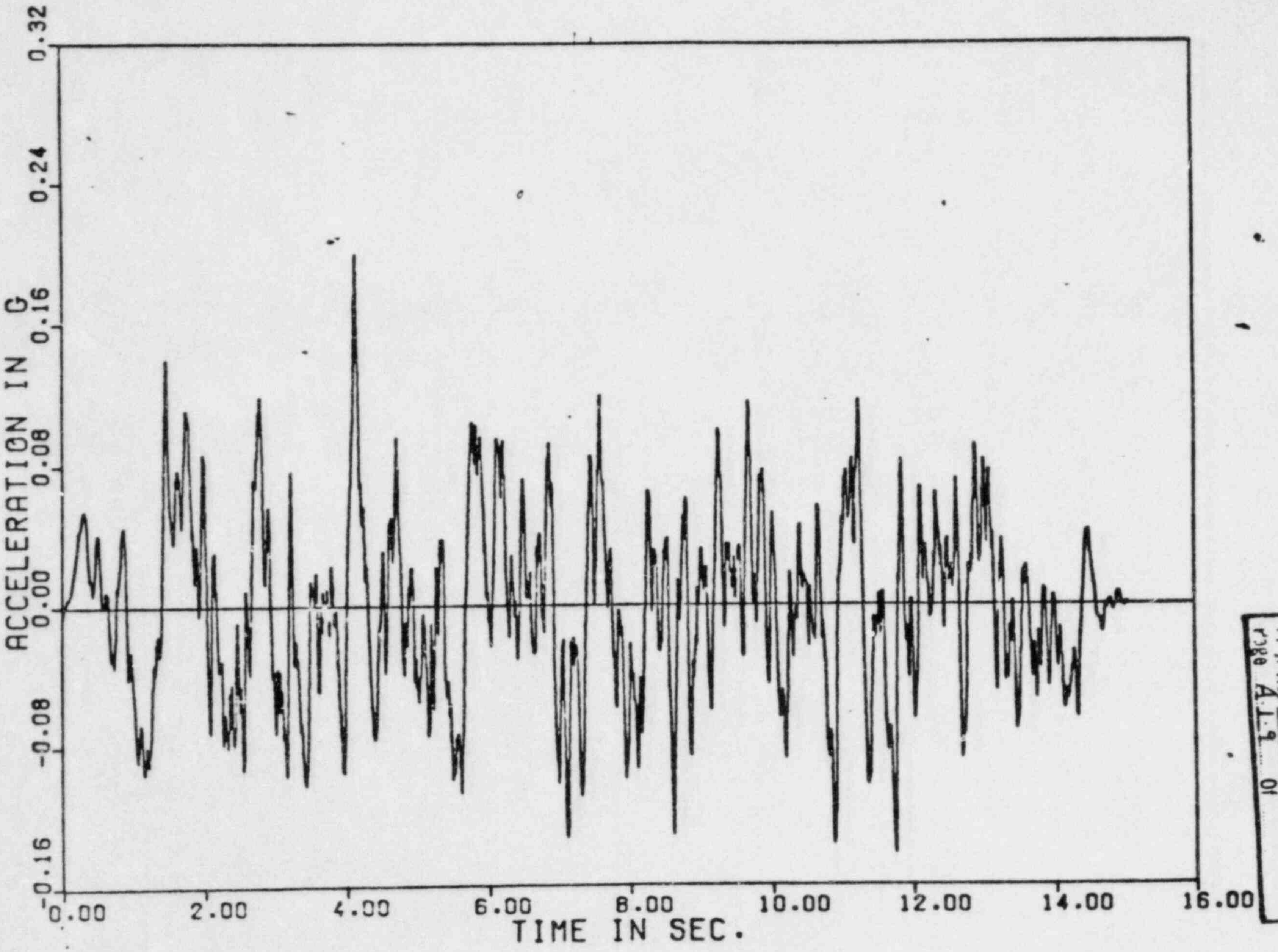
Prepared by M. DeMarz Date 5-17-84
Reviewed by Ismail Kisisel Date 5-18-84
Approved by Y. A. Patel Date 5-19-84



☑ HORIZONTAL SPECTRA - SSE
☐ VERTICAL SPECTRA
COMBINED BY ☐ ABS ☐ SRSS N/A

LOCATION: EDGS
ELEVATION: BASE
DAMPING: 3%

KC
10 MAY 8



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BASE SSE 3% (BASE) GROUND EDGS HOR N/A

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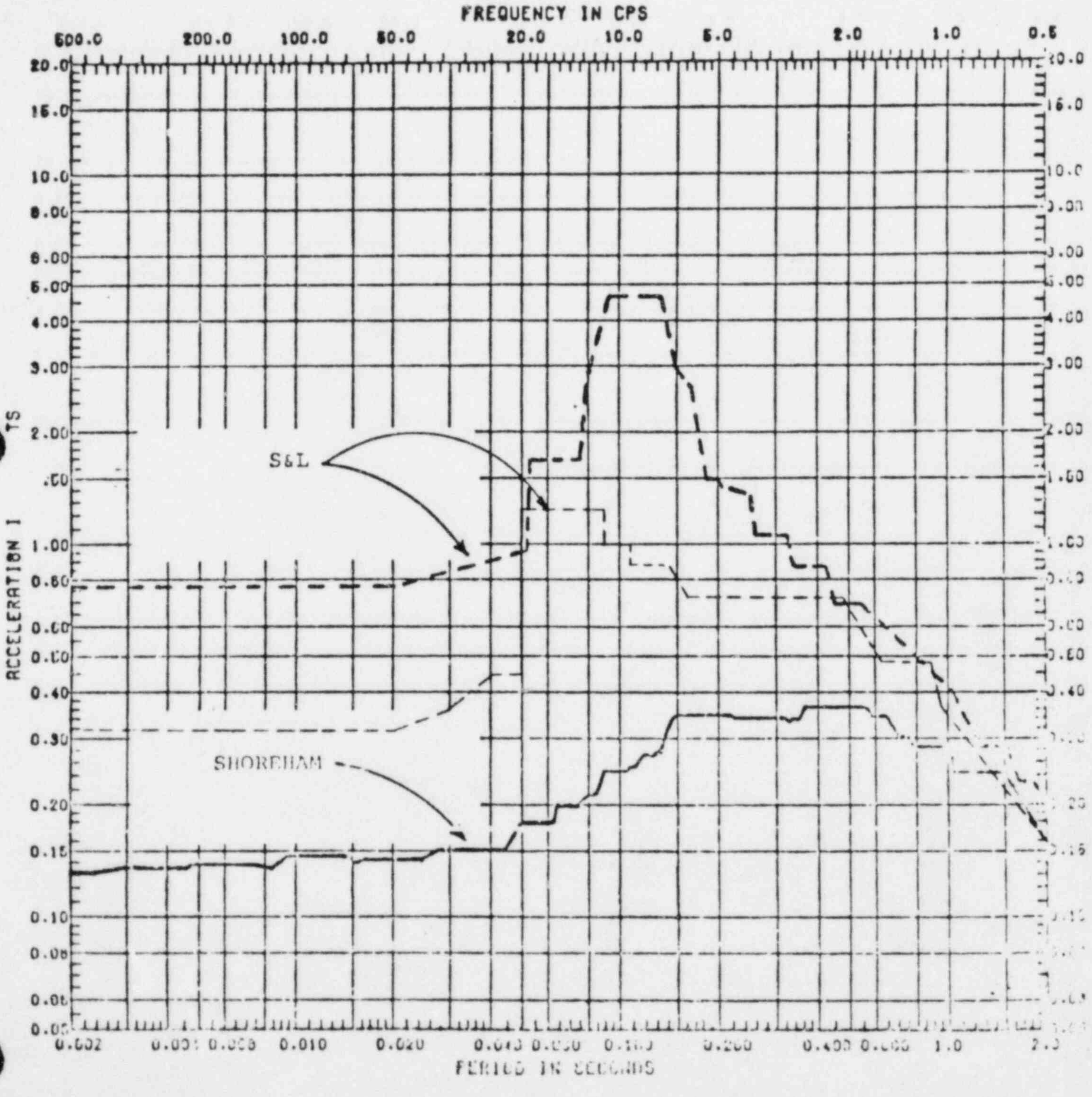
Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Calc No. CQD-014046
Rev. 00 Date 6-1-84
Page A.1.10 of

Client
Project
Proj. No. 6495-00 Equip No.

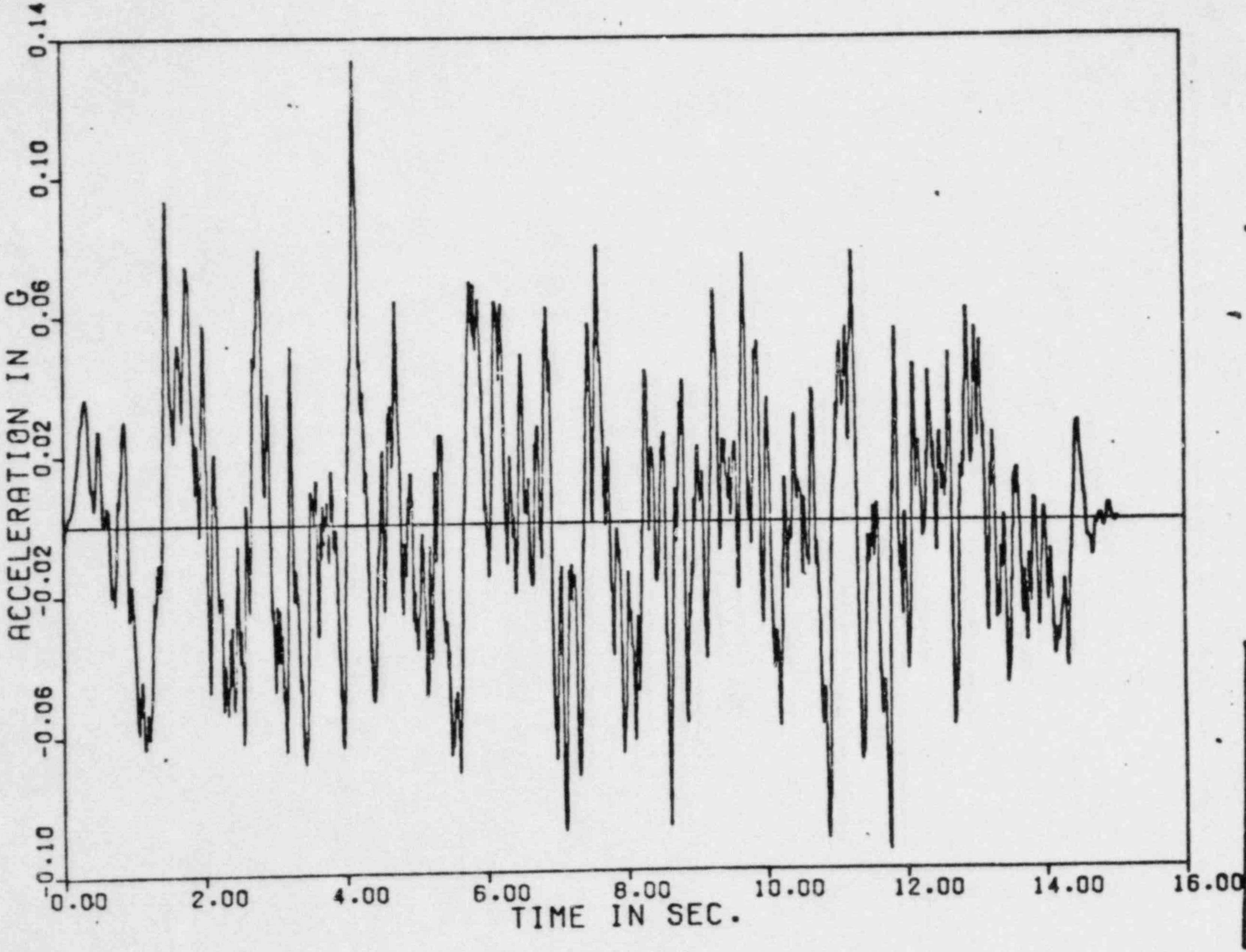
Prepared by M. Darrag Date 5-17-84
Reviewed by Ismail Kisisel Date 5-18-84
Approved by Y A Patel Date 5-19-84



☐ HORIZONTAL SPECTRA
☑ VERTICAL SPECTRA -- SSE
COMBINED BY ☐ ABS ☐ SRSS N/A

LOCATION: EDGS
ELEVATION: BASE
DAMPING: 3 %

10 MAY '84



Calc. No: CQD- 014046
Rev: 00 Date: 6-1-84
Proj. No: 6995-00
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BASE SSE 3% (BASE) GROUND EDGS VER N/A

Elevated Spectra Comparison:

Envelope of the elevated spectra for horizontal and vertical direction of OBE and SSE load cases obtained through the time history analyses (see Appendix "C.1(b)" of this report for details -Ref. 4) are given on the following pages. The Shoreham curves on these plots are the envelope of the spectra obtained at nodes 15,16 and 60 of the cubicle 1A door panel model. The dashed line curves on these plots represent the generic final envelope of elevated response spectra (Ref. 3) used by S&L previously for similar equipment and components. As the comparison indicate, S&L curves envelope the Shoreham curves in the frequency range of 0.5 Hz to 500 Hz with a significant margin of conservatism.

Summary:

S&L response spectra curves for OBE and SSE load cases in horizontal and vertical directions for the BASE and for the ELEVATED locations of EDGS envelope presently obtained corresponding Shoreham repsonse spectra curves in the frequency range of interest with significant margins.

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Project
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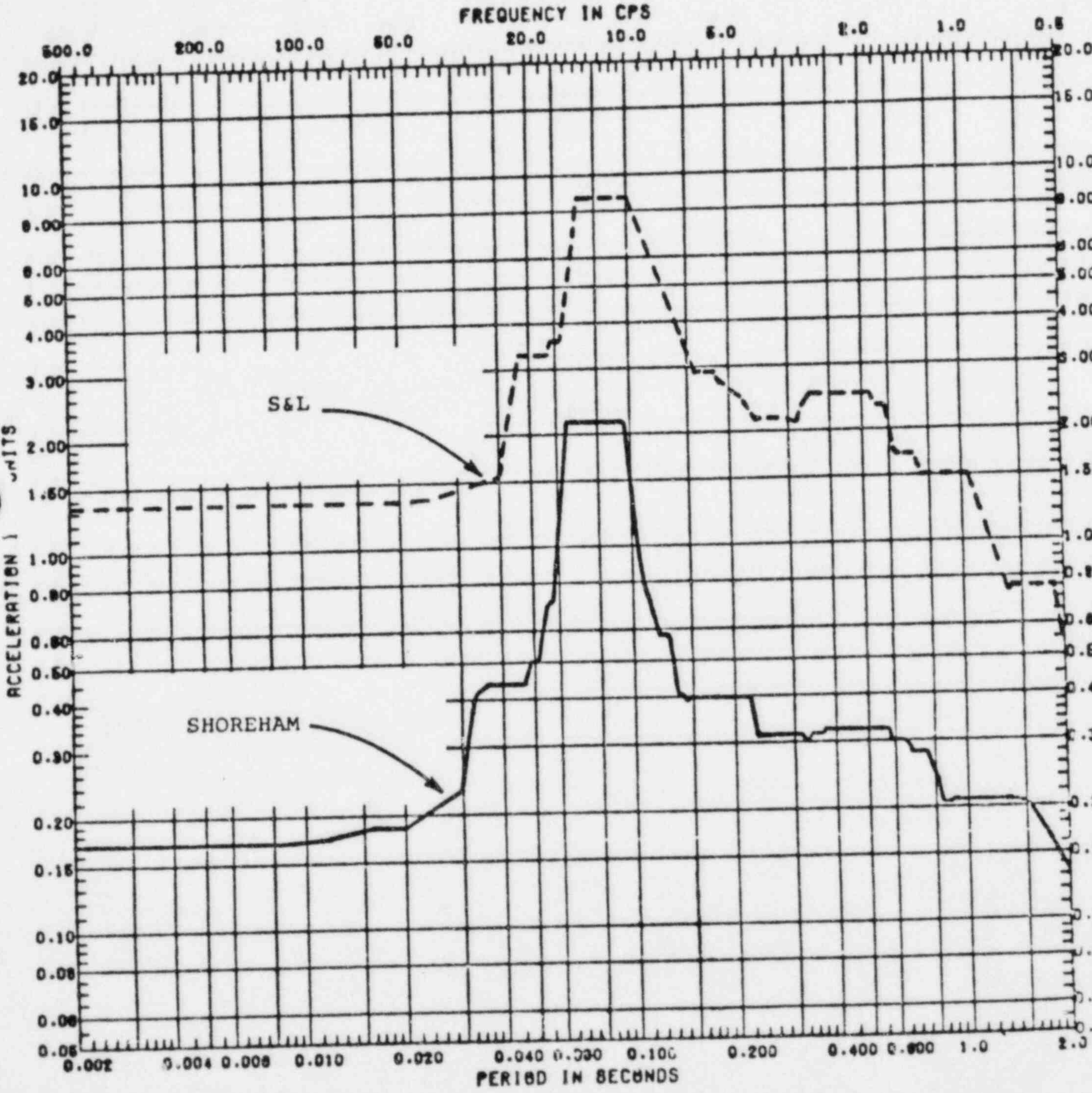
Prepared by M. Damaz Date 5-17-84
Reviewed by Ismail Kaiser Date 5-18-84
Approved by Y A Patel Date 5-19-84



☑ HORIZONTAL SPECTRA - OBE
☐ VERTICAL SPECTRA
COMBINED BY ☐ ABS ☐ SRSS N/A

LOCATION: EDGS
ELEVATION: ENVELOPE ALL
DAMPING: 2%

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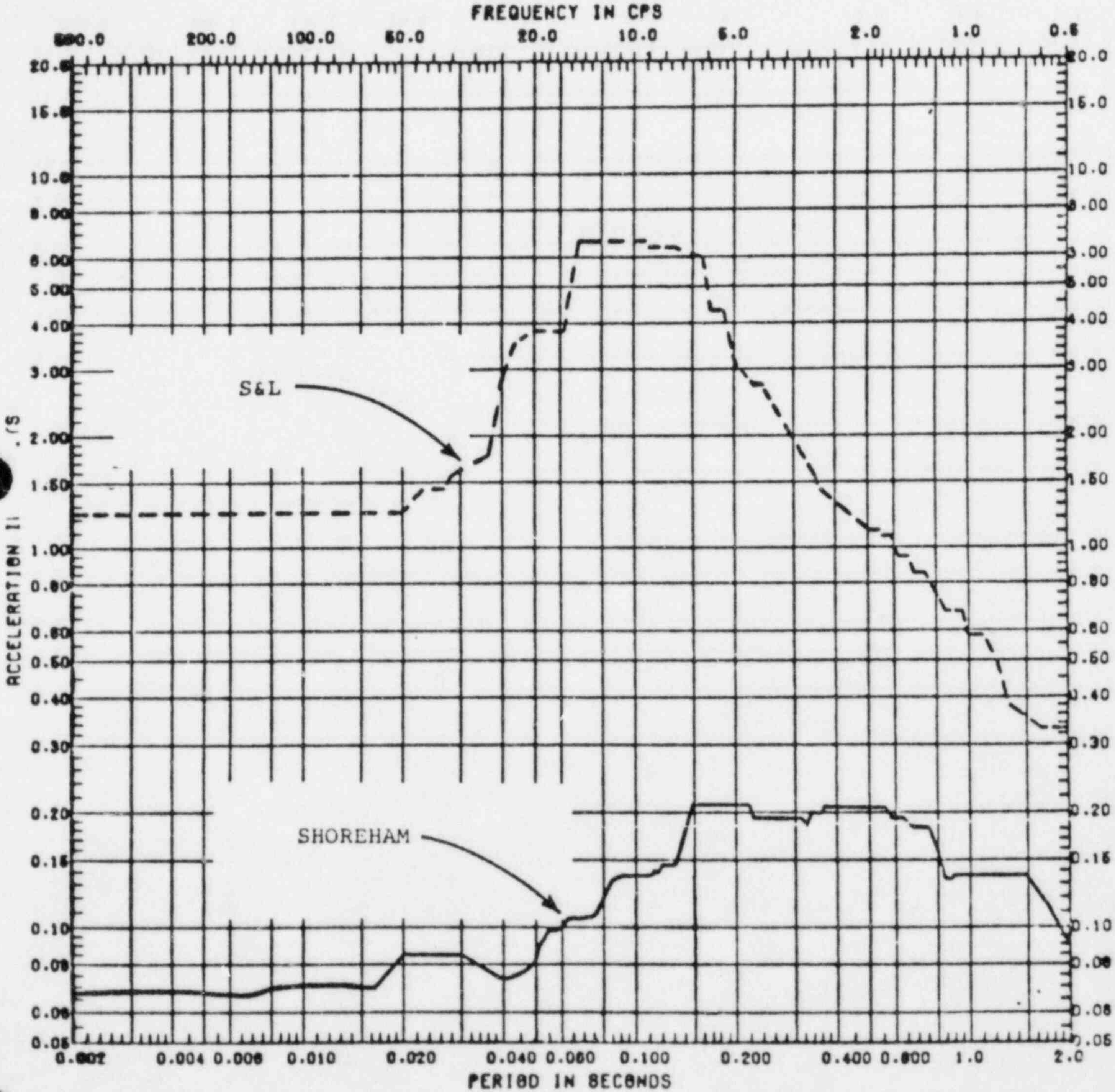
Project
Proj. No. 6995-00 Equip. No.

Prepared by M. Danay Date 5-18-84
Reviewed by Ismail Kisser Date 5-18-84
Approved by Y. A. Patel Date 5-19-84



☐ HORIZONTAL SPECTRA
☑ VERTICAL SPECTRA – OBE
COMBINED BY ☐ ABS ☐ SRSS N/A

LOCATION: EDGS
ELEVATION: ENVELOPE ALL
DAMPING: 2%

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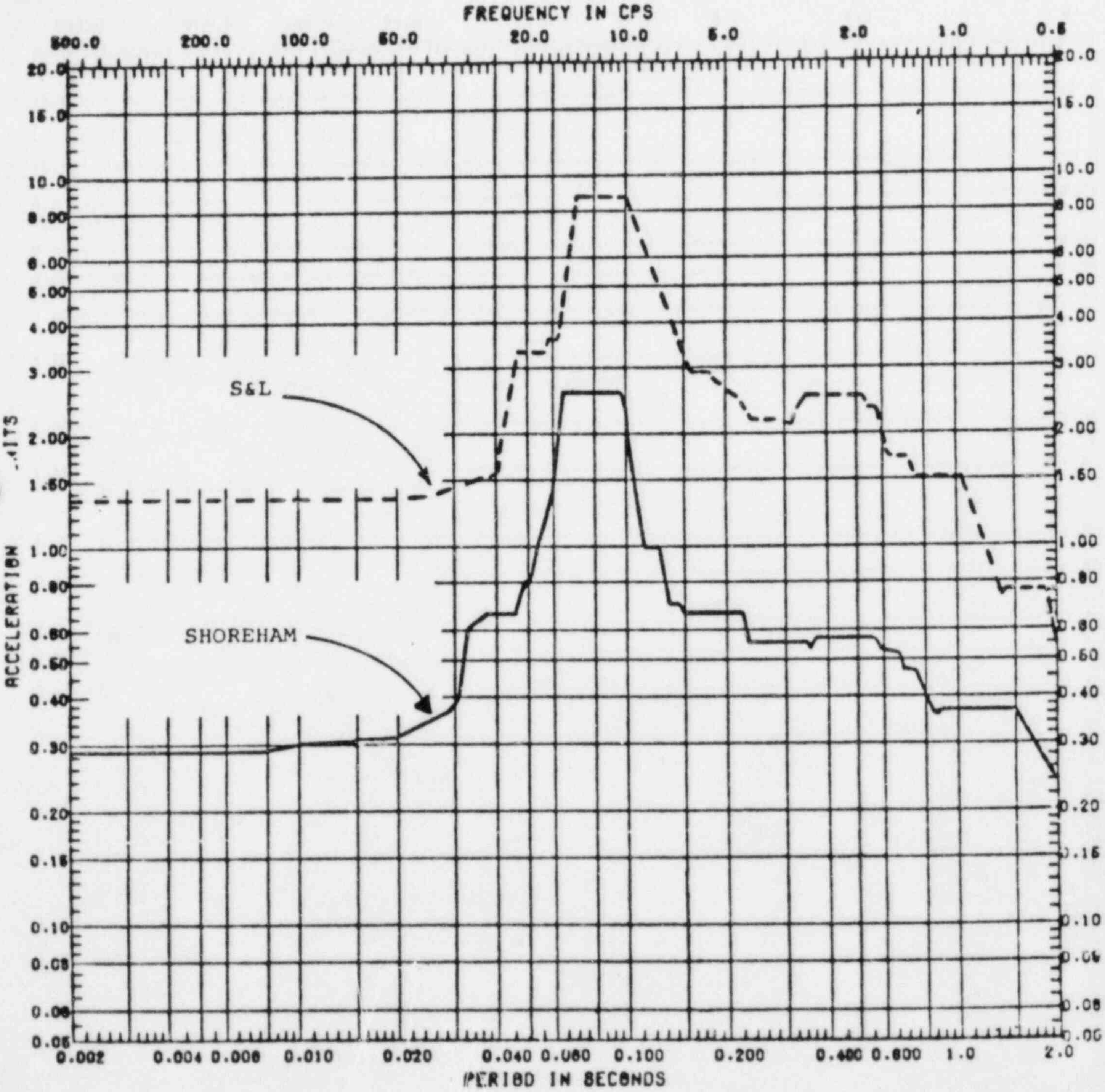

☑ HORIZONTAL SPECTRA - SSE

☐ VERTICAL SPECTRA

COMBINED BY ☐ ABS ☐ SRSS N/A

LOCATION: EDGS

ELEVATION: ENVELOPE ALL

DAMPING: 3%

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Prepared by M. Danag	Date 5-18-84
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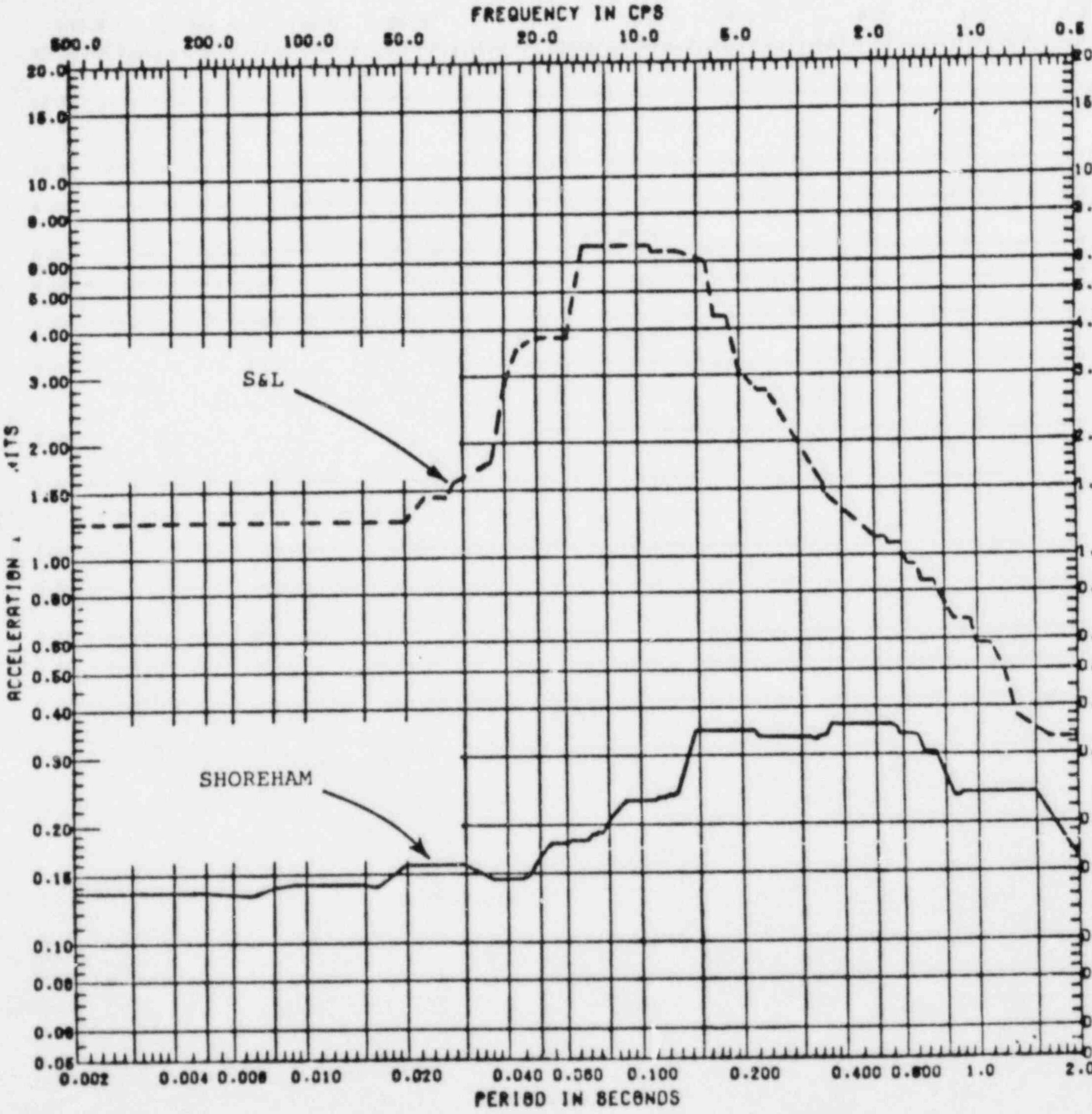

☐ HORIZONTAL SPECTRA
☑ VERTICAL SPECTRA - SSE
COMBINED BY ☐ ABS ☐ SRSS N/A

LOCATION: EDGS
ELEVATION: ENVELOPE ALL
DAMPING: 3%

FINAL

Response Spectrum Design Basis Earthquake 1% Damping Used as Input for LaSalle EDGS Finite Element Analysis Found in EMD-008028

HORIZONTAL (East-West)		HORIZONTAL (North-South)		VERTICAL	
Equipment Frequency (cps)	Acceleration Response (g's)	Equipment Frequency (cps)	Acceleration Response (g's)	Equipment Frequency (cps)	Acceleration Response (g's)
0.5	0.27	0.5	0.28	0.5	0.16
0.61	0.36	0.6	0.34	1.04	0.36
0.76	0.36	0.74	0.34	1.11	0.49
1.11	0.59	1.11	0.7	1.61	0.49
1.32	0.59	1.67	0.7	2.08	0.72
1.89	1.35	2.08	0.9	3.33	0.72
2.82	1.35	2.78	0.9	6.25	0.72
3.55	1.67	4.76	1.7	7.14	0.89
5.88	1.67	5.41	1.7	9.09	0.89
6.67	1.35	5.62	2.4	9.26	1.0
8.70	1.35	7.25	2.4	11.11	1.0
9.01	0.92	10.53	0.73	11.76	1.25
11.63	0.92	12.82	0.73	19.23	1.25
12.20	0.42	20.0	0.32	20.83	0.45
15.15	0.42	25.0	0.32	25.0	0.45
18.18	0.3	33.33	0.32	33.33	0.38
50.0	0.3	50.0	0.32	50.0	0.32

B.1

[redacted]

B.1 LUBE OIL SYSTEM

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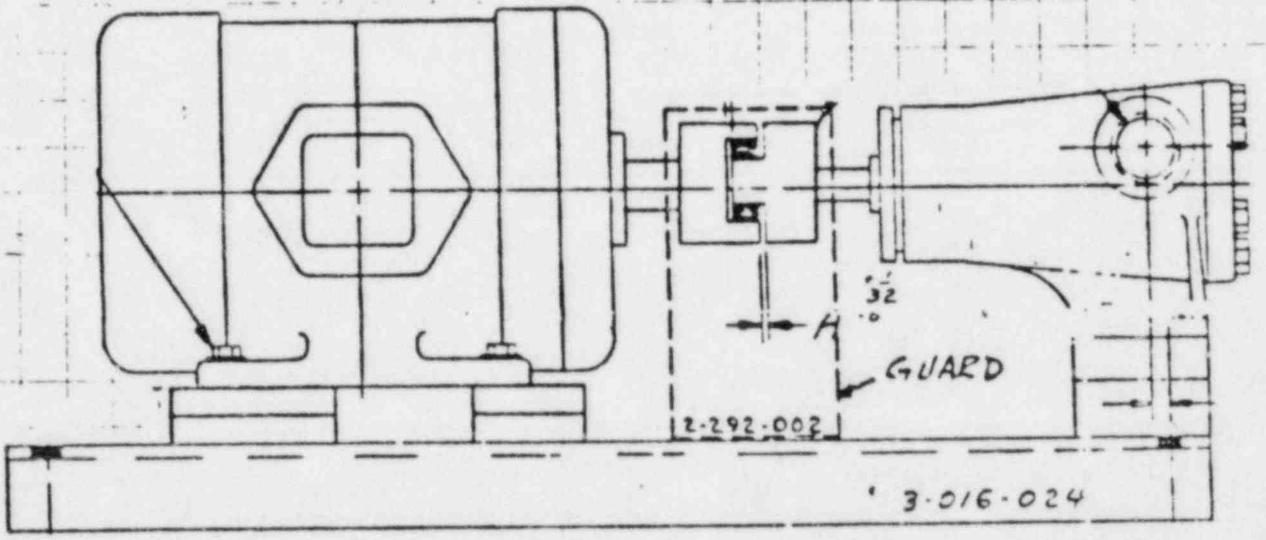
Client		Prepared by David Wright	Date 5/28/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Lube Oil Circulating Pump

Log No. 803160500



The Lube oil circulating pump consists of an electric motor and a small pump unit. The Assy. is mounted to the side of the accessory rack by welds along the platform. The unit will be shown to maintain both pressure boundry and structural integrit... Operability of the pump will be shown using simple deflection analysis. The allowable deflection of the motor shaft will be the air-gap between the rotor and housing.

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Client		Prepared by	Date
Project		Reviewed by	Date
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Electric Motor Analysis

The following information was obtained from the generator unit (Engine #2):

Manufacturer — Delco Products

Model No. — B6733-A

Frame — 184

Serial No. — D67

(see telephone memo dated 5/17/84 between D. Wright and C. Gangone)

Information provided by Delco Products (see phone memo dated 5/17/84 between D. Wright and J. Miller)

Model No. — B6733-A

Frame Size — 184

Horsepower — 1 H.p.

Shaft O.D. — 1.125 in

Estimated wt. — 100 Lbs.

Note — for mounting details use the Nema standard frame size.

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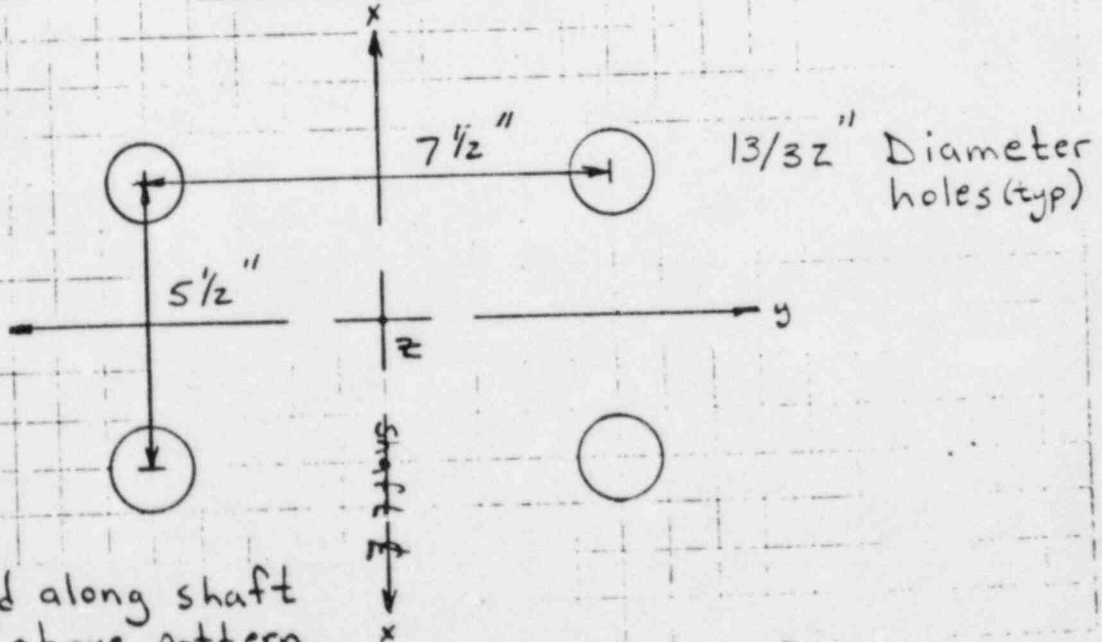
Method of Analysis

The motor analysis will be done in two areas. The mounting bolts of the motor to the platform will be performed to ensure that the motor will remain in place during and after a seismic event. The operability of the motor will be demonstrated using a simple deflection analysis. The allowable deflection will be taken as the air gap between the rotor and stator.

Mounting Bolt Analysis

*Bolt Pattern



C.G. located along shaft ¢ at 4 1/2" above pattern.

*Note - Mounting details are taken from a Delco products T-Line catalog # 1-2000 dated May 1, 1978.

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Loading, Seismic + Weight

note- starting torque due to 1 H.p. is negligable and will not be considered.

$F_x = (wt)(A_x) = 100 \text{Lbs} (2.5 g's)^* = 250 \text{Lbs}$

$F_y = (wt)(A_y) + (wt) = 100 \text{Lbs} (2.5 g's) + 100 \text{lbs} = 350 \text{Lbs}$

$F_z = (wt)(A_z) = 100 \text{Lbs} (2.5 g's) = 250 \text{Lbs}$

$M_x = F_y (7.5/2) = 350 \text{Lbs} (7.5/2) = 1313 \text{ in-lbs}$

$M_y = F_x (5.5/2) = 250 \text{Lbs} (5.5/2) = 688 \text{ in-lbs}$

$M_z = 0.0 \text{ in-Lbs}$

*assume an acceleration level of 2.5g's to account for any amplification in the mounting.

Geometric properties of the bolts

*Hole size - 13/32"

Bolt size - 3/8", SAE Gr. 5 bolt (A-449) (Assumed)

$A_s = 0.0775 \text{ in}^2$ (from "Machinery's Handbook", 12th edition, 3/8" Coarse-Thread series Page 1282)

*taken from Delco products T-Line catalog # 1-2000, 5/1/78

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Client		Prepared by	Date
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$$I_{xx} = \Sigma(\overset{\approx 0}{\cancel{I_x}} + Ad^2) = (0.0775\,in^2)(7.5/2)^2(4) = 4.4\,in^4$$

$$I_{yy} = \Sigma(\overset{0}{\cancel{I_y}} + Ad^2) = (0.0775\,in^2)(5.5/2)^2(4) = 2.3\,in^4$$

Check of Stress levels

$$\sigma_t = F_z/4A_t + M_x C_y/I_{xx} + M_y C_x/I_{yy}$$

$$\sigma_t = \frac{250\,lbs}{4(0.0775)} + \frac{1313\,in\text{-}lbs\,(7.5/2)}{4.4\,in^4} + \frac{688\,in\text{-}lbs\,(5.5/2)}{2.3\,in^4}$$

$$\sigma_t = 2748\,psi$$

Note - Due to the conservatism in the loading, all stress based on tensile Area

$$\tau_v = \left(\frac{F_x^2 + F_y^2)^{1/2}}{4A_t}\right) = \frac{(250^2 + 350^2)^{1/2}}{4(0.0775)} = 1387\,psi$$

Comparison to Allowable

(From ASME B+PV Code, 1977 edition; summer 79 addendum, section III, article 2460 of Appendix XVII.)

for combined tensile and shear, $\frac{\sigma_t^2}{S_t^2} + \frac{\tau_v^2}{S_v^2} \leq 1.00$

for service levels A+B, $S_t = 0.35u$, $S_v = 0.124 S_u$, $^*S_u = 120\,ksi$

$$\frac{(2748)^2}{(36{,}000)^2} + \frac{(1387)^2}{(14{,}880)^2} = 0.015 \leq 1.00$$

{ the bolts are adequate for levels A, B + C loadings.

* ASME B+PV Appendix I, table I-7.3, 1" and under page 98.

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Calcs. For

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Client		Prepared by	Date
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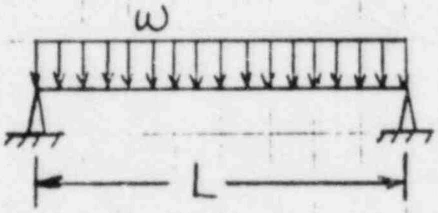
Operability Check

The operability of the motor will be shown through a simple deflection analysis. The total weight of the motor will be assumed to act on the rotor. The bearings will be modeled as simple supports. The bearing to bearing length will be estimated from Delco products catalog 1-2000 dated 4/1/79.

Analytical Model



$L = 14\ {}^{15}/_{16}$ (dimension "C" from above, consult NEMA Standard on electric motor dimensions, standard frame 184)

$\omega = \text{Load} / L$

$\text{Load} = wt\,(A_x^2 + A_y^2 + A_z^2)^{1/2} = 100\,\text{lbs}\,(2.5^2 + 2.5^2 + 2.5^2)^{1/2}$

$\text{Load} = 433\,\text{Lbs}$

$\Delta_{max} = \frac{5\omega L^4}{384EI}$ (from "Design of Welded Structures" by Blodgett, page 8.1-7, example 3B)

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Approved by	Date

$$I = \frac{\pi d^4}{64} = \frac{\pi (1.1250\,in)^4}{64} = 0.079\,in^4$$

$E = 30 \times 10^6 psi$ (assume carbon steel)

$$\omega = 433\,Lbs / (14\tfrac{15}{16}\,in) = 28.99\,Lbs/in$$

$$\Delta_{max} = \frac{5(28.99\,Lbs/in)(14\tfrac{15}{16})^4}{384(30 \times 10^6)(0.079)} = 0.0079\,in$$

$\Delta_{max} \ll 0.30$in air gap. There will be no contact between the rotor and the stator, Nor will there be any plastic deformation occuring. The shaft will not bind.

Form GQ-3.08.1 Rev. 7

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Client		Prepared by	Date
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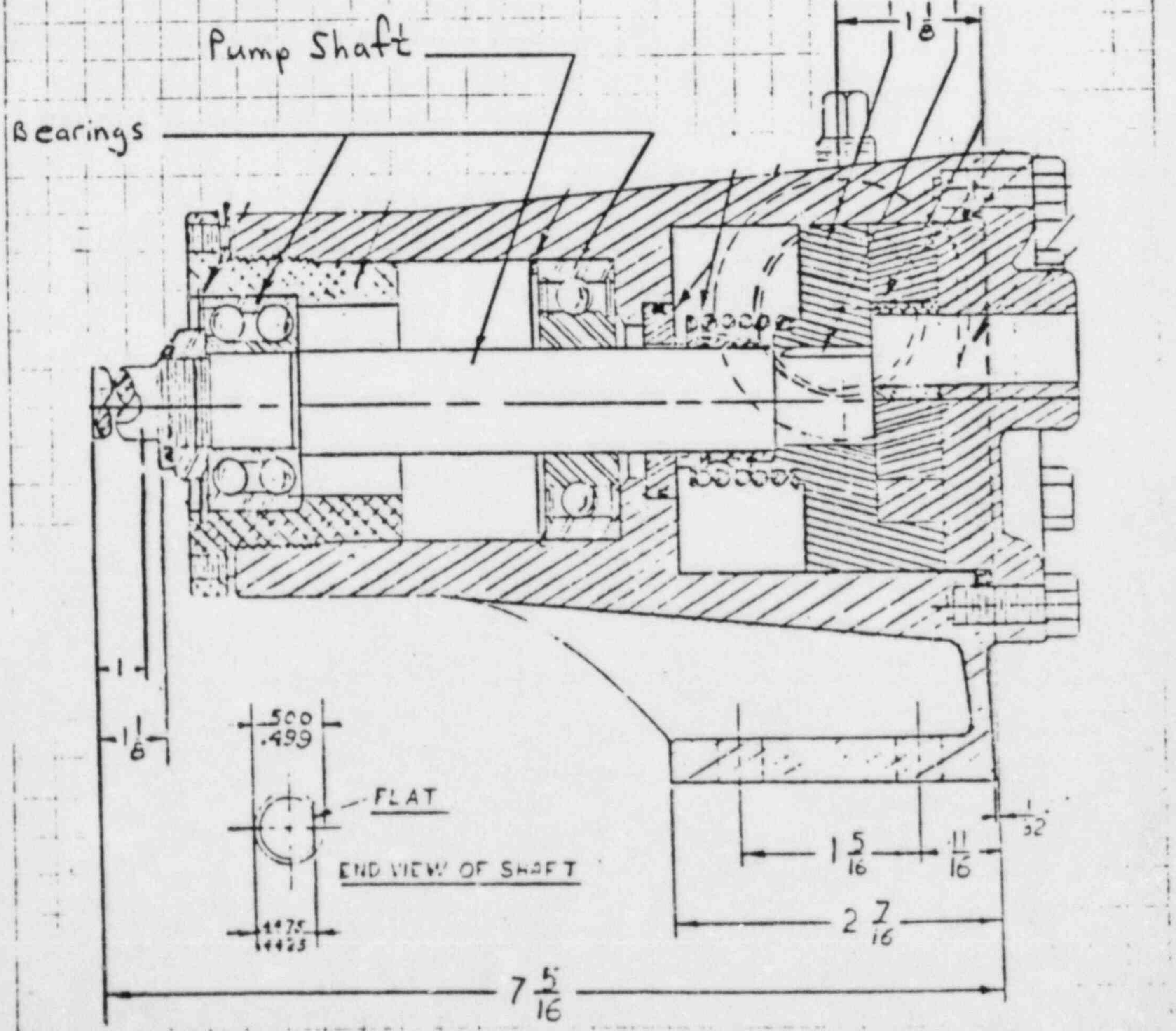
Mechanical Pump Assy.



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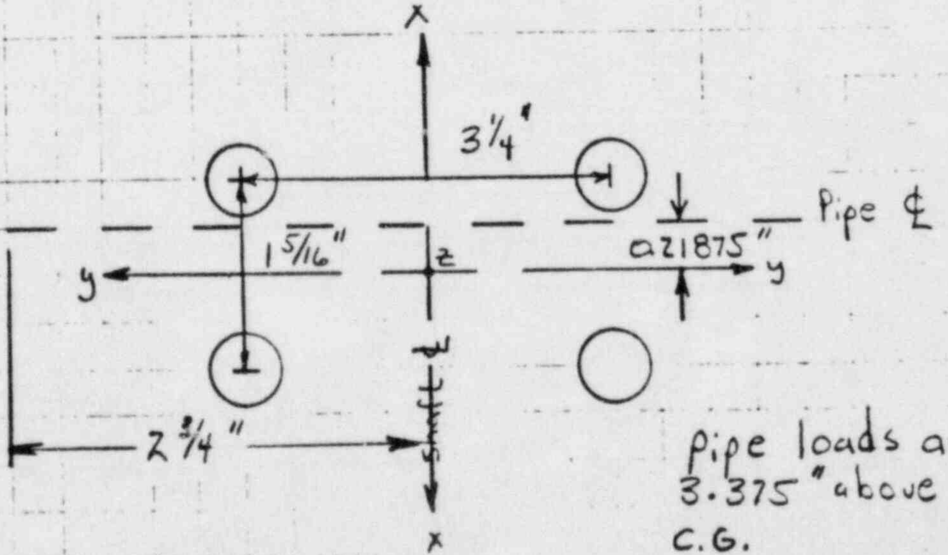
Pump Analysis

The pump has a total length of 7 5/16 in with a shaft O.D. of 1/2 in. It is obvious by the motor analysis that deflection of the shaft will not be a factor. There will be no weight on the shaft (due to vibration) other than its own weight. The structural integrity of the pump will be checked through an analysis of the mounting bolts.

Mounting Bolt Check

*Bolt Pattern



pipe loads are
3.375" above bolt
C.G.

*All Dimensions are estimated from EMD drawing EMD-8274509.
This is a representative drawing of a similar assembly.

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Loading - The pump assembly has no extended masses, is rigid and has a relatively small mass. For these reasons the only loading to be considered will be the nozzle loads. The nozzle loads will be conservatively estimated by loading found in S+L's CQD-005513, rev.00 dated 1/21/83 in titled "Nozzle Loads For Equipment". These are nozzle loads based on the section properties and allowable stress values of the attached piping. It is obvious through a simple inspection of the piping attached to the pump that the actual loads will be much lower than those used.

From CQD-005513, Equipment wt ≤ 1Kip, Class 2+3 nozzle size - 1" NPT, service level C

$$F_R = \sqrt{F_x^2+F_y^2+F_z^2} = 0.19 \text{ Kips} = 190 \text{ Lbs}$$

$$M_R = \sqrt{M_x^2+M_y^2+M_z^2} = 0.10 \text{ Kip-ft} = 1200 \text{ in-Lbs}$$

$$F_{x_p} = F_{y_p} = F_{z_p} = 190/\sqrt{3} = 110 \text{ Lbs}$$

$$M_{x_p} = M_{y_p} = M_{z_p} = 1200/\sqrt{3} = 693 \text{ in-Lbs}$$

Note - subscript "p" refers to the pipe load.
subscript "r" refers to resultant load.

Loading at Bolt Centroid

$F_x = 2F_{x_p} = 2(110\,Lbs) = 220\,Lbs$

$F_y = 2F_{y_p} = 220\,Lbs$

$F_z = 2F_{z_p} = 220\,Lbs$

$M_x = 2M_{x_p} + F_y(C_z) = 2(693) + 220(3.375'') = 2129$ in-lbs

$M_y = 2M_{y_p} + F_z(C_x) + F_x(C_z) = 2(693) + 220(.21875) + 220(3.375)$

$M_y = 2177$ in-lbs

$M_z = 2M_{z_p} + F_x C_y = 2(693) + 220(2.75) = 1991$ in-lbs

Bolt Geometric Properties (see note)

Bolt Hole size — 11/32 in

Assumed Bolt Size — 5/16 in, SAE Gr. 5 (SA-449)

$A_t = 0.0524\,in^2$ (taken from "Machinery's Handbook", 12th edition page 1282, tensile stress area for UNC screw)

Number of bolts — 4

Note — all dimensions are taken from EMD (GM) drawing Number EMD-8274509. This is a representative drawing of a similar assembly.

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Reviewed by Date
Approved by Date

Bolt Geometric Properties

$$I_{xx} = \Sigma(\cancel{I_x}^{0} + Ad^2) = 4(0.0524)(3.25/2)^2 = 0.553 \text{ in}^4$$

$$I_{yy} = \Sigma(\cancel{I_y}^{0} + Ad^2) = 4(0.0524)(1^{5/16}/2)^2 = 0.090 \text{ in}^4$$

$$I_{zz} = 0.553 + 0.090 = 0.643 \text{ in}^4$$

Check of Stress levels

$$\sigma_t = \frac{F_z}{4A_t} + \frac{M_x C_y}{I_{xx}} + \frac{M_y C_x}{I_{yy}} = \frac{220 \text{ Lbs}}{4(0.0524)} + \frac{2129(3.25/2)}{0.553} + \frac{2177(1^{5/16}/2)}{0.090} = 23,180 \text{ psi}$$

$$\tau_v = \frac{(F_x^2 + F_y^2)^{1/2}}{4(A_t)} + \frac{M_z C_{xy}}{I_{zz}}$$

$$\tau_v = \frac{(220^2 + 220^2)^{1/2}}{4(0.0524)} + \frac{1991\left[(3.25/2)^2 + (1^{5/16}/2)^2\right]^{1/2}}{0.643 \text{ in}^4} = 6911 \text{ psi}$$

Comparison To Allowable -

From ASME B+PV code, 1977 edition with summer 79 addendum, Section III, article 2460 of Appendix XVII,

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For Combined tensile and shear the following conditions must be met;

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \leq 1.00$$

where:

F_t = allowable bolt tensile stress = 0.3Su (service level A,B)

F_v = allowable bolt shear stress = 0.124Su (service level A,B)

f_t = calculated bolt tensile stress based on tensile area

f_v = calculated bolt shear stress based on shear area.

Note-The stress will be calculated using the tensile stress area. Due to the very high nozzle loads the calculated tensile stress will still be conservative.

Su = 120 ksi {From ASME B+PV code, 1977 edition, section III, table I-7.3 of appendix I, page 98}

F_t = 0.3(120,000psi) = 36,000psi

F_v = 0.124(120,000 psi) = 14,880psi

f_t = 23,180psi

f_v = 6911psi

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Approved by Date

$$\frac{(23,180)^2}{(36,000)^2} + \frac{(6911)^2}{(14,880)^2} = 0.63 \leq 1.00$$

The bolts are adequate for service level A+B allowables and are therefore adequate for Level c also. This is because Level C loads were used to calculate the actual stress and levels A+B allowables were used for comparison for adequacy.

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MEMORANDUM OF
TELEPHONE CONVERSATION

CALC NO. CQD-014046
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Date: 5/17/84

Time: 1:00 pm

Person Called: David Wright (Name) of Sargent & Lundy (Company)

Person Calling: Charlie Gangone (Name) of LILCo (Company)

Project: Shoreham Project No. 6995-00

Subject Discussed: Identification of the Lube Oil Circulating Pump and Fuel Oil Transfer Pump Assemblies

Summary of Discussion, Decisions and Commitments:

Charlie (LILCo) provided the following information:

Lube Oil Circulating Pump	Fuel Oil Transfer Pumps	
	Upper	Lower
Mfg. - Delco	Mfg. - Delco	Delco
Model - B6733-A	Model - A9916M1	020908CC
Frame - 184	Frame - N/A	N/A
Serial - D67	Serial - G63	N/A
	Type - CS	CC

This information was obtained from the master unit (Engine #2).

DW/eg

cc Charlie Gangone - LILco
J. Sinnappan - 30
AEM/APD/DMW - 30

David Wright
Signature

File: CQD-014046 - 30

SARGENT & LUNDY

MEMORANDUM OF TELEPHONE CONVERSATION

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.1.15

Date: 5/17/84

Time: 2:00 pm

Person Called: David Wright (Name) of Sargent & Lundy (Company)

Person Calling: Jim Miller (Name) of Delco Products (Company)

Project: Shoreham **Project No.** 6995-00

Subject Discussed: Electric Motors Model B6733-A and A9961M1

Summary of Discussion, Decisions and Commitments:

The following information was provided by Jim (Delco)

Model No. - B6733-A	Model No. - A9916M1
Frame - 184	Frame - 56
Serial No. - D67	Serial No. - G63
Horsepower - 1 H.P.	Horsepower - 3/4 H.P.
Estimated Wt. - 100 lbs.	Estimated Wt. - N/A
Shaft O.D. - 1.125 in	Shaft O.D. - 0.5 in. or 0.625 in. (consult Nema Standard)

Jim also stated that bearing to bearing dimensions could be estimated from a Nema chart of standard frame sizes.

DW/eg

cc
J. Miller - Delco
J. Sinnappan - 30
AEM/APD/DMW - 30

David Wright
Signature

File: CQD-014046 - 30

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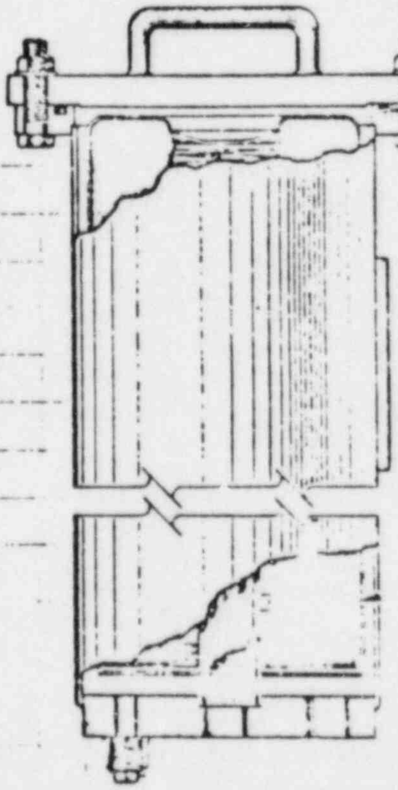
CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.1.16

Client		Prepared by Bruce M Forry	Date 5-14-84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Lube Oil Soakback Filter

Log No. 0803160700



Mfg: Commercial Filters

Model: 1750-10SP

The Lube oil Soakback filter is mounted by a welded bracket to the accessory rack. The purpose of the soakback filter is to take impurities out of the Lubricating system. The filter has no extended masses or moving parts so operability need not be shown. The filter will be shown to maintain structural integrity and pressure boundry.

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CALC NO. CQD-014046
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Client		Prepared by	Date
Project		Reviewed by	Date
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Analysis Methodology :

1. All stress analysis calculations for the Shoreham Lube Oil Soakback Filters are based upon the method of static coefficients per IEEE-344 (1975) - ie. using 1.5 x peak acceleration values. These values are the enveloped accelerations taken from the Shoreham elevated response spectra - pgs. A.1.13, A.1.15 & A.1.16 of CQD File # 014046.

2. All of the most probable modes of failure are examined in this analysis. This includes stress analysis of the bracket-to-filter weld pattern, bracket hold down bolts and stresses with the bracket itself and in the filter.

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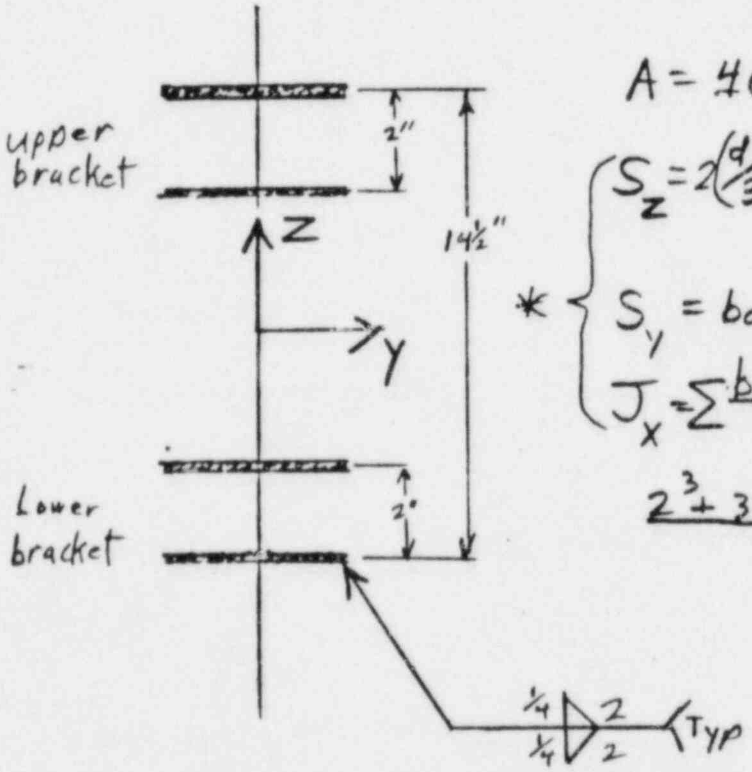
Mounting-Bracket-to-Filter Weld Pattern:

(Minimum Available Weld Area) $W_{ASSEMBLY} \cong 155$ lbs {Note - same filter as found on fuel system, see B.2.53,54 for weight}



$A = 4(2.0 in) = 8.0 in$

$$* \begin{cases} S_z = 2\left(\frac{d^2}{3}\right) = 2\left(\frac{(2)^2}{3}\right) = 2.67 in^2 \\ S_y = bd_1 = (2.0)(10.5 + 14.5) = 47.0 in^2 \\ J_x = \sum \frac{b^3 + 3bd^2}{6} = \frac{2^3 + 3(2)(10.5)^2}{6} + \end{cases}$$

$$\frac{2^3 + 3(2)(14.5)^2}{6} = 323.17 in^3$$

Max. Loading on Weld Pattern: (Dead wt. + Max. Seismic)

$F_x = 155 lb(2.6g)(1.5) = 605 lb$

$F_y = F_x = 605 lb$

$F_z = 155 lb (0.36g)(1.5) = 84 lb$

$\Delta x = 1.25"$
$\Delta y = 0"$
$\Delta z = 7.25"$

Note: See Appendix A for appropriate response spectra.

$M_x = F_y \Delta z + F_z \Delta y = 605(7.25) + 0 = 4,386$ in-lb
$M_y = F_x \Delta z + F_z \Delta x = 605(7.25) + 84(1.25) = 4,491$ in-lb
$M_z = F_x \Delta y + F_y \Delta x = 0 + 605(1.25) = 756$ in-lb

* Design of Welded Structures - O. Blodgett - 8th ed; fig. 7.4-7

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☒ Safety-Related ☐ Non-Safety-Relate

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Max. Corresponding Weld Stresses:

$$f_x = \frac{F_x}{A} + \frac{M_z}{S_z} + \frac{M_y}{S_y} = \frac{605}{8.0} + \frac{756}{2.67} + \frac{4,491}{49.0} = .450 \text{ lb/in}$$

$$f_y = \frac{F_y}{A_w} + \frac{M_x c_z}{J_x} = \frac{605}{8.0} + \frac{4,386(7.25)}{323.17} = 174 \text{ lb/in}$$

$$f_z = \frac{F_z}{A_w} + \frac{M_x c_x}{J_x} = \frac{84}{8.0} + \frac{4,386(1.25)}{323.17} = .28 \text{ lb/in}$$

$$f_r = \sqrt{(450)^2 + (174)^2 + (28)^2} = 483 \text{ lb/in}$$

Allowable Weld Stress $-f_{allow}$: (Design of Welded Structures - p. 7.4-3)

(Assume Weakest Weld Material - E60xx welded onto A-36 steel)

$$f_{allow} = 9,600\, w \text{ lb/in}$$

For Adequacy : $f_r = 483 \text{ lb/in} = 9,600\, w(0.707)$

$$\therefore w = \frac{483}{0.707(9,600)} = \underline{\underline{0.071''}}$$

Since the actual weld leg size is 0.25", the subject weld pattern is adequate as designed

1.

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Client		Prepared by	Date
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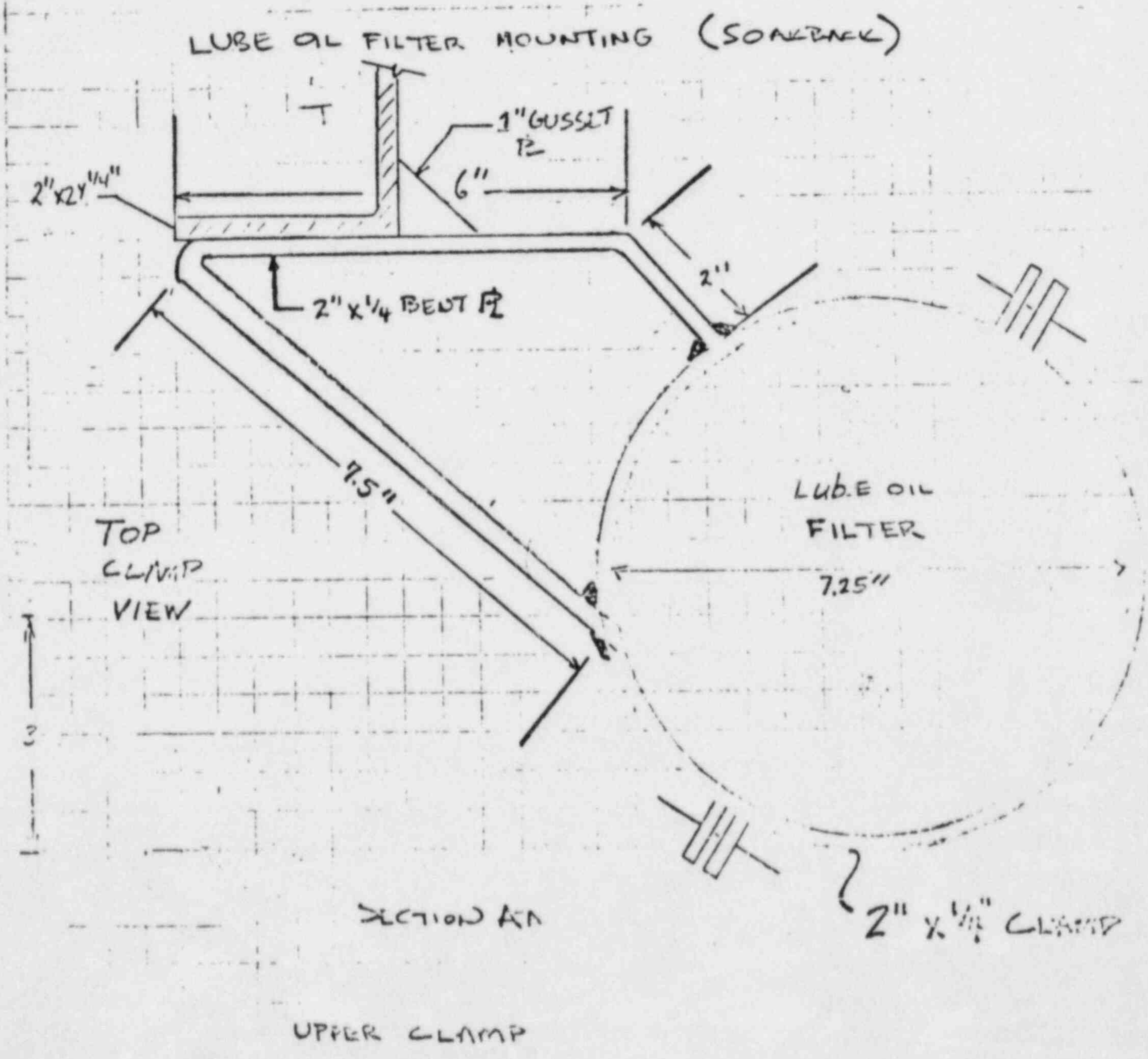
Proj. No. 6995-00	Equip. No.	Approved by	Date

Mounting Bracket Evaluation: (least X-sections used)

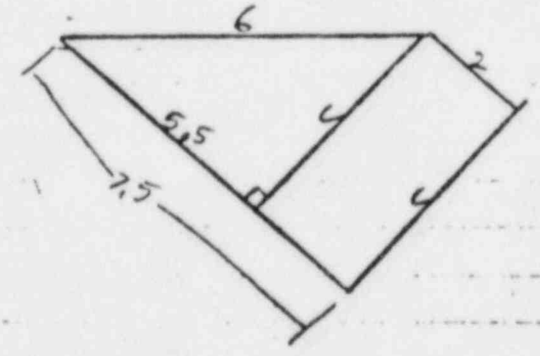


LUBE OIL FILTER IS A MICHIANA FILTER ([illegible])
MODEL NO. 17050-10EP & SERIAL NO. 30487

Form GO 3.08.1 Rev. 2

2. Shoreham Bracket X-Sections (As-Built)

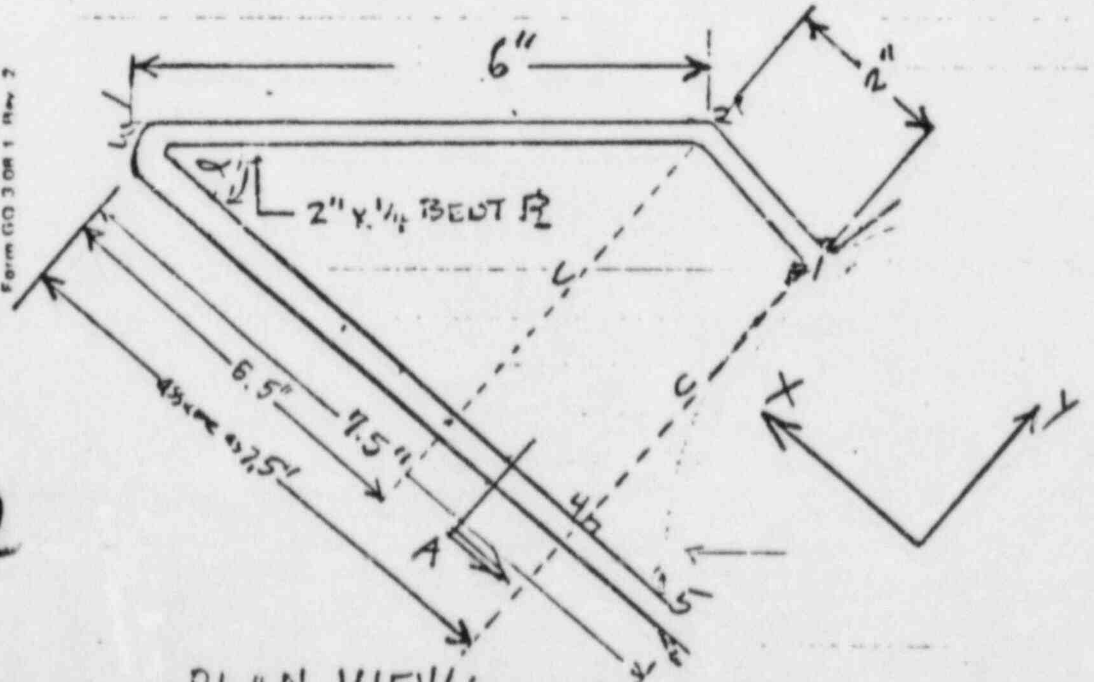
Derivation of Critical X-Section - The critical X-section for these brackets is viewed along the X axis. The value of "c" determines the brackets' moments of inertia and by assuming that the length of $\overline{3\text{-}5}$ = length of $\overline{3\text{-}4}$ the angle α is smaller than in actuality. By having a smaller angle α, the value of "c" is less than it is actually and consequently so are the critical x-section's moments of inertia.



$$6^2 = c^2 + 5{,}5^2$$

$$c^2 = 36 - 30{,}25$$

$$c = \sqrt{5{,}75} = \underline{\underline{2.40''}} \text{ (conservative)}$$



PLAN VIEW (1 Bracket)

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Calcs. For

☒ Safety-Related ☐ Non-Safety-Relat

CALC NO. CQD-014046
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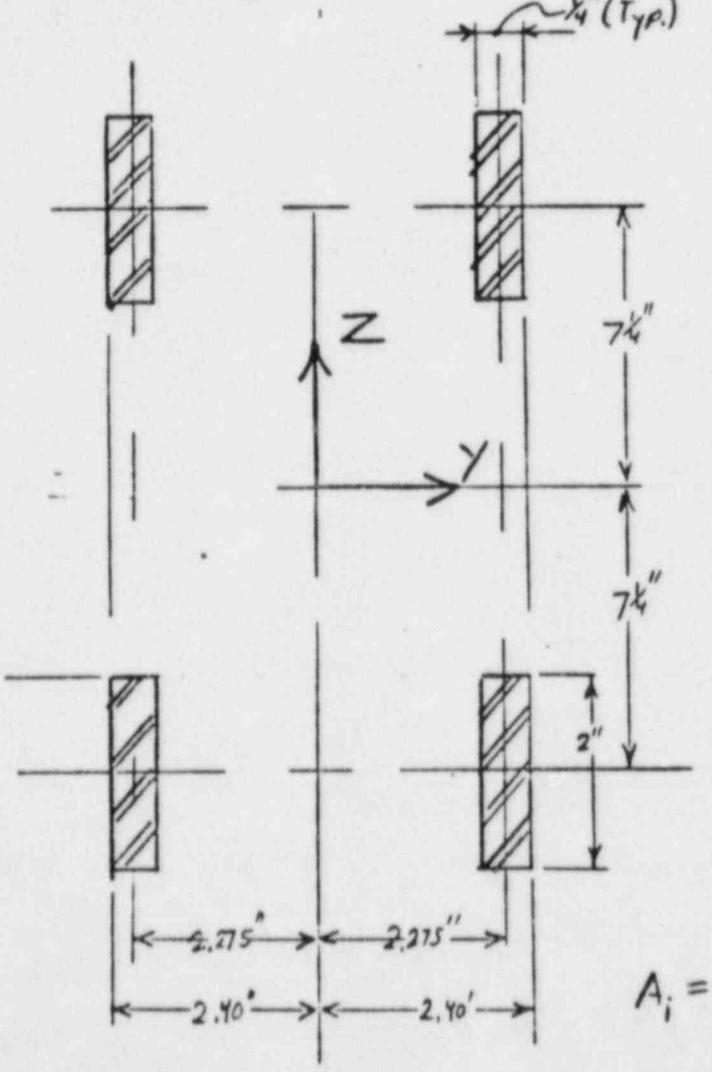
Proj. No. 6995-00	Equip No.	Approved by	Date

Corresponding Critical X-Section Properties:



$A_i = 0.25(2) = 0.50 in^2$

$A = 4(2.0)(0.25) = \underline{\underline{4.0 in^2}}$

$I_{YY} = 4[\frac{1}{12}(0.25)(2)^3] + 4(0.50)(7.25)^2 = \underline{\underline{105.8 in^4}}$

$I_{ZZ} = 4[\frac{1}{12}(2)(0.25)^3] + 4(0.50)(2.275)^2 = \underline{\underline{10.4 in^4}}$

$J_{XX} = 4R = 4\beta b d^3 = 4(0.307)(2)(0.25)^3 = \underline{\underline{0.038 in^4}}$
(Blodgett - pg. 2.10-2)

Form GO 3.08.1 Rev. 2

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Bracket Stress Analysis:

$$\tau_r = \frac{\left[(605)^2+(.84)^2\right]^{1/2}}{2.0} + \frac{(4,386)(0.25)}{0.038} = 7,519 \text{ psi}$$

$$\sigma_t = \frac{605}{2.0} + \frac{756\left(\frac{2.40}{2}\right)}{10.4} + \frac{4,491\left(\frac{8.25}{2}\right)}{105.8}$$

$$\sigma_t = 565 \text{ psi}$$

$$\sigma_p = \frac{565}{2} + \sqrt{\left(\frac{565}{2}\right)^2 + (7,519)^2} = \underline{\underline{7,807 \text{ psi}}}$$

Allowable Stress: (Assuming A-36 steel)

$S_{all} = 1.5 S k_m$ where: $k_m = 1.33$ from Table NF-3552(b)-1 (level B) of the 1977 ASME B&PV Code, Div. 1, Sect. III, Subsection NF.

1.5 = from NF-3260 & 3552 of the 1977 ASME B&PV Code, Div. 1, Sect. III, Subsection NF.

S = 12,600 from Appendices of 1977 ASME B&PV Code, Div. 1, Sect. III, Table I-7.1.

$S_{all} = 1.5(1.33)(12,600) = 25,137 \text{ psi} > \sigma_p = 7,807 \text{ psi}$

∴ brackets are adequate as designed

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CALC NO. CQD-014046
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Calcs. For

☒ Safety-Related ☐ Non-Safety-Rela

Client		Prepared by	Date
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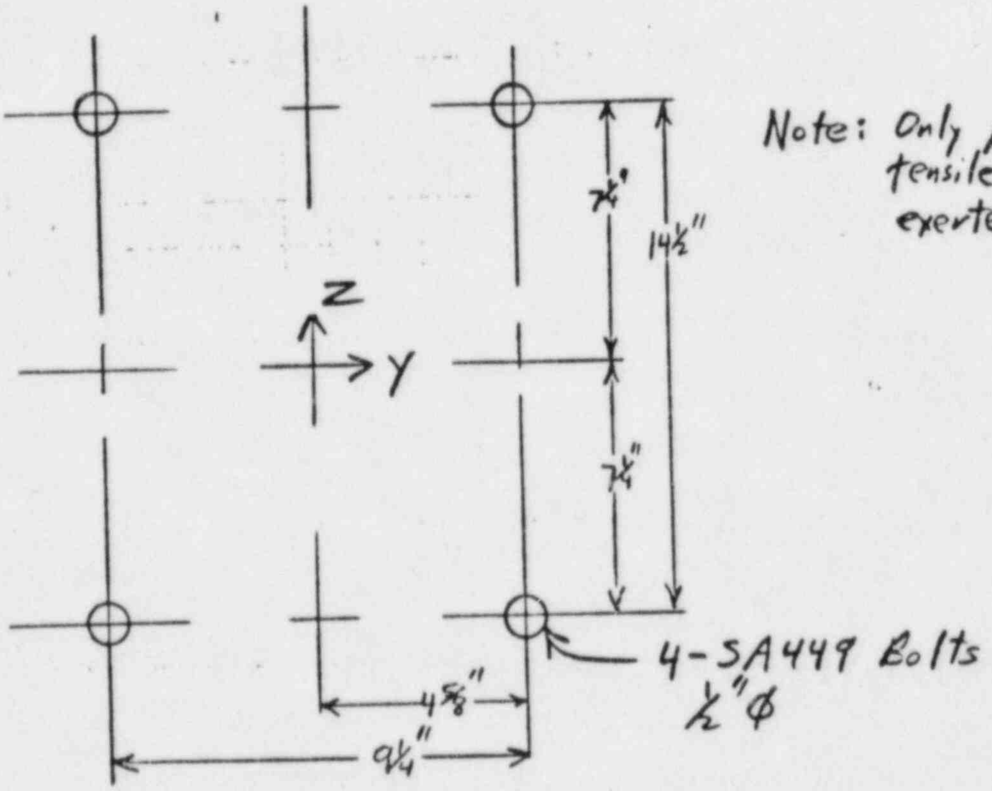
Proj. No. 6995-00	Equip. No.	Approved by	Date

Filter Clamp Bolt Pattern Evaluation:



Note: Only pure shear & tensile forces are exerted on bolts.

$F_x = F_y = 605 \text{ lbs}$

$F_z = 84 \text{ lb}$

$A = 4(0.142) = 0.568 \text{ in}^2$

$I_{yy} = 4(0.142)(7.25)^2 = 29.86 \text{ in}^4$

$I_{zz} = 4(0.142)(4.625)^2 = 12.15 \text{ in}^4$

$J_{xx} = I_{yy} + I_{zz} = 42.01 \text{ in}^4$

Form GQ-3.08.1 Rev. 2

Calcs. For

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Client

Project

Proj. No. 6995-00 Equip. No.

Prepared by Date

Reviewed by Date

Approved by Date

Bolt Stresses: $A_{tension_{\frac{1}{2}''}} = 0.142 in^2$ $A_{shear_{\frac{1}{2}''}} = 0.126 in^2$ } AISC Hdbk 8th ed - pg. 4-14.

$$\sigma_T = f_t = \frac{605}{4(0.142)} = 1,065 psi$$

$$\gamma_{max} = f_v = \sqrt{\left(\frac{605}{4(0.126)}\right)^2 + \left(\frac{84}{4(0.126)}\right)^2} = 1,212 psi$$

Bolt Adequacy Criteria: [1977 ASME B&PV Code, Div. 1, Sect. III, (up to and including 1977 Summer Addenda) Article 2460 of Appendix XVII]

Governing Eqn.: $$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \leq 1.0$$

→ Table 7.3 of Appendix I of 1977 ASME B&PV Code, Div. 1, Sect. III.

where $F_t = 0.30 S_{u_{A\cdot 449}} = 0.30(120) = 36,000 psi$

$F_v = 0.124 S_{u_{A-449}} = 0.124(120) = 14,880 psi$

$$\therefore \frac{(1,065)^2}{(36,000)^2} + \frac{(1,212)^2}{(14,820)^2} \leq 1.0$$

$0.008 < 1.0$ ✓ Adequate as Designed

Conclusions:

The Shoreham E.D.G.S. Lube Oil Soakback Filter is adequate as designed to withstand the maximum loading conditions for Normal, Upset & Emergency plant operating conditions without loss of pressure or structural integrity.

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Calcs. For FIELD SKETCHES

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CALC NO. COD-014046
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PROJ. NO. 6995-00
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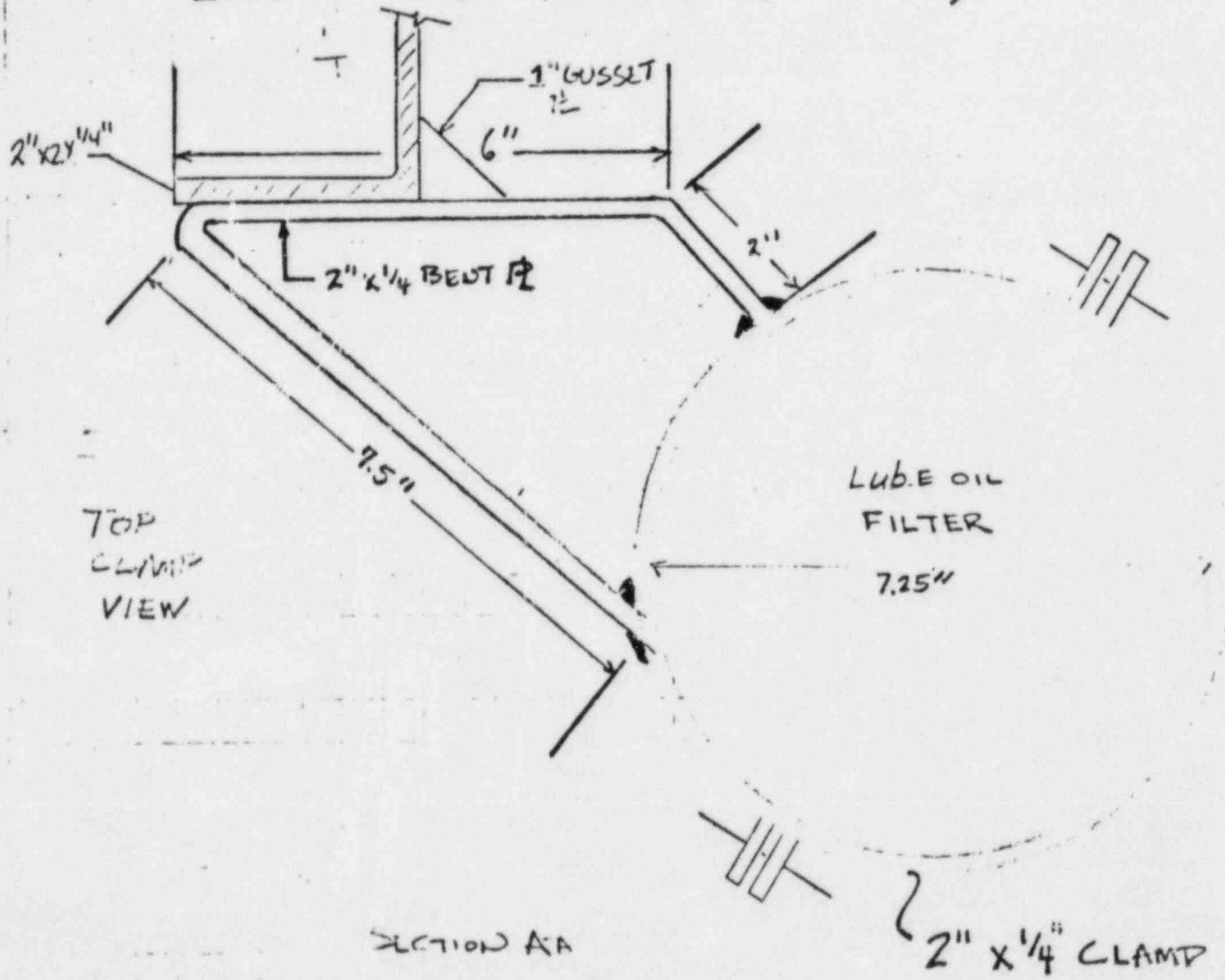
Client		Prepared by	Date 5/4/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

LUBE OIL FILTER MOUNTING (SOAKBACK)



UPPER CLAMP

LUBE [illegible] FILTER IS A [illegible] [illegible] ([illegible])
MODEL NO. 17050-105P ; SERIAL NO. 30487

Sketch obtained during field walkdown

Calcs. For FIELD SKETCHES

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
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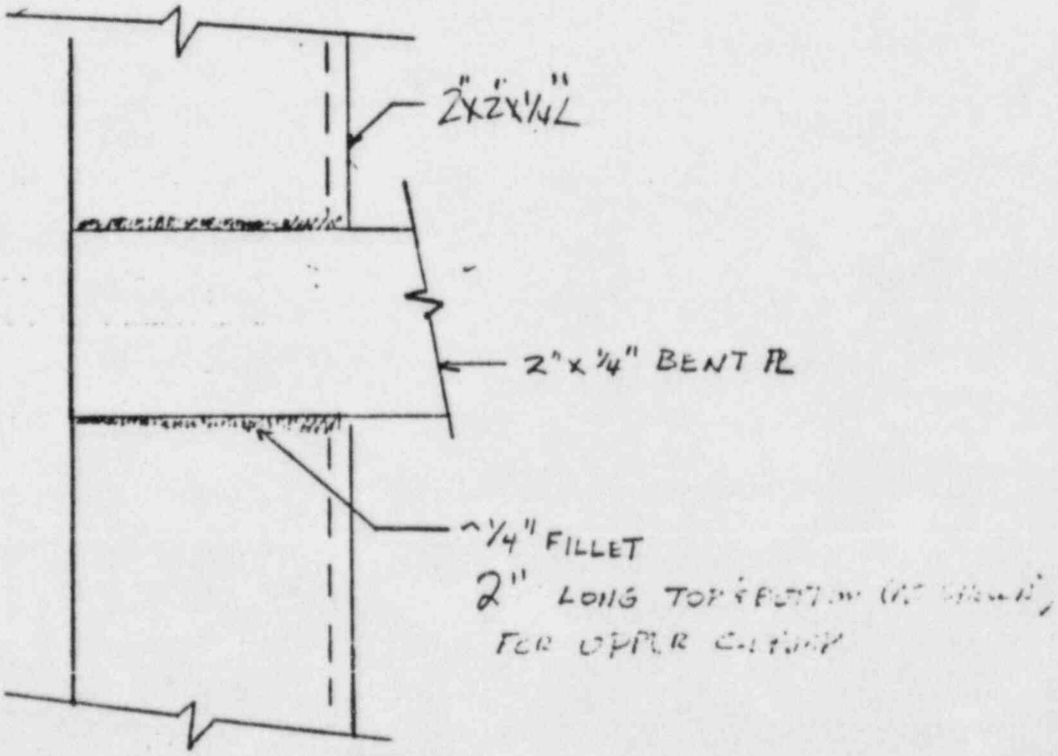
☒ Safety–Related ☐ Non–Safety–Related

Client		Prepared by M Web	Date 5/4/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

THE MOUNTING FOR THE LOWER CLAMP ON THE LUBE OIL FILTER (TO THE ACCESSORY RACK) IS A MUCH MORE RIGID ATTATCHMENT THAN THE UPPER CLAMP. THEREFORE, WE WILL CONSIDER THE LOWER CLAMP MTG. SIMILIAR TO THE UPPER CLAMP MTG. WHICH IS CONSERVATIVE.



Sketch obtained during field walkdown

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Calcs. For FIELD SKETCHES

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Client	Prepared by	Date 5/4/84
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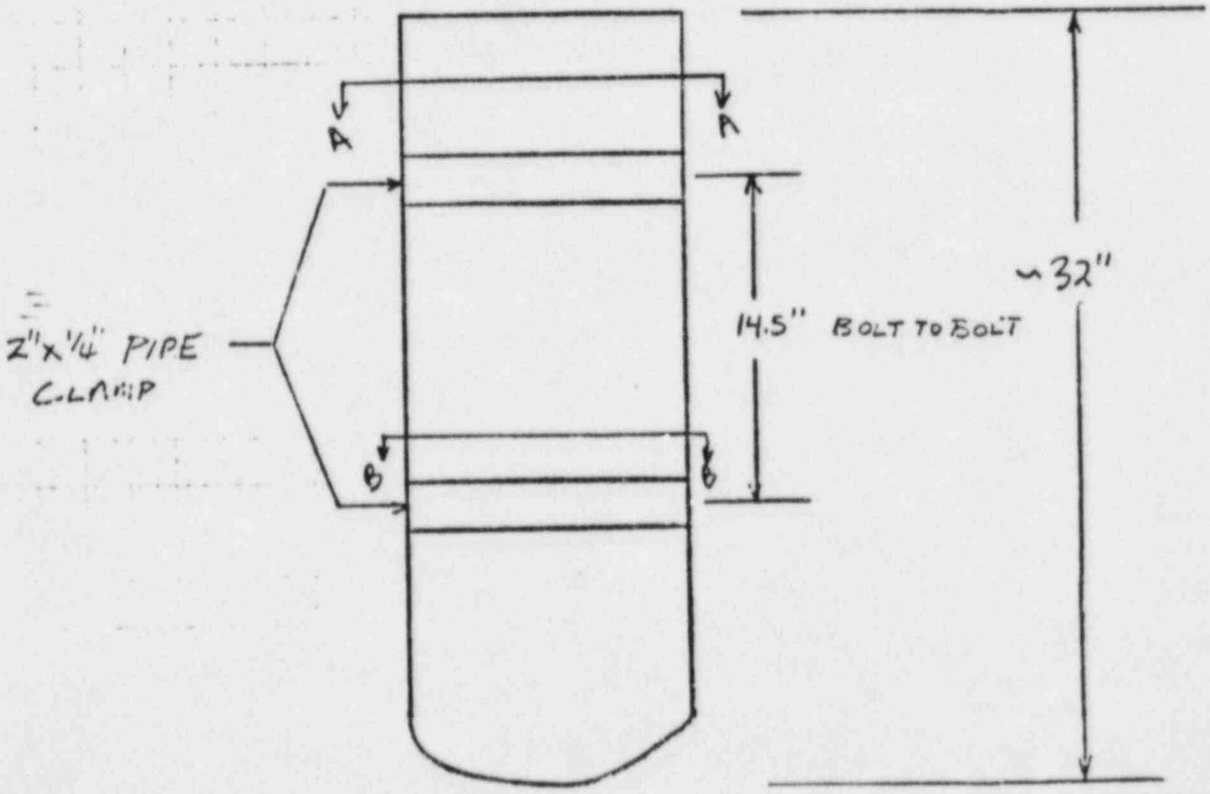
Fuel oil filter Mounting



Sketch obtained during field walkdown

Form GQ 3.08.1 Rev. 2

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Client		Prepared by David Wright	Date 5/17/84
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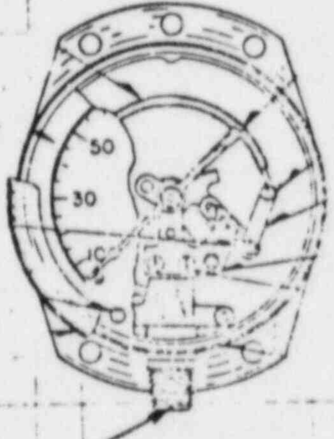
LUBE OIL PRESSURE GAUGE

LOG NO 0803161903



THE LUBE OIL PRESSURE GAUGE WILL NOT CAUSE THE ENGINE TO STOP OR CAUSE FAILURE OF THE SYSTEM TO START. FOR THIS REASON, THE GAUGES ARE NOT ESSENTIAL AND NEED NOT FUNCTION. THE ONLY MODE OF FAILURE THAT COULD CAUSE A PROBLEM WOULD BE A RUPTURE OF THE BOURDON TUBE OR HOUSING CAUSING LEAKAGE OF THE OIL IN THE CONTROL PANEL. THE LEAKAGE RATE WOULD BE SO SLOW THAT IT WOULD NOT CAUSE ANY SIGNIFICANT LOSS OF LUBRICANT. THE MOUNTING SCREWS ON THE SHOREHAM GAUGE WILL BE CHECKED FOR ADEQUACY AND THE REST OF THE GAUGE WILL BE PROVEN ADEQUATE BY THE LASALLE (CECo) ANALYSIS. DUE TO THE EXTREME SIMILARITY (EVEN BETWEEN DIFFERENT MFG.), THIS WILL SHOW THE ADEQUACY OF THE GAUGES FOR SEISMIC LOADING.

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Mounting Screw Check

Screw Size - #10 {note- typical mounting screw size for similar gauges. See ashcroft analysis page B.1.44 of this report.}

Material - SA-307 (assumed material, very low grade)

$A_t = 0.0175 \text{ in}^2$ {from "Machinery's Handbook", 12th edition page 1282, UNC screw, tensile Area.}

No. of Screws - 4

Weight - 2 3/4 Lbs {enveloped maximum weight of the gauges shown in the analysis on the following pages.}

Assumption 8 For ease of analysis assume that the offset of the C.G. of the gauge from the bolt centroid is negligable. Use a 10"g" acceleration to make up for neglicting the moments.

Loading

$$F = F_x = F_y = F_z = 2.75 \text{ Lbs } (10g\text{'s}) = 27.5 \text{ Lbs}$$

$$\sigma_t = F/4A_t = 27.5 \text{ Lbs}/4(0.0175 \text{ in}^2) = 393 \text{ psi}$$

$$\tau_r = F\sqrt{2}/4A_t = 27.5 \text{ Lbs}\sqrt{2}/4(.0175 \text{ in}^2) = 556 \text{ psi}$$

$$\sigma_p = \frac{393}{2} + \sqrt{\left(\frac{393}{2}\right)^2 + 556^2} = 786 \text{ psi} < S_a = 7000 \text{ psi}$$ (ASME Appendix I, table I-7.3)

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PAGE B.1.31

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Marshalltown Figure 24-F
4½ inch Dial Size

Figure 24-F

USAGE—For pressures above atmospheric. Usable on air, oil, gas, water or any other pressure medium which does not attack brass. Install protective syphon when used on steam.

DIAL SIZES—3½", 4½" 6", 8½" and 12", except Fig. No. 24P and Fig. No. 24PC are available in the 4½" and 6" sizes only.

BOURDON TUBE—Phosphor bronze.

MOVEMENT—Rugged - bronze bushed - independent mounting.

ACCURACY—Within 1% of total scale range in middle half of scale - 1½% elsewhere.

FIG. NO. 24F

The front flange of the aluminum case has holes for three mounting screws which are covered by the ring when flush mounted in a panel. The edge of the front flange is threaded to receive an aluminum die cast ring on 3½", 4½" and 6" sizes. 8½" and 12" sizes have a sand cast aluminum hinged and pinned ring. Case and ring have oven baked black enamel finish. ¼" male lower back connection is standard.

Molded Case

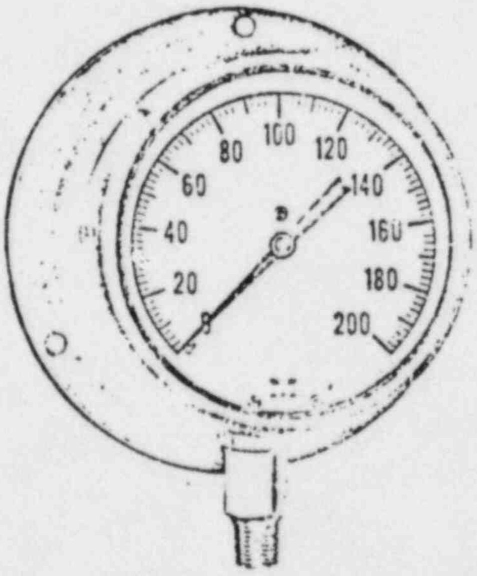
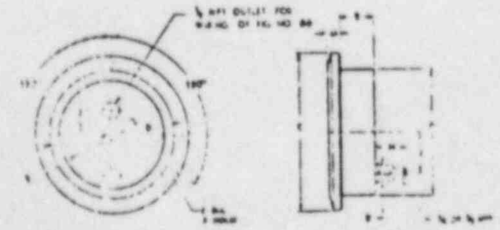


Fig. No. 24

Fig. No. 24F

MOLDED CASE -- FLUSH MOUNTING



Gauge Size	A	B	C	D	E	Panel Holes	H ¼ NPT	H ½ NPT	J	K	Panel Cutout	L	[illegible]
3½	3⅝	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1⅜	[illegible]	[illegible]	⅝	[illegible]		[illegible]
4½	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	⁹⁄₃₂	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
6	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	⁹⁄₃₂	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
8½	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1½	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
12	[illegible]	[illegible]	[illegible]	13	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

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Client		Prepared by	Date
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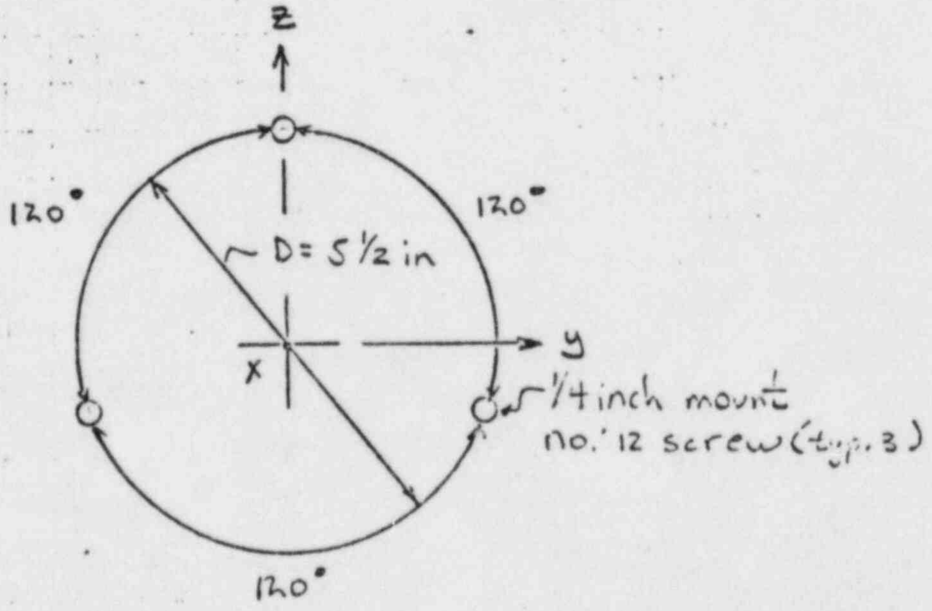
Check of Mounting Screws

The Figure 24-F is panel mounted. The mounting holes are 1/4 inch diameter so assume a #12 mounting screw.



Weight – 2 Lbs { See Phone memo dated 3/27/84 between M. Dixon (Marshalltown) and D. Wright (S+L) attached on Pg. B.1.59 }

Because of the size, shape and weight of the gauge along with the method of mounting it is obvious that the gauge will be seismically rigid ie. $f_1 > 33$ Hz ZPA levels will be used for seismic loading criteria as follows:

ZPA Horizontal = 1.4 g's

ZPA Vertical = 1.5 g's

{ from CQD(S+L)-009249, Rev. 00 dated 8/10/83 on the subject of Elevated Response Spectrum for EDGS Panel mounted Instruments. }

Note – it has been shown in Appendix

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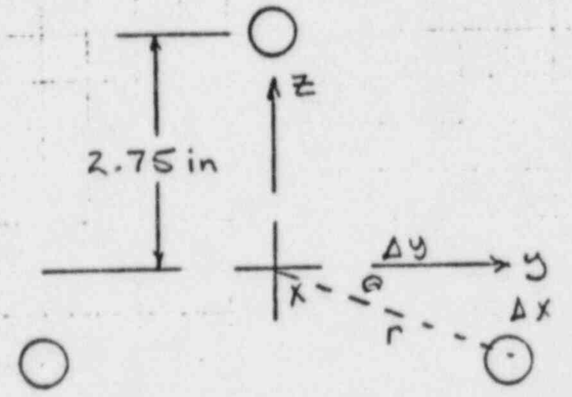
Geometric properties of bolt pattern



r = 2.75in , θ = 120° - 90° = 30°

Δx = r sinθ = 2.75 sin 30° = 1.375in

Δy = r cos θ = 2.75 cos 30° = 2.38 in

$$\bar{Z} = \frac{\bar{Z}_1 A_1 + \bar{Z}_2 A_2 + \bar{Z}_3 A_3}{A_1 + A_2 + A_3} = \frac{Z_1 + Z_2 + Z_3}{3} = \frac{(2.75 - 1.375)\times 2 + 5.5}{3}$$

from base of gauge

$\bar{Z}$ = 2.75 in = center of the circle D = 5.5in

Form G3.3.08.1 Rev. 2

Loading, seismic + Weight

$F_x = Wt(a_x) = 2Lbs(1.4g's) = 2.8Lbs$

$F_y = Wt(a_y) = 2Lbs(1.4g's) = 2.8Lbs$

$F_z = Wt + wt(a_z) = 2Lbs + 2Lbs(1.5g's) = 5Lbs$

$M_x = 0.0$ in-lbs (no offset of c.g. from bolt centroid)

$M_y = 0.0$ in-Lbs "

$M_z = 0.0$ in-Lbs "

Nozzle Loading Due to pressure

The area will be conservatively based on outside diameter of the 1/4 NPT stem. O.D. = 0.540 from Machinery's Handbook, 12th edition page 1399

$F_{x\,\text{Pressure}} = (\text{Pressure})(\text{Area})$

$\text{Area} = \text{Area at stem} = \frac{1}{4}\pi d^2 = \frac{1}{4}\pi(.540)^2 = 0.229\ in^2$

Pressure = 300psi (maximum gauge value)

$F_x = 300psi(.229\ in^2) = 69\ Lbs$

Note - the tubing will transmit very little seismic loading to the gauge. This is due to its small weight and many supports.

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Loading, seismic + weight + Pressure

$F_x = 2.8 \text{ Lbs} + 69 \text{ Lbs} = 72 \text{ Lbs}$

$F_y = 2.8 \text{ Lbs}$

$F_z = 5 \text{ Lbs}$

$M_x = 0.0 \text{ in-lbs}$

$M_y = 0.0 \text{ in-lbs}$

$M_z = 0.0 \text{ in-lbs}$

Bolt Data

Size – #12 {see phone memo dated 3/27/84 between M. Dixon (Marshalltown and D. Wright (S+L) see page B.1.59}

Material – assume SAE Grade 1, 2 or 3, SA-307

$\text{Area}_t = 0.0242 \text{ in}^2$ {From Machinery's Handbook, 12th edition page 1282, UNC screws, tensile stress Area}

$\sigma_t = F_x / 3A_t = 72/3(.0242) = 992 \text{ psi}$

$\tau_v = [F_y^2 + F_z^2]^{1/2} / 3A_t = [2.8^2 + 5^2]^{1/2} / 3(.0242) = 79 \text{ psi}$

Check Against Allowable (see pages B.1.13, .13.1 and .13.2)

*$S_u = 60{,}000 \text{ psi}$

* From ASME B+PV code, 1977 edition, section III, table I-7.3

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For details on symbols see page B.1.13, 1.13.1, 1.13.2

$F_t = 0.3\, S_u$ (for service levels A+B) $= 0.30(60) = 18000$psi

$F_v = 0.124\, S_u$ (") $= 0.124(60) = 7{,}440$psi

$f_t = 992$ psi

$f_v = 79$ psi

For Combined Shear + Tension the following Condition Must be Satisfied:

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \leq 1.00 \quad \text{so:}$$

$$\frac{992^2}{18{,}000^2} + \frac{79^2}{7440^2} = 0.0032 \leq 1.00$$

The bolts are adequate to withstand service level A, B + C loadings.

Inlet Stem

Stem size: 1/4" NPT

inlet hole: 0.16in (see phone memo on page B.1.59)

Tube, Connection Size: 1/4 in O.D. x 0.035in wall
stainless steel TP-304,

Section Modulas of Stem

From Machinery's Handbook, 12th edition,

1/4" NPT: Pipe O.D.: 0.540

Thread Height: 0.04444in

$\text{Min. O.D} = 0.540 - 2(0.04444) = 0.451''$

$\text{Max I.D.} = 0.16\text{in}$

$A = \frac{\pi}{4}(OD^2 - ID^2)$

$A = \frac{\pi}{4}(.451^2 - .16^2) = 0.14\text{in}^2$

$Z = \frac{\pi(OD^4 - ID^4)}{32\,OD} = \frac{\pi(.451^4 - .16^4)}{32(.451)}$

$Z = 0.009\text{ in}^3$

Section Modulas of Tube

$O.D. = 0.25\text{in}$

$I.D. = 0.25\text{in} - (0.035)2 = 0.18\text{in}$

$\text{Area} = \frac{\pi}{4}(OD^2 - ID^2)$

$= \frac{\pi}{4}(.25^2 - 0.18^2) = 0.024\text{in}^2$

$Z = \frac{\pi(OD^4 - ID^4)}{32\,OD} = \frac{(.25^4 - .18^4)\pi}{32(.25)}$

$Z = 0.001\text{ in}^3$

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Stem Material - Extruded Brass (Phone memo, page B.1.59)

Allowable Stress - 8000 psi (ASME Appendix I, table I-5.4 for all Brass shown, pg. 118 @100°F, Section III)

Tube Material - stainless steel TP-304 (assumed for a conservi analysis, not actual.

Allowable Stress - 18,800 psi (ASME Appendix, table I-7.2 for TP304 stainless, page 84. @100°F. Section III)

Comparison of strength:

$$A_{stem}\left(\frac{S_{stem}}{S_{tube}}\right) = 0.14\,in^2\left(\frac{8,000}{18,800}\right) = 0.059\ in^2$$

$$Z_{stem}\left(\frac{S_{stem}}{S_{tube}}\right) = 0.009\left(\frac{8}{18}\right) = 0.004\ in^3$$

$$A_{stem}/A_{tube} = 0.062\,in^2/.024\,in^2 = 2.6$$

$$Z_{stem}/Z_{tube} = .004/.001 = 4.0$$

The stem is 4 times stronger than the tube in bending. The stem will not fail before the tube.

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Marshalltown Figure 23

2½ inch Dial Size

Drawn Steel Case

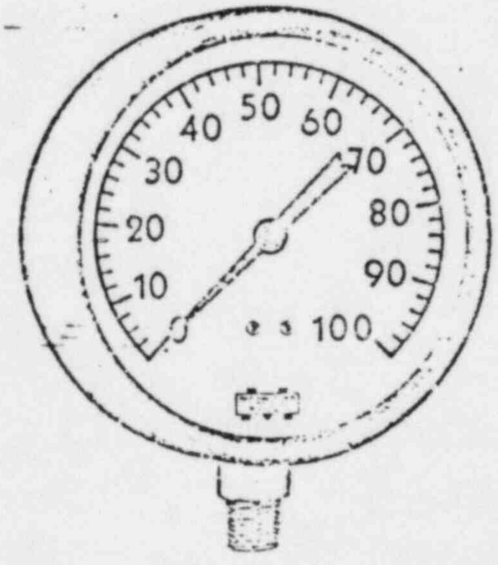


Fig. No. 23

USAGE—For pressures above atmospheric. Usable on air, oil, gas or water or any other pressure medium which does not deteriorate brass. Install protective syphon when used on steam.

DIAL SIZES—2", 2½", 3½" and 4½", except that the 4½" is not available in Fig. Nos. 23B and 23D.

CASE—Drawn steel - phosphatized for rust resistance and finished in oven baked black enamel.

RING—Same as above.

BOURDON TUBE—Phosphor Bronze.

MOVEMENT—Brass - Independent Mounting.

ACCURACY—Within 2% of total scale range in middle half of scale - 3% elsewhere.

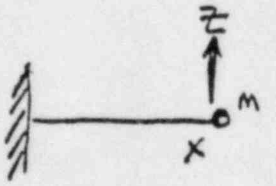
FIG. NO. 23

¼" male bottom connection is standard. When so specified, a ⅛" male bottom connection can be furnished on 2" and 2½" sizes.

Mountings: The Figure 23 gauge is stem Mounted.

Weight: ½ Lb (phone memo page B.1.59)

Stem Size: ¼ in Npt (phone memo page B.1.59)



$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{m}}$$

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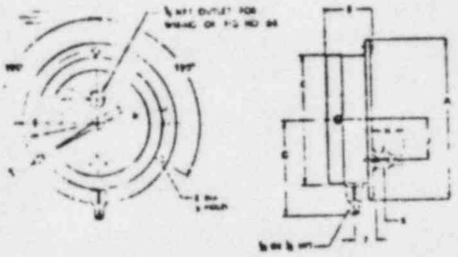
with $K = \frac{3EI}{l^3}$ { from "Marks Standard Handbook for Mechanical Engineers, 8th edition, Mc Graw-Hill Books, page 5-70 }

where l = length of the stem which will be estimated from the table below.

Gauge Dimensions



Gauge Size	A	B	C	D	E	Panel Holes	F	¼ NPT G	¼ NPT H	½ NPT G	½ NPT H	J	K	Panel Holes	L
3½	4¾	1½	3 11/16	4¼	7/32	¼	11/16	3⅛	1 5/16			1 5/32	⅜	1	
4½	5⅞	1 27/32	4¾	5⅜	9/32	5/16	⅞	4	1⅝	4⅛	1 11/16	1 15/32	⅜	1¼	1⅛
6	7⅝	1 27/32	6 5/16	7	9/32	5/16	1 3/16	5⅛	1 9/16	5¼	1 11/16	[illegible]	⅞	1¼	1⅛
8½	10¾	2⅛	8 13/16	[illegible]	9/32	5/16	1 1/16	6 11/16	1⅝	[illegible]	1 9/16	2 13/32	⅞	1¼	1⅛
12	14	2½	12 5/16	13⅝	9/32	5/16	[illegible]	8¼	1⅝	8½	[illegible]	[illegible]	⅞	1¼	1¼

stem length = $G-(C/2) = 3\frac{1}{8}'' - (3\frac{11}{16}'')/2 = 1.28$ in ($3\frac{1}{2}''$ gauge

$4'' - (4\frac{3}{4}'')/2 = 1.625$ in ($4\frac{1}{2}''$ gauge

$5\frac{1}{8}'' - (6\frac{5}{16}'')/2 = 1.97$ in ($6''$ gauge

Since our gauge is $2\frac{1}{2}''$ we will use $l = 1.28$ in 8.

Using the relationship developed previously:

¼ Npt O.D. min (inside of thread) = 0.451 in
I.D Max = 0.15 in (phone memo page B.1.59)

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$$I = \frac{\pi(OD^4 - ID^4)}{64} = \frac{\pi(.451^4 - .15^4)}{64}$$

$$I = 0.002 \text{ in}^4$$

$E_{ss} = 15 \times 10^6$ psi {From "Strength of Materials", 2nd edition by John Cernica, copyright 1977 by Holt, Rineholt + Winston, page 463, mean value}

$$K = \frac{3EI}{l^3} = \frac{3(15\times10^6)(0.002)}{(1.28)^3} = 42,915 \text{ in/Lb}$$

$f_n = \frac{1}{2\pi}\sqrt{\frac{42,915}{(1/2 \text{ Lb}/386.4)}} = 917$ Hz, the gauge is rigid

<u>loading, weight + seismic</u>

$F_x = Wt(a_x) = 0.5$ Lbs $(1.4g's) = 0.7$ Lbs

$F_y = wt(a_y) = 0.5$ lbs $(1.4g's) = 0.7$ Lbs

$F_z = wt(a_z) + wt = 0.5(1.5) + 0.5 = 1.25$ lbs

(Note - apply weight to cause bending moment to be as large as possible.)

$M_x = F_z G = 1.25(3\tfrac{1}{8}'') = 3.91$ in-lbs

$M_y = 0.0$ in-lbs

$M_z = F_x G = 0.7(3\tfrac{1}{8}'') = 2.19$ in-lbs

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Loading, Pressure

$$F_{y_p} = (\text{Pressure})(\text{Area})$$
$$= (300\text{psi})\left(\frac{\pi}{4}(0.540_{\text{in}}^{2}\right) = 69\text{ Lbs}$$

$$\sigma_t = (F_y + F_{y_p})/A_{st} + \frac{M_x + M_z}{Z_{st}}$$ (from previous Marshalltown 1/4 in NPT stem)

$$\sigma_t = \frac{(0.7\text{lbs} + 69\text{lbs})}{0.14\text{in}^2} + \frac{2.91 + 2.19}{.009\text{in}^3} = 1176\text{psi}$$

$$\tau_v = [F_x^2 + F_z^2]^{1/2}/A_s = [0.7^2 + 1.25^2]^{1/2}/0.14\text{in}^2 = 10.\text{psi}$$

$$\sigma_p = \frac{\sigma_t}{2} + \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau_v^2} = \frac{1176}{2} + \sqrt{\left(\frac{1176}{2}\right)^2 + 10^2}$$

$$\sigma_p = 1176\text{psi}$$

S = 8000 psi (ASME Appendix I, table I-8.4, same value for all Brass listed, Section III)

$S > \sigma_p \Rightarrow$ acceptable.

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Bourdon Tube Analysis

Bourdon tube information was not available due to the proprietary nature of the data. This was the policy expressed by MR. MiLt Dixon of Marshalltown (see ref. 3 phone Memo.). We can however draw some conclusions from the information provided by Marsh gauges. The actual stress in the bourdon tube of the Marsh gauge was found to be: (seismic only)

$$\sigma_{seis} = 377psi$$

This level is much smaller than the stress in the tube due to internal pressure. For this reason the seismic loading is assumed to have a negligible affect on the gauge integrity and the gauge is qualified for use.

Note - for a more detailed bourdon tube analysis see the Ashcroft bourdon tube check in this analysis. The deflection due to normal use was 113 times greater than the deflection due to seismic loading

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Fatigue Analysis, Figure 23, 24-F

Engine opperating Vibrations:

A conservative estimate of the g' stress due to Engine opperational vibration is the following:

$$\sigma_{p_{\text{eng. vibe}}} = \sigma_{p_{\text{seismic}}} \left[\frac{\text{maximum Acc, Eng. vib.}}{\text{Min Acc, seismic}} \right]$$

minimum Seismic = 1.4 g's

Maximum Engine Vibration = 1.0 g's {From CQD-012628, Rev.00 dated 2/22/84 in titled Response of an EMD 645 Engine to impulsive and Steady state loads}

The critical area of the gauges (envelope both) is the stem of the Figure 23 gauge;

$$\sigma_{p_{\text{seis}}} = 1176 \text{ psi}$$

$$\sigma_{p_{\text{Eng Vibration}}} = 1176 \text{psi} \left(\frac{1.0}{1.4} \right) = 840 \text{psi}$$

Since both values are well below the endurance limit of 12,000psi the gauges are qualified for use on the engines.

(ASME B+PV code, Appendix I, section III table I-9.1 fig. I-9.2. Su = 18,000psi)

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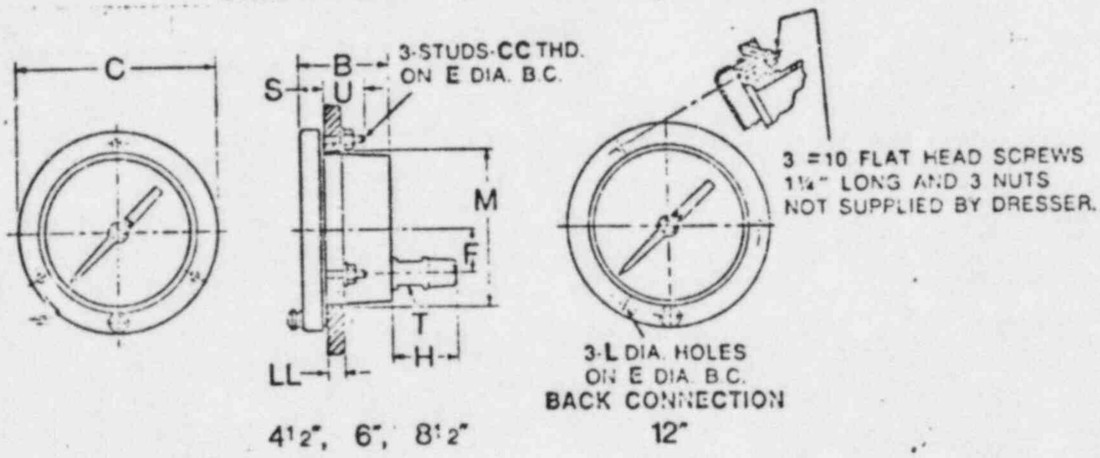
Ashcroft 45-1377A,S-02B

4½ inch Gauge

CASE TYPE 1377 (OPEN FRONT)/1377-S (SOLID FRONT)



CASE TYPE 1377 (OPEN FRONT)

Dial Size Inches	B	C	E	F	H	L	M	S	T	U	CC	LL	Weight (lbs.)
4½	2¾	6¾	5⅞	1⅜	1⁹⁄₁₆	—	4⅜	⅜	⅜	¾	#10-24	¼ / ½	2½
6	2½	7⅝	7	2¼	1⁹⁄₁₆	—	6½	⅜	⅜	¾	¼-20	½ / ½	2½
8½	2¼	10⁹⁄₁₆	9½	2⅝	1⁹⁄₁₆	—	9	⅜	⅜	¾	½-20	½ / ½	5½
12	3	13⁹⁄₁₆	13	2⅛	1½	.196	12½	⅜	⅜	¾	#10	½ / ½	15½

CASE TYPE 1377 (SOLID FRONT)

Dial Size Inches	B	C	E	F	H Max.	H Min.	M	S	T	U	CC	LL	Weight (lbs.)
4½	2¾	6¾	5⅞	1⅜	1⅛	1⅝	4⅜	⅜	⅜	¾	#10-24	[illegible]	2½
6		7⅝	7	2¼			6½				¼-20		[illegible]
8½		10⁹⁄₁₆	9⅛	2⅝			9				½-20		[illegible]

Note - since there may be both open and solid front items in the plant, both will be addressed Via an enveloping approach.

Maximum pressure at which a gauge is continually operated should not exceed 75% of full scale range.

Select:

1. Case type number — Table A
2. Dial size — Table A
3. Bourdon System (ordering code) — Table B
4. Connection: Location — Table A: Size — Table B
5. Mounting accessory or variation (if required) — Table A
6. Pressure Range — Table C on page 12
7. Accessories and optional features — pages 15-20

Example:

1279-S	4½"	TA	Back ½ NPT	w/1278 Ring	0/2000 psi
phenol-solid front	4½"	AISI 316 st. st. system	Back Conn. ½" NPT	with 1278 Ring	0/2000 psi

TABLE A — CASE SELECTION

Case Type Number	Dial Size	Case Style	Case: Material Finish	Style Ring: Material Finish	Mounting and Connection
1279	4½, 6, 8½	Open Front	Phenol Black	Snap Stainless Steel	Stem – Lower or Back Surface – Lower or Back Flush – Back: order 1278 ring. (see page 18)
1279(*)SL** 1279(*)SH**	4½	Solid Front	Phenol Black	Threaded (reinforced) polypropylene)	
1320*	4½	Open Front	Stainless Steel Polished	Bayonet Lock Stainless Steel Polished	Stem – Lower or Back Surface–Lower: Specify XBF Surface–Back: Specify XBQ Flush – Back: Specify XBC
1377	4½, 6, 8½, 12	Open Front	Aluminum: 4½, 6, 8½ Cast iron: 12" Black epoxy coated	Hinged Steel Black epoxy coated	Flush – Back connection only
1377-S	4½, 6, 8½	Solid Front			
1379	4½, 6, 8½, 12	Open Front	Aluminum Black epoxy coated	Threaded Aluminum: 4½, 6 Bronze: 8½, 12 Black epoxy coated	Stem – Lower or Back Surface – Lower or Back Flush – Back
1379-S	4½, 6, 8½	Solid Front			
2462	6	Solid Front	Polypropylene (fiberglass reinforced) Black	Bayonet Lock Polypropylene Black	Stem – Lower or Back Surface – Lower or Back: Specify XBF Flush – Back: Specify XBQ

*Can be Liquid Filled - see page 13. ** - Lower conn. only. For field conversion order Liquid Filled Conversion Kit.

TABLE B — BOURDON SYSTEM SELECTION

Ordering Code	Bourdon Tube and Tip Material (all joints TIG welded except "A")	Socket Material	Tube Type: Drawn or Bored	Pressure Range (PSI)	NPT Connection (3)	
					Open Front	Solid Front
A	Grade A Phosphor Bronze Tube — Brass Tip. Silver brazed	Brass	Drawn	12/1000	¼	½
B	AISI 4130 alloy steel	AISI 1019 steel	Drawn	12/1000	¼	½
D	AISI 4130 alloy steel	AISI 1019 steel	Bored	1000/20,000	½	½
			Drawn (spiral)	30,000/100,000†	—	¼ high pressure
R	AISI 316 stainless steel	AISI 1019 steel	Drawn	12/1000	¼	½
RT	AISI 316 st. st. tube AISI 1019 steel tip	AISI 1019 steel	Bored	1000/20,000	½	½
S	AISI 316 stainless steel	AISI 316 stainless steel	Drawn	12/1000	¼	½
TA	AISI 316 stainless steel	AISI 316 stainless steel	Bored	1000/20,000	½	½
			Drawn (spiral)	20,000/80,000†	—	¼ high pressure
P	K Monel	R Monel	Drawn	12/1000	¼	½
O	K Monel	R Monel	Bored	1000/20,000	½	½
			Drawn (spiral)	[illegible]†	—	¼ high pressure
F(1)	E-Brite 26-1	E-Brite 26-1	Drawn	[illegible]/1000(2)	—	½ only
FT(1)	E-Brite 26-1	E-Brite 26-1	Bored	[illegible]	—	½ only

(1) available with 4½" solid front cases only.

†100,000 [illegible] — available only in 6 and 8½" Type 1377/1379 solid front cases.

(2) other ranges on application.

(3) Optional connections [illegible]
½ NPT [illegible]
¼ NPT [illegible]

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Mounting Screw Check

Screw Data:

Screw size : #10 {From Dresser Industries, Ashcroft gauge bulletin DU-1, Duragauge Press. Gauge Data found on B.1.44 of this report.}

Number : 3

Material : SA-307 (assumed, SAE grades 1, 2 or 3)

A_t : 0.0175 in^2 {From Machinery's Handbook, 12th edition page 1282 for UNC screw, tensile stress Area.}

Gauge Wt : $2\frac{3}{4}$ Lbs (env[open face + solid face])

The gauge is seismically rigid as can be seen by the method of mounting and by no extended masses being present. Zpa levels will be used.

loading, Seismic + Weight

$F_x = Wt(a_x) = 2.75 \text{Lbs}(1.4 g\text{'s}) = 3.85 \text{Lbs}$

$F_y = Wt(a_y) = 2.75 \text{Lbs}(1.4 g\text{'s}) = 3.85 \text{Lbs}$

$F_z = Wt(a_z) = 2.75 \text{Lbs}(1.5 g\text{'s}) = 4.125 \text{Lbs}$

$M_x = M_y = M_z = 0.0$, No extended masses

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Calcs. For

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Pressure loading

F_{x_p} = Pressure (stem Area)

Stem Size – 1/2" NPT (solid front, 1/4" npt open front)

Max. O.D. = 0.840 in (standard table)

$$A = \frac{\pi}{4} OD^2 = \frac{\pi}{4}(.840^2) = 0.554 \text{ in}^2$$

$$F_{x_p} = 300 \text{ Lbs}(.554 \text{ in}^2) = 166 \text{ Lbs}$$

$$\sigma_t = (F_x + F_{xp})/3A_t = (3.85+166)/3(.0175) = 3235 \text{ psi}$$

$$\tau_v = [F_y^2+F_z^2]^{1/2}/3A_t = [3.85^2+4.125^2]^{1/2}/3(.0175) = 107 \text{ psi}$$

Check Against Allowable (treating Stem as Pipe)

Treating the stem as a pipe, From ASME B+PV codes, Section III, Division 1, subsection ND-3000 paragraph ND-3652.1 page 183

$$S_{OL} \leq 1.2 S_h$$

S_{OL} = Occasional Load Stress, This is the stress due to combined loads.

S_h = allowable stress at "T".

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$$S_{OL} \overset{*}{=} \sigma_p = \frac{\sigma_t}{2} + \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau_r^{\;2}} = \frac{3235}{2} + \sqrt{\left(\frac{3235}{2}\right)^2 + 107^2}$$

σ_p = 3239 psi

S_h = 7000 psi [for SA-307 from ASME B+PV code, Section III, Appendix I, table I-7.3 page 99.]

$S_h > \sigma_p \Rightarrow$ the stem is adequate

*Note -- $S_{OL} = \frac{P_{m}D_o}{4t_n} + \frac{0.75i(M_A + M_B)}{Z}$

σ_p is more conservative due to the method of determining the loads and the high seismic coefficients use.

Inlet Stem check

1/4" Npt Data (from Marshalltown Fig. 24-F Analysis)

$A_{\substack{min\\stem}} = 0.14 in^2$

$Z_{\substack{min\\stem}} = 0.009 in^3$ (assuming inlet hole ≈ tube I.D.)

$A_{tube} = 0.024 in^2$

$Z_{tube} = 0.001 in^3$

1/2" Npt Data with 3/8 x 0.035 in thick tube

1/2" Npt O.D. = 0.840 in

thread depth = 0.05714 in

{From Machinery's Handbook, 12th edition pages 1399 for 1/2" NPt}

Min O.D. = O.D. − 2 t_d = .840 − 2(.05714) = 0.726 in

Max I.D. = tube I.D. (assumed) = 3/8 − 2(.035) = 0.305 in

$$A = \frac{\pi}{4}(OD^2 - ID^2) = \frac{\pi}{4}(0.726^2 - 0.305^2) = 0.341 in^2$$

$$Z = \frac{\pi(OD^4 - ID^4)}{32(O.D.)} = \frac{\pi}{32(.726)}(.726^4 - .305^4) = 0.036 in^3$$

S = 8000 psi (ASME Appendix I, table I-8.4, for all Brass Shown pg. 118)
Section III.

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tube Material – TP 304 stainless (worst case)

$$A_{tube} = \frac{\pi}{4}(OD^2 - ID^2) = \frac{\pi}{4}(3/8^2 - .305^2) = 0.037 in^2$$

$$Z_{tube} = \frac{\pi(OD^4 - ID^4)}{32\,OD} = \frac{\pi(3/8^4 - .305^4)}{32(3/8)} = 0.003 in^3$$

S_{tube} = 18,800psi (ASME B+PV code, table I-7.2 of Appendix I, section III, TP-304 stainless pg. 84)

Strength Comparison:

1/4" – O.K. per Marshalltown Fig. 24-F analysis

1/2" stem check

$$A_{stem}\left(\frac{S_{stem}}{S_{pipe}}\right) = 0.341\left(\frac{8}{18.8}\right) = 0.145 in^2$$

$$Z_{stem}\left(\frac{S_{stem}}{S_{pipe}}\right) = 0.036\left(\frac{8}{18.8}\right) = 0.015 in^3$$

Area stem/Area tube = 0.145/0.037 ≅ 4

Z_{stem}/Z_{tube} = 0.015/0.003 = 5

The stem is 5 times stronger than the tube, the tube will fail before the stem.

Form GQ-3.03.1 Rev 2

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Bourdon Tube Check

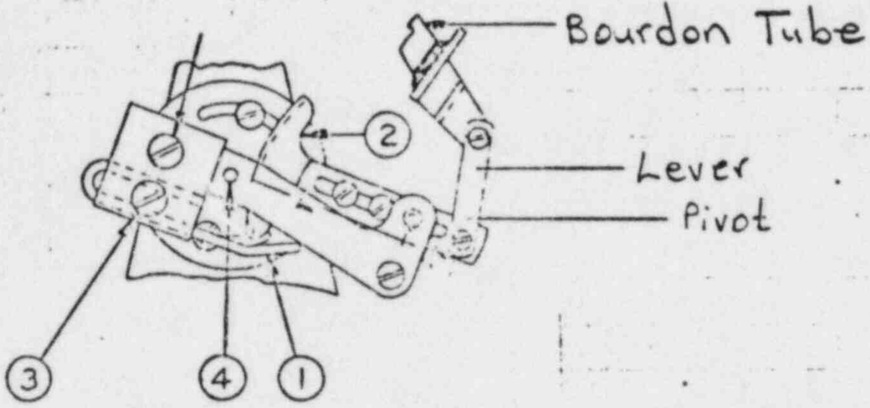
Although no specific details were available as to the thickness and cross sectional properties it will be shown that the contribution to the tube stress due to seismic loading will be negligable. This will be done by developing a relationship between tube deflection and the rated pressure. This will be done by making use of the gauge movement ratios shown below (Table taken from Engineering Data, Ashcroft Gauges Maintenance and repair form No. 250-1353-E.)

Gauge Movement Ratios

Gauge sizes to 3½" inclusive — 12:1 Ratio
"Quality" Gauges, All Sizes — 12:1 Ratio
"Duragauges" — As listed below in Table

Duragauge Type in Terms of Tube Material	Maximum Dial Graduation P.S.I.	Movement Ratio		
		12:1	16:1	20:1
All Materials (Drawn Tubes)	1000	x		
4½" Size — (Bored Tubes)	Over 1000 to 20,000 inclusive		x	
6" Size and Larger (Bored Tubes)	Over 1000 to 14,999 inclusive		x	
6" Size and Larger—Beryllium Copper Only	100 to 500 inclusive	x		
	Over 500 to 20,000 inclusive		x	
6" Size and Larger—All Materials Except Beryllium Copper	15,000 and 20,000 inclusive			x
6" Size 1332	20,000 to 100,000 inclusive		x	

Gauge "Works"



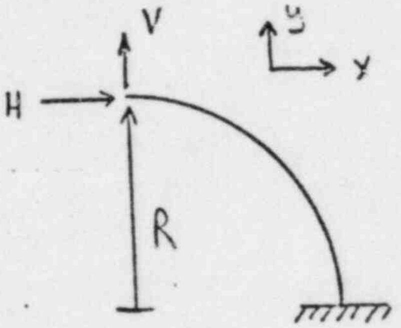
SHCROFT DURAGAUGE GAUGE SINGE RETARD
(Ashcroft Form # 250-1353-E)

Gauge Operation: When pressure is input into the curved bourdon tube it attempts to straighten. As it deflects the attached lever pulls on the driving arm. At the other end of the driving arm there is a gear (2) driving another gear (4) which turns the dial. Making use of this information a relative movement can be obtained. We cannot find the exact deflection of the tube without more information but we can approximate the deflection for comparison purposes.

For purposes of comparison assume $l = 0.5$ in

$\delta = l \sin 22.5° = 0.5 \sin 22.5° = 0.19$ in total tip deflection

For comparison purposes the Marsh J-4853 Bourdon tube dimensions will be used



$R = 4\tfrac{1}{2}$ in $/2$

$R = 2.25$ in assumed because gauge is $4\tfrac{1}{2}$ inch (Ashcroft)

{From "Formulas For Stress and Strain, 5th edition By Roark + Young copyright 1975, McGraw Hill, page 215}

$$\frac{\delta_y}{3} = \frac{(\pi/4)VR^3 + \tfrac{1}{2}HR^3 + M_oR^2}{EI}$$ ⇒ assumed to be ⅓ of the deflection since other sections not included.

$$\frac{\delta_x}{3} = \frac{\tfrac{1}{2}VR^3 + (\tfrac{3}{4}\pi - 2)HR^3 + (\pi/2 - 1)M_oR^2}{EI}$$

$E = 15 \times 10^6$ psi {Strength of Materials, 2nd edition By John Cernica, copyright 1977 by Holt, Rinehart and winston}

bronze

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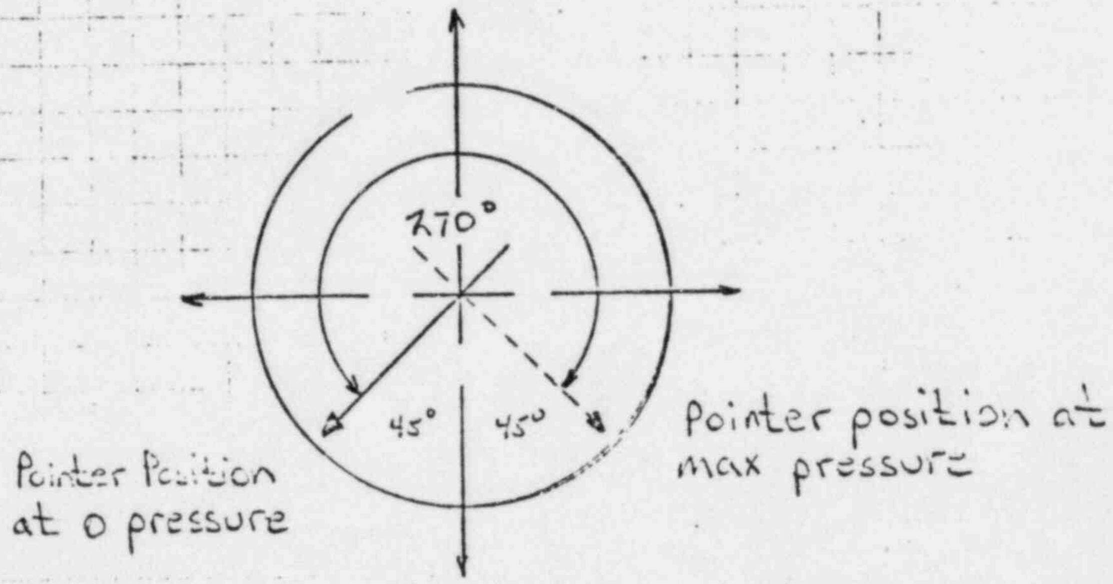
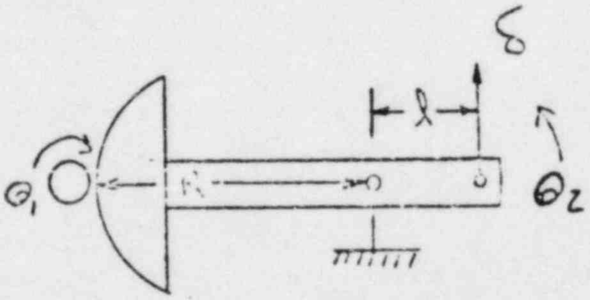
Dial Arrangement



The small gear must turn 270° for an entire scale deflection. Simplifying the works:



$$\frac{\theta_2}{\theta_1} = 12:1 \text{ ratio}$$

so for a full deflection of the scale (270°) the lever arm will deflect:

$$(270°)\left(\frac{1}{12}\right) = 22.5°$$

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H seismic = Wt (a$_x$) = 0.041 Lbs (1.4 g's)(2) = 0.115 Lbs

V seismic = Wt (a$_z$) = 0.041 Lbs (1.5 g's)(2) = 0.123 Lbs

Note – no torsional deflection will occur, tube fixed at both ends. The above weights will be doubled to account for larger tube.

I = 6.47×10^{-4} in^4 (from previous section)

$$\delta_y = \frac{\left(\frac{\pi}{4}\right)(.123)(2)^3 + \frac{1}{2}(.115)(2^3)}{(15\times10^6)(6.47\times10^{-7})} = 1.27\times10^{-4}(3) = 3.8\times10^{-4}\text{in}$$

$$\delta_x = \frac{\frac{1}{2}(.123)(2)^3 + \left(\frac{3}{\text{[illegible]}}\,\text{[illegible]} - 2\right)(.115)(2^3)}{(15\times10^6)(6.47\times10^{-7})} = 8.446\times10^{-5}(3) = 2.534\times10^{-4}\text{in}$$

$$\delta_{total} = (\delta_x^2+\delta_y^2)^{1/2} = (3.8\times10^{-4\,2} + 2.84\times10^{-4\,2})^{1/2} = 0.0004567\,\text{in}$$

$$\frac{\delta_{normal\,operation}}{\delta_{seismic}} = \frac{0.19\,\text{in}}{0.0004567} = \text{416 times more.}$$

The seismic stress (deflection) will have little if any adverse affects on the tube. The tube is adequate.

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Fatigue Analysis

For the purpose of fatigue evaluation only the point of maximum stress will be addressed. This will be the brass stem of the 1009 at a $\sigma_p = 3122 psi$

Engine Operational Vibration

For ease of analysis the engine operating stress (due to vibration) will be conservatively estimated by the relationship:

$$\underset{\text{eng vibe}}{\sigma_p} = \underset{\text{seis+pressure}}{\sigma_p} \left[\frac{\text{max "g" level vibe}}{\text{min "g" level seis}}\right]$$

The engine vibration levels are obtained from CQD-012628 rev.00 dated 2/22/84 "Response of An EMD 645 E-4 Engine".

$$a_x = 0.95g's \ , \ a_y = 0.85g's \ , \ a_z = 1.00g's$$

The minimum seismic "g" level was 1.4g's in the Horizontal. Using the appropriate levels from above:

$$\underset{\text{Eng Vibe}}{\sigma_p} = \underset{\text{Seis}}{\sigma_p} \left[\frac{1.0g's}{1.4g's}\right] = 3122(1/1.4) = 2230 \text{ psi}$$

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Since the socket is threaded a fatigue strength reduction factor of 4 (as required by ASME B+PV codes for bolts section III, Division 1, subsection NB; paragraph NB-3230.)

Loading Combination	σ_p	n	N
seismic + Pressure + vibe	(3122 + 2230) 4 21,408 psi	60 cycles	18,342 cycles
Pressure + Vibe	21,408 psi	(5500 - 60) 5440 cycles	18,342 cycles
Vibration	(2230) 4 8920 psi	2.8×10^9 cycles	∞

From ASME B+PV code, section III, Appendix I table I-9.1, fig I-9.3
$S_y = 18$ ksi

$$N_1 = 10000\left(\frac{20000}{10000}\right)^{\left[\log\left(\frac{24.5}{21.408}\right)/\log\left(\frac{24.5}{21.0}\right)\right]} \simeq 18,342 \text{ cycles}$$

$N_2 = N_1 = 18,342$ cycles

$N_3 = \infty$, below the endurance limit of 12,000 psi

$$\text{Usage factor} = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} = \frac{60}{18,342} + \frac{5440}{18,342} + \frac{2.8\times10^9}{\infty} = 0.30 < 1.0$$

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MEMORANDUM OF
TELEPHONE CONVERSATION

Date: 3/15/84

Time: 11:30 am

Person Called: David Strayhorn (Name) of Marsh Instrument Co. (Company)

Person Calling: David Wright (Name) of Sargent & Lundy (Company)

Project: LaSalle/Clinton Project No. 6549-30/4536-32

Subject Discussed: Design Dimensions and Criteria on the Model J-4858 Gauge

Summary of Discussion, Decisions and Commitments:

Dave (Marsh) stated that the J-4858 gauge is designed and built to conform to ANSI B.40.1 standards. He also stated that the volume of the bourdon tube for the 4858 can vary by as much as 25%. The following design dimensions were provided by Dave at my request. (See attachment)

DW/cg

cc: M. Mohaghegh - 30
J. Sinnappan/P. Hlepas - 30
AEM/APD/DHW - 30
CQD File - 30

D. Strayhorn - Marsh Instrument Co.

Dav
Signature

File: CQD-012825

File no. CQD-012825
date 3/15/84
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Manufacturer/Model Number - Marsh/J-4858

Mounting - stem mounted from the bottom

Total Weight (Dry) - 5-3/4 ounces

Stem Size - 1/4 in. NPT

Stem Material - Copper Alloy

Stem Inlet Hole Diameter - 1/8 in. min., 3/16 in. max.

Bourdon Tube Material - Copper Alloy

Bourdon Tube Thickness - 1/16 in. to 1/8 in.

Bourdon Tube Arc. Length - 1-11/16 in.

Bourdon Tube X-Sect - approximate circle, 7/16 in. O.D.

Dial Size - 2-1/2 in.

Maximum Rated Pressure - 300 psi

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MEMORANDUM OF TELEPHONE CONVERSATION

Date: March 27, 1984

Time: 10:30 am

Person Called: Milt Dixon (Name) of Marshalltown (Company)

Person Calling: David Wright (Name) of S&L (Company)

Project: LaSalle/Clinton Project No. 6548-30/ 4536-32

Subject Discussed: Marshalltown Pressure Gauge, Figure 23 and Figure 24-F with Dial Ranges 0 - 100psi, 0 - 300psi, 0 - 160psi and 0 - 240psi.

Summary of Discussion, Decisions and Commitments:

The following information was provided by Mr. Dixon:

Figure 23	Figure 24-F
Approximate Wt. 1/2 lbs.	2 lbs.
Stem Size 1/4 in.	1/4 in.
Stem hole I.D. 0.15 in.	0.16 in.
Mounting Screw N/A	0.23 in. (hole size)
Stem Material Extruded Brass	Forged Brass
Bourdon Tube Material Phosphor Bronze	Phosphor Bronze

Mr. Dixon stated that any information regarding the size, x-section or thickness of the Bourdon tube was considered proprietary information and could not be released.

DW/cg

cc M. Dixon - Marshalltown Mfg.
S. A. Ahmad - 30
AEM/APD/DW - 30

Signature

File: CQD-013106 - 30

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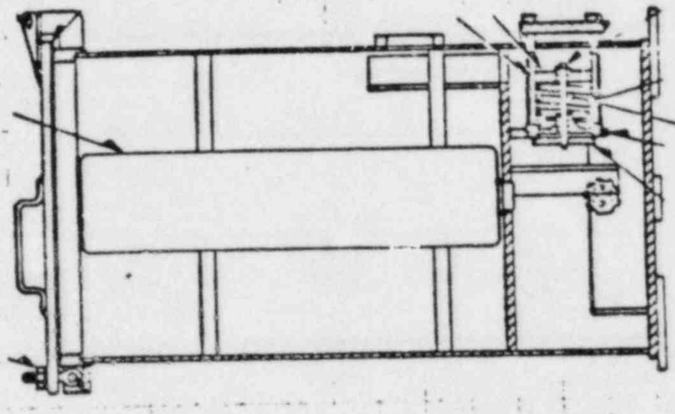
Client
Project
Proj. No. Equip. No.

Prepared by David Wright Date 5/17/84
Reviewed by Date
Approved by Date

Primary Lube Oil filter Relief Valve

Log No. 0803161707



The primary lube oil filter relief valves found on the shoreham (LILco) Diesels are identical to those found on the La Salle (CECo) Diesels. Since these valves were shown to be adequate for the LaSalle diesels, they are also adequate for the Shoreham engines. As shown in the LaSalle report (CQD-010229 Rev.00) the releif valve is totally encased in the filter housing. Even if the valve fails the filter would still maintain its pressure boundry and would allow the main flow of oil to run through the shell.

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Calcs. For 1 of 11

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VALVES NOTED ON
EMD DWG NO 8320705

VALVE LOG
NO

0303161707
0601161707

ADDITIONAL INFORMATION

PASSIVE
WORKING TEMP = 235°F
WORKING PRESS = 40 PSI
GRAY IRON T - 30,000 PSI MIN
[illegible]

PRIMARY FILTER RELIEF VALVE

THIS RELIEFVALVE IS INTERNAL TO THE LUBE OIL FILTER WITH A COVER BOLTED TO THE LUBE OIL FILTER. THEREFORE THERE IS NO PIPING END LOADS ASSOCIATED WITH THEM, ANY SEISMIC LOADS WOULD BE SEEN FIRST BY THE FILTER. IF THE FILTER WAS NOT RIGIDLY ATTACHED THE VALVE MIGHT SEEN SOME RESPONSE (SEISMIC) HOWEVER. THE LUBE OIL FILTER IS CONSIDERED A RIGID ATTACHMENT. THE PRESSURE BOUNDRY IS THE COVER THAT IS BOLTED TO THE LUBE OIL FILTER, THE VALVE BY ITSELF, THERE IS NOTHING TO CHECK FOR IN THIS SECTION.

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Calcs. For LOG NO 0803160

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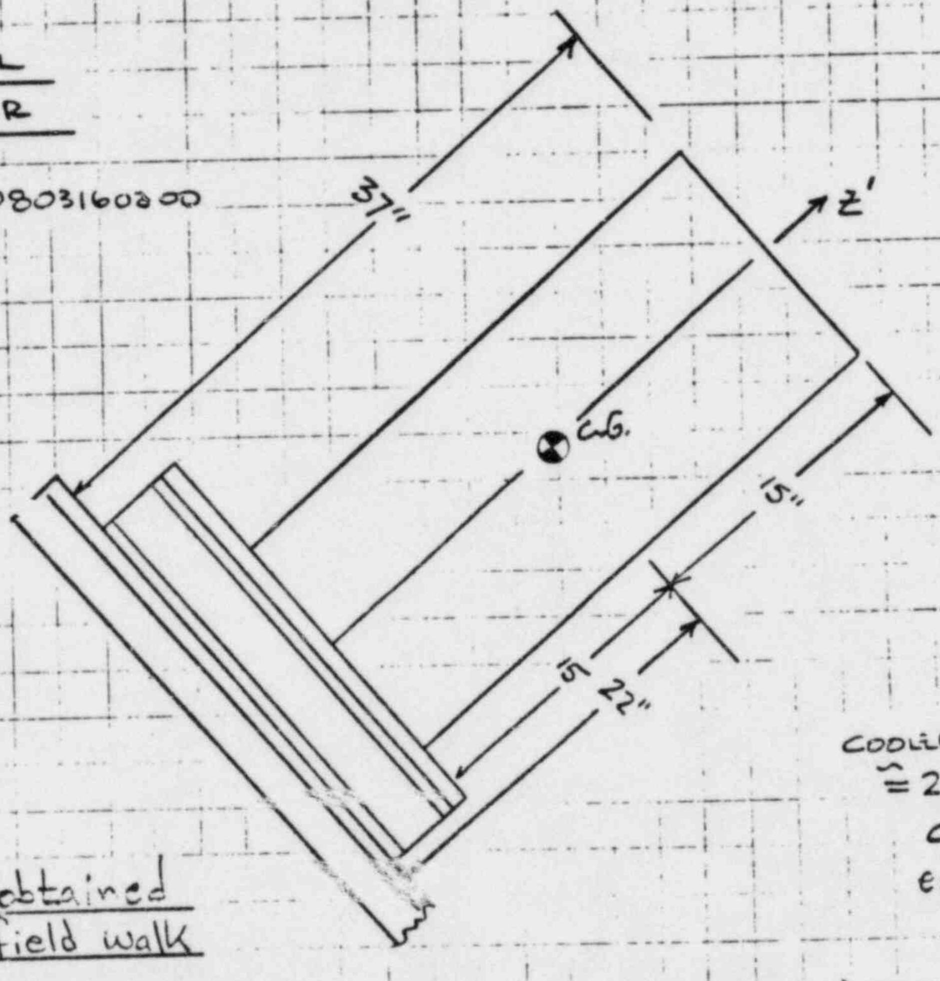
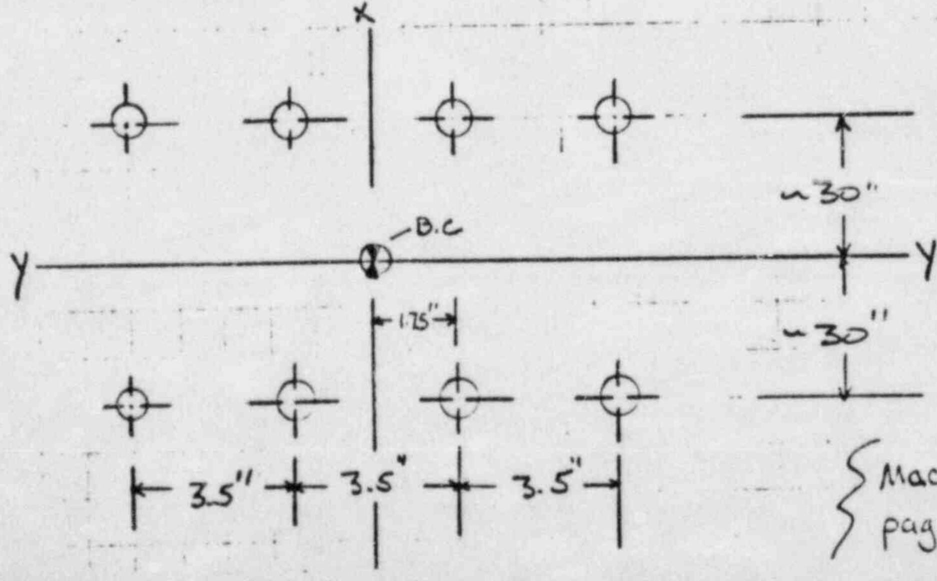
LUBE OIL COOLER

LOG NO 0803160800



COOLER WT
≃ 2000 LBS
CONSERVATIVELY
estimated.

sketch obtained during field walk-down



(8) BOLTS
1/2 DIA.
SAE 5
STRESS
AREA
$= .1416 \text{ IN}^2$

{Machinery's Handbook, page 1282, UNC}

Form GO-3.08.1 Rev. 2

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Calcs. For FIELD SKETCH

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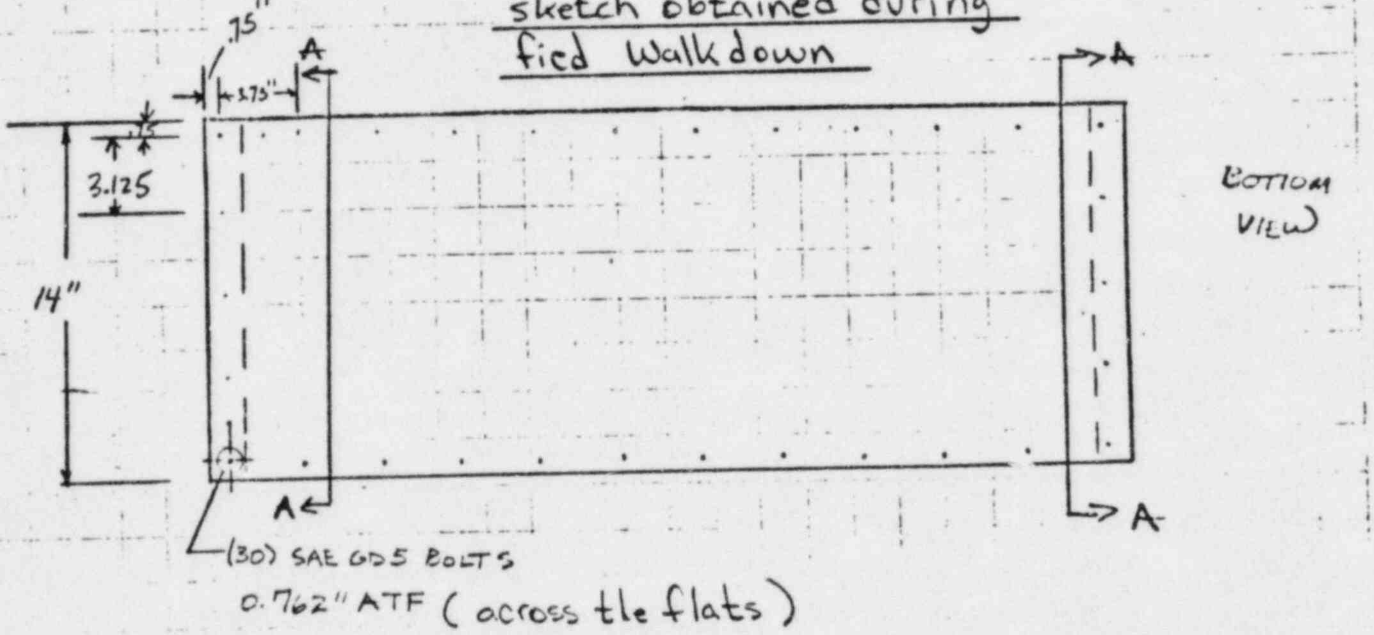
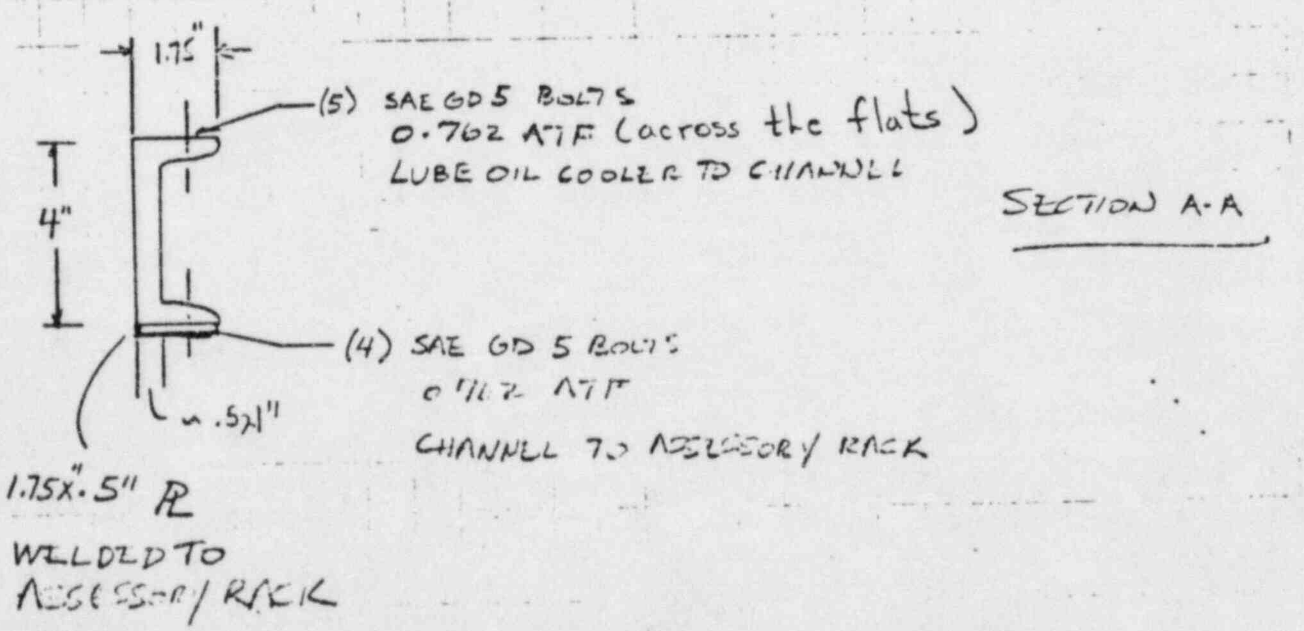
LUBE OIL COOLER

sketch obtained during fied walkdown





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Loading Due to Normal Engine Operation

BECAUSE THE LUBE OIL COOLER IS MOUNTED ON THE ACCESSORY RACK AND KNOWING THAT THE RACK IS RIGID (PER EMD FILE NO.*EMD-008028 REV 00 DATED 4/19/77) THE ACCELERATION LEVELS MEASURED FROM THE ACCESSORY RACK FRAME WILL BE USED TO QUALIFY THE LUBE OIL COOLER

$x = 0.126g \quad y = 0.142g \quad z = 0.111g$ (Location 15)

[Taken from S+L-CQD-012628, Rev.00 dated 2/22/84 in titled Response of An EMD645 Engine And Selected Accessories To Impulsive and Steady State Loads]

$Acc\ x = x' = 0.126$

$Acc\ z' = .707z + .707y = 0.1789g$

$Acc\ y' = .707y + .707z$ (WORST CASE) $= 0.1789g$

LOADING

$F_{x'} = 2000\ (0.126) = 252$ LBS
$F_{y'} = 2000\ (0.1789) = 357.8$ LBS
$F_{z'} = 2000\ (0.1789) = 357.8$ LBS
$M_{x'} = F_{y'}\bar{z} = 357.8\ (22) = 7871.6$ IN-LBS
$M_{y'} = F_{x'}\bar{z} = 252\ (22) = 5544$ IN-LBS
$M_z = 0.0$

* Note- EMD-008028 (Sargent+Lundy) is a finite element analysis of the LaSalle EMD Diesel generator

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INERTIA PROPERTIES

$$I_{xx} = 4(.1416)(1.75)^2 + 4(.1416)(5.25)^2 = 17.346\ IN^4$$

$$I_{YY} = 4(.1416)(30)^2 = 509.76\ -IN^4$$

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BOLT STRESS Due to Normal Operating Vibration

$$\sigma_t = \frac{F_Z}{8A_s} + \frac{M_x c_y}{I_x} + \frac{M_y C_x}{I_y}$$

$$= \frac{357.8}{8(.1416)} + \frac{7871.6\,(6.125)}{17.346} + \frac{5544\,(30)}{509.76}$$

$$\sigma_t = 316 + 2780 + 327 = 3423 \text{PSI}$$

$$\tau_v = ({F_x'}^2 + {F_y'}^2)^{1/2} / 8A_s$$

$$= (252^2 + 357.8^2)^{1/2} / 8(.1416) = 387 \text{PSI}$$

$$\sigma_p = \frac{3423}{2} + \sqrt{\left(\frac{3423}{2}\right)^2 + (387)^2} = 3467 \text{PSI}$$

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SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

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Loading, Seismic + Weight (stress)

From CQD-010330, Rev.00 on the subject of Qualification of an EMD-645-E4 Diesel Engine ;

The Lube oil cooler from the above mentioned report is identical (except size) to the unit found on the Shoreham MP-45's. The weight of the shoreham cooler has been estimated at 2000 Lbs. The previous cooler had a weight of 1190 Lbs. The actual stress in the Shoreham bolts (mounting) will be estimated by increasing the previously obtained stress level by a % that is proportional to the larger Shoreham weight. The factor that the stress from the old analysis will be increased by is:

$$\text{factor} = \text{Weight(New)} / \text{Weight(old)}$$

$$= 2000\ \text{Lbs} / 1190\ \text{Lbs} = 1.68$$

$\text{Stress}_{\text{New}} = \text{Stress}_{\text{old}} (1.68)$, using the previouse values from the above referenced report:

$$\sigma_{t_{\text{New}}} = \sigma_{t_{\text{old}}} (1.68) = 2761\ \text{psi} (1.68) = 4638\ \text{psi}$$

$$\tau_{v_{\text{New}}} = \tau_{v_{\text{old}}} (1.68) = 2782\ \text{psi} (1.68) = 4674\ \text{psi}$$

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Combining the operating stress with the seisimic + weight stress we have:

$\sigma_{t\,\text{total}} = 3423\text{psi} + 4638\text{psi} = 8061\text{ psi}$ (tension)

$\tau_{v\,\text{total}} = 387\text{psi} + 4674\text{ psi} = 5061\text{ psi}$ (shear)

Allowable Stress

From ASME B+PV code, 1977 edition with summer 79 addendum, section III, Article 2460 of Appendix XVII;

For combined shear and tensile loading the following condition must be met:

$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \leq 1.0$, f_t = actual tensile stress, f_v = act shear,

F_t = Allowable tensile stress = 0.3Su (service level A+B)

F_v = Allowable shear stress = 0.124 (").

$F_t = (0.3)(\overset{*}{1}20{,}000\text{psi}) = 36{,}000\text{psi}$, $F_v = (0.124)(\overset{*}{1}20{,}000) = 14{,}880\text{psi}$

$\frac{8061^2}{36{,}000^2} + \frac{5061^2}{14{,}880^2} = 0.166 \leq 1.00 \Rightarrow$ bolts are adequate

*note - from ASME B+PV. Section III table I-7.3 of Appendix I SA-449

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Client	Prepared by David Wright Date 5/17/84
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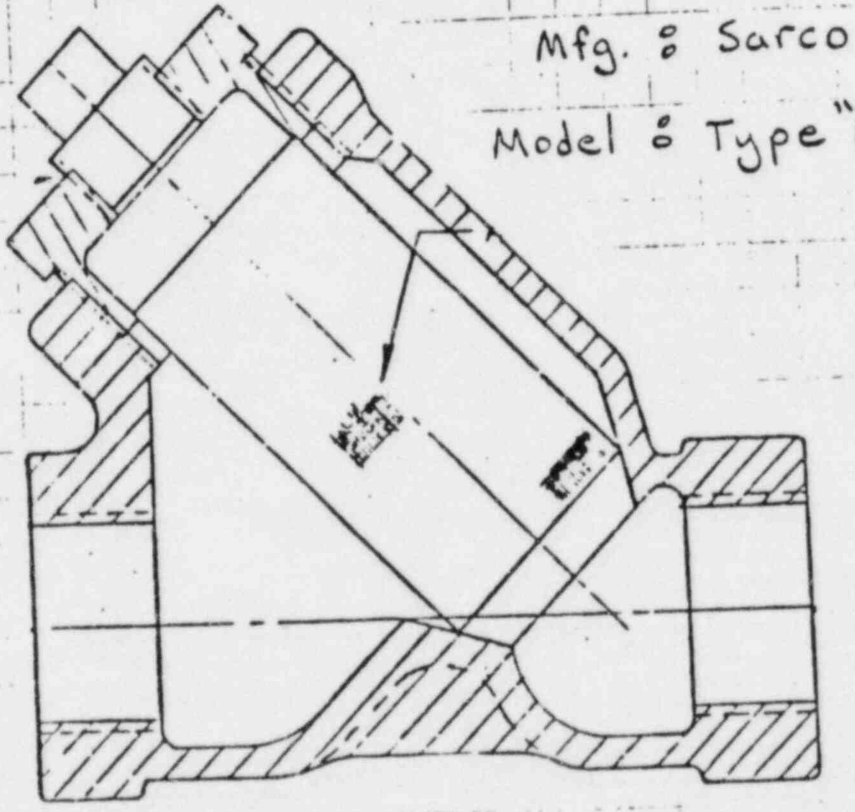
Lube Oil "Y" Strainer Assy.

Log No. 0803161800

Mfg. : Sarco

Model : Type "AT"



The Lube oil "Y" strainer found on the Shoreham engines are identicle to those found on the LaSalle (ceco) engines. Since it was shown that the LaSalle strainers were adequate, the Shoreham strainers are also adequate for use on their E.D.G.S.

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ANALYSIS OF
SARCO 'Y' STRAINER -TYPE IT

LOG NO. SIZE N.P.T

0303161800
0201161800 N/A 1 IN.
0702161800
0002116180D

ADDITIONAL INFORMATION

~~PASSIVE~~ ~~(SEE SUBJECT STRAINER LISTING)~~

WORKING TEMP = 235°F
WORKING PRESS = 50 PSI } SEE ATTACHED WORK ORDER 75744

VALVE BODY MATERIAL = ASTM A126 CLASS B (PHONE MEMO) Attached.

RATED MAX PRESS = 200 PSIG @ 150°F
EMD DWG B280951 } (SEE SARCO CATALOGUE). Attached at end.
WT = 4.3 LBS

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MATERIAL PROPERTIES

ASTM A126 GD B (PHONE MEMO) attached

MIN. TENSILE STRENGTH = 31.0 KSI ASME B+PV Code, Section III Div. 1, subsection ND class 3 components, 1983 Summer Addenda.

ID OF STRAINER AT CROTCH REGION = 1 1/2 IN (PHONE MEMO)
WALL THICKNESS = 9/32 IN = t_m (")

$\therefore$ OD = $2t_m + ID$ = $2(9/32'') + 1\frac{1}{2}''$ = 2.0625 IN.

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BODY STRUCTURAL INTEGRITY

$S_{y_{st}}$ = STRAINER BODY YIELD STRENGTH (MINIMUM) FOR ASTM A126 GD B MATERIAL —
= 31.0 KSI (ASME B+PV code section III Div. 1 subsection ND, class 3 components, 83 summer addendum.)

S_{y_p} = PIPE YIELD STRENGTH (MIN) FOR ASTM A106 GD B (ASSUMED) MATERIAL.
= 35.0 KSI (ASME B+PV Code section III Div. 1, Appendix I, 1983 summer Addendum.)

SHOW THAT $\frac{S_{y_{st}}}{S_{y_p}} \times \frac{Z_{st}}{Z_p} > 1.1$

$$Z_{st} = \frac{\pi}{32}\left[\frac{OD^4 - ID^4}{OD}\right]$$

$$= \frac{\pi}{32}\frac{\left[(2.0625\,IN)^4 - (1.5\,IN)^4\right]}{(2.0625\,IN)} = 0.6204\ IN^3$$

$Z_p = 0.1606\ IN^4$ (FOR 1" SCH 80 - CAMERON CAT. P.171)

$$\frac{S_{st}}{S_p} \times \frac{Z_v}{Z_p} = \frac{31}{35} \times \frac{0.6204}{.1606} = 3.42 > 1.1$$

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SHOW THAT $\frac{S_{Y_{SG}}}{S_{Y_P}} \cdot \times \frac{A_V}{A_P} > 1.1$

$$A_{st} = \frac{\pi}{4}\left[OD^2 - ID^2\right]$$

$$= \frac{\pi}{4}\left[(2.0625 IN^2 - (1.5 IN^2)\right] = 1.574$$

$A_P = 0.639 \ IN^2$ (1" SCH 80 - CAMERON CAT. P171)

$$\frac{S_{Y_{SG}}}{S_{Y_P}} \times \frac{A_V}{A_P} = \frac{31}{35} \times \frac{1.574}{.639} = 2.18 > 1.1$$

THESE RESULTS SHOW THE ADEQUACY OF THE BODY TO WITHSTAND PIPE END LOADS IN ACCORDANCE WITH PROCEDURES FOR VALVES PER ASME B&PV CODE SECTION III SUBSECTION ND-3521 (a).

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CAP THREAD STRESS

$n = 11.5$

$Le = .42$

$K_n MAX = 1.555$

$E_s MIN = 1.5756$

} AMERICAN NATIONAL STANDARD STRAIGHT PIPE THREADS FOR MECH JOINTS MACHINERY HANDBOOK 21ST ED P. 1402

TOTAL WT OF STRAINER = 4.3 LBS
ASSUME CAP WT = 1 LB
(CONSERVATIVE)

$$A_s = 3.1416(n)(Le)(K_n MAX)\left[\frac{.5}{n} + 57735\,(E_s MIN - K_n MAX)\right]$$

$$= 3.1416(11.5)(.42)(1.555)\left[\frac{.5}{11.5} + .57735\,(1.5756 - 1.555)\right]$$

$$= 1.306\ IN^2$$

SHEAR STRESS

$$\tau_s = \frac{F + P\frac{\pi}{4}D^2}{A_s}$$

$F = 6.23g\ (1\ lb) = 6.23\ LB$

$P = 300\ PSI$ MAX RATING

$D = 1.660\ IN$ (Nominal Pipe size of cap)

$$\tau_s = \frac{6.23\ LBS + 300\frac{LB}{IN^2}\ \frac{\pi}{4}\ (1.660\ IN)^2}{1.306\ IN^2} = 502\ PSI$$

NEGLIBLE

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TENSILE STRESS

$$A_t = 0.7854 \left(D - \frac{.9743}{n} \right)^2$$

$$= 0.7854 \left(1.660 - \frac{.9743}{11.5} \right)^2$$

$$= 1.45 \text{ IN}^2$$

$$\sigma_t = \frac{6.73 \text{ LBS} + 300 \frac{\text{LB}}{\text{IN}^2} \frac{\pi}{4} (1.660 \text{ IN})^2}{1.95 \text{ IN}^2}$$

$$= 336 \text{ PSI}$$

NEGLIBLE

THESE RESULTS SHOW THAT THE CAP (PRESSURE BOUNDARY) THREAD STRESSES DUE TO SEISMIC & PRESSURE LOADINGS ARE NEGLIBLE AND ARE THEREFORE ACCEPTABLE

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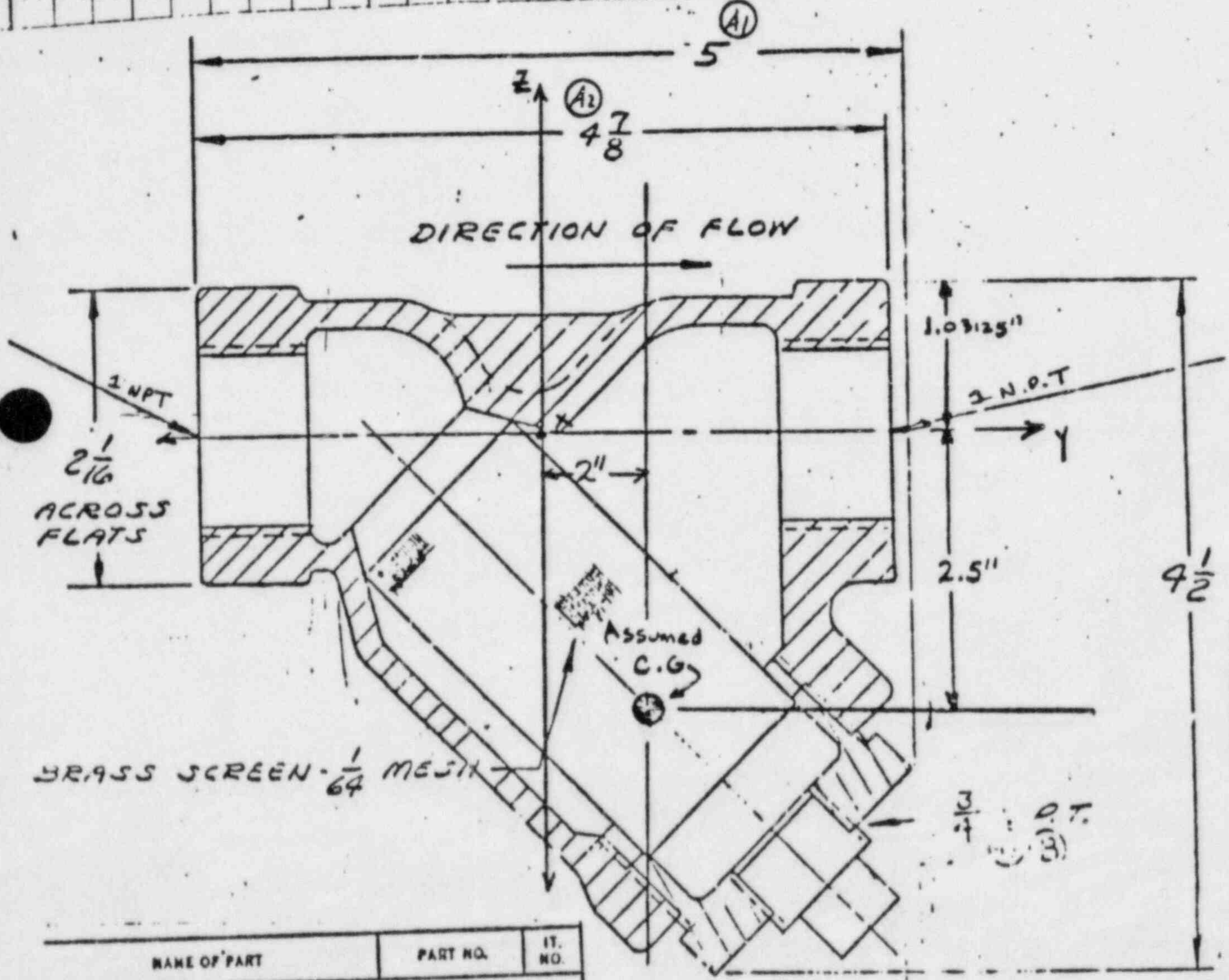
FIGURE A



NAME OF PART	PART NO.	IT. NO.

ELECTRO-MOTIVE DIVISION
GENERAL MOTORS CORPORATION
LA GRANGE, ILLINOIS, U.S.A.

STRAINER

8280951

SARCO COMPANY INC.
Ⓐ
TYPE "AT" SEMI-STEEL 250#
SCREWED STRAINER CAP
WITH PIPE PLUG

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ACTUAL STRESS CALC

NOZZLE LOADING

* LOADING FOR ASME CLASS 3, EQUIPMENT WEIGHT ≤ 1 KIP (SERVICE LEVEL C)

FOR 1 INCH PIPE

$$F = \sqrt{F_x^2 + F_y^2 + F_z^2} = 0.19 \text{ KIPS} = 190 \text{ LBS}$$

$$M = \sqrt{M_x^2 + M_y^2 + M_z^2} = 0.10 \text{ KIP-FT} = 1200 \text{ IN-LBS}$$

ASSUMING ALL FORCE COMPONENTS EQUAL TO ONE ANOTHER
& " " MOMENT " " " " "

THEREFORE:

$$F_x = F_y = F_z = 190/\sqrt{3} = 110 \text{ LBS}$$

$$M_x = M_y = M_z = 1200/\sqrt{3} = 693 \text{ IN-LBS}$$

$$F_{AXIAL} = 110 \text{ LBS}$$

$$F_{SHEAR} = 110 \text{ LBS} \cdot \sqrt{2} = 156 \text{ LBS}$$

$$M_{TORSIONAL} = 693 \text{ IN-LBS}$$

$$M_{BENDING} = 693. \text{ IN-LBS } \sqrt{2} = 980 \text{ IN-LBS}$$

*Note – From S+L file CQD-005513, Rev.00 dated 1/21/83 in titled Nozzle Loads For Equipment. These loads are based on the section properties and allowables of the pipe.

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SEISMIC LOADING

RESPONSE SPECTRA USED IS THE LASALLE/CLINTON DIESEL GEN. ENVELOPED SPECTRA FOUND IN CQD-011392 DATED 12-19-83 PEAK ACCELERATION LEVELS ARE:

ENV [VERT] = 4.75 g's , ENV [HORZ] = 1.85 g's

A STATIC COEFFICIENT ANALYSIS WILL BE USED PER IEEE-344-1975. THEREFORE THE SEISMIC COEFFICIENTS ARE AS FOLLOWS:

ENV [OBE & SSE HORIZONTAL 2% & 7%] = 1.85 (1.5)* = 2.8 g's

ENV [OBE & SSE VERTICAL 2% & 3%] = 4.75 (1.5)* = 7.1 g's

* RECOMMENDED STATIC COEFFICIENT FROM IEEE-344-1975

$F_x = F_y = (4.3\text{ lbs})(2.8g) = 12\text{ lbs}$

$F_z = (4.3\text{ lbs})(7.1g) = 31\text{ lbs}$

$M_x = F_y(z) + F_z(y) = 12\text{ lbs}(2.5\text{ in.}) + 31\text{ lbs}(2\text{ in}) = 92\text{ in-lbs}$

$M_y = F_x(z) + F_z\cancelto{0}{(x)} = 12\text{ lbs}(2.5\text{ in}) = 30\text{ in lbs}$

$M_z = F_x(y) + F_y\cancelto{0}{(x)} = 12\text{ lbs}(2\text{ in}) = 24\text{ in lbs}$

Note $x = 0.0$, $y = 2''$, $z = 2.5''$ SEE FIGURE A

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TENSILE STRESS

$$\sigma_t = \frac{F_{\text{AXIAL}}}{\text{Area}} + \frac{M_{\text{BENDING}}}{Z} + \frac{PD_o}{4t_m}$$

$$F_{\text{AXIAL}} = [\underbrace{110}_{\text{NOZZLE}} + \underbrace{31}_{\text{SEISMIC}} + \underbrace{4.3}_{\text{DEAD WT.}}]\text{ lbs} = 145.3\text{ lbs}$$

$$M_{\text{BENDING}} = [980 + (92^2 + 30^2)^{\frac{1}{2}}]\text{ in lbs} = 1077\text{ in lbs}$$

$$\underbrace{P = 300\ \frac{\text{lb}}{\text{in}^2}}_{\text{SARCO CATALOGUE}},\quad D_o = 2.0625\text{ in}\quad \underbrace{t_m = 9/32\text{ in}}_{\text{PHONE MEMO}}$$

$$\sigma_t = \frac{145.3\text{ lbs}}{1.574\text{ in}^2} + \frac{1077\text{ in lbs}}{0.6204\text{ in}^3} + \frac{300\ \frac{\text{LB}}{\text{IN}^2}\ 2.0625\text{ in}}{4\ (9/32\text{ in})}$$

$$\sigma_t = [92.3 + 1736 + 550]\text{ PSI} = 2378\text{ PSI}$$

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SHEAR STRESS

$$\tau_v = \frac{M_t r_o}{I_t} + \frac{F_{shear}}{A}$$

$M_t = [693 + 24]$ in-lbs $= 717$ in lbs

$F_{shear} = [156 + (12^2 + 12^2)^{1/2}]$ lbs $= 173$ lbs

$$I_t = 2\left[\frac{\pi\,(2.0625^4 - 1.5^4)}{64}\right] = 1.28 \text{ in}^4$$

$$\tau_v = \frac{717 \text{ inlbs } (2.0625''/2)}{1.28 \text{ in}^4} + \frac{173 \text{ lbs}}{1.574 \text{ in}^2}$$

$$= [578 + 110] \text{ PSI} = 688 \text{ PSI}$$

PRINCIPAL STRESS

$$\sigma_p = \frac{2378}{2} + \sqrt{\left(\frac{2378}{2}\right)^2 + (688)^2} = 2563 \text{ PSI}$$

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FATIGUE ANALYSIS

ENGINE VIBRATION - THE ENGINE VIBRATION LEVEL AT THE STRAINER IS OBTAINED FROM LOCATION 8 ~~[illegible]~~ see below ① (LUBRICATOR LOCATION IS ON SAME LINE AS STRAINER) THE ACCELERATION LEVELS MEASURED FROM THE BASE OF THE RESERVOIR OF THE LUBRICATOR WERE : X = 0.732, Y = 0.347, Z = 0.847

ASSUMPTIONS - BY INSPECTION OF THE FORCES AND MOMENTS USED TO DETERMINE THE SEISMIC (incl. Nozzle) STRESS LEVELS AT THE BODY CROTCH OF THE STRAINER ARE GREATER THAN THE LEVELS THAT WILL BE SEEN DURING ENGINE OPERATION. THEREFORE WE WILL ASSUME THAT σ_P DUE TO NORMAL OPERATION IS LESS THAN σ_P DUE TO SEISMIC LOADING

IT'S CONSERVATIVE TO ASSUME

$$\sigma_{P_{\text{Norm. Engine Vibe}}} = \sigma_{P_{\text{Seismic}}}$$

NOTE: THE STATEMENTS ABOVE WILL BE USED THROUGHOUT THIS ANALYSIS FOR ALL STRAINERS

① S+L file No. CQD-012628 Rev. 00 dated 2/22/84 in titled Response of An EMD 645 Engine And Selected Accessories To Impulsive and Steady State Loads.

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ENGINEERS
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LOADING COMBO.	σ_p	n (required)	N
SEISMIC + ENGINE VIBRATION	(2121+2121) = 4242 psi	60 cycles	SEE NOTE BELOW
ENGINE VIBE	2121 psi	* 2.97×10^9 cycles	∞

section III

THE ASME B&PV CODE, APPENDIX DOES NOT HAVE AN S-N CURVE FOR CAST IRON, HOWEVER, PER REF. 10 PAGE 5-9 SHOWS THE TYPICAL APPROXIMATE FATIGUE ENDURANCE LIMITS FOR CAST IRON

UTS = 20-50 KSI

FATIGUE ENDURANCE LIMIT = 6-18 KSI

NOTE:

As can be seen from the level, Note THAT THE SEISMIC + ENGINE VIBRATION CYCLES ARE VERY LOW SO THAT FAILURE DUE TO FATIGUE IS NOT A CONCERN

*REQ'D ENGINE VIBE CYCLES = 11,000 hours @ 75 HZ

$N = (11{,}000 \text{ hours})(3600 \text{ seconds/hr})(75 \text{ cycles/sec})$

$N \approx 2.97 \times 10^9$ cycles

(also from "Response of An EMD 645 Engine" noted on last page)

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It should be noted that iF THE MAX. PRINCIPAL STRESS FALLS BELOW 7250PSI (50 MN/m²) ENDURANCE LIMIT THAN THE STRAINOR (MADE OF CAST IRON) IS QUALIFIED

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ITEM	PART N°	LOCATION	WORKING TEMP. (°F)	WORKING PRESS. (PSI)	MATERIAL	* ITEMS THAT APPLY TO EMD O# 7B3014 & 16 W.O. 75744 MODEL - SIGE4
OIL FILTER ASM	8379120	ACCESS. RACK (LUBE OIL SYS.)	235	40	STL HOUSING - SAE 1020-1025 HOT ROLLED SHEET STL	*
CIRC. PUMP (ELEC.)	8262483	BASE MTD (LUBE OIL SYS.)	235	50	CAST IRON PUMP T-35000 PSI (MIN.) C.I. MOTOR HSG. T-20000 PSI (MIN.)	9099638 APPLIES
"Y" STRAINER	3280951	ACCESS. RACK (LUBE OIL SYS.)	235	50	SEMI-STL. T-35000 PSI (MIN.)	*
CHECK VALVE	8366533	ENGINE ATTACH. (LUBE OIL SYS)	210	70	STEEL - B1113	*
TURBO FILTER	8366743	ENGINE ATTACH. (LUBE OIL SYS.)	210	70	GREY IRON HEAD T-30000 PSI (MIN.) STL SHELL - SAE 1010 HOT ROLLED WELD STL TUBE	*
CHECK VALVE	8366577	ENGINE ATTACH. (LUBE OIL SYS.)	210	70	STEEL - B1113	*
SOAK BACK PUMP	8455815	BASE MTD. (LUBE OIL SYS.)	235	125	CAST IRON PUMP T-35000 PSI (MIN) H. ROLLED STL. MOTOR HSG. SAE 1010	*
PRESS. & COOL. PUMP	8360554	ENGINE ATTACH. (LUBE OIL SYS.)	210	95	CAST IRON T-25000 PSI (MIN.)	
PRESS. SWITCH	8362040	ENGINE ATTACH (LUBE OIL SYS.)	210	0	CAST ALUM - ALLOY 356-T6	*
LOW LEVEL SWITCH	8445672	ENGINE ATTACH (LUBE OIL SYS.)	235	0	STL. CAP - SAE 1010 STAINLESS STL. 303 - STEM	*
PRESS. SWITCH	8358930	NOT BY EMD (LUBE OIL SYS.)	—	—	ALUM CASING - ALLOY 380 COLD ROLLED STL. - MTG. FLANGE	
TEMP. GAGE	9086874	ON SCAV. PUMP (LUBE OIL SYS.)	235	35	STAINLESS STL - AISI 304	*
PRESS. SWITCH	8475932	ENGINE ATTACH. (LUBE OIL SYS.)	210	0	PHENOLIC	*

SARGENT & LUNDY

MEMORANDUM OF TELEPHONE CONVERSATIOI

Date: 3/4/84

Time: 8:45 am

Person Called: Mark Weber (Name) of Sargent & Lundy (Company)

Person Calling: Joe Radle (Name) of Sarco (215-797-5830) (Company)

Project: LaSalle/Clinton Project No. 6548-30/4536-32

Subject Discussed: Sarco Type IT Pipeline Strainer (1" NPT)

Summary of Discussion, Decisions and Commitments:

Mr. Radle provided me with following data pertaining to the subject strainer:

The threads on the cup are American National Standard 11-1/2 threads per inch with O.D. - 1.660 in. and minor dia. - 1.4877 in. The wall thickness at the crotch region: S 9/32 in. and the inner diameter at the same region: S 1-1/2 in.

MLW/eg

cc: D. Wright - 30
J. Radle - Sarco
AEM/APD/MLW - 30

Signature

File: COD-013400

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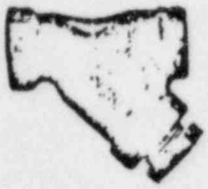
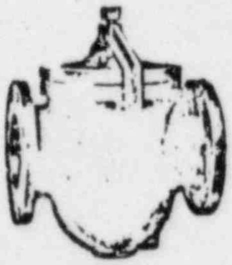
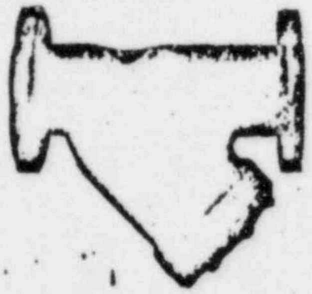
SARCO PIPELINE STRAINERS

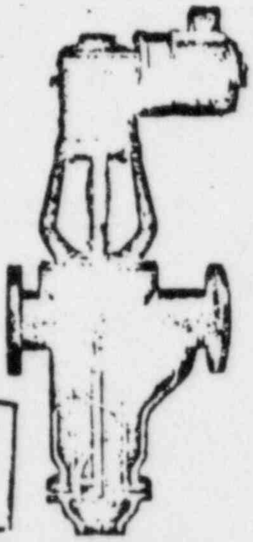
One of the White Consolidated Industries







Calc. No: CQD-
Rev: Date:
Proj. No:
Page Of



[illegible]	Body Material	Connections	Standard Sizes, Inch	Steam, Liquid, Gas or Oil, Non-Shock		Liquid, Gas or Oil Only at 150°F Max. Temp. Max. Press. Non-Shock	See Bulletin Page
				Max. Pr. PSIG	Max. Temp. °F	PSIG	
IT	Cast Iron	Screwed	¼ to 3	250	406	300	2
IF-125	Cast Iron	Flanged	2 to 12	125	353	200	3
			14 to 18	100	340	150	
AF-250	Cast Iron	Flanged	2 to 12	250	406	500	
			14 to 18	200	390	300	
BT	Bronze	Screwed	¼ to 3	250	406	400	4
		Sweat	½ to 3				
BF-150	Bronze	Flanged	2 to 8	150	406	225	
BF-300	Bronze	Flanged	2 to 8	300	450	500	
CT	Cast Steel	Screwed and Socket Weld	½ to 2	[illegible]	850	1480	5
CF-150	Cast Steel	Flanged	2 to 12	150	609	285	[illegible]
CF-300	Cast Steel	Flanged	2 to 12	300	838	740	
CF-600	Cast Steel	Flanged	2 to 12	600	839	1480	
DT	Ductile Iron	Screwed	½ to 1	400	450	600	[illegible]
[illegible]	316 Stn. Stl.	Screwed	[illegible] to 2	600	1165	1235 *	[illegible]
[illegible]	Cast Iron	Flanged-125	2 to 12	125	353	200	7
			14 to 20	100	340	150	
		Flanged-250	2 to 12	250	406	500	
			14 to 20	200	390	300	
	Cast Steel (Stn. Stl.)	Flanged-150	2 to 10	150	809 (600)	285 (235)*	
		Flanged 300	2 to 10	300	838 (1165)	740 (615)*	
Scraper	To Specification	Screwed	¾ to 3	To Specification			8-9
		Flanged	3 to 10				

For screens other than standard see table page 2.

*Cold rating at 100°F

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
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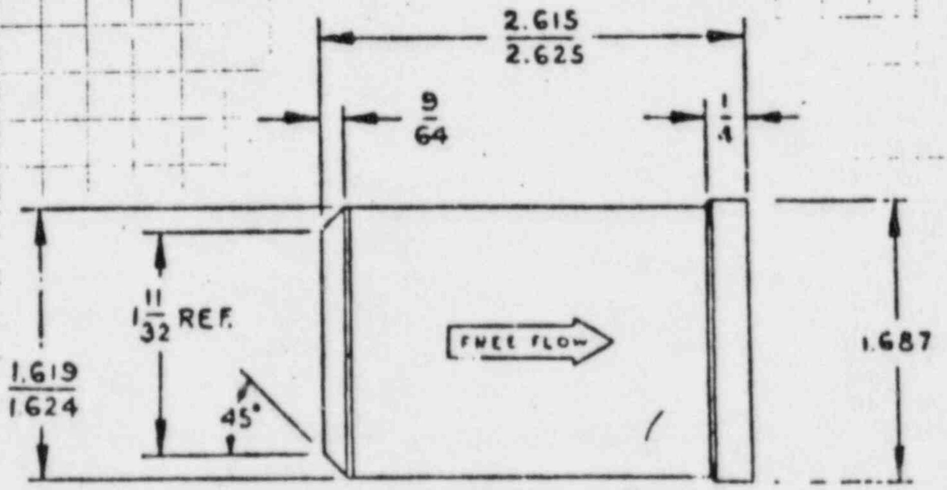
PROJ. NO. 6995-00
PAGE B.1.87

Client
Project
Proj. No. Equip. No.

Prepared by David Wright Date 5/17/84
Reviewed by Date
Approved by Date

Lube Oil Check Valve on Turbo Filter

Log No. 0803161704



The Turbocharger Filter check valve found on the Shoreham Diesels (LILco) is identical to those found on the La Salle Diesels (CECo). Since these valves were found to be adequate for the La Salle engines they are also adequate for use on the Shoreham engines. (original report (S+L) CQD-010330, Rev. 00). A copy of the previous work is attached.

Form GQ-3.08.1 Rev. 2

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CALC NO. CQD-014046
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Client		Prepared by	Date
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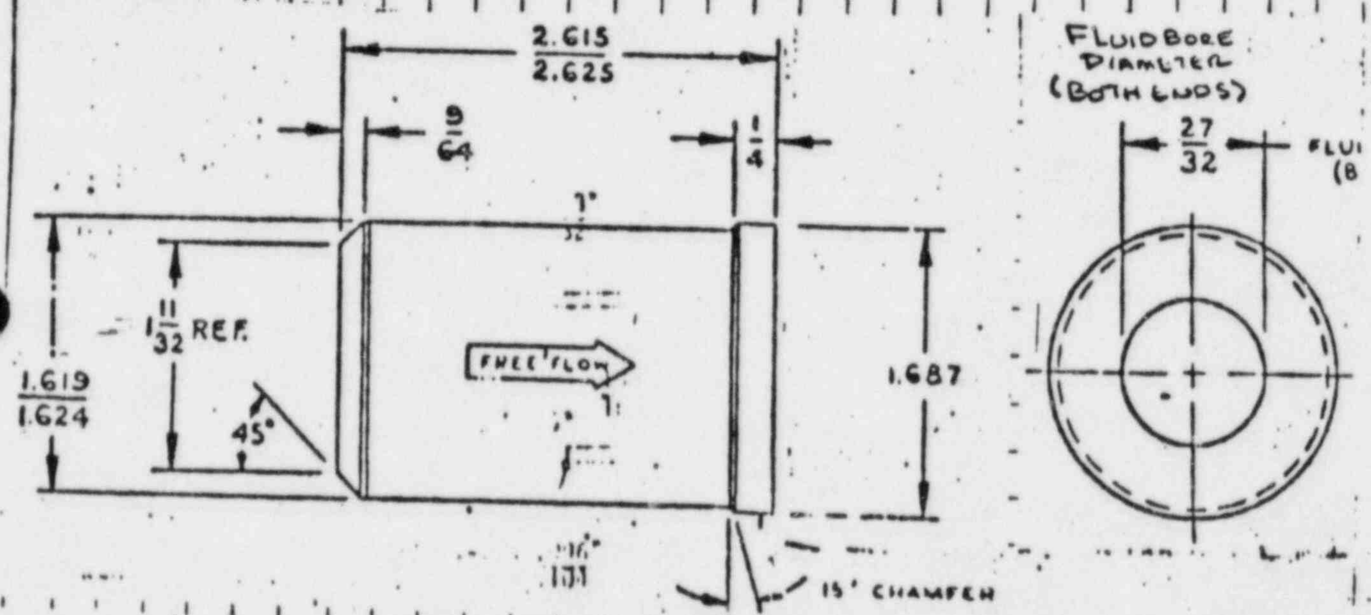
Proj. No.	Equip. No.	Approved by	Date

KEPNER CHECK VALVE FIGURE NO 1290



NOTE:

MATERIAL: BODY, CAP, RETAINER, AND POPPET TO BE B 1113 STEEL. SPRING IS AISI 302 STAINLESS STEEL.

FINISH: CARBON STEEL PARTS CADMIUM PLATED

SEALS: FLEXIBLE SEALS ARE VITON MATERIAL
[EXTERIOR SEAL IS UNIFORM SIZE #127
NOT INCLUDED WITH VALVE.]

CRACKING PRESSURE: 1 TO 2 PSIG

VALVE-CHECK
KEPNER PRODUCTS
PT. & DWG. NO.1290

Log No.'s
0303161704
0201161704
0202161704
0601161704

REV. A	RFC. NO. 36764 DATE 1-4-71	MATERIAL WAS POLYACRYLATE RUBBER	VKP
FIRST USED			
REV.	R.F.C. / DATE 1-13-65	[illegible]	

8366577

ELECTRO-MOTIVE DIV.
GENERAL MOTORS CORP.
LA GRANGE, ILLINOIS—U.S.A.

Form GQ-3.08.1 Rev. 2

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Rela

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.1.89

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

KEPNER FIG. 1290 CHECK VALVE

VALVE LOG
NO.

0303161704
0201161704
0202161704
0601161704

ADDITIONAL INFORMATION

ACTIVE

CONCLUSION

THESE CHECK VALVES ARE * LOCATED INSIDE THE SOAKBACK FILTER. THERE IS NO PIPING DIRECTLY ATTACHED TO THESE VALVES THEREFORE THERE IS NO PIPING END LOADS ASSOCIATED WITH THEM. THEREFORE, THERE IS NOTHING TO CHECK AS FAR AS ASME REQUIREMENTS ND-3521 (a) IS CONCERNED. SINCE THESE VALVES HAVE NO EXTENDED STRUCTURES THAT ARE ESSENTIAL TO MAINTAINING PRESSURE INTEGRITY THEN THE REQUIREMENTS OF ASME ND-3521(C) ARE SATISFIED.

* SEE THE ATTACHED SHEETS OF LAP-400-3 REV. 0 AUG 10, 1983 FOR THEIR LOCATION THIS DOCUMENTATION IS ATTACHED AT THE REAR OF THIS SECTION

Form GO-3.08.1 Rev. 2

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
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PAGE B.1.90

SARGENT & LUNDY
ENGINEERS
CHICAGO

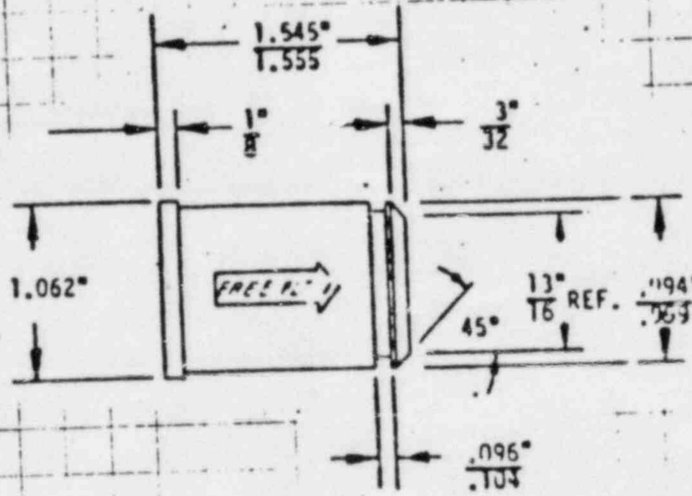
Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client	Prepared by Dave Wright Date 5/17/84
Project	Reviewed by Date
Proj. No. Equip. No.	Approved by Date

Check Valve on Soakback filter

Log No. 0803161703



The Soakback filter check valve found on the Shoreham Diesels (Lilco) is identicle to those on the LaSalle (CECo) Diesels. They were found to be adequate for use on the LaSalle engines and are therefore adequate for use on the Shoreham engines. (original report (S+L) CQD-010330, Rev. 00) A copy of the previous Analysis is attached.

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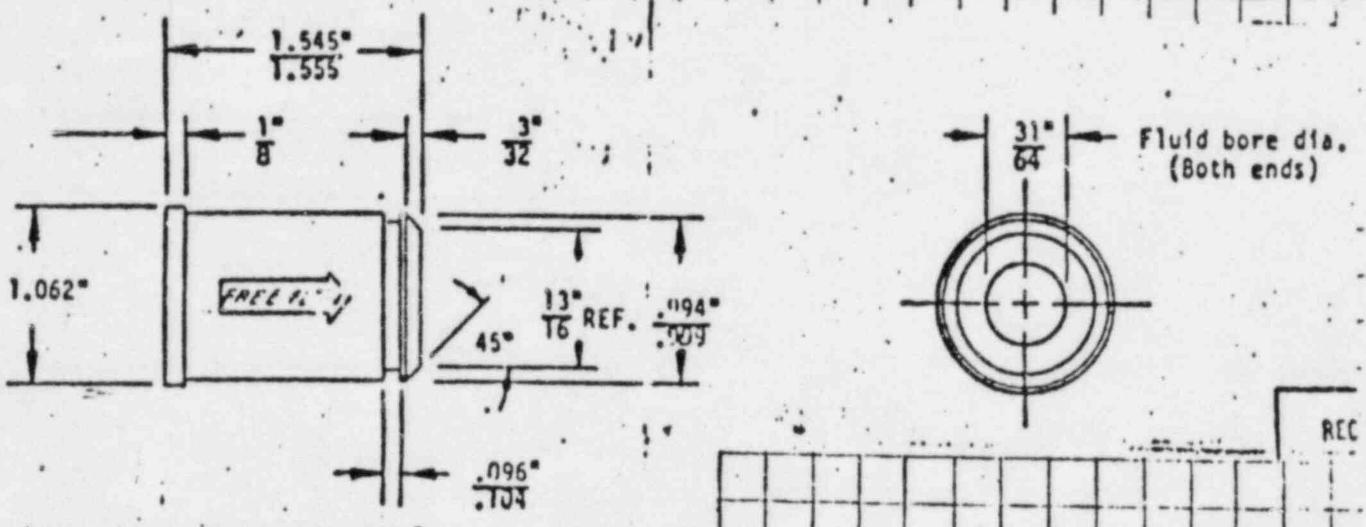
Client
Project
Proj. No. Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

KEPNER CHECK VALVE FIGURE NO 1283



NOTE:

Material: Body, cap, retainer, and poppet to be B-1113 Steel. Spring is AISI 302 Stainless Steel.

Finish: Carbon steel parts cadmium plated.

Seals: Flexible seals are VITON MATERIAL
(Exterior seal is Uniform size #020.
NOT SUPPLIED WITH VALVE.)

Cracking Pressure: 1 to 2 psig.

REV. A
RFC. NO. 36764
DATE 1-4-71
FIRST USED
MATERIAL WAS POLYACRYLATE RUBBER.

Log No.'s

0303161703
0201161703
0202161703
0601161703

NAME
VALVE CHECK

VENDOR
KEPNER PRODUCTS
PT & DWG .C 1283

FIRST USED
16-645-E-3

REV. —
RFC. -
DATE 1-13-5

8366533

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GENERAL MOTORS CORP.
LA GRANGE, ILLINOIS—U.S.A.

SARGENT & LUNDY
ENGINEERS
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Calcs. For

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Calc. No. CQD-014046
Rev. 00 Date 6-1-94
Page B1.91.2 of

Client
Project
Proj. No. 6995-00 Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

KEPNER FIG. 1283 CHECK VALVE

VALVE LOG NO.

0303161703
0201161703
0202161703
0601161703

ADDITIONAL INFORMATION

ACTIVE

CONCLUSION

THESE CHECK VALVES ARE LOCATED* INSIDE THE SOAKBACK FILTER. THERE IS NO PIPING DIRECTLY ATTATCHED TO THESE VALVES THEREFORE THERE IS NO PIPING END LOADS ASSOCIATED WITH THEM. THEREFORE, THERE IS NOTHING TO CHECK AS FAR AS ASME REQUIREMENTS ND-3521(a) IS CONCERNED. SINCE THESE VALVES HAVE <u>NO</u> EXTENDED STRUCTURES THAT ARE ESSENTIAL TO MAINTAINING PRESSURE INTEGRITY THEN THE REQUIREMENTS OF ASME ND-3521 (c) ARE SATISFIED.

* SEE THE ATTACHED SHEETS OF LAP-400-3 REV. 0 AUG. 10, 1983 FOR THEIR LOCATION. THIS DOCUMENTATION IS ATTACHED AT THE REAR OF THIS SECTION.

SARGENT & LUNDY
ENGINEERS
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Calcs. For

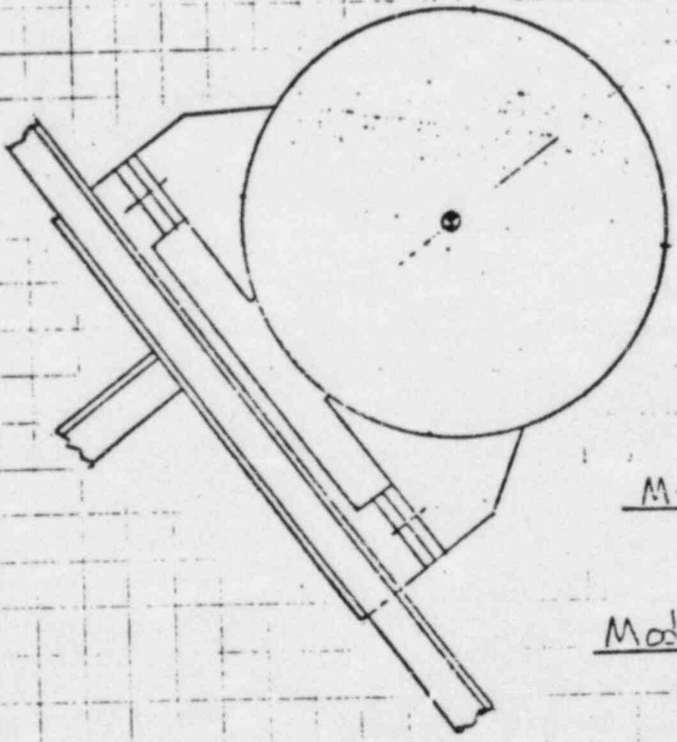
☒ Safety-Related ☐ Non-Safety-Related

Client
Project
Proj. No. Equip. No.

Prepared by Dave Wright Date 5/17/84
Reviewed by Date
Approved by Date

Lube Oil Filter Assy.

Log NO. 0803160300



Mfg. : Commercial Filters

Model No. : 23800-86L

The filter modeled in the previous analysis was an EMD-8362640. The vendor for EMD on this item was commercial filters and the model No. 23800-86 PLA. This is identical to the Shoreham filter so all of the *previous analysis applies.

* File no. S+L-EMD-00808 Rev. 00 dated 4/19/77 on the subject of Qualification of E.D.Gs.

Form GQ-3.08.1 Rev. 2

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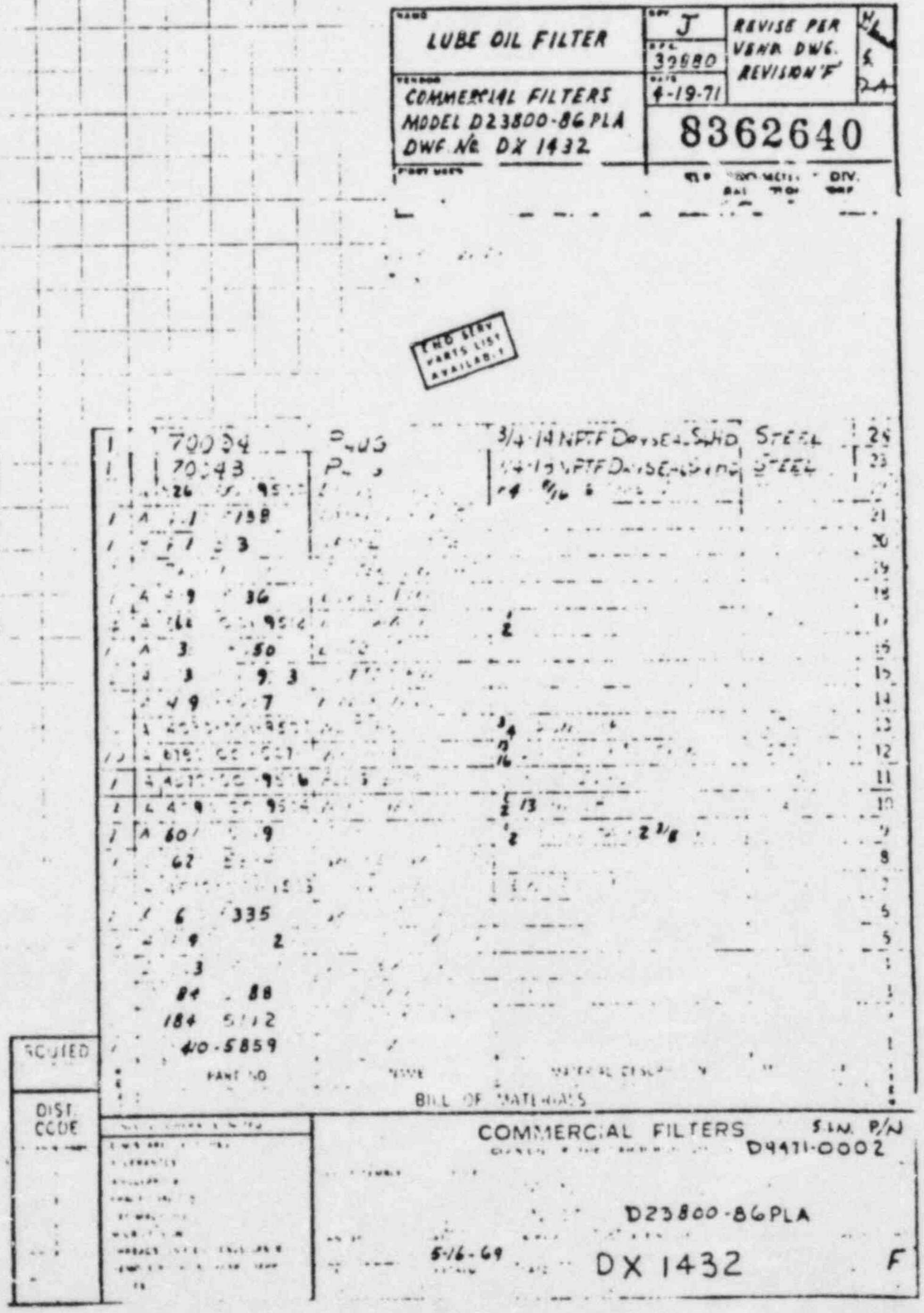
GM-EMD Drawing Label



NAME	REV		
LUBE OIL FILTER	J	REVISE PER VEND. DWG. REVISION 'F'	H.L. S. R.A.
	39880		
VENDOR COMMERCIAL FILTERS MODEL D23800-86PLA DWG. No. DX 1432	4-19-71		
	8362640		

DIV.

END SERV PARTS LIST AVAILABLE

1 70034 PLUG 3/4-14 NPTF DRYSEAL-STND STEEL 25
1 70043 P[illegible] 1/4-18 NPTF DRYSEAL-STND STEEL 23

PART NO. NAME MATERIAL DESCRIPTION

BILL OF MATERIALS

COMMERCIAL FILTERS S.N. P/N D4471-0002

D23800-86PLA

5-16-69

DX 1432 F

MAXIMUM STRESSES

DAT. 2/11/77 | W.O. 66875
PERFORMED BY: Arun.

DIESEL GENERATOR SET –

S. NO.	STRUCTURE REGION	ELEMENT NO.	NODE NO.	STRESS PSI
1	GENERATOR STATOR	329	1088-1086-1087	2262.
2	GENERATOR ROTOR	52	1128-1129	2016.
3	GENERATOR BRG. BRACKETS	396	1225-1241	4748.
4	EXCITER STATOR	1023	847-855	415.
5	ACCESSORY RACK FRAME	489	334-345	5786.
6	LUBE OIL FILTER	529	523-527-531	5553.
7	AMOT THERMOSTAT	581	612-641-613	1320.
8	WATER EXPANSION TANK	664	733-731-739	3235.
9	HEAT EXCHANGER	713	246-240-239-245	20783.
10	JACKET WATER PIPING	777	444-443-449-451	12621.
11	ACCESSORY RACK PIPING	676	659-660	1058.
12	BASE	329	135-136	8821.

AIR-START SKID-MOUNTED ASSEMBLY –

1	AIR RECEIVERS	16	30-31	107.
2	BASE, COMPRESSORS AND AIR DRYERS	26	24-34	1937.

ENGINEERING DEPARTMENT

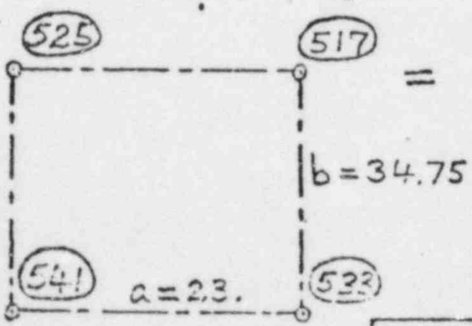
LUBE OIL FILTER
CONNECTION BOLT LOADS

DATE: 2/17/77
FILE: W.O. 63683
PERFORMED BY: Arun

S. No.	ELEMENT NO.	NODE NO.	ELEMENT FORCES		
			Fx #	Fy #	Fz #
1	517	521-537	474.	35.	184.
2	525	526-542	163.	167.	360.
3	533	531-547	528.	74.	256.
4	541	536-552	188.	189.	432.

$FX_{(MAX.)} = 528.^{\#}$ $\sqrt{F_y^2 + F_z^2}_{(MAX.)} = \sqrt{189.^2 + 432.^2} = 472.^{\#}$

$$F_{TENSILE\,(MAX.)} = \underset{El.533}{F_x} + \underset{El.517}{\frac{M_y}{b}} + \underset{El.541}{\frac{M_z}{a}}$$



$$= \underset{El.533}{528.} + \underset{El.517}{126./34.75} + \underset{El.541}{417./23.}$$

$$\underline{\underline{F_{TENSILE(MAX)} = \underline{550.}^{\#}}}$$

$$\underline{F_{SHEAR\,(MAX.)}} = \sqrt{\left(\underset{El.541}{\frac{(\Sigma M_x)a}{2(a^2+b^2)} + F_y}\right)^2 + \left(\underset{El.541}{\frac{(\Sigma M_x)b}{2(a^2+b^2)} + F_z}\right)^2}$$

ΣM_x for elements 517, 525, 533 AND 541 ≈ 0

$$\underline{\underline{= \sqrt{189.^2 + 432.^2} = \underline{472.}^{\#}}}$$

Mounting Bolt Stress

Bolt Size – 5/8"

Material – SAE Gr. 5, SA-449

Stress Area – 0.2256 in² (Machinery's handbook page 1282, UNC screw)

$$\sigma_t = \frac{F_{t\,max}}{A} = \frac{550\ \text{Lbs}}{0.2256\ \text{in}^2} = 2438\ \text{psi}$$

$$\tau_r = \frac{F_{r\,max}}{A} = \frac{472\ \text{Lbs}}{0.2256\ \text{in}^2} = 2092\ \text{psi}$$

Allowable Stress

Per ASME B+PV code, 1977 edition with summer 79 addendum, section III, article 2460 of Appendix XVII,

for combined loading (tensile, shear) the following must be true:

$$\frac{f_t^2}{F_t^2} + \frac{f_r^2}{F_r^2} \leq 1.00$$ (see page B.1.13.1 for symbols)

$F_t = 0.3 S_u = (.3)(\overset{*}{120}\text{ksi}) = 36{,}000\text{psi}$ / $F_r = 0.124(S_u) = .124(\overset{*}{120}) = 14{,}880\text{psi}$

$$\frac{2438^2}{36{,}000^2} + \frac{2092^2}{14{,}880^2} = 0.024 \leq 1.00 \Rightarrow \text{adequate.}$$

*Note – From ASME B+PV code, Section III table I-7.3 of Appendix I.

Calcs. For

☒ Safety–Related ☐ Non–Safety–Related

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Proj. No.	Equip. No.	Approved by	Date

Maximum Shell Stress

The maximum shell stress was found to be 5553 psi in the previous analysis. If we assume the material to be A-36 carbon steel we have an allowable stress of:

$S_a = 13.7$ Ksi (for SA-36 plate steel per ASME B+PV codes, table I-7.1 of Appendix I.

$S_a > \sigma_p$ so the shell is adequate.

Conclusions: The Lube oil filter is adequate for use in the shoreham Nuclear plant.

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ENGINEERS
CHICAGO

Calcs. For

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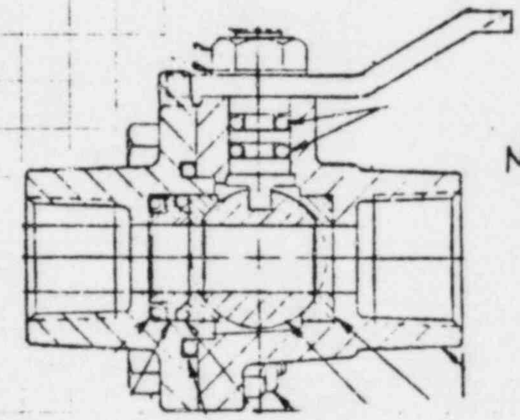
CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.1.98

Client		Prepared by Dail Wright	Date 5/17/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Lube Oil System Ball Valve

Log No. 0803161601



Mfg: Smith Valve

Model : 718-125 (1¼")

The Lube oil Ball Valve is identical to equipment found on the LaSalle (CECo) Diesels. The valves in the previous report (CQD-010330, Rev.00) were qualified in accordance with ASME B+PV code sect. III subsection ND-3500 (for class 3 components.) In addition, the applicable valves are qualified in conformance with ANSI B16.34. By using the documents as design rules, all valves will be qualified and shown to maintain structural and pressure integrity after the postulated seismic occurance. A copy of the analysis from the above report is attached.

Form GQ-3.08.1 Rev. 2

Calcs. For

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Reviewed by Date

Approved by Date

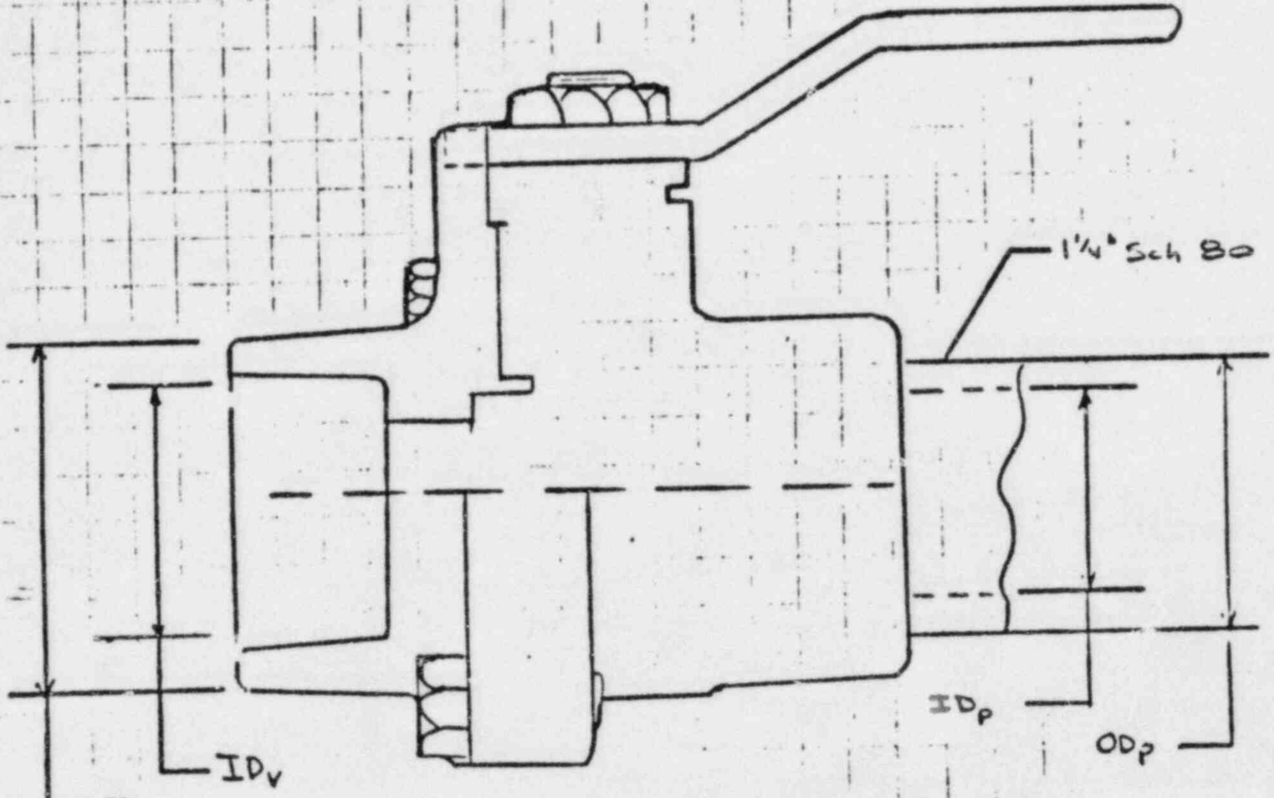

FIGURE 11

ODv = 1.9375 IN
IDv = 1.1875 IN
ODp = 1.660 IN
IDp = 1.278 IN

(1.9375" − 2(.375")) = 1.1875 IN. (SEE PHONE MEMO attached)

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Client		Prepared by	Date
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Proj. No.	Equip. No.	Approved by	Date

BODY STRUCTURAL INTEGRITY

S_V = VALVE BODY ALLOWABLE STRESS LIMIT FOR ASTM B283 (ALLOY 377) MATERIAL AT 100°F
= 12.0 KSI (ANSI / ASME B 31.1 - 1983 ED. APPENDIX A, TABLE A-6
Section III.

S_P = PIPE ALLOWABLE STRESS LIMIT FOR ASTM A106 GD B (ASSUMED) MATERIAL @ 100°F
= 15.0 KSI (TABLE I-7.1 P.103 - ASME APPENDICES, Section III.

SINCE S_V/S_P = 12.0/15.0 = 0.8 < 1.0 THEN:

SHOW THAT $\frac{S_V}{S_P} \times \frac{Z_V}{Z_P} > 1.1$

Z_V = SECT MOD. OF VALVE BODY

$$= \frac{3.14}{32} \frac{\left[(OD_V)^4 - (ID_V)^4\right]}{OD_V}$$

$$= \frac{3.14}{32} \frac{\left[(1.9375\ IN)^4 - (1.1875\ IN)^4\right]}{(1.9375\ IN)} = 0.61328\ IN^3$$

Z_P = SECT MOD OF PIPE
= 0.2914 IN^3 (FOR 1¼" SCH 80, CAMERON CAT. P 172)

$$\frac{S_V}{S_P} \times \frac{Z_V}{Z_P} = 0.8 \times \frac{0.61328}{0.2914} = 1.63 > 1.1$$

Form GQ-3 08.1 Rev. 2

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ENGINEERS
CHICAGO

☒ Safety-Related ☐ Non-Safety-Related

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Proj. No. / Equip. No.	Approved by / Date

SHOW THAT $\frac{S_v}{S_p} \times \frac{A_v}{A_p} > 1.1$

A_v = CROSS SECT. AREA OF VALVE BODY

$= \frac{\pi}{4}\left[OD_v^2 - ID_v^2\right]$

$= \frac{\pi}{4}\left[(1.9375\ IN)^2 - (1.1875\ IN)^2\right] = 1.8407\ IN^2$

A_p = CROSS SECT. AREA OF CONNECTING PIPE

$= 0.881\ IN^2$ (FOR 1¼ SCH 80 PIPE FR. CAMERON CAT. P.172)

$\frac{S_v}{S_p} \times \frac{A_v}{A_p} = .8 \times \frac{1.8407}{0.881} = 1.67 > 1.1$

∴ THE VALVE BODY IS STRONGER THAN THE PIPE

Form GQ-3.08.1 Rev. 2

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☒ Safety-Related ☐ Non-Safety-Relat

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

FIGURE 718 - 1¼" SIZE

THE BOLT THREAD SERIES & SIZE & MATERIAL, AND THE PRESSURE CLASS ARE THE SAME AS FIG. 709 - 1¼" SIZE. (4 BOLTS)

GASKET IS BUNA-N 'O' RING WITH AN ID = 1 IN AND THICKNESS = 0.103 IN

$$OD = ID + 2\,\text{thickness} = 1 + 2(.103) = 1.200\ \text{IN}$$

$$A_g = \frac{\pi}{4}(1.206\ \text{IN})^2 = 1.1423\ \text{IN}^2$$

$$F = S_r W_v = 6.23g\ (5.0\ \text{LBS}) = 31.15\ \text{LBS}$$

$$\frac{P_c \times A_g + F}{A_b} = \frac{400\ \frac{\text{LB}}{\text{IN}^2} \times 1.1423\ \text{IN}^2 + 31.15\ \text{LBS}}{0.904\ \text{IN}^2}$$

$$= 540\ \text{PSI}$$

$$540\ \text{PSI} < 7000\ \text{PSI} < 8050\ \text{PSI}$$

Neither the threads nor the bolts will Fail.

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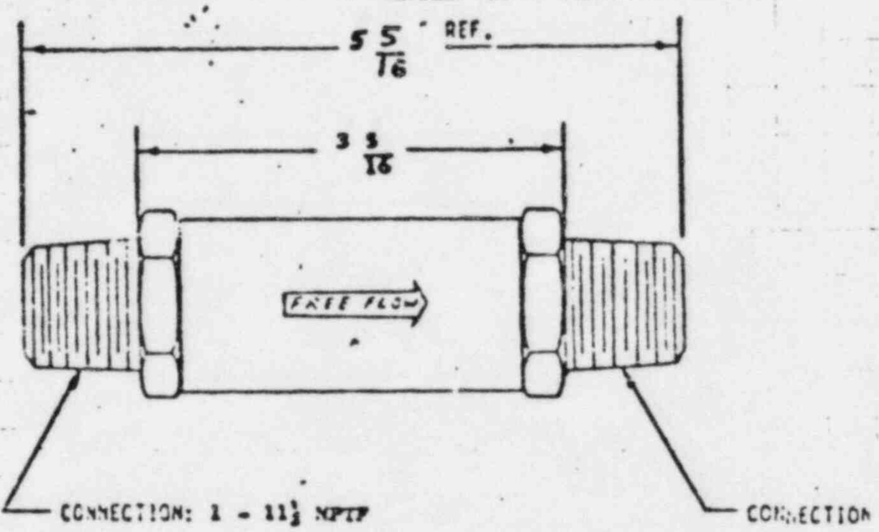
☒ Safety–Related ☐ Non–Safety–Relat

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.1.103

Client		Prepared by Dave Wright	Date 5/17/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Primary Lube Oil filter Check Valve

Log No. 0803161705



The primary Lube oil filter check valve found on the shoreham (LILco) Diesels are identical to those found on the LaSalle (CEco) Diesels. Since they were shown to be adequate for the LaSalle diesels they are also adequate for the Shoreham diesels. (original qualification under S+L-CQD-010330, Rev. 00). The copy of the section where this valve is addressed in the other analysis is attached.

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CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

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Proj. No.	Equip. No.	Approved by	Date

SEISMIC DISTURBANCE EFFECT ON CRACKING PRESSURE

THE SEISMIC RESULTANT FORCE, S_r, HAS AN EFFECT ON THE VALVE CRACKING PRESSURE DETERMINED AS FOLLOWS

FIGURE 316C-1-30

$S_r = 6.23$ g's

PA = PRESSURE AREA = 0.85 IN^2 (PHONE MEMO) 3/15/84 attached

W = WT OF MOVING PARTS = WT OF SPRING + WT. OF POPPET

$$= \frac{0.9 \text{ OZ}}{16 \text{ OZ/LB}} + \frac{3.65 \text{ OZ}}{16 \text{ OZ/LB}} = 0.284375 \text{ LB}$$ (PHONE MEMO) 3/15/84 attached

$$\Delta \text{ CRACKING PRESSURE} = \pm \frac{S_r(W)}{PA}$$

$$= \frac{6.226 \text{ g's } (0.284375 \text{ LB})}{0.85 \text{ IN}^2} = 2.08 \text{ PSI}$$

THE CRACKING PRESSURE FOR THIS VALVE IS 30PSI

Form GQ-3.08.1 Rev. 2

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VALVE BODY DATA

FIGURE NO	* VALVE BODY O.D (IN) $O.D_v$	* NOMINAL WALL THICKNESS t_n	VALVE BODY I.D (IN) $I.D_v$
106C-1	0.800	0.150	0.50
212C-2	1.430	0.233	0.964
316C-1-30	1.800	0.220	1.36

$$O.D_v = ID_v + 2(t_n) \Rightarrow I.D_v = O.D_v - 2(t_n)$$

* SEE PHONE MEMO FOR DETAILED DATA (attached)

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☒ Safety-Related ☐ Non-Safety-Relate

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BODY STRUCTURAL INTEGRITY 316Cr1 -3D

S_{vy} = VALVE BODY YIELD STRENGTH FOR AISI B1113 MATERIAL (B1117 MATERIAL)
= 68.0 KSI (Marks Standard Handbook For Mechanical Engineers, 8th edition, Mcgraw Hill)

S_{vp} = PIPE YIELD STRENGTH FOR SA 106 GRADE B MATERIAL
= 35.0 KSI (ASME B+PV code, Section III, Appendix I, table I-7.1)

SINCE S_{vy} / S_{vp} = 68KSI/35 KSI = 1.94 71.0

SHOW THAT THE VALVE IS 1.1 TIMES STRONGER THAN THE PIPE, THE SECTION MODULUS OF THE VALVE (Z_v) MUST BE 1.1 TIMES LARGER THAN THE SECTION MODULUS OF THE PIPE (Z_p). SEE PHONE MEMO FOR DIMENSIONS

$$Z_v = \frac{3.14}{32} \frac{\left[(1.80\,IN)^4 - (1.36\,IN)^4\right]}{1.80\,IN} = 0.3859\,IN^3$$

$Z_p = 0.1606\,IN^3$ (1" SCH80 CAMERON CATALOGUE P.171)
$(1.1)Z_p = 1.1(0.1606\,IN^3) = 0.1766\,IN^3$ $0.3859\,IN^3\ 70.1766\,IN^3$

SHOW THAT THE VALVE IS 1.1 TIMES STRONGER THAN THE PIPE, THE CROSS SECTION AREA OF THE VALVE MUST BE 1.1 TIMES LARGER THAN THE CROSS SECTION AREA OF THE PIPE (A_p). SEE PHONE MEMO FOR DIMENSIONS

$$A_v = \frac{3.14}{4}\left[(1.80\,IN)^2 - (1.36\,IN)^2\right] = 1.092\,IN^2$$

$A_p = 0.639\,IN^2$ (1" SCH80 - CAMERON CATALOGUE P.171)

$1.1(A_p) = 1.1(0.634\,IN^2) = 0.7009\,IN^2$

Form GO-3.08.1 Rev. 2

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Prepared by Date

Reviewed by Date

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WE HAVE SHOWN THAT THE VALVE BODY IS 1.1 TIMES STRONGER THAN THE CONNECTING PIPE BY THE SECTION MODULUS METHOD AND CROSS SECTION AREA METHOD. THIS SATISFIES THE REQUIREMENTS OF ASME ND-3521 (a) Section III, ASME B+PV Code.

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Client		Prepared by	Date
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Proj. No.	Equip. No.	Approved by	Date

SPRING NATURAL FREQUENCY

NATURAL FREQUENCY OF THE SPRING IS CALCULATED ASSUMING A COMPRESSION SPRING FIXED AT BOTH ENDS PER "Handbook of Mechanical Spring Design, Associated Spring Corp, Bristol Conn. 1964. (SEE PHONE MEMO'S FOR DATA BELOW)

GOVERNING EQN

$$F_n = (d/9D^2N)\sqrt{Gg/\rho}$$

FIGURE 106C-1 :

$d = 0.028$ IN. WIRE DIA

$O.D = 0.40$ IN. SPRING OUTER DIA

$D = O.D - 2(d/2) = O.D - d = .40'' - .028'' = 0.386$ MEAN SPRING DIA.

* $\rho = 0.285$ LB/IN³ (AISI 302 STAINLESS STEEL) DENSITY

* $G = 10.6 \times 10^6$ LB/IN² MOD. OF RIGIDITY

* $g = 386.4$ IN/SEC²

$N = 10$ NO. OF ACTIVE COILS

$$F_n = 0.028/(9(0.386)^2 10 \sqrt{\frac{10.6 \times 10^6 (386.4)}{0.285}}$$

$F_n = 250$ HZ RIGID

NOTE: ABOVE DATA CAN BE FOUND IN PHONE MEMO & MARKS HANDBOOK 8th ED.

* VALUES ARE CONSTANT FOR FIGURE 106C-1, 212C-2, 316C-1-3D (SAME SPRING MATERIAL)

Calcs. For

☒ Safety–Related ☐ Non–Safety–Related

Client

Project

Proj. No. Equip. No.

Prepared by Date

Reviewed by Date

Approved by Date

N/A

~~FIGURE 212C-2:~~

~~d = 0.048 IN~~
~~O.D = 0.620 IN~~
~~D = 0.620" - 0.048" = 0.572 IN~~

~~SEE PHONE MEMOS FOR DATA~~

~~$F_n = 0.048/(9(0.572)^2\ 8.4)\sqrt{\frac{10.6\times10^6(386.4)}{0.285}}$~~

~~F_n = 232 HZ RIGID~~

FIGURE 316C-1-30

d = 0.110 IN
O.D = 1.087 IN
D = 1.087" - 0.110" = 0.977

$$F_n = 0.110/(9(0.977)^2\ 5.1)\sqrt{\frac{10.6\times10^6(386.4)}{0.285}}$$

F_n = 301 HZ RIGID

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Client		Prepared by	Date
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BENDING STRESS ON POPPETT (FIG. 316C-1-30)

SEE PROCEDURE FROM FIG 106C-1

CASE 1

$d_p = 1.36$ IN

CASE 3

$r_o = D/2 = 0.977''/2 = 0.4885$ IN

$a = d_p/2 = 1.36''/2 = 0.68$ IN

BOTH CASES

$t = 7/16'' = 0.4375$ IN

$W_p = 3.65 OZ \times \frac{1 LB}{16 OZ} = 0.228$ LB

$F = S_r W_p = 6.23 g\,(0.228\text{ LB}) = 1.42$ LB

$$\sigma_b = 3000 \frac{\pi}{4}(1.36)^2 \times \frac{3(3\times3.28+1)}{8\pi(3.28)(.4375)^2} + \left[\frac{3(1.42)}{2\pi(3.28)(.4375)^2}\left\{4.28 \times \ln\frac{(.68)}{(.4885)} + \frac{2.28}{2} - (2.28)\left(\frac{(.4885)^2}{2(.68)^2}\right)\right\}\right]$$

$$= 8982 + [1.08\{1.41 + 1.14 - .588\}] = 8982 + 2.11$$

$$\sigma_b \hat{=} 8984 \text{ PSI} < 17.5 \text{ KSI}$$

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☒ Safety-Related ☐ Non-Safety-Related

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Prepared by Date
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SHEAR STRESS ON POPPETT (FIG. 316C-1-30)

$$\sigma_s = \frac{F + \frac{P\pi d_p^2}{4}}{\pi d_p t}$$

SHEAR STRESS ALONG PERIMETER OF POPPETT

$$= \frac{1.42 + 3000\pi \frac{(1.36)^2}{4}}{\pi (1.36)(.4375)} = 2333 \text{ PSI}$$

< 14.0 KSI

(Allowable stress from ASME B+PV code, Section III, Appendix I.)

Form GQ-3.08.1 Rev. 2

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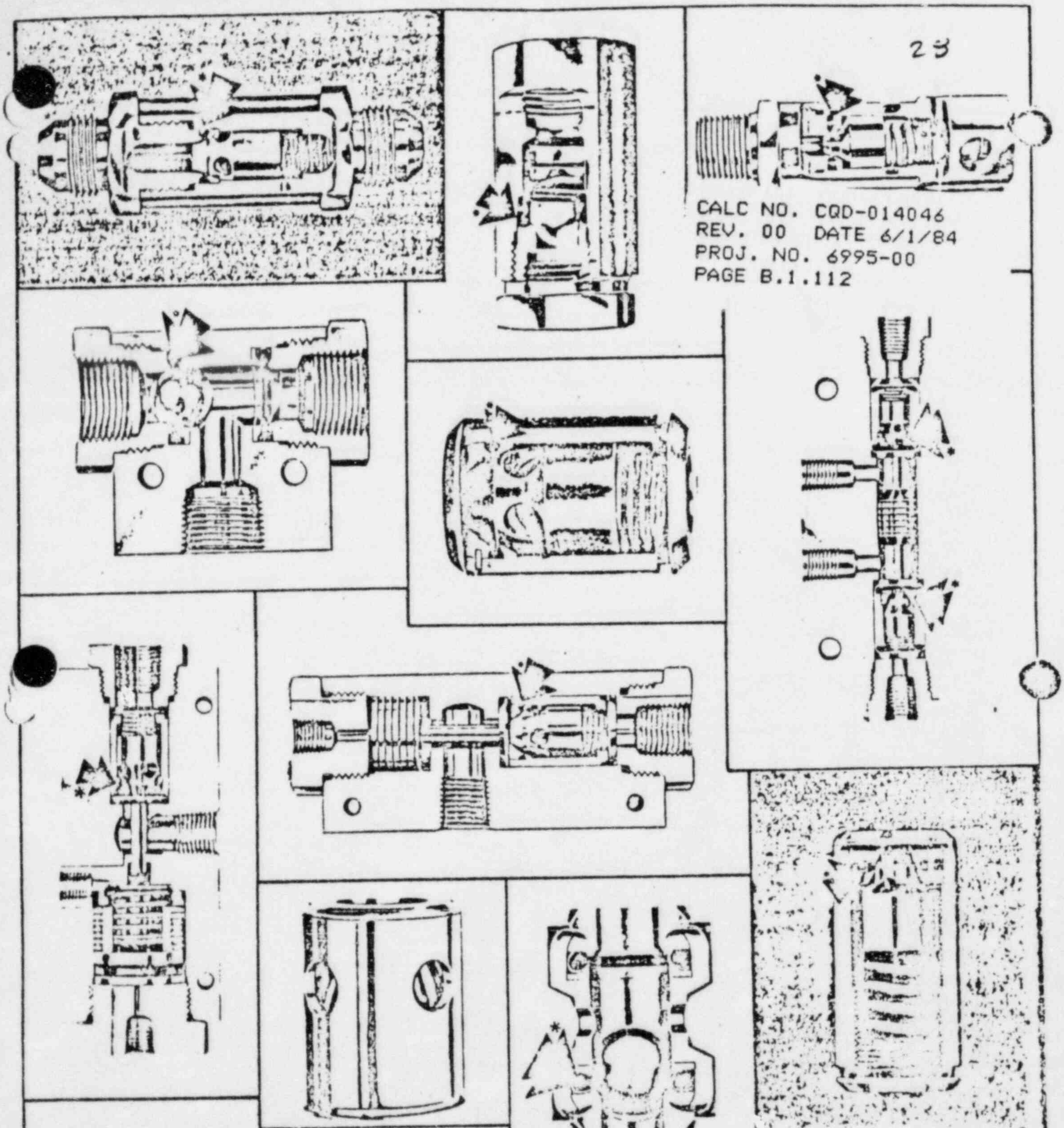

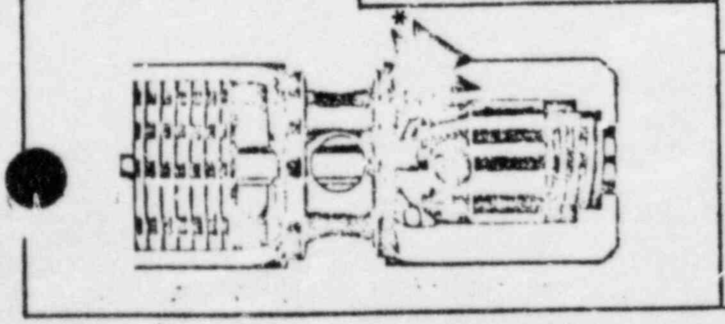

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PRODUCTS COMPANY
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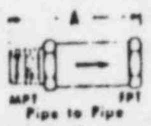
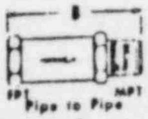
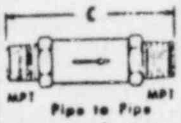
Kep-O-seal Valve Division

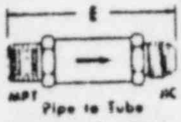
Calc. No: CQD-
Rev: Date:
Proj. No:

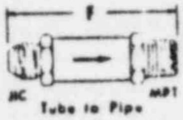
Check and Relief Check Valve



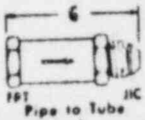


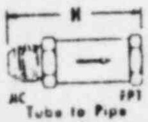


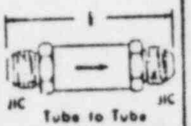


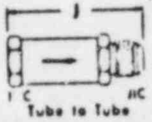


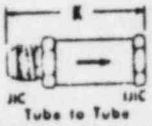
SIZE NPT	SIZE JIC	MODEL	DIM. A	MODEL	DIM. B	MODEL	DIM. C	MODEL	DIM. D	MODEL	DIM. E	MODEL	DIM. F	DIM. HEX.
⅛	¼	104	2 5/32	204	1 29/32	304	2 11/32	404	1¾	504	2 19/32	704	2 15/32	11/16
¼	⅜	106	2⅞	206	2 15/32	306	2 21/32	406	2⅜	506	2 15/16	706	2 15/16	13/16
⅜	½	108	3 1/16	208	2 27/32	308	3½	408	2 7/16	508	3½	708	3½	1
½	⅝	110	3¾	210	3 7/32	310	4 [illegible]/32	410	2 15/16	510	3 31/32	710	3 31/32	1⅛
¾	¾	112	4 3/16	212	3⅝	312	4 13/32	412	3 7/16	512	4 7/16	712	4 7/16	1 7/16
1	1	116	5	216	4 5/16	316	5 11/32	416	4	516	5 7/32	716	5 7/32	1 13/16
1¼	1¼	120	•	220	•	320	•	420	4 15/16	520	•	720	•	2⅜
1½	1½	124	•	224	•	324	•	424	5 3/16	524	•	724	•	2⅝
2	2	132	•	232	•	332	•	432	6¼	532	•	732	•	3½

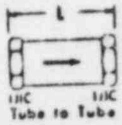












SIZE NPT	SIZE JIC	MODEL	DIM. G	MODEL	DIM. H	MODEL	DIM. I	MODEL	DIM. J	MODEL	DIM. K	MODEL	DIM. L	DIM. HEX.
⅛	¼	904	2 1/32	1104	2 9/32	1304	2 19/32	1504	2 15/32	1704	2½	1904	2 13/32	11/16
¼	⅜	906	2 7/16	1106	2 13/16	1306	2⅞	1506	2 21/32	1706	2 29/32	1906	2 11/16	13/16
⅜	½	908	2 29/32	1108	3 3/32	1308	3 17/32	1508	3 9/32	1708	3¼	1908	2 31/32	1
½	⅝	910	3 5/32	1110	3 11/16	1310	3 29/32	1510	3⅝	1710	3⅞	1910	3 19/32	1⅛
¾	¾	912	3 11/16	1112	4 7/32	1312	4 15/32	1512	4 5/32	1712	4 7/16	1912	4⅛	1 7/16
1	1	916	4 7/32	1116	4 29/32	1316	5⅛	1516	4 13/16	1716	5	1916	4 23/32	1 13/16
1¼	1¼	920	5 3/16	1120	5 29/32	1320	6⅛	1520	•	1720	•	1920	•	2⅜
1½	1½	924	5 9/16	1124	6 1/32	1324	6⅞	1524	•	1724	•	1924	•	2⅝
2	2	932	6 25/32	1132	7 19/32	1332	8 3/32	1532	•	1732	•	1932	•	3½

Legend:
MPT — Male pipe thread (Dryseal)
FPT — Female pipe thread (Dryseal)
JIC — Male tube connection (SAE Compatible)
IJIC — Female tube connection (SAE Compatible)
• — Consult Factory for availability
→ Arrow indicates direction of FREE FLOW
Dimensions in inches

CONSTRUCTION MATERIALS

CODE LETTER — SPECIFICATION

A	Anodized aluminum alloy
B	Commercial brass
C	Cadmium plated steel
D	Stainless steel (Type 303)

All standard springs are made of stainless steel. CONSULT FACTORY FOR ALTERNATE MATERIALS.

SERVICE APPLICATIONS:

#1 Standard — General purpose oils and lubricants, hydraulic oil, air, water.

#2 Standard — Kerosene, fuel oil, gasoline, aromatic fuels, and liquified petroleum gas.

See Kepner Bulletin #1244-6 for added details on alternate fluids, temperature, and seal data.

When ordering, be sure to specify type of service desired. Signify by attaching standard type number as dashed suffix to model number. See order example.

If no service is specified, Standard Type #1 will be furnished.

OPERATING CONDITIONS:

Standard Working Pressure -- 3000 psi

Production Proof Testing available to 10,000 psi — CONSULT FACTORY.

Standard temperature range: -40°F to 300°F.

Special Kep-O-Seal Check Valves for extreme operating conditions are also available. CONSULT FACTORY.

VALVE CRACK & RELIEF PRESSURES:

Standard Cracking Pressure — 1 to 2 psi

Optional Cracking Pressures — Relief checks 5 psi, 10 psi, 25 psi, 50 psi, and 65 psi normally stocked. Intermediate settings to 150 psi to order. Consult current price sheet for details.

When ordering non-standard cracking pressure add desired pressure setting expressed in PSI as second dashed suffix to part number — SEE ORDERING EXAMPLE.

SIZES AND CONNECTIONS:

Standard — as shown above. There are no mounting limitations.

A wide variety of SAE and other common port configurations are also available to order.

ORDERING EXAMPLE

Select base number from above table. Attach code letter indicating desired construction material. Show fluid service by first dash number. Add second dash number to specify non standard cracking pressure.

EXAMPLE:

408C-1-25

Base Model Number — 408
Material Code — C
Fluid Service — 1
Cracking Pressure (PSI) — 25

Calc. No: CQD-
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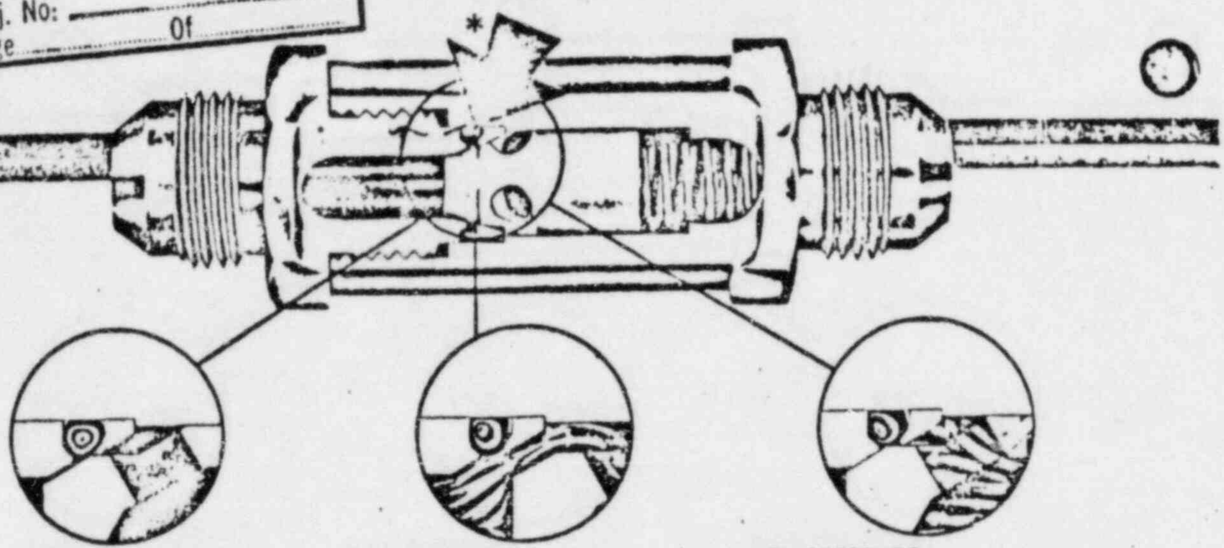
Page Of

Flexible S



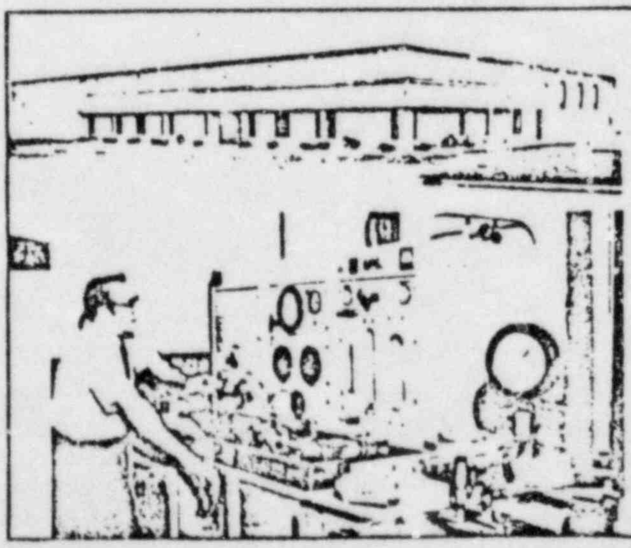
ZERO PRESSURE
NO FLOW
Relaxed seal ring and gentle seal-to-poppet contact guarantees low pressure sealing and eliminates valve chatter.

HIGH PRESSURE
FULL FLOW
Seal flexes to close off all external leakage around end cap. Enclosure protects seal ring, prevents seal displacement.

HIGH PRESSURE
REVERSE FLOW CHECKED
Seal still holding external leakage now also flexes around poppet. Higher pressures tighten the seal. LEAKAGE ZERO.

- **Positive leak tight sealing**
- **Positive action at very low pressure**
- **Full flow with low pressure drop**
- **Long life — easy maintenance**
- **Freedom from vibration and chatter**

The Kep-O-Seal Flexible Seal Seat* Valve design enjoys an enviable world-wide reputation for ingenuity and excellence in positive control of all kinds of fluid and gas systems. First developed through incredibly detailed laboratory testing, this design has been applied through the years to hundreds of thousands of control situations — some simple and some very complicated. Our seat design, utilizing a carefully engineered combination of metal-to-metal and resiliant seal contact, has been incorporated in a broad family of related in-line and cartridge type control valves. All Kepner valves provide very positive control with long service life and simple — easy maintenance. Seat "wire drawing" common to conventional valves has been completely eliminated by the Flexible Seal Seat* design as the seal closes around scratches, dents, and other irregularities and effectively prevents even the smallest leakage past the seat. All Kep-O-Seal Valves have generous flow passages, heavy internal wall sections, and low rate springs, for safety, efficiency and extra long service life. Kep-O-Seal in-line valves, Kepsēl Cartridge valves, and Kep-O-Lok valves are well designed, carefully manufactured, and competively priced.



Kepner Products Company owns and operates this completely modern production facility located in a northwestern suburb of Chicago. We are equipped to manufacture every machine part used in our product line. Quality Control is independently supervised, and manufacturing methods are constantly improved to keep pace with technical advances in the metal working industries.

Our laboratory is equipped for development and production testing of high pressure hydraulic and pneumatic systems. This test facility is fully instrumented for high and low temperature extremes, pressurized fluid and gas flow, and very high static pressures. It is also equipped for high pressure cyclic endurance testing.

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* PATENTED

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Villa Park, Illinois 60181

SOUTHWEST RESEARCH INSTITUTE
ATTN: STEVE STEWART
6220 CULEBRA ROAD
PO DRAWER 28510
SAN ANTONIO, TX 78284

Thank you for your interest in . . .

KEP-O-SEAL HYDRAULIC & PNEUMATIC CHECK VALVES

The attached literature will give you the information you requested. Should you have further questions about your specific requirements, they can be answered by:

INDUSTRIAL AIR & HYDRAULICS
12311 AMELIA DRIVE
HOUSTON, TX 77045

713-434-0111

Hydraulic and Pneumatic Valves

Calc. No: CQD-
Rev: Date:
Proj. No:
Page Of

Hydraulic and Pneumatic Check and Relief Check Valves

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Kep-O-Seal Valve Division

SARGENT & LUNDY

MEMORANDUM OF
TELEPHONE CONVERSATION

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.1.116

Date: 3/30/84

Time: 9:00 am

Person Called: David Trommeter (Name) of Kepner Products (Company)

Person Calling: Mark L. Weber (Name) of Sargent & Lundy (Company)

Project: LaSalle/Clinton **Project No.** 6548-30/4536-32

Subject Discussed: Kepner Check Valves Figures 106C-1, 212C-2, 316C-1-30

Summary of Discussion, Decisions and Commitments:

Mr. Trommeter provided me with the following data pertaining to the subject valves:

	106C-1	212C-2	316C-1-30
Spring: O.D. (in.)	.40-.39	.62-.61	1.087
wire diameter (in.)	.028	.048	.110
No. of active coils	10	8.4	5.1
Thickness of poppet at nose (in.)	7/32	23/64	7/16

Thickness of poppet are approximate valves measured off of drawings.

MLW/eg

cc D. Wright - 30
D. Trommeter - Kepner
AEM/APD/MLW - 30

Signature

File: [illegible]

SARGENT & LUNDY

MEMORANDUM OF
TELEPHONE CONVERSATION

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.1.117

Date: 3/16/84

Time: 1:15 pm

Person Called: David Trommeter (Name) of Kepner Products (Company)

Person Calling: Mark L. Weber (Name) of Sargent & Lundy (Company)

Project: LaSalle/Clinton Project No. 6548-30/4536-32

Subject Discussed: Kepner Check Valves

Figure 106C-1, Figure 212C-2, Figure 316C-1-30

Summary of Discussion, Decisions and Commitments:

Mr. Trommeter (Kepner) provided me (S&L) with the following data pertaining to the subject valves:

	Fig. 106C-1	Fig. 212C-2	Fig.316C-1-30
Valve Body O.D. (in.)	0.800	1.430	1.800
Nom. Wall Thickness (in.)	0.150	0.233	0.220

MLW/eg

cc D. Wright - 30
D. Trommeter - Kepner
AEM/APD/MLW - 30

Signature

SARGENT & LUNDY

MEMORANDUM OF
TELEPHONE CONVERSATION

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.1.118 of B.1.118
FINAL

Date: 3/15/84

Time: 1:00 pm

Person Called: Norm Engstrom (Name) of Kepner Products Company (Company)

Person Calling: Mark L. Weber (Name) of Sargent & Lundy (Company)

Project: LaSalle/Clinton Project No. 6548-30/4536-32

Subject Discussed: Kepner Check and Relief Check Valves
Figures 106C-1, 212C-2, 316C-1-30

Summary of Discussion, Decisions and Commitments:

Mr. Engstrom supplied me with the following data pertaining to the subject valves.

	Figure 106C-1	Fig. 212C-2	Fig. 316C-1-30
Wt. of spring (pz.)	0.05	0.30	0.90
Wt. of poppet (oz.)	0.20	1.35	3.65
Pressure area (in^2)	0.099	0.41	0.85
Spring rate (LB/IN)	1.55	6.5	38.6

MLW/eg

cc D. Wright - 30
N. Engstrom - Kepner
AEM/APD/MLW - 30

Signature

File: CQD-013404 - 30

B.2

B.2 FUEL OIL SYSTEM

SARGENT & LUNDY
ENGINEERS
CHICAGO

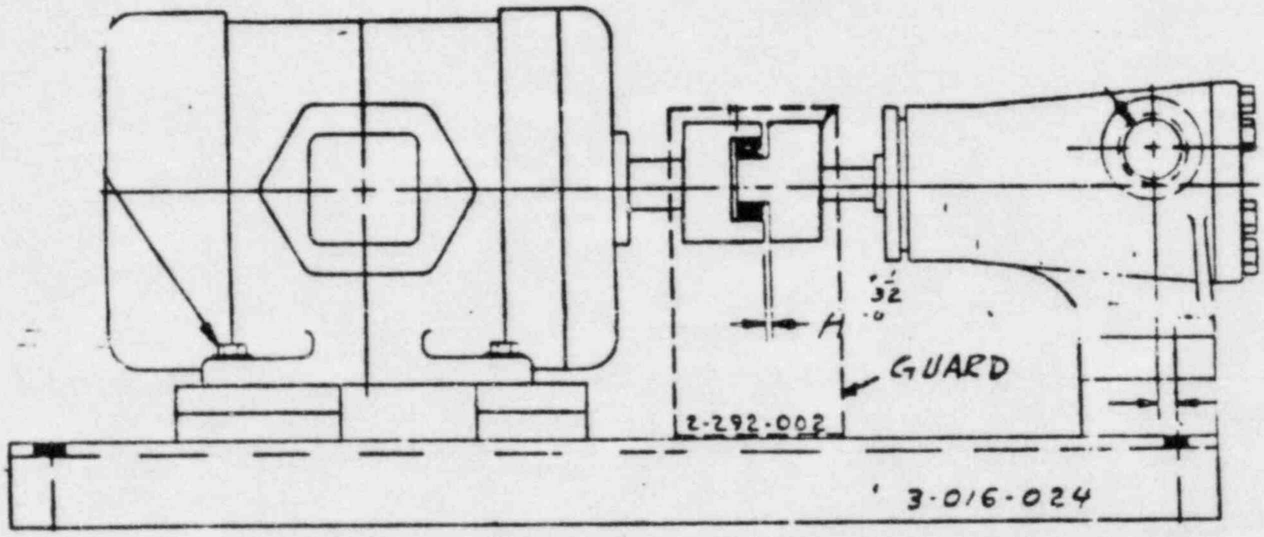
Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client	Prepared by David Wight Date 5/11/84
Project	Reviewed by Date
Proj. No. Equip. No.	Approved by Date

Fuel Oil Transfer Pumps (2)

Log No.'s 0803060303, 304, 309, 310



The fuel oil transfer pumps are found only on the master unit, engine number 2. This pump and motor assembly is identical, with the exception of the motor size, to the lube oil circulating pump assembly. Since the pump was addressed in the lube oil section it is only necessary to examine the electric motor drive. The motor drive found on the transfer system are smaller in size and output than the lube oil electric drive motor.

Calcs. For

☒ Safety–Related ☐ Non–Safety–Related

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Electric Motor Drive Identification

The following information was obtained from the master unit (engine 2) and supplemented with information obtained from Delco Products. (see phone memorandums between D. Wright and C. Gangone and D. Wright and J. Miller)

	Upper Motor	Lower Motor
Manufacturer :	Delco Products	Unable to determine
Model Number :	A9916M1	020908CC
Serial Number :	G63	none found
frame size :	56	56
type :	CS	CC
Rated Power :	3/4 H.P.	3/4 H.P.
Shaft size :	0.625 in	0.625 in

Method of Analysis

The motor will be checked for structural integrity and will be shown to be operable by a deflection analysis of the rotor using the air gap of 0.30 in as an allowable.

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Calcs. For

☒ Safety–Related ☐ Non–Safety–Related

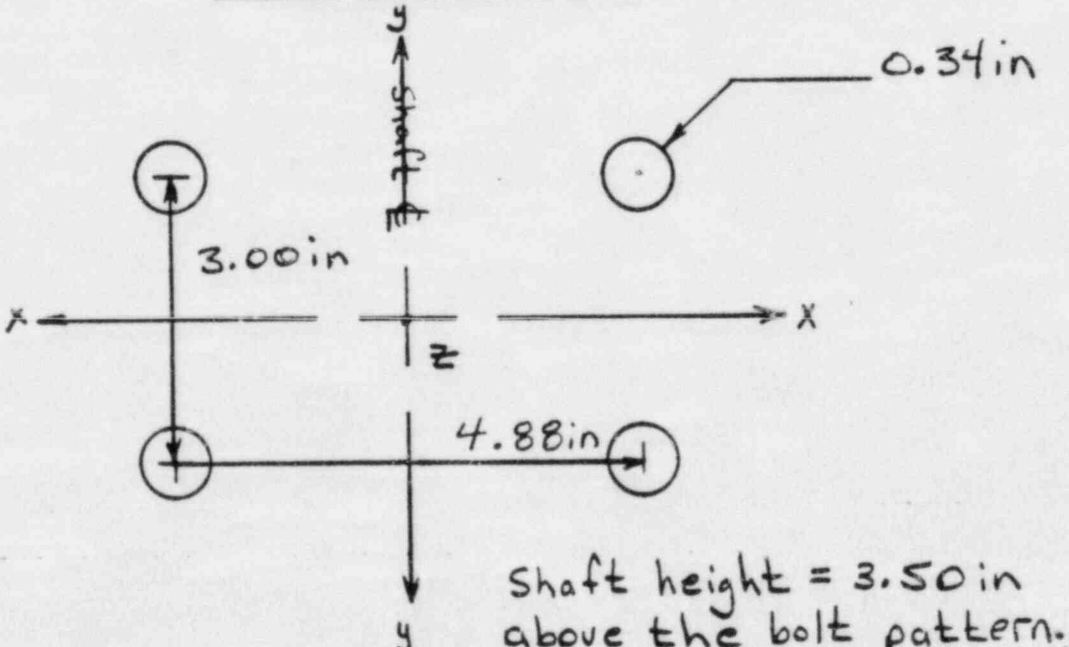
CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.2.4

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Structural Integrity Check

Due to the fact that the motor has no significant extended masses the most critical mode of failure would be the mounting bolts. The dimensions of the motor will be taken from National Electric Manufacturers Association standard MG1-1978, Parts 4 and 11 for frame size 56.

Bolt Pattern



Shaft height = 3.50 in above the bolt pattern. (from Above NEMA std.)

Since the motor weight is not known it will be estimated at 100 Lbs. This is the same weight used in the Lube oil Circulating pump analysis of the larger 184 framed motor.

Bolt Geometric Properties

Bolt hole Size– 0.34 in

assumed Bolt Size– 5/16 in SAE Gr. 5 (SA-449)

$A_t = 0.0524 \text{ in}^2$ { Machinery's Handbook, 12th edition
Page 1282, UNC screw, tensile "A_t" }

$I_{xx} = \Sigma(\cancelto{0}{I_x} + Ad^2) = 4(0.0524)(1.5)^2 = 0.4716 \text{ in}^4$

$I_{yy} = \Sigma(\cancelto{0}{I_y} + Ad^2) = 4(0.0524)(2.44)^2 = 1.248 \text{ in}^4$

Loading, Seismic + Weight

Since the motors are mounted in an upright position the loading due to weight will not be seen by the bolts. In order to account for any amplification of the seismic motion in the mounting frame an acceleration level of 2.5 g's will be used in each direction. Startup torque is negligable for a 3/4 Hp motor.

$F_x = Wt\, A_x = 100 \text{Lbs} (2.5 g\text{'s}) = 250 \text{ Lbs}$

$F_y = Wt\, A_y = 100 \text{Lbs} (2.5 g\text{'s}) = 250 \text{ Lbs}$

$F_z = Wt\, A_z - Wt = 100 \text{Lbs} (2.5 g\text{'s}) - 100 \text{Lbs} = 150 \text{ Lbs}$

$M_x = F_y C_z = 250 \text{Lbs} (3.50 \text{in}) = 875 \text{ in-Lbs}$

$M_y = F_x C_z = 250 \text{Lbs} (3.50 \text{in}) = 875 \text{ in-Lbs}$

$M_z = 0.0 \text{ in-Lbs}$ (no offset of the C.G.)

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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.2.6

Client

Project

Proj. No. Equip. No.

Prepared by Date

Reviewed by Date

Approved by Date

Check of Bolt Stress

$$\sigma_t = \frac{F_z}{4A_t} + \frac{M_x C_y}{I_{xx}} + \frac{M_y C_x}{I_{yy}}$$

$$\sigma_t = \frac{150\text{ Lbs}}{4(0.0524)} + \frac{875\text{ in-lbs}(1.50\text{ in})}{0.4716\text{ in}^4} + \frac{875\text{ in-lbs}(2.44\text{ in})}{1.248\text{ in}^4}$$

$$\sigma_t = 5209\text{ psi}$$

$$\tau_v = \frac{(F_x^2+F_y^2)^{1/2}}{4A_t} = \frac{250\text{ Lbs}\sqrt{2}}{4(0.0524)} = 1687\text{ psi}$$

Allowable Stress, Servicel level A+B

ASME B+PV code, 1977 edition with 1979 summer addendum, Section III, article 2460 of Appendix XVII :

$$F_t = 0.3\,S_u = 0.3(\overset{*}{1}20{,}000) = 36{,}000\text{ psi}$$

$$F_r = 0.124\,S_u = 0.124(\overset{*}{1}20{,}000) = 14{,}880\text{ psi}$$

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_r^2} \le 1.0 = \frac{5209^2}{36{,}000^2} + \frac{1687^2}{14{,}880^2} = 0.034 \le 1.00$$

The bolts are adequate.

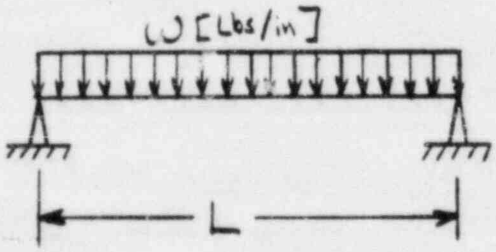
*Note- From Section III of ASME B+PV code, table I-7.3 of Appendix I for SA-449, D<1.0"

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Operability Check

As mentioned previously, operability will be demonstrated by a deflection analysis of the rotor. Allowable deflections will be assumed to be equal to the air gap of 0.30 in. The bearing/Rotor assembly will be modeled as a simply supported beam, with the entire 100 Lbs acting as a distributed load.

Analytical Model



Note - "L" will be conservatively estimated as the entire shaft length, not Brg. to Brg. distance. It is obtained from photo # 21.

$L \cong 13 in$

$Wt = 100 Lbs$

$\omega = 100 Lbs / 13 in = 7.69 Lbs/in$

$\omega_{seismic} = (2.5 g's \sqrt{2})(7.69 Lbs/in) = 27.2 Lbs/in$

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

$$\Delta_{max} = \frac{5wL^4}{384EI}$$

(from Design of Welded Structures By Blodgett, page 8.1-7)

$$I_{shaft} = \frac{\pi d^4}{64} = \frac{\pi(0.625)^4}{64} = 0.007\,in^4$$

$E = 30 \times 10^6$ psi (assume carbon steel.)

$$\Delta_{max} = \frac{5(27.2\,Lbs/in)(13\,in^4)}{384(30 \times 10^6)(0.007)} = 0.05\,in$$

since 0.05in is much less than the airgap of 0.30 in the motor will not bind and will remain operable during and after a seismic event.

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ENGINEERS
CHICAGO

Calcs. For

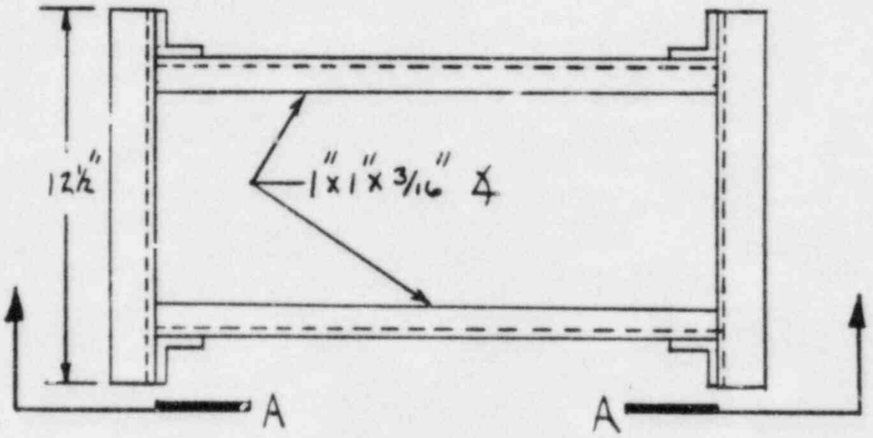
☒ Safety-Related ☐ Non-Safety-Related

Client		Prepared by David Wright	Date 5/11/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Diesel Fuel Transfer Pump Mounting Frame

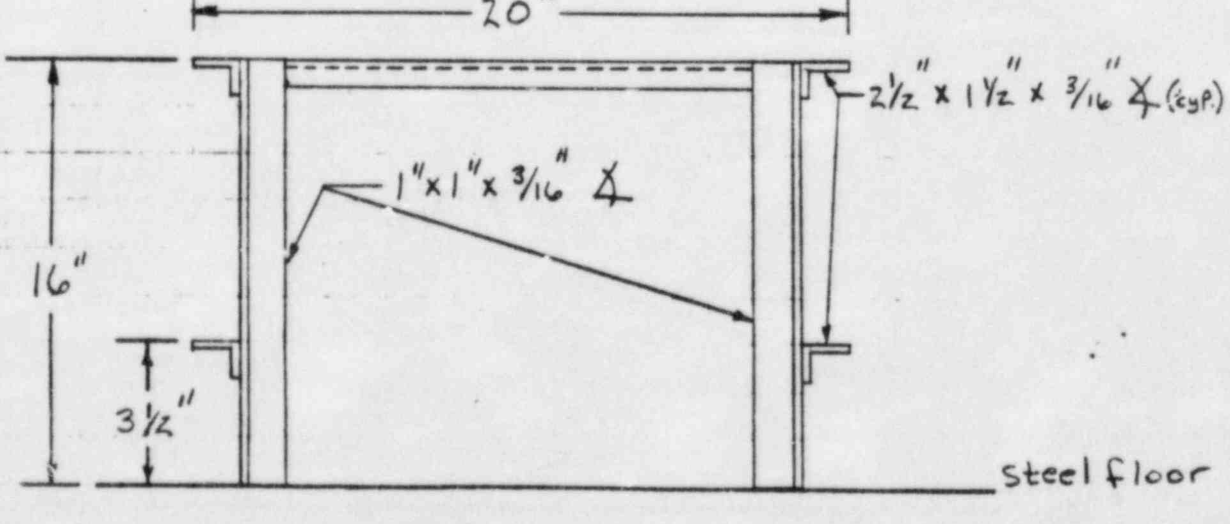
Log No. 080306031l

Sketch obtained during field walkdown



Plan View

Note- all welded construction



Section A-A

SARGENT & LUNDY
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Calcs. For

☒ Safety–Related ☐ Non–Safety–Related

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.2.10

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Method of Analysis

For conservatism and ease of analysis the frame will be modeled as 8

Actual System
2 D.O.F.

Assumed System
1 D.O.F.

This is very conservative since we are not taking credit for the stiffness in the frame, but treating it as a cantilever beam. The acceleration levels to be used are the shoreham peak (1.5).

$K_{assumed} = 4$(stiffness of a 16 in long $1 \times 1 \times \frac{3}{16}$ ∠, cantilever connection.)

$$K = 4\left[\frac{3EI}{l^3}\right] = 4\left[\frac{3(30\times10^6)(0.03)}{16^3}\right] = 2637 \text{ Lbs/in}$$

(from Marks Standard Handbook for Mechanical Engineers, 8th edition page 5-70)

Form GQ-3.08.1 Rev. 2

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety–Related ☐ Non–Safety–Related

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

$m_1+m_2=$ (weight of Both Pump/motor units)/386.4

$m_1+m_2=(\overset{*}{1}50\,Lbs+\overset{*}{1}50\,lbs)/386.4=0.776$

$$f_n=\frac{1}{2\pi}\sqrt{\frac{K}{m}}=\frac{1}{2\pi}\sqrt{\frac{2637}{0.776}}=9.3\ Hz$$

It can be seen from the Shoreham Base response spectrum that the energy content lies between 1 Hz and 7 Hz. We have shown (by a very conservative assumption) the first predominant frequency to be 9.3 Hz. For this reason a peak (1.5) acceleration level is conservative.

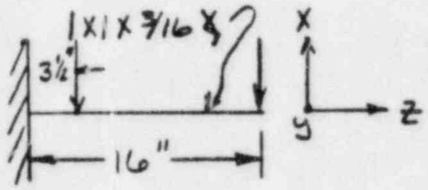
$A_x=A_y=A_z=$ ENV [2% OBE, 3% SSE from S+L generated Shoreham spectra. see A.1]

$A_x=A_y=A_z=0.55\,g's\,(1.5)=0.825\,g's$

* Note – since the motor was conservatively estimated at 100 Lbs, the unit weight will be estimated at 150 Lbs.

Form GQ-3.08.1 Rev. 2

Analytical Model



Weight / Beam = 300Lbs/4 = 75Lbs

Weight / Location = 75/2 = 37.5Lbs

Assumptions: 1.) Assume each of the 4 legs takes 1/4 of the loading.

2.) Assume 37.5Lbs acts at z = 3½" and 37.5Lbs at z = 16".

Loading, Seismic + weight

$F_x = wt(A_x) = 75 Lbs (0.825 g's) = 61.9 Lbs$

$F_y = wt(A_y) = 75 Lbs (0.825 g's) = 61.9 Lbs$

$F_z = wt + wt(A_z) = 75 + 75 Lbs (0.825 g's) = 136.9 Lbs$

$M_x = (F_y/2)(3½) + (F_y/2)(16) = (61.9/2)(3.5) + (61.9/2)(16) = 604 Lbs$

$M_y = M_x$ (by symmetry) = 604Lbs

$M_z = 0.0$ in-lbs

Beam Stress

$$\sigma_t = \frac{F_z}{A} + \frac{M_x}{S_x} + \frac{M_y}{S_y}$$

$$\sigma_t = \frac{136.9\,\text{lbs}}{0.340\,\text{in}^2} + \frac{604\ \text{in-lbs}}{0.044\,\text{in}^3} + \frac{604\ \text{in-lbs}}{0.044\,\text{in}^3}$$

$$\sigma_t = 27{,}836\,\text{psi}$$

$$\tau_v = [F_x^2 + F_y^2]^{1/2} / A = [61.9^2 + 61.9^2]^{1/2} / 0.340$$

$$\tau_v = 257\ \text{psi}$$

$$\sigma_p = \left(\frac{27{,}836}{2}\right) + \left[\left(\frac{27{,}836}{2}\right)^2 + 257^2\right] = 27{,}838\,\text{psi}$$

$$^{*}S_{all} = 0.95\,S_y = 0.95\,(36{,}000\,\text{psi}) = 34{,}200\,\text{psi}$$

$S_{all} > \sigma_p \Rightarrow$ the frame is sound.

* assuming A-36 steel, S+L standard MSS-6.2-D for service level C loading conditions.

Form GQ-3.08.1 Rev. 2

SARGENT & LUNDY
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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.2.14

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Weld Check

The entire structure was welded to the floor. By inspection it can be concluded that the entire assembly will act as a rigid frame when the pump/motor platforms are attached. For this reason and due to the wide weld pattern the welds do not require extensive analysis. They will be adequate to withstand a seismic event.

Form GQ-3.08.1 Rev. 2

SARGENT & LUNDY
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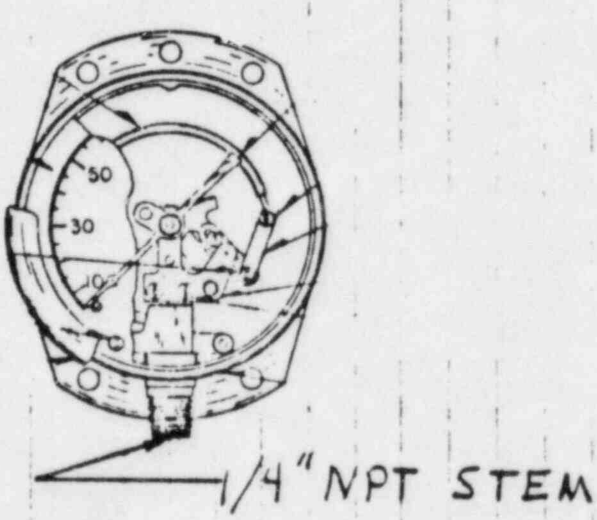
Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client		Prepared by Dave Wright	Date 5/17/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Fuel Oil Pressure Gauge

Log NO. 0803060302



The Fuel oil pressure gauge will not cause the engine to stop or cause failure of the system to start. For this reason the gauges are not essential and need not function. The only mode of failure that could cause a problem would be a rupture of the bourdon tube or housing causing leakage of the oil in the control panel. The leakage rate would be so slow that it would not cause any significant loss of Lubricant. The mounting screws on the Shoreham gauge will be checked for adequacy and the rest of the gauge will be proven adequate by the La Salle (CECO) Analysis. Due to the extreme similarity (even between different mfg.) this will show the adequacy of the gauges for seismic loading.

Form GO-3.08.1 Rev. 2

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Mounting Screw Check

Screw Size - #10 {note- typical mounting screw size for similar gauges. See ashcroft analysis page B.1.44 of this report.}

Material - SA-307 (assumed material, very low grade)

$A_t = 0.0175\ \text{in}^2$ {from "Machinery's Handbook", 12th edition page 1282, UNC screw, tensile Area.}

No. of Screws - 4

Weight - 2 3/4 Lbs {enveloped maximum weight of the gauges shown in the analysis on the following pages.}

Assumption 8 For ease of analysis assume that the offset of the C.G. of the gauge from the bolt centroid is negligable. Use a 10"g" acceleration to make up for neglicting the moments.

Loading

$F = F_x = F_y = F_z = 2.75\ \text{Lbs}\ (10g\text{'s}) = 27.5\ \text{Lbs}$

$\sigma_t = F/4A_t = 27.5\ \text{Lbs}/4(0.0175\text{in}^2) = 393\ \text{psi}$

$\tau = F\sqrt{2}/4A_t = 27.5\ \text{Lbs}\sqrt{2}/4(.0175\text{in}^2) = 556\ \text{psi}$

$\sigma_p = \frac{393}{2} + \sqrt{\left(\frac{393}{2}\right)^2 + 556^2} = 786\ \text{psi} < S_a = 7000\ \text{psi}$ (ASME Appendix I, table I-7.3)

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.2.17

Client	Prepared by ___ Date
Project	Reviewed by ___ Date
Proj. No. ___ Equip. No.	Approved by ___ Date

Marshalltown Figure 24-F
4½ inch Dial Size

Figure 24-F

USAGE—For pressures above atmospheric. Usable on air, oil, gas, water or any other pressure medium which does not attack brass. Install protective syphon when used on steam.

DIAL SIZES—3½", 4½" 6", 8½" and 12", except Fig. No. 24P and Fig. No. 24PC are available in the ½" and 6" sizes only.

BOURDON TUBE—Phosphor bronze.

MOVEMENT—Rugged - bronze bushed - independent mounting.

ACCURACY—Within 1% of total scale range in middle half of scale - 1½% elsewhere.

FIG. NO. 24F

The front flange of the aluminum case has holes for three mounting screws which are covered by the ring when flush mounted in a panel. The edge of the front flange is threaded to receive an aluminum die cast ring on 3½", 4½" and 6" sizes. 8½" and 12" sizes have a sand cast aluminum hinged and pinned ring. Case and ring have oven baked black enamel finish. ¼" male lower back connection is standard.

Molded Case

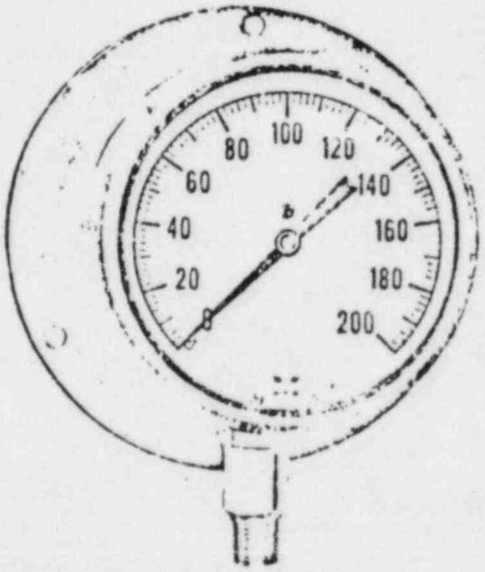
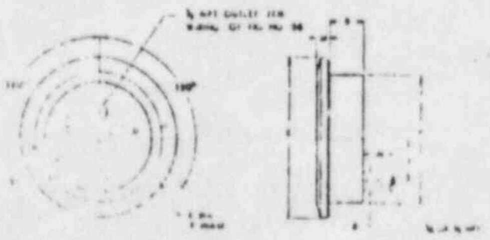


Fig. No. 24

Fig. No. 24F

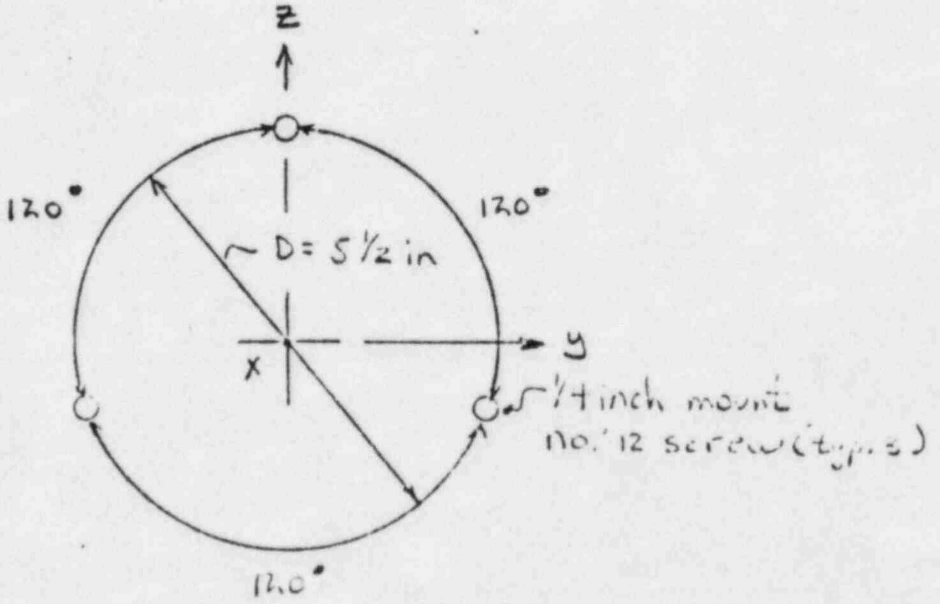
MOLDED CASE -- FLUSH MOUNTING



Gauge Size	A	B	C	D	E	Panel Holes	H ¼ NPT	H ½ NPT	J	K	Panel Cutout	L	4
3½	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]
4½	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
6	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
8½	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
12	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Check of Mounting Screws

The Figure 24-F is panel mounted. The mounting holes are 1/4 inch diameter so assume a #12 mounting screw.



Weight – 2 Lbs } See Phone memo dated 3/27/84 between M. Dixon (Marshalltown) and D. Wright (S+L) attached on pg. B.1.59

Because of the size, shape and weight of the gauge along with the method of mounting it is obvious that the gauge will be seismically rigid ie. $f_1 > 33$ Hz. Zpa levels will be used for seismic loading criteria as follows:

ZpA Horizontal = 1.4 g's
ZpA Vertical = 1.5 g's
{ from CQD(S+L)-009249, Rev. 00 dated 8/10/83 on the subject of Elevated Response Spectrum for EDGS Panel mounted Instruments.

Note – it has been shown in Appendix A that the shoreham spectrum is bounded by the above.

SARGENT & LUNDY
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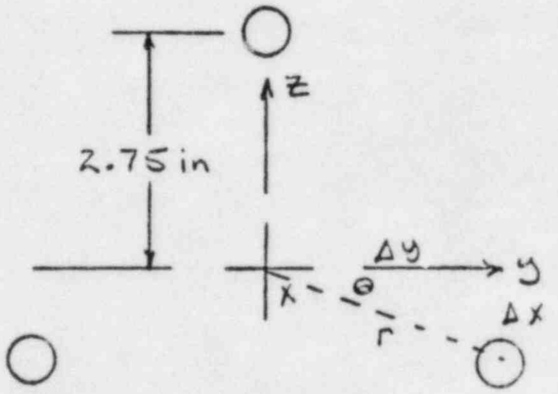
Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.2.19

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Geometric properties of bolt pattern



$r = 2.75 in$, $\theta = 120^\circ - 90^\circ = 30^\circ$

$\Delta x = r \sin\theta = 2.75 \sin 30^\circ = 1.375 in$

$\Delta y = r \cos\theta = 2.75 \cos 30^\circ = 2.38 in$

$$\bar{z} = \frac{\bar{z}_1 A_1 + \bar{z}_2 A_2 + \bar{z}_3 A_3}{A_1 + A_2 + A_3} = \frac{z_1 + z_2 + z_3}{3} = \frac{(2.75 - 1.375) \times 2 + 5.5}{3}$$

from base of gauge

$\bar{z} = 2.75 in =$ center of the circle $D = 5.5 in$

Form GQ 3.08.1 Rev. 2

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. EQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.2.20

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Loading, seismic + Weight

$F_x = Wt(a_x) = 2 Lbs(1.4 g's) = 2.8 Lbs$

$F_y = Wt(a_y) = 2 Lbs(1.4 g's) = 2.8 Lbs$

$F_z = Wt + Wt(a_z) = 2 Lbs + 2 Lbs(1.5 g's) = 5 Lbs$

$M_x = 0.0$ in-lbs (no offset of c.g. from bolt centroid)

$M_y = 0.0$ in-Lbs "

$M_z = 0.0$ in-Lbs "

Nozzle Loading Due to pressure

$F_{x_{\text{pressure}}} = (\text{Pressure})(\text{Area})$

The area will be conservatively based on outside diameter of the 1/4 NPT stem. O.D. = 0.540 from Machinery's Handbook, 12th edition page 1399

Area = Area at stem = $\frac{1}{4}\pi d^2 = \frac{1}{4}\pi(.540)^2 = 0.229 \text{ in}^2$

Pressure = 300 psi (maximum gauge value)

$F_x = 300 \text{ psi}(.229 \text{ in}^2) = 69 \text{ Lbs}$

Note – the tubing will transmit very little seismic loading to the gauge. This is due to its small weight and many supports.

Loading, seismic + weight + Pressure

$F_x = 2.8 \text{ Lbs} + 69 \text{ Lbs} = 72 \text{ Lbs}$

$F_y = 2.8 \text{ Lbs}$

$F_z = 5 \text{ Lbs}$

$M_x = 0.0 \text{ in-lbs}$

$M_y = 0.0 \text{ in-lbs}$

$M_z = 0.0 \text{ in-lbs}$

Bolt Data

Size – #12 { See phone memo dated 3/27/84 between M. Dixon (marshalltown and D. wright (S+L) see page B.1.59 }

Material- assume SAE Grade 1, 2 or 3, SA-307

$\text{Area}_t = 0.0242 \text{ in}^2$ { From Machinery's Handbook, 12th edition page 1282, UNC screws, tensile stress Area }

$\sigma_t = F_x / 3A_t = 72 / 3(.0242) = 992 \text{ psi}$

$\tau_r = [F_y^2 + F_z^2]^{1/2} / 3A_t \quad [2.8^2 + 5^2]^{1/2} / 3(.0242) \quad 79 \text{ psi}$

Check Against Allowable (see pages B.1.13, .13.1 and .13.2)

$^*S_u = 60{,}000 \text{ psi}$

* from ASME B[illegible] code, 1977 edition, section III, table I-7.3 of Appendix I, page 98

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For		Calc. No. CQD-014046
☐ Safety-Related	☐ Non-Safety-Related	Rev. 00 Date 6/1/84
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Client	Prepared by	Date
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Proj. No. Equip. No.	Approved by	Date

For details on symbols see page B.1.13, 1.13.1, 1.13.2

$F_t = 0.3\,S_u$ (for service levels A+B) $= 0.30(60) = 18000$psi

$F_v = 0.124\,S_u$ (") $= 0.124(60) = 7{,}440$psi

$f_t = 992$ psi

$f_v = 79$ psi

For Combined Shear + Tension the following Condition Must be satisfied:

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \le 1.00 \quad \text{so:}$$

$$\frac{992^2}{18{,}000^2} + \frac{79^2}{7440^2} = 0.0032 \le 1.00$$

The bolts are adequate to withstand service level A, B + C loadings.

Inlet Stem

Stem size : 1/4" NPt

inlet hole : 0.16in (see phone memo on page B.1.59)

Tube, Connection Size : 1/4 in O.D. X 0.035in wall
stainless steel TP-304,

Section Modulas of Stem

From Machinery's Handbook, 12th edition,

1/4" NPt : Pipe O.D. : 0.540

Thread Height : 0.0444in

$\text{Min. O.D} = 0.540 - 2(0.0444) = 0.451''$

$\text{Max I.D.} = 0.16\text{in}$

$A = \frac{\pi}{4}(OD^2 - ID^2)$

$A = \frac{\pi}{4}(.451^2 - .16^2) = 0.14\text{in}^2$

$Z = \frac{\pi(OD^4 - ID^4)}{32OD} = \frac{\pi(.451^4 - .16^4)}{32(.451)}$

$Z = 0.009\text{ in}^3$

Section Modulas of Tube

$O.D. = 0.25\text{in}$

$I.D. = 0.25\text{in} - (0.035)2 = 0.18\text{in}$

$\text{Area} = \frac{\pi}{4}(OD^2 - ID^2)$

$= \frac{\pi}{4}(.25^2 - 0.18^2) = 0.024\text{in}^2$

$Z = \frac{\pi(OD^4 - ID^4)}{32OD} = \frac{(.25^4 - .18^4)\pi}{32(.25)}$

$Z = 0.001\text{ in}^3$

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ENGINEERS
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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
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Client		Prepared by	Date
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Proj. No.	Equip. No.	Approved by	Date

Stem Material - Extruded Brass (Phone memo, page B.1.59)

Allowable Stress - 8000 psi (ASME Appendix I, table I-5.4 for all Brass shown, pg. 118 @ 100°F, Section III)

Tube Material - stainless steel, TP-304 (assumed for a conser. analysis, not actual.

Allowable Stress - 18,800 psi (ASME Appendix, table I-7.2 for TP304 stainless, page 81 @ 100°F, Section III)

=

Comparison of strength:

$$A_{stem}\left(\frac{S_{stem}}{S_{tube}}\right) = 0.14 in^2\left(\frac{8,000}{18,800}\right) = 0.059\ in^2$$

$$Z_{stem}\left(\frac{S_{stem}}{S_{tube}}\right) = 0.009\left(\frac{8}{18}\right) = 0.004\ in^3$$

$$A_{stem}/A_{tube} = 0.062\ in^2/.024 in^2 = 2.6$$

$$Z_{stem}/Z_{tube} = .004/.001 = 4.0$$

The stem is 4 times stronger than the tube in bending. The stem will not fail before the tube.

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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
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PAGE B.2.24

Client		Prepared by	Date
Project		Reviewed by	Date
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Marshalltown Figure 23

2 1/2 inch Dial Size

Drawn Steel Case

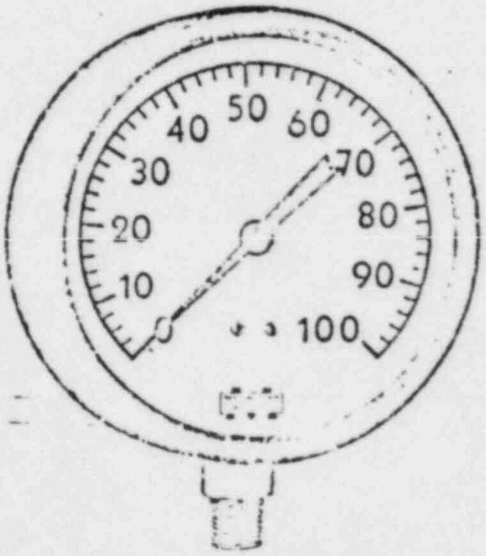


Fig. No. 23

USAGE—For pressures above atmospheric. Usable on air, oil, gas or water or any other pressure medium which does not deteriorate brass. Install protective syphon when used on steam.

DIAL SIZES—2", 2 1/2", 3 1/2" and 4 1/2", except that the 4 1/2" is not available in Fig. Nos. 23B and 23D.

CASE—Drawn steel - phosphatized for rust resistance and finished in oven baked black enamel.

RING—Same as above.

BOURDON TUBE—Phosphor Bronze.

MOVEMENT—Brass - Independent Mounting.

ACCURACY—Within 2% of total scale range in middle half of scale - 3% elsewhere.

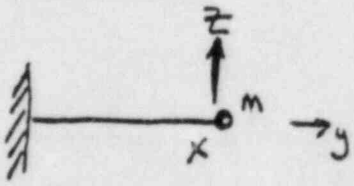
FIG. NO. 23

1/4" male bottom connection is standard. When so specified, a [illegible] bottom connection can be furnished on 2" and 2 1/2" sizes.

Mountings: The Figure 23 gauge is stem Mounted.

weight: 1/2 Lb (phone memo page B.1.59)

Stem Size: 1/4 in Npt (phone memo page B.1.59)



$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$

SARGENT & LUNDY ENGINEERS CHICAGO	Calcs. For		CALC NO. CQD-014046 REV. 00 DATE 6/1/84 PROJ. NO. 6995-00 PAGE B.2.25
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Client		Prepared by	Date
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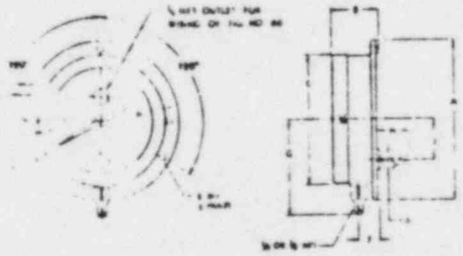
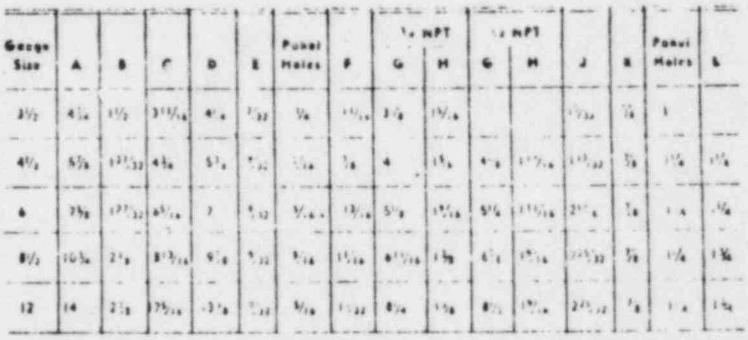
with $K = \frac{3EI}{l^3}$ { from "Marks Standard Handbook for Mechanical Engineers, 8th edition, McGraw-Hill Books, page 5-70 }

where l = length of the stem which will be estimated from the table below.

Gauge Dimensions





Gauge Size	A	B	C	D	E	Panel Holes	F	¼ NPT G	¼ NPT H	½ NPT G	½ NPT H	J	K	Panel Holes	L
3½	[illegible]	1½	3 11/16	[illegible]	7/32	[illegible]	[illegible]	3⅛	[illegible]			[illegible]	[illegible]	1	
4½	[illegible]	1 21/32	4¾	[illegible]	[illegible]	[illegible]	⅞	4	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
6	[illegible]	1 27/32	6 5/16	7	[illegible]	5/16	13/16	5⅛	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
8½	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
12	14	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Stem length = $G-(C/2) = 3\,1/8'' - (3\,11/16'')/2 = 1.28$ in ($3\frac{1}{2}''$ gauge

$4'' - (4\,3/4'')/2 = 1.625$ in ($4\frac{1}{2}''$ gauge

$5\,1/8'' - (6\,5/16'')/2 = 1.97$ in ($6''$ gauge

Since our gauge is $2\frac{1}{2}''$ we will use $l = 1.28$ in :

<u>using the relationship developed previously:</u>

¼ NPT O.D. min (inside of thread) = 0.451 in

= O.D Max = 0.15 in (phone memo page B.1.59)

$$I = \frac{\pi(OD^4 - ID^4)}{64} = \frac{\pi(.451^4 - .15^4)}{64}$$

$$I = 0.002 \text{ in}^4$$

$E_{ss} = 15 \times 10^6$ psi { From "Strength of Materials", 2nd edition by John Cernica, copyright 1977 by Holt, Rineholt + Winston, page 463, mean value }

$$K = \frac{3EI}{\ell^3} = \frac{3(15\times10^6)(0.002)}{(1.28)^3} = 42,915 \text{ in/Lb}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{42,915}{(1/2 \text{ Lb}/386.4)}} = 917 \text{ Hz},$$ the gauge is rigid

loading, weight + seismic

$F_x = Wt(a_x) = 0.5 \text{ Lbs}(1.4 g's) = 0.7 \text{ Lbs}$

$F_y = Wt(a_y) = 0.5 \text{ Lbs}(1.4 g's) = 0.7 \text{ Lbs}$

$F_z = Wt(a_z) + Wt = 0.5(1.5) + 0.5 = 1.25 \text{ Lbs}$

(Note - apply weight to cause bending moment to be as large as possible.)

$M_x = F_z G = 1.25(3\,1/8) = 3.91$ in-lbs

$M_y = 0.0$ in-lbs

$M_z = F_x G = 0.7(3\,1/8'') = 2.19$ in-lbs

Loading, Pressure

$$F_{y_p} = (\text{Pressure})(\text{Area})$$

$$= (300\text{psi})\left(\frac{\pi}{4}(0.540_{in}^{2})\right) = 69\ \text{Lbs}$$

$$\sigma_t = (F_y + F_{y_p})/A_{st} + \frac{M_x + M_z}{Z_{st}}$$ (from previous Marshalltown 1/4in NPT stem)

$$\sigma_t = \frac{(0.7\text{lbs} + 69\text{lbs})}{0.14\text{in}^2} + \frac{2.91 + 2.19}{.004\text{in}^3} = 1176\text{psi}$$

$$\tau_v = \left[F_x^2 + F_z^2\right]^{1/2}/A_s = \left[0.7^2 + 1.25^2\right]^{1/2}/0.14\text{in}^2 = 10.\text{psi}$$

$$\sigma_p = \frac{\sigma_t}{2} + \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau_v^2} = \frac{1176}{2} + \sqrt{\left(\frac{1176}{2}\right)^2 + 10^2}$$

$$\sigma_p = 1176\text{psi}$$

S = 8000 psi (ASME Appendix I, table I-B.4, same value for all Brass listed, Section III)

$S > \sigma_p \Rightarrow$ acceptable.

SARGENT & LUNDY ENGINEERS CHICAGO	Calcs. For ☒ Safety-Related ☐ Non-Safety-Related	CALC NO. CQD-014046 REV. 00 DATE 6/1/84 PROJ. NO. 6995-00 PAGE B.2.28

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip No.	Approved by	Date

Bourdon Tube Analysis

Bourdon tube information was not available due to the proprietary nature of the data. This was the policy expressed by MR. MiLt Dixon of Marshalltown (see ref. 3 phone memo.). We can however draw some conclusions from the information provided by Marsh gauges. The actual stress in the bourdon tube of the Marsh gauge was found to be: (seismic only)

$$\sigma_{seis} = 377 psi$$

This level is much smaller than the stress in the tube due to internal pressure. For this reason the seismic loading is assumed to have a negligible effect on the gauge integrity and the gauge is qualified for use.

Note - for a more detailed bourdon tube analysis see the Ashcroft bourdon tube check in this analysis. The deflection due to normal use was 113 times greater than the deflection due to seismic loading

Fatigue Analysis, Figure 23, 24-F

Engine opperating Vibrations

A conservative estimate of the g' stress due to Engine opperational vibration is the following:

$$\sigma_{p_{\text{eng. vibe}}} = \sigma_{p_{\text{seismic}}} \left[\frac{\text{maximum Acc, Eng. vibe}}{\text{Min Acc, seismic}} \right]$$

Minimum Seismic = 1.4 g's

Maximum Engine Vibration = 1.0 g's {From CQD-012628, Rev. 00 dated 2/22/84 in titled Response of an EMD 645 Engine to Impulsive and Steady state loads}

The critical area of the gauges (envelope both) is the stem of the Figure 23 gauge;

$$\sigma_{p_{\text{seis}}} = 1176 \text{ psi}$$

$$\sigma_{p_{\text{Eng Vibration}}} = 1176 \text{psi} \left(\frac{1.0}{1.4} \right) = 840 \text{psi}$$

Since both values are well below the endurance limit of 12,000psi the gauges are qualified for use on the engines.

(ASME B+PV code, Appendix I, section III table I-9.1 fig I-9.3, $S_y = 18,000$ psi)

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Calcs. For

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CALC NO. CQD-014046
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PAGE B.2.30

Client
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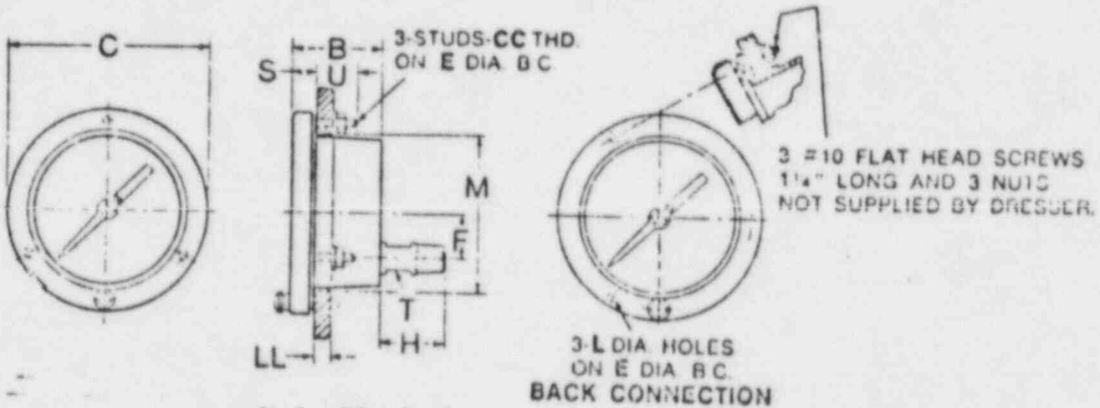
Proj. No. Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

Ashcroft 45-1377A,S-02B

4½ inch Gauge

CASE TYPE 1377 (OPEN FRONT)/1377-S (SOLID FRONT)



CASE TYPE 1377 (OPEN FRONT)

Dial Size Inches	B	C	E	F	H	L	M	S	T	U	CC	LL	Weight (lbs.)
4½	[illegible]	6[illegible]	5[illegible]	1[illegible]	1[illegible]	—	4[illegible]	[illegible]	[illegible]	[illegible]	#10-24	[illegible]	2[illegible]
6	2[illegible]	7[illegible]	7	2[illegible]	1[illegible]	—	6[illegible]	[illegible]	[illegible]	[illegible]	¼-20	[illegible]	2[illegible]
8½	2[illegible]	10[illegible]	9[illegible]	2[illegible]	1[illegible]	—	9	[illegible]	[illegible]	[illegible]	¼-20	[illegible]	5[illegible]
12	3	13[illegible]	13	2[illegible]	1½	.196	12½	[illegible]	[illegible]	[illegible]	#10	[illegible]	15[illegible]

CASE TYPE 1377 (SOLID FRONT)

Dial Size Inches	B	C	E	F	H Max.	H Min.	M	S	T	U	CC	LL	Weight (lbs.)
4½		6[illegible]	5[illegible]	1[illegible]			4[illegible]				10-[illegible]		2[illegible]
6	2[illegible]	7[illegible]	7	2[illegible]	1[illegible]	1[illegible]	6[illegible]	[illegible]	[illegible]	[illegible]	¼-20	[illegible]	[illegible]
8½		10[illegible]	9[illegible]	2[illegible]			[illegible]				¼-20		

Note - since there may be both open and solid front items in the plant, both will be addressed via an enveloping approach.

CALC NO. CQD-014046
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35 ✓

Maximum pressure at which a gauge is continually operated should not exceed 75% of full scale range.

'O ORDER A GAUGE

Select:

1. Case type number — Table A
2. Dial size — Table A
3. Bourdon System (ordering code) — Table B
4. Connection: Location — Table A: Size — Table 5
5. Mounting accessory or variation (if required) — Table A
6. Pressure Range — Table C on page 12
7. Accessories and optional features — pages 15-20

Example:

1279-S	4½"	TA	Back ½ NPT	w/1278 Ring	0/2000 psi
phenol-solid front	4½"	AISI 316 st. st. system	Back Conn. ½" NPT	with 1278 Ring	0/2000 psi

TABLE A — CASE SELECTION

Case Type Number	Dial Size	Case Style	Case: Material Finish	Ring: Style Material Finish	Mounting and Connection
1279	4½, 6, 8½	Open Front	Phenol Black	Snap Stainless Steel	Stem – Lower or Back Surface – Lower or Back Flush – Back: order 1278 ring (see page 18)
1279(*)SL** 1279(*)SH**	4½	Solid Front	Phenol Black	Threaded (reinforced) polypropylene)	
1320*	4½	Open Front	Stainless Steel Polished	Bayonet Lock Stainless Steel Polished	Stem – Lower or Back Surface–Lower: Specify XBF Surface- Back: Specify XBQ Flush – Back: Specify XUC
1377	4½, 6, 8½, 12	Open Front	Aluminum: 4½, 6, 8½ Cast Iron: 12" Black epoxy coated	Hinged Steel Black epoxy coated	Flush – Back connection only
1377-S	4½, 6, 8½	Solid Front			
1379	4½, 6, 8½, 12	Open Front	Aluminum Black epoxy coated	Threaded Aluminum: 4½, 6 Bronze: 8½, 12 Black epoxy coated	Stem – Lower or Back Surface – Lower or Back Flush – Back
1379-S	4½, 6, 8½	Solid Front			
2462	6	Solid Front	Polypropylene (fiberglass reinforced) Black	Bayonet Lock Polypropylene Black	Stem – Lower or Back Surface – Lower or Back: Specify XBF Flush – Back: Specify XBQ

*Can be Liquid Filled - see page 13 ** - Lower conn only. For field conversion order Liquid Filled Conversion Kit

TABLE B — BOURDON SYSTEM SELECTION

Ordering Code	Bourdon Tube and Tip Material (all joints TIG welded except "A")	Socket Material	Tube Type: Drawn or Bored	Pressure Range (PSI)	NPT Connection (3)	
					Open Front	Solid Front
A	Grade A Phosphor Bronze Tube — Brass Tip, Silver brazed	Brass	Drawn	12/1000	¼	½
B	AISI 4130 alloy steel	AISI 1019 steel	Drawn	12/1000	¼	½
D	AISI 4130 alloy steel	AISI 1019 steel	Bored	1000/20,000	½	½
			Drawn (spiral)	30,000/100,000 †	—	¼ high pressure
R	AISI 316 stainless steel	AISI 1019 steel	Drawn	12/1000	¼	½
RT	AISI 316 st. st. tube AISI 1019 steel tip	AISI 1019 steel	Bored	1000/20,000	½	½
S	AISI 316 stainless steel	AISI 316 stainless steel	Drawn	12/1000	¼	½
TA	AISI 316 stainless steel	AISI 316 stainless steel	Bored	1000/20,000	½	½
			Drawn (spiral)	30,000/80,000 †	—	¼ high pressure
P	K Monel	R Monel	Drawn	12/1000	¼	½
Q	K Monel	R Monel	Bored	1000/20,000	½	½
			Drawn (spiral)	30,000/80,000 †	—	¼ high pressure
F(1)	E-Brite 26-1	E-Brite 26-1	Drawn	100/1000 (2)	—	½ only
FT(1)	E-Brite 26-1	E-Brite 26-1	Bored	2000 (2)	—	½ only

(1) available with 4½" solid front cases only

†30,000 psi and up — available only in 6" & 8½" Type 1377-1379 solid front cases.

(2) other ranges on application.

(3) Optional connections available.
½ NPT where ¼ NPT is standard
¼ NPT where ½ NPT is standard

Mounting Screw Check

Screw Data:

Screw size : #10 {From Dresser Industries, Ashcroft gauge bulletin DU-1, Duragauge Press. Gauge Data found on B.1.44 of this report.}

Number : 3

Material : SA-307 (assumed, SAE grades 1, 2 or 3)

A_t : 0.0175 in^2 {From Machinery's Handbook, 12th edition page 1282 for UNC screw, tensile stress Area.}

Gauge Wt : $2\frac{3}{4}$ Lbs (env[open face + solid face])

The gauge is seismically rigid as can be seen by the method of mounting and by no extended masses being present. ZPA levels will be used.

Loading, Seismic + Weight

$F_x = wt(a_x) = 2.75 Lbs (1.4 g's) = 3.85 Lbs$

$F_y = wt(a_y) = 2.75 Lbs (1.4 g's) = 3.85 Lbs$

$F_z = wt(a_z) = 2.75 Lbs (1.5 g's) = 4.125 Lbs$

$M_x = M_y = M_z = 0.0$, No extended masses

Pressure loading

F_{x_p} = Pressure (stem Area)

Stem Size – 1/2" NPT (solid front, 1/4" npt open front)

Max. O.D. = 0.840 in (standard table)

$A = \frac{\pi}{4} OD^2 = \frac{\pi}{4}(.840^2) = 0.554 in^2$

F_{x_p} = 300 Lbs (.554 in²) = **166** Lbs

$\sigma_t = (Fx + Fxp)/3A_t = (3.85 + 166)/3(.0175) = 3235$ psi

$\tau_v = [F_y^2 + F_z^2]^{1/2}/3A_t = [3.85^2 + 4.125^2]^{1/2}/3(.0175) = 107$ psi

Check Against Allowable (treating Stem as Pipe)

Treating the stem as a pipe, From ASME B+PV codes, Section III, Devision 1, subsection ND-3000 paragraph ND-3652.1 page 133

$S_{OL} \leq 1.2 S_h$

S_{OL} = Occasional Load Stress, This is the stress due to combined loads.

Sh = allowable stress at "T".

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Calcs. For		Calc. No. CQD-014046
		Rev. 00 | Date 6/11/84
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Client	Prepared by	Date
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Proj. No. 6995-00 Equip. No.	Approved by	Date

$$S_{OL} - {}^{*}\sigma_p = \frac{\sigma_t}{2} + \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau_r^{\,2}} = \frac{3235}{2} + \sqrt{\left(\frac{3235}{2}\right)^2 + 107^2}$$

$$\sigma_p = 3239 \text{ psi}$$

$S_h = 7000$ psi [for SA-307 from ASME B+PV code, Section III, Appendix I, table I-7.3 page 99.]

$\therefore S_h > \sigma_p \Rightarrow$ the stem is adequate

*Note - $S_{OL} = \dfrac{P_{max} D_o}{4 t_n} + \dfrac{0.75 i (M_A + M_B)}{Z}$

σ_p is more conservative due to the method of determining the loads and the high seismic coefficients use.

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Inlet Stem check

1/4" Npt Data (from Marshalltown Fig. 24-F Analysis)

$A_{\substack{min\\stem}} = 0.14 \text{ in}^2$

$Z_{\substack{min\\stem}} = 0.009 \text{ in}^3$ (assuming inlet hole @ tube I.D.)

$A_{tube} = 0.024 \text{ in}^2$

$Z_{tube} = 0.001 \text{ in}^3$

1/2" Npt Data with 3/8 x 0.035 in thick tube

1/2" Npt O.D. = 0.840 in

thread depth = 0.05714 in

{ From Machinery's Handbook, 18th edition pages 1399 for 1/2" NPt }

$\text{Min O.D.} = \text{O.D.} - 2t_d = .840 - 2(.05714) = 0.726 \text{ in}$

$\text{Max I.D.} = \text{tube I.D. (assumed)} = 3/8 - 2(.035) = 0.305 \text{ in}$

$$A = \frac{\pi}{4}(OD^2 - ID^2) = \frac{\pi}{4}(0.726^2 - 0.305^2) = 0.341 \text{ in}^2$$

$$Z = \frac{\pi(OD^4 - ID^4)}{32(O.D.)} = \frac{\pi(.726^4 - .305^4)}{32(.726)} = 0.036 \text{ in}^3$$

S = 8000 psi (ASME Appendix I, table I-8.4, for all Brass shown pg. 118) Section III.

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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
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tube Material – TP 304 stainless (worst case)

$$A_{tube} = \frac{\pi}{4}(OD^2 - ID^2) = \frac{\pi}{4}(3/8^2 - .305^2) = 0.037 in^2$$

$$Z_{tube} = \frac{\pi(OD^4 - ID^4)}{32 OD} = \frac{\pi(3/8^4 - .305^4)}{32(3/8)} = 0.003 in^3$$

$S_{tube} = 18{,}800 psi$ (ASME B+PV code, table I-7.2 of Appendix I, section III, TP-304 stainless pg. 84)

Strength Comparison:

1/4" – ok. per Marshalltown Fig. 24-F analysis

1/2" stem check

$$A_{stem}\left(\frac{S_{stem}}{S_{pipe}}\right) = 0.341\left(\frac{8}{18.8}\right) = 0.145 in^2$$

$$Z_{stem}\left(\frac{S_{stem}}{S_{pipe}}\right) = 0.036\left(\frac{8}{18.8}\right) = 0.015 in^3$$

Area stem/Area tube = 0.145/0.037 ≅ 4

Zstem/Ztube = 0.015/0.003 = 5

The stem is 5 times stronger than the tube, the tube will fail before the stem.

Form GQ 3.08.1 Rev. 2

Calcs For

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Bourdon Tube Check :

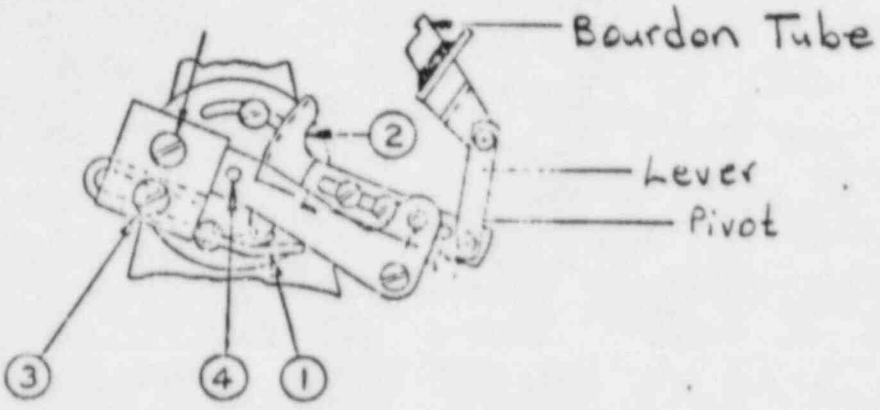
Although no specific details were available as to the thickness and cross sectional properties it will be shown that the contribution to the tube stress due to seismic loading will be negligible. This will be done by developing a relationship between tube deflection and the rated pressure. This will be done by making use of the gauge movement ratios shown below (Table taken from Engineering Data, Ashcroft Gauges Maintenance and repair form no. 250-1353-E.)

Gauge Movement Ratios

"Gauge sizes to 3½" inclusive – 12.1 Ratio
"Quality" Gauges, All Sizes – 12.1 Ratio
"Duragauges" – As listed below in Table

Duragauge Type in Terms of Tube Material	Maximum Dial Graduation P.S.I.	Movement Ratio		
		12:1	16:1	20:1
All Materials [illegible] Tubes)	1000	x		
4½" Size – (Series T [illegible])	Over 1000 to 20,000 inclusive		x	
6" Size and Larger [illegible] Tubes	Over 1000 to 14,[illegible] inclusive		x	
6" Size and Larger – Beryllium Copper Only	100 to 500 inclusive	x		
	Over 500 to 20,000 inclusive		x	
6" Size and Larger – All Materials except Beryllium Copper	15,000 and 20,000 inclusive			x
6" Size 1332	[illegible] to 100,000 inclusive		x	

Gauge "Works"



ASHCROFT DURAGAUGE GAUGE SINGE RETARD
(Ashcroft Form # 250-1353-E)

Gauge Operation: When pressure is input into the curved bourdon tube it attempts to straighten. As it deflects the attached lever pulls on the driving arm. At the other end of the driving arm there is a gear (2) driving another gear (4) which turns the dial. Making use of this information a relative movement can be obtained. We cannot find the exact deflection of the tube without more information but we can approximate the deflection for comparison purposes.

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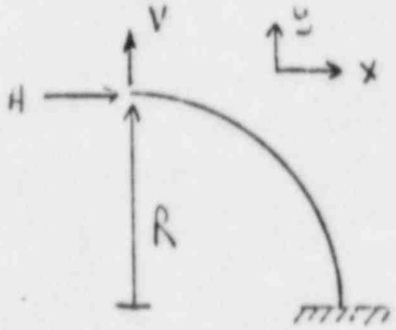
For purposes of comparison assume $l = 0.5$ in

$\delta = l \sin 22.5° = 0.5 \sin 22.5° = 0.19$ in total tip deflection

For comparison purposes the Marsh J-4858 Bourdon tube dimensions will be used



$R = 4\frac{1}{2}$ in. $/2$

$R = 2.25$ in assumed because gauge is $4\frac{1}{2}$ inch (Ashcroft)

{From "Formulas For Stress and Strain, 5th edition By Roark + Young copyright 1975, McGraw Hill, page 215}

$$\frac{\delta_y}{3} = \frac{(\pi/4)VR^3 + \frac{1}{2}HR^3 + M_oR^2}{EI}$$ ⇒ assumed to be 1/3 of the deflection since other sections not included.

$$\frac{\delta_x}{3} = \frac{\frac{1}{2}VR^3 + (\frac{3}{4}\pi - 2)HR^3 + (\pi/2 - 1)M_oR^2}{EI}$$

$E = 15 \times 10^6$ psi {Strength of Materials, 2nd edition By John Cernica, copyright 1977 by Holt, Rinehart and Winston}

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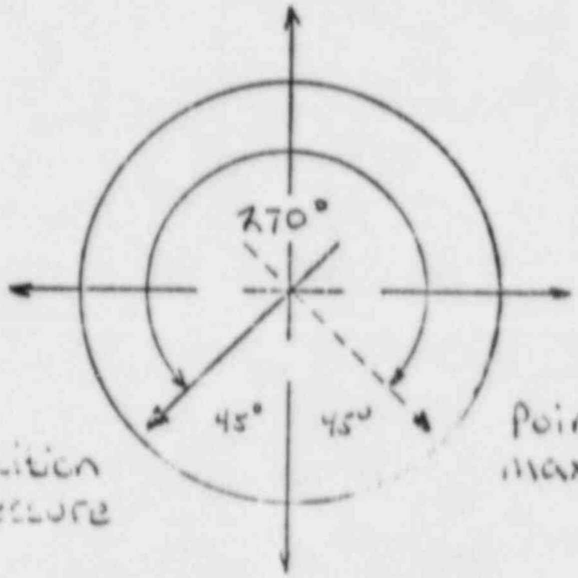
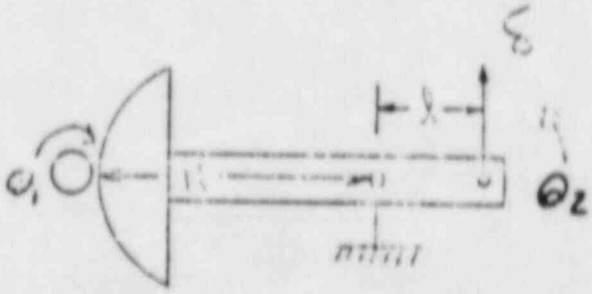
Dial Arrangement



Pointer Position at 0 pressure

Pointer position at max pressure

The small gear must turn 270° for an entire scale deflection. Simplifying the works:



$\frac{\theta_2}{\theta_1} = 12:1$ ratio

so for a full deflection of the scale (270°) the lever arm will deflect:

$$(270^\circ)\left(\frac{1}{12}\right) = 22.5^\circ$$

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H seismic = Wt(a_x) = 0.041 Lbs(1.4 g's)(2) = 0.115 Lbs

V seismic = Wt(a_z) = 0.041 lbs(1.5 g's)(2) = 0.123 Lbs

Note – No torsional deflection will occur, tube fixed at both ends. The above weights will be doubled to account for larger tube.

$I = 6.47 \times 10^{-4} in^4$ (from previous section)

$$\delta_y = \frac{(\frac{\pi}{4})(.123)(2)^3 + \frac{1}{2}(.115)(2^3)}{(15 \times 10^6)(6.47 \times 10^{-4})} = 1.27 \times 10^{-4}(3) = 3.84 \times 10^{-4} in$$

$$\delta_x = \frac{\frac{1}{2}(.123)(2)^2 + (\frac{\pi}{4} - \frac{2}{\pi})(.115)(2^3)}{(15 \times 10^6)(6.47 \times 10^{-4})} = 8.446 \times 10^{-5}(3) = 2.534 \times 10^{-4} in$$

$$\delta_{total} = (\delta_x^2 + \delta_y^2)^{1/2} = (3.8 \times 10^{-4^2} + 2.84 \times 10^{-4^2})^{1/2} = 0.0004567 in$$

$$\frac{\delta_{normal\ operation}}{\delta_{seismic}} = \frac{0.19 in}{0.0004567} = 416 \text{ times more.}$$

The seismic stress (deflection) will have little if any adverse affects on the tube. The tube is adequate.

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Fatigue Analysis

For the purpose of fatigue evaluation only the point of maximum stress will be addressed. This will be the brass stem of the 1004 at a $\sigma_p = 3122$ psi

Engine Operational Vibration

For ease of analysis the engine operating stress (due to vibration) will be conservatively estimated by the relationships

$$\underset{\text{eng vibe}}{\sigma_p} = \underset{\text{seis+pressure}}{\sigma_p}\left[\frac{\text{max "g" level vibe}}{\text{min "g" level seis}}\right]$$

The engine vibration levels are obtained from CQD-012628 rev.00 dated 2/22/84 "Response of An EMD 645 E-4 Engine".

$a_x = 0.95$ g's , $a_y = 0.85$ g's , $a_z = 1.00$ g's

The minimum seismic "g" level was 1.4 g's in the Horizontal. Using the appropriate levels from above:

$$\underset{\text{Eng Vibe}}{\sigma_p} = \underset{\text{seis}}{\sigma_p}\left[\frac{1.00\text{ g's}}{1.4\text{ g's}}\right] = 3122(1/1.4) = 2230\text{ psi}$$

Since the socket is threaded a fatigue strength reduction factor of 4 (as required by ASME B+PV codes for bolts section III, Division 1, Subsection NB; paragraph NB-3230.)

Loading Combination	σ_p	n	N
seismic + Pressure + vibe	(3122 + 2230) 4 21,408 psi	60 cycles	18,342 cycles
Pressure + Vibe	21,408 psi	(5500-60) 5440 cycles	18,342 cycles
Vibration	(2230)4 8920 psi	2.8×10^9 cycles	∞

From ASME B+PV code, section III, Appendix I table I-9.1, fig I-9.3
$S_y = 18$ ksi

$$N_1 = 10000\left(\frac{20000}{10000}\right)^{\left[\log\left(\frac{24.5}{21.408}\right)/\log\left(\frac{24.5}{21.0}\right)\right]} = 18{,}342 \text{ cycles}$$

$N_2 = N_1 = 18{,}342$ cycles

$N_3 = \infty$, below the endurance limit of 12,000 psi

$$\text{Usage factor} = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} = \frac{60}{18{,}342} + \frac{5440}{18{,}342} + \frac{2.8\times10^9}{\infty} = 0.30 < 1.0$$

MEMORANDUM OF
TELEPHONE CONVERSATION

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Date: 3/15/84

Time: 11:30 am

Person Called: David Strayhorn (Name) of March Instrument Co. (Company)

Person Calling: David Wright (Name) of Sargent & Lundy (Company)

Project: LaSalle/Clinton **Project No.** 6548-20/4[illegible]-32

Subject Discussed: Design Dimensions and Criteria on the Model J-4858 Gauge

Summary of Discussion, Decisions and Commitments:

Dave (March) stated that the J-4858 gauge is designed and built to conform to ANSI B.40.1 standards. He also stated that the volume of the bourdon tube for the 4858 can vary by as much as 25%. The following design dimensions were provided by Dave at my request. (See attachment)

DW/cq

cc M. Molaghugh - 30
J. Sinnappan/P. Hlepas - 30
AEM/APD/DEM - 15
CQD File - 30

D. Strayhorn - March Instrument

Signature

lc: CQD-012825

Manufacturer/Model Number - Marsh/J-4858
Mounting - stem mounted from the bottom
Total Weight (Dry) - 5-3/4 ounces
Stem Size - 1/4 in. NPT
Stem Material - Copper Alloy
Stem Inlet Hole Diameter - 1/8 in. min., 3/16 in. max.
Bourdon Tube Material - Copper Alloy
Bourdon Tube Thickness - 1/16 in. to 1/8 in.
Bourdon Tube Arc. Length - 1-11/16 in.
Bourdon Tube X-Sect - approximate circle, 7/16 in. O.D.
Dial Size - 2-1/2 in.
Maximum Rated Pressure - 300 psi

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MEMORANDUM OF
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CALC NO. CQD-014046
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Date: March 27, 1984

Time: 10:30 am

Person Called: Milt Dixon (Name) of Marshalltown (Company)

Person Calling: David Wright (Name) of S&L (Company)

Project: LaSalle/Clinton Project No. 6548-30/ 4536-32

Subject Discussed: Marshalltown Pressure Gauge, Figure 23 and Figure 24-F with Dial Ranges 0 - 100psi, 0 - 300psi, 0 - 160psi and 0 - 240psi.

Summary of Discussion, Decisions and Commitments:

The following information was provided by Mr. Dixon:

Figure 23	Figure 24-F
Approximate Wt. 1/2 lbs.	2 lbs.
Stem Size 1/4 in.	1/4 in.
Stem hole I.D. 0.15 in.	0.16 in.
Mounting Screw N/A	0.23 in. (hole size)
Stem Material Extruded Brass	Forged Brass
Bourdon Tube Material Phosphor Bronze	Phosphor Bronze

Mr. Dixon stated that any information regarding the size, x-section or thickness of the bourdon tube was considered proprietary information and could not be released.

DW/eg

cc: M. Dixon - Marshalltown Mfg.
S. A. Ahmad - 30
AEM/APD/DW - 30

David Wright
Signature

File CQD-012106 - 30

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Client		Prepared by David Wright	Date 5/17/84
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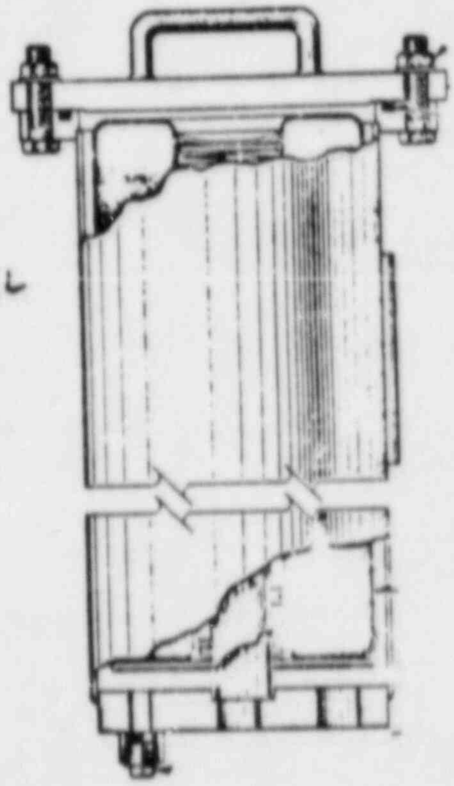
FUEL OIL TRANSFER SYSTEM FILTER

LOG NO 0803060502



MFG: COMMERCIAL FILTERS

MODEL: 1750-105A

THE FUEL OIL TRANSFER FILTERS FOUND ON THE SHOREHAM (LILCO) DIESELS ARE IDENTICAL STRUCTURALLY TO THE LASALLE LUBE OIL SOAKBACK FILTERS, THE ONLY DIFFERENCE BEING THE FILTER ELEMENT. SINCE THEY WERE SHOWN TO BE ADEQUATE FOR THE LASALLE DIESELS, THEY ARE ALSO ADEQUATE FOR THE SHOREHAM DIESELS. The only significant difference between the two is the mounting arrangement. The Shoreham filters are mounted through holes in the radiator seperation wall.

Check of The Body Stress

$I_{\text{section}} = 33.72 \text{ in}^4$ (from previous analysis, (S+L) CQD-010330 Rev. 00)

Weight = 155 Lbs (conservative estimate, Previous analysis (S+L) CQD-010330, Rev. 00)

Pressure = 50 psi (assumed, this is the pressure at the circulating pump, From EMD list of parameters for 20-645E4 engine.

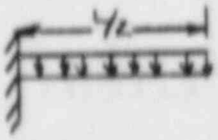
Pressure Stress

$$\sigma_L = \frac{PR}{2t} = \frac{(50\,\text{psi})(7.25/2)}{2(0.25)} = 363\ \text{psi}$$

$$\sigma_t = \frac{PR}{t} = \frac{(50\,\text{psi})(7.25/2)}{(0.025)} = 725\ \text{psi}$$

weight + seismic stress

model of filters



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Weight = 155 Lbs/2 = 77.5 Lbs

$\omega_{weight} = 77.5 \text{ Lbs}/(32\tfrac{3}{4} \div 2) = 4.73 \text{ Lbs/in}$

The seismic loads will be determined using the static coefficient analysis outlined in IEEE-344(1975) (i.e. peak of RRS multiplied by 1.5).

Env(OBE 2%, SSE 3% Horizontal, vert) = 0.38g's (1.5) = 0.57g's
(see A.1, shoreham base spectrum)

$\omega_{seismic} = 4.73 \text{ Lbs/in} (0.57 g\text{'s}) = 2.70 \text{ Lbs/in}$

$\omega_{seis} + \omega_{wt} = 4.73 + 2.70 = 7.43 \text{ Lbs/in}$

$$M_{seis+wt} = \frac{\omega L^2}{2} = \frac{(7.43)(32\tfrac{3}{4} \div 2)^2}{2} = 996 \text{ in-Lbs}$$

Note - this will be in 2 directions so:

$$\sigma_t = 2\left(\frac{Mr}{I}\right) = 2\left(\frac{996 \text{ in-Lbs} (7.25/2)}{33.72 \text{ in}^4}\right) = 214 \text{ psi}$$

$$\tau_v = \frac{2\ \omega L/2}{\pi(r_o^2 - r_i^2)} = \frac{2(7.43)(32.75/2)}{\pi(3.625^2 - 3.375^2)} = 43 \text{ psi}$$

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$\sigma_{Long} = 214 psi + 363 psi = 577 psi$

$\sigma_{tan} = 725 psi$

$\tau_{r} = 43 psi$

$$\sigma_{p_{max}} = \frac{(577+725)}{2} + \sqrt{\left(\frac{725-577}{2}\right)^2 + 43^2} = 737 psi$$

$^{*}S_a = 12,600 psi < 737 psi \Rightarrow$ acceptable

* Note - Allowable stress from ASME B&PV codes, section III, table I-7.1 of Appendix I, page 70 for SA-36 steel @ 200°

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I. Soakback Filters

Log Number		Mfg. Model Number
0303160700	(Lasalle 1+2)	EMD(G.M.)-8379120
0503160700	(Lasalle 3	"
C201160700	(Clinton 1+2)	"
C202160700	(Clinton 1+2	"
0601160700	(Clinton 3)	"

All of the soak back filters are of the same type and manufacturer. It can be seen that there are no extended masses on the filters (see photo and that all of them (Clinton + LaSalle) are mounted in the same location (Acc. Rack) and have the same type of mounting bracket. Because the filter has no extended masses, and since the body is very strong compared to the brackets, they will be the critical areas to be checked. Due to the conservative nature of the analysis and the small external pipe size the nozzle loads are assumed to be negligible.

Form GQ-3.08.1 Rev. 2

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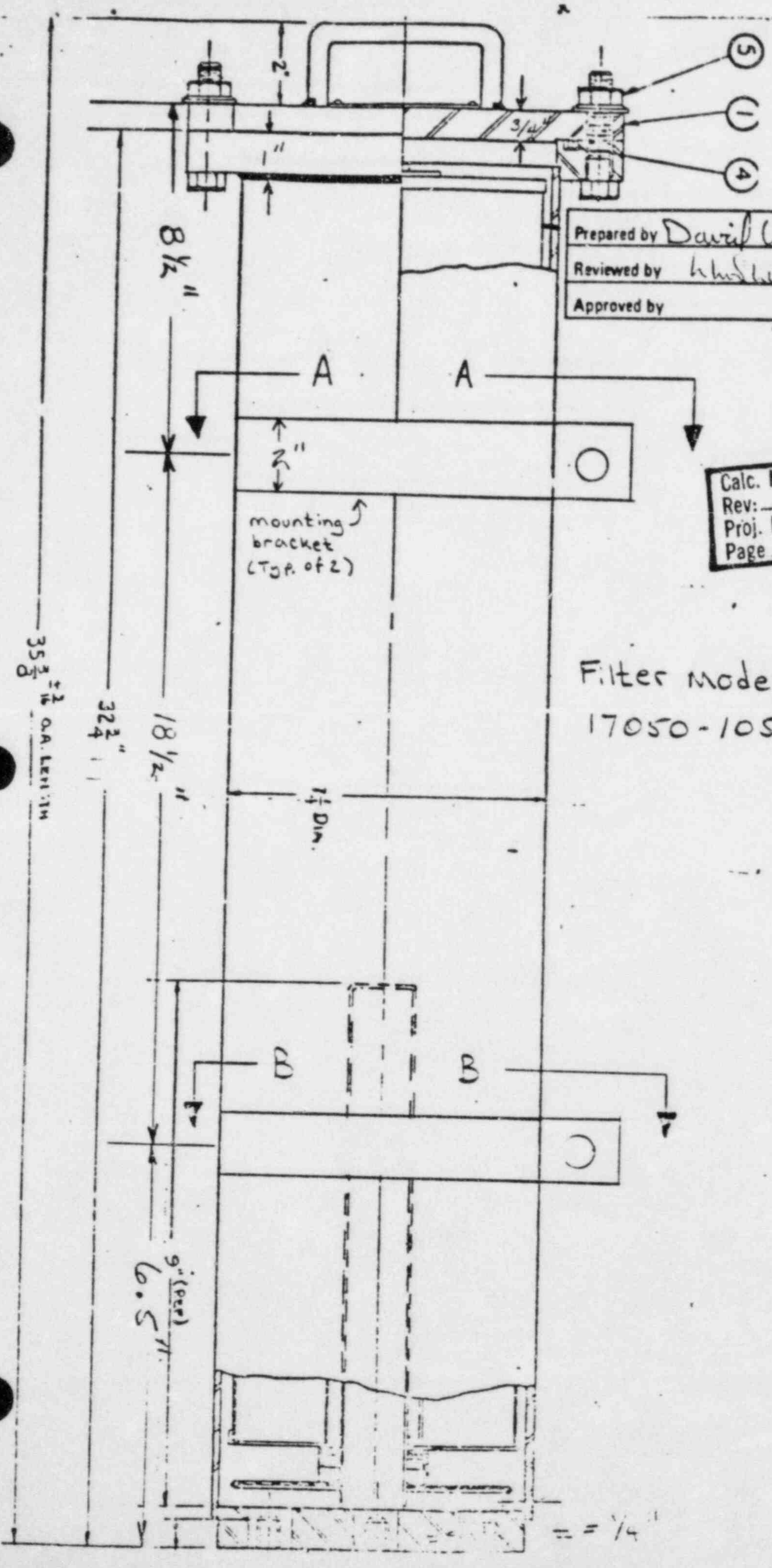

Calc. No: CQD
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Filter model number
17050-105P

Maximum weight

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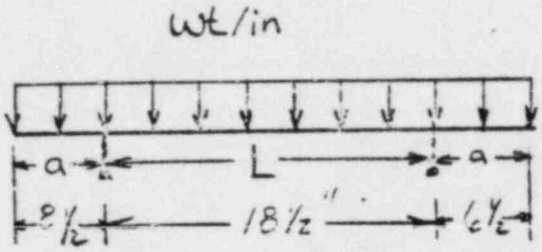
Soakback Filter Dynamic Characteristics

Filter Body Frequency



Note - the deflection formula used to determine the natural frequency of the body was derived for equal overhang distance. The overhang on the supports are approximately equal and the long dimension is used for conservatism.

$$\delta_{end} = \frac{Wa}{24EI}(L^3 - 6a^2L - 3a^3)$$

{from Design of Welded Structures by Blodgett, copyright J.F Lincoln, page 8.1-20}

$$\delta_{\mathbb{C}} = \frac{WL^2(5L^2 - 24a^2)}{384EI}$$

"

$$I = \frac{\pi}{4}(R_o^4 - R_i^4)$$

{Formulas For Stress and Strain by Roark + Young, 5th edition copyright McGraw Hill, page 66}

$R_o = 7.25''/2 = 3.625 in$

$R_i = 3.625'' - 0.25'' = 3.375 in$

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$$I = \frac{\pi}{4}\left(3.625^4 - 3.375^4\right) = 33.72 \text{ in}^4$$

$$E = 29 \times 10^6 \text{ psi} \quad (\text{assume A-36 steel})$$

Estimation of the weight

$$\text{Weight} = [\text{head Weight} + \text{base weight} + \text{body weight} + \text{Element Weight} + \text{Oil Weight}]$$

$$\text{head Weight} = \forall\rho = (\pi)(1/4)(8.4)^2(0.75)(.283 \text{ Lbs/in}^3)$$
$$= 11.96 \cong 12 \text{ Lbs}$$

$$\text{base Weight} = \forall\rho = (\pi)(1/4)(6.75^2)(1\text{in})(.283 \text{ Lbs/in}^3)$$
$$= 10.13 \cong 10 \text{ Lbs}$$

$$\text{body Weight} = \forall\rho = (\pi)(1/4)(7.25^2 - 6.75^2)(32.125\text{in})(.283)$$
$$= 49.98 \cong 50 \text{ Lbs}$$

$$^{*}\text{Liquid Weight} = \forall\rho = \pi(1/4)(6.75^2)(32.125)(62.4 \text{ Lbs/ft}^3)(1\text{ft}^3/1728\text{in}^3)$$
$$= 41.51 \cong 42 \text{ Lbs}$$

*assume water as the fluid, it is heavier than oil.

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Filter Element weight = 15 Lbs (assumed, the filter is a paper element)

Total Weight ≅ [12 + 10 + 50 + 42 + 15] = 129 Lbs

Total Weight + 20% margin for security ≅ 155 Lbs

wt/in length = 155 Lbs/33.5 in = 4.6 Lbs/in

$$\delta_{end} = \frac{(4.6)(8.5)[18.5^3 - 6(8.5)^2(18.5) - 3(8.5^3)]}{24(29 \times 10^6)(33.72)} = -5.9 \times 10^{-6} \text{ in}$$

$$\delta_{℄} = \frac{(4.6)(18.5)^2[5(18.5)^2 - 24(8.5^2)]}{384(29 \times 10^6)(33.72)} = -9.5 \times 10^{-8} \text{ in}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta_{max}}} = \frac{1}{2\pi}\sqrt{\frac{386.4}{5.9 \times 10^{-6}}} = 1288 \text{ HZ}$$

The body is very rigid and will be modeled as a rigid link with mass.

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FUEL OIL SUCTION STRAINERS AND FUEL OIL TRANSFER STRAINERS

(LOG NO 0803060500, 0803060501)

THE FUEL OIL SUCTION STRAINER MARVEL ENGINE CO MODEL NO. 1520800 IS ACCEPTABLE BASED ON THE SIMILIARITY OF THE QUALIFIED COMMERCIAL FILTER MODEL B5 FUEL STRAINER PER CQD FILE NO. CQD-010330 REV. 00. BASED ON THE MOUNTING AND SIZE OF THE MARVEL FUEL STRAINER (SEE PHOTOGRAPHS 24 & 25) SHOWS SIMILIARITY WITH THE COMMERCIAL FILTER. IN ADDITION THE OPERATING TEMP + PRESS. ARE ALSO SIMILIAR THEREFORE WE CAN CONCLUDE THAT LOG 0803060501 IS adequate for use.

THE FUEL OIL TRANSFER STRAINER (LOG NO 0803060502) IS PIPE SUPPORTED LIKE THE QUALIFIED PAGET STRAINER. IN ADDITION, THE SPANS OF PIPE THAT ATTACH TO THE STRAINER ARE SHORT TO THE NEAREST SUPPORT AND THE WEIGHT OF STRAINER IS NEGLIBLE (SEE PHOTOGRAPH 23) THEREFORE, BY ENGINEERING JUDGEMENT AND SIMILIARITY THE FUEL OIL TRANSFER STRAINER (LOG NO. 0803060502) IS ACCEPTABLE. NOTE, THE QUALIFIED PAGET STRAINER CAN BE FOUND IN CQD FILE NO. CQD-010330 REV 00.

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

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REV. 00 DATE 6/1/84
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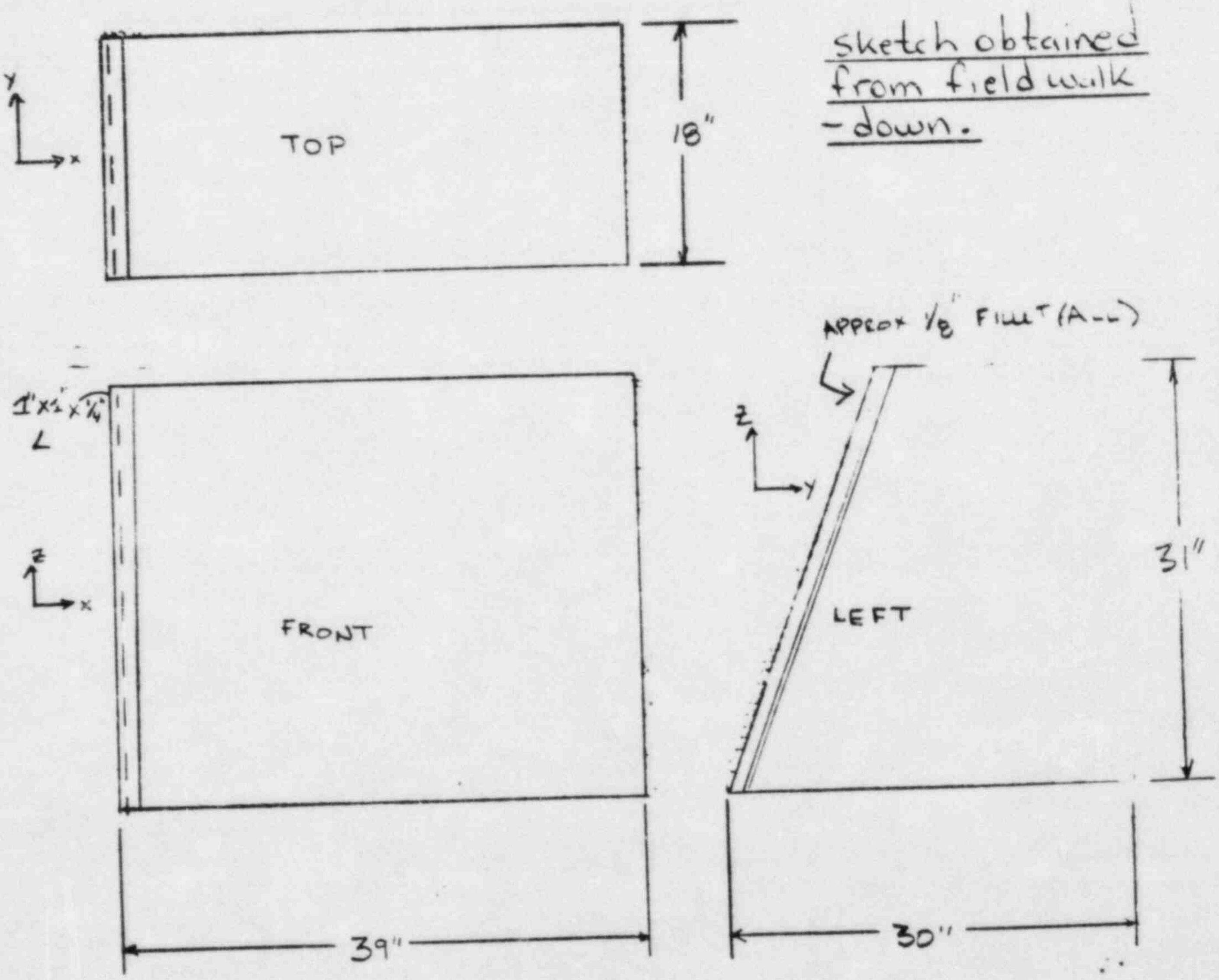
DISEL FUEL DAY TANKS

LOGNO 0803190100

sketch obtained from field walk-down.



THE FUEL DAY TANK IS HOUSED IN BY A THICK [illegible] GA E STEEL WITH 1"x1"x1/4" ANGLES RUNNING ALONG THE FREE EDGES WELDED TO HOUSING.

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BASED ON THE FIELD OBSERVATION OF THE ATTACHMENT OF THE FUEL DAY TANKS TO THE UNIT HOUSING & RADIATOR COMPARTMENT HOUSING BY ENGINEERING JUDGEMENT THE STRUCTURAL INTEGRITY OF THE WELDS WILL BE MAINTAINED. IT WAS OBVIOUS THAT THE AMOUNT OF WELD WILL PREVENT HORIZONTAL AND VERTICAL MOVEMENT, IN ADDITION THE PRESSURE IN THE TANK IS VERY LOW THEREFORE STRESSES ON THE TANK SHELL (INTERNALLY) ARE LOW. BELOW IS SUPPORTING DOCUMENTATION TO strengthen the STATEMENTS ABOVE.

VOLUME OF TANK = LENGTH × AREA

$= 39'' \times \frac{1}{2} (18'' + 30'') 31'' = 29016 \text{ IN}^3$

DENSITY OF FUEL OIL = $65 \frac{\text{LB}}{\text{FT}^3}$ ASSUMED

WT OF FUEL OIL $= 29016 \text{ IN}^3 \times 65 \frac{\text{LBS}}{\text{FT}^3} \times \frac{1 \text{ FT}^3}{1728} = 1092 \text{ LBS}$

STATIC LOADS

$F_X = 0$
$F_Y = 0$
$F_Z = 1092 \text{ LBS}$
$M_X = 1092 (9) = 9{,}828 \text{ IN LBS}$
$M_Y = 1092 (\frac{39}{2}) = 21{,}294 \text{ IN LBS}$

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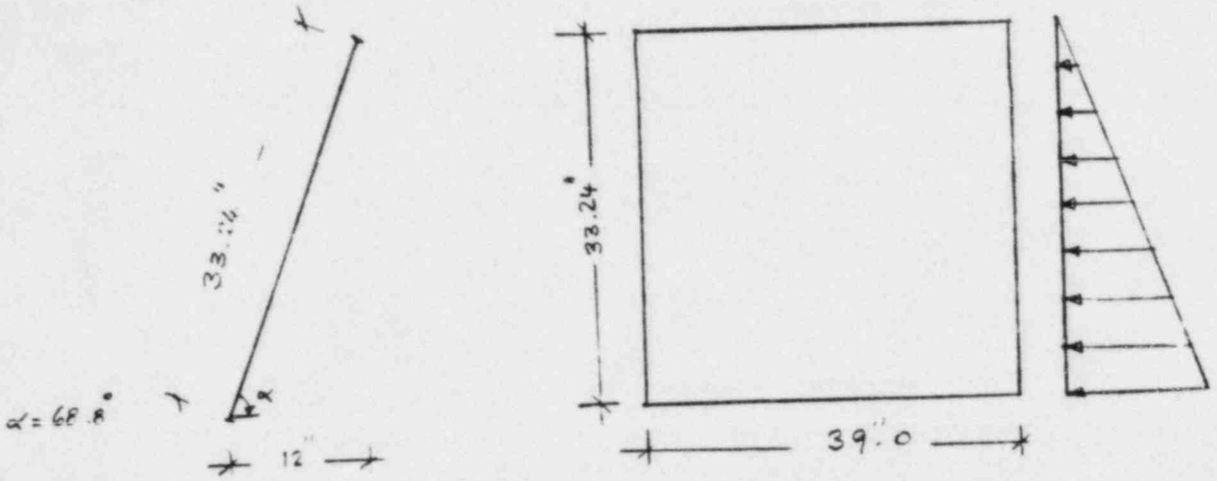
Check of Plate Stresses



1) From the Fluid Pressure

$$\gamma = \frac{624}{(12)^3} = 0.0361 \ \#/In^3$$

$$\frac{39}{33.24} = 1.17$$

$$M_{Fluid} = 0.0338 \times (33.24)^2 \times 0.0361$$

$$= 1.348 \ \#\text{-In.}$$

$$S = \frac{1.0 \times (0.125)^2}{6} = 0.0026$$

$$\sigma_{Fluid} = \frac{1.348}{0\ 0026} = 531 \text{ psi}$$

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2) From the dead weight

$q = 0.283 \times 0.125 \times \cos\alpha$

$= 0.283 \times 0.125 \times 0.36$

$= 0.0127 \ \#/in^2$

$M_{weight} = 0.0336 \times (33.24)^2 \times 0.0127$

$= 0.474 \ \#\text{-}In$

$$\sigma_{weight} = \frac{0.474}{0.0026} = 182.4 \text{ psi}$$

3) Final Stress

$\sigma_{Total} = 531 + 182.4 = 713.4$ psi

Maximum ordinate of Resp. Spec. Curve : ~ 0.9 g.

$\sigma_{max} = 1.8 \times 1.5 \times 713.4 = 1926$ psi

*Sa = 12,600psi > 1926 psi => adequate

* Note - allowable from ASME B+PV code, Section I, 1977 edition of table I-7.1 from Appendix I for SA-36 steel @ 200°F

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Calcs. For LOG NO 08030608
& 0803060900

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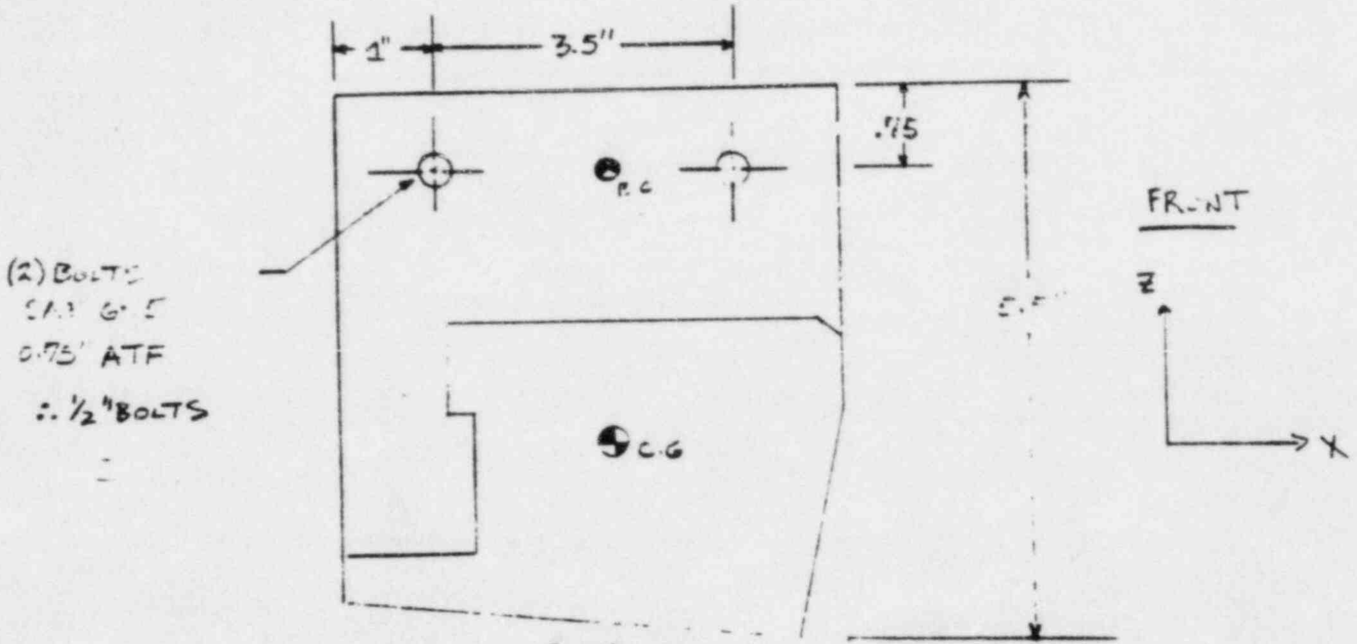
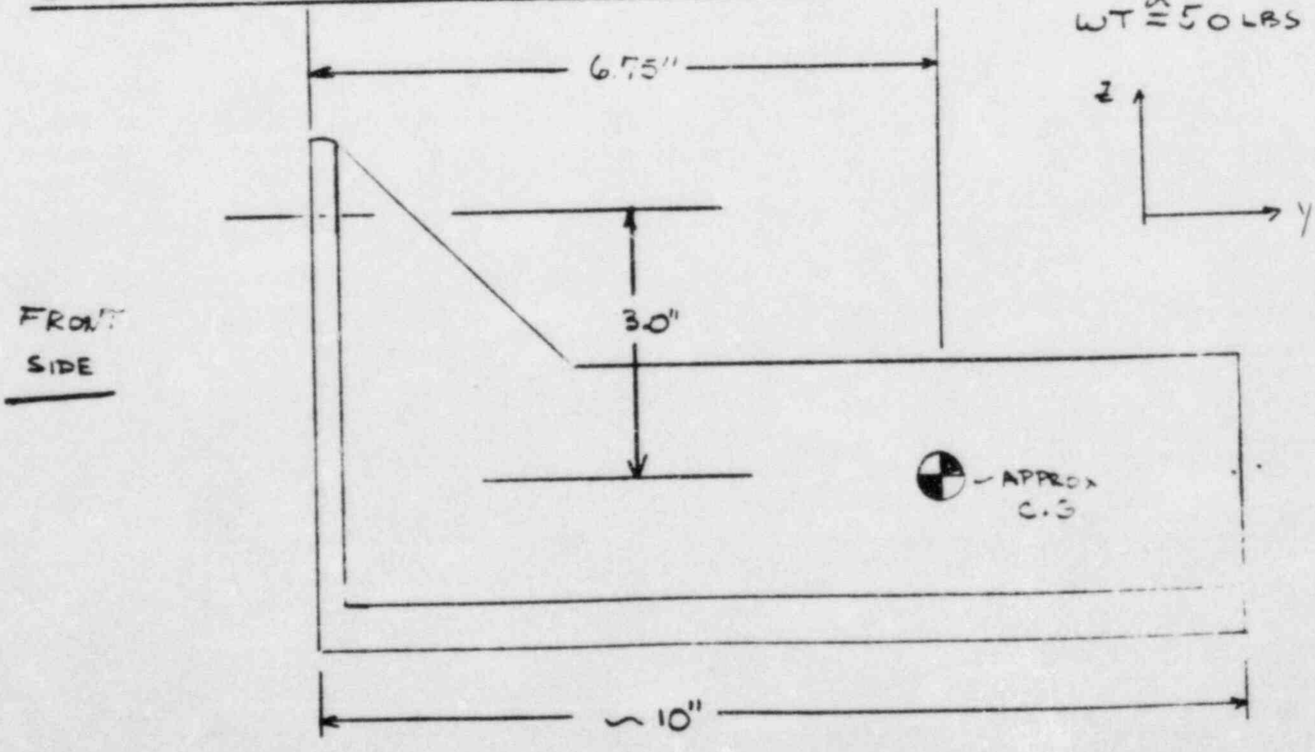
FUEL OIL MANIFOLD ASSEMBLY WTG
(LOG NO 0803060800, 0803060900)



sketch obtained from field walkdown

WT ≅ 50 LBS



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Calcs. For

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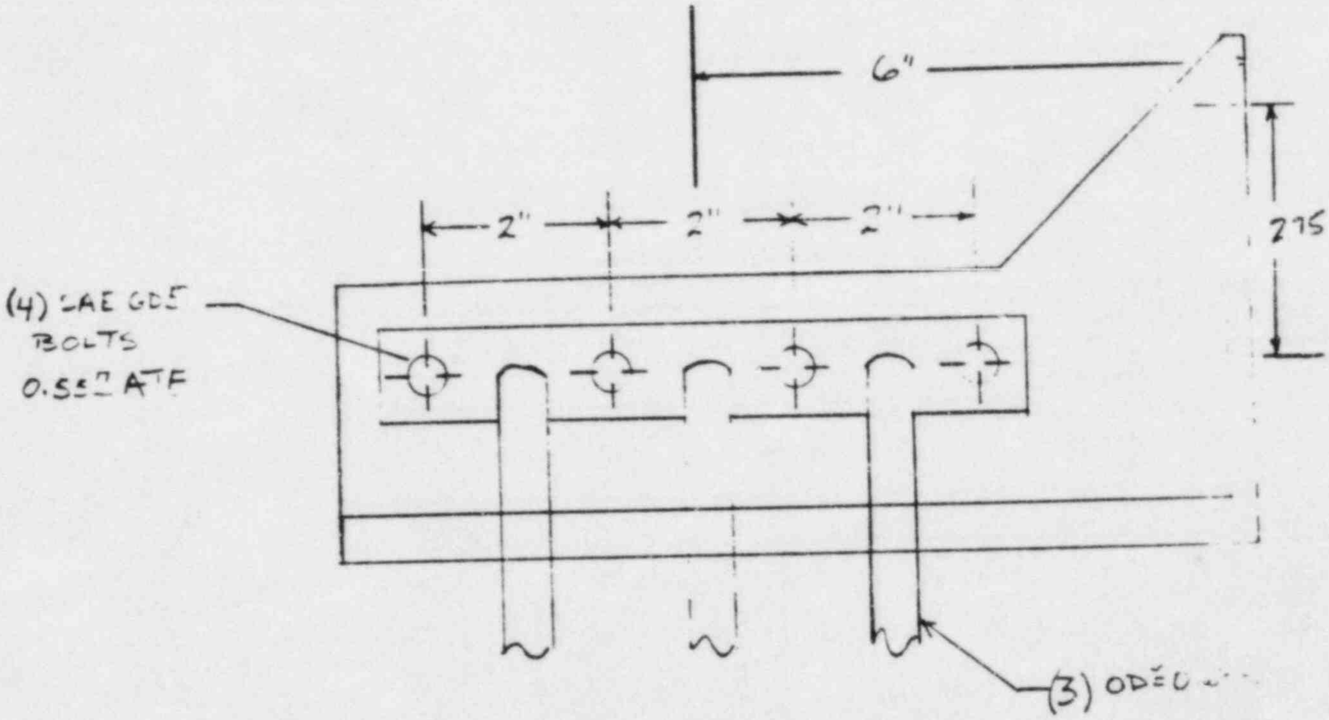
FUEL OIL MANIFOLD ASSEMBLY

REAR SIDE



Sketch obtained from field walkdown

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Calcs. For

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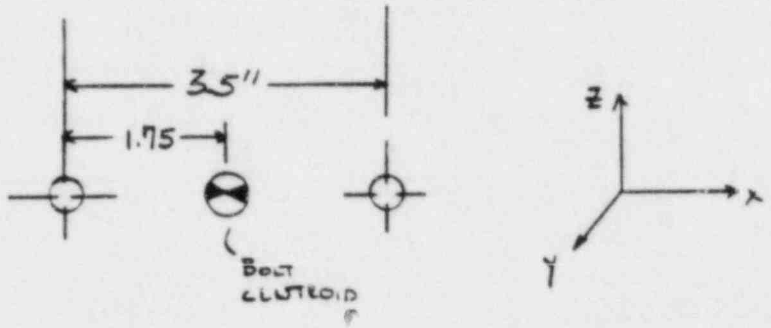
BOLT PATTERN



BOLTS ARE ½" SAE GD 5 (SA-449)

$S_a^* = 23{,}000$ psi BOLTS < 1"
$S_u^* = 120{,}000$ psi

BOLT AREA = $\frac{\pi}{4}(.5)^2 = 0.196$ IN^2

TOTAL BOLT AREA = $2 \times 0.196 = 0.392$ IN^2

$I_{zz} = 2(1.75)^2(0.196) = 1.20$ IN^4
$I_{xx} = 0$
$J = I_{xx} + I_{zz} = 1.20$ IN^4

*note - from ASME B+PV code, Section III, 1977 edition, table I-7.3, page 98 of Appendix I for SA-449 bolting material, O.D. < 1"

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Calcs. For

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DEAD WT LOADS

$F_Z = 50$ LBS (ASSUMED - CONSERVATIVE)

NOZZLE LOADS

FROM THE REAR SIDE OF THE FUEL MANIFOLD THERE ARE 3 SMALL PIPES LEADING OUTWARD WITH [illegible] 10 FT OF SPAN TOTAL TO THE ENGINE BASE. THESE LINES WILL NOT PRODUCE SIGNIFICANT LOADING ON THE MANIFOLD HOWEVER WE WILL CONSERVATIVELY ASSUME ALL 3 LINES REPRESENTING ONE 1 INCH LINE AND WE WILL USE THE LOADS GIVEN IN CQD-LP-03 APPENDIX H P. 18 TABLE 1 FOR CLASS 2 & 3 EQUIPMENT "Nozzle Loads". EQUIP. DRY WT < 1 KIP SERVICE LEVEL C,

$$F_R = \sqrt{Fx^2 + Fy^2 + Fz^2} = 190 \text{ LBS}$$

$$M_R = \sqrt{Mx^2 + My^2 + Mz^2} = .10 \text{ KIP.FT}$$

$$.10 \text{ KIP FT} \times \frac{12 \text{ IN}}{\text{FT}} \times \frac{1000 \text{ LBS}}{\text{KIP}} = 1200 \text{ IN LBS}$$

$$\therefore Fx = Fy = Fz = 190/\sqrt{3} = 110 \text{ LBS}$$

$$Mx = My = Mz = 1200/\sqrt{3} = 693 \text{ IN-LBS}$$

SINCE THE LOADS ABOVE ARE VERY CONSERVATIVE WE WILL ASSUME THAT THEY ACT AT THE BOLT CENTROID

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SARGENT & LUNDY
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CHICAGO

Calcs. For

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SEISMIC LOADS

$F_x = 50 \text{ LBS } (2.8g) = 140$

$F_y = 50 \text{ LBS } (2.8g) = 140$

$F_z \quad 50 \text{ LBS } (7.1g) = 355$

$M_x = F_y(z) + F_z(y) = 140(3) + 355(6.75) = 2816.25 \text{ IN-LBS}$

$M_y = F_x(z) + F_z\cancelto{0}{(x)} = 140(3) = 420 \text{ IN-LBS}$

$M_z = F_x(y) + F_y\cancelto{0}{(x)} = 140(6.75) = 945 \text{ IN LBS}$

note – conservatively assume a horizontal acceleration of 2.8 g's in both x and y and a vertical acceleration of 7.1 g's in z. (see Appendix A for spectrum comparison).

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THE NORMAL ENGINE VIBRATION LOADS DUE TO NORMAL OPERATING CONDITIONS CAN BE OBTAINED BY USING THE TEST DATA PER *REF. (BELOW) CONSERVATIVE ENG. VIBE LOADS ARE AS FOLLOWS

$x = .95g$
$y = .85g$
$z = 1.0g$

NORMAL ENGINE VIBRATION LOADS

Fx = 50 LBS (.95g) = 47.5 LBS

Fy = 50 LBS (.85g) = 42.5 LBS

Fz = 50 LBS (1.0g) = 50 LBS

Mx = 42.5 LBS (3)" + 50 LBS (6.75)" = 465 IN LBS

My = 47.5 LBS (3)" = 142.5 IN LBS

Mz = 47.5 LB (6.75)" = 320.625 IN LBS

* REF: Values taken from S+L file CQD-012628, Rev. 00, dated 2/22/84 in titled "Response of An EMD 645 Engine And Selected Accessories to Impulsive and Steady State Loads".

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Calcs. For		
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TOTAL LOADS

Fx = 110 + 140 + 47.5 = 297.5 LBS

Fy = 110 + 140 + 42.5 = 292.5 LBS

Fz = 50 + 110 + 355 + 50 = 565 LBS

Mx = 693 + 2816.25 + 465 = 3971.25 IN LBS

My = 693 + 420 + 142.5 = 1255.5 IN LBS

Mz - 693 + 945 + 320.625 = 1958.625 IN LBS

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Calcs. For

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TENSION DUE TO FY

$$\frac{F_Y}{2} = \frac{292.5}{2} = 146.25 \text{ LBS}$$

2 Active Bolts

TENSION DUE TO MZ

$$MZ = 1 F_B (3.5)$$

$$T_B = \frac{1958.625}{3.5} = 557.6 \text{ LBS}$$

1 Active Bolt

TENSION DUE TO MX

$$MX = 2 F_e (5.5)$$

$$F_B = \frac{3974.25}{2(5.5)} = 361.3 \text{ LBS}$$

2 Active Bolts

PRYING ACTION

SHEAR DUE TO FZ

$$\frac{F_Z}{2} = \frac{565}{2} = 282.5 \text{ LBS}$$

2 Active Bolts

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SHEAR DUE TO FX

1 Active Bolt

$F_x = 297.5$ LBS

TENSILE STRESS

$$\sigma_t = \frac{146.25}{.392} + \frac{559.6}{.196} + \frac{361.3}{.392} = 4150 \text{ PSI}$$

SHEAR STRESS

$$\tau_{max} = \frac{292.5}{.392} + \frac{297.5}{.196} = 2239 \text{ PSI}$$

Allowable Stress

From ASME B+PV code, 1977 edition with summer 79 addendum, section III, article 2460 of Appendix XVII;

$F_t = 0.3 S_u = 0.3(120,000 psi) = 36,000 psi$ (service levels A+B)

$F_v = 0.124 S_u = 0.124(120,000 psi) = 14,880 psi$ (")

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for Combined Tensile and Shear Loading the following condition must be met:

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \leq 1.00$$

$$\frac{4150^2}{36,000^2} + \frac{2239^2}{14,880^2} = 0.036 \leq 1.00 \Rightarrow$$ the bolts are adequate.

NOTE: SINCE THE STRESS IS LOW WE CAN CAN CONSERVATIVELY ASSUME THAT STRESSES DUE TO OPERATIONAL LOADING WILL NOT HAVE A SIGNIFICANT IMPACT TO OVERSTRESS THIS SYSTEM.

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Calcs. For LOG NO 080206110

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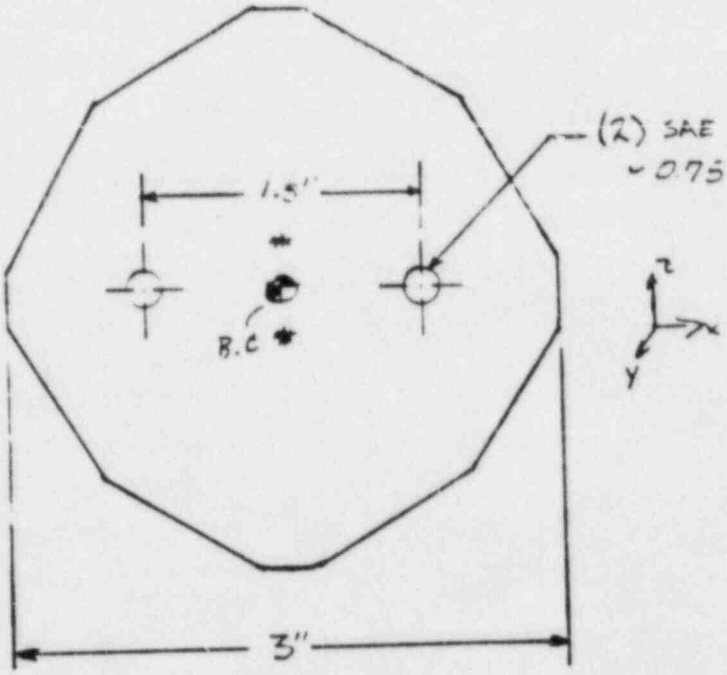
FUEL OIL CROSSOVER MANIFOLD ASSEMBLY

(LOG NO. 080206110)



Sketch obtained from field walkdown

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DEAD WT

ASSUME THE DIFFUSER ACCOUNTS FOR 50 LBS DEAD WT (CONSERVATIVE)

$F_z = 25$ LBS

SEISMIC LOADS

$Fx = 25$ LBS $(2.8g) = 70$ LBS

$FY = 25.0$ LBS $(2.8)g = 70$ LBS

$FZ = 25.0^{\#}(7.1) = 177.5$ LBS
$MX = 177.5(3) = 532.5$ IN-LBS
$MZ = 70\ (3) = 210$ IN-LBS
$MY = 0$

NOZZLE LOADS

TAKEN FROM CQD-LP-03 APPENDIX H, "Equipment Nozzle loads"
CLASS 2 & 3 EQUIP. DRYWEIGHT < 1 KIP
SERVICE LEVEL C.

$\left[F_x^2+F_y^2+F_z^2\right]^{1/2} = 190$ LBS $190/\sqrt{3} = 110$ LBS $= F_x = F_y = F_z$

$\left[M_x^2+M_y^2+M_z^2\right]^{1/2} = .10$ KIP FT $= 1200$ IN LBS $1200/\sqrt{3} = 693$ IN LBS
$693^{\text{IN LBS}} = MX = MY = MZ$

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Calcs. For

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ENGINE VIBRATION LOADS

LOADS DUE TO NORMAL OPERATING CONDITION ON THE FUEL CROSSOVER CAN BE CONSIDERED NEGLIGIBLE SINCE THE PIPE RUNS ARE SMALL (WT IS SMALL) AND THERE ARE NO SIGNIFICANT EXTERNAL MASSES. THE ENGINE ACCELERATION LEVELS WOULD BE LESS THAN THE SEISMIC ACCEL LEVEL AND SINCE THE SEISMIC LOADS ARE SMALL THEN THE ENGINE OPERATING LOADS ARE EVEN SMALLER.

SEISMIC
$a_x = 2.0$ g's
$a_y = 2.0$ g's
$a_z = 7.1$ g's
} assumed levels, see app. A for actual spectra values.

ENGINE
$a_x = .95$ g's
$a_y = .95$ g's
$a_z = 1.02$ g's
} obtained from CQD-012623, Rev. 00 dated 2/22/84 for force of AN EMD 645 Engine.

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Calcs. For

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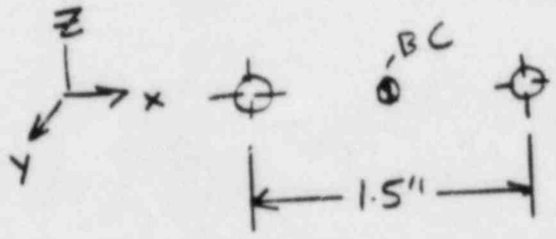
TOTAL LOADS

FX = 70 + 110 = 180LBS
FY = 70 + 110 = 180 LBS
FZ = 25 + 177.5 + 110 = 213.5 LBS.
MX = 532.5 + 693 = 1225.3 INLBS
MY = 693 INLBS
MZ = 210 + 693 = 903 INLBS

BOLT PATTERN



(2) SAE GD 5
(SA 449 [illegible]
1/2" BOLTS

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Calcs. For

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TENSION DUE TO Fy

$\frac{F_y}{2} = \frac{180}{2} = 90$ LBS $\qquad \frac{90\text{ LBS}}{.392\text{ IN}^2} = 229.6$ PSI

TENSION DUE TO MZ

1 ACTIVE BOLT

$MZ = 1\ F_B\ (1.5')$

$\therefore \frac{903}{1.5} = 602$ LBS $\qquad \frac{602}{.196} = 3071.5$ P

TENSION DUE TO MX (PRYING ACTION)

$Mx = 2\ F_B\ (1.25)$

$\frac{1225.3}{2(1.25)} = 490$ $\qquad \frac{490}{.392} = 1250$ PSI

ASSEMBLY CAP IS ≈ 2.5" O.D.
∴ ½(2.5) = 1.25 IN

SHEAR DUE TO FZ

$\frac{FZ}{2} = \frac{213.5\text{ LBS}}{2} = 106.75$ LBS $\qquad \frac{106.75}{.392} = 272.3$ PSI

SHEAR DUE TO FX = 180 LBS $\qquad \frac{180}{.196} = 918.4$ PSI

1 ACTIVE BOLT

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TENSILE STRESS

σ_t 29.6 + 3071.5 + 1250 = 4551.1 PSI

SHEAR STRESS

$\tau_{MAX} = 272.3 + 918.4 = 1190.7$ PSI

Allowable Stress

From ASME B+PV code, 1977 edition with summer 79 addendum section III, Article 2460 of Appendix XVII;

$F_t = 0.3 S_u = (.3)(\overset{*}{1}20{,}000) = 36{,}000$ psi (service levels A+B)

$F_v = 0.124 S_u = (.124)(\overset{*}{1}20{,}000) = 14{,}880$ psi (")

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \le 1.00 \Rightarrow \frac{4551.1^2}{36{,}000^2} + \frac{1191^2}{14{,}880^2} = 0.022 < 1.00$$

The bolts are adequate.

* Note - from ASME B+PV code, 1977 edition, section III, table I-7.3 of Appendix I for SA 449 steel, D ≤ 1"

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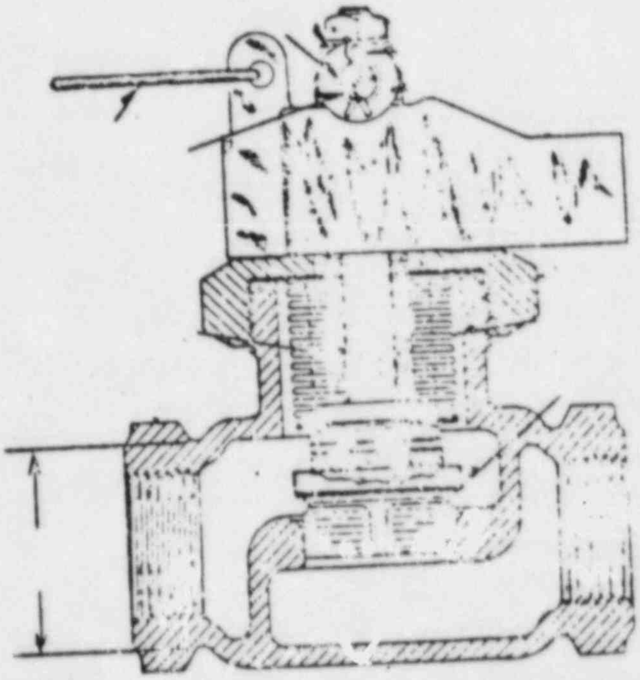
Fuel Oil Cut Off Valve

Log No. 0805C60100



MFG: ELECTRO-MOTIVE DIVISION GENERAL MOTORS CORP.

PARTS LIST NO B723-F

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Calcs. For

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FINAL

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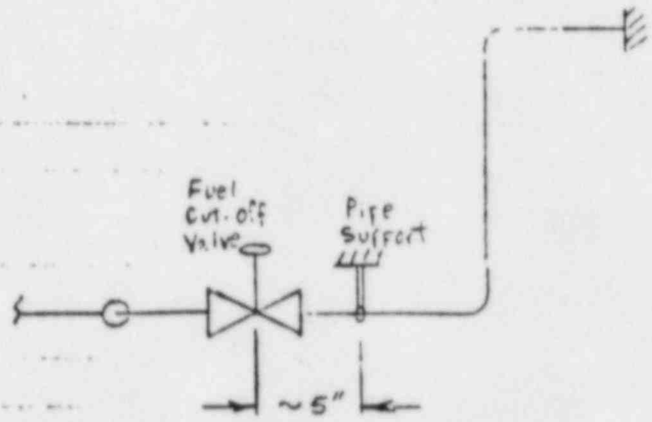
Design Parameters Relative to Valve Structural Integrity

Pipe System : Photos #9 & #19 with information obtained during field trip

Pipe Support : Photo #18 ; see figure below
Pipe support is located on structural member of skid assembly ~1ft above base of skid



Response Spectra:

Maximum (peak) accelerations : 0.28g vertical , 0.38g horizontal

These values are applicable at the base of the skid assembly. Due to the proximity of the subject valve and pipe support to the skid base, minimal amplification will occur. Note that these accelerations are at least 1/3 less than the respective LaSalle/Clinton values.

Conclusion : Since, in separate analysis, the structural integrity of the attached piping has been demonstrated and the subject valve is near a support for this piping, the valve is considered structurally adequate to withstand the loads which could occur at its location.

Form GQ-3.08.1 Rev. 2

B.3

B.3 COOLING SYSTEM

SARGENT & LUNDY
ENGINEERS
CHICAGO

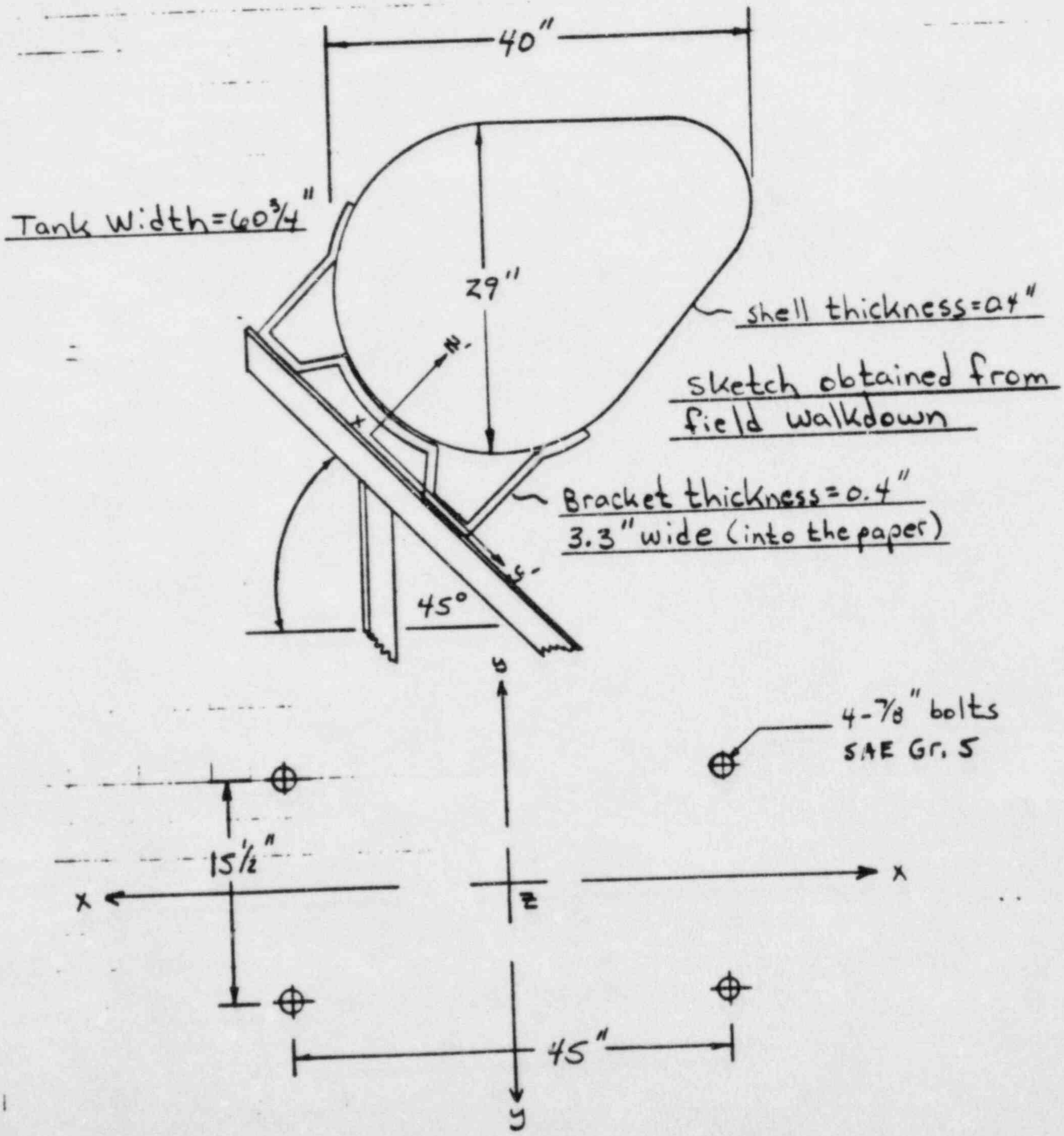
Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client		Prepared by David Wright	Date 5/10/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Water Expansion Tank

Log No. 080304 1400



sketch obtained from field walkdown

Form GQ-3.08.1 Rev. 2

Estimated Tank Weight

Tank Weight = Weight of the material + Weight of the water

$$\text{Water Wt} = \forall_{tank}\,\rho_{water}$$

For conservatism assume the tank to be a 40" O.D. cylinder, 60 3/4" long.

$$\forall_{water} = \frac{\pi D^2}{4} L = \frac{(\pi)(40)^2(60.75)}{4} = 76{,}341 \text{ in}^3$$

$$\rho_{water} = 62.4 \text{ Lbs/ft}^3 \left(\frac{1 \text{ ft}^3}{1728 \text{ in}^3}\right) = 0.036 \text{ Lbs/in}^3$$

$$\text{Water Wt} = (76{,}341 \text{ in}^3)(0.036 \text{ Lbs/in}^3) = 2{,}757 \text{ Lbs}$$

$$\text{Tank Weight} = \forall_{metal}\,\rho_{metal}$$

Using the same assumption of a 40" O.D. tank:

$$\forall_{metal} = 2\left(\frac{\pi D^2}{4}\right)t + \frac{\pi}{4}(OD^2 - ID^2)L$$

Note - ID = O.D. - 2t = 40" - 2(.40) = 39.2"

$$\forall_{metal} = 2\left(\frac{\pi 40^2}{4}\right)(.40) + \frac{\pi}{4}(40^2 - 39.2^2)(60.75)$$

$$\forall_{metal} = 4028 \text{ in}^3$$

$\rho_{metal} = 0.283 \text{ Lbs/in}^3$ (assuming A-36 steel)

$$W_t = (4028 \text{ in}^3)(0.283 \text{ Lbs/in}^3) = 1140 \text{ Lbs}$$

Total Weight = 2757 + 1140 = 3897 Lbs

Method of Analysis

Since the Shoreham engine is identical to the La Salle engine and the accessory racks are identical a comparison of the expansion tanks will be made and adjustments done to account for any weight differences. It has already been shown that the shoreham spectrum is bound by the La Salle spectrum so the seismic criteria is satisfied. (see Appendix A for spectrum comparison.)

ENGINEERING DEPARTMENT

Acc. Rack & H. Exch.

DATE: 5-76
FILE: 68693
PERFORMED BY: SHEKAR

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
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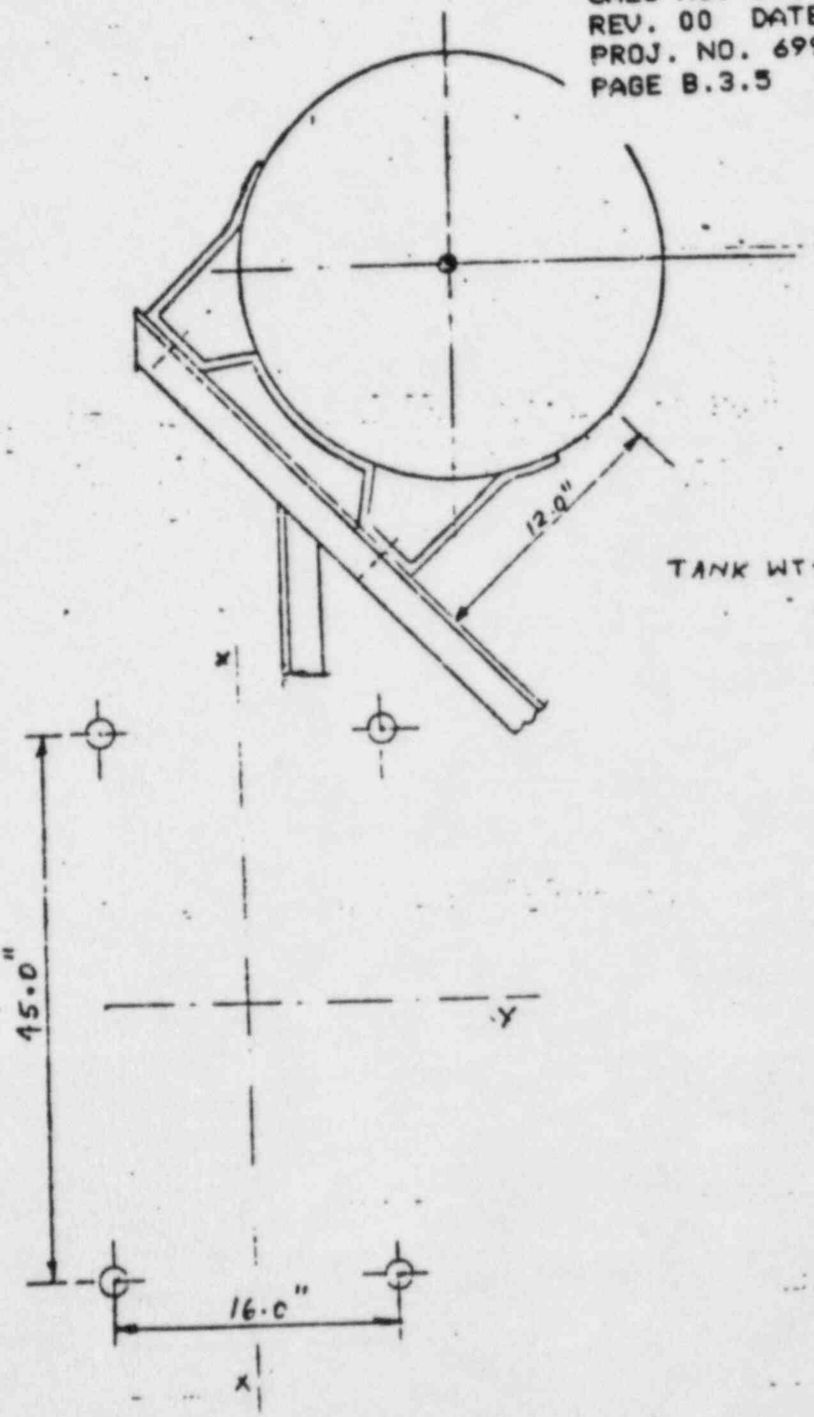
WATER EXPANSION TANK CONNECTION BOLTS



BOLTS : 7/8" DIA SAE 5
QUANTITY : 4
STRESS AREA : 0.4612 in^2

SHEET 12 OF [illegible]

FORM NO. 57-SERV.

ENGINEERING DEPARTMENT

WATER EXPANSION TANK
CONNECTION BOLT LOADS

STEWART & STEVENSON

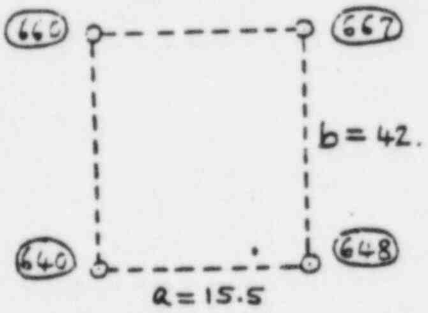
DATE: 2/17/77
FILE: W.O 6Y6RS
PERFORMED BY:

WATER EXPANSION TANK CONNECTION BOLT LOADS

S.No.	ELEMENT NO.	NODE No.	ELEMENT FORCES		
			Fx #	Fy #	Fz #
1	640	709-363	93.	10.	51.
2	648	719-362	459.	79.	372.
3	660	729-361	131.	8.	45.
4	667	739-360	475.	43.	335.

F_x (MAX.) = $\underline{475}^{\#}$

$$\sqrt{F_y^2 + F_z^2} = \sqrt{79^2 + 372^2} = \underline{380.}^{\#}$$



$$F_{\text{TENSILE (MAX.)}} = \underset{\text{El.667}}{F_x} + \underset{\text{El.643}}{\frac{M_y}{b}} + \underset{\text{El.660.}}{\frac{M_z}{a}}$$

$$= 475. + \frac{287.}{42.} + \frac{3.}{15.5}$$

$$= \underline{482.}^{\#}$$

$$F_{\text{SHEAR (MAX.)}} = \sqrt{\left(\frac{(\Sigma M_x)a}{2(a^2+b^2)} + \underset{\text{El.648}}{F_y}\right)^2 + \left(\frac{(\Sigma M_x)b}{2(a^2+b^2)} + \underset{\text{El.648}}{F_z}\right)^2}$$

ΣM_x for elements 640, 648, 660 AND 667. ≈ 0

$$= \sqrt{79.^2 + 372.^2}$$

$$= \underline{380.}^{\#}$$

SHEET

AXIMUM STRESSES

DATE: 2/11/77

FILE: W.O. 68683

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.3.7

DIESEL GENERATOR SET -

S. NO.	STRUCTURE REGION	ELEMENT NO.	NODE NO.	STRESS PSI
1	GENERATOR STATOR	329	1088-1086-1087	2262.
2	GENERATOR ROTOR	52	1128-1129	2016.
3	GENERATOR BRG. BRACKETS	396	1225-1241	4748.
4	EXCITER STATOR	1023	847-855	415.
5	ACCESSORY RACK FRAME	489	334-345	5786.
6	LUBE OIL FILTER	529	523-527-531	5553.
7	AMOT THERMOSTAT	581	612-641-613	1320.
8	WATER EXPANSION TANK	664	735-731-733	3235.
9	HEAT EXCHANGER	713	246-240-239-245	20783.
10	JACKET WATER PIPING	777	444-443-449-451	12621.
11	ACCESSORY RACK PIPING	676	659-660	1058.
12	BASE	829	135-136	8821.

AIR-START SKID-MOUNTED ASSEMBLY -

1	AIR RECEIVERS	16	30-31	107.
2	BASE, COMPRESSORS AND AIR DRYERS	26	24-34	1037.

SHEET II-3

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Similarity Analysis

Bolt pattern used in finite element-analysis 42" x 15.5"

Bolt pattern found at Shorham - 45" x 15.5"

Mounting, Previous - Accy. rack, bolted

Mounting, Shoreham - Accy. rack, bolted

Bolt size Previous - 7/8 in

Bolt size Shoreham - 7/8 in

weight previous - 1500 Lbs

Weight Shoreham (conservative estimate) - 3897 Lbs

Since the weight of the shoreham tank is greater than that of the previous analysis an adjustment will be made to account for this. Due to the very conservative adjustment and tank weight, any ecentricity differences have been accounted for.

3897 Lbs / 1500 Lbs = 2.598 times heavier.

Note: Previous Analysis found in (S+L) EMD-008028 Rev.00 dated 4/19/77, Seismic Qualification of EMD 645 Engine.

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Calcs. For

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Project		Reviewed by	Date
Proj. No.	Equip No.	Approved by	Date

Maximum Bolt Loads

Tensile Previous = 475 Lbs

Shear Previous = 380 Lbs

Tensile New = 475 Lbs (2.598) = 1234 Lbs

Shear New = 380 Lbs (2.598) = 987 Lbs

Stress Level

$$\sigma_t = \frac{F_t}{A} = \frac{1234\text{ Lbs}}{0.4612\text{ in}^2} = 2676\text{ psi}$$

$$\tau_v = \frac{F_v}{A} = \frac{987\text{ Lbs}}{0.4612\text{ in}^2} = 2140\text{ psi}$$

Allowable Stress

From ASME B+PV code, 1977 edition with summer 79 addendum, Section III, article 2460 of Appendix XVII

for service levels A+B, $F_t = 0.3 S_u$, $F_v = 0.124 S_u$

Form GQ-3.08.1 Rev. 2

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Client
Project
Proj. No. Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

$F_t = 0.3(120{,}000) = 36{,}000$psi , $F_v = .124(120{,}000) = 14{,}880$psi

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \le 1.00 \Rightarrow \frac{2676^2}{36{,}000^2} + \frac{2140^2}{14{,}880^2} = 0.026 \le 1.00$$

note- Su obtained from ASME B+PV code, 1977 edition, section III, table I-7.3 of Appendix I for SA-449.

The Maximum stress in the expansion tank (Previously analyzed) was found to be 3,235psi using the same method to adjust the level we find:

Maximum Adjusted Stress = 3235 (2.598) = 8404psi

S_{all} = 13.7 Ksi { for SA-36 plate material per ASME B+PV code, 1977 edition, table I-7.1 of Appendix I for plate material }

$S_{all.} > \sigma_p$ so the tank is adequate.

Conclusion: The tank will survive an eartquake of a magnitude greater than is expected to occur at shoreham.

Note: Previous Analysis found in (S+L) EMD-008028 Rev. 00 dated 4/19/77, Qualification of EMD 645 Diesel.

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ENGINEERS
CHICAGO

Calcs. For

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Client		Prepared by David Wright	Date 5/17/84
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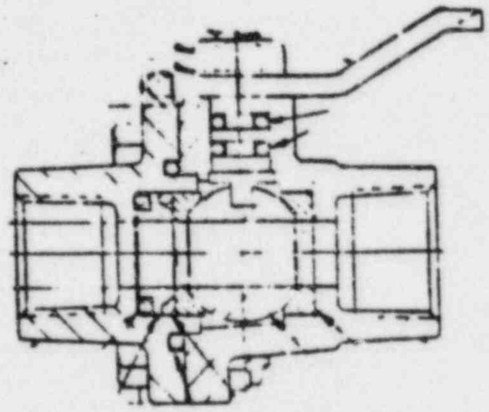
Proj. No.	Equip. No.	Approved by	Date

Cooling System Drain Valve

Log NO. 0803041201



EMD NO.: 8354227

mfg.: Smith Valve

Model: 718-15

The Cooling system drain valve found on the Shoreham engine is identical to the one found on the La Salle engines (CECo.). The valves in the previous report (CQD-010330 Rev. 00) were qualified in accordance with ASME B+PV code sect. III subsection ND-3500. (The valves are all class 3 items.) In addition, the applicable valves are qualified in conformance with ANSI B16.34. By using the documents as design rules, all valves will be qualified to show structural and pressure integrity. The previous analysis is attached.

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REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
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SARGENT & LUNDY
ENGINEERS
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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client

Project

Proj. No. Equip. No.

Prepared by Date

Reviewed by Date

Approved by Date

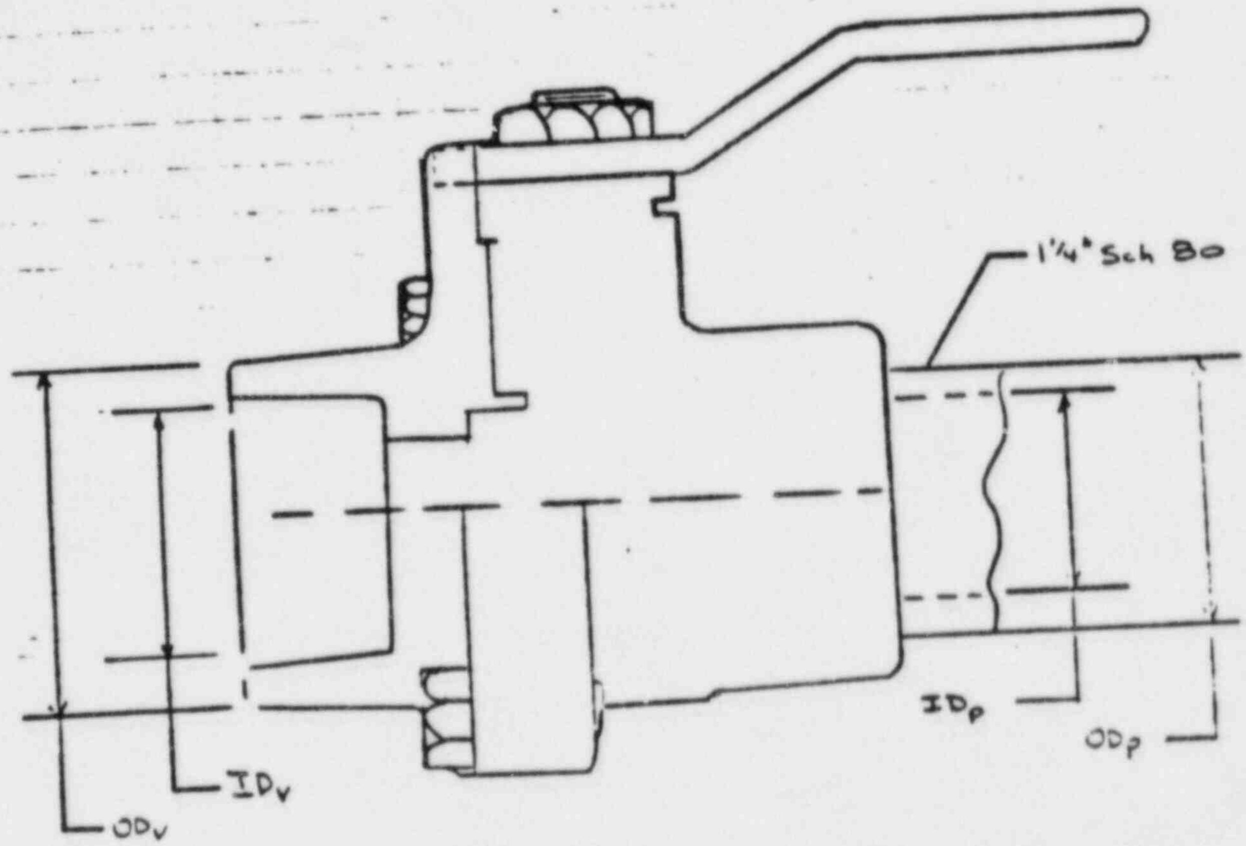

FIGURE 11

OD_v = 1.9375 IN
ID_v = 1.1875 IN
OD_p = 1.660 IN
ID_p = 1.278 IN

(1.9375" − 2(.375") = 1.1875 IN. SEE PHONE MEMO attached

Note – Data for 1¼" Shc 80 pipe obtained from Cameron Engineering Tables For Pipe.

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REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
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SARGENT & LUNDY
ENGINEERS
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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client		Prepared by	Date
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Proj. No.	Equip. No.	Approved by	Date

BODY STRUCTURAL INTEGRITY

S_V = VALVE BODY ALLOWABLE STRESS LIMIT FOR ASTM B283 (ALLOY 377) MATERIAL AT 100°F
= 12.0 KSI (ANSI/ASME B 31.1-1983 ED. APPENDIX A, TABLE A-6

S_P = PIPE ALLOWABLE STRESS LIMIT FOR ASTM A106 GD B (ASSUMED) MATERIAL @ 100°F
= 15.0 KSI (TABLE I-7.1 P.103 - ASME APPENDICES)

SINCE S_V/S_P = 12.0/15.0 = 0.8 < 1.0 THEN:

SHOW THAT $\frac{S_V}{S_P} \times \frac{Z_V}{Z_P} > 1.1$

Z_V = SECT MOD. OF VALVE BODY

$$= \frac{3.14}{32} \frac{[(OD_V)^4 - (ID_V)^4]}{OD_V}$$

$$= \frac{3.14}{32} \frac{[(1.9375 \text{ IN})^4 - (1.1875 \text{ IN})^4]}{(1.9375 \text{ IN})} = 0.61326 \text{ IN}^3$$

Z_P = SECT MOD. OF PIPE
= 0.2914 IN³ (FOR 1¼ SCH 80 CAMERON CAT. P 172)

$$\frac{S_V}{S_P} \times \frac{Z_V}{Z_P} = 0.8 \times \frac{0.61326}{0.2914} = 1.68 > 1.1$$

Form GO-3081 Rev. 2

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Client

Project

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Prepared by Date

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Approved by Date

SHOW THAT $\frac{S_V}{S_P} \times \frac{A_V}{A_P} > 1.1$

A_V = CROSS SEC. AREA OF VALVE BODY

$$= \frac{\pi}{4}\left[OD_V^2 - ID_V^2\right]$$

$$= \frac{\pi}{4}\left[(1.9375\ IN)^2 - (1.1875\ IN)^2\right] = 1.8407\ IN^2$$

A_P = CROSS SECT. AREA OF CONNECTING PIPE

$= 0.881\ IN^2$ (FOR 1¼ SCH 80 PIPE PER CAMERON CAT. P.172)

$$\frac{S_V}{S_P} \times \frac{A_V}{A_P} = .8 \times \frac{1.8407}{0.881} = 1.67 > 1.1$$

∴ THE VALVE BODY IS STRONGER THAN THE PIPE

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FIGURE 718 - 1¼" SIZE

THE BOLT THREAD SERIES & SIZE & MATERIAL, AND THE PRESSURE CLASS ARE THE SAME AS FIG. 709 - 1¼" SIZE. (4 BOLTS)

GASKET IS BUNA-N 'O' RING WITH AN ID = 1 IN AND THICKNESS = 0.103 IN (PHONE MEMO)

$$OD = ID + 2\,\text{thickness} = 1 + 2(.103) = 1.206\ IN$$

$$A_g = \frac{\pi}{4}(1.206\,IN)^2 = 1.1423\ IN^2$$

$$F = S_r W_v = 6.23g\ (5.0\,LBS) = 31.15\ LBS$$

$$\frac{P_c \times A_g + F}{A_b} = \frac{400\,\frac{LB}{IN^2} \times 1.1423\,IN^2 + 31.15\,LBS}{0.904\,IN^2}$$

= 540 PSI ⇒ Negligable stress for steel.

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Client		Prepared by Andrew R Wisniewski	Date
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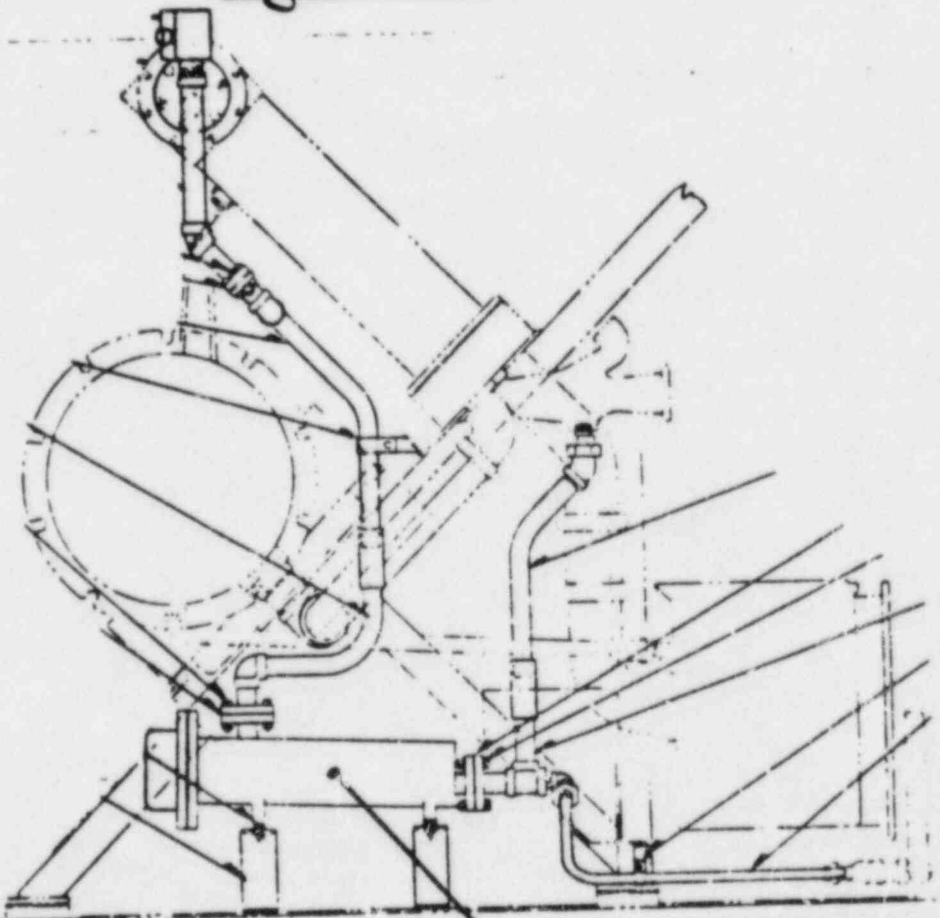
Proj. No.	Equip. No.	Approved by	Date

Immersion Heater

Log Number 0803040800



Immersion Heater

The immersion heater is an electrical device that was found not to be essential to the operation of the engine. The only requirement is that the heater remain structurally intact.

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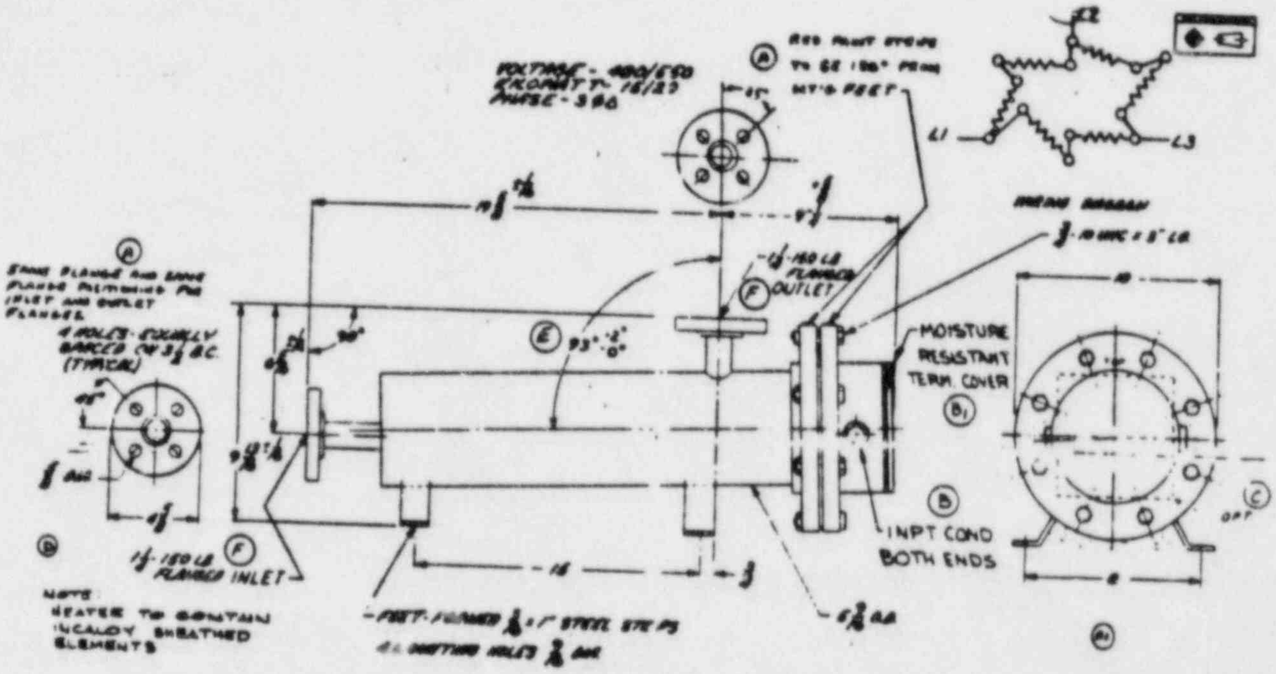
Client		Prepared by	Date
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date

Immersion Heater



Heater Shipping Weight - 125 lb (dry) (see phone memo between D. Wright + C. Eckberg)
Working Temperature - 190°F (from G.M. list of parameters attached)
Working Pressure - 25 psig (" " ")
Heater Shell Material - assumed to be SA-106 grade A

The modes of failure which are considered critical and will be investigated are:

- stresses at the heater-to-1¼" pipe nozzle juncture
- mounting bolt stresses

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CALC NO. CQD-014046
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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

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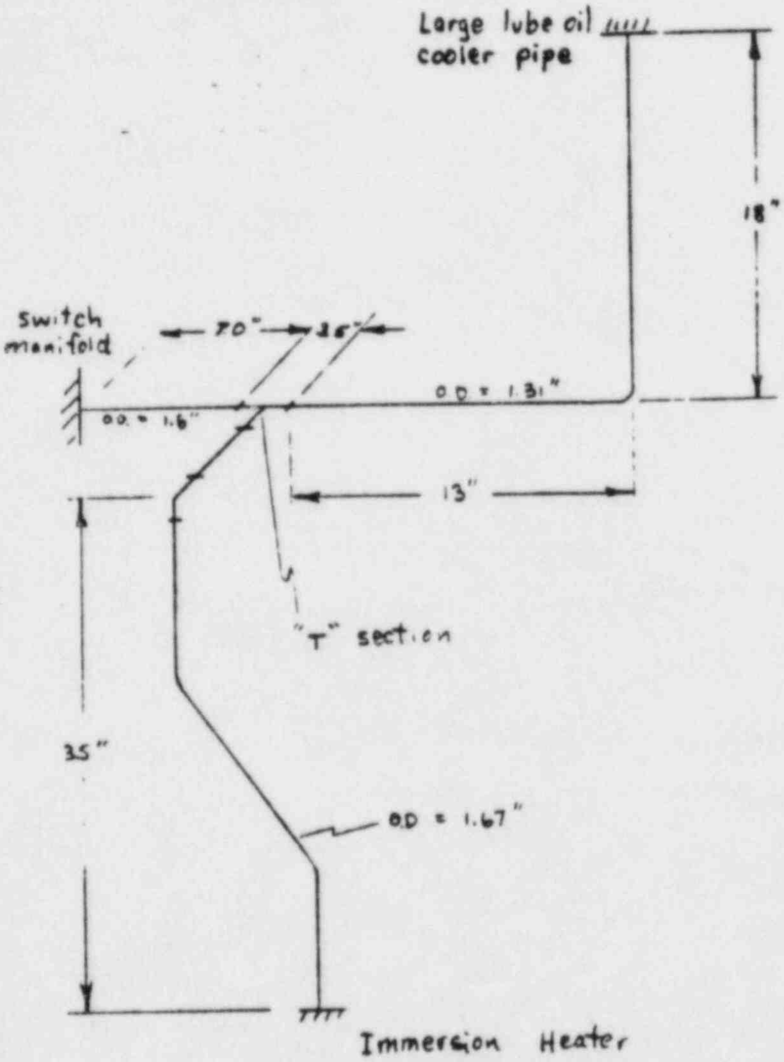

IMMERSION HEATER OUTLET PIPING DIAGRAM

sketch obtained during field walk-down

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Calcs. For

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Client

Project

Proj. No. Equip. No.

Prepared by Date

Reviewed by Date

Approved by Date

Heater - pipe nozzle juncture stresses

Applied nozzle loads are obtained from CQD-005513, 1/21/83, "Nozzle Loads for Equipment". Loads are based on a 2" nozzle size (conservative). Stresses are calculated on the following page using the procedure given in Welding Research Council Bulletin No 107, Rev. 3, April 1972

Maximum stress = 12.636 ksi

Design criteria : Maximum stress ≤ 1.5 S, where
S = allowable stress of shell material

From ASME B+PV code, section III, 1983 edition, table I-7.1 of Appendix I,
S = 12.0 ksi for SA-106 grade A steel at 190°F

Maximum stress = 12.636 ksi < 1.5(12.0) = 18.0 ksi
⇑ for service level B loading, ASME B+PV code, sect III, subsection ND for class 3 components.

∴ heater shell stresses are acceptable

CALC NO. CUD-014046
REV. 00 DATE 6/1/84
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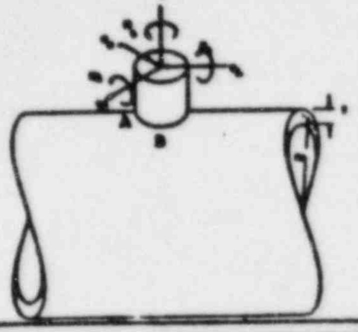
LOCAL MEMBRANE STRESSES IN A CYLINDRICAL SHELL DUE TO NOZZLE LOADS AND PRESSURE



Project Name:
Project No:
Spec. No:

Equip. # Log # 0803040800	Pipe Line #

Equipt. Description: IMMERSION HEATER

Nozzle Description: OUTLET

Subsystem #	Node #	PIPSYS

Internal Pressure P_o = 250 Psig

APPLIED NOZZLE LOADS

AXIAL LOAD	P =	329.1	lbs
SHEAR LOAD	V_c =	329.1	lbs
SHEAR LOAD	V_L =	329.1	lbs
TORSIONAL MOMENT	M_T =	2771	in-lbs
CIRCUMFERENTIAL MOMENT	M_c =	2771	in-lbs
LONGITUDINAL MOMENT	M_L =	2771	in-lbs

GEOMETRY

VESSEL MEAN RADIUS	R_m =	2.65	in.
VESSEL THICKNESS	T =	0.258	in.
NOZZLE RADIUS	r_o =	0.625	in.

GEOMETRIC PARAMETERS

$\gamma = \frac{R_m}{T}$ = 10.27

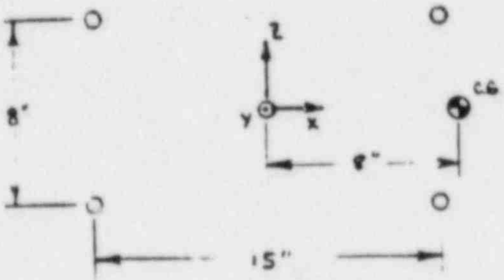
$\beta = (0.875)\frac{r_o}{R_m}$ = 0.206

	MULTIPLY AND ENTER ABSOLUTE VALUE OF RESULT		STRESSES (AT LOCATIONS A,B)	
	READ CURVES FOR	CALCULATE	A	B
CIRCUMFERENTIAL STRESSES	$\frac{N_\phi}{P/R_m}$ = 1.5	$\frac{P}{R_m T}$ = 481.4	722.1	722.1
	$\frac{N_\phi}{M_c/R_m^2 \beta}$ = 0.32	$\frac{M_c}{R_m^2 \beta T}$ = 7424	▨	2375.7
	$\frac{N_\phi}{M_L/R_m^2 \beta}$ = 1.2	$\frac{M_L}{R_m^2 \beta T}$ = 7424	8908.8	▨
	Circumferential Pressure Stress, $P_o R_m/T$		256.8	256.8
	Sum of Circumferential Stresses σ_ϕ =		9887.7	3354.6
LONGITUDINAL STRESSES	$\frac{N_x}{P/R_m}$ = 1.8	$\frac{P}{R_m T}$ = 481.4	866.5	866.5
	$\frac{N_x}{M_c/R_m^2 \beta}$ = 0.51	$\frac{M_c}{R_m^2 \beta T}$ = 7424	▨	3786.2
	$\frac{N_x}{M_L/R_m^2 \beta}$ = 0.33	$\frac{M_L}{R_m^2 \beta T}$ = 7424	2449.9	▨
	Longitudinal Pressure Stress, $P_o R_m/2T$		128.4	128.4
	Sum of Longitudinal Stresses σ_x =		3444.8	4781.1
SHEAR STRESSES	Due to Torsion M_T	$\frac{M_T}{2\pi r_o^2 T}$	4376.0	4376.0
	Due to Load V_c	$\frac{V_c}{\pi r_o T}$	649.6	▨
	Due to Load V_L	$\frac{V_L}{\pi r_o T}$	▨	649.6
	Sum of Shear Stresses τ =		5025.6	5025.6
Maximum Stress	$= 1/2 \left[(\sigma_\phi + \sigma_x) + \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2} \right]$ =		12,636 psi	9144 psi

Maximum stress should be less than or equal to 1.5S where S is the allowable stress of shell material.

Mounting Bolt Stresses

Mounting is achieved by 4 - ½"ϕ bolts, SA-449 steel (assumed)



Based on the heater construction, the center of gravity is assumed to be offset from the bolt centroid 8" towards the flanged end of the component.
Y_{CG} assumed to be 4"

$e_x = 8"$ $e_y = 4"$ $e_z = 0$

Component weight

Dry weight = 125 lb (see attached phone memo)

Assuming the heater is filled with water, the liquid weight is approximately:

$(62.4\ lb/ft^3)(\pi)(2.524\ in)^2(24\ in)\left(\frac{1\ ft}{12\ in}\right) = 17.3\ lb$

∴ Total weight = 125 + 17.3 = 142.3 lb

Seismic Plus Dead Weight Loads:

For a conservative analysis, the heater is assumed flexible in all directions. Applied g-values of 1.5 times peak response spectra values will be used. SSE values are used (see Shoreham base spectrum in Appendix A of this report.)

$a_x = a_z = 1.5(0.55) = 0.825\,g$
$a_y = 1.5(0.36) + 1.0 = 1.54\,g \downarrow$ or $a_y = -1.5(0.36) + 1.0 = 0.46\,g \downarrow$

Client		Prepared by	Date
Project		Reviewed by	Date
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$F_x = F_z = (142.3)(0.825) = 117.4$ lb
$F_y = (142.3)(1.54) = 219.1$ lb ↓ or $F_y = (142.3)(0.46) = 65.5$ lb ↓

$M_x = (219.1)(0) + (117.4)(4) = 469.6$ in-lb
$M_y = (117.4)(0) + (117.4)(8) = 939.2$ in-lb
$M_z = (117.4)(4) + (219.1)(8) = 2222.4$ in-lb

Nozzle Loads : Calculated in the table on the following page.

$F_x = F_y = F_z = 658.2$ lb

$M_x = 0.827$ ft-k $= 9924$ in-lb
$M_y = 1.00$ ft-k $= 12000$ in-lb
$M_z = 1.365$ ft-k $= 16380$ in-lb

Total Equipment Loads Acting on Bolt Pattern : (at centroid)

$F_x = F_z = 117.4 + 658.2 = 775.6$ lb
$F_y = 219.1 + 658.2 = 877.3$ lb ↓ or $F_y = 658.2 - 65.5 = 592.7$ lb ↑

$M_x = 469.6 + 9924 = 10394$ in-lb
$M_y = 939.2 + 12000 = 12939$ in-lb
$M_z = 2222.4 + 16380 = 18602$ in-lb

Bolt Pattern Properties :

$A_{bolt} = 0.126$ in² (root area used for all properties, conservative)
$\Sigma A = 4(0.126) = 0.504$ in²

$I_{xx} = 4(0.126)(4)^2 = 8.064$ in⁴
$I_{zz} = 4(0.126)(7.5)^2 = 28.35$ in⁴
$J_{yy} = I_{xx} + I_{zz} = 36.41$ in⁴

NOZZLE #	LINE NUMBER	SIZE	NOZZLE DESCRIPTION	SUBSYSTEM	NODE #	LOCATION (FT.)		
						X / X'	Y	Z / Z'
1		1¼" ϕ	INLET			0.948	0.292	0
2		1¼" ϕ	OUTLET			0.688	0.818	0
3								
4								
5								

NOZZLE #	NOZZLE FORCES (KIP)			MOMENTS DUE TO F_x, F_y, F_z (FT-KIP)						NOZZLE MOMENTS (FT-KIP)		
	F_x	F_y	F_z	$(F_x)(Y)$	$(F_x)(Z')$	$(F_y)(X)$	$(F_y)(Z)$	$(F_z)(X')$	$(F_z)(Y)$	M_{xn}	M_{yn}	M_{zn}
1	0.3291	0.3291	0.3291	.0961	0	0.312	0	0.312	.0961	0.231	0.231	0.231
2	0.3291	0.3291	0.3291	0.269	0	0.226	0	0.226	0.269	0.231	0.231	0.231
3												
4												
5												
TOTAL	0.6582	0.6582	0.6582	0.3651	0	0.538	0	0.538	0.3651	0.462	0.462	0.462

$M_x = M_{xn} + (F_y)(Z) + (F_z)(Y) =$ 0.827

$M_y = M_{yn} + (F_x)(Z') + (F_z)(X') =$ 1.000

$M_z = M_{zn} + (F_x)(Y) + (F_y)(X) =$ 1.365

PLANT CONDITION:

☐ Normal

☐ Upset

☒ Emergency

NOZZLE LOADS FROM:

☐ Piping Stress Report

☐ EMD Tech. Proc. #26

☐ Vendors Report

☒ CQD-005513, Nozzle Loads"

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Maximum Bolt Stresses :

$$\sigma_t = \frac{F_y}{\Sigma A} + \frac{M_x C_z}{I_{xx}} + \frac{M_z C_x}{I_{zz}}$$

$$= \frac{592.7}{0.504} + \frac{(10394)(4.0)}{8.064} + \frac{(18602)(7.5)}{28.35} = 11253 \text{ psi}$$

$$\tau = \frac{\sqrt{F_x^2 + F_z^2}}{\Sigma A} + \frac{M_y \sqrt{C_x^2 + C_z^2}}{J_{yy}}$$

$$= \frac{\sqrt{7756^2 , 7756^2}}{0.504} + \frac{(12939)\sqrt{7.5^2 + 4.0^2}}{36.41} = 5197 \text{ psi}$$

Allowable Stress

Per ASME B+PV code, 1977 edition with summer 79 addendum, Section III, article 2460 of Appendix XVII,

$F_t = 0.3 S_u = 0.3(\overset{*}{1}20{,}000\text{psi}) = 36{,}000\text{psi}$ (service level A+B)

$F_r = 0.124 S_u = 0.124(\overset{*}{1}20{,}000\text{psi}) = 14{,}880\text{psi}$ (service levels A+B)

$$\frac{f_t^2}{F_t^2} + \frac{f_r^2}{F_r^2} \leq 1.00 \Rightarrow \frac{11{,}253^2}{36{,}000^2} + \frac{5197^2}{14{,}880^2} = 0.22 \leq 1.0$$

The bolts are adequate.

*Note - From ASME B+PV code, 1977 edition, Section III, table I-7.3, of Appendix I, SA-449 steel with D<1"

ITEM	PART NO	LOCATION	WORKING TEMP. (°F)	WORKING PRESS. (PSI)	MATERIAL	* ITEMS APPLY TO EMC 183078 W.O. 74080 MODEL- 520E4
FUEL PUMP	8080284	ENGINE ATTACH (FUEL OIL SYS)	90	45	CAST IRON T-35000PSI (MIN.)	
GAGE VALVE	8422197	NOT BY EMD (FUEL OIL SYS)	–	–	BRASS	
FLEX CONNECTION	8347788	INLET TO J.W. PUMP (COOLING SYS)	190	15	STL. FLANGE - SAE 1015-1018 STL. CONNECTOR - SAE 1026	*
WATER PUMP LH	8347607	ENGINE ATTACH (COOLING SYS)	190	70	CAST IRON T-25000PSI (MIN.)	*
WATER PUMP RH	8249002	ENGINE ATTACH (COOLING SYS.)	190	70	CAST IRON T-25000PSI (MIN.)	*
ASPIRATOR	8367825	ACCESS. RACK (COOLING SYS)	190	15	CAST IRON T-25000PSI (MIN.)	*
LUBE OIL COOLER	8437992	ACCESS. RACK (COOLING SYS.)	190 (H2O) 235 (OIL)	25 (H2O) 5 (OIL)	C.I. HDR. T-25000 PSI (MIN.) TANK - HOT ROLLED SHEET STL. TUBES - RED BRASS ASTM B-135 ALLOY No1	*
SIGHT GLASS	8157497	ACCESS. RACK (COOLING SYS.)	200	10	BRASS FITTINGS - BS-565 T-55000PSI (MIN.)	*
EXPANSION TANK	8413656	ACCESS. RACK (COOLING SYS)	200	10	HOT ROLLED SHEET STEEL	*
IMMERSION HEATER	8398082	ACCESS. RACK (COOLING SYS.)	190	25	STEEL - SAE 1010	*
THERM. VALVE.	8394034	ACCESS. RACK (COOLING SYS)	200	35	GREY C.I. - ASTM A48-64 CLASS 25B	*
IMMERSION HEATER	8394593	ACCESS. RACK (COOLING SYS.)	190	25	STEEL - SAE 1010	
FLEX CONNECTION	8350434	ACCESS. RACK (COOLING SYS)	190	15	STEEL - SAE 1026	

SARGENT & LUNDY

MEMORANDUM OF TELEPHONE CONVERSATION

Date: 5/17/84

Time: 1:30-pm

Person Called: Chris Eckberg (Name) of Thermolink (Company)

Person Calling: David Wright (Name) of S&L (Company)

Project: Shoreham Project No. 6995-00

Subject Discussed: Design Dimensions of the Thermolink Model 78V10091-874 Immersion Heater

Summary of Discussion, Decisions and Commitments:

The following data was provided by Thermolink:

Shipping Wt. (incl. crate) - 125 lbs.
Nozzle Sizes - 1-1/4" sch 40 standard pipe
Shell Size - 5" sch 40 standard pipe
Mounting Brackets - 1/4" plate
Bolt Hole Size - 9/16" O.D.

Note - The overall dimensions of the heater described were almost identical to those on EMD drawing no. EMD-8398082. Thermolink also stated that the heating coils in the units were interchangeable with all of EMD (GM) units.

DW/eg

David Wright
Signature

cc C. Eckberg - Thermolink
P. Hlepas - 30
AEM/APD/DMW - 30

File: ~~[illegible] 30~~

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Client
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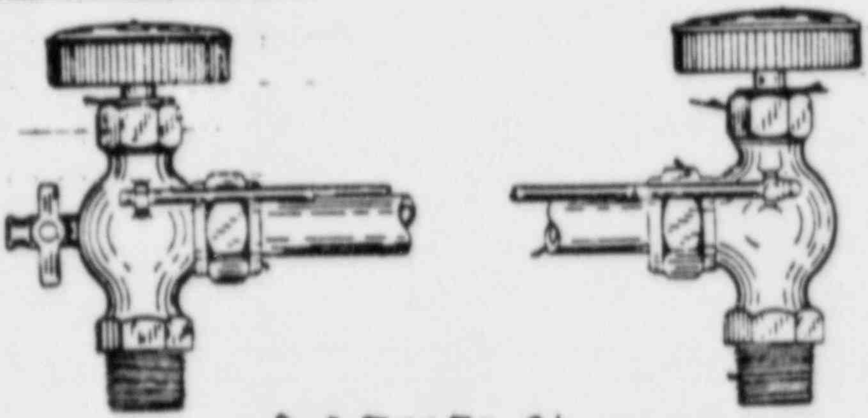
Proj. No. Equip. No.

Prepared by Dale Wright Date 5/17/84
Reviewed by Date
Approved by Date

Expansion Tank Sight Glass

Log No. 0803041401



Mfg: ESSEX Glass

The Sight Glass found on the Shoreham expansion tank is identical to the one found on the LaSalle (CECo) engine. The results of the previous analysis ((S+L) CQD-010330 Rev.00) proved the equipment to be adequate for use on the engine. The glass was shown to remain structurally intact during a seismic occurance. The maximum stress levels were:

$\sigma_{\substack{max \\ glass}} = 406 psi$ (CQD-010330 Rev.00 (S+L))

$\sigma_{\substack{max \\ body}} = 3557 psi$ (" ")

The Shoreham response spectrum has been proven to be <u>less</u> severe in Appendix A of this report.

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Calcs. For LOG NO. 0803040
& 0803040901

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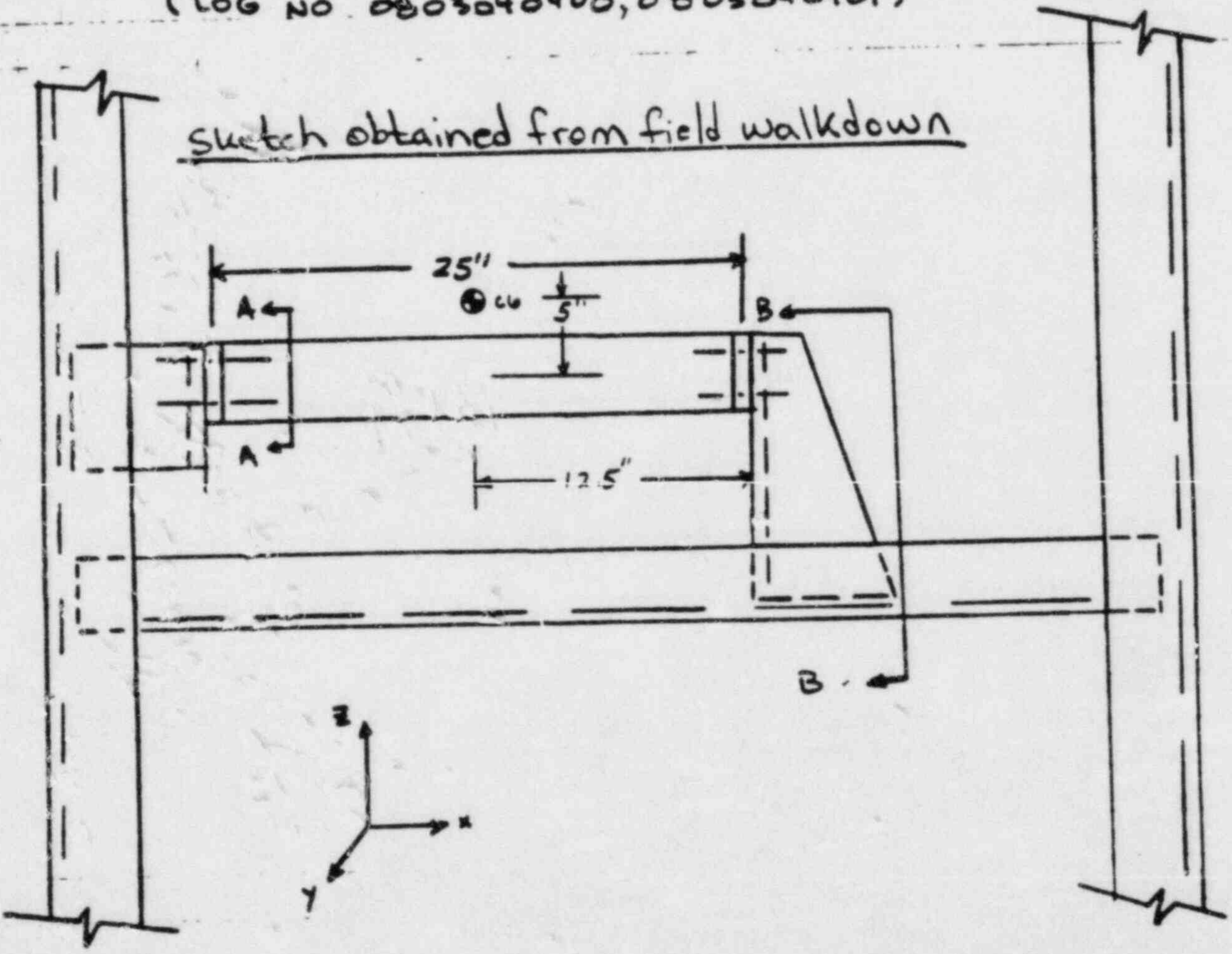
Project
Proj. No. Equip. No.

Prepared by
Date 5/4/84
Reviewed by
Date
Approved by
Date

TEMP . CONTROL MANIFOLD ASSEMBLY

(LOG NO 0803040900, 0803040901)



sketch obtained from field walkdown

ESTIMATED WT ≅ 75 LBS
(CONSERVATIVELY)

SEE PHOTOS 13 & 14

CG IS ASSUMED (CONSERVATIVE)

DISTANCES FROM C.G TO BC ARE x = 12.5 y = 1 z = 5

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SINCE THE WEIGHT IS DISTRIBUTED EVENLY ACROSS THE MANIFOLD THEN WE CAN ANALYZE EITHER SECTION AA OR SECTION BB & CHECK THE BOLT STRESSES

SINCE WE ASSUMED TOTAL WT OF 75 LBS THE BOLTS ON SECTION AA SEE 1/2 THIS AMOUNT 75/2 = 37.5 LBS

SEISMIC LOADS

Fx = 37.5 LBS (*2.8) = 105 LBS
Fy = 37.5 LBS (*2.8) = 105 LBS
Fz = 37.5 LBS (*7.1) = 266.25 LBS
Mx = 105 (5) + 266.25 (1) = 791.25 IN LBS
My = 105 (5) + 266.25 (12.5) = 3853.125 IN LBS
Mz = 105 (1) + 105 (12.5) = 1417.5 IN LBS

VIBRATION LOADS due to normal operation

The acceleration levels are taken from S+L's CQD-012628, Rev. 00 dated 2/22/84, "Response of an EMD 645 Engine and Selected Accessories to Impulsive and Steady State Loads".

x = .126, y = .142, z = .111

Fx = 37.5 LBS (0.126) = 4.725 LBS
Fy = 37.5 LBS (0.142) = 5.325 LBS
Fz = 37.5 LBS (0.111) = 4.16 LBS
Mx = 5.325 (5) + 4.16 (1) = 30.785 IN LBS
My = 4.725 (5) + 4.16 (12.5) = 75625 IN LBS
Mz = 4.725 (1) + 5.325 (12.5) = 71.2875 IN LBS

* Note - assumed "g" levels can be seen to be conservative in Appendix A.

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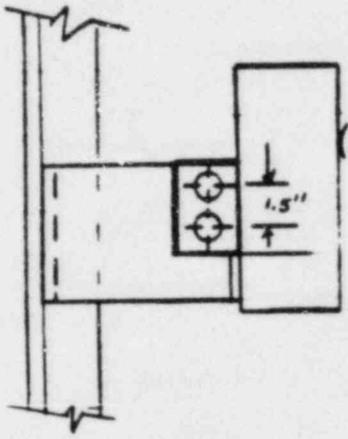
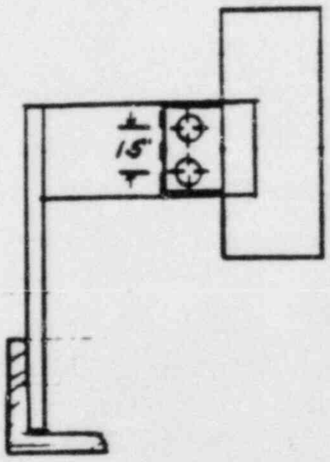
Temperature Manifold



(2) SAE GD5 BOLTS
MEASURED 0.75" ATF
∴ ½" BOLTS

SECTION A-A



(2) SAE GD 5 BOLTS
MEASURED 0.75" ATF
∴ ½" BOLTS

SECTION B-B

sketch obtained from field walkdown

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TOTAL LOADS

FX = 105 + 4.725 = 110 LBS

FY = 105 + 5.325 = 111 LBS

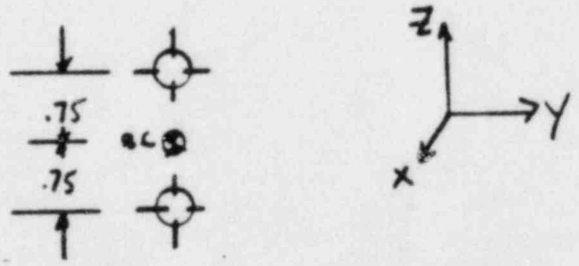
FZ = 266.25 + 4.16 + 37.5 = 308 LBS

MX = 791.25 + 30.78 = 822 INLBS

MY = 3853 + 76 = 3929 INLBS

MZ = 1418 + 72 = 1490 INLBS

BOLT PATTERN



BOLT AREA = $\frac{\pi}{4}(.5)^2$ = 0.196 IN^2

TOTAL BOLT AREA = 2(0.196) = 0.392 IN^2

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TENSION DUE TO FX

$\frac{FX}{2\ Bolts} = \frac{110}{2} = 55$ LBS

$\frac{55}{.342} = 140$ PSI

TENSION DUE TO MY

(1 ACTIVE BOLT)

$MY = 1\ F_R (1.5")$

$\frac{3929^{\text{IN LBS}}}{1.5\ \text{IN}} = F_R$ $F_R = 2619.3$ LBS

$\frac{2619.3}{.196} = 13,364$ PSI

SHEAR DUE TO FZ

$\frac{FZ}{1\ BOLT} = 308$ LBS

$\frac{308}{.196} = 1572$ PSI

SHEAR DUE TO FY

$\frac{FY}{2\ Bolts} = \frac{111}{2} = 55.5$ LBS

$\frac{55.5}{.392} = 142$ PSI

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TENSILE STRESS

$\sigma_t = 13,364 + 140 = 13,504$ PSI

SHEAR STRESS

$\tau_{max} = 1572 + 142 = 1714$ PSI

Allowable Stress

From ASME B+PV code, 1977 edition with Summer 79 addendum, section III, article 2460 of Appendix XVII, Service levels A+B;

$F_t = 0.3S_u = 0.3(\overset{*}{120}\text{ ksi}) = 36000\text{psi}$, $F_v = 0.124S_u = (.124)(\overset{*}{120}\text{ksi}) = 14{,}880$ psi

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \leq 1.00 \Rightarrow \frac{13,504^2}{36,000^2} + \frac{1714^2}{14,880^2} = 0.154 \leq 1.00$$

The bolts are adequate.

NOTE THE SEISMIC LOADS USED IN THIS ANALYSIS WERE EXTREMELY CONSERVATIVE WHICH RESULTED IN THE HIGH STRESSES ABOVE. NOTE THE ACTUAL BOLT LOADS DUE TO SEISMIC EFFECTS COULD BE REDUCED BY USING THE PERTINENT SHOREHAM SPECTRA RESPONSE SPECTRA PEAK G VALUES (TIMES 1.5) & THE PERTINENT WEIGHT OF THE MANIFOLD ASSEMBLY. HOWEVER, THIS ANALYSIS SHOWS BY CONSERVATIVE RESULTS THAT THE TEMP CONTROL MANIFOLD ASSEMBLY IS ADEQUATE.

*Note- From ASME B+PV CODE, 1977 edition, Section III, table I-7.3 of appendix I for SA-449 steel

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THE TEMPERATURE CONTROL MANIFOLD BRACKET THAT ATTACHES THE TEMP CONTROL MANIFOLD TO THE ACCESSORY RACK IS ACCEPTABLE BY ENGINEERING JUDGEMENT BASED ON THE FOLLOWING STATEMENTS

1) THE ACCESSORY RACK IS RIGID

2) THE ACCESSORY RACK VIBRATION ACCELERATION LEVELS ARE VERY SMALL AS SHOWN IN THIS ANALYSIS

3) THE ACTUAL WT. OF TEMP CONTROL MANIFOLD IS SMALL

4) THE BRACKETS HAVE SUFFICIENT WELD SIZE AND LENGTHS

5) THE BRACKET THICKNESS IS 1/4 INCH

6) THE ECCENTRICITIES OF THE TEMP CONTROL MANIFOLD WITH RESPECT TO BRACKET SUPPORT IS SMALL (WT IS SMALL)

7) THE WEIGHT IS DISTRUBUTED EVENLY ACROSS THE TEMP CONTROL MANIFOLD

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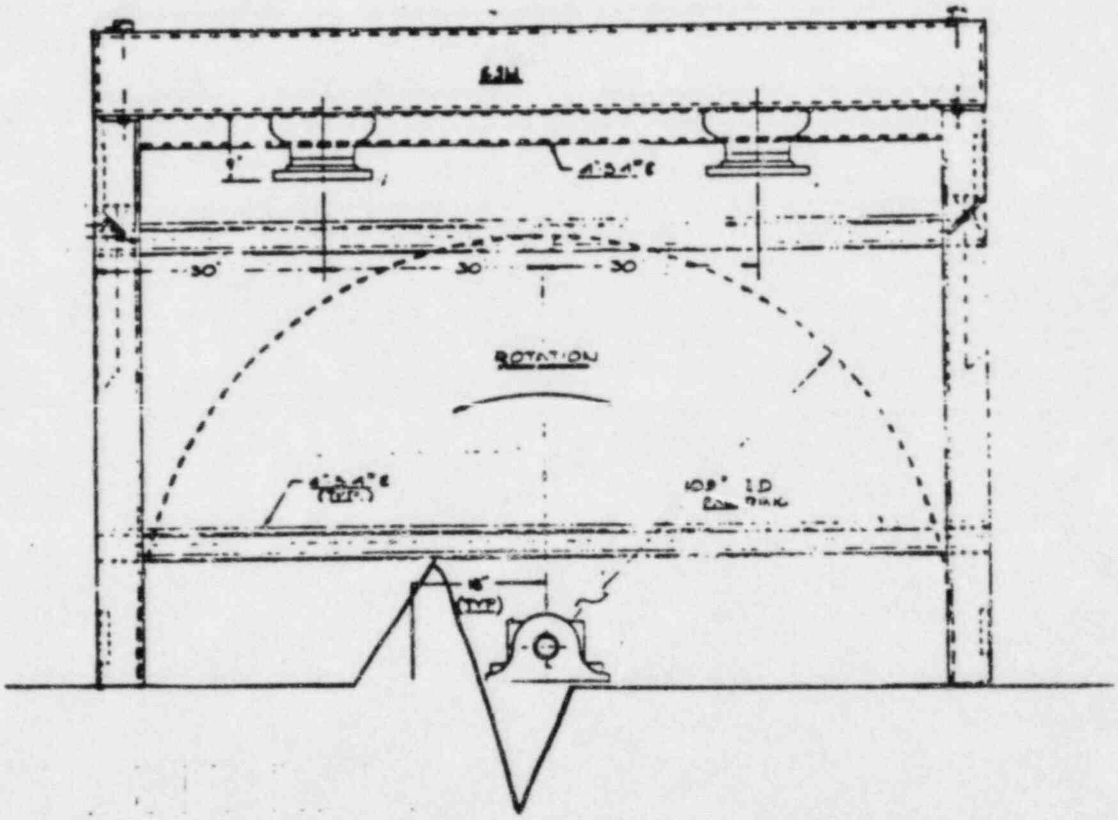
Radiator Structural Integrity Analysis

Log No. 0803110000



Conclusions:

The existing two channel sections (3"x1¾") at the centerline are found to be overstressed for OBE and SSE loading conditions. The addition of 2 - 3 in I-Beams (S 3 x 7.5) spanning from side wall to side wall is necessary. This modification will bring the stresses to within allowable limits. Location details are found inside.

1-d

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References:

1) Memorandum of Telephone Conversation between Mark Shoam of O&M Manufacturing and David Wright of S&L, dated 5/8/84.

2) MSS-6.2-D, S&L Standart Specification for Dynamic Qualification Criteria for Nuclear Safety Related Equipment.

3) Formulas for Stress and Strain, Roark and Young, 5th Edition, 1975.

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Analysis:

Following assumptions are made:

i) Radiators are simply supported along the walls, rigid I beam and on the channel members at the center,

ii) No credit is taken for the radiator rigidity,

iii) No credit is taken for the web of the channel sections due to perforations,

iv) Load distribution is based on spans of 48" and 76" (See figure on next page)

v) Weight of radiator is 9000 lb total (4500 lb per side) per Ref. 1.

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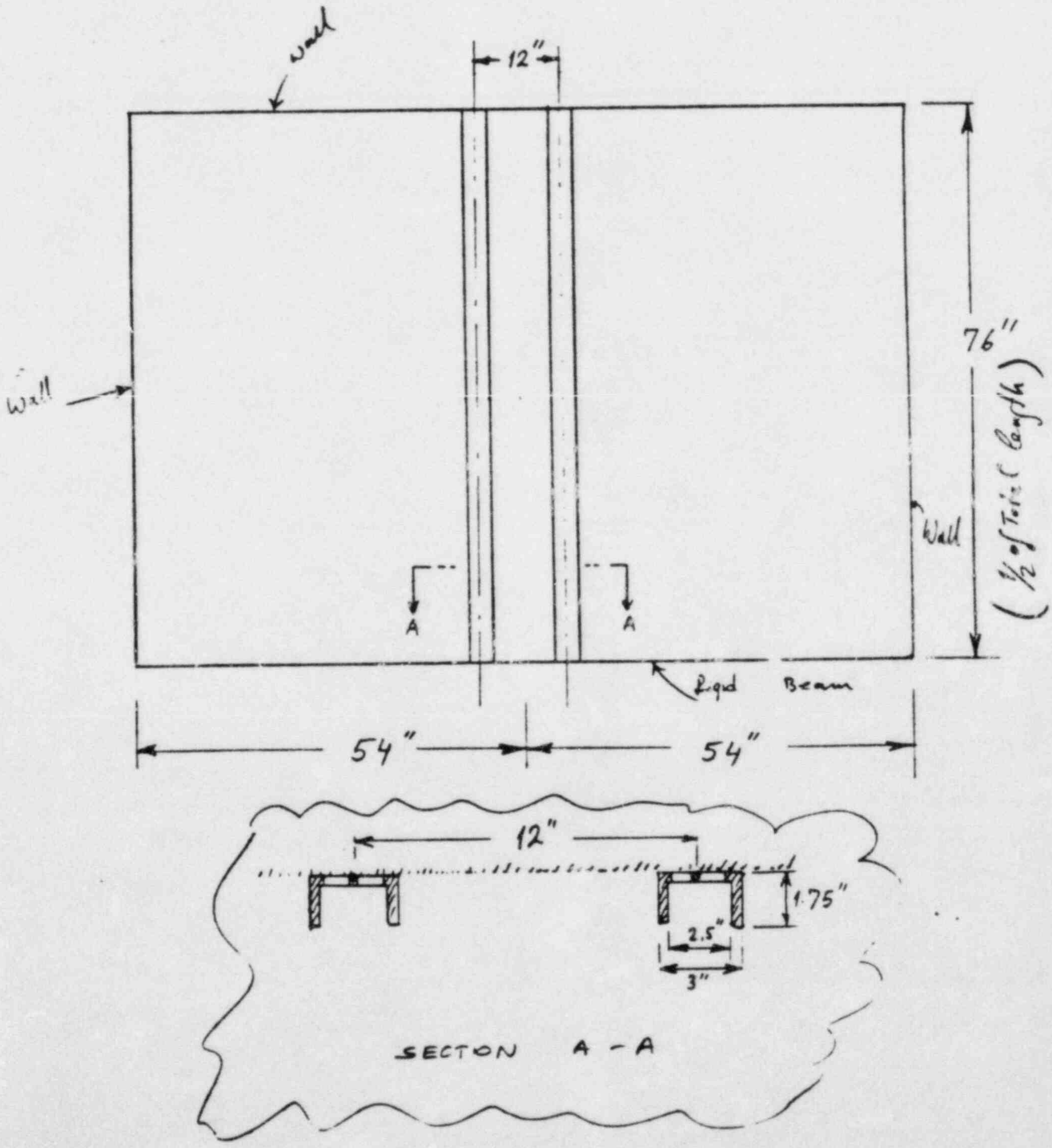
Sketch obtained from field walkdown



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Load distribution on edges of 48" x 76" slab:

Assume:

q_1 = distributed load carried along short span (48")

q_2 = distributed load carried along long span (76")

Compatibility:

$$q = q_1 + q_2$$

$\delta_1 = \delta_2$ (deflections at ½ span)

Assuming simply supported beam strips:

$$\delta_1 = (5 q_1 l_1^4)/384\ EI$$

$$\delta_2 = (5 q_2 l_2^4)/384\ EI$$

$$q_1 l_1^4 = q_2 l_2^4$$

$$q_1/q_2 = \left(l_2/l_1\right)^4 = \left(\frac{76}{48}\right)^4 = 6.28$$

$$q = 7.28 q_2$$
$$q = 1.159 q_1$$

$\Rightarrow$ q_2 = 14% of q

q_1 = 86% of q

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Total weight = 9000 lb.

Weight on 76"x108" section = 4500 lb.

$$q = \frac{4500}{76 \times 108} = 0.548 \ lb/in^2$$

Load per lineal inch of 2 channel sections:

$$q' = 2 \times (0.548 \times 48/2 \times 0.86) + 12 \times 0.548$$
$$+ \ 4 \times 1.75 \times 0.25 \times 0.283$$

$$q' = 29.69 \ lb/in$$

Properties of channel sections (only flanges considered)

$A = 1.75 \times 0.25 \times 4 = 1.75 \ in^2$ (area)

$I = 4 \times \left(\frac{0.25 \times (1.75)^3}{12}\right) = 0.45 \ in^4$ (moment of Inertia

Consider the two channel sections as uniformly loaded simply supported beam and calculate frequencies, deflections & stresses :

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Frequency : (Ref. 3)

$$f_n = \frac{K_n}{2\pi}\sqrt{\frac{E\,I\,g}{q'\,l^4}}$$

$$f_n = \frac{K_n}{2\pi}\sqrt{\frac{30\times10^6\times0.45\times386.4}{29.69\times76^4}}$$

$$f_n = K_n \cdot 0.365$$

1^{st} mode : $K_1 = 9.87 \Rightarrow f_1 = 3.60$ cps

2^{nd} mode : $K_2 = 39.50 \Rightarrow f_2 = 14.43$ cps

Seismic coefficients :

Peak of vertical base spectra will be increased by 50% for multimode participation in both OBE & SSE load cases. (Ref. 2)

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OBE : $a_{OBE} = 1.5 \times 0.21 \times g = 0.315\,g$

SSE : $a_{SSE} = 1.5 \times 0.36 \times g = 0.540\,g$

1 g acceleration will be added to these accelerations to account for weight.

Deflections: (Ref. 3)

$$\delta_{OBE} = \frac{(1.315 \times 29.69) \times 5 \times l^4}{384 \times E \times I}$$

$$\delta_{OBE} = \frac{1.315 \times 29.69 \times 5 \times (76)^4}{384 \times 30 \times 10^6 \times 0.45}$$

$$\delta_{OBE} = 1.25 \text{ in.}$$

Similarly,

$$\delta_{SSE} = \frac{1.540 \times 29.69 \times 5 \times (76)^4}{384 \times 30 \times 10^6 \times 0.45}$$

$$\delta_{SSE} = 1.47 \text{ in.}$$

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Stresses:

At mid-span:

Shear = 0

OBE : Moment $= M_{OBE} = (1.315 \times 29.69) \times (76)^2 / 8$

$M_{OBE} = 28189$ in-lb.

$\sigma_{OBE} = \frac{28189 \times 1.75/2}{0.45}$

$\sigma_{OBE} = 54811$ psi

SSE : Moment $= M_{SSE} = (1.540 \times 29.69) \times (76)^2 / 8$

$M_{SSE} = 33012$ in-lb

$\sigma_{SSE} = \frac{33012 \times 1.75/2}{0.45}$

$\sigma_{SSE} = 64189$ psi

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Stress Limits:

Allowable stress limits for non-ASME equipment supports per Ref. 2, Table 10.3 are:

OBE ⇒ $\sigma_{All} = 0.60\, S_y \times 1.33 \cong 0.8\, S_y$ } Bending

SSE ⇒ $\sigma_{All} = 0.95\, S_y$

S_y = yield stress

Material of the channel sections:
A-36 Structural Steel

$S_y = 36.\text{ksi}$

∴

OBE: $\sigma_{All} = 0.8 \times 36.0 = 28.8$ ksi

SSE: $\sigma_{All} = 0.95 \times 36.0 = 34.2$ ksi

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Since:

$$\sigma_{OBE} = 54811 > 28800 \text{ psi}$$

and

$$\sigma_{SSE} = 64189 > 34200 \text{ psi}$$

The two channel sections do not meet the seismic criteria under the assumed structural behavior.

Modification

A design modification is proposed here to reduce the stresses and the deflections of these critical support members. Modification consists of placing a structural member parallel to the end wall, spanning from side wall to side wall at about the mid-length

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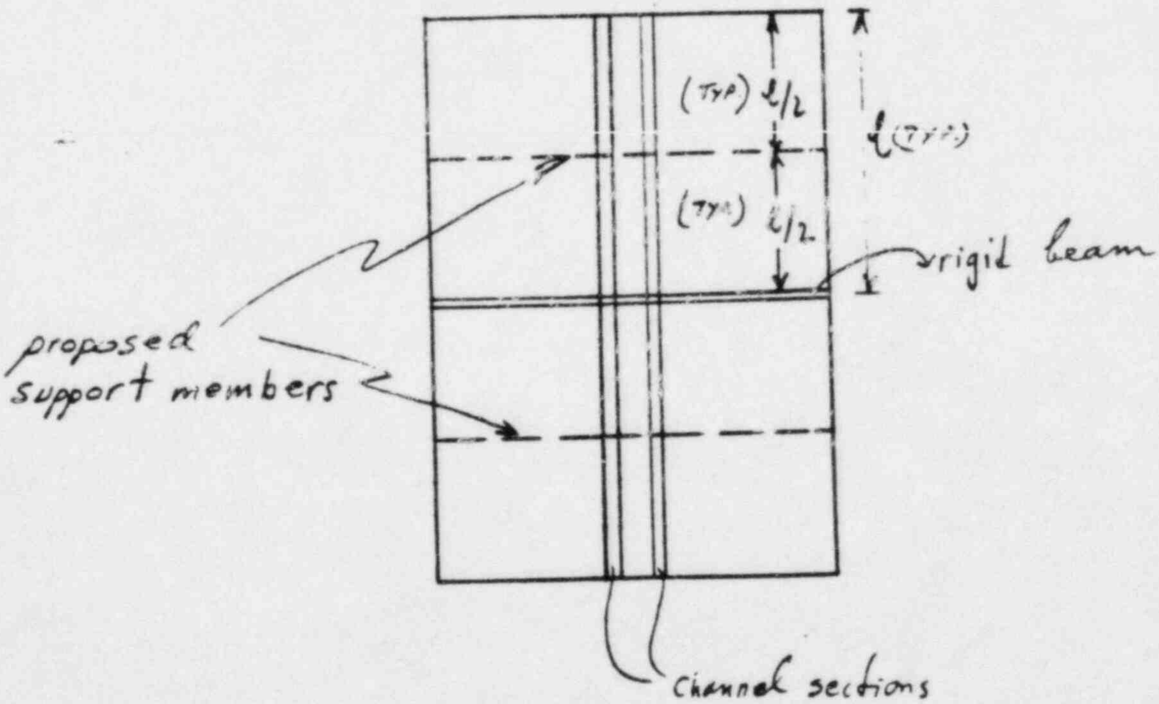
of the 76" span of the two Channel sections. Two such members are needed on both sides of the existing rigid beam. (Refer to the following sketch)



Cross-section determination for proposed support members:

These members will be considered simply

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(17)

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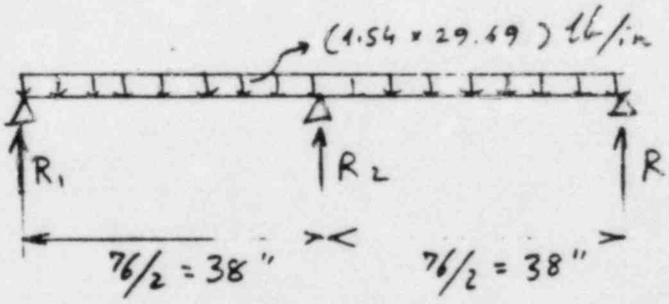
supported members loaded at center with a concentrated load equivalent to the reaction received from the two channel sections.

If propesed members are placed at the midspan of channel sections, channel section beam idealisation will be as follows (USE SSE loading only):



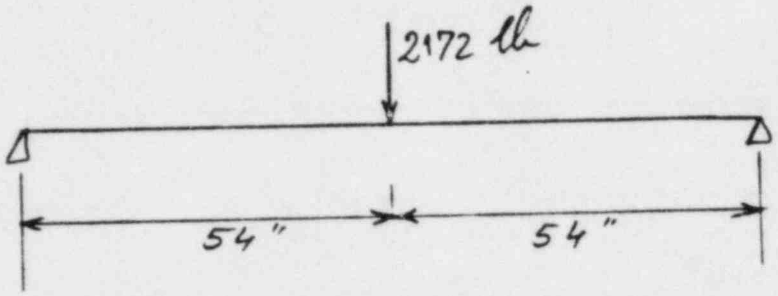
By method of superposition it can be shown that:

$$R_1 = \frac{3 \times (1.54 \times 29.69) \times 38}{8} = 652 \text{ lb}$$

and

$$R_2 = \frac{5 \times (1.54 \times 29.69) \times 38}{4} = 2172 \text{ lb}$$

Accordingly, proposed support member will be idealised as the following beam:



Max. Moment : $M = \frac{2174 \times 108}{4} = 58698$ in-lb

Max Shear : $V = 2174/2 = 1087$ lb

Select a cross-section :

3 in I beam S 3 x 7.5

Area : $A = 2.21\ in^2$

Moment of Inertia : $I = 2.93\ in^4$

Section Mod : $S = 1.95\ in^3$

Stress :

$$\sigma = \frac{M}{S} = \frac{58698}{1.95} = 30102 \text{ psi}$$

$$\tau = \frac{1087}{2.21} = 492 \text{ psi}$$

$$\sigma_{max} = \frac{30102}{2} + \sqrt{\left(\frac{30102}{2}\right)^2 + 492^2}$$

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(SSE) $\sigma_{max} = 30110$ psi

(SSE) $\sigma_{All} = 34200$ psi

$\sigma_{max} < \sigma_{All}$, section OK.

Verify for (OBE):

$$R_2 = \frac{5 \times (1.315 \times 29.69) \times 38}{4} = 1854 \text{ lb.}$$

$$M = \frac{1854 \times 108}{4} = 50072 \text{ in-lb.}$$

$$V = \frac{1854}{2} = 927 \text{ lb.}$$

$$\sigma = \frac{50072}{1.95} = 25678 \text{ psi}$$

$$\tau = \frac{927}{2.21} = 420 \text{ psi}$$

$$\sigma_{max} = \frac{25678}{2} + \sqrt{\left(\frac{25678}{2}\right)^2 + 420^2}$$

(OBE) $\sigma_{max} = 25685$ psi

(OBE) $\sigma_{All} = 28800$ psi

$\sigma_{max} < \sigma_{All}$, section OK

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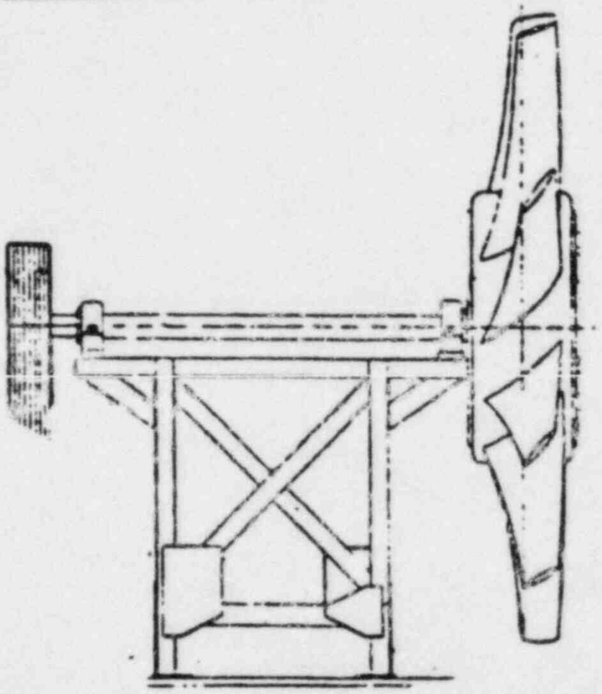
RADIATOR FAN, SHAFT, AND PEDESTAL

LOG NO. 0803101150
LOG NO. 0803101100
LOG NO. 0803101000



Structural integrity of the fan shaft and pedestal will be investigated by calculating stresses at several critical locations. A simple check for operability of the fan shaft will be performed by calculating deflections of the shaft and its supports.

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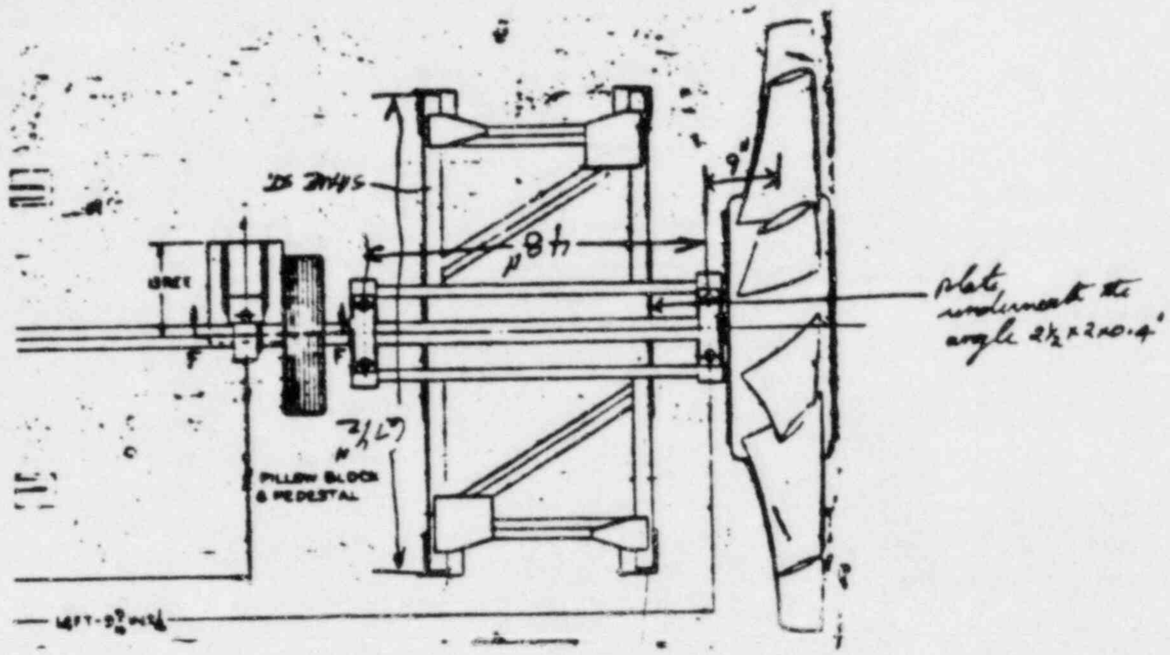

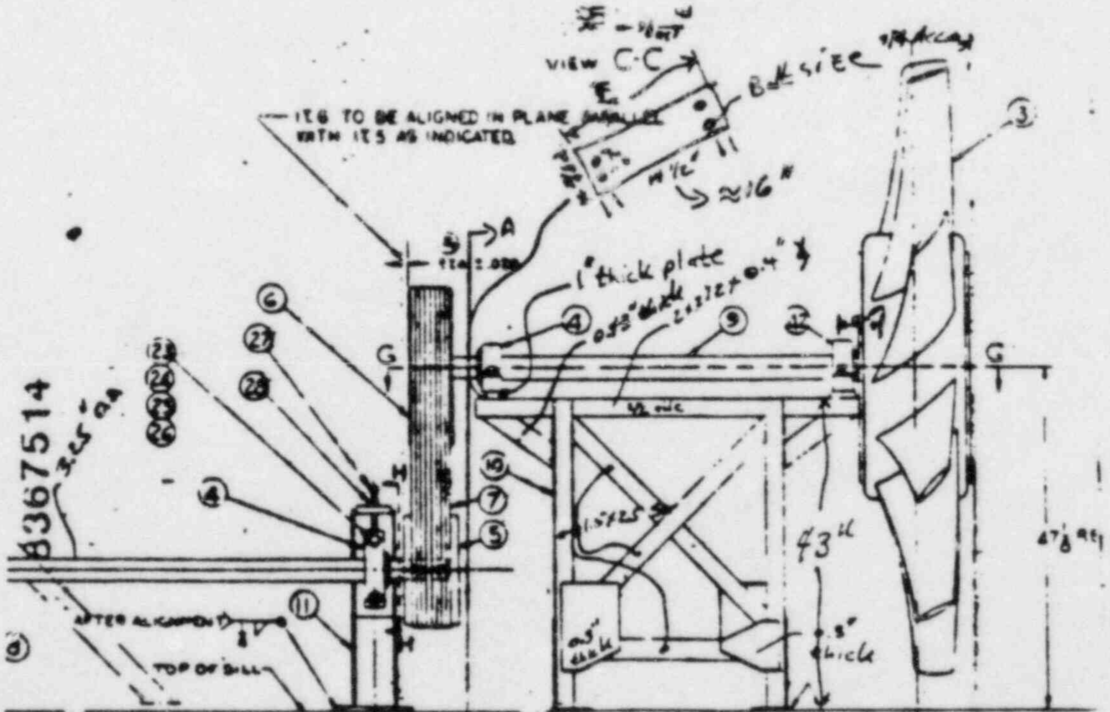

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The fan shaft has bearing at the either end and they are supported by two horizontal 2×2½"×0.4" L's which runs ⊥ to the axis of the fan shaft. {note- dimensions from previous page obtained during walkdown, Drawing is EMD NO. 8367514, Rev. AB4}

OD of the shaft = 3¼"

length of the supported shaft = 48"

weight of the shaft = $\pi \times 3.25^2 \times 0.25 \times 48 \times .283 = 112.69$ lbs

From the referenced telephone conversation memo the weight of the fan = 251 lbs (see attached memo)

total weight = 251 + 112.69 = 363.69 lbs ≃ 370 lbs

To consider the weight of the two bearings and to calculate safely increasing the weight 2½ times as much.

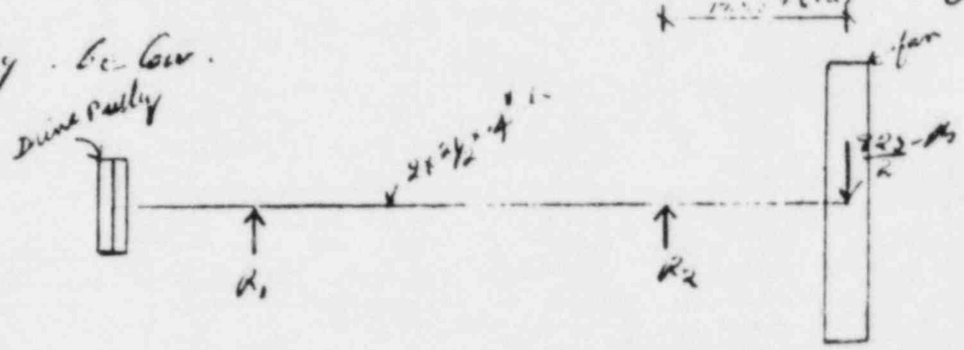
total weight = 370×2.5 = 925 lb

From the reference response spectra peak g value 0.3g using a conservative factor 1.5
the g value = 0.3×1.5 = 0.45g

(See Shoreham base spectrum in Appendix A of this report)

SUPPORT ANGLE STRESSES

The most critical direction for the supporting frame is vertical. Assuming the Horizontal angle as simply supported with the loads concentrated at the end of the overhang portion see fig. below.



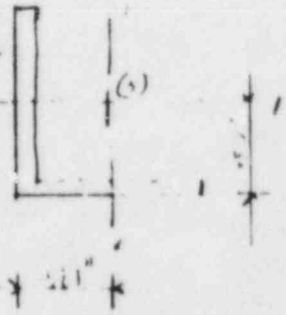
The conservative way of analyzing the above model would be to load the beam at the fan end only, without any load on the pulley end.

Max moment occurs at $R_2 = \frac{925 \times 17.58}{2} = \frac{16261.5}{2} = 8130.75$ in-lb

Considering the angle 4×4×1/4 will be conservative
(All properties of the ∠ from AISC standards)

$I_{11} = 0.91 in^4$

$\sigma_{11} = \frac{16261.5 \times 0.831}{0.912 \times 2} = 7408.38$ psi × 1.45
= 10742 psi



In 2-2 direction, the moment would be ≈ 3659 of proportion of the load in the 3-3 direction.

(8130.75 × .45y) = 3659 in-lbs. Using same moment arm of 17.58 in

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$I_{22} = 0.514$

$$\sigma_{22} = \frac{16261.8}{2 \times 0.514} \times 0.581 = 9190.595 \text{ PSI} \times .45g = 4136 \text{ psi}$$

The torsion is negligible

max tensile stress $= \{10743^2 + 4536^2\}^{1/2} = 11{,}512$ psi

Deflection of the Fan Shaft

cross-sectional area of the shaft $= \frac{\pi \times 3.25^2}{4} = 8.295$ in²

Assuming the same analytical model as for the horizontal support (see figure on B.3.51), that the entire load 925 will be resisted by one single shaft

moment of inertia of the shaft $= \pi \times 3.25^4 \div 64 = 5.475$ in⁴
section modulus of the shaft $= \pi \times 3.25^3 \div 32 = 3.370$ in³
max deflection of the shaft at the over-hanging portion (fan end).

Taken from "Design of Welded Structures" by Blodgett, page 8.1-19, case 6ab:

$$\Delta max \text{ for the over-hang} = \frac{Pa^2}{3EI}(L+a)$$

$a = 17.58''$
$P = 925$ lbs
L = considered to be $(48-17.58)$ bet. center line $= 30.42''$
$E = 29 \times 10^6$
$I = 5.476$ in⁴

$\therefore \Delta max = 0.086''$

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Δmax between the supports = $\frac{PaL^2}{9\sqrt{3}EI}$

= 0.015"

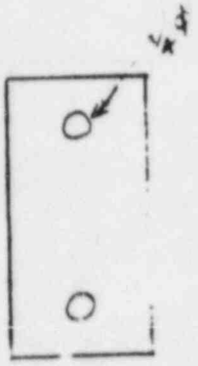
The above deflections are to small to interfere with the operation of the fan.

<u>check on Bearing bolts</u>

cross sectional area of the bolts = $\frac{\pi \times 3/4^2}{4}$

= 0.4417 in²

Tension in the bolts = $\frac{925}{0.4417 \times 2}$ = 1047.09 PSI



shear would be the same

maximum stress = abs sum = 1047.09 PSI x 2 = 2094.18 PSI

It is assumed that the bolts will not see any moments.

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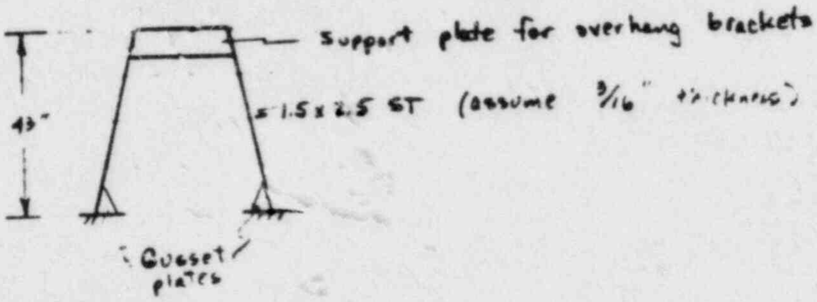
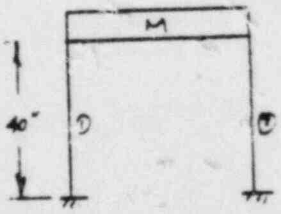
PEDESTAL ANALYSIS

Simplified sketch of Sec. A-A from page B.3.51



The rigidity of this frame section (considered least rigid, since not braced) will be checked by modelling it as shown below: (conservative)



$M = 925$ lb

$I_1 = I_2 = \frac{1}{12}(1.5(2.5)^3 - 1.125(2.125)^3) = 1.05$ in⁴

$E_1 = E_2 = 29 \times 10^6$ psi

$A_1 = A_2 = 1.5(2.5) - 1.125(2.125) = 1.359$

$M_1 = M_2 = (0.283)(1.359)(40) = 15.4$ lbm

From Formulas for Natural Frequency and Mode Shape, by RD Blevins, 1979, page 221:

$$f_1 = \frac{1}{2\pi}\left[\frac{12\,\Sigma E_i I_i}{L^3(M + 0.37\,\Sigma M_n)}\right]^{1/2}$$

$$f_1 = \frac{1}{2\pi}\left[\frac{12(2)(29\times10^6)(1.05)(386.4)}{(40)^3(925 + (0.37)(2)(15.4))}\right]^{1/2} = 10.9 \text{ Hz}$$

Since this conservative model shows the fundamental frequency of the frame is above 10 Hz, where the Shoreham base response spectra accelerations are low (approaching ZPA), the frame is considered rigid and structurally adequate to withstand seismic loadings.

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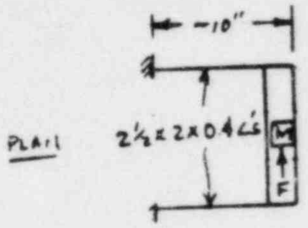
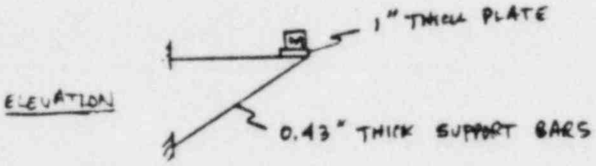
FAN-SIDE PILLOW BLOCK SUPPORT

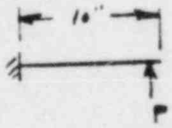
Since the pedestal frame acts rigidly, the support can be analyzed as shown below:



$M = \frac{925}{2} \doteq 463$ lb to account for weight of fan, one bearing, and structural elements.

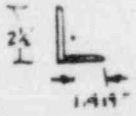


The critical direction will be for the force F shown in the plan view
Considering the 2 angles acting as one element (conservative):



$I = 2I_L = 2(0.514) = 1.028 \text{ in}^4$

$A = 2A_L = 2(1.55) = 3.1 \text{ in}^2$



F will be the seismic force, assume the mass vibrates flexibly

$a_{horiz} = 0.55g$ (peak response spectra value)

$F = 1.5(0.55)(463) = 382$ lb

$M_{max} = (382)(10) = 3820$ in-lb $\qquad \sigma_{max} = \frac{(3820)(1.419)}{1.028} = 5273$ psi OK

$\Delta_{max} = \frac{PL^3}{3EI} = \frac{(382)(10)^3}{3(29\times10^6)(1.028)} = .0043''$ Deflections are minimal and will not affect fan operation

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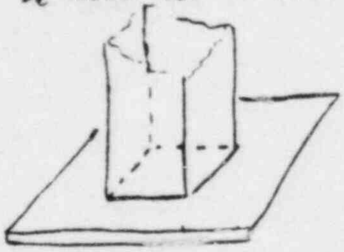
check on the welds

The critical most weld would be the junction of 2x2½x¼" Horizontal angle and vertical tube. This weld need not be checked because there is a vertical plate is also welded underneath the angle 2x2½x.¼", which takes most of the loadings. see figures in Page 2

check on the weld at the bottom of the four slanting vertical legs.



Since the structure is very sturdy and the load is [illegible]

it is judged all the welds would be sufficient to withstand the applied loading.

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**MEMORANDUM OF
TELEPHONE CONVERSATION**

Date: 5/15/84

Time: 9:55 AM

Person Called: Morice Goza (Name) of OM manufacturing company (Company)

Person Calling: S. JAGADEESAN (Name) of SARGENT & LUNDY ENGRS (Company)

Project: **Project No.** 6995-00

Subject Discussed: Regarding the weight of Radiator fan for the Diesel generator, fan model # 8-108-AL

Summary of Discussion, Decisions and Commitments:

Mr. GOZA informed that the weight of the ~~the~~ fan to be 251 lbs.

cc:

S Jagadeesan
Signature

File:

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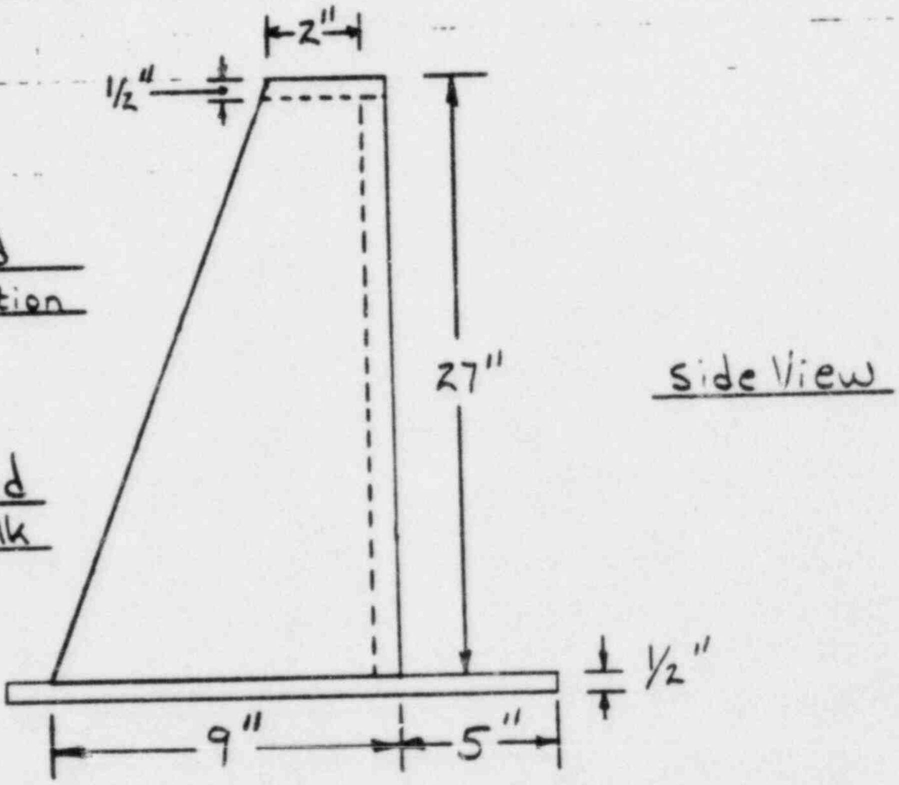
Prepared by David Wright Date 5/11/84
Reviewed by Date
Approved by Date

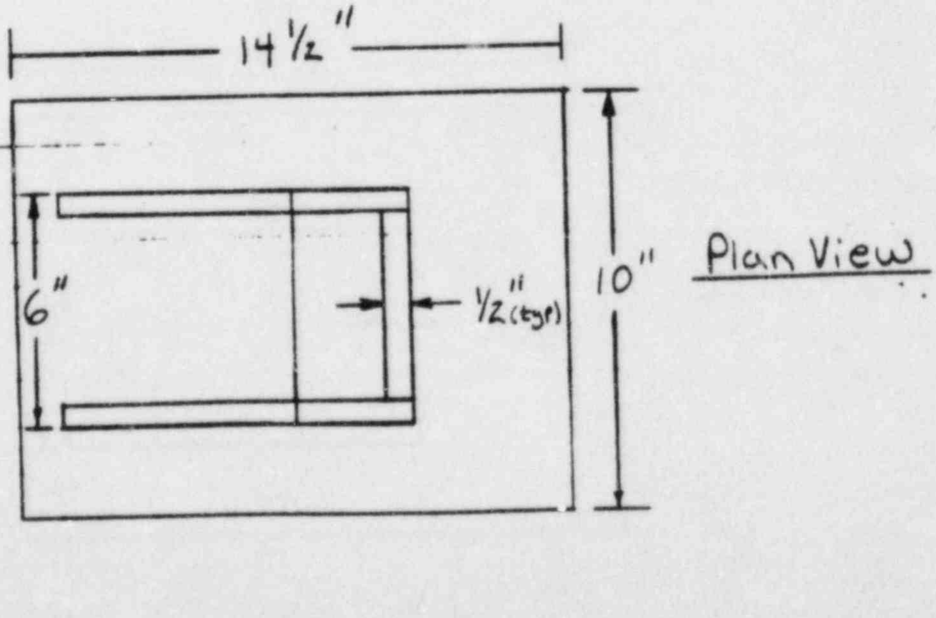
Forward Fan Shaft Bracket

Log No. 0803101140

Note- all welded construction

Sketch obtained during field walk-down





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LOWER FAN-DRIVE SHAFT

The shaft will be analyzed as a 9 ft span, 3.25" O.D beam.

$A_{shaft} = 8.295 \text{ in}^2$
$I = 5.476 \text{ in}^4$ } from fan shaft analysis
$S = 3.370 \text{ in}^3$

$W_{shaft} = (9)(12)(8.295)(0.253) = 254 \text{ lb}$ $w = \frac{254}{108} = 2.35 \text{ lb/in}$

Considering the shaft as simply supported for conservatism:

∴ Natural frequency: $f_1 = \frac{9.87}{2\pi}\sqrt{\frac{(29\times10^6)(5.476)(386.4)}{(2.35)(108)^4}} = 21.8 \text{ HZ}$

Seismic coefficients (@ 21.8 Hz)
(see base spectrum for Shoreham in appendix A of this report.)

$a_{vert} = 0.18g + 1.0 = 1.18g$
$a_{horiz} = 0.26g$
$a_{res} = \sqrt{a_{vert}^2 + a_{horiz}^2} = 1.21g$

$\omega = (2.35)(1.21) = 2.84 \text{ lb/in}$

$M_{max} = \frac{wl^2}{8} = \frac{(2.84)(108)^2}{8} = 4141 \text{ in-lb}$

$\sigma_{max} = \frac{M_{max}}{S} = \frac{4141}{3.370} = 1229 \text{ psi}$

$\Delta_{max} = \frac{5wl^4}{384EI} = \frac{5(2.84)(108)^4}{384(29\times10^6)(5.476)} = .032 \text{ in}$

Since the shaft center deflection is so small, no misalignment or operational problems will occur.

By comparing the overhanging portion of the lower shaft with that of the fan shaft, the lower shaft case is less severe and can be assumed adequate based on the fan shaft analysis done previously.

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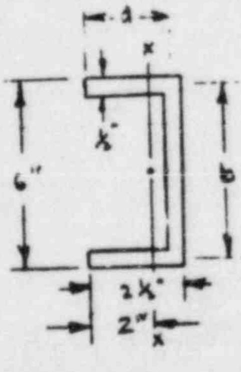
LOWER FAN-DRIVE SHAFT PILLOW BLOCK AND PEDESTAL

Assuming the pillow block and pedestal resist one-half of the lower shaft loads plus one-half the total load attributed to the fan shaft and end drives.

$$F_{,} = \frac{1}{2}\left[(1.21)(254 + 925)\right] = 713 \text{ lb}$$

This is lower than the load used to analyze the fan shaft pillow block, so the lower shaft pillow block can be qualified by similarity.

The pedestal is constructed of ½" thick plates welded together as shown on the first page of this section. For simplicity and conservatism, consider the pedestal as a C-channel as shown below.



from "Design of Welded Structures" by Blodgett Page 2.2-8,

$$I_{xx} = \frac{td^3(2b+d)}{3(b+2d)} = \frac{(0.5)(2.25)^3(2(5.5)+2.25)}{3(5.5+2(2.25))} = 2.51 \text{ in}^4$$

$$c = 2.5 - \frac{(2.25)^2}{5.5+2(2.25)} = 2.0$$

Applying all of the above resultant force at the top of the pedestal perpendicular to x-x:

$$M_{max} = (713 \text{ lb})(27 \text{ in}) = 19251 \text{ in-lb} \text{ at the pedestal base}$$

$$\sigma_{max} = \frac{M_{max}\,C}{I_{xx}} = \frac{(19,251)(2.0)}{2.51} = 15,339 \text{ psi}$$

Assuming A-36 structural steel ($S_y = 36$ ksi) $\sigma_{allow} = \frac{4}{3}(0.6)(36) = 28.8$ ksi

Since $\sigma_{max} = 15.399$ ksi $< \sigma_{allow} = 28.8$ ksi, the structural integrity of the pedestal is shown.

Form GQ-3.08.1 Rev. 2

SARGENT & LUNDY
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Calcs. For Shutter Linkage Assem

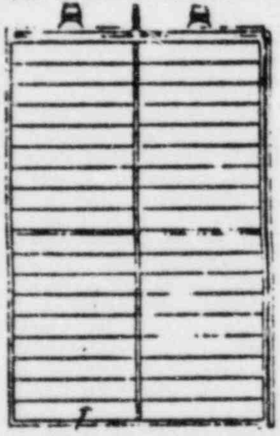
☒ Safety-Related ☐ Non-Safety-Related

Client LILCo	Prepared by / Date
Project Shoreham-1	Reviewed by / Date
Proj. No. 6995-00 Equip. No.	Approved by / Date

Radiator Shutter Assembly

Log # 0803101110

Note: The conclusions reached below are based on a physical inspection of the equipment, pictures and Drawings provided by Morrison Knudeson Co. These drawings contain similar equipment and may not be the exact system at Shoreham. This information should be obtained from another field walkdown.



The shutter linkage assembly on the Shoreham-1 E.D.G.S.'s is a device used for regulating the intake air across the radiator. This assembly is controlled by the MP9750 actuator (based on M-K dwg's).

By comparison of this assembly (from field trip) with similar HVAC dampers used in the nuclear power industry and through past experience with these structurally similar dampers we are confident that the shutter linkage assembly is seismically adequate as designed for maintaining its structural integrity.

Note - A simple analysis should be performed to substantiate the conclusions reached above.

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Calcs. For Shutter Actuator

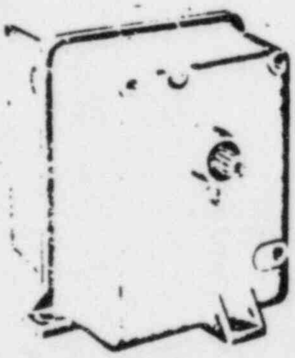
☒ Safety-Related ☐ Non-Safety-Related

Client LILCo
Project Shoreham-1
Proj. No. 6995-00 Equip. No.

Prepared by Date 5-31-8
Reviewed by Date
Approved by Date

Shutter Actuator

Log # 0803101120



MF. : Barber Colman Co.
Model # : MP9750

Note: Morrison-Knudsen Co. supplied LILCo with representative drawings for similar E.D.G.S. systems. This device is indicated on M-K dwg #6041-F04001, rev. B and is considered to be applicable here. Positive identification should be obtained.

The shutter actuators located on the Shoreham MP-45 Generators are used to open/close the shutter assemblies that are located in front of the radiators. The E.D.G.S. engine temperature is monitored by the actuator's microtherm control circuit. The appropriate electrical signal is sent to the actuators internal transformer where the voltage input is stepped up. This higher voltage is converted to a high torque on the splined output shaft via a gear reduction system. Due to the very compact construction no deflection of the gear shafts is expected. For this reason the mounting bolts will be analyzed as the most critical mode of failure.

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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client		Prepared by	Date
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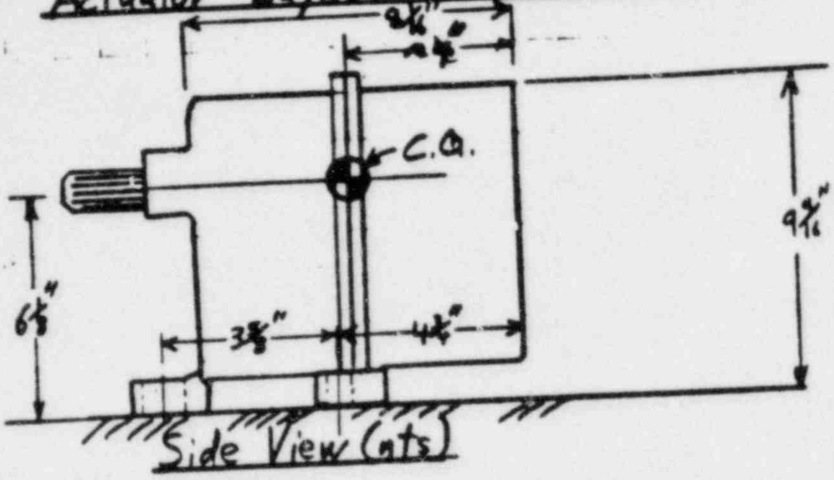
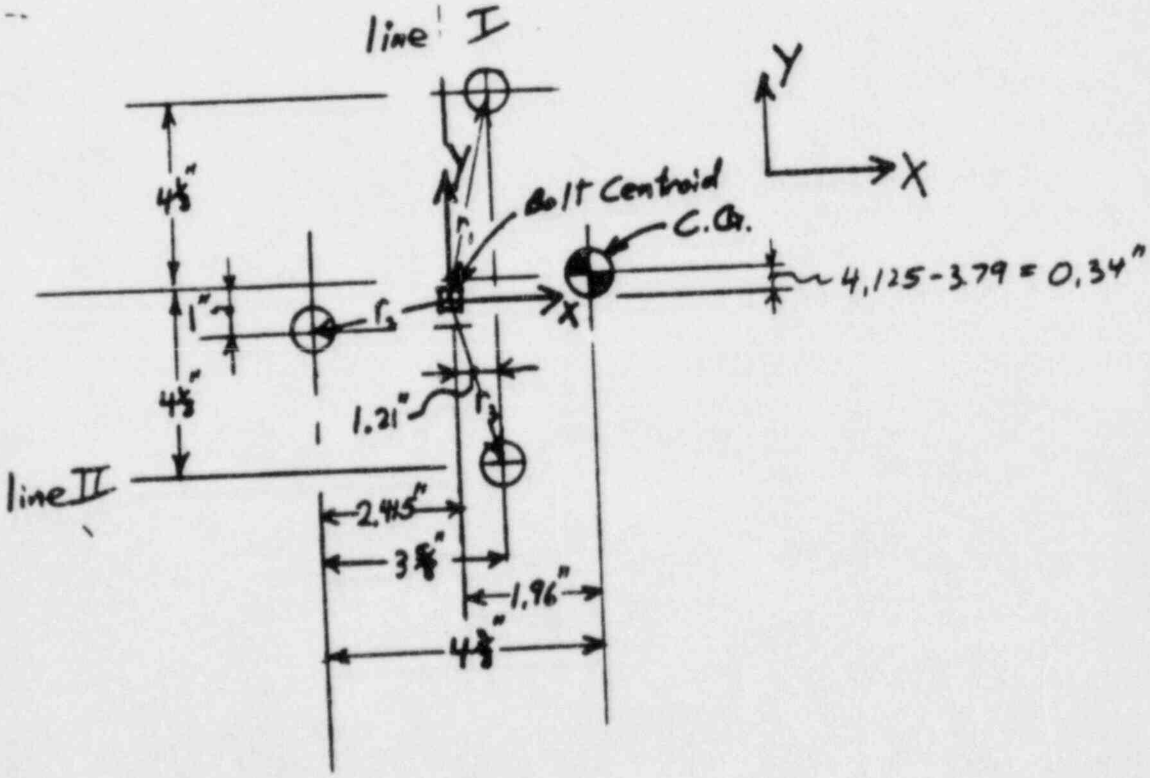
Proj. No.	Equip. No.	Approved by	Date

Actuator Layout - C.G. & Bolt Centroid:



Side View (nts)



Note- Data Obtained from Barber-Coleman GENERAL INSTRUCTIONS NO. F-11331-5 for High Torque Oil Submerged Actuators MC, MF, MP-9000.

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Calcs. For

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Client		Prepared by	Date
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Bolt Centroid Location:

$$\Sigma M_{line\ I} : \bar{x} = \frac{2A(0) + A(3.625)}{3A} = \underline{\underline{1.21''}}$$

$$\Sigma M_{line\ II} : \bar{z} = \frac{A(3.125) + A(8.25)}{3A} = \underline{\underline{3.79''}}$$

∴ C.G. - Bolt Centroid Eccentricities Are:

$\underline{\underline{e_x = 1.96''}}$ $M_x = F_y e_z + F_z e_y$

$\underline{\underline{e_y = 0.34''}}$ $M_y = F_x e_z + F_z e_x$

$\underline{\underline{e_z = 6.125''}}$ $M_z = F_y e_x + F_x e_y$

Dead Wt Loads:

$F_x = F_y = M_z = \underline{\underline{0.0}}$

$F_z = \underline{\underline{-100\ lb}}$ (conservatively assumed)

$M_x = 100(0.34) = \underline{\underline{34\ in\text{-}lb}}$

$M_y = 100(1.96) = \underline{\underline{196\ in\text{-}lb}}$

Calc. For

☒ Safety-Related ☐ Non-Safety-Related

Client	Prepared by / Date
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$$\tau_y = \frac{F_y}{A_T} + \frac{M_z(1.21)}{J_{zz}}$$

$$\tau_y = \frac{150}{0.33} + \frac{345(1.21)}{5.14} = \underline{\underline{536\,psi}}$$

$$\tau_{max} = \sqrt{(\tau_x)^2+(\tau_y)^2} = \sqrt{(754)^2+(536)^2} = \underline{\underline{925\,psi}}$$

2. Tension –

Note: Bolt tensile failure due to M_y would occur by the pivoting about the 2-bolt centerline. This is the worst possible failure due to M_y.

$$\sigma_z = \frac{F_z}{A_T} + \frac{M_x(c_y)}{I_{xx}} + \frac{M_y(c_x)}{I_{yy}} \quad : \; c_x = 3.625\,in \quad c_y = 4.47\,in$$

where: $I_{xx} = 0.11[(4.125+0.34)^2+(4.125-0.34)^2+(1-0.34)^2]$

$$I_{xx} = \underline{\underline{3.82\,in^4}}$$

$$I_{yy} = 0.11(3.625)^2 = \underline{\underline{1.45\,in^4}}$$

$$\therefore \sigma_z = \frac{(150-100)}{3(0.11)} + \frac{(34+969.8)(4.47)}{3.82} + \frac{(196+1,212.8)(3.625)}{1.45}$$

$$\sigma_z = \underline{\underline{4,848\,psi}}$$

Max. Seismic Loads: (using 1.0g as peak – envelopes Shoreham R.S.)

$F_x = F_y = F_z = 100 \text{ lb} (1.0g)(1.5) = 150 \text{ lb}$

$M_x = 150(0.34 + 6.125) = 969.8 \text{ in-lb}$

$M_y = 150(1.96 + 6.125) = 1,212.8 \text{ in-lb}$

$M_z = 150(1.96 + 0.34) = 345.0 \text{ in-lb}$

Bolt Pattern Stress Analysis:

1. Shear –

$r_1 = \sqrt{(1.21)^2 + (4.125 + 0.34)^2} = 4.626''$ $A_{\frac{3}{8}''\phi} = 0.110 \text{ in}^2$ (AISC, 8th ed. pg. 4-141)

$r_2 = \sqrt{(1)^2 + (2.415)^2} = 2.614''$

$r_3 = \sqrt{(1.21)^2 + (4.125)^2} = 4.299''$

Torsional Moment of Inertia $= J_{zz} = 1(0.110)[(4.626)^2 + (2.614)^2 + (4.299)^2]$

$J_{zz} = \underline{\underline{5.14 \text{ in}^4}}$

$$\tau_x = \frac{F_x}{A_T} + \frac{M_z(4.125 + 0.34)}{J_{zz}}$$

$$\tau_x = \frac{150}{3(0.11)} + \frac{(345)(4.465)}{5.14} = \underline{\underline{754 \text{ psi}}}$$

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CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
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FINAL

Client		Prepared by	Date
Project		Reviewed by	Date
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Bolt Pattern Acceptance Criteria: From ASME B+PV code, 1977 edition, 71 summer addendum, section III, article 2460 of Appendix XVII;

1st Condition: $\sigma_z = 4.85 ksi \le 0.33 S_u$: $S^{*}_{u_{Bolt}}$ for A-36 = 58.0 ksi

$4.85 \le 0.33(58.0)$

$\underline{\underline{4.85 \le 19.14}}$ ✓ Adequate

2nd Condition: $\sigma_z = 4.85 \le 0.43 S_u - 1.8 \tau_{max}$

$\underline{\underline{4.85 \le 23.28}}$ ✓ Adequate

Conclusion:

The shutter actuator hold-down mounting bolt pattern has been stress analyzed and is adequate as designed for the maximum postulated seismic loading, using a very conservative methodology.

* From ASME B+PV code, 1977 edition, Section III, table I-7.1 of Appendix I.

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B.4

B.4 ELECTRIC START SYSTEM

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Calcs. For

☒ Safety-Related ☐ Non-Safety-Relat

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
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PAGE B.4.2

Client		Prepared by	Date
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BATTERIES

LOG. NO. 0803220800

The batteries are located on both sides of the diesel generator very close to the base of the skid. Seven batteries are contained on each side in a strongly reinforced tray attached to the diesel generator skid. Due to rigidity and location of the battery tray, the seismic input will be ZPA values from the base response spectra.

Horizontal seismic accelerations are approximately 0.22g, while vertical accelerations are approximately 0.15g. Since vertical seismic accelerations are well below 1.0g, no bouncing or impact loading will occur. S&L suggests that the spaces between and around the batteries be filled with rubber pads or other energy-absorbing material to prevent bumping due to horizontal movement.

With the horizontal movement of the batteries restricted as described above, it can be judged that the low seismic input will not affect structural integrity or operability of the batteries.

Form GQ-3.08.1 Rev. 2

Calcs. For

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Client		Prepared by	Date
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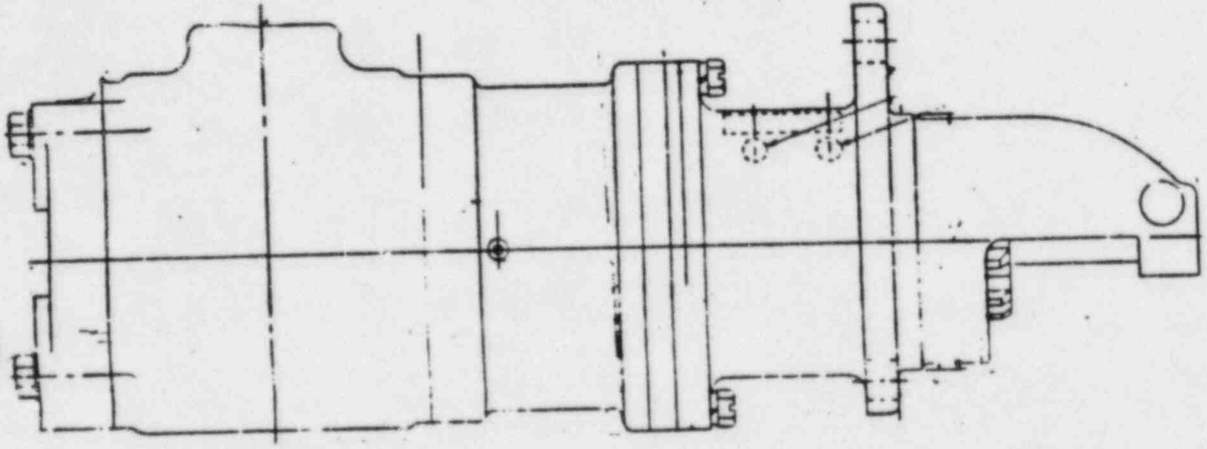
Electric Starter Motors

Log. Number 0803220100



MF: Delco Remy
Model #: 1109758

The electric start motors on the Shoreham E.D.G.S.'s are used for initiating the diesel engine running cycle just prior to generation of on-site power by the E.D.G.S's.

The analysis required to ensure structural integrity and operability of these motors is accomplished by examining the most probable modes of failure. This analysis includes stress analysis of the motor's hold-down bolt pattern and operability check via a deflection analysis using conservative parameters.

note - Data requested from Delco-Remy was not obtained in time to use in this analysis. All of this work is based on conservative assumptions and should be compared to the actual data to substantiate the analysis

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ENGINEERS
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Calcs. For

☒ Safety-Related ☐ Non-Safety-Rela

CALC NO. CQD-014046
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PAGE B.4.4

Client
Project
Proj. No. Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

Structural Integrity Check:

THE BOLT PATTERN FOR THE ELECTRIC START MOTORS IS SHOWN IN THE FIELD SKETCHES (SHOWN IN THIS SECTION) FROM PICTURE #32 & 33 WE CAN CONSERVATIVELY ASSUME THE *CENTER OF GRAVITY IS 12 INCHES FROM THE BOLT CENTROID IN THE AXIAL DIRECTION. THE MEASURED DISTANCE ACROSS THE FLATS OF THE BOLTS WERE 0.94" WHICH CORRESPONDS TO A 5/8 INCH BOLT (Machinery's Handbook, 12th edition). FROM THE EMD PARTS CATALOGUE FOR SERVICE DATA ENGINE INFORMATION (ATTACHED) WE NOTE THAT THE STARTING MOTOR (ELECTRIC) WEIGHS 78 LBS AND ITS MOUNTING BRACKET IS 58 LBS

THE MEASURED DISTANCE ACROSS THE FLATS OF THE BRACKET TO ENGINE BOLTS WERE 1.75 IN WHICH CORRESPONDS TO A ~1 1/8 INCH BOLT CONSERVATIVELY

* NOTE: WE WILL ALSO CONSERVATIVELY ASSUME THE C.G IS 2 INCHES OFF CENTER DUE TO THE ELECTRIC WIRE HOOKUP ATTACHMENT ON THE MOTOR (SEE PICTURES 32 & 33).

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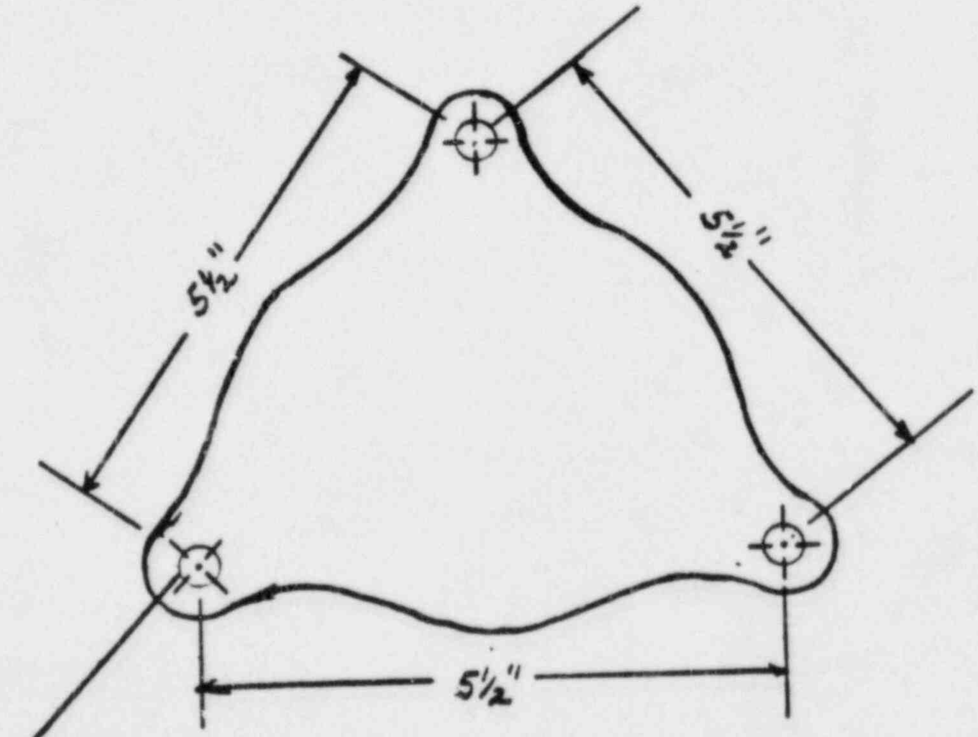
Calcs. For FIELD SKETCHES

☒ Safety-Related ☐ Non-Safety-Rela

Client
Project
Proj. No. Equip. No.

Prepared by David Wright Date 5-4-84
Reviewed by Date
Approved by Date

ELECTRIC STARTER MOTOR BOLT PATTERN



(3) SAE GD 5 BOLTS
0.94" ATF

Form GQ-3.08.1 Rev. 2

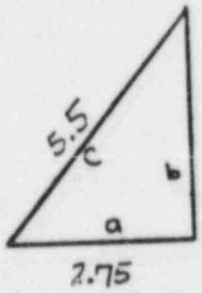
BOLT CENTROID ANALYSIS

$$AREA = \frac{\pi}{4}(5/8\ IN)^2 = 0.3068\ IN^2$$ (based on outside diameter.)

$$\Sigma AREA = 3 \times 0.3068\ IN^2 = 0.920\ IN^2.$$

Z_0 FROM TOP BOLT

$$= \left\{0.3068\,(4.763)\,2\right\}\frac{1}{0.920} = 3.176$$



$$\frac{5.5}{2} = 2.75$$

$$a^2 + b^2 = c^2$$

$$(2.75)^2 + b^2 = (5.5)^2$$

$$b = \left[(5.5)^2 - (2.75)^2\right]^{1/2} = 4.763$$

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Client		Prepared by	Date
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WEIGHT LOADS

$F_z = 78$ LBS

SEISMIC LOADS

BASED ON THE ENVELOPED RESPONSE SPECTRA FOR THE DIESEL GENERATORS LOCATED IN LASALLE & CLINTON STATIONS THE PEAK VALUES ARE LISTED BELOW, IN ADDITION A RECOMMENDED SEISMIC COEFFICIENT OF *1.5 PER IEEE-1975 WILL BE INCLUDED

ENV [HORZ (OBE @ 2% & SSE @ 3%)] = 1.8g (1.5) ≈ 2.8 g's

ENV [VERT (OBE @ 2% & SSE @ 3%)] = 4.75g (1.5) = 7.1g's

note- these values have been shown to be higher than
the shoreham levels (in appendix A of this report.)

$F_x = 78$ LBS $(2.8g) = 218.4$ LBS
$F_y = 78$ LBS $(2.8g) = 218.4$ LBS
$F_z = 78$ LBS $(7.1g) = 553.8$ LBS
$M_x = F_y\cancel{(z)}^{0} + F_z(y) = 553.8$ LBS (12 IN) $= 6645.6$ IN-LBS

$M_y = F_x\cancel{(z)}^{0} + F_z(x) = 553.8$ LBS (2.0 IN) $= 1107.6$ IN-LBS

$M_z = F_x(y) + F_y(x) = 218.4$ LBS (12 IN) $+ 218.4$ LBS (2.0 IN)
$= 3057.6$ IN-LBS

Form GQ-3.08.1 Rev. 2

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ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Re

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TOTAL LOADS

$F_x = 218.4$ LBS

$F_y = 218.4$ LBS

$F_z = 78$ LBS $+ 553.8$ LBS $= 631.8$ LBS

$M_x = 6645.6$ IN-LBS

$M_y = 1107.6$ IN-LBS

$M_z = 3057.6$ IN-LBS

It should be noted that the electric motor torque was not available so no operating loads will be included in this analysis. The gear loading due to the operating loads will cause a shear load on the bolts. It will be shown that the seismic levels of shear stress are very small therefore would not cause any significant increase in the total shear stress due to seismic + startup torque.

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

Client		Prepared by	Date
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MOUNTING BOLT ANALYSIS

$$I_{zz} = 2(.3068\,IN^2)(2.75\,IN)^2 = 4.64\ IN^4$$

$$I_{xx} = 0.3068\ IN^2\ (3.176\,IN)^2 + 2(.3068\,IN^2)(1.587\,IN)^2$$
$$= 4.64\ IN^4$$

$$J = I_{zz} + I_{xx} = 4.64\ IN^4 + 4.64\ IN^4 = 9.28\ IN^4$$

TENSILE STRESS

$$\sigma_t = \frac{Fy}{\Sigma A} + \frac{Mx\, Cz_{max}}{I_{xx}} + \frac{Mz\, Cx_{max}}{I_{zz}}$$

$$= \frac{218.4\,LBS}{0.920\,IN^2} + \frac{6645.6\,IN\,LBS\,(3.176\,IN)}{4.64\,IN^4} + \frac{3057.6\,IN\,LBS\,(2.75\,IN)}{4.64\,IN^4}$$

$$= [237.4 + 4548.8 + 1812.2]\,PSI = 6598.4\,PSI$$

SHEAR STRESS

$$\tau_z = \frac{Fz}{\Sigma A} + \frac{M_y\, C_{z\,max}}{J} = \frac{553.8\,LBS}{0.92\,IN^2} + \frac{1107.6\,IN.LBS\,(3.176\,IN)}{9.28\,IN^4}$$

$$= 981.\ psi$$

$$\tau_x = \frac{Fx}{\Sigma A} + \frac{M_y\, C_{z\,max}}{J} = \frac{218.4\,LBS}{0.92\,IN^2} + \frac{1107.6\,IN.LBS\,(2.75\,IN)}{9.28\,IN^4}$$

$$= 565\ PSI$$

$$\tau_{MAX} = \left[\tau_z^2 + \tau_x^2\right]^{1/2} = \left[(981)^2 + (565)^2\right]^{1/2}$$

$$= 1132\ psi$$

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SARGENT & LUNDY
ENGINEERS
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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

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Check, ASME

Allowable Stress

From ASME B+PV CODE, 1977 edition, 79 Summer Addendum, section III, article 2460 of Appendix XVII ;

$F_t = 0.3 S_u = 0.3(\overset{*}{1}20{,}000 psi) = 36{,}000 psi$ (service level A+B)

$F_v = 0.124 S_u = 0.124(\overset{*}{1}20{,}000 psi) = 14{,}880 psi$ (")

for Combined tension and shear:

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \le 1.00$$

$$\frac{6598^2}{36{,}000^2} + \frac{1132^2}{14{,}880^2} = 0.039 \le 1.00$$ so the bolts are adequate.

* from ASME B+PV Code, 1977 edition, section III, table I-7.3 of Appendix I, page 98 for SA-449.

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SARGENT & LUNDY
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Calcs. For

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Proj. No.	Equip. No.	Approved by	Date

Note - Redundant Check, Not needed!

SATISFY THE FOLLOWING 2 CONDITIONS

EMD FILE NO
EMD-029453
DATED APRIL 3, 1981

1) $\sigma_t \le .44\ S_u$

2) $\sigma_t \le .57 S_u - 1.8\ \tau_{max}$

$S_u = 120$ KSI FOR SA-449 BOLTS (SAE GR5)
< 1 INCH DIAMETER
ASME SECT. III DIV. 1. APPENDICES
APPENDIX I TABLE I-7-3
SUMMER - 1983

$S_a = 23.0$ KSI

1) $\sigma_t \le .44\ (120)$ KSI

6598 PSI < 52,800 PSI OK

2) 6598 PSI $\le$ .57(120) KSI $-$ 1.8(3095.4) PSI

6598 PSI $\le$ 62,828 PSI OK

ALSO
MAX PRINCIPAL STRESS = 6787 PSI < ALLOWABLE STRESS = 23,000 PSI

OK.

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ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Rel

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Project
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Prepared by Date
Reviewed by Date
Approved by Date

Operability Check:

The possibility of the starting motor pinion gear binding against the diesel engine flywheel is examined in this section. The maximum possible motor deflection is conservatively calculated for the motor-side of the hold-down bolts and then converted to the corresponding angular displacement of the pinion gear side.

This analysis is conservative because:

1. Max. seismic accelerations of 2.8g horizontally and 7.1g vertically, corresponding to the enveloped Clinton/LaSalle response spectra, are utilized.
2. The motor weight used in the analysis corresponds to that of the motor and mounting bracket.
3. Minimal motor casing wall thickness is assumed.

Form GQ-3.08.1 Rev 2

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety–Related ☐ Non–Safety–Relat

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Client

Project

Proj. No. Equip. No.

Prepared by Date

Reviewed by Date

Approved by Date

Model:

$$W_{max} = F_z = F_{z_{dead}} + F_{z_{seismic}} = (78 + 58 \text{ lb}) + 7.1(78+58) = 1{,}102 \text{ lb}$$

$$\Delta_{max} = \frac{WL^3}{3EI} \quad , \quad E = 30\times10^6 psi \text{ (Carbon Steel)}$$

$$I = \frac{\pi}{64}\left[(3.25)^4 - (3.0)^4\right] = 1.50 in^4$$

$$\Delta_{max} = \frac{1{,}102(12)^3}{3(30\times10^6)(1.50)} = \underline{0.014''}$$

$$\therefore \theta = \frac{\Delta}{L} = \frac{0.014}{2.5} = 0.06°$$

{This is a very small amount of deflection and will not bind the motor shaft.}

Note – this is a very conservative analysis based on assumptions that are much more severe than the actual data.

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REV. 00 DATE 6/1/84
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CHICAGO

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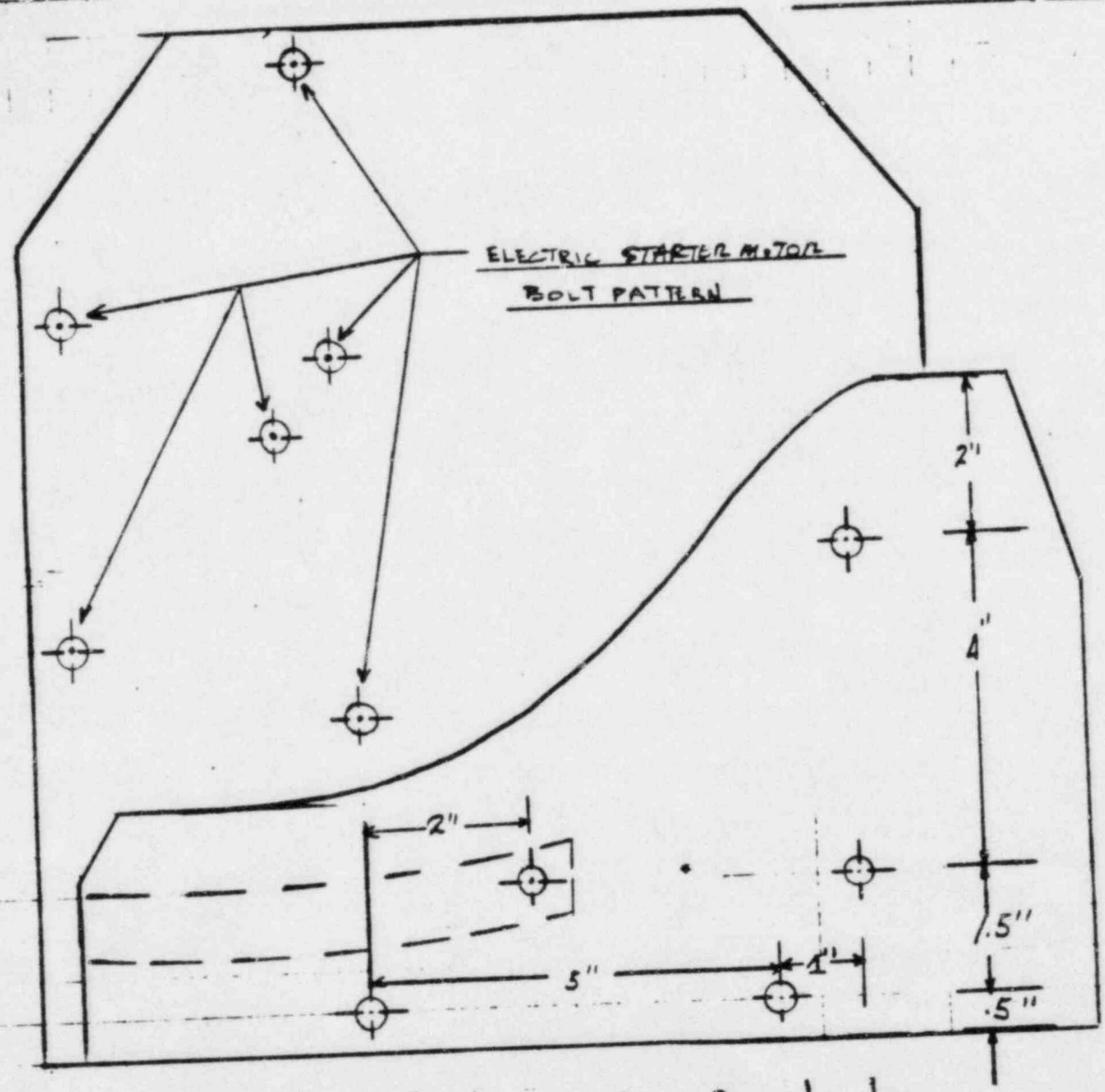
Client		Prepared by	Date 5/4/84
Project		Reviewed by	Date
Proj. No.	Equip. No.	Approved by	Date



Electric Starter Motor Brackets

Log Numbers 0803220600, 0700

sketch obtained during field walkdown.

Note - Bracket to Engine bolts (5) are SAE GD5
and 1.75" across the flats.

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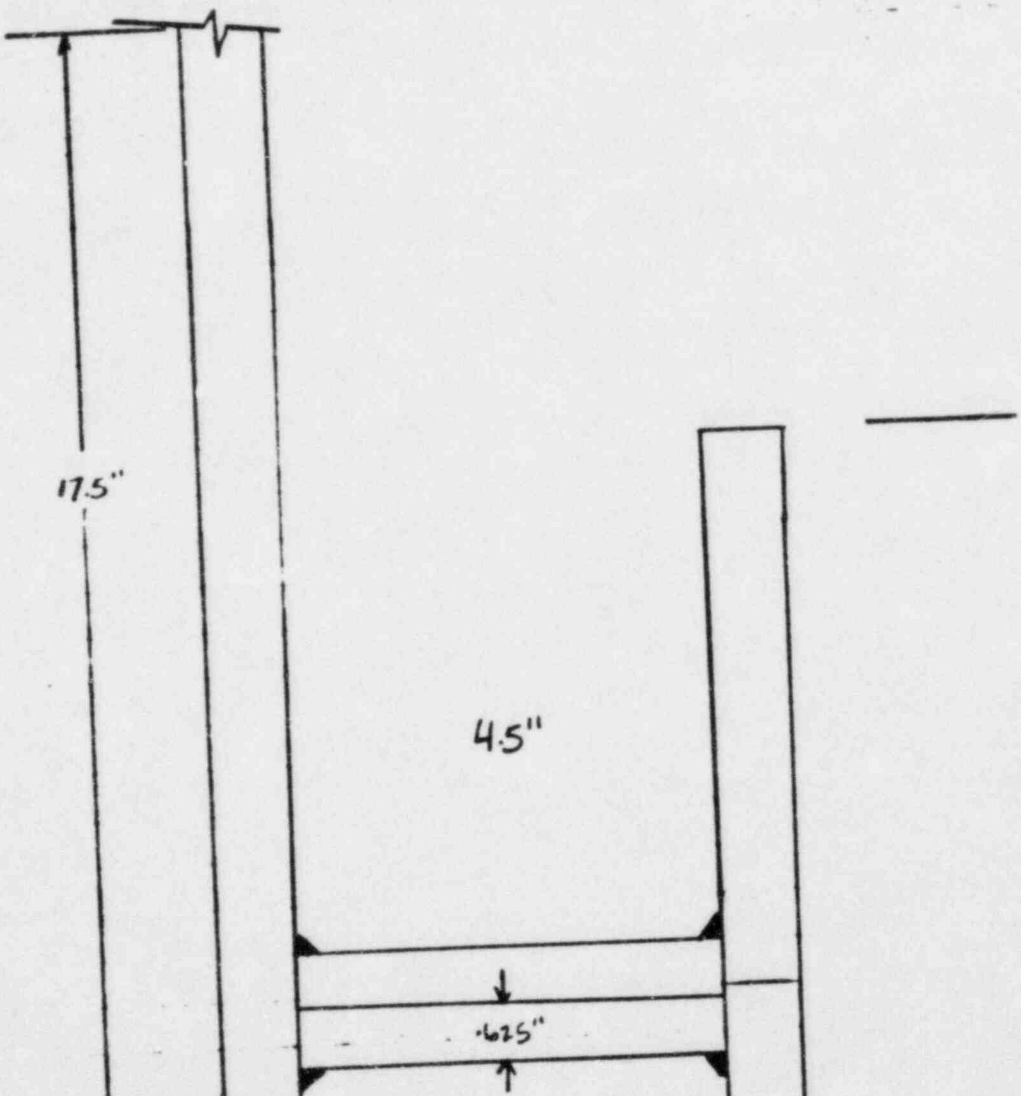
Proj. No. Equip. No.

Prepared by David Wright Date 5/4/84
Reviewed by Date
Approved by Date



Electric Starter Motor Brackets

sketch obtained during field Walkdown

Method Of Analysis

TO SHOW THAT THE ELECTRIC STARTER MOTOR BRACKET ASSEMBLY IS RIGID WE WILL SHOW THAT THE LINK BETWEEN THE TWO VERTICAL PLATES IS RIGID WE CAN CONSERVATIVELY ASSUME THAT THE VERTICAL PLATE THAT ATTACHES TO THE ENGINE IS A RIGID CONNECTION

MODEL THE HORIZONTAL PLATE (LINK) AS A CANTILEVER BEAM FIXED AT ONE END AND LOAD CONCENTRATED AT THE FREE END.

THE MAX DEFLECTION IS GIVEN BY:

$$\Delta_{MAX} = \frac{PL^3}{3EI}$$ ("Design of Welded Structures", Blodgett)

$$P = 2\left(78\ \frac{LBS}{MOTOR}\right) + 58\ LBS_{BRACKET} = 214\ LBS$$

$E = 29.6 \times 10^6$ PSI FOR CARBON STEEL IN GENERAL

$$I = \frac{(9\ IN)(.625\ IN)^3}{12} = 0.183\ IN^4$$

$L = 4.5\ IN$

$$\Delta_{MAX} = \frac{214\ (4.5)^3}{3(29.6 \times 10^6)(0.183)} = 0.0012\ IN$$

THE FREQUENCY AS DETERMINED FROM THE STATIC DEFLECTION Δ_{max} IS BASED ON:

$$f_n = \frac{1}{2\pi}\sqrt{g/\Delta_{max}} = \sqrt{\frac{3864 \text{ in/sec}^2}{0.0012 \text{ IN}}}$$

of the Link plate

$$= 90.2 \text{ Hz}$$

VIBRATION ANALYSIS BY
ROBERT K VIERCK 2nd ED
HARPER & ROW PUBL.

BY ENGINEERING JUDGEMENT THE WELDS BETWEEN THE HORIZONTAL PLATE LINKING THE 2 VERTICAL PLATES are ACCEPTABLE. This is based on their large size and wide patterns.

NATURAL FREQUENCY OF PLATE (ELECTRIC START MOTORS MOUNTED ON)

$$f_1 = \frac{\lambda_1}{2\pi a^2}\left[\frac{Eh^3}{12\gamma(1-\nu^2)}\right]^{1/2} Hz$$

$\frac{a}{b} = \frac{11}{17} \doteq .65 \quad \therefore \quad \lambda_1 = 8.946$

NAT FREQ & MODE SHAPES BY BLEVINS TABLE 11-4 #1 (CONSERVATIVE)

ALL ENDS FREE

$E = 29{,}000{,}000 \ LB/IN^2$

$h = 0.82$ IN.

$a = 11$ IN

$\nu = .3$

$$\gamma = \frac{214 \ LBS}{(144 \ IN^2)(386.4 \ \frac{IN}{SEC^2})} = 0.003846 \ \frac{LBS \cdot SEC^2}{IN}$$

$$f_1 = \frac{8.946}{2\pi(11)^2}\left[\frac{29{,}000{,}000 \ (.82)^3}{12(0.003846)(1-(.3)^2)}\right]^{1/2} = 230 Hz$$

THIS ANALYSIS SHOWS THAT THE BRACKET (PLATE) FOR WHICH THE ELECTRIC START MOTORS ARE MOUNTED BEHAVES IN A RIGID MANNER ($\frac{\text{compared}}{\text{TO 33HZ}}$) IN ADDITION WE HAVE ALSO SHOWN THAT THE LINK (PLATE) BETWEEN THE VERTICAL PLATES OF THE BRACKET IS RIGID, SINCE WE CONSERVATIVELY HAVE SHOWN THE MOUNTING BRACKET AS BEING RIGID, THEN WE WILL TRANSFER ALL THE LOADS TO THE BOLT

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CENTROID OF THE BRACKET TO ENGINE ASSEMBLY AND WILL CHECK THE STRESSES IN THE BOLTS

Form GC-3.08.1 Rev. 2

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Bracket to Engine Mounting Bolt Analysis

$$I_{xx} = 0.974\left[(3.8)^2 + 2\times(1.7)^2 + 2\times(0.2)^2\right] = 20.17 \quad In^4$$

$$I_{zz} = 0.994\left[2\times(2.2)^2 + (1.2)^2 + (1.8)^2 + (3.8)^2\right] = 28.80 \quad In^4$$

$$I_p = 20.17 + 28.80 = 48.97 \quad In^4$$

$$C_{z\,max} = 3.8'' \qquad C_{x\,max} = 3.8''$$

Tensile Stress:

$$\sigma_t = \frac{600}{4.97} + \frac{14520\times 3.8}{20.17} + \frac{7800\times 3.8}{28.80}$$

$$= 3885 \text{ Psi}$$

$$\tau_z = \frac{1734}{4.97} + \frac{16040\times 3.8}{48.97} = 1594 \text{ Psi}$$

$$\tau_x \; \frac{600}{4.97} + \frac{16040\times 3.8}{48.97} = 1365 \text{ Psi}$$

$$\tau = \tau_z + \tau_x = 1594 + 1365 = 2960 \text{ psi}$$

Allowable Stress - From ASME B+PV code, 77 edition, 79 addendum section III, article 2460 of Appendix XVII for service level A+B, SA449 material:

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \le 1.0 \Rightarrow \frac{3885^2}{36000^2} + \frac{2960^2}{14{,}880^2} = 0.05 \le 1.00 \Rightarrow \text{bolts adequate}$$

Form GO-3.08.1 Rev. 2

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FINAL

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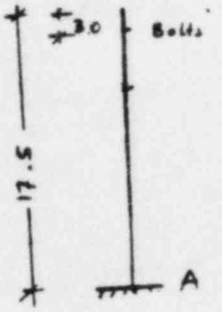
Bending Stresses in the Plate:



$M_A \approx 14.5(78+58) \cdot 2.8 + 14250$

$= 19771$ #-In.

$S = \frac{b\,t^2}{6} = \frac{75\,(082)^2}{6} = 0.84$

effective width of plate assumed to be w=7.6"

$\sigma_{max} = \frac{19771}{084} = 23536$ psi

(Based on AISC code Allowables,)

$\sigma_a = .66\,S_y^*(1.33) = 31,601$ psi > 23536 psi, adequate.

*S_y = 36,000 psi, assume A-36 carbon steel.

Conclusions: The starter motor brackets will be able to withstand a seismic event of greater than the SSE level for Shoreham and still remain structurally intact.

B.5

B.5 AIR INTAKE & EXHAUST SYSTEM

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Project		Reviewed by	Date
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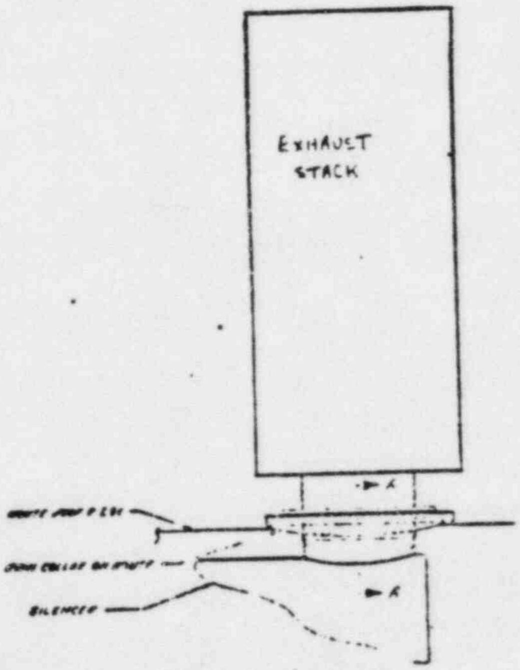
EXHAUST STACK AND SNUBBER

LOG NO. 0803150300



The exhaust stack is bolted with (20) - 1⅛"ϕ bolts to the rain shield which sits on the rain collar of the diesel generator house roof. A negligible amount of the exhaust stack loads are carried by the rain shield, almost all will be transmitted to the exhaust silencer. The exhaust stack -to- silencer juncture is the critical element for structural integrity of the exhaust stack; stresses at this location will be calculated.

(see sect A-A on following page)

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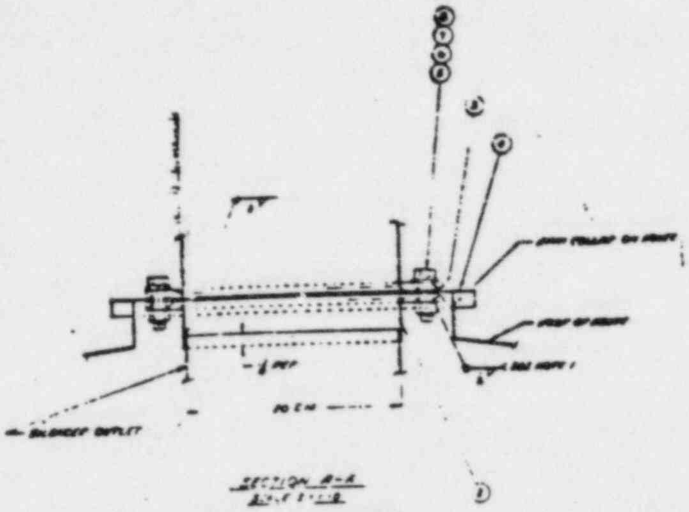
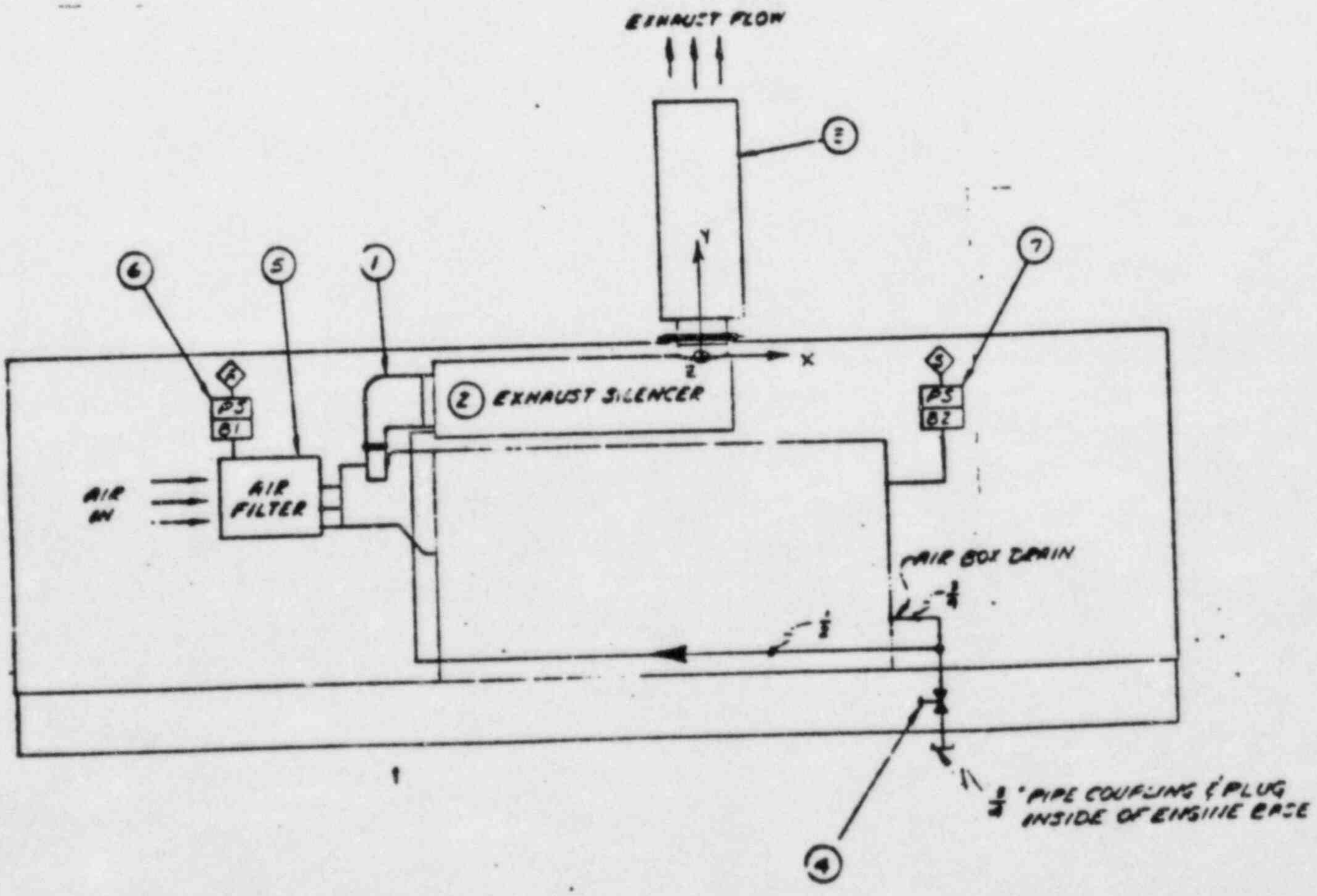


SECTION A-A
(FROM PREVIOUS PAGE)

8		1/2 FLAT WASHER, STD, STEEL			20
7		1/2 LOCK WASHER, SPLIT, STEEL			20
6		1/2-13UNC HEX NUT, STEEL			20
5		1/2-13UNC x [illegible] BOLT, STEEL			20
4		EXHAUST RAIN SHIELD		[illegible]	1
3		GASKET		[illegible]	2
2		EXHAUST COLLAR		[illegible]	1
1		EXHAUST STACK ASSEMBLY		[illegible]	1



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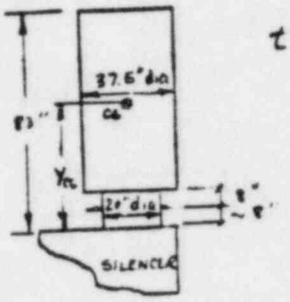
EXHAUST STACK NATURAL FREQUENCY



$t = 11$ Ga. ⇒ 0.1196"

Note: Dimensions are per telecon memo between David Wright and Ken Lewis dated 5/29/84.

Y_{CG} assumed to be 50"

The stack will be modelled as a 20" dia column of 83" height to conservatively approximate its natural frequency.

From Pressure Vessel Design Handbook by Henry H Bednar, 1981, page 16

$$f_1 = 0.560 \left[\frac{g E I}{w H^4} \right]^{1/2}$$

Weight of stack:

$$W = 0.283 \text{ lb/in}^3 \left[\pi (0.1196 \text{ in}) \left((37.5 \text{ in})(67 \text{ in}) + (20 \text{ in})(16 \text{ in}) \right) \right] = 301 \text{ lb}$$

$$w = \frac{301}{83} = 3.627 \text{ lb/in}$$

$$I = \frac{\pi}{64} \left[20^4 - (20 - 2(0.1196))^4 \right] = 369 \text{ in}^4$$

$$f_1 = 0.56 \left[\frac{(386.4)(29 \times 10^6)(369)}{(3.627)(83)^4} \right]^{1/2} = 86.8 \text{ Hz}$$

The exhaust stack is attached to the silencer which was found to have a fundamental natural frequency of 165 Hz. A conservative approximation of the systems resultant natural frequency is:

$$f_R = \left(\frac{1}{f_1} + \frac{1}{f_2} \right)^{-1} = \frac{f_1 f_2}{f_1 + f_2}$$

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$$f_R = \frac{(168)(86.8)}{(168 + 86.8)} = 57.2 \text{ Hz}$$

Therefore seismic coefficients will be based on elevated response spectra accelerations at 57 Hz times 1.5 to account for multi-mode effects.

SEISMIC LOADS (includes dead weight)

SSE coefficients : $a_x = a_z = 1.5(0.32) = 0.48g$
$a_y = 1.0 \pm 1.5(0.15) = 1.225g \downarrow$ or $0.775g \downarrow$

Note - accelerations from elevated spectrum appendix A.

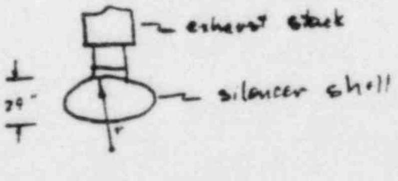
$F_x = F_z = (301)(0.48) = 144.5$ lb
$F_y = (301)(1.225) = 368.7$ lb $\downarrow$
$M_x = (144.5)(50) = 7225$ in-lb
$M_y = 0$
$M_z = (144.5)(50) = 7225$ in-lb

Note : These loads will be included in analysis of silencer.

EXHAUST STACK - SILENCER JUNCTURE STRESSES

The silencer shell is 10 Ga (0.1345") carbon steel elliptically shaped; the shell radius at the nozzle attachment is ~1.5 d_{minor}

$r \cong 1.5(29) = 43.5$ in



From the table on the following page $\sigma_{max} = 5062$ psi.
from ASME B+PV code, 83 edition, section III, table I-7.1 of Appendix I, SA-285 grade A carbon steel $S = 11.2$ ksi

Since $\sigma_{max} = 5.062$ ksi $< S = 11.2$ ksi, the exhaust stack and its attachment are considered structurally adequate.

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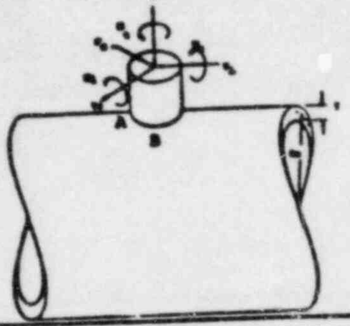
LOCAL MEMBRANE STRESSES IN A CYLINDRICAL SHELL DUE TO NOZZLE LOADS AND PRESSURE



Project Name: SHOI
Project No: 6995
Spec. No:

Equip. #: LOG. NO 0803150300	Pipe Line #
Equipt. Description: EXHAUST STACK / SILENCER	
Nozzle Description: SILENCER OUTLET TO STACK	
Subsystem #	Node # / PIPSYS
Internal Pressure P_o = ~0 Psig	

APPLIED NOZZLE LOADS

AXIAL LOAD	P =	368.7	lbs
SHEAR LOAD	V_c =	144.5	lbs
SHEAR LOAD	V_L =	144.5	lbs
TORSIONAL MOMENT	M_T =	0	in-lbs
CIRCUMFERENTIAL MOMENT	M_c =	7225	in-lbs
LONGITUDINAL MOMENT	M_L =	7225	in-lbs

GEOMETRY

VESSEL MEAN RADIUS	R_m =	43.5	in.
VESSEL THICKNESS	T =	0.1345	in.
NOZZLE RADIUS	r_o =	10.0	in.

GEOMETRIC PARAMETERS

$\gamma = \frac{R_m}{T}$ =	323
$\beta = (0.875)\frac{r_o}{R_m}$ =	0.20

	MULTIPLY AND ENTER ABSOLUTE VALUE OF RESULT		STRESSES (AT LOCATIONS A,B)	
	READ CURVES FOR	CALCULATE	A	B
CIRCUMFERENTIAL STRESSES	$\frac{N_\phi}{P/R_m}$ = 6.6	$\frac{P}{R_m T}$ = 63.0	4158	415.8
	$\frac{N_\phi}{M_c/R_m^2\beta}$ = 7.0	$\frac{M_c}{R_m^2\beta T}$ = 142	▨	994
	$\frac{N_\phi}{M_L/R_m^2\beta}$ = 14.0	$\frac{M_L}{R_m^2\beta T}$ = 142	1988	▨
	Circumferential Pressure Stress, P_o*R_m/T		0	0
	Sum of Circumferential Stresses σ_ϕ =		2404	1410
LONGITUDINAL STRESSES	$\frac{N_x}{P/R_m}$ = 24.0	$\frac{P}{R_m T}$ = 63.0	1612	1512
	$\frac{N_x}{M_c/R_m^2\beta}$ = 25.0	$\frac{M_c}{R_m^2\beta T}$ = 142	▨	3550
	$\frac{N_x}{M_L/R_m^2\beta}$ = 6.0	$\frac{M_L}{R_m^2\beta T}$ = 142	852	▨
	Longitudinal Pressure Stress, $P_o*R_m/2T$		0	0
	Sum of Longitudinal Stresses σ_x =		2364	5062
SHEAR STRESSES	Due to Torsion M_T	$\frac{M_T}{2\pi r_o^2 T}$	0	0
	Due to Load V_c	$\frac{V_c}{\pi r_o T}$	34	▨
	Due to Load V_L	$\frac{V_L}{\pi r_o T}$	▨	34
	Sum of Shear Stresses τ =		34	34
Maximum Stress	$= 1/2\left[(\sigma_\phi + \sigma_x) + \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2}\right]$ =		2423	5062

Maximum stress should be less than or equal to 1.5S where S is the allowable stress of shell material.

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MEMCRANDUM OF
TELEPHONE CONVERSATION

Proj.
Page

Date: 5/29/84

Time: 2:30 pm

Person Called: David Wright (Name) of Sargent & Lundy (Company)

Person Calling: Ken Lewis (Name) of Morrison Knudsen (Company)

Project: Shoreham Project No. 6995-00

Subject Discussed: Diesel Generator Exhaust Stack Model No. 6041F09002

Summary of Discussion, Decisions and Commitments:

The following information was provided by Mr. Lewis:

*Height: 83 in.

Body O.D.: 37.5 in.

Nozzle Height: 8 in.

Nozzle O.D.: 20 in.

Material: Hot rolled carbon steel

Thickness: 11 Gauge with a 7 gauge reinforcement at the nozzle section

Note - The dimensions noted above are for a similar stack which may not be on every engine.

* Top of silencer to top of stack.

cc: DW/eg
K. Lewis - Morrison Knudsen
J. Sinnappan - 30
AEM/APD/DW - 30

David Wright
Signature

File: CQD-014046 - 30

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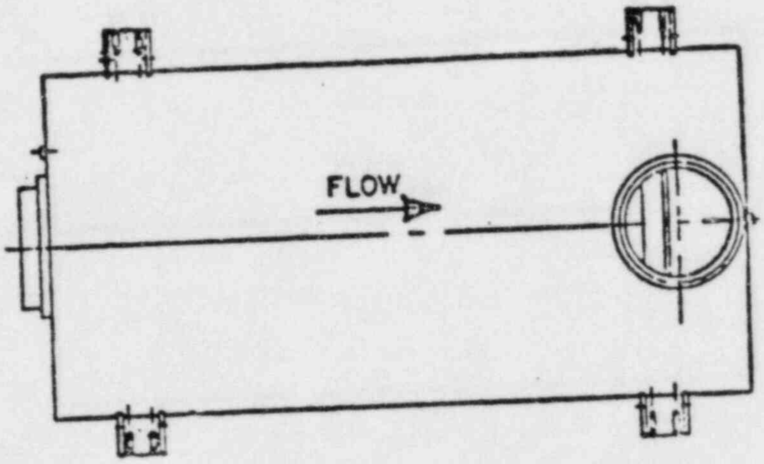
EXHAUST SILENCER LOG NO. 0803150100



MFG: Universal Silencer

PART NO: WAS 23-3902

Structural integrity will be investigated by calculating stresses due to dead weight, seismic accelerations, and piping loads as applicable for the following items:

- Mounting bolts
- Mounting brackets
- shell of silencer body

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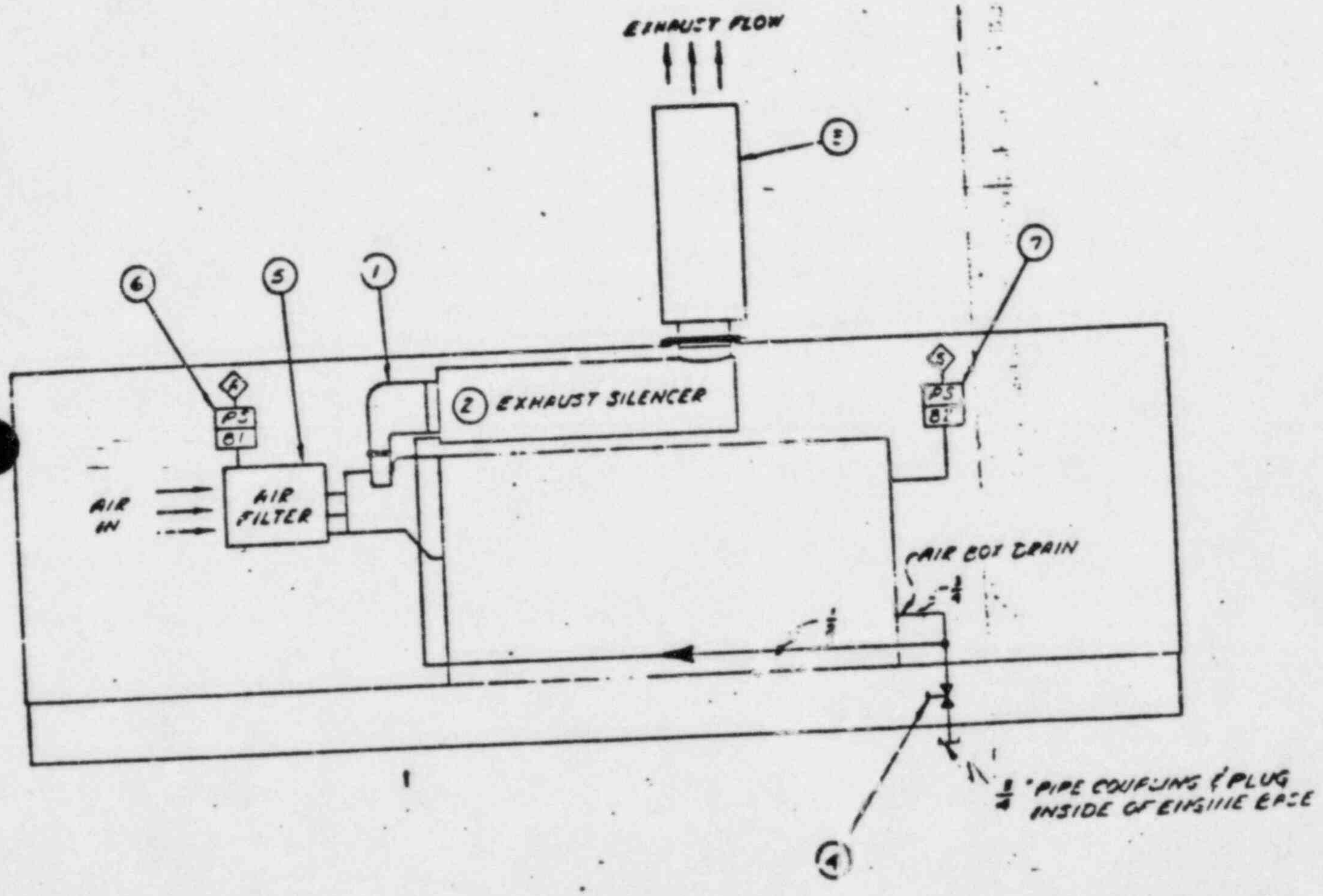

7		PRESSURE SWITCH PS-B2			
6		PRESSURE SWITCH PS-B1			
5	204	AIR FILTER			1
4	807	VALVE 3/4" NPT, GATE		609/F09007	1
3		EXHAUST STACK & SHUEESE MOUNTING ASS'Y		29-3902	1
2	901	EXHAUST SILENCER UNIVERSAL		8366591	1
1	2013	EXHAUST ELBOW EMD			

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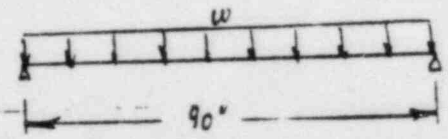
MOUNTING BOLT STRESSES

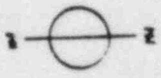
The exhaust silencer is mounted by means of 4 brackets attached to the shell. Since the bracket mounting slots are 3/4" wide, the mounting bolts will be assumed as 5/8" ϕ, SA-449 steel.

Note : For design details, see Universal Silencer Drawing No. A-23-902-XX, Rev. 1 and Telecon Memo between J. Harris of Universal Silencer and A. Wisniewski of S& L (CQD), dated 5/22/84.

Equipment Natural Frequency :

To approximate the fundamental natural frequency of the silencer, it will be modelled as a simply supported circular tube of 29" diameter. This conservatively approximates the 29" x 58" elliptical cross-section





$d_{outside} = 29"$
$t = 0.1345"$ (10 Ga.)
$d_{inside} = 28.731"$

$W = 1700$ lb $\Rightarrow$ $\omega = \frac{1700}{90} = 18.89$ lb/in

$I_z = \frac{\pi}{64}(29^4 - 28.731^4) = 1270$ in^4 $E = 29 \times 10^6$ psi

$f_n = \frac{9.87}{2\pi}\sqrt{\frac{(29\times10^6)(1270)(386.4)}{(18.89)(90)^4}} = 168$ Hz

"Formulas For Stress and Strain", Roark + Young 5th edition, McGraw Hill

$\therefore$ the component can be analyzed as rigid

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Equipment Loads :

Since the silencer is mounted near the top of the Diesel Generator assembly, elevated response spectra are used to determine seismic accelerations. The seismic accelerations (ZPA values, since rigid) will be multiplied by a factor of 1.5 to account for exhaust piping loads.

<u>Seismic Coefficients</u> (includes dead weight)

SSE $a_x = a_z = 1.5(0.29) = 0.435\,g$

$a_y = 1.5(0.14) + 1 = 1.21\,g \downarrow$ or $a_y = 1 - 1.5(0.14) = 0.79\,g \downarrow$

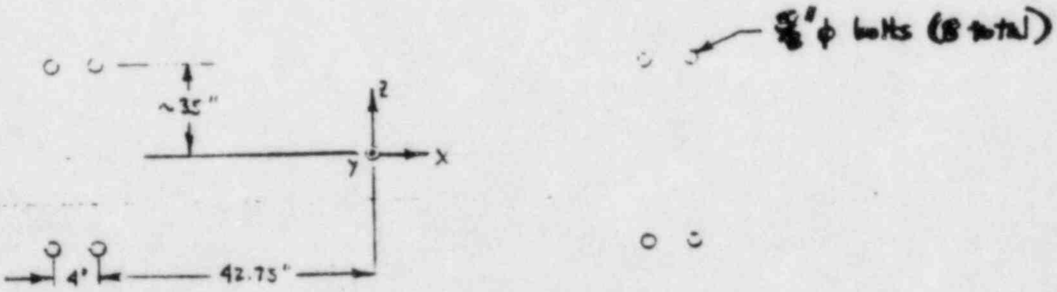
note - accelerations obtained from elevated spectrum, Shoreham, Appendix A.

The component center of gravity will be very close to the bolt centroid, so the eccentricities are assumed to be zero.

$F_x = F_z = (1700)(0.435) = 739.5$ lb

$F_y = (1700)(1.21) = 2057$ lb $\downarrow$ or $F_y = (1700)(0.79) = 1343$ lb $\downarrow$

$M_x = M_y = M_z = 0$



$A_{bolt} = 0.226\ in^2$ (Tensile stress area, per AISC Manual)

$\Sigma A = 8(0.226) = 1.808\ in^2$

$I_x = 8(0.226)(35)^2 = 2215\ in^4$

$I_z = 4(0.226)(42.75^2 + 46.75^2) = 3628\ in^4$

$I_y = I_x + I_z = 5843\ in^4$

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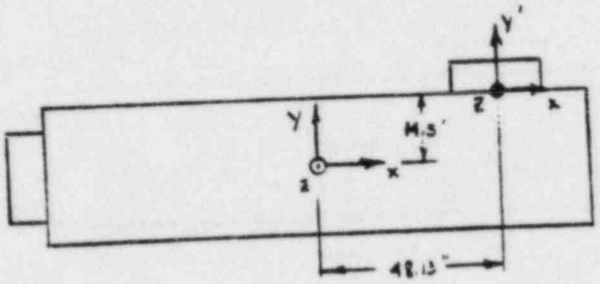
The loads determined from the analysis of the exhaust stack will be included: (see previous pages for calculated values)



$F_x' = F_z' = 144.5$ lb
$F_y' = 368.7$ lb ↓

$M_x' = M_z = 7225$ in-lb
$M_y' = 0$

Total Loads at Bolt Centroid:

$F_x = F_z = 739.5 + 144.5 = 854$ lb
$F_y = 2037$ lb ↓ $+ 368.7$ lb ↓ $= 2426$ lb ↓
$M_x = 7225 + (144.5)(14.5) = 9320$ in-lb
$M_y = (144.5)(48.13) = 6955$ in-lb
$M_z = 7225 + (144.5)(14.5) + (368.7)(48.13) = 27066$ in-lb

$$\sigma_t = \frac{F_y \uparrow}{\Sigma A} + \frac{M_x C_z}{I_x} + \frac{M_z C_x}{I_z}$$

$$\sigma_t = 0 + \frac{(9320)(35)}{2215} + \frac{(27066)(46.75)}{3628} = 496 \text{ psi}$$

$$\tau_x = \frac{F_x}{\Sigma A} + \frac{M_y C_z}{J_y} = \frac{884}{1.808} + \frac{(6955)(46.75)}{5843} = 545 \text{ psi}$$

$$\tau_z = \frac{F_z}{\Sigma A} + \frac{M_y C_x}{J_y} = \frac{884}{1.808} + \frac{(6955)(35)}{5843} = 531 \text{ psi}$$

$$\tau = \sqrt{\tau_x^2 + \tau_z^2} = 761 \text{ psi}$$

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$f_t = 496 \text{ psi}$

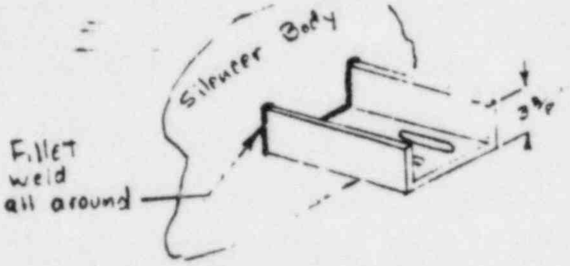
$f_v = 761 \text{ psi}$

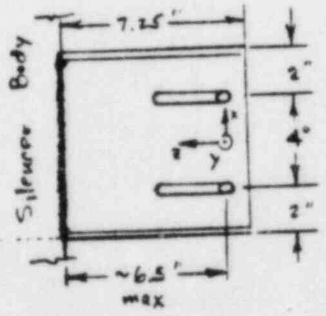
as can be seen from similar bolt analysis throughout this report, based on ASME B+PV code, 77 edition, 79 summer addendum, section III, article 2460, that these levels are insignificant. The bolts are adequate.

MOUNTING BRACKET STRESSES

Bracket thickness = 3/8"

Bracket stresses will be highest just before attachment to body of silencer





$F_x \cong 2(\tau_x)_{bolt} A_{bolt} = 2(545)(0.226) = 246 \text{ lb}$

$F_y = 2(\tau_t)_{bolt} A_{bolt} = 2(496)(0.226) = 224 \text{ lb}$

$F_z \cong 2(\tau_z)_{bolt} A_{bolt} = 2(531)(0.226) = 240 \text{ lb}$

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At bracket-to-silencer body juncture:

$F_x = 246$ lb
$F_y = 224$ lb
$F_z = 240$ lb
$M_x = (224)(6.5) + 240(0.73) = 1631$ in-lb
$M_y = (246)(6.5) = 1599$ in-lb
$M_z = (246)(0.73) = 179.6$ in-lb

From Blodgett, p. 2.2-8:

$S_x = \dfrac{td^2(2b+d)}{3(b+d)}$, $S_y = \dfrac{tb}{6}(b+6d)$

$R = \frac{t^3}{3}(b+2d)$

$t = 0.375''$
$b = 7.625''$
$d = 3.188''$

$S_x = \dfrac{(0.375)(3.188)^2[2(7.625) + 3.188]}{3(7.625 + 3.188)}$

$\bar{y} = \dfrac{d^2}{b+2d} = 0.73''$ $S_x = 2.166$ in^3

$S_y = \dfrac{(0.375)(7.625)(7.625 + 6(3.188))}{6} = 12.75$ in^3

$R = \dfrac{(0.375)^3}{3}(7.625 + 2(3.188)) = 0.246$

$A = (0.375)(7.625 + 2(3.188)) = 5.25$ in^2

$\sigma_t = \dfrac{F_z}{A} + \dfrac{M_x}{S_x} + \dfrac{M_y}{S_y}$

$\sigma_t = \dfrac{240}{5.25} + \dfrac{1631}{2.166} + \dfrac{1599}{12.75} = 924.1$ psi

$\tau = \dfrac{\sqrt{F_x^2 + F_y^2}}{A} + \dfrac{M_z t}{R} = \dfrac{\sqrt{246^2 + 224^2}}{5.25} + \dfrac{(179.6)(0.375)}{0.246} = 336$ psi

$\sigma_{max} = \dfrac{\sigma_t}{2} + \left[\left(\dfrac{\sigma_t}{2}\right)^2 + \tau^2\right]^{1/2} = 1033$ psi

Assuming bracket material is A-36 steel (S_y = 36 ksi)

$\sigma_{allow} = \frac{4}{3}(0.6)(36.0) = 28.8$ ksi (per AISC Manual)

Since $\sigma_{max} = 1033$ psi $< \sigma_{allow} = 25{,}800$ psi, the mounting bracket stresses are acceptable.

SHELL STRESSES

The silencer is modelled again as a simply supported beam of 29" diameter for conservatism. A uniform loading equivalent to the resultant of F_y, F_z, M_y and M_z will be applied (total loads at [illegible])

$$F_R = \sqrt{F_y^2 + F_z^2} = \sqrt{2426^2 + 884^2} = 2582 \text{ lb}$$

$$\therefore \quad w = \frac{2582}{90} = 28.69 \text{ lb/in}$$

$$M_{max} = \frac{wL^2}{8} + \sqrt{M_y^2 + M_z^2} = \frac{(28.69)(90)^2}{8} + \sqrt{6955^2 + 27066^2} = 56{,}444 \text{ [illegible]}$$

$$\sigma = \frac{Mc}{I_z} = \frac{(56{,}994)(14.5)}{1270} = 651 \text{ psi}$$

$S_a = 12{,}600$ psi {From ASME B+PV code, 1977 edition, Section III, table I-7.1 of Appendix I, for SA-36 steel @ 300°F}

CONCLUSION

The subject exhaust silencer is structurally adequate to withstand the applicable dead weight, seismic, and piping loads

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MEMORANDUM OF TELEPHONE CONVERSATION

Calc. No: CQD-
Rev: Date:
Proj. No:
Page Of

Date: 5/22/84

Time: 2:15 pm

Person Called: John Harris (Name) of Universal Silencer (Company)

Person Calling: Andy Wisniewski (Name) of S&L (CQD) (Company)

Project: Shoreham Project No. 6995-00

Subject Discussed: Design Specifications for Qualification of Universal Silencer 20" Exhaust Silencer, P/N WAS 22-3902.

Summary of Discussion, Decisions and Commitments:

The following information was provided by Mr. Harris:

Mounting Slot Size: 3/4" width

Mounting Bracket Thickness: 3/8"

Shell Thickness: 10 gauge

Mounting brackets attached to body by fillet welding all around

Component Weight: Approx. 1700 lbs.

Conclusion: The information provided shall be used for equipment qualification.

DW/eg

cc J. Harris - Universal Silencer
D. Wright - 30
AEM/APD/AW - 30

Andy Wisniewski
Signature

File: CQD-014046 - 30

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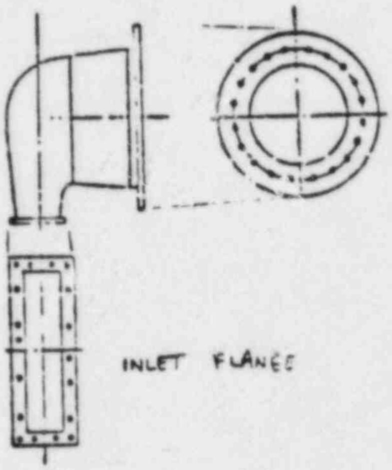
EXHAUST OUTLET ADAPTER

LOG NO. 0803150200



The exhaust outlet adapter is connected in-line between the diesel generator and the exhaust silencer. Analysis of the exhaust silencer showed that the assembly is rigid and structurally adequate to withstand applicable loadings. The outlet flange mounting to the silencer is by (20) - 1⅛" ϕ bolts while the inlet flange mounting is by (18) - 9/16" ϕ bolts. Comparison of the exhaust outlet adapter with the silencer indicates that the adapter will also be rigid.

Based on the above information, the exhaust outlet adapter and its mountings are considered structurally adequate by engineering judgment without the need of further analysis

Exhaust Outlet Adapters Turbocharged Engines

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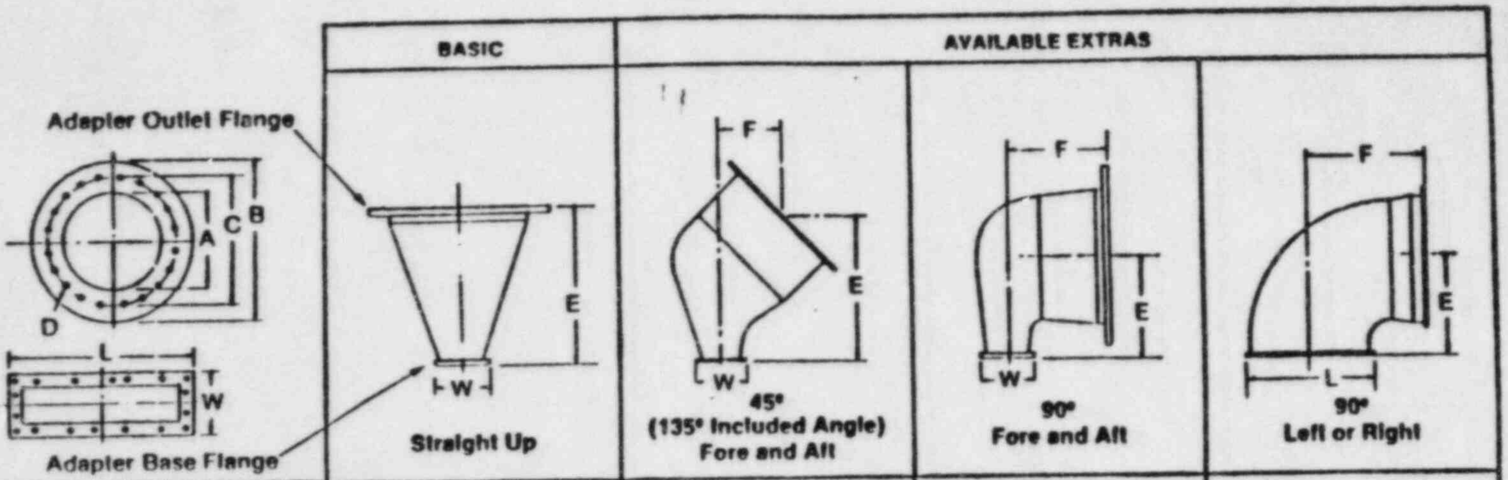

Turn Angle From Vertical Outlet Facing Longitudinal Load (F_l) - Max. *lb. (kg.) Transverse Load (F_t) Max. *lb. (kg.) Vertical Load (F_v) Max. *lb. (kg.)	0° Up 300 (136) 3000 (1361) 1000 (454)		45° (135 Inc. Angle) Fore or Aft 300 (136) 3000 (1361) 1000 (454)		90° Fore or Aft 300 (136) 3000 (1361) 800 (363)		90° Left or Right 300 (136) 3000 (1361) 1000 (454)	
Combined Loading of F_l, F_t, & F_v	$3.3\ F_l + .33\ F_t +$ $F_v \leq 1000$		$3.3\ F_l + .33\ F_t +$ $F_v \leq 1000$		$3.3\ F_l + .33\ F_t +$ $1.25\ F_v \leq 1000$		$3.3\ F_l + .33\ F_t +$ $F_v \leq 1000$	
ENGINE MODEL	8-645E4C & E9C 12-645F4B & F9B	16-645F4B & F9B 20-645F4B & F9B	8-645E4C & E9C 12-645F4B & F9B	16-645F4B & F9B 20-645F4B & F9B	8-645E4C & E9C 12-645F4B & F9B	16-645F4B & F9B 20-645F4B & F9B	8-645E4C & E9C 12-645F4B & F9B	16-645F4B & F9B 20-645F4B & F9B
Dimensions in Inches (mm) A - Outlet Dia. I.D.	20 (508)	22 (559)	20 (508)	22 (559)	20 (508)	22 (559)	20 (508)	22 (559)
B - Outlet Flange O.D.	27-1/2 (699)	29-1/2 (749)	27-1/2 (699)	29-1/2 (749)	27-1/2 (699)	29-1/2 (749)	27-1/2 (699)	29-1/2 (749)
C - Bolt Circle - Dia.	25 (635)	27-1/4 (692)	25 (635)	27-1/4 (692)	25 (635)	27-1/4 (692)	25 (635)	27-1/4 (692)
D - Bolts (not furn.) No. - Hole Size	20 - 1-3/16 (30)	20 - 1-3/8 (35)	20 - 1-3/16 (30)	20 - 1-3/8 (35)	20 - 1-3/16 (30)	20 - 1-3/8 (35)	20 - 1-3/16 (30)	20 - 1-3/ (35)
E - Outlet Flange Vertically from Base	26-1/2 (673)	26-1/2 (673)	23-3/4 (603)	23-3/4 (603)	18-29/32 (480)	22-11/16 (576)	20-9/16 (522)	20-9/16 (522)
F - Outlet Flange Horizontally from Base ℄	0	0	10-1/4 (260)	10-1/4 (260)	16-1/2 (419)	16-1/2 (419)	28 (711)	28 (711)

L x W - Adapter Base Flange (ALL) 32-3/4 x 9-9/32 (832 mm x 236 mm)

Bolts - Hole Size (Bolts Furnished) 18 - 9/16 (14 mm)

Location of Turbocharger Exit Flange:

Flange face above ℄ crankshaft	70-3/4 (1797 mm)
Flange ℄ of rear finished face of engine	22-19/64 (566 mm)

*NOTE: Maximum permissible loading on exhaust outlet flanges - individual and not occurring together.

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Client LILCo
Project Shoreham - 1
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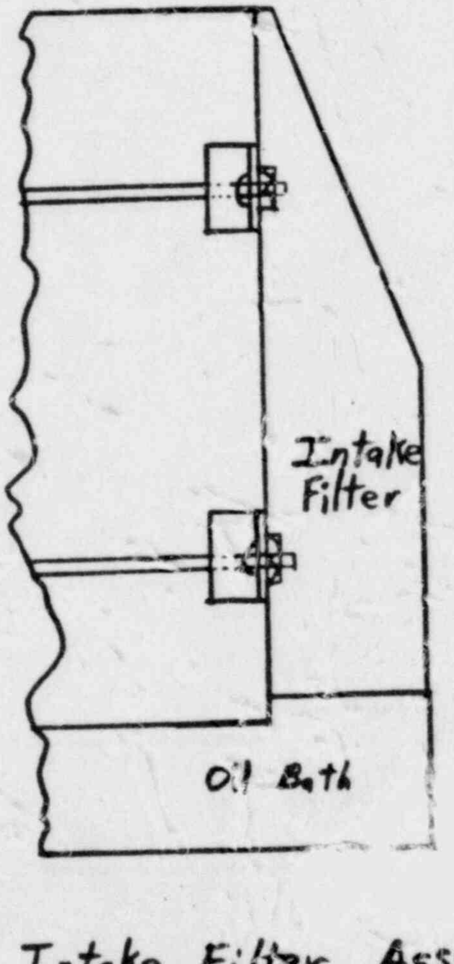
Prepared by Bruce M. Foy Date 5-29-84
Reviewed by Date
Approved by Date

Air Intake Filter Assembly

Log # 0803140100



The Air Intake Filter Assembly found on the Shoreham E.D.G.S.'s consists of a simple sheet metal enclosure with two open sides (mesh-see photos 29,30). This assembly serves only as the initial entry point for the diesel engine intake. Inside the assembly rests an air filter solely for the purpose of trapping particulate matter out of the intake air.

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From photo #'s 29 & 30 it is evident that the assembly in question swings out horizontally from one side when filter changing is required.

The oil bath container is welded to the diesel intake substructure and is only connected to the assembly via two buckle tie downs. Since the welds are much more rigid than these buckles, any induced seismic loads from the oil bath container would be directly transfered through the welds into the diesel substructure.

From the photos it is also apparent that there are no extended masses or extraneous portions of the assembly. Consequently the assembly is quite rigid and poses no danger to any adjacent safety-related subcomponents.

From the field sketch it is obvious that the hinge bolt assembly, comprised of 2-3/8"∅ bolts pin-connected to the substructure via 2-1/8" thick welded brackets (1/4" fillet-3 sides), is more than adequate for withstanding the maximum possible seismic loading.

For these reasons it is determined that these air intake assemblies are adequate as designed and require no detailed analysis. If subsequent loss of structural integrity of this assembly were to occur, the E.D.G.S.'s will still function as intended. This assembly only serves as a first stage of filtering the intake air, and is not a crucial item required to function in order for the E.D.G.S.'s to perform their intended safety function.

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Client
Project
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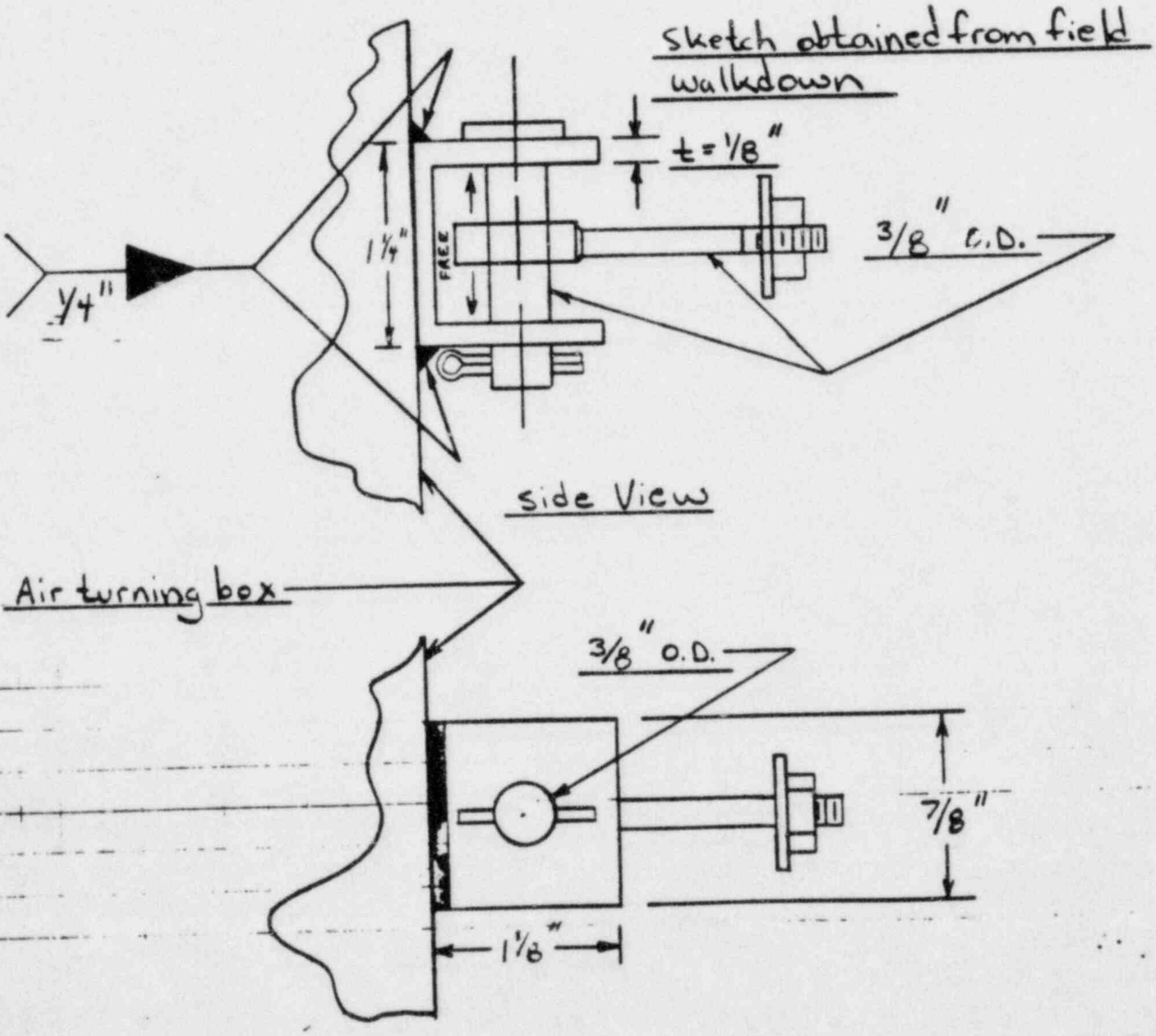
Prepared by David Wright Date 5/16/84
Reviewed by Date
Approved by Date

Air Cleaner Filter Bracket

Log No. 0803140100



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Client LILCo	Prepared by	Date 5-29-94
Project Shoreham-1	Reviewed by	Date
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Air Turning Box

Log # 0803140400

(View Not Available)

The Air turning box is found in the generator portion of the building known as the engine enclosure. This box only serves as a mechanism to divert engine air flow. The box is centrally located on top of the generator between a variety of other subcomponents. The air turning box is mounted (as can be seen in the mounting detail) to the top of the generator, and to the Nozzle leading to the turbocharger inlet. The generator has been shown to be rigid and will transmit the ground motion (ZPA) directly to the box without any significant amplification. Since loss of the turning box (i.e. falling or breaking loose) would not cause any loss of air to the engine we do not consider it an essential component and is adequate for use on the engine.

Note- further calculations should be performed to ensure that no essential equipment would be hit by the loose equipment.

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FINAL

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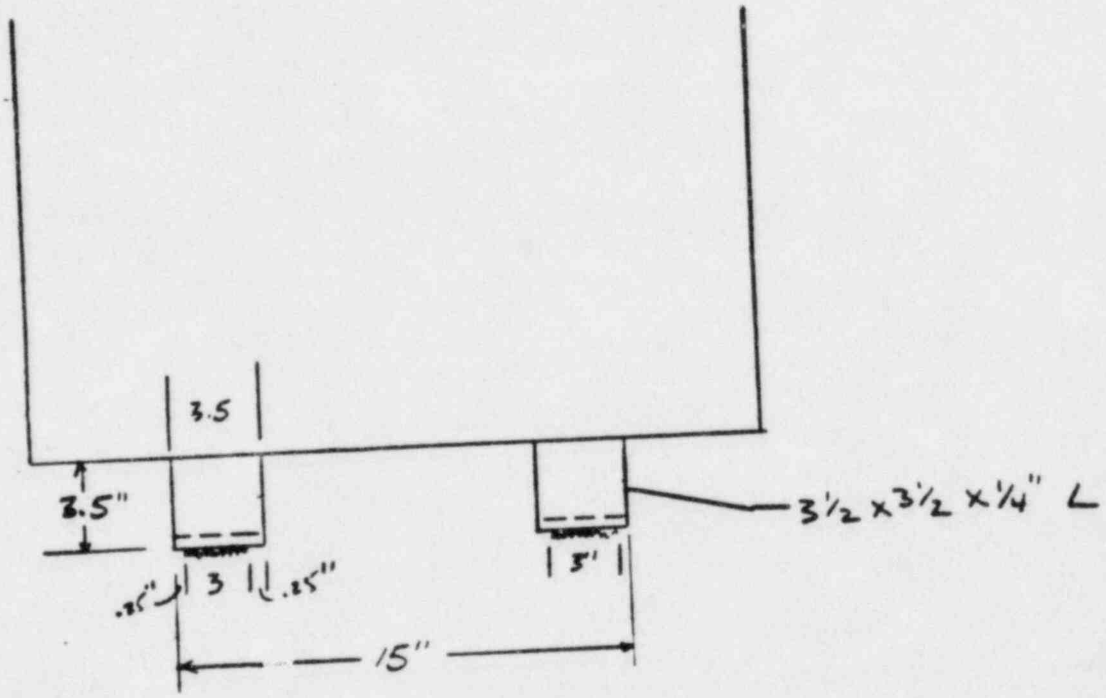
AIR TURNING BOX MTG



Sketch obtained from field walkdown

NOTE: WELDS ON BOTTOM ARE BAD WELDS AND SHOULD BE CHECKED

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B.6

B.6 GENERATOR SYSTEM

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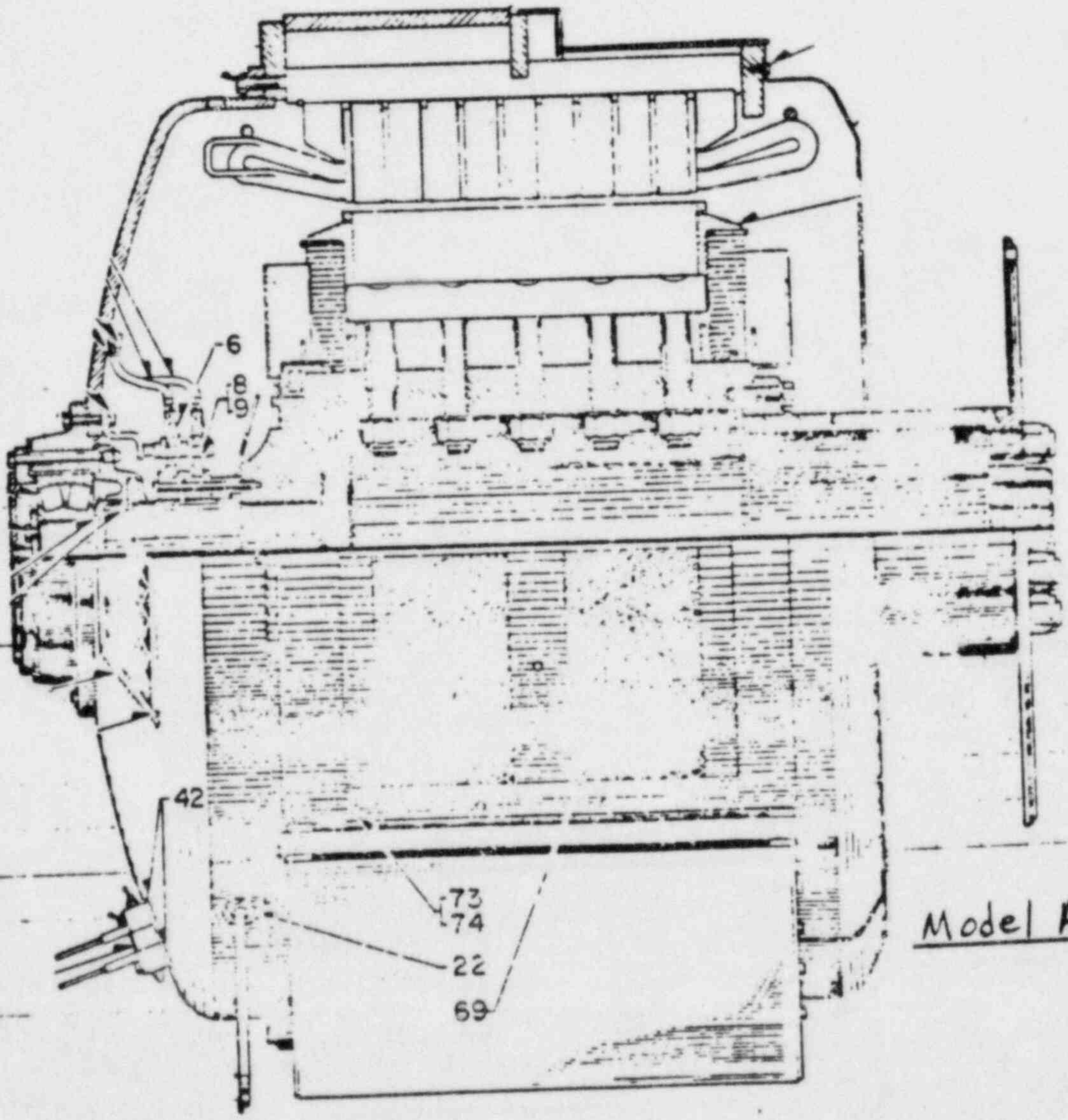
Client
Project
Proj. No. Equip. No.

Prepared by Dave Wright Date 5/17/84
Reviewed by Date
Approved by Date

Generator Assembly

Log No. 0806070000



Model A-20

Generator Assembly

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Method of Analysis

The generator will be addressed in two areas;

I.) structural integrity

A structural integrity analysis of the generator will demonstrate that the equipment will remain intact after a seismic occurance has taken place. It will be shown that the generator will remain fixed in its operating position with no appreciable damage occurring.

II. Operability

The ability of the generator to function after it has been subjected to a seismic loading condition will be demonstrated by performing a deflection analysis on the rotor assembly. The allowable deflection of the rotor will be taken as the air gap between the rotor and the stator.

The analysis performed in this report will be suplimented with the work done on the LaSalle (C.E.Co.) report found in (S+L) EMD-008028 Rev. 00 dated 4/77.

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Structural Inegrity Check

The EMD(G.M.) Model A-20 generator is equipped with a single outboard bearing. The inboard side of the generator is supported, through a large coupling, by the outboard engine bearing. The shaft is connected to the engine by a large coupling ring bolted to the Engine flywheel. Since the coupling ring and flywheel are only inches away from the engine bearing there will be no appreciable shaft deflection and no analysis of the engine shaft will be performed. The outbord bearing housing and support will be assumed rigid in the verticle direction. The representative crossection can be seen on the cutaway view of the generator. Since the housing takes no thrust loads (the generator rotor is restrained axially by the engine shaft) there is no need to perform any extensive analysis at those points. It should also be noted that the generator has no extended masses or components that would tend to be excited by a low frequency seismic input. For this reason the worst mode of failure will be the mounting bolts that attach the generator to the skid frame.

Mounting Bolt Check

Bolt Size - 1½ in (EMD(GM) drawing NO. 8367514)

No. of Bolts - 4 "

Type - SAE Grade 5 (A-449 material assumed)

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Loading:

The loading on the bolts will be estimated such that it would simulate the worst possible combination of occurances. The Loading will include a startup torque taken as double the operating torque and an SSE seismic event.

Starting Torque:

assumptions: 1.) The maximum possible input H.P. will be assumed equal to the Nominal engine output.

2.) Starting torque will be estimated at double the torque at max H.P. output.

Nominal Engine BHP = 3600 H.P. @ 900 rpm (from General Data section of the MP-45 owners manual)

$$^*T = \frac{5252(\text{H.P.})}{N} = \frac{(5252)(3600)}{(900)} = 21{,}008 \text{ ft-lbs}$$

$$T = 21{,}008 \text{ ft-lbs } (12\text{in}/1\text{ft}) = 252{,}096 \text{ in-lbs}$$

*Note - From Internal Combustion Engines and Air Pollution by EDward Obert, copyright 1973 by Harper and Row

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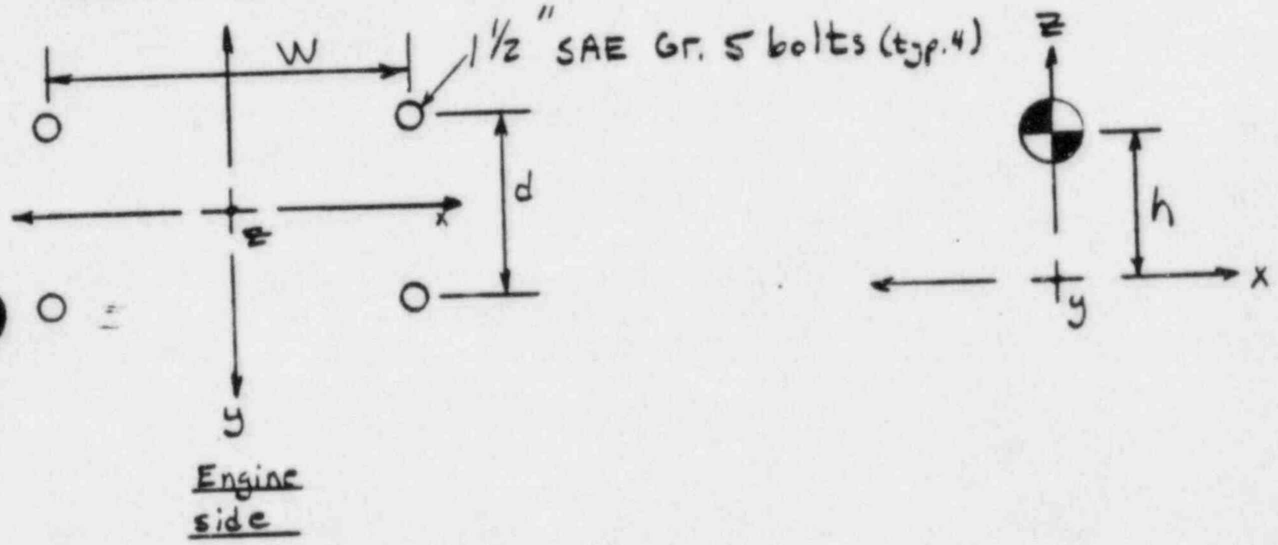
starting torque = $2T = 2(252,096) = 504,192$ in-lbs

so: $M_{y_{start}} = 504,192$ in-lbs

Bolt Pattern



$W \cong$ width at outside of skid flange − width of the flange

$= 74.5$in $- 11.5$in $= 63$in (all dimensions from LILco drawing no. F50272-1, Rev 1, 4-25-84)

$h \cong$ engine shaft height $= 20.125$in (from LaSalle (ceco) report, Stewart + Stevenson drawing No. 22080 rev. F, 12/19/75)

$d \cong 12.75$in (estimated from EMD dwg No. 8367514 rev. D)

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Bolt Pattern Geometric Properties:

$A_t = 1.405 \text{ in}^2$ (from standard table, 1½ in bolt size)

$$I_{yy} = \Sigma \overset{\cong 0}{\cancel{I_y}} + A\bar{x}^2 = 4(1.405)[63/2]^2 = 5576 \text{ in}^4$$

$$I_{xx} = \Sigma \overset{\cong 0}{\cancel{I_x}} + A\bar{y}^2 = 4(1.405)[12.75/2]^2 = 228 \text{ in}^4$$

(Seismic + Weight) Loading

The acceleration of the generator will be assumed to be 1"g" in each direction. This is a bounding level that conservatively envelopes the 0.2g maximum zpa for Shoreham (SSE level).

Note - see Appendix A for Shoreham Ground spectrum

Generator Weight = 18,100 Lbs (from MP-45 owners manual, model AB-20 similar to A-20)

$F_x = W_t(1g) = 18,100 \text{Lbs}(1g) = 18,100 \text{Lbs}$

$F_y = W_t(1g) = 18,100 \text{Lbs}(1g) = 18,100 \text{Lbs}$

$F_z = W_t(1g) + (-W_t) = 18,100 \text{Lbs}(1g) - 18,100 \text{Lbs} = 0.0 \text{Lbs}$

$M_x = M_y = F_{x \text{ or } y}(h) = 18,100 \text{Lbs}(20.125 \text{in}) = 364263 \text{ in-lbs}$

Total Loading (Weight + Start Torque + Seismic)

$F_x = 18,100$ lbs

$F_y = 18,100$ lbs

$F_z = 0.0$ in-lbs

$M_x = 364,263$ in-lbs

$M_y = 364,263$ in-lbs + 504,192 in-lbs = 868,455 in-lbs

$M_z = 0.0$ in-lbs

Estimated Stress

$$\sigma_t = F_z \cancelto{0}{/4A} + M_x C_y / I_x + M_y C_x / I_y$$

$$= \frac{(364,263)(12.75/2)}{(228)} + \frac{(868,455)(63/2)}{(5576)} = 15,091 \text{ psi}$$

$$\tau_v = [F_x^2 + F_y^2]^{1/2} / 4A + \cancelto{0}{M_z C_{x,y}} / I_z$$

$$\tau_v = [18,100^2 + 18,100^2]^{1/2} / 4(1.405) = 4555 \text{ psi}$$

for Service level A conditions the following conditions must be satisfied (Per AISC guidelines)

$\sigma_t \leq 0.33 S_u$ and

$\sigma_t \leq 0.43 S_u - 1.8 \tau$

$S_u^* = 105{,}000 \text{psi}$ (from ASME Appendix I, table I-7.3 for SA-449 material, 1" to 1½".)

$\sigma_t = 15{,}091 \text{psi} \leq 0.33(105{,}000) = 34{,}650 \text{psi} \Rightarrow$ ok

$\sigma_t = 15{,}091 \text{psi} \leq 0.43(105{,}000) - 1.8(4555) = 36{,}951 \Rightarrow$ ok

Since both conditions were satisfied for service level A using leve C loads the bolts are adequate and will not fail in service or under seismic loading conditions.

*Note – From ASME B+PV CODE, 1977 Edition, Section III, table I-7.3 of Appendix I for SA-449 steel

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Operability Check

The following information was provided by EMD (GM):
(see attached Telephone memorandum)

Model A-20 data: Rotor Weight - 8100 Lbs

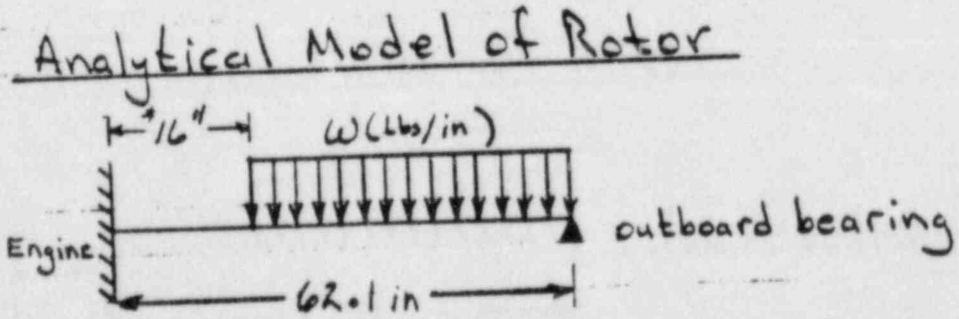
Rotor Length - 62.1 in

Air Gap - 0.40 in

minimum Shaft - 4.725 in
O.D. (found at outboard bearing location)

The deflection of the rotor will be checked to show that no contact will be made between the rotor and stator during a seismic event. For conservatism the rotor will be assumed to have the minimum shaft size over the entire length. The seismic accelerations will be applied as a resultant uniform load over the shaft.

Analytical Model of Rotor



* Note - assumed, no data on the length but estimated from visual inspection of the system.

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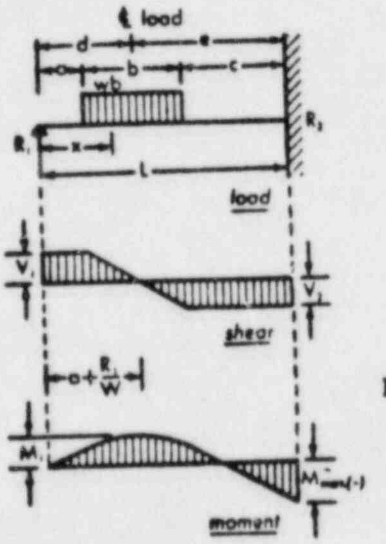
$\omega = 8100\,\text{Lbs}/(62.1-16) = 176\,\text{Lbs/in}$

$\underset{\text{Seismic Resultant}}{\omega} = [(176\,\text{Lbs/in} + 176\,\text{Lbs/in})^2 + 176^2]^{1/2} = \underline{394\,\text{Lbs/in}}$ (resultant for ℄ all dir.)

From <u>Design of Welded Structures</u> (by Blodgett) it can be seen that the maximum deflection for the model shown will occur at the ℄ of the Load.

(8C) **Beam fixed at one end and supported at the other end**
Uniform load partially distributed over span



$$R_1 = V_1 = \frac{wb}{8L^3}(12e^2L - 4e^3 + b^2d)$$

$$R_2 = V_2 = wb - R_1$$

$$M_{max\,(-)} = \frac{wb}{8L^2}(12e^2L - 4e^3 + b^2d - 8eL^2)$$

$$M_1 = R_1\left(a + \frac{R_1}{2w}\right)$$

When $x < a$ $\quad M_x = R_1 x$

When $x > a$ but $x < (a+b)$ $\quad M_x = R_1 x - \frac{w}{2}(x-a)^2$

When $x > (a+b)$ but $x < L$ $\quad M_x = R_1 x - wb(x-d)$

When $x < a$ $\quad \Delta_x = \frac{x}{24EI}[4R_1(x^2 - 3L^2) + wb(b^2 + 12e^2)]$

When $x > a$ but $x < (a+b)$ $\quad \Delta_x = \frac{1}{24EI}[4R_1x(x^2 - 3L^2) + wbx(b^2 + 12e^2) - w(x-a)^4]$

When $x > (a+b)$ but $x < L$ $\quad \Delta_x = \frac{1}{6EI}[3M_{max}(L-x)^2 + R_2(L-x)^3]$

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Δ will be checked at $\Rightarrow$ $x = (62.1'' - 16'')/2 = 23.05''$ (from Brg end)

$a = 0.0$ in

$e = 23.05'' + 16'' = 39.05$ in

$$\Delta_x = \frac{x}{24EI}\left[4R_1(x^2-3L^2)+wb(b^2+12e^2)\right]$$

$E = 30 \times 10^6$ psi (assume carbon steel)

$I = \frac{\pi R^4}{4}$, where R = minimum shaft radius

$I = \frac{\pi(4.725/2)^4}{4} . 24.5 \text{ in}^4$

$b = 62.1 - 16 = 46.1$ in

$d = 46.1 \text{ in}/2 = 23.1$ in

$$R_1 = \frac{wb}{8L^3}\left(12e^2L - 4e^3 + b^2d\right)$$

$$R_1 = \frac{(394)(46.1)}{8(62.1)^3}\left(12(39.05)^2(62.1) - 4(39.05)^3 + (46.1)^2(23.1)\right)$$

$R_1 = 8981$ Lbs

$$\underset{x=23.05}{\Delta @} = \frac{23.05}{24(30\times10^6)(24.5)}\left[4(8981)(23.05^2 - 3(62.1)^2) + (394)(46.1)(46.1^2 + 12(39.05)^2)\right]$$

$$\underset{x=23.05}{\Delta e} = 0.03\text{ in}$$

Conclusions: Since the deflection was shown by a very conservative method to be much less than the airgap of 0.4 in the rotor has been shown to retain its shape and not strike the stator. The Generator has been shown to operate under a 1g loading condition (x, y + z). It should be noted that a very detailed finite element analysis was performed on the LaSalle (ceco) Generator. Although different in some ways the LaSalle generator was shown to have stresses on the order of:

*2262 psi for the Stator

2016 psi for the Rotor

4748 psi for the BRG. Bracket.

*Note - (from S+L EMD-008028 Rev 00 dated 4/77)

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MEMORANDUM OF TELEPHONE CONVERSATION

Date: 5/14/84

Time: 1:30 pm

Person Called: David Wright (Name) of S&L (Company)

Person Calling: Charlie Gangone (Name) of LILCo (Company)

Project: Shoreham Project No. 6995-00

Subject Discussed: Generator Identification

Summary of Discussion, Decisions and Commitments:

Charlie identified the generators as Model No. A-20-0, having the following serial numbers:

71-H3-1056

*67-F4-1003

70-J3-1007

73-K3-1128

*Note - The F4 portion of the number was hard to see and could have been F7.

DW/eg

cc C. Gangone - LILCo
J. Sinnappan - 30
AEM/APD/DMW - 30

David Wright
Signature

File: CQD-014118 - 30

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MEMORANDUM OF TELEPHONE CONVERSATION

Date: 5/16/84

Time: 4:00 P.M.

CQD-014098

Person Called: M. Fleckenstein (Name) of Electro-Motive Div. (Company)

Person Calling: S. A. Ahmad (Name) of S&L (Company)

Project: Shoreham Project No. 6995-00

Subject Discussed: EMD Diesel Generators (S. No. 63610-63611)
Generator Dimensions

Summary of Discussion, Decisions and Commitments:

Mr. Fleckenstien gave the following dimensions relative to the generator sketch prepared by D. M. Wright of S&L, which I delivered to Mr. Fleckenstein on 5/14/83.

Air Gap 0.4"
Shaft Dia. @ the outboard bearing 4.725"
Shaft length between O.B bearing and gearbox coupling 62.1"
Weight of Rotor 8100 lbs.

cc: D. M. Wright -30
AEM/APD/SAA -30
CQD File -30

SAA
Signature

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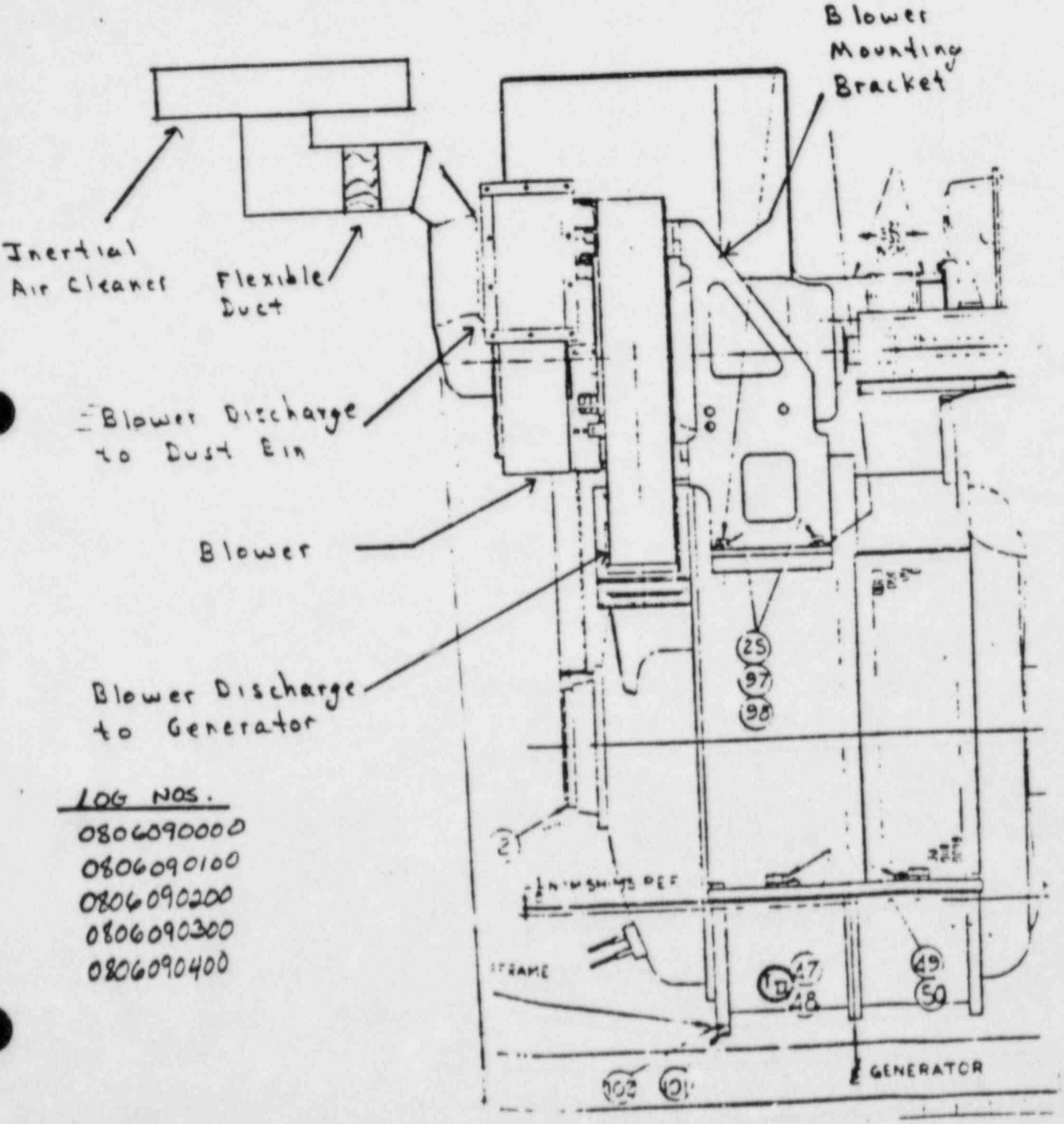
Generators Cooling Systems



LOG NOS.
0806090000
0806090100
0806090200
0806090300
0806090400

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Diesel Generator Cooling System

Description:

The Diesel Generator Cooling System removes heat from the generator casing area by a forced air cooling system. Outside air enters the cooling system and passes through an inertial air Cleaner, where any foreign material is Filtered out. The Filtered air then enters the blower assembly which is driven by the diesel generator. The blower air then passes to the generator casing where heat is removed and is then discharged outside the generator area.

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Study of the Diesel Generator Cooling System

The system study will concentrate on the following areas:

1) Inertial Air Cleaner
2) Cooling System Blower
3) Cooling System Blower Duct (To generator)
4) Cooling System Blower Duct (To Dust Bin)
5) Cooling System Blower Flexible Duct

1) Inertial Air Cleaner

The inertial air cleaner is attached to the generator cooling system by means of a circular duct approximately [illegible] inches in diameter. The mounting details and cleaner configuration is not available at present and will be addressed in the phase III assessment.

It should be noted, however, that if failure of this cleaner should occur during a seismic event, cooling air for the generator will still be maintained.

2) Generator Cooling System

The generator cooling blower is attached to the diesel generator by means of a large steel bracket approximately [illegible] by 13 by 1 inch thick.

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These brackets are mounted onto the generator by means of 4 - 3/4 inch bolts.

These brackets are also used to support the blower motor.

Exact blower/motor weights and dimensions are not available at present, therefore, seismic integrity will shown in the phase III assessment.

It should be noted that if the blower/motor/bracket assembly is considered a rigid assembly the maximum seismic acceleration the assembly will see is the floor response spectra (ZPA - .35g)

Cooling System Blower Duct (To generator)

The cooling system blower duct, which passes air to the diesel generator is approximately 11 x 8 inches and has a running length of 2 to 3 feet from the center line of the blower to the generator casing.

There is also a flexible duct section between the blower and generator casing.

Details of this duct and the flexible duct section are not available at this time and will be addressed in the phase III assessment.

It should be noted, however, that any

seismic loads or movements by the blower will not cause any detrimental effects on the exhaust duct, i.e. the flexible duct section, by its inherent properties, will not allow any load transfer through its body.

4) Cooling System Blower Duct (To Dust Bin)

This cooling duct passes air from the blower to the Dust Bin.

The size of this duct is approximately 18 x 8 inches and also has a flexible duct section between the blower and the dust bin.

Details of the duct and flexible section are not available at this time and will be addressed in the phase II assessment.

It should be noted, however, that failure of this ductwork during a seismic event will not cause a cessation of cooling air to the diesel generator.

5) Cooling System Blower, Flexible Duct

These items are addressed in sections 3 and 4 above.

B.7 SMALL BORE PIPING AND COUPLERS

B.7

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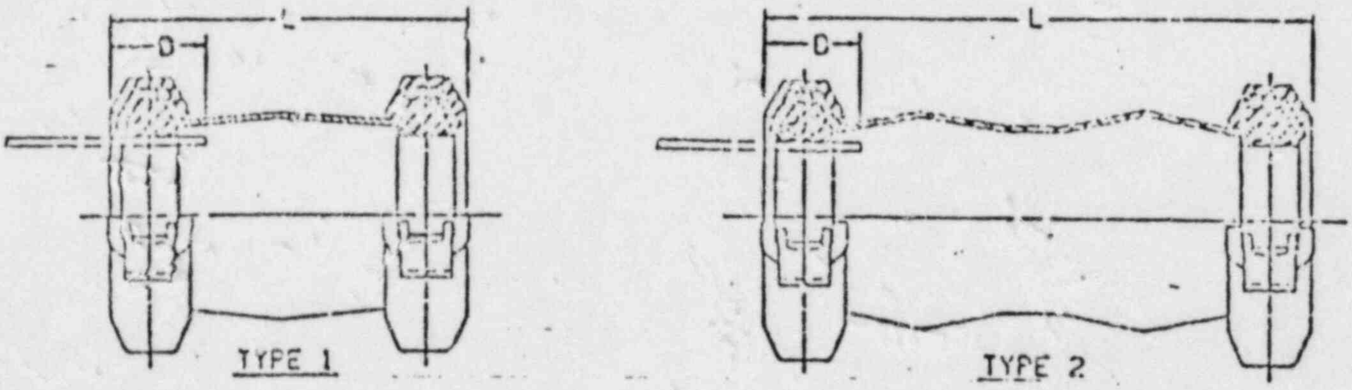
E.D.G.S. On-Skid Pipe Joints

Log Numbers : 0803040100
0803041003
0803041004
0803041008
0803041009
0803161202
0803162301
0803162302



TYPE 1

TYPE 2

The Shoreham E.D.G.S. pipe joints are also manufactured by AEROQUIP Co.. These joints serve as flexible connections for joining two aligned/misaligned pipes. These joints are required to maintain their pressure boundary during Normal, Upset & Emergency plant operating conditions. Since the operating parameters are the same for Shoreham as they are for LaSalle, the following LaSalle pipe joint analysis is deemed to be bounding.

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References:

1. Clinton/LaSalle Enveloped base response spectra - CQD# 011392, rev. 00 - 12/17/83
2. E.D.G.S. Mech. Equip. Qualification List by Log# - 2/7/84
3. E.M.D. dwg.# 8488397; rev. L - 8/9/82
4. E.M.D. dwg.# 8277713, rev. C1 - 5/16/77
5. E.M.D. dwg.# 8350434, - 6/12/64
6. E.M.D. Spare Parts List - 11/74; P.L.# B1810
7. CQD File# 013571, rev. 00 - / / ; Telephone Memorandum of Telephone conversation between Mr. B. Lory of S&L and Mr. K. Swanson of Aeroquip
8. CQD File# 013926, rev. 00 - 4/27/84 ; Telephone Memorandum of of Telephone conversation between Mr. B. Lory of S&L and Mr. Dick Baker of Aeroquip
9. LaSalle E.D.G.S. Piping Walkdown by B. Lory - 1/24/84
10. Formulas for Stress and Strain; Roark & Young - 5th ed.
11. NAVCO Piping Datalog -
12. E.M.D. Component Parameter Listing
13. 1977 ASME Code, Section III of Div. 1; Subsection ND
14. 1977 ASME Appendix I, section III.
15. CQD File# 013933, rev. 00 - 4/30/84; Telephone Memorandum of telephone conversation between Mr. B. Lory of S&L and Mr. D. Baker of Aeroquip
16. Aeroquip Catalog# 869,
17. AISC hdbook - 8th ed - pg. 4-141

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Procedure :

1. The analysis addresses the following items as being the critical modes of failure :

   a. Slippage of whole pipe joint assembly resulting in loss of pressure integrity. This problem is resolved by ensuring proper placement of welded hemispheres ("buttons") around the connected pipes' perimeters.

   b. Overstressing of the pipe joint sleeve due to pressure, dead wt. and thermal effects. The stress analysis criteria shall be the ASME methodology done for piping analysis.

   c. Seismic & Engine Vibration Loads being induced into the pipe joint sleeve because the angular mis-alignment between the pipe joint and a connected pipe has exceeded 4°. This failure mode possibility is resolved by conservatively calculating the longest unsupported pipe spans slope wrt. pipe joint while being subjected to peak seismic accelerations x 1.5.

   d. Tensile failure of the clamp bolt, used for maintaining the pressure integrity of the seal assembly inside the clamp.

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Assumptions & Simplifications:

1. Due to the current lack of dimensions and EPID. part #'s certain pipe joints are assigned the necessary dimensions by correlation of their log #'s and joint interfaces with similar log #'s whose parameters are known.

2. Per ref. #'s 3 and 7 it is apparent that seismic and engine vibration induced loads are absorbed in the pipe joints' BUNA-N seals. This phenomena occurs via seal distortion and changes in the pipe-pipe joint angular misalignment. Minimum wall thicknesses and pipe materials are used in calculating θ's.

3. It is assumed that the axial thermal expansion of the two connected pipes is much less than the space between them, ie. L-2D. This is reasonable since via the field sketches and pictures the lengths of the connected pipes are relatively small.

4. In calculating the pipe joint edge-to-pipe joint stop distances (ie. $\bar{X}$), we assume that initially the pipe joints are centered during their installation.

5. By considering each connected 3-D pipe run as one long cantilever beam in calculating θ wrt to its pipe joint we have a conservative value of θ — in actuality, during a seismic event, 3-D pipe runs with a few bends will have discreet pipe slopes (θ's) which may add or subtract to the resultant θ at the pipe joint location.

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6. The thermal stress for ASME stress analysis, M_c, is considered as the SRSS value of the hoop and circumferential pipe joint stresses. This is assumed to be equivalent to the equivalent bending moment M_c due to thermal effects.

7. The stress intensification factor used in the stress analysis at the pipe joint sleeves is equal to 2.3 which is for threaded pipe joints. It is assumed that the minute differences between the Aeroquip pipe joints used and a threaded pipe joint are negligible.

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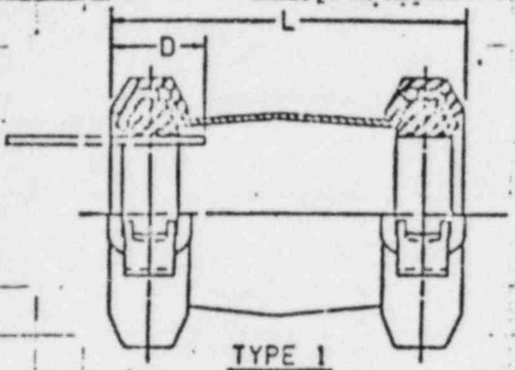
Summary Table of Type #1 Flexmaster Pipe Joints : (ref. #2-6.)



TYPE 1

*

	Joint #	Pipe Size	Loc #	EMD Pt #	Joint Interface	Working Fluids	L (in)	D (in)
	1	4"	0303041003	8350434	Water Pump Inlet	Water	6.50	1.07
+	2	4"	0303041004	8393725	Engine Water Outlet	"	7.50	1.88
**	3	6"	0303041005	8393725	AMOT to Oil Cooler.	"	6.50	1.26
	5	1"	0303041009	8470340	Overflow Tee	"	2.88	0.81
□	6	1¼"	0303162301	----	Soak & Circ Pumps	Lube Oil	3.25	0.93
	7	4"	0303162302	----	Oil Cooler to Oil Strainer	"	7.50	1.88
	8	4"	0503041003	8277713	Water Pump Inlet	Water	6.50	1.07
+	9	4"	0503041004	8252815	Engine Water Outlet	"	7.50	1.88
+	10	4"	0503041004	8393725	"	"	7.50	1.88
**	11	6"	0503041005	8393725	AMOT to Oil Cooler	"	6.50	1.26
	15	1"	0503041009	8470340	Overflow Tee	"	2.88	0.81
□	16	1¼"	0503162301		Soak & Circ Pumps	Lube Oil	3.25	0.93
	17	4"	0503162302	8252895	Oil Cooler to Oil Strainer	"	7.50	1.88
+	18	4"	0201041004	8252895	Engine Water Outlet	Water	7.50	1.88
	22	1"	0201041009	8470340	Overflow Tee	"	2.88	0.81
□	23	1¼"	0201162301		Soak & Circ. Pumps	Lube Oil	3.25	0.93
	24	4"	0201162302	8252895	Oil Cooler to Oil Strainer	"	7.50	1.88
+	25	4"	0202041004	8252895	Engine Water Outlet	Water	7.50	1.88
	27	1"	[illegible]	8470340	Overflow Tee	"	2.88	0.21

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Joint #	Pipe Size	Loc #	EMD Part #	Joint Interface	Working Fluid	L (in)	D (in)
□ 28	1¼"	0202162301		Soak & Circ Pumps	Lube oil	3.25	0.93
29	4"	0202162302	8252895	Oil Cooler to Strainer	"	7.50	1.88
30	4"	0601041003	8277713	Water Pump Inlet	Water	6.50	1.07
† 31	4"	0601041004	8147711	Engine Water Outlet	"	6.50	1.86
*† 32	6"	0601041005	8393725	AMOT to Oil Cooler	"	6.50	1.86
36	1"	0601041009	8470340	Overflow Tee	"	2.88	0.81
□ 37	1¼"	0601162301		Soak & Circ Pump	Lube Oil	3.25	0.93
38	4"	0601162302		Oil Cooler to Strainer	"	7.50	1.88

** See [illegible]

[illegible] is only for [illegible] purposes [illegible] See [illegible] = see [illegible]

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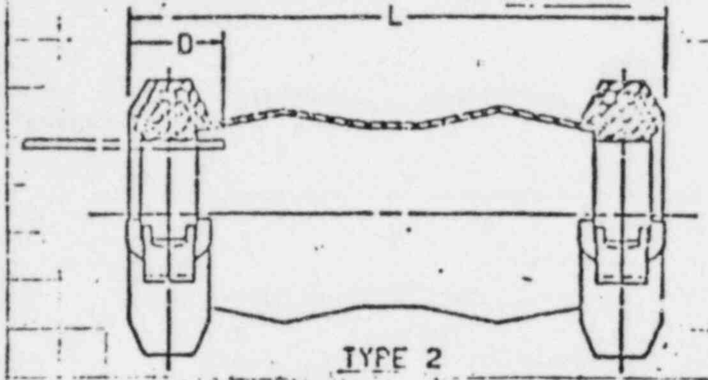
Summary Table of Type #2 Flexmaster Pipe Joints:



	Joint #	Pipe Size	Loc #	EMD Part #	Joint Interface	Working Fluid	L (in)	D (in)
++	11	"	0303041008	8362588	Exp. Tank to Aspirator	Wat.	4.50	0.75
	12	3/4"	0503041007	8470605	Cooling System Vent	"	4.50	0.75
++	13	"	0503041008	8362588	Exp. Tank to Aspirator	"	4.50	0.75
++	14	"	0503041008	8166524	"	"	4.50	0.75
+	19	3/4"	0201041007	8470605	Cooling System Vent	"	4.50	0.75
*+	20	"	0201041008	8362588	Exp. Tank to Aspirator	"	4.50	0.75
*+	21	"	0201041008	8166524	"	"	4.50	0.75
+	25	3/4"	0202041007	8470605	Cooling System Vent	"	4.50	0.75
+	33	3/4"	0601041007	8470605	"	"	4.50	0.75
*	34		0601041003	8362588	Exp. Tank to Aspirator	"	4.50	0.75
*	35		0601041008	8166524	"	"	4.50	0.75

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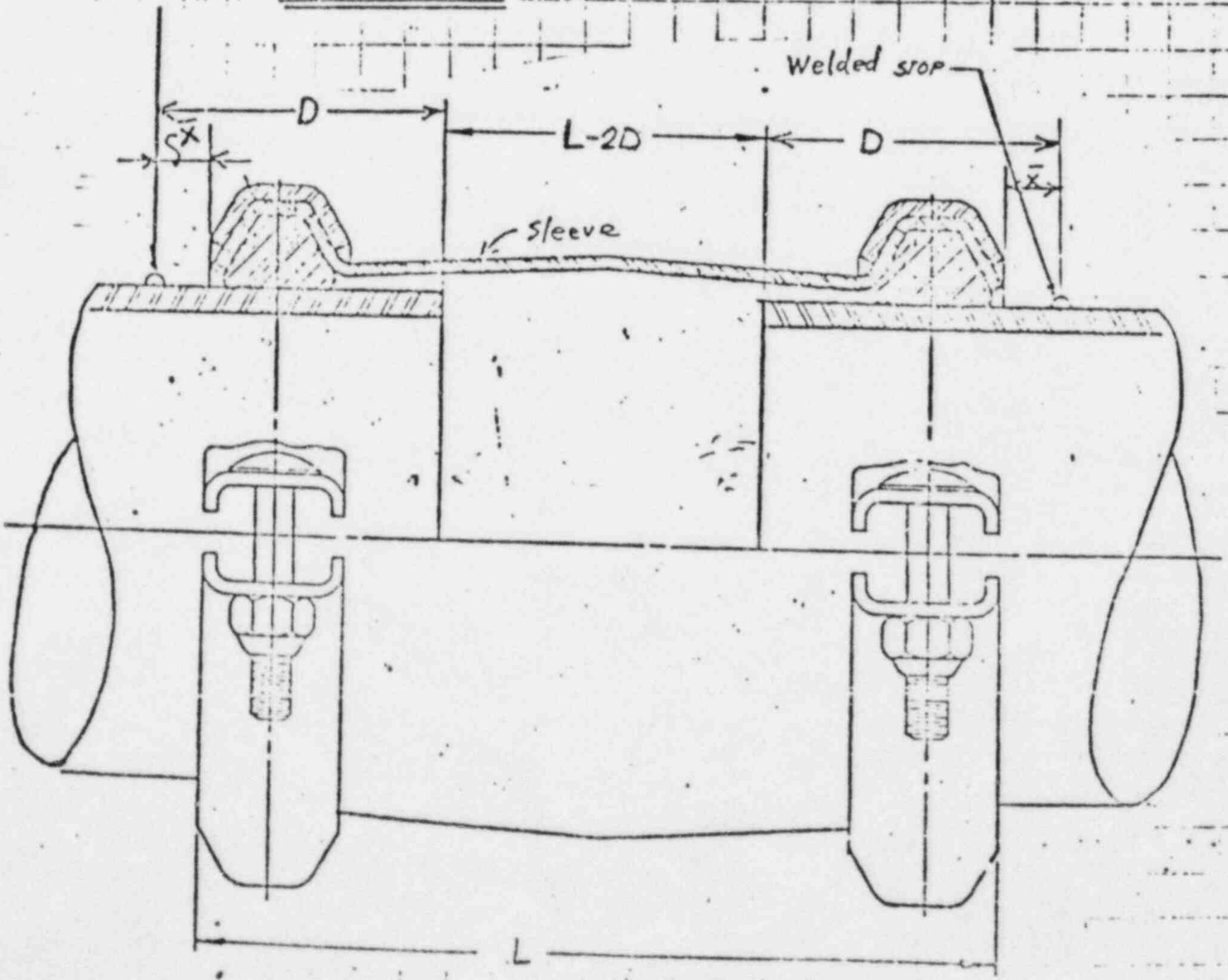
Welded Pipe Joint Stop Placement Criteria :

The following table represents calculated distances from the pipe joints' ends to the pipe joint stops. Let this distance be denoted as $\overline{X}$. This value of $\overline{X}$ is arbitrarily calculated to be : (using engineering judgement)

$$\overline{X} = \frac{D}{2}$$



Hence the following table contains the recommended $\overline{X}$ as a fn. ϕ Pipe-

Pipe O.D.	3/4"	1"	1¼"	4"	6"

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Calculation of Max. Pipe Slopes for Pipe Joints:

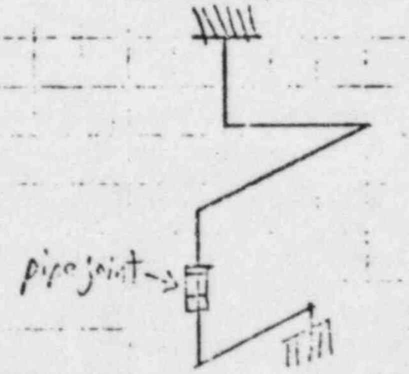
Principle: Per ref. #8 if the connected pipes angular misalignments with the pipe joint is ≤ 4°, the seismic, dead wt. and engine vibration loads will only cause distortion in the BUNA-N seals and possibly a slightly larger misalignment (θ).

For each pipe run having a pipe joint the following steps will be used for conservatively calculating the maximum possible pipe slope (θ) during a seismic event. Vertical pipe runs will utilize peak SRSS g value of two horizontal response spectra x 1.5 and horizontal pipe runs will utilize V.S. response spectra peak value x 1.5.

Steps:

1. given 3-D pipe run:

3-D Pipe Run



Deflected under Seismic

2. Split the pipe run into two pipe segments at the pipe joint

3. If each pipe segment is the same O.D. select the longer one for analysis.

4. Conservatively model this one pipe segment as a cantilever beam subjected to the appropriate seismic acceleration and calculate the beam's slope (θ)

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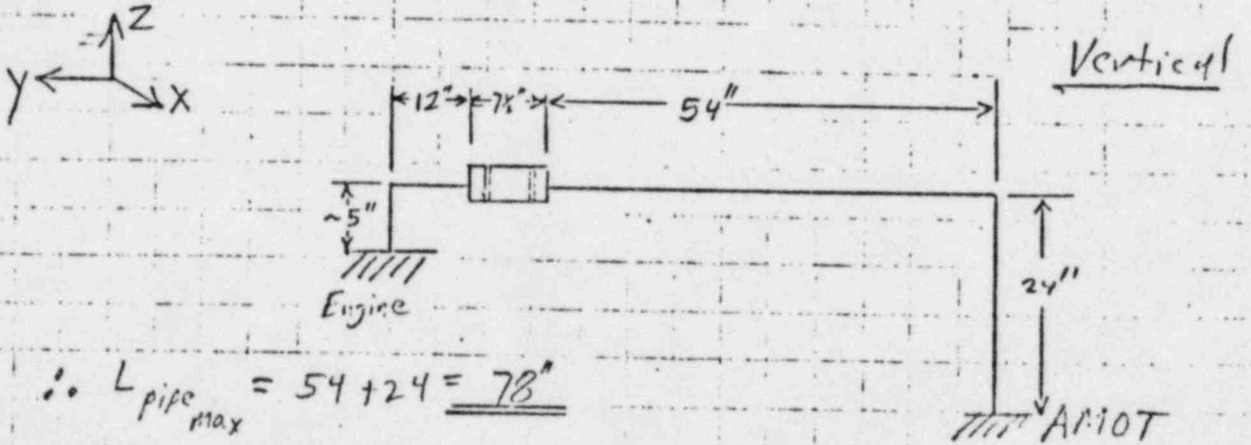
The following pipe runs are either from ref. #9 or are approximations derived through examination of, pictures included in this calculation.

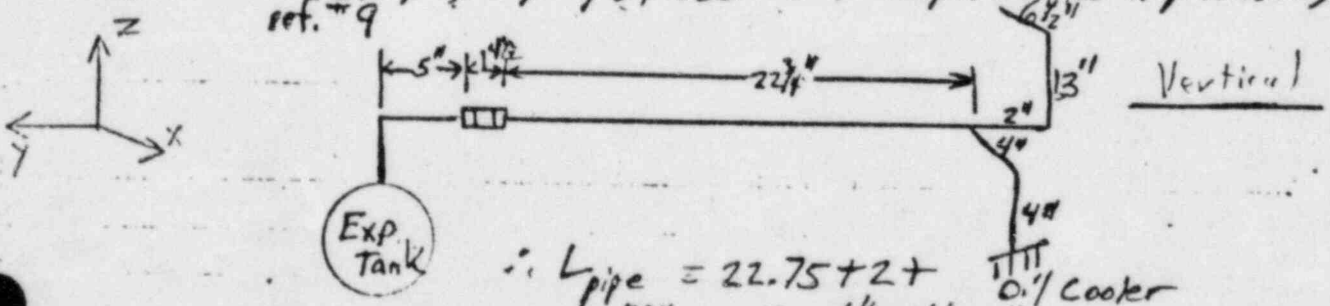
I. Pipe Joint #s 2, 9, 10, 18, 25 & 31: (4" ϕ - Engine Water Outlet)

These Clinton/LaSalle pipe runs having pipe joints would approximately look like this - (see picture #'s IE-4 & 7E-2)



$\therefore L_{pipe_{max}} = 54 + 24 = \underline{78"}$

II. Pipe Joint #s 4, 13, 14, 20, 21, 34 & 35 (¾" Exp. Tank to Aspirator)
ref. #9



$\therefore L_{pipe_{max}} = 22.75 + 2 + 13 + 6\frac{1}{2} = \underline{44.75"}$

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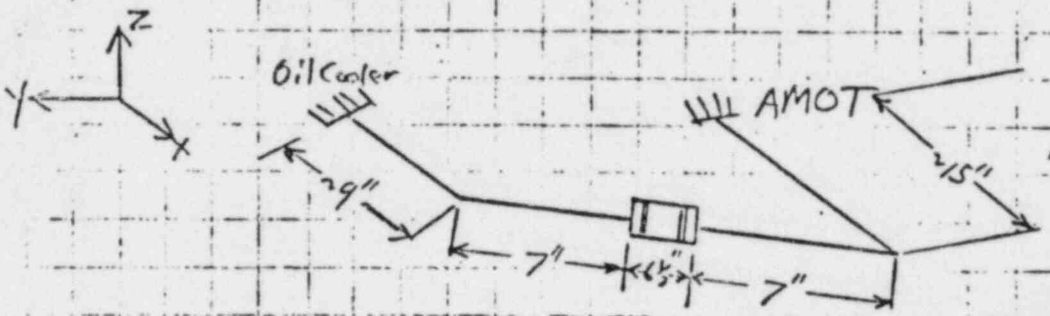
III. Pipe Joint #'s 3, 11, & 32 (6" AMOT to Oil Cooler)
(see picture #'s 1X-15 & 2F-7)

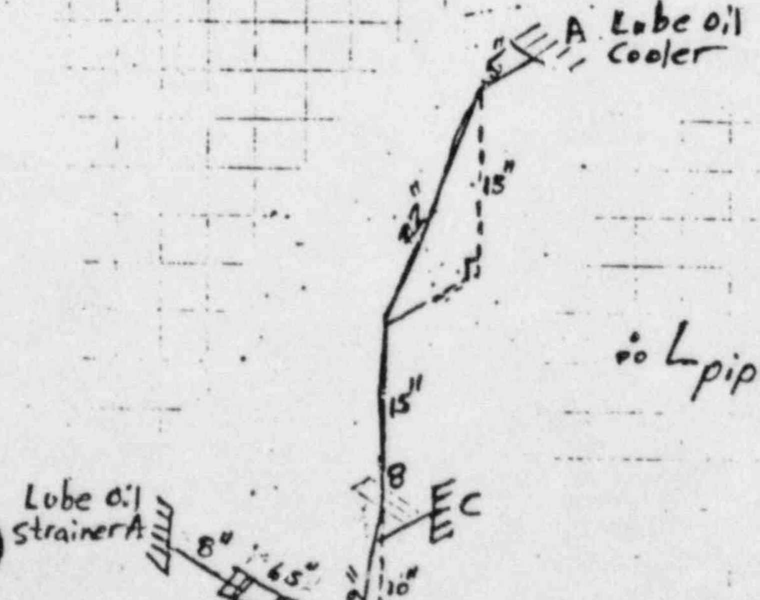
Horizontal



$\therefore L_{pipe_{max}} = 7 + 15 = \underline{\underline{22"}}$

IV. Pipe Joint #'s 7, 17, 24, 29 & 38 (4"Φ Oil Cooler to Oil Strainer)
(ref. #9

Horizontal



$\therefore L_{pipe_{max}} = 9 + 10 = \underline{\underline{19"}}$

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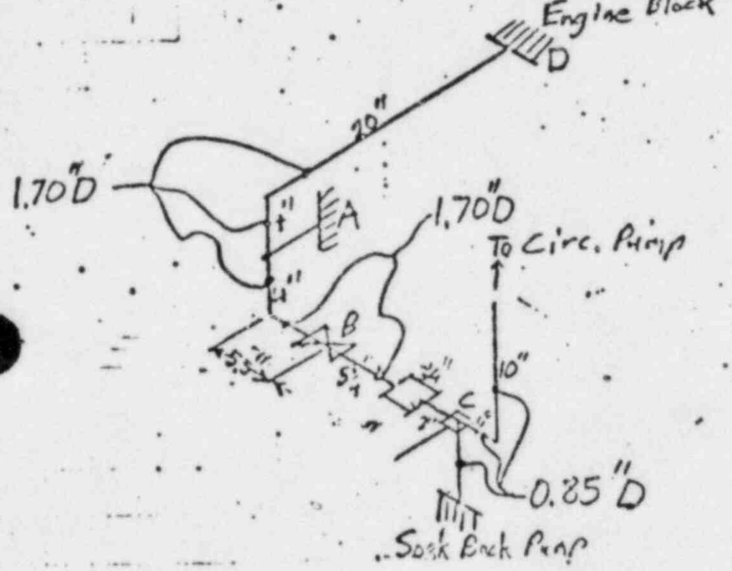
V, Pipe Joint #'s 6, 16, 23, 28 & 37 (1¼" ∅ Soak-Circ Pumps)
(ref. #9, pgs. & 12 of this calc.)

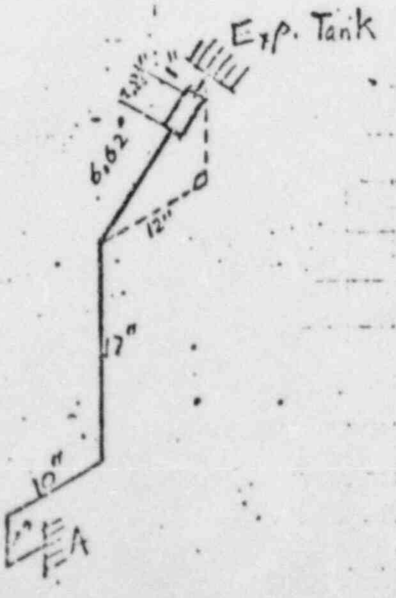


$L_{pipe_{max}} = 5.5 + 4 + 5\frac{1}{4} = 14\frac{3}{4}''$

(Note: have concentrated mass at Pt. B also)

Horizontal

VI Pipe Joint #'s 5, 15, 22, 27 & 36 (1"∅ overflow Tee)
(ref. #9, see p. of this calc)



$L_{pipe_{max}} = 7 + 10 + 17 + 6.62 \approx 40.62''$

Vertical

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VII. Pipe Joint #'s 1, 8, & 30 (4"ϕ Water Pump Inlet)

From picture 1A-13 it is intuitively obvious that the unsupported pipe lengths connected by these pipe joints is much less than $L_{pipe_{max}}$ calculated earlier.

VIII. Pipe Joint #'s 12, 19, 26 & 33 (3/4"ϕ Cooling System Vent)

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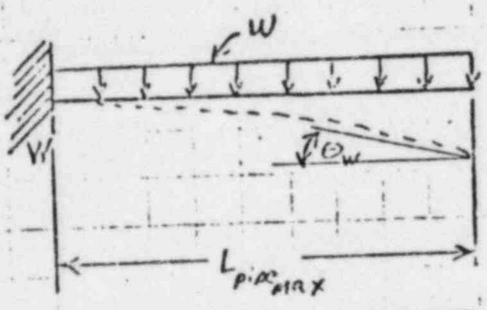
Summary Table for Calculating Max. Pipe-Pipe Joint θ:

Beam Model:



$E = 29.0 \times 10^6 psi$

$$\theta_w = \frac{wL^3}{6EI}$$ (ref. #10 - p. 98)

Note: W for vertical pipe = $w(1.85g)(1.5) = 2.78g$
w for horizontal pipe = $w(4.75g)(1.5) = 7.13g$

Summary Table

	Pipe Size	$L_{pipe_{max}}$	*$I_{(in^4)}$	Max Work Press	Design Press.	Fluid Temp (max)	$w_{seismic}$ (lb/in)	θ_w (rad)	θ_w (°)
	3/4"		0.0370	15psi		190°F	$(1.13+0.231)\left(\frac{}{12}\right)$ $= 0.$		
V	1"	40.62"	0.0873	10psi		200°F	$(1.68+0.374)\left(\frac{2.78}{11.2}\right)$ $=0.476$	0.0021	0.12°
H	1¼"	14.75"	0.1947	50psi		235°F	$(2.27+0.647)\left(\frac{7.13}{12}\right)$ $=1.733$	0.00016	0.01°
V	4"	78.00	7.23	70psi		235°F	$(10.79+5.51)\left(\frac{12.78}{12}\right)$ $=3.777$	0.00142	0.08°
H	6"	22.00	28.14	35psi		200°F	$(18.97+12.51)\left(\frac{7.13}{12}\right)$ $=18.704$	0.0000	0.0°

From the above table it is obvious that even using a very conservative model with maximum seismic loading, the maximum pipe-to-pipe joint angular misalignment is much less than the max. allowed 4°. Hence seismic dead wt. and engine vibration loads are omitted from the pipe joint sleeve stress analysis.

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Pipe Joint Sleeve Stress Analysis:

I. Methodology - These pipe joint sleeves will be analyzed just like the on-skid pipe stress analysis was done. The following ASME criteria is applicable. Note that M_A & M_B in the following equations are zero.

1st Condition: Sustained & Sustained + Occasional Loads -

$$*S_L = S_{OL} = \frac{P_{max} D_o}{4 t_n} \le 1.0 S_h \; : \; \frac{0.75 i M_A}{Z} = \frac{0.75 i (M_A + M_B)}{Z} = 0$$

2nd Condition: Thermal Expansion -

$$*S_{TE} = \frac{P D_o}{4 t_n} + i\left(\frac{M_c}{Z}\right) \le (S_h + S_A)$$

where: $Z = \pi r_m^2 t_n$

S_h = Allowable stress for A-513 @ T_{hot} from ref. #14

r_m = mean sleeve radius

$** S_A = f(1.25 S_c + 0.25 S_h)$ where $f = 1.0$ for 7,000 or less full temp cycles

S_c = allowable Stress for A-513 @ 70°F

let $\frac{M_c}{Z} = \sqrt{(\sigma_1)^2 + (\sigma_2)^2}$

σ_1 = hoop stress due to ΔT

σ_2 = circumferential stress due to ΔT

II. Sleeve Material Allowable Stresses —

Sleeve Mat'l : A-513 (ref. #7)

$S_{u_{min}} = 40{,}000\,psi$ } ref. #15

$S_{y_{min}} = 25{,}000\,psi$

$\nu = 0.27$ (ref. #5 - pg. 607)

$E = 27.7 \times 10^6\,psi$ @ 200°F (ref. #14 - Table I-6.0)

$\alpha = 6.77 \times 10^{-6}\,in/in\text{-}°F$ @250°F (ref. #14 - Table I-5.0)

$S = 11{,}200\,psi$ for SA-155, typ. C45

($S_{y_{min}} = 24.5\,ksi$, $S_{u_{min}} = 45.0\,ksi$)

$i = 2.3$ (ref. #14 - Fig. ND-3673.2(b)-1)

$S_c = S_h = 11{,}200\,psi$

$S_A = f(1.25\,S_c + 0.25\,S_h) = 1.0(1.5(11{,}200)) = 16{,}800\,psi$

III. Pressure Stresses —

Note: For cylindrical shells - if $\frac{R_o}{t} \geq 10$ use thin-shell theory

" " " - if $\frac{R_o}{t} < 10$ use thick-shell theory

∴ For 3/4"φ Pipe Joint : $\frac{R_o}{t} = \frac{\frac{1}{2}(1.312)}{0.065} = 10.09$

" 1"φ " " : $\frac{R_o}{t} = \frac{\frac{1}{2}(1.625)}{0.065} = 12.50$

" 1¼"φ " " : $\frac{R_o}{t} = \frac{\frac{1}{2}(1.975)}{0.065} = 15.19$

" 4" " " : $\frac{R_o}{t} = \frac{\frac{1}{2}(5.00)}{0.120} = 20.83$

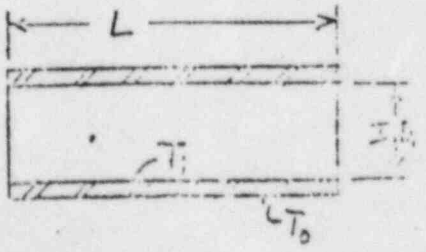
" 6" " " : $\frac{R_o}{t} = \frac{\frac{1}{2}(7.00)}{0.120} = 29.17$

$$\text{hoop stress} = \sigma_1 = \frac{PR_o}{2t}$$
$$\text{circumferentinl stress} = \sigma_2 = \frac{PR_o}{t}$$
} ref. # 10 - pg 448

pipe size	working pressure	working stresses σ_1	σ_2	Design pressure	Max stresses σ_1	σ_2
3/4"	15psi	75.7psi	151psi	300psi	1,513.5psi	3,027psi
1"	10psi	62.5psi	125psi	200psi	1,250psi	2,500psi
1 1/4"	50psi	380psi	760psi	200psi	1,519psi	3,038psi
4"	70psi	729.1psi	1,458psi	150psi	1,562.3psi	3,125psi
6"	35psi	1510.5psi	1,0[illegible]psi	150psi	2,187.8psi	4,376psi

IV. Thermal Stresses:

Since the pipe joints and connected piping are not restrained from longitudinal (axial) expansion due to ΔT, the corresponding hoop stresses are 0. However there is another hoop stress indice due to a ΔT between the inside & outside pipe joint surfaces.



Assume uniform T_i on inside pipe surface and uniform T_o on the outside pipe surface.

Pipe Joint Cross Section (lits)

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Assuming a linear temperature gradient across the pipe joint sleeve and a maximum ΔT of $10^{\circ}F$ the following table of circumferential and hoop stresses is generated using the following equations from ref. #5 - pg. 585.

$$\sigma_{circumferential_{max}} = \frac{\frac{1}{2}\Delta T \alpha E}{1-\nu} = \sigma_{hoop_{max}}$$

$$\therefore \sigma_{1_{max}} = \sigma_{2_{max}} = \frac{\frac{1}{2}(10^{\circ}F)(6.77\times10^{-6}\,\frac{in}{in\,^{\circ}F})(27.7\times10^{6}\,psi)}{1-0.27} = \underline{\underline{1,284.4\,psi}}$$

$$\therefore \frac{M_c}{Z} = \sqrt{2(1,284.4)^2} = \underline{\underline{1,816.4\,psi}}$$

V. Pipe Sleeve Stress Criteria -

1^{st} Condition -

$$S_L = S_{OL} = \frac{P_{max}\,D_o}{4\,t_n} \leq 1.0\,S_h \quad ie.$$

Using max. P, D_o and min. t_n we address all the applicable pipe joints in one calculation:

$$S_L = S_{OL} = \frac{300\,(7.0)}{4(0.065)} = \underline{8,077\,psi} \leq 11,200\,psi \checkmark \text{ Adequate}$$

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2nd Condition -

$$S_{TE} = \frac{P_{max} D_o}{4 t_n} + i\left(\frac{M_c}{Z}\right) \leq S_h + S_A$$

$$\leq 11,200 + 16,800$$

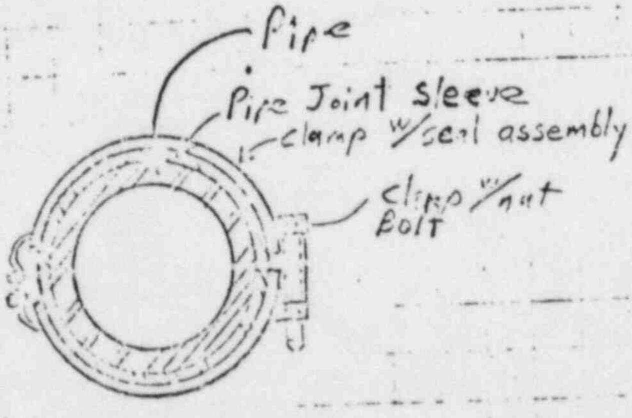
$$\leq 28,000$$

For all the Pipe Joints -

$$S_{TE} = 8,077 + 2.3(1,816.4) \leq 28,000$$

$$S_{TE} = 12,254.7\ psi \leq 28,000\ psi$$ ✓ Pipe Joint Sleeves are allowable

Clamp Bolt Tensile Stress Analysis :



Due to the ΔT in the radial direction & assuming that the pipe radial expansion is not permitted via seal distortion we can calculate the tensile bolt force by multiplying the circumferential stress due to ΔT by the pipe's longitudinal area.

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$\sigma_{c_{max}}$ = 642.2 psi for all size pipe based upon 5°F ΔT

To compute pipe longitudinal areas we use D_{max} from ref. 7-3 as pipe length L and by assuming that the connected pipes are standard we generate the following table:

Pipe size	D_{max} (in)	t_n (in)	Area (in)²	F_{bolt} (lbs)	Bolt Tensile Area (in²)	Bolt Tensile Stress
¾"	1.00	0.113	0.226	145	0.032	4.53ksi
1"	1.25	0.133	0.333	214	0.052	4.12ksi
1¼"	1.38	0.140	0.386	248	0.052	4.77ksi
4"	2.81	0.237	1.332	855	0.068	12.57ksi
6"	1.86	0.280	1.042	669	0.142	4.71ksi

Equations: $Area = 2(t_n)(D_{max})$

$$F_{bolt_1} = (\sigma_{c\,max})(Area_1)$$

Allowable Tensile Stress = $0.61\,S_y$ = 0.61(36.0ksi) = <u>21.85ksi</u>
(ref. 7-13 - NF3280)

∴ Since all clamp bolt stresses are well below its tensile stress allowable, Hence Bolt pre-torque induced stresses will not adversely affect these results and the bolts are adequate.
[illegible]

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Calcs. For Seismic Qualification of E.D.G.S. On-Skid Piping & Pipe Jo

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ent LILCo		Prepared by Bruce H. Lory	Date 5-9-84
Project Shoreham - 1		Reviewed by	Date
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Purpose: The purpose of this calculation is to ensure that the subject piping is built to withstand the applicable piping loads for the Normal, Upset and Emergency plant operating conditions. The pipings' structural & pressure integrity are examined in accordance with the 19 ASME Code, Div.1, Section III, Subsection ND.

Note- for the purpose of this analysis the following will be used:

The pipe will be assumed to be ASME Class III.

References:

1. CQD File - 010330, rev.00 - / 84; Seismic Qualification Analysis of LaSalle E.D.G.S.
2. LaSalle Response Spectra;

Direction	Spectra No.	rev.	date
ØBE-EW	102&111-OB-EW	01	4-26-73
ØBE-NS	102&111-03-NS	01	4-23-73
ØBE-VS	110&111-OB-VS	01	7-19-73
SSE-EW	102&111&118-DB-EW	01	12-9-73
SSE-NS	102&111-DB-NS	01	12-9-73
SSE-V.S	102&111-DB-VS	01	7-25-73

3. Not Used

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4. Shoreham Seismic Response Spectra: RSG Program: RunID SIKX - 6/12/84

5. S&L Qualification status of E.D.G.S. Mechanical Components for LILCo MP-45 D.G.'s - rev.00 - 4/30/84

6. S&L Field Trip to Shoreham site - 5/3/84, 5/4/84

7. NAVCo Piping Datalog - 10th ed - 6/1/74

8. AISC Hdbook - 8th ed.

9. 1977 ASME Code, Div. 1, Sect. III, Subsection ND

10. 1977 ASME B+PV code, section III, Appendix I.

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Assumptions :

1. It is assumed that the pressure/temperature parameters for the Shoreham E.D.G.S. are identical to those for LaSalle. (Good assumption, same Engine type + model #.)
2. The stress analysis performed on the Shoreham pipe runs are resulting from the assumption that the runs exhibit non-rigid body natural frequencies. This is a very conservative approach since the seismic loads are derived from peak seismic accelerations X 1.5.

3. Since the LaSalle analysis utilized 1.5 x peak seismic accelerations in all three (X-Y-Z) orthogonal directions, it is assumed that the similarity study can justifiably be based upon comparing max. unsupported pipe lengths for each log # -

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The following log numbers correspond to the E.D.G.S. on-skid piping runs addresed in this section:

0803040101
0803040104
0803040106
0803040107
0803040108
0803040109
0803040110
0803040111
0803040603

0803040604
0803040703
0803040704
0803040705
0803040706
0803060701
0803060702
0803060703
0803060708

0803161400
0803161500
0803162201
0803162202
0803162203
0803162204
0803162205
0803162206
0803162207
0803162208

Analysis Methodology:

1. Both the LaSalle /Shoreham Emergency Diesel Generator Engines are manufactured by EMD, and upon examination of available pictures it becomes clear that a majority of the Shoreham piping runs are replicate to those on LaSalle. This is especially true for the accessory rack piping. Therefore a majority of the Shoreham E.D.G.S. piping is qualified on the basis of similarity. These pipe runs are addressed and deemed adequate as designed by comparison of similar or identical pipe runs on LaSalle because:

   a. It can be shown that, for a particular pipe run (Log #), the LaSalle pipe run analyzed

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in ref. #1 is of longer, unsupported length than the corresponding pipe run on Shoreham's E.D.G.S..

b. Most of the LaSalle piping was analyzed using peak seismic accelleration values x 1.5 which is based upon the assumption that the LaSalle pipe runs exhibit non-rigid body natural frequencies.

c. The LaSalle analysis (in ref. #1) utilized "weakest" pipe material, consequently therefore the smallest ASME allowable stresses were used.

d. The LaSalle analysis assumed minimum wall thickness for calculating the corresponding pipe section modulus. This resulted in max. pipe stresses. Shoreham's piping is Sch 40.

e. The LaSalle EDGS enveloped response spectra completely envelopes the Shoreham OBE (1%) and SSE (2%) response spectra.

For these reasons replicate or nearly replicate Shoreham pipe runs are seismically adequate as designed provided it is shown that for each pipe run, the LaSalle maximum unsupported pipe length is greater than or equal to the corresponding Shoreham pipe run.

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2. Any unique Shoreham pipe runs are analyzed using the same conservative technique used in ref.# 1 for the LaSalle piping analysis. Consequently the Shoreham pipe runs are assumed to exhibit non-rigid body natural frequencies, which is a conservative assumption.

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Qualified Piping Runs by Similarity :

The following table represents those Shoreham pipe runs which are deemed to be seismically adequate based upon the similarity methodology outlined earlier.

Cooling System

Pipe Size D.L.	Shoreham Log #	Pipe Run Label	Max. LaSalle Pipe length (in)	LaSalle pg. # in App.	Max. Shoreham Pipe Length (in)	Shoreham Picture #'s
*	0803040106	Oil Cooler to Water Pumps	*	*	*	11
1¼	0803040107	Drain Piping	52"	B.7.34	L<<52	8, 8.1
1¼	0803040108	Immersion Heater Piping	40.5"	B.7.35	L<<40.5	54
1	0803040110	Exp. Tank Overflow	47.5"	B.7.36	39	10, 11
2½	0803040111	Exp. Tank to Water Pumps	8,25"	B.7.37	8,25	12

Form GQ 3.08.1 Rev. 2

* Unique to EMD Engines ∴ L. =L.

CALC NO. COD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.7.27

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Re

Client
Project
Proj. No. 6995-00 Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

Fuel Oil System

Pipe Size (in.)	Shoreham Log #	Pipe Run Label	Max. LaSalle Pipe length (in)	LaSalle pg. # in Log	Max. Shoreham Pipe Length (in)	Shoreham Pict. #
5/8	0803060708	Suction Strainer to Fuel Pump	98.5	B.7.38	L < 72	9, 24

Form GO-3.08.1 Rev. 2

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Re

Client

Project

Proj. No. 6995-00 Equip. No.

Prepared by Date

Reviewed by Date

Approved by Date

Fuel Oil System

Pipe Size O.D.	Shoreham Log #	Pipe Run Label	Max. LaSalle Pipe length (ft)	LaSalle pg. # in App.	Max. Shoreham Pipe Length (ft)	Shoreham [illegible] #
5/8	0803060708	Suction Strainer to Fuel Pump	98.5	B.7.38	L < 72	9, 24

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
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PAGE B.7.28

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Rel.

Client
Project
Proj. No. 6995-00 Equip No.

Prepared by Date
Reviewed by Date
Approved by Date

Lube Oil System

Pipe Size O.D.	Shoreham Log #	Pipe Run Label	Max. LaSalle Pipe length (in)	LaSalle pg. # in Log	Max. Shoreham Pipe Length (in)	Shoreham Picture #s
½	0803162201	Y-Strainer to Soak Back Filter	59"	B.7.39	56"	32-36, 38
1	0803162203	Circ Pump to Y-Strainer	25"	B.7.39	18"	32,
3	0803162205	Scavenging Pump to Oil Filter	47"	B.7.40	47"	43, 44
3	0803162206	Oil Filter to Oil Cooler	32"	B.7.41	32"	45
4	0803162207	Oil Cooler to Oil Strainer	45"	B.7.42	44"	46-49
½	0803162208	Oil Filter Inlet from Y-strainer	35.5"	B.7.39	12"	32, 43

Form GQ 3.01.1 Rev. 2

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Relate

Client
Project
Proj. No. 6995-00 Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

A majority of the Shoreham E.D.G.S. piping runs are replicate to the corresponding LaSalle piping runs. It is shown on the following pages that the applicable LaSalle OBE (1%) and SSE (2%) max. response spectra peak accelerations (3.5g-vert. & 1.8g-horiz.) are much larger than the 0.62g Shoreham peak acceleration. (see pg. B.7.33).

The LaSalle analysis utilized the method of static coefficients per IEEE-344 (1975) - ie. 1.5 x peak acceleration values were used in the LaSalle piping analysis. Since the LaSalle pipe runs are seismically adequate (and conforms to the stress requirements of the 1977 ASME B&PV Code, Division 1, Section III, Subsection ND-3600), the replicate pipe runs on Shoreham are obviously adequate as designed.

The unique pipe runs on Shoreham are analyzed in the same manner, ie. 1.5 x peak accelerations in all three directions for base-mounted pipe runs. Additional conservatism was utilized when possible by calculating pipe stresses based upon 2 x peak seismic acceleration values.

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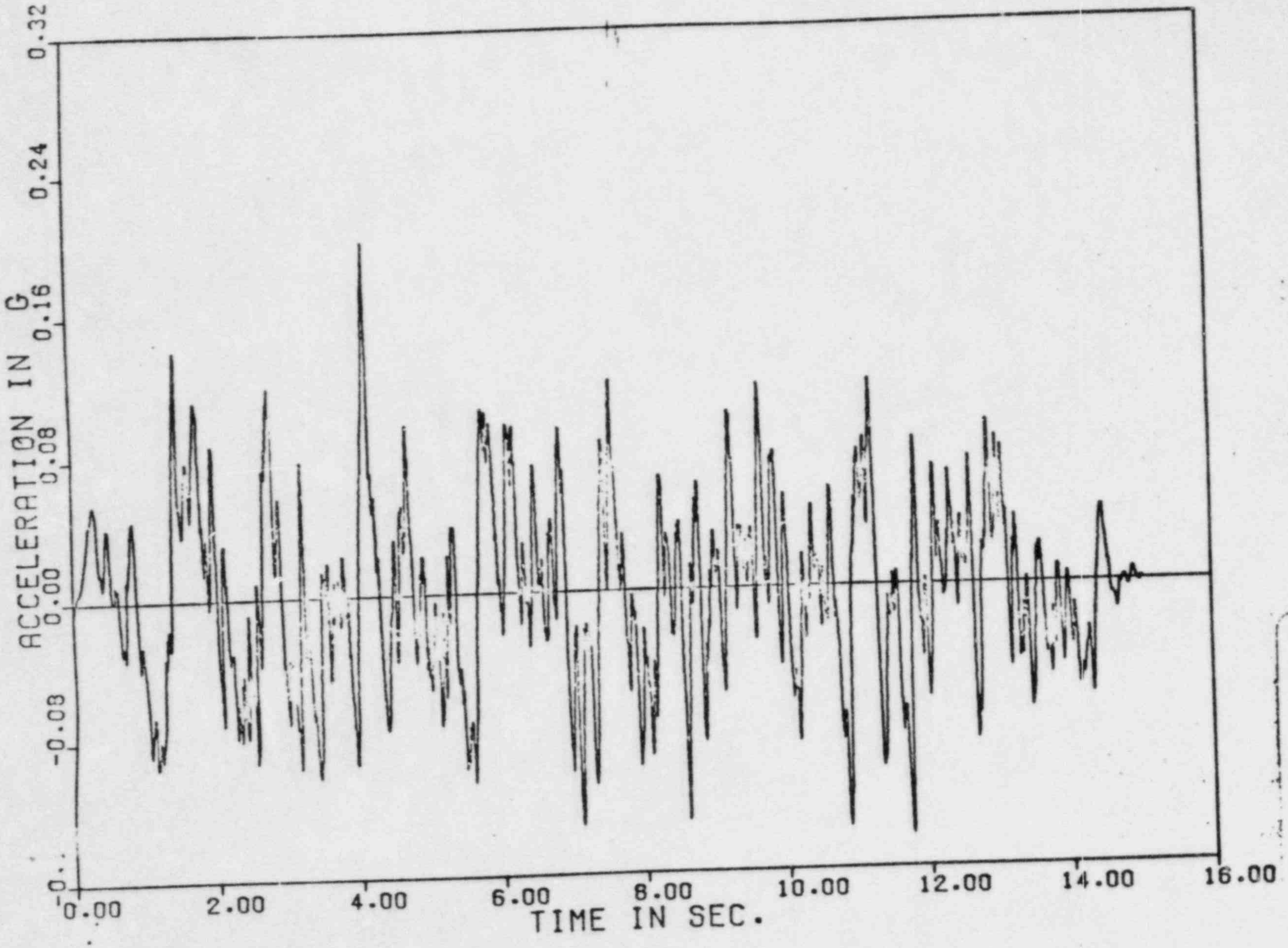
IKX JUN 84

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.7.32



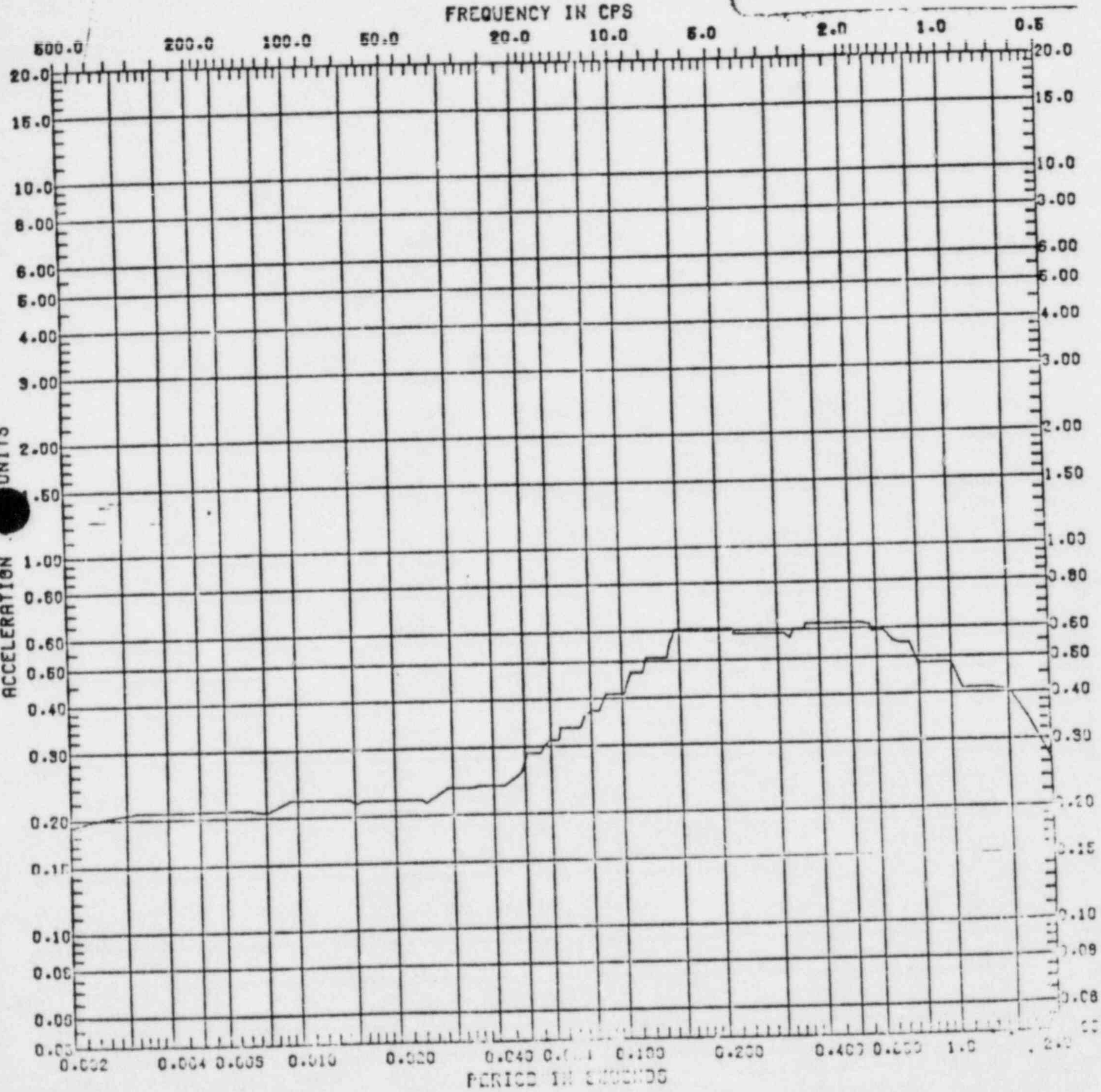
BASE SSE 2% GROUND EDGS HOR N/A

SARGENT & LUNDY
ENGINEERS

12 JUN 84
SIKX

CALC NO. CQD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.020
PAGE

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.7.33



LOADING:SSE(2%) EDGS
NODE BASE
DIRECTION HOR ANGLE N/A

SPECTRA NO. SSE 2%
ELEVATION GROUND
LOCATION EDGS

1

LaSalle EDGS Piping Qualification

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.7.34

"Critical" Cooling Drain Piping System Isometric

rre quip. No.: 0,1,2DG01K;1,2E22-S001
Pipe Run(s): C/4
rresp. Log. No.(s): 0303040107
0503040107

rresp. Photo No.(s): e1, 0E-9, 3E-14

rresp. Spec.: J-2541, T-2500

ared by Bruce M. Long Date 1-24-84
ewed by David Wright Date 1-31-84
roved by Date

Parameters

Op. Pressure: 25 psi
Dsgn. " :
Op. Temp$_{max}$: 200° (assumed)

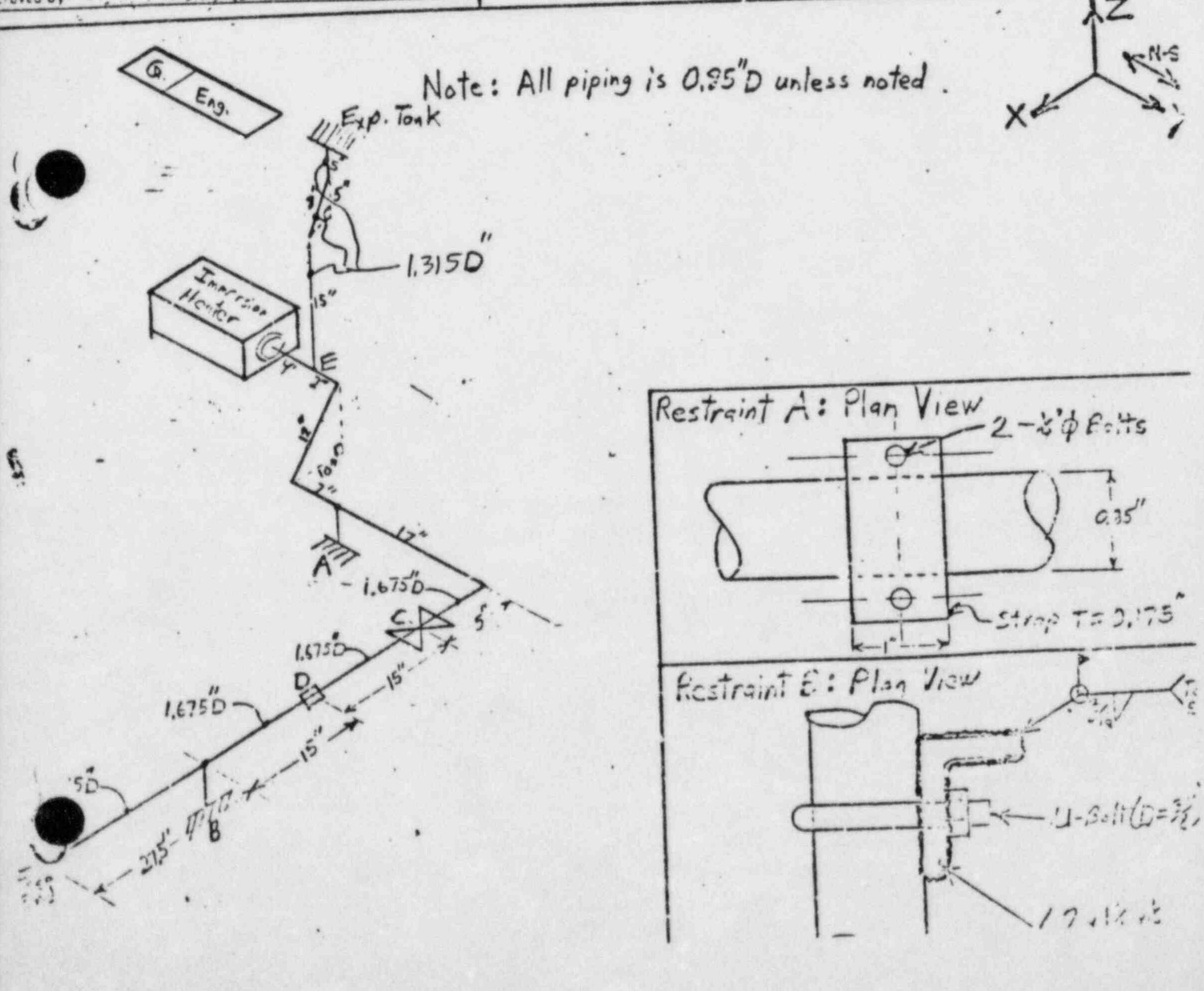
Concentrated Masses

pt.	Item	Est. wt. (lbs)
C	Shutoff Valve	4.5 lb
D	3/4" Coupling	Negligible

Pipe Materials

Segment	Mat'l.

Note: All piping is 0.85"D unless noted.



LaSalle EDGS Piping Qualification

"Critical" Pin for D Piping System Isometric

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.7.35

Corresp. Equip. No.: 0125C011; 1E22-S001
Corresp. Pipe Run(s): D
Corresp. Log. No.(s): 0307D4D102
0207D4D103

Corresp. Photo No.(s): 3A-14, 5E-13 ex.

Corresp. Spec.: J-2544, J-2500

Prepared by Bruce M. Long	Date 1-24-84
Reviewed by David Wright	Date 1-31-84
Approved by [illegible]	Date [illegible]

Parameters

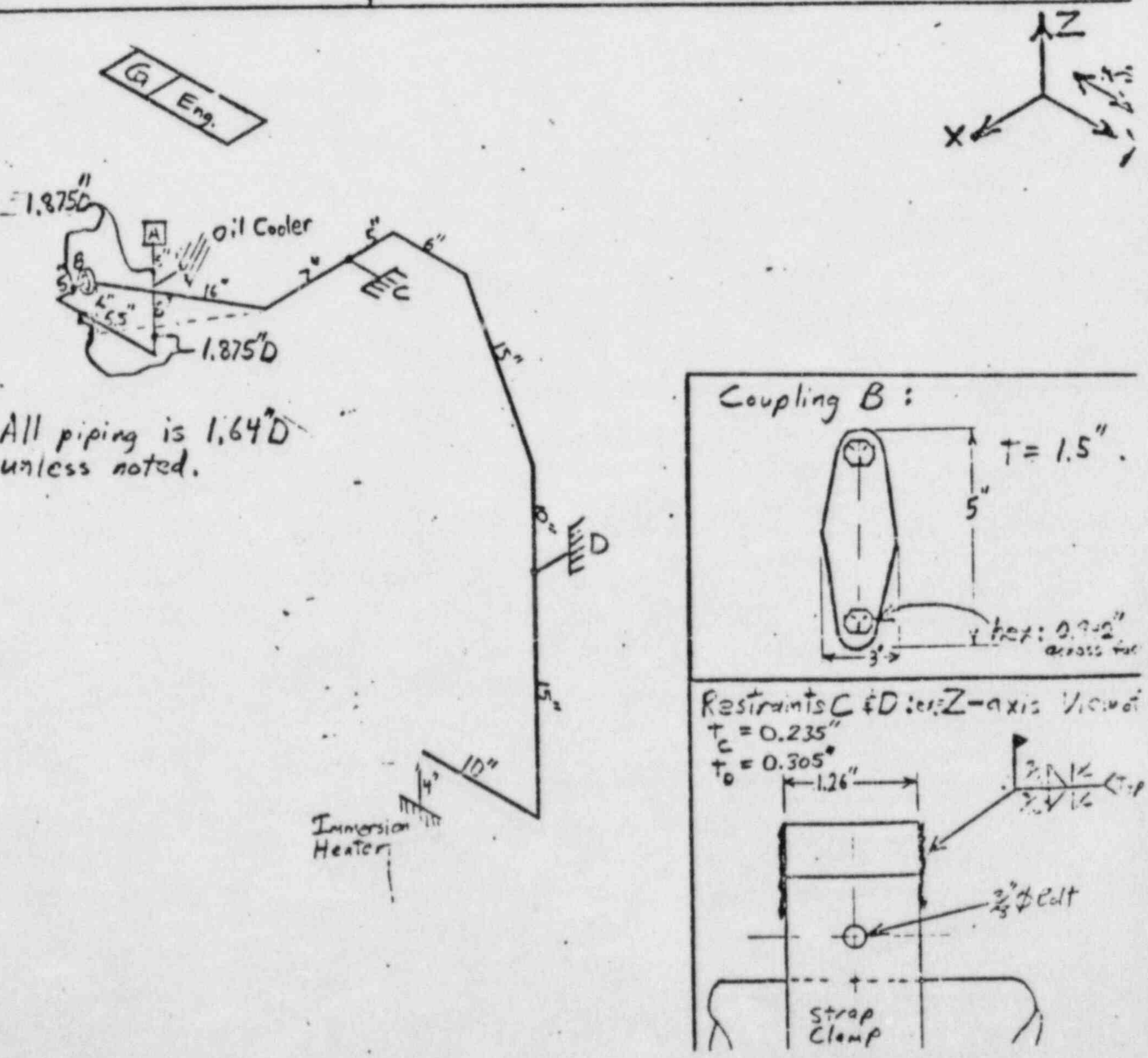
Op. Pressure: ________
Dsgn. " : ________
Op. Temp : ________

Concentrated Masses

pt.	Item	Est. Wt. (lbs)
A	Flow Switch	4
B	2-Bolt Coupling	

Pipe Materials

Segment	Mat'l.



Note: All piping is 1.64"D unless noted.

LaSalle EDGS Piping Qualification

"Critical" Pipe Run F Piping System Isometric

Equip. No. : 0,1 2 DG01K; 1,2E22-S001
Pipe Run(s): F
orresp. Log. No.(s): 23030'D110
orresp. Photo No.(s): 1A-14
rresp. Spec. : J-2544, J-2500

ared by Bruce M. Long Date 1-25-84
ewed by Dail Wright Date 1-31-84
roved by Date 1-31-84

Parameters

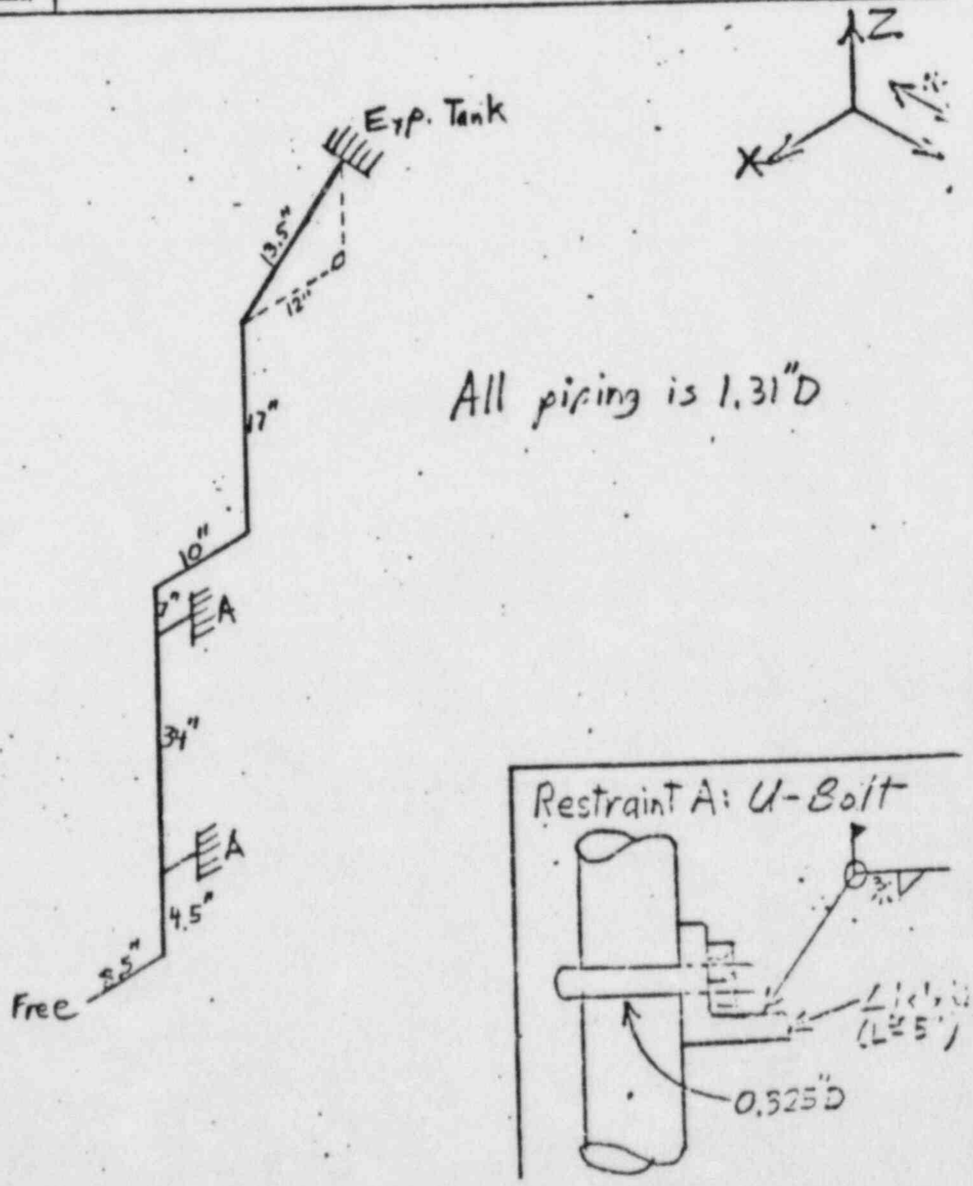
Op. Pressure :
Dsgn. " :
Op. Temp :

Concentrated Masses

pt.	Item	est. wt. (lbs)

Pipe Materials

Segment	Mat'l



All piping is 1.31"D

LaSalle EDGS Piping Qualification

"Critical" Exp Tank-[illegible] Piping System Isometric

CALC NO. CQD-014046
REV. 00 DATE 6/1/64
PROJ. NO. 6995-00
PAGE B.7.37

orr Equip. No.: 2[illegible]E30[illegible]E22-S001
. Pipe Run(s): 6
orresp. Log. No.(s): 0203042111
orresp. Photo No.(s): 2C-13 Gray.
orresp. Spec.: J-2540, J-2500

epared by Bruce M. [illegible] Date 1-24-84
viewed by David M. Wright Date 1-31-84
proved by [illegible] Date [illegible]

Parameters

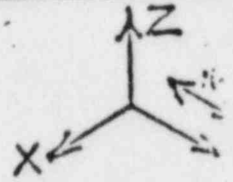
Op. Pressure: ____
Dsgn. " : ____
Op. Temp: ____

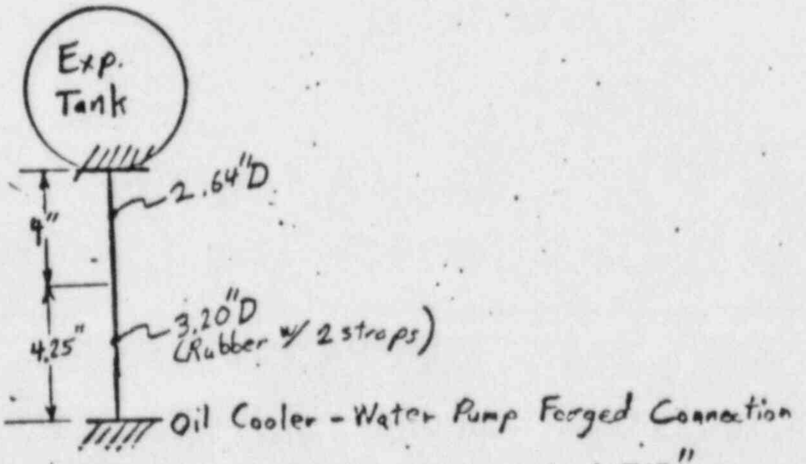
Concentrated Masses

Pt.	Item	Est. Wt. (lbs)

Pipe Materials

Segment	Mat'l





(Each Strap: W = 0.767"
t = 0.05"
1 Bolt - 3/8"ø)

Elevation View Along X-Axis

LaSalle EDG'S Piping Qualification

"Critical" Pipe K & L Piping System Isometric

Equip. No.: 0[illegible]1,2E22-S001
Pipe Run(s): K & L/J & L
orresp. Log. No.(s): 0303 060708
0303060710
0503060708
0503060710
orresp. Photo No.(s): 7C-6, 1A-9 [illegible]
orresp. Spec.: J-2514, J-2500

pared by Bruce M. [illegible]	Date 1-25-84
iewed by David Wright	Date 1-31-84
proved by R. H[illegible]	Date 1-31-84

Parameters

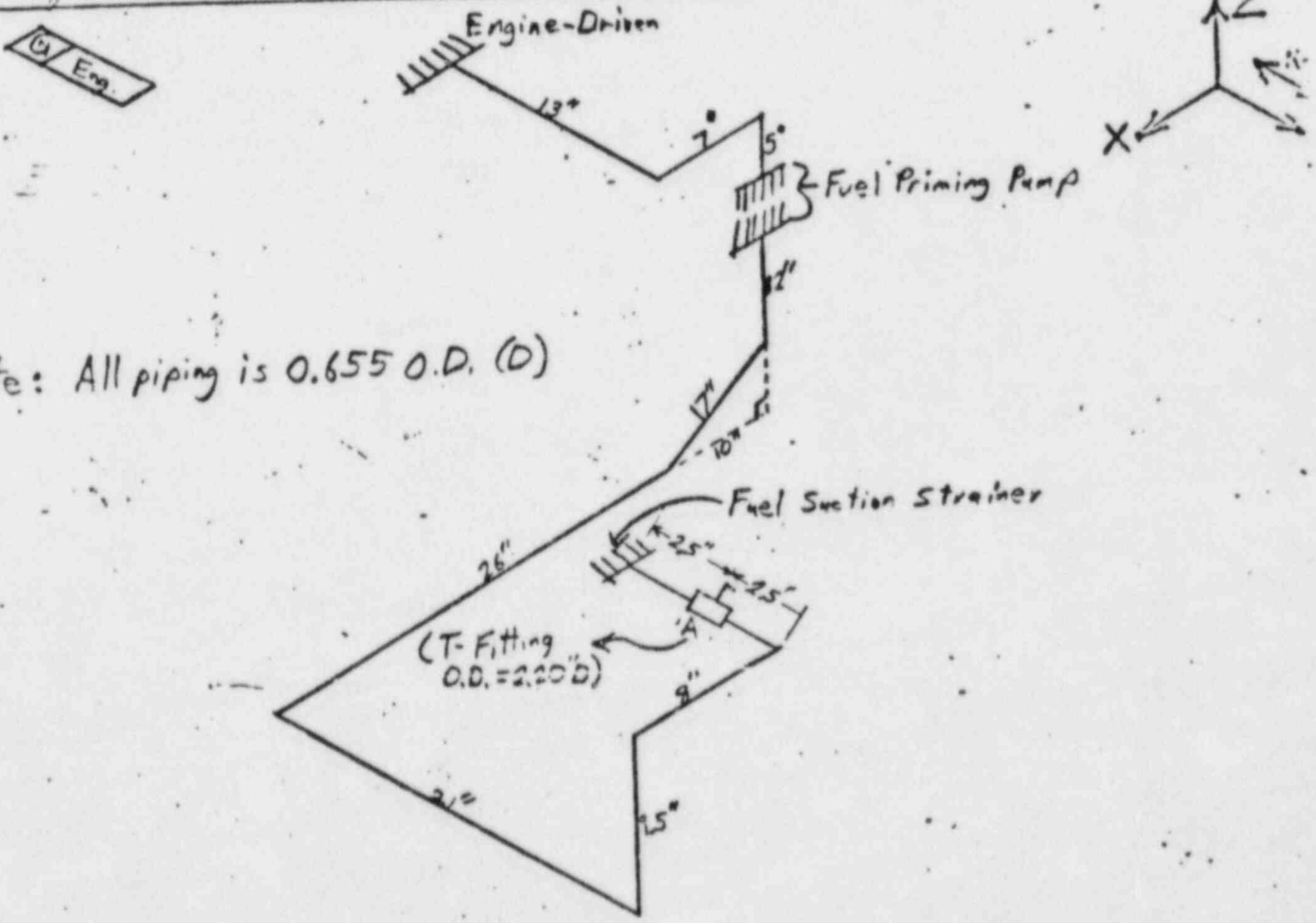
Op. Pressure :
Dsgn. " :
Op. Temp :

Concentrated Masses

pt.	Item	Est. Wt. (lbs)

Pipe Materials

Segment	Mat'l



Note: All piping is 0.655 O.D. (D)

LaSalle EDG&S Piping Qualification

"Critical" Pipe Run M&S Piping System Isometric

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.7.39

orresp. Equip. No. : 0,1,2 D/S01K ; 1,2 E22-S001
Pipe Run(s): M & N & S / U
orresp. Log. No.(s): 0303162201
0303162206
0303162203
0503162206

orresp. Photo No.(s): 4C-2, 9E-12 for ey. 3C-11

orresp. Spec. : J-2544, J-2500

pared by Bruce M. Fong	Date 1-25-84
viewed by David Wright	Date 1-31-84
proved by D. Hlayna	Date 1-31-84

Parameters

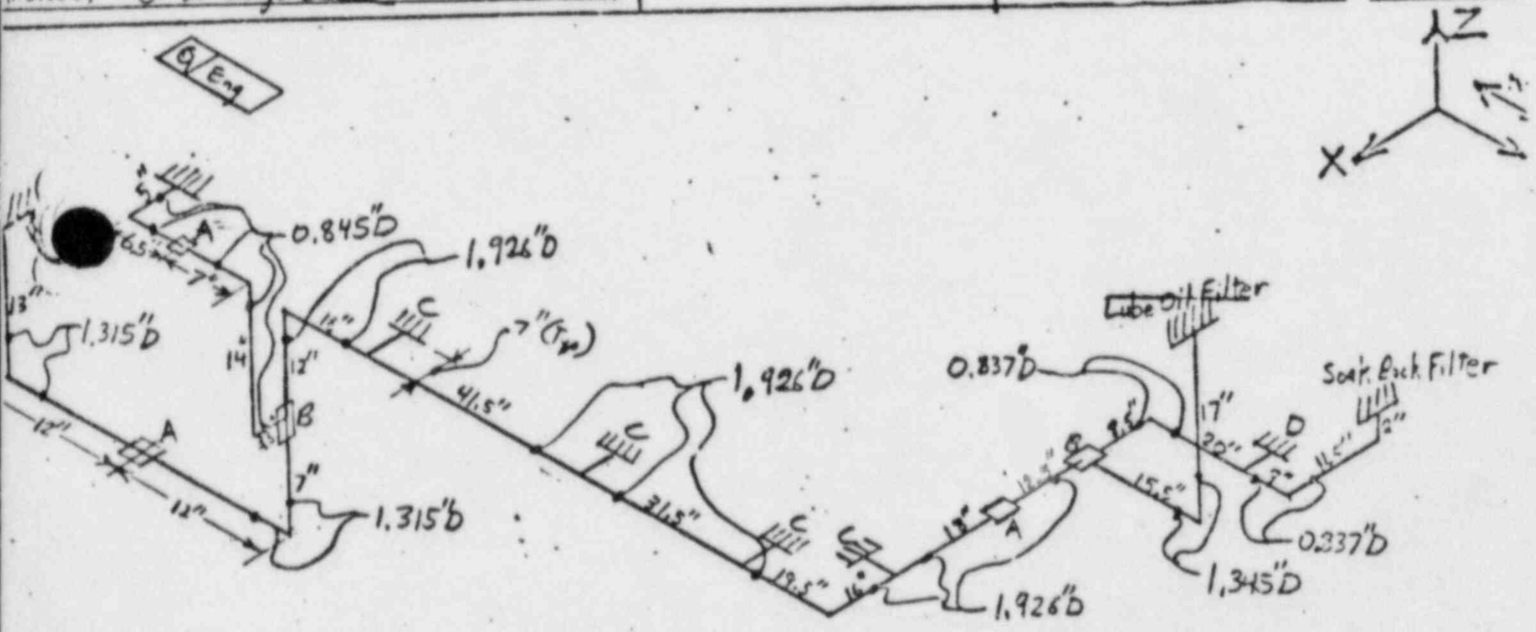
Op. Pressure : 125 psi max
Dsgn. " :
Op. Temp : 235°F

Concentrated Masses

pt.	Item	Est. Wt. (lbs)
A	strainers	
B	T-Branches	
C	Strap Clamps	
D	strap Clamp	

Pipe Materials

Segment	Mat'l



A: Y-Strainers - 3"L x 2.0" Hex Fit

B: T-Branches - 4½"L x 1.83"D

C.: Strap Clamps: W=1.25", t=0.[illegible]", L≅7
1 Bolt - 0.375"D, welded w/ 2-¼ [illegible]

D: Strap clamp: W=1.20", t=0.[illegible]", L≅7'
1 Bolt - 0.375"D, welded w/ 2-⅛ [illegible]

LaSalle EDGS Piping Qualification

"Critical" Pipe Run P Piping System Isometric

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.7.40

Equip. No.: 0,1,2DG01P; 1,2E22-S001
Pipe Run(s): P/R
rresp. Log. No.(s): 0303162205
0503162205

orresp. Photo No.(s): 3A-11

rrcsp. Spec.: J-2544, J-2500

ared by	Date 1-25-84
ewed by Dale Wright	Date 1-31-84
oved by	Date 1-31-84

Parameters

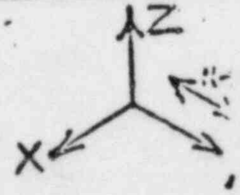
Op. Pressure: ______
Dsgn. " : ______
Op. Temp : ______

Concentrated Masses

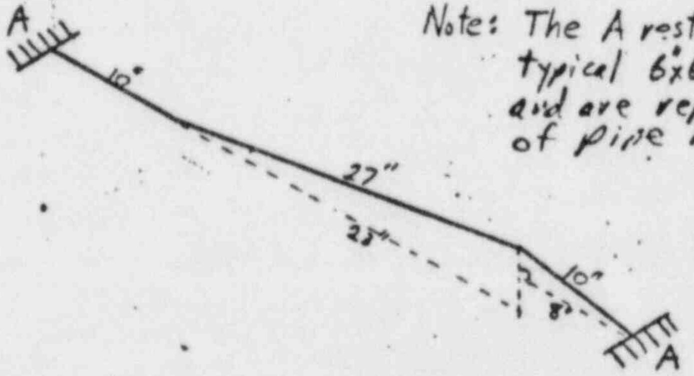
pt.	Item	Est. Wt. (lbs)

Pipe Materials

Segment	Mat'l



Eng / G



All piping is 3.50 D

Note: The A restraints are the typical 6"x6" flange connections and are replicate with Anchor D of pipe run Φ

LaSalle EDG:S Piping Qualification

"Critical" Pipe Run Q Piping System Isometric

Equip. No.: 0,1,2DG01K; 1,2E22-S001
p. Pipe Run(s): Q
Corresp. Log. No.(s): 03031522AG

orresp. Photo No.(s): 3C-15

orresp. Spec.: J-2544, J-2500

pared by Bruce M. Long	Date 1-25-84
viewed by David Wright	Date 1-31-84
proved by	Date 1-31-84

Parameters

Op. Pressure: ______
Dsgn. " : ______
Op. Temp : ______

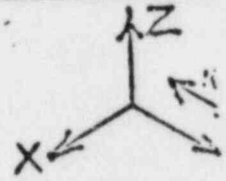
Concentrated Masses

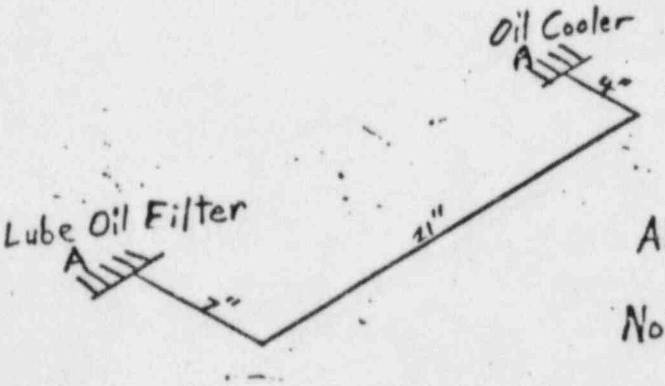
pt.	Item	Est. Wt. (lbs)

Pipe Materials

Segment	Matl.

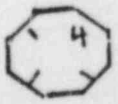
Eng G





All Piping is 3.482" O.D. (D)

Note: The A anchors are replicate to the D. anchor of pipe run Ø except bolt marking is:



LaSalle EDG:S Piping Qualification

"Critical" Pipe Run R Piping System Isometric

Cor Equip. No. : 0,1,2DG01K; 1,2E22-S001
C p. Pipe Run(s): R/T
Corresp. Log. No.(s): 0303162207
0503162207
Corresp. Photo No.(s): 1A-15, 2C-5
Corresp. Spec. : J-2544, J-2500

epared by Bruce F.H.	Date 1-25-84
eviewed by David W.	Date 1-31-84
pproved by D. Hl	Date 1-31-84

Parameters

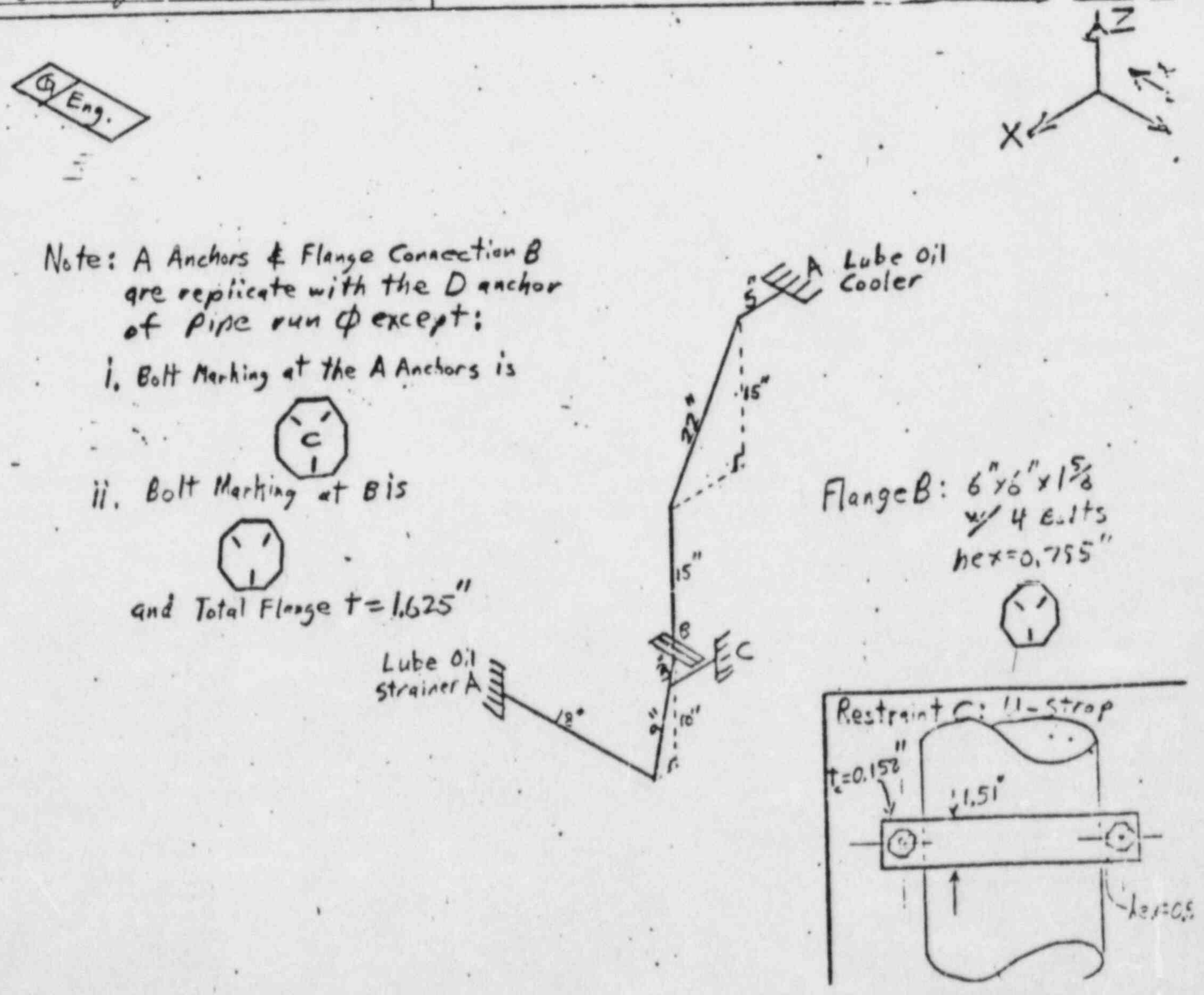
Op. Pressure :
Dsgn. " :
Op. Temp :

Concentrated Masses

pt.	Item	Est. Wt. (lbs)
B	Flange Connt.	16.6 lb

Pipe Materials

Segment	Mat.



Note: A Anchors & Flange Connection B are replicate with the D anchor of pipe run ϕ except:

i. Bolt Marking at the A Anchors is

ii. Bolt Marking at B is

and Total Flange t = 1.625"

Flange B: 6" x 6" x 1 5/8
w/ 4 Bolts
hex = 0.755"

Restraint C: U-strap
t = 0.158"
1.51"
hex = 0.5

```
                         FINAL RESPONSE SPECTRUM VALUES

BASE SSE 3%            GROUND       SDSS                NOR    N/A

IDENED SPECTRUM

AMPING FACTOR      .020          NUMBER OF SPECTRUM POINTS        59.

  PERIOD   ACCELERATION        PERIOD   ACCELERATION        PERIOD   ACCELERATION        PERIOD   ACCELERATION
  SECOND              G        SECOND              G        SECOND              G        SECOND              G
    .002         .19105          .002         .19875          .003         .20751          .005         .20751
    .005         .20845          .007         .20848          .008         .20598          .010         .23207
    .014         .22207          .015         .21555          .016         .22000          .024         .22000
    .025         .21556          .029         .22504          .025         .23504          .035         .23769
    .043         .23789          .045         .25385          .043         .25922          .050         .28241
    .054         .28841          .057         .29552          .059         .31085          .063         .31085
    .064         .33472          .074         .33472          .075         .35764          .080         .37116
    .084         .37116          .088         .40957          .102         .40857          .103         .43724
    .106         .48839          .115         .48839          .116         .50892          .137         .50852
    .139         .54751          .145         .50831          .216         .60531          .220         .58707
    .216         .55707          .227         .56247          .334         .58957          .360         .58957
    .264         .52040          .545         .52040          .571         .51250          .574         .55088
    .531         .58088          .857         .55766          .706         .54841          .750         .54714
    .793         .48274         1.000         .48274         1.075         .41181         1.333         .41181
   1.500         .40720         1.715         .34242         2.400         .26438
```

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                                         I N P U T     D A T A

   N O T E :   THE DATA PROVIDED EITHER IN ARRAY FORMS OR THROUGH REPEATED CARD GROUPS ARE NOT PRINTED OUT
               CONSULT  (R S G)  MANUAL FOR SYMBOLS.

   NSIG      =          1                    NT        =        80                  NBACR     =          0
   IBACR     =          0                    NPLTM     =         1                  ICYCLE    =          0
   IFFT      =          0                    IPLT      =         4                  NPUNCH    =          0
   INTERP    =          0                    NTHIS     =         1                  NOTH      =          0
   IENVLP    =          0                    NPLCON    =         0                  NSUB      =          0

   IUNIT     =          0                    IDRW      =         1                  NBTH      =          0
   NTYPE     =          0                    NHCYCL    =         1                  NPNCH     =          0
   TSCALE    =      .0000                    FAC       =     .2000                  NPLB      =          0
   ASCALE    =      .0000                    NCYCL     =         0                  NVEL      =          0
   NCASE     =          0                    BSTRT     =     .0000                  NDF       =          0

   NTITLE    =          0                    INCUV     =         0                  NFLOOR    =          0
   ISRS      =          7                    NTIME2    =         0                  B         =      .0000
   ISRSS     =          3                    NINTC     =         0                  ACCL      =      .0000
   NFILTR    =          0                    ALFA      =     .0000                  IPFD      =          0
   IPSDF     =          0                    LOOPTH    =         0                  MSIG      =          0

   NTROP     =          0                    NFILTD    =         0                  MPLTM     =          0
   MAXREC    =          0                    ISPECD    =         0                  ERROR     =      .0000
   NTT       =          0                    NN        =         0                  IVNT      =          0
   IFF       =          0                    NSPDF     =         0                  NTY       =          0
   NDAMP     =          1                    NSPMF     =         0                  IPACT     =          0

   NPER      =          0                    ISPLIT    =         0

@PMD.LEAP
```

3308

CALC NO. CQD-014046
REV. 00   DATE 6/1/84
PROJ. NO. 6995-00
PAGE B.7.44

```
13.4800     -.052440   13.4900     -.058250   13.5000     -.047060   13.5100     -.025150
13.5200     -.026600   13.5300     -.005500   13.5400     -.003860   13.5500      .015140
13.5600      .018480   13.5700      .018800   13.5800      .011400   13.5900      .021660
13.6000      .022560   13.6100      .013620   13.6200      .014220   13.6300      .003660
13.6400     -.002040   13.6500     -.016660   13.6600     -.021560   13.6700     -.035320
13.6800     -.050420   13.6900     -.024080   13.7000     -.026540   13.7100     -.021980
13.7200     -.029700   13.7300     -.0405"?   13.7400     -.052020   13.7500     -.041760
13.7600     -.020600   13.7700     -.014940   13.7800     -.024120   13.7900     -.031380
13.8000     -.035700   13.8100     -.030420   13.8200     -.003240   13.8300      .001380
13.8400      .009520   13.8500      .007120   13.8600      .008520   13.8700     -.003920
13.8800     -.018900   13.8900     -.024620   13.9000     -.033860   13.9100     -.045590
13.9200     -.036220   13.9300     -.033100   13.9400     -.018340   13.9500     -.014360
13.9600     -.005160   13.9700      .001100   13.9800      .005360   13.9900      .002280
14.0000     -.000120   14.0100     -.011420   14.0200     -.024660   14.0300     -.032640
14.0400     -.025100   14.0500     -.030780   14.0600     -.023460   14.0700     -.012700
14.0800     -.018420   14.0900     -.026600   14.1000     -.028040   14.1100     -.045220
14.1200     -.048880   14.1300     -.057860   14.1400     -.057740   14.1500     -.065060
14.1600     -.052520   14.1700     -.047080   14.1800     -.051140   14.1900     -.052800
14.2000     -.050980   14.2100     -.047320   14.2200     -.042340   14.2300     -.038620
14.2400     -.035540   14.2500     -.034080   14.2600     -.033340   14.2700     -.025680
14.2800     -.030000   14.2900     -.038300   14.3000     -.048300   14.3100     -.059960
14.3200     -.033100   14.3300     -.051080   14.3400     -.036440   14.3500     -.025960
14.3600     -.015740   14.3700     -.012600   14.3800     -.004760   14.3900      .002660
14.4000      .007660   14.4100      .013420   14.4200      .022300   14.4300      .031220
14.4400      .035980   14.4500      .038920   14.4600      .041640   14.4700      .042540
14.4800      .042560   14.4900      .034660   14.5000      .029600   14.5100      .028160
14.5200      .024720   14.5300      .019420   14.5400      .017040   14.5500      .011740
14.5600      .011780   14.5700      .008420   14.5800      .000920   14.5900     -.003700
14.6000     -.007060   14.6100     -.006760   14.6200     -.007560   14.6300     -.007640
14.6400     -.008500   14.6500     -.004520   14.6600     -.011340   14.6700     -.015640
14.6800     -.015540   14.6900     -.012980   14.7000     -.007600   14.7100     -.003240
14.7200     -.000800   14.7300     -.003700   14.7400     -.002080   14.7500      .000350
14.7600      .000720   14.7700      .001320   14.7800      .002300   14.7900      .001720
14.8000      .002220   14.8100      .000720   14.8200     -.002660   14.8300     -.002760
14.8400     -.003320   14.8500     -.002520   14.8600     -.000200   14.8700      .000780
14.8800      .003180   14.8900      .005480   14.9000      .006560   14.9100      .006680
14.9200      .005220   14.9300      .004140   14.9400      .002980   14.9500      .001490
14.9600      .000320   14.9700     -.000240   14.9800     -.000660   14.9900     -.001020
15.0000     -.001300   15.0100      .001120   15.0200      .001100   15.0300      .001080
```

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                         R E S P O N S E   S P E C T R U M   V A L U E S

BASE SSE 2%              GROUND       EDGS                    HOR    N/A



DAMPING FACTOR         .020     NUMBER OF SPECTRUM POINTS      80     MAX ACCN OF TIME HISTORY       .200000 G
  PERIOD   PSU.ABS.ACC.            PERIOD   PSU.ABS.ACC.            PERIOD   PSU.ABS.ACC.            PERIOD   PSU.ABS.ACC.
    SEC               G              SEC               G              SEC               G              SEC               G

   .0020          .10105            .0030          .18875            .0040          .20751            .0050          .19238
   .0060          .20588            .0080          .18879            .0100          .20530            .0120          .22207
   .0150          .19870            .0200          .22000            .0250          .18058            .0350          .23504
   .0400          .27803            .0450          .23769            .0500          .22855            .0600          .25385
   .0610          .25863            .0630          .28641            .0670          .26102            .0690          .28475
   .0710          .29862            .0740          .31089            .0770          .28311            .0800          .33472
   .0830          .30446            .0870          .32711            .0910          .32238            .0950          .35784
   .1000          .37116            .1050          .35792            .1110          .40967            .1200          .38974
   .1250          .39569            .1280          .43724            .1330          .46539            .1380          .43831
   .1430          .45161            .1480          .50892            .1540          .41984            .1600          .49431
   .1670          .45066            .1740          .54751            .1820          .50531            .1910          .51886
   .2000          .46577            .2080          .49351            .2170          .45797            .2380          .53843
   .2400          .53088            .2500          .52748            .2630          .56707            .2780          .55960
   .2900          .55177            .3030          .49168            .3170          .48375            .3330          .53427
   .3450          .51818            .3570          .50001            .3700          .52758            .3850          .49855
   .4000          .53927            .4170          .59957            .4350          .57892            .4550          .52040
   .5780          .51380            .5000          .51878            .5260          .58089            .5560          .55765
   .6880          .54841            .6250          .54714            .6670          .47700            .8100          .46049
   .7690          .42048            .8330          .48274            .9090          .40213           1.0000          .34867
  1.1110          .41181           1.2500          .40720           1.4290          .34242           2.0000          .28438
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SARGENT & LUNDY
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Piping Analysis Section:

There are three pipe systems which are unique to these Shoreham E.D.G.S.'s and consequently require analysis for conformance to ASME III criteria.

1st System - Cooling System Vent Piping (Log # 0803040109)

2nd System - Engine to Circ. Pump (Log # 0803162204)

3rd System - Engine Water Outlet to Radiator (Log # 0803040101)
Radiator to Lube Oil Cooler (Log # 0803040104)

This analysis assumes that these systems exhibit non-rigid body natural frequencies which for static analysis requires peak "g" value's x 1.5 in all three (X, Y & Z) orthogonal directions.

For additional conservatism this analysis will utilize seismic accelerations on the order of 2 x peak "g" values.

Note: The 1st system's horizontal seismic accelerations are taken as $a = K a_{ZPA}$ where K is an amplification factor derived through a conservative, realistic time history-based approach.

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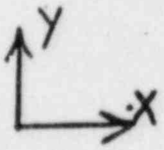
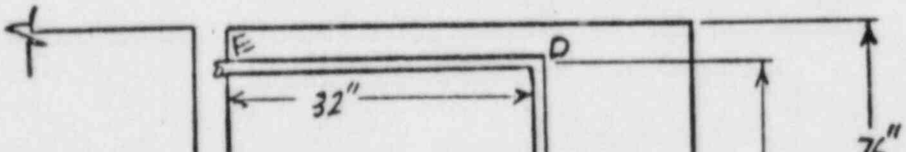
Cooling System Vent Piping : ($1''\phi$, sch.40)

From picture #s 29.-c., ;57 it is obvious that the longest unsupported vent piping span is the external pipe run above the radiator housing. A conservative pipe run (4 total) taken from pg. A10 of Appendix and from photos yields the following 3-D layout:



Note: The whole E.D.G.S. & shed are located on the soil. The horizontal response spectra experienced by the vent piping on the roof is not the peak "g" value but some amplitude of the ZPA. A conservative amplification



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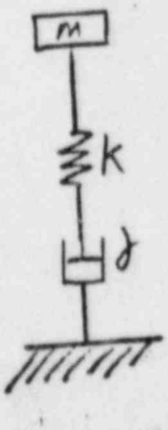
Max. Horizontal Seismic Acceleration on Vent Piping:

Because the E.D.G.S. is resting directly on the soil the vent piping located on top of the E.D.G.S. shed will experience amplification of the ground motion ZPA. The methodology of peak seismic "g" value x 1.5 being considered the bounding case is not necessarily the worst case condition here because soil behavior is indicative of ZPA since the soil stiffness is very, very large.

A conservative approximation of the maximum horizontal seismic acceleration experienced at the vent piping location would be derived by the following model. A single-degree-of-freedom model with viscous damping, γ, can be used.



The max. horizontal acceleration at the mass (vent piping) would be if the mass experienced the same resonance as the soil, let's say 30 Hz. Then if one also assumes this occurs at critical damping, the max. horizontal acceleration is:

$$a_{h_{max}} = a_{ground\ ZPA} \left(\frac{1}{2\gamma}\right)$$

$$\therefore \quad a_{h_{OBE}} = 0.10g\left(\frac{1}{2(0.01)}\right) = \underline{\underline{5.0g}}$$

$$a_{h_{SSE}} = 0.20g\left(\frac{1}{2(0.02)}\right) = \underline{\underline{5.0g}}$$

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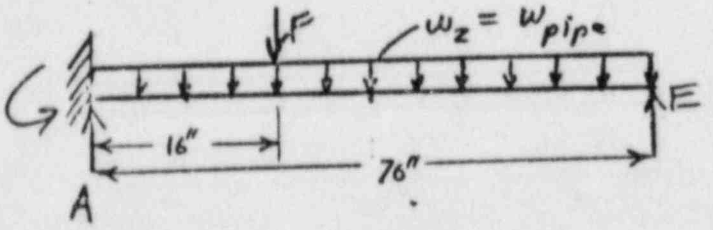
From picture #'s 2.a, &c, it is apparent that these 4 identical pipe runs are resting on the roof's inverted C-channel (pt. E). This is a high friction pt. and the horizontal seismic acceleration are adjusted when one considers that the pipe run's dead wt. $= 0.14[2(16)+76+32]$ $= 19.6$ lbs being supported vertically at its two ends and horizontally supported at one end (welded by brazing) and a friction support. This friction induces out-of-phase resonance between vibration modes. Consequently the system would not be at critical damping and the horizontal max. seismic acceleration is $5.0(0.33) = 1.65g$.

<u>Dead Wt. Piping Moment - M_A</u> (Z-Axis Projection & Loading)

Note: In the vertical load cases pt. E. is a simple support
and this



$$^{*}w_z = 1.68\ \text{lb}/\text{ft} = 0.14\ \text{lb}/\text{in}$$

$$F = 0.14(0.5(38)+38) = 7.98\ \text{lb}$$

$$M_{A_{max}} = 16F + \frac{w_z L^2}{8} = 16(7.98) + \frac{0.14(70)^2}{8} = \underline{\underline{213.4\ \text{in-lb}}}$$

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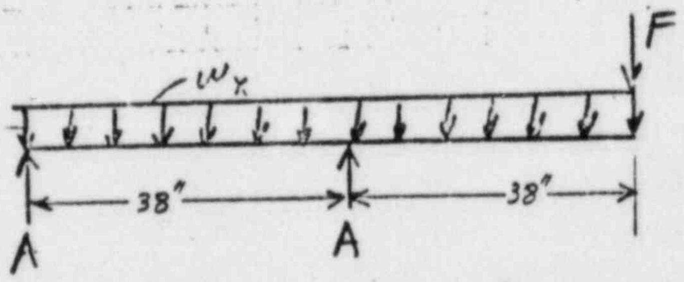
Seismic Moment M_B:

1. Z-Axis Projection - $a_{vs} = \frac{2}{3} a_{horiz.}$

$$M_{1_{max}} = \frac{2}{3}(0.62g)(2)(213.4) = 176.5 \text{ in-lb}$$

2. X-Axis Projection -

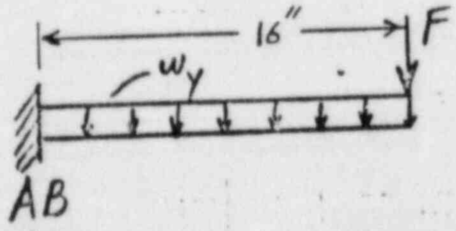


$w_x = 0.14(1.65g) = 0.23$ lb/in

$F = 0.23(32) = 7.36$ lb

$$M_{2_{max}} = 38F + \frac{w_x(38)^2}{2} = 38(7.36) + \frac{0.23(38)^2}{2} = 445.7 \text{ in-lb}$$

3. Y-Axis Projection :



or
AC

$w_y = w_x = 0.23$

$F = \frac{1}{2}(76 + 32)(0.23) = 12.42$ lb

$$M_3 = 12.42(16) + \frac{0.23(16)^2}{2} = 228.2 \text{ in-lb}$$

$$\therefore M_{B_{max}} = \sqrt{(176.5)^2 + (445.7)^2 + (228.2)^2} = 530.9 \text{ in-lb}$$

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Selected "Worst Case" Pipe Segment Data Summary for Log #0803040109

System : Cooling Vent Piping
Working Fluid : condensed vapor
From : Radiator
To : open atmosphere
Pipe O.D. : $D_o = 1.315$ in
Pipe Nominal Wall Thickness : $t_n = 0.133$ in (min. wall T. assumed)
Pipe Mean Radius : $r_m = \frac{1}{2}(D_o) - \frac{1}{2}(t_n) = 0.658 - 0.067 = 0.592$ in
Pipe Distributed Wt. : $w = 1.68\ \frac{lb}{ft}$
Pipe Section Modulus (per ASME) : $Z = \pi r_m^2 t_n = 3.14(0.592\,in)^2(0.133\,in) = 0.146\ in^3$
Max. Pressure : $P_{max} = 10$ psi
ASME Allowable Stress : $S_c = 15,000$ ksi (ref. #10 - Table I)
$S_h = 15.00$

Analysis Results -

Static Bending Moment : $M_A = 131.0$ in-lb
Seismic Resultant Bending Moment : $M_B = 530.9$ in-lb
Shape Factors :

$$h_{elbow} = \frac{t_n R_o}{r_m^2} = \frac{0,133\left(\frac{1,315}{2}\right)}{(0,592)^2} = 0,250$$

$$h_{T\text{-}Joint} = \frac{t_n}{r_m} = \frac{0,133}{0,592} = 0,225$$

Stress Intensification Factors : $i_{max} = \frac{0.9}{(h_{min})^{2/3}} = \frac{0.9}{(0.225)^{2/3}} = 2,44$

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Calculated Max. Pipe Stresses -

Pipe Material : A-105, grd. B
Allowable Stress : $S_c = 15,000$ psi
$S_h = 15,000$ psi

1st Condition : Sustained Loads -

$$S_L = \frac{PD_o}{4t_n} + 0.75(i_{max})\left(\frac{M_A}{Z}\right) \leq 1.0 S_h$$

$$S_L = \frac{10(1.315)}{4(0.133)} + 0.75(2.44)\left(\frac{131.0 \text{ in-lb}}{0.146 \text{ in}^3}\right) \leq 15,000 \text{ psi}$$

$$\underline{\underline{S_L = 1,666.7 \text{ psi} \leq 15,000 \text{ psi}}}$$ ✓ Adequate

2nd Condition : Sustained + Occasional Loads

$$S_{OL} = \frac{PD_o}{4t_n} + 0.75(i_{max})\left(\frac{M_A + M_B}{Z}\right) \leq 1.2 S_h$$

$$S_{OL} = \frac{10(1.315)}{4(0.133)} + 0.75(2.44)\left(\frac{131.0 + 530.9 \text{ in-lb}}{0.146 \text{ in}^3}\right) \leq 18,000 \text{ psi}$$

$$\underline{\underline{S_{OL} = 8,321 \text{ psi} \leq 18,000 \text{ psi}}}$$ ✓ Adequate

∴ Log #0803040109 Pipe Runs are Adequate as designed to withstand the maximum postulated seismic loads due to an OBE (1%) or SSE (2%) event without loss of pressure or structural integrity.

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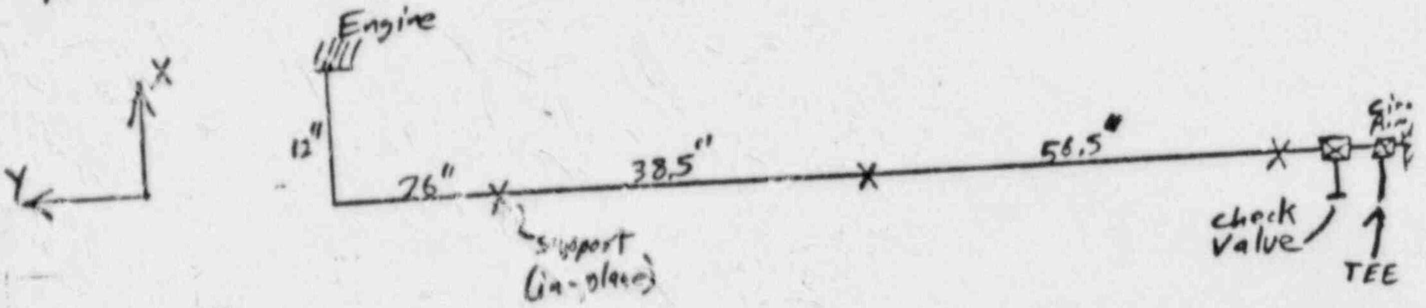
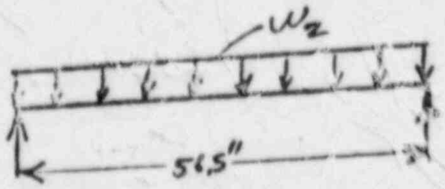
Engine to Fire Pump Piping: (1½"φ) Log#0803162204

The precise measurement of this pipe run (3-D space run) is obtained from photo #'s 39-42 of ref. #6.



Max. Dead Wt Moment (MA): Z-Axis Projection

$w = {}^{*}(2.72 + 0.882)\frac{1}{12} = 0.30\ ^{lb}/_{in}$



$M_{A_{max}} = \frac{wL^2}{2} = \frac{0.30(56.5)^2}{2} = \underline{\underline{478.8 \text{in-lb}}}$

Max. Seismic Moment (MB): Z-A

1. Z-Axis Projection

$M_1 = \frac{3}{3}(0.62g)(2)(478.8) = 396.0 \text{in-lb}$

(#)

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2. X-Axis Projection -

$M_2 = 2(0.620)(478.8) = 593.7$ in-lb

3. Y-Axis Projection -

$M_3 = M_2 = 593.7$ in-lb

$\therefore M_B = \sqrt{(396.0)^2 + (593.7)^2(2)} = \underline{\underline{928.3 \text{ in-lb}}}$

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Selected "Worst Case" Pipe Segment Data Summary for Log # 0803/62204

System : Lube Oil
Working Fluid : Lubricating Oil
From : Engine
To : Circ. Pump
Pipe O.D. : $D_o = 1.900 in$
Pipe Nominal Wall Thickness : $t_n = 0.145 in$
Pipe Mean Radius : $r_m = \frac{1}{2}(D_o) - \frac{1}{2}(t_n) = \frac{1}{2}(1.90) - \frac{1}{2}(0.145) = 0.878 in$
Pipe Distributed Wt. : $w = 0.30 \, {}^{lb}/_{in}$
Pipe Section Modulus (per ASME) : $Z = \pi r_m^2 t_n = 3.14(0.878 in)^2(0.145 in) = 0.351 in^3$
Max. Pressure : $P_{max} = 50 psi$
ASME Allowable Stress : $S_c = 15{,}000 ksi$ (ref. # 10 - Table I)

Analysis Results -

Static Bending Moment : $M_A = 478.8$ in-lb
Seismic Resultant Bending Moment : $M_B = 928.3$ in-lb
Shape Factors :

$$h_{elbow} = \frac{t_n R_o}{r_m^2} = 0.179$$

$$h_{T\text{-}Joint} = \frac{t_n}{r_m} = 0.165$$

Stress Intensification Factors : $i_{max} = \frac{0.9}{(h_{min})^{2/3}} = \frac{0.9}{(0.165)^{2/3}} = 2.99$

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Calculated Max. Pipe Stresses –

Pipe Material : A-106, grd. B
Allowable Stress : $S_c = 15,000$ psi
$S_h = 15,000$ psi

1st Condition : Sustained Loads –

$$S_L = \frac{PD_o}{4t_n} + 0.75(i_{max})\left(\frac{M_A}{Z}\right) \leq 1.0\,S_h$$

$$S_L = \frac{50\,(1.90)}{4(0.145)} + 0.75(2.99)\left(\frac{478.8 \text{ in-lb}}{0.351 \text{ in}^3}\right) \leq 15,000 \text{ psi}$$

$$\underline{\underline{S_L = 3,222.8 \text{ psi} \leq 15,000 \text{ psi}}}$$ ✓ Adequate

2nd Condition : Sustained + Occasional Loads

$$S_{OL} = \frac{PD_o}{4t_n} + 0.75(i_{max})\left(\frac{M_A + M_B}{Z}\right) \leq 1.2\,S_h$$

$$S_{OL} = \frac{50\,(1.90)}{4(0.145)} + 0.75(2.99)\left(\frac{478.8 + 928.3 \text{ in-lb}}{0.351 \text{ in}^3}\right) \leq 18,000 \text{ psi}$$

$$\underline{\underline{S_{OL} = 9,154 \text{ psi} \leq 18,000 \text{ psi}}}$$ ✓ Adequate

∴ The piping from the engine to the oil circ. pump is adequate as designed and is capable of withstanding the maximum combined effect of loading without loss of structural or pressure integrity.

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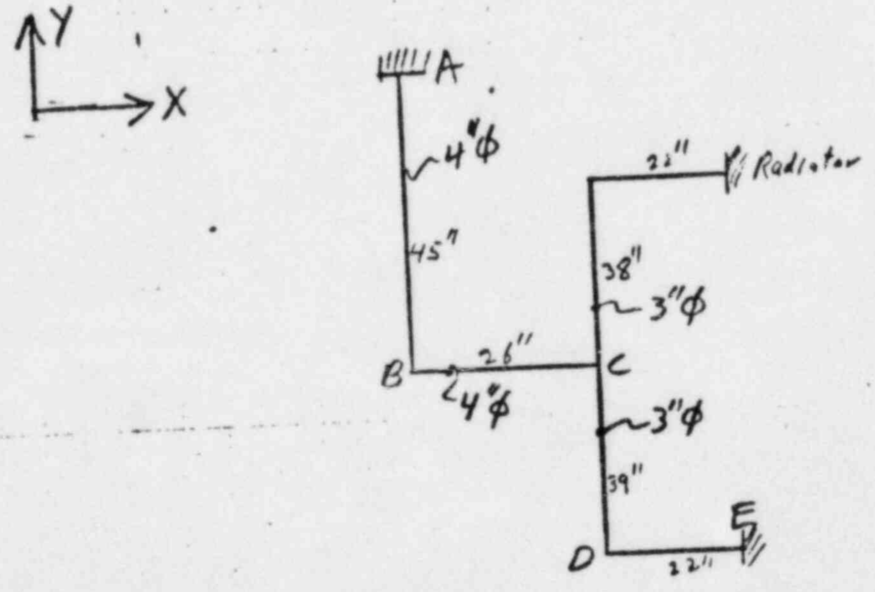
Engine to Radiator Cooling Piping & Radiator to Lube Oil Cooler Piping:

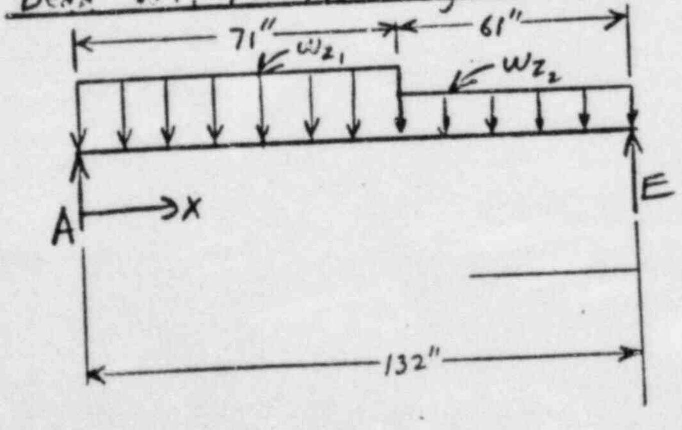
Log#'s 0803040101 & 0803040104 respectively

Two pipe spans present the maximum pipe bending moment locations, one of them also envolves 3"φ & 4"φ pipe joined together. Note that these pipe runs are in 2-D space.

Pipe Run I - Log# 0803040101 (see pgs A11-A14, they are from ref #6)



Dead Wt. Max. Bending Moment : Z-Axis Projection



$w_{z_1} = w_{4"\phi} = 10.79 + 5.51 = 16.31 \text{lb/ft}$

$w_{z_2} = w_{3"\phi} = 7.58 + 3.20 = 10.78 \text{lb/ft}$

$L = 132$

$a = 71$

$b = 61$

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Using case 6 on pg. 2-115 of ref #8 :

Bending moment M_A in 4"φ Pipe Region - $M_A = R_A(x) - \frac{w_{z_1} x^2}{2}$

where $R_A = \frac{w_{z_1} a(2L-a) + w_{z_2} c^2}{2L} = \frac{16.3(1/12)(71)(2(132)-71) + 10.78(1/12)(61)^2}{2(132)}$

$R_A = 83.17$ lb

$M_{A(x)}$ is max. when $M_A' = 0$: $R_A - \frac{2 w_{z_1} x}{2} = 0$

$R_A - w_{z_1} x = 0$

$R_A = w_{z_1} x$

$x = \frac{R_A}{w_{z_1}} = \frac{83.17}{16.3(1/12)} = \underline{\underline{61.23''}}$

$\therefore M_{A_{max}} = R_A(61.23) - \frac{w_{z_1}(61.23)^2}{2}$

$= 83.17(61.23) - \frac{(16.3(1/12)(61.23)^2}{2} = \underline{\underline{2,545.2 \text{ in-lb}}}$

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Calcs. For

☒ Safety-Related ☐ Non-Safety-

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Project
Proj. No. 6995-00 Equip. No.

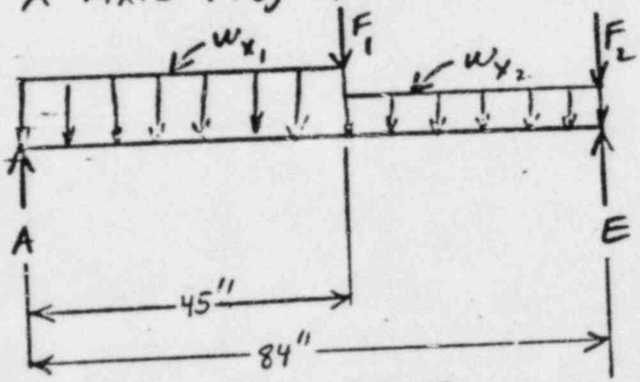
Prepared by Date
Reviewed by Date
Approved by Date

Max. Seismic Moment M_B :

1. Z-Axis Projection -

$$M_1 = \tfrac{2}{3}(0.62g)(2)(2{,}546.2) = 2{,}105.9 \text{ in-lb}$$

2. X-Axis Projection -



L = 84 a = 45 c = 39

$$w_{x_1} = 16.3(1/12)(0.62g)(2) = 1.68 \text{ lb/in}$$

$$w_{x_2} = 10.78(1/12)(0.62g)(2) = 1.11 \text{ lb/in}$$

$$F_1 = 26(1.68) = 43.7 \text{ lb}$$

$$F_2 = 22(1.11) = 24.4 \text{ lb}$$

$$M_{max_w} \text{ at } x = \frac{R_A}{w_{x_1}}, \quad R_A = \frac{1.68(45)(2(84)-45) + 1.11(39)^2}{2(84)}$$

$$R_A = \underline{\underline{65.4 \text{ lb}}}$$

$$\therefore x = \frac{65.4}{1.68} = \underline{\underline{38.93''}}$$

$$\therefore M_{max_w} = 65.4(38.93) - \frac{1.68(38.93)^2}{2} = 1{,}273 \text{ in-lb}$$

$$M_{max_{F_1}} = \frac{F_1 a c}{L} = \frac{43.7(45)(39)}{84} = 913 \text{ in-lb}$$

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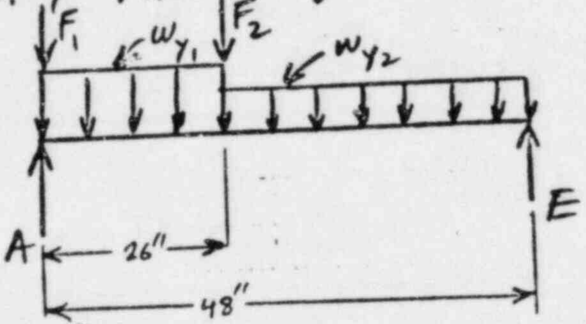
$\therefore M_{2_{max}}$ (regardless of location along beam) $= 1,273 + 913 = 2,186$ in-lb

3. Y-Axis Projection -



$w_{Y_1} = w_{X_1} = 1.68$ lb/in

$w_{Y_2} = w_{X_2} = 1.11$ lb/in

$F_2 = 39(1.11) = 43.3$ lb

$L = 48 \quad a = 26 \quad c = 22$

$$R_A = \frac{1.68(26)(2(48)-26) + 1.11(22)^2}{2(48)} = 37.5 \text{ lb}$$

$M_{3_{max_w}}$ occurs at $x = \frac{37.5}{1.68} = \underline{\underline{22.3''}}$

$$M_{3_{max_w}} = 37.5(22.30) - \frac{1.68(22.30)^2}{2} = 418.5 \text{ in-lb}$$

$$M_{3_{max}} = 418.5 + \frac{37.5(26)(22)}{48} = 865.4 \text{ in-lb}$$

$$\therefore M_B = \sqrt{(2,105.9)^2 + (2,186)^2 + (865.4)^2} = \underline{3,156 \text{ in-lb}}$$

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Selected "Worst Case" Pipe Segment Data Summary for Log #0803040101 ¢

System : Radiator - Engine Cooling
Working Fluid : Water
From : Engine / Radiator
To : Radiator / Lube Oil Cooler
Pipe O.D. : $D_o = 4.50$ in
Pipe Nominal Wall Thickness : $t_n = 0.237$ in
Pipe Mean Radius : $r_m = \frac{1}{2}(D_o) - \frac{1}{2}(t_n) = \frac{1}{2}(4.50) - \frac{1}{2}(0.237) = 2.132$ in
Pipe Distributed Wt. : $w =$
Pipe Section Modulus (per ASME) : $Z = \pi r_m^2 t_n = 3.14(2.132 in)^2(0.237 in) = 3.383 in^3$
Max. Pressure : $P_{max} = 70$ psi ← Max. Value on High P side of Water Pumps
ASME Allowable Stress : $S_c = 15{,}000$ ksi (ref. # 10 - Table I7.1)

Analysis Results -

Static Bending Moment : $M_A = 2{,}546.2$ in-lb
Seismic Resultant Bending Moment : $M_B = 3{,}156$ in-lb
Shape Factors :

$$h_{elbow} = \frac{t_n R_o}{r_m^2} = \frac{0.237(2.25)}{(2.132)^2} = 0.117$$

$$h_{T\text{-}Joint} = \frac{t_n}{r_m} = \frac{0.237}{2.132} = 0.111$$

Stress Intensification Factors : $i_{max} = \frac{0.9}{(h_{min})^{2/3}} = \frac{0.9}{(0.111)^{2/3}} = 3.90$

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Calculated Max. Pipe Stresses -

Pipe Material : A-106, grd B
Allowable Stress : $S_c = 15{,}000$ psi
$S_h = 15{,}000$ psi

1st Condition : Sustained Loads -

$$S_L = \frac{PD_o}{4t_n} + 0.75(i_{max})\left(\frac{M_A}{Z}\right) \leq 1.0\,S_h$$

$$S_L = \frac{70(4.5)}{4(0.237)} + 0.75(3.90)\left(\frac{2{,}546.2\text{ in-lb}}{3.383\,in^3}\right) \leq 15{,}000\,psi$$

$$\underline{\underline{S_L = 2{,}534\ psi \leq 15{,}000\ psi}}$$ ✓ Adequate

2nd Condition : Sustained + Occasional Loads

$$S_{OL} = \frac{PD_o}{4t_n} + 0.75(i_{max})\left(\frac{M_A + M_B}{Z}\right) \leq 1.2\,S_h$$

$$S_{OL} = \frac{70(4.5)}{4(0.237)} + 0.75(3.90)\left(\frac{2{,}546.2 + 3{,}156\text{ in-lb}}{3{,}383\ in^3}\right) \leq 18{,}000\,psi$$

$$\underline{\underline{S_{OL} = 5{,}263\ psi \leq 18{,}000\ psi}}$$ ✓ Adequate

∴ The 4"φ piping under this Log# is adequate to withstand the maximum seismic loads due to OBE (1%) or SSE (2%) event without loss of pressure or structural integrity.

Form GQ-3.03.1 Rev. 2

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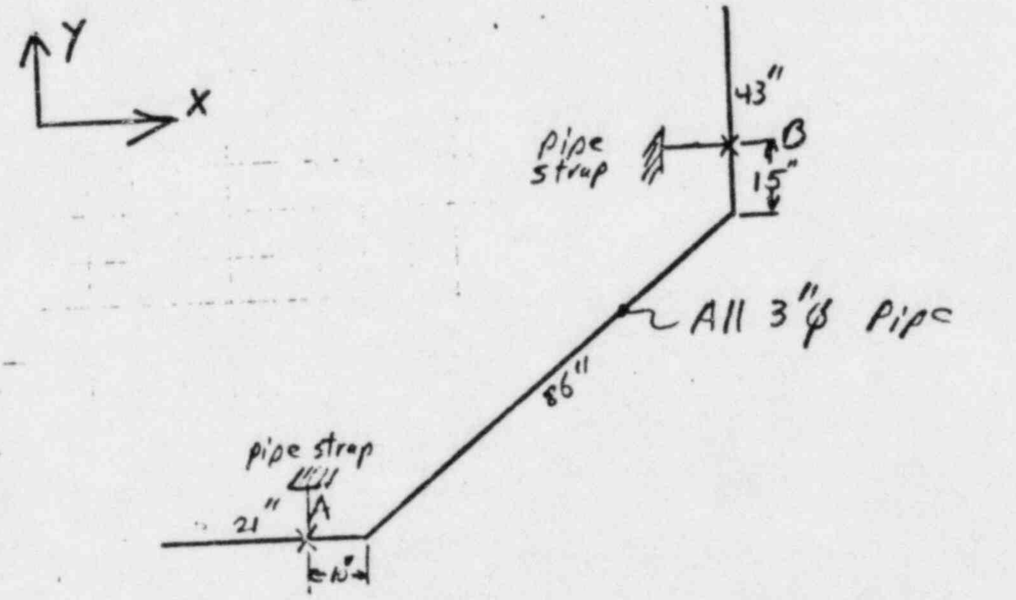
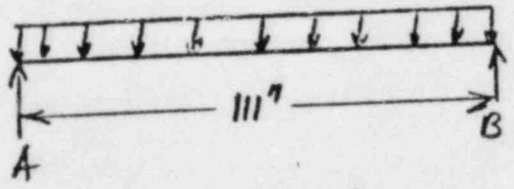
Pipe Run II - Log# 0803040101

Dead wt. Max. Bending Moment : Z-Axis Projection





$w_z = 10.78(1/12) = 0.90$ lb/in

$$M_{A_{MAX}} = \frac{w_z L^2}{2} = \frac{0.90(111)^2}{2} = \underline{\underline{5,544.5 \text{ in-lb}}}$$

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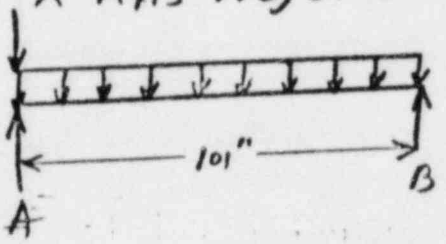
Seismic Max. Bending Moment:

1. Z-Axis Projection –

Factor reduced to 1.5 from 2.0 since 2.0 is too conservative

$$M_1 = \tfrac{2}{3}(0.62g)(1.5)(5,544.5) = 3,439 \text{ in-lb}$$

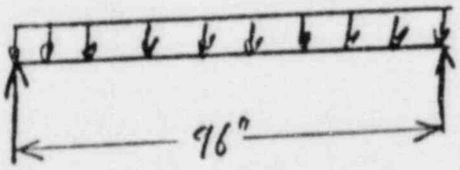
2. X-Axis Projection –



$$w_x = 0.90(1.5)(0.62) = 0.84 \text{ lb/in}$$

$$M_{2_{max}} = \frac{0.84(101)^2}{2} = 4,284 \text{ in-lb}$$

3. Y-Axis Projection –



$$M_{3_{max}} = \frac{0.84(96)^2}{2} = 3,871 \text{ in-lb}$$

$$M_B = \sqrt{(3,439)^2 + (4,284)^2 + (3,871)^2} \qquad \underline{6,720 \text{ in-lb}}$$

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Selected "Worst Case" Pipe Segment Data Summary for Log #0803040101

System : Cooling
Working Fluid : Water
From : Radiator
To : Oil Cooler
Pipe O.D. : $D_o = 3.50$ in
Pipe Nominal Wall Thickness : $t_n = 0.226$ in
Pipe Mean Radius : $r_m = \frac{1}{2}(D_o) - \frac{1}{2}(t_n) =$ $= 1.637$ in
Pipe Distributed Wt. : $w = 0.83$ lb/in
Pipe Section Modulus (per ASME) : $Z = \pi r_m^2 t_n = 3.14(\quad \text{in})^2(\quad \text{in}) = 1.903 \text{ in}^3$
Max. Pressure : $P_{max} = 70$ psi
ASME Allowable Stress : $S_c = 15.00$ ksi (ref. #10 - Table I)

Analysis Results -

Static Bending Moment : $M_A = 5,544.5$ in-lb
Seismic Resultant Bending Moment : $M_B = 6,720$ in-lb
Shape Factors :

$$h_{elbow} = \frac{t_n R_o}{r_m^2} = \frac{0.226(1.75)}{(1.837)^2} = 0.148$$

$$h_{T\text{-}Joint} = \frac{t_n}{r_m} = \frac{0.226}{1.637} = 0.138$$

Stress Intensification Factors : $i_{max} = \frac{0.9}{(h_{min})^{2/3}} = \frac{0.9}{(0.138)^{2/3}} = 3.37$

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Calculated Max. Pipe Stresses -

Pipe Material : A-106, grd. B
Allowable Stress : $S_c = 15,000$ psi
$S_h = 15,000$ psi

1^{st} Condition : Sustained Loads -

$$S_L = \frac{PD_o}{4t_n} + 0.75(i_{max})\left(\frac{M_A}{Z}\right) \leq 1.0 S_h$$

$$S_L = \frac{70(3.50)}{4(0.226)} + 0.75(3.37)\left(\frac{5,544.5 \text{ in-lb}}{1.903 \text{ in}^3}\right) \leq 15,000 \text{ psi}$$

$S_1 = 7,835.0$ psi ≤ 15,000 psi ✓ Adequate

2^{nd} Condition : Sustained + Occasional Loads

$$S_{OL} = \frac{PD_o}{4t_n} + 0.75(i_{max})\left(\frac{M_A + M_B}{Z}\right) \leq 1.2 S_h$$

$$S_{OL} = \frac{70(3.50)}{4(0.226)} + 0.75(3.37)\left(\frac{5,544.5 + 6,720 \text{ in-lb}}{1.903 \text{ in}^3}\right) \leq 18,000 \text{ psi}$$

$S_1 = 16,560$ psi ≤ 10,000 psi ✓ Adequate

∴ Piping under Log # 0803040101 is adequate as designed.

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Conclusions:

1. All E.D.G.S. on-skid piping runs have been examined for seismic adequacy based upon the ASME III, (Div. 1), Subsection ND criteria and evaluated via similarity with the LaSalle E.D.G.S. on-skid piping runs. Based upon this evaluation all of the Shoreham on-skid E.D.G.S. pipe runs are adequate as designed for Normal, Upset & Emergency plant operating conditions except the following pipe runs which were unique and consequently stress analyzed per ASME:

    1. Log # 0803040109
    2. Log # 0803162204
    3. Log # 0803040101
    4. Log # 0803040104

2. The piping runs (4) listed in 1. were stress analyzed using a very conservative method and checked for adequacy to the ASME code. All of these piping runs are adequate as designed for Normal, Upset & Emergency plant operating conditions.

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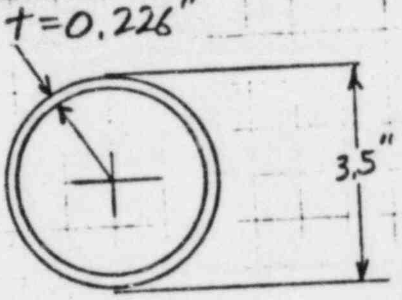
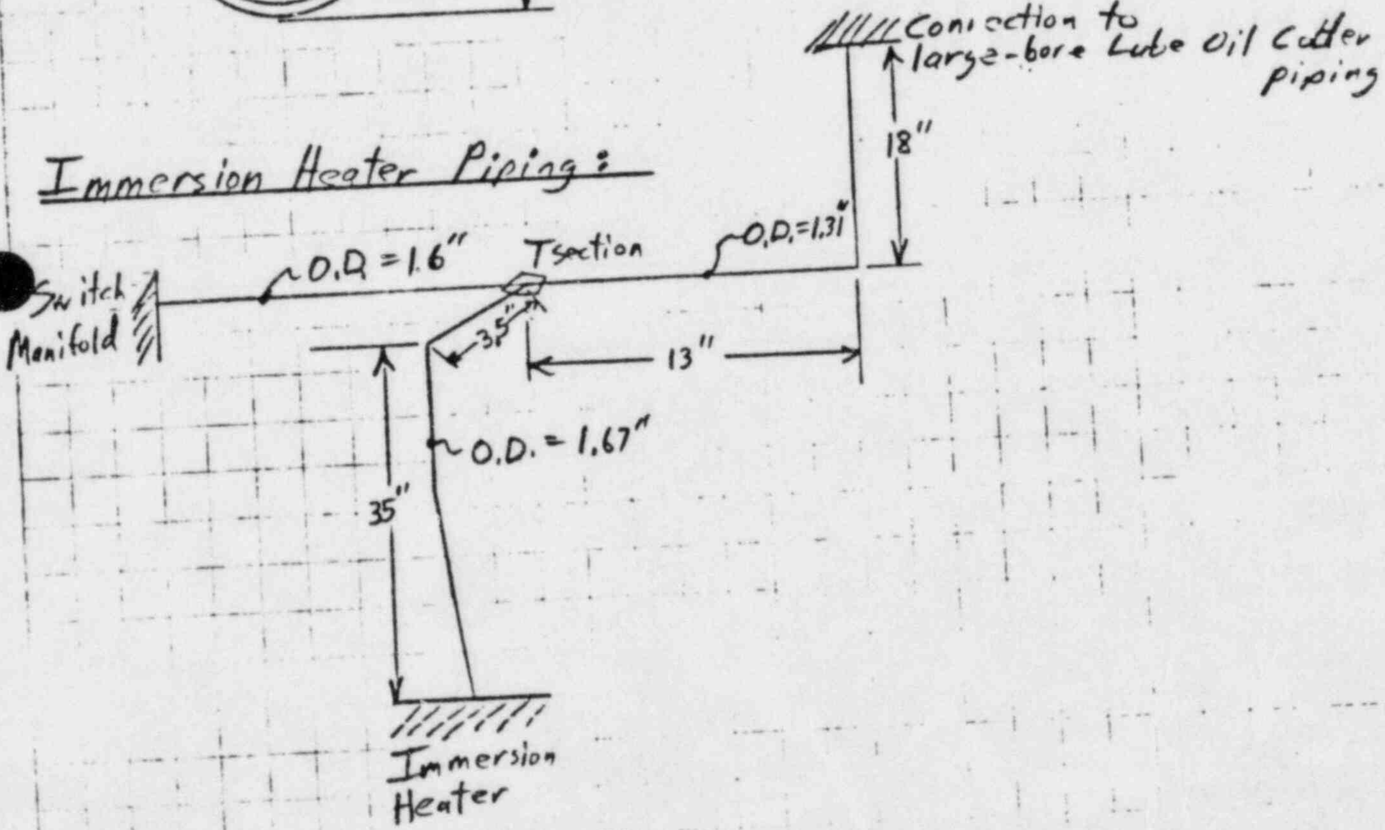
Radiator Inlet Water Piping:



$L_{max} = 4'$

Immersion Heater Piping:



Field sketch obtained during walkdown.

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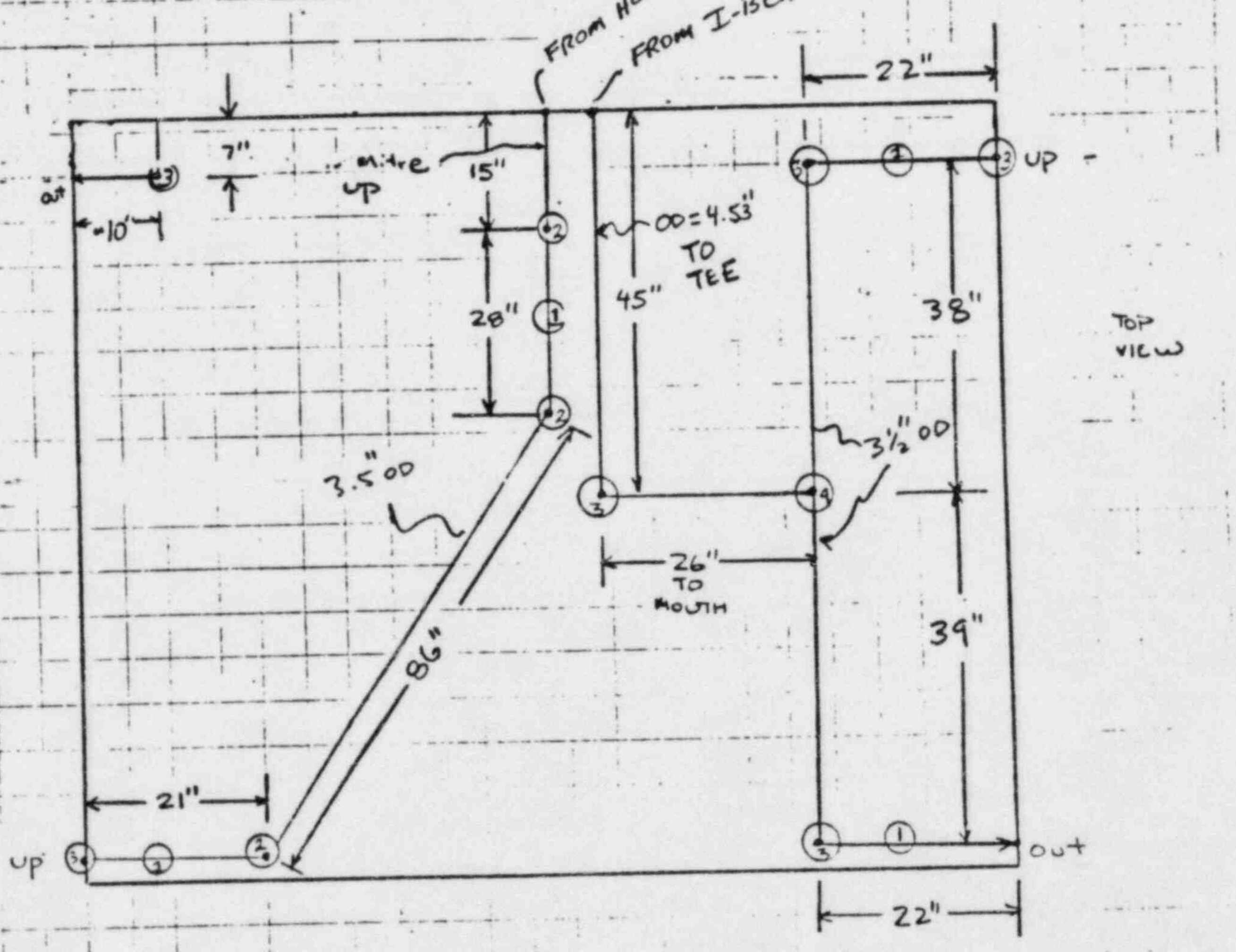
RADIATOR HOUSING PIPING



KEY
① AEROQUIP COUPLING
② 45° ELBOW
③ 90° ELBOW
④ TEE

Sketch Obtained during Field Walkdown

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B.8

B.8 ENGINE ENCLOSURE

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The enclosure of the Diesel Generator is analyzed as a frame. The radiator constitutes a horizontal beam with very high stiffness. The stiffeners located every 24" with the 3/8" plate considered as the vertical members of the frame structure. Among the three subdivisions of the enclosure, the area with the radiators are considered the most critical one. Therefore only this section of the enclosure is analyzed in details. The Eigenfrequency analysis show that the frame is very flexible. Therefore 1.5 times the maximum ordinates of the response spectra are employed to analyze the seismic loads. The wind loads are taken from the booklet issued by American National Standard with the title "Minimum Design Loads for Buildings and Other Structures.

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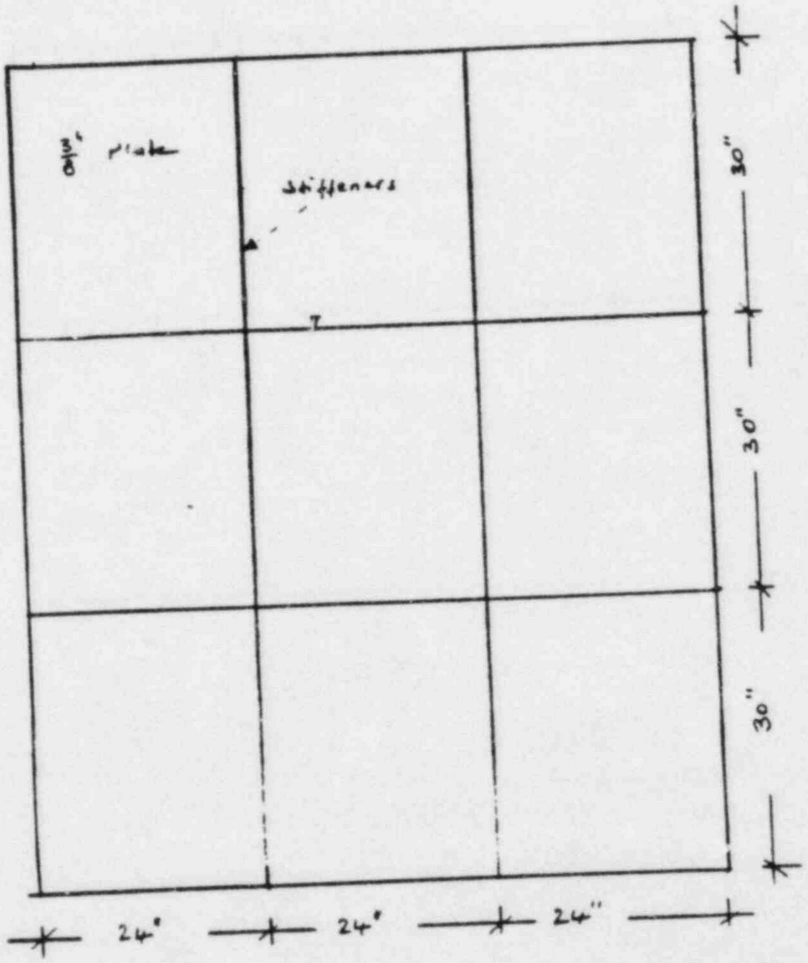
Distribution of Wall-Stiffeners:



Reference I: " Formulas for Natural Frequency and Mode Shapes"

by Robert D. Blevins

Reference 2: Betonkalender 1970, pages 501 & 502

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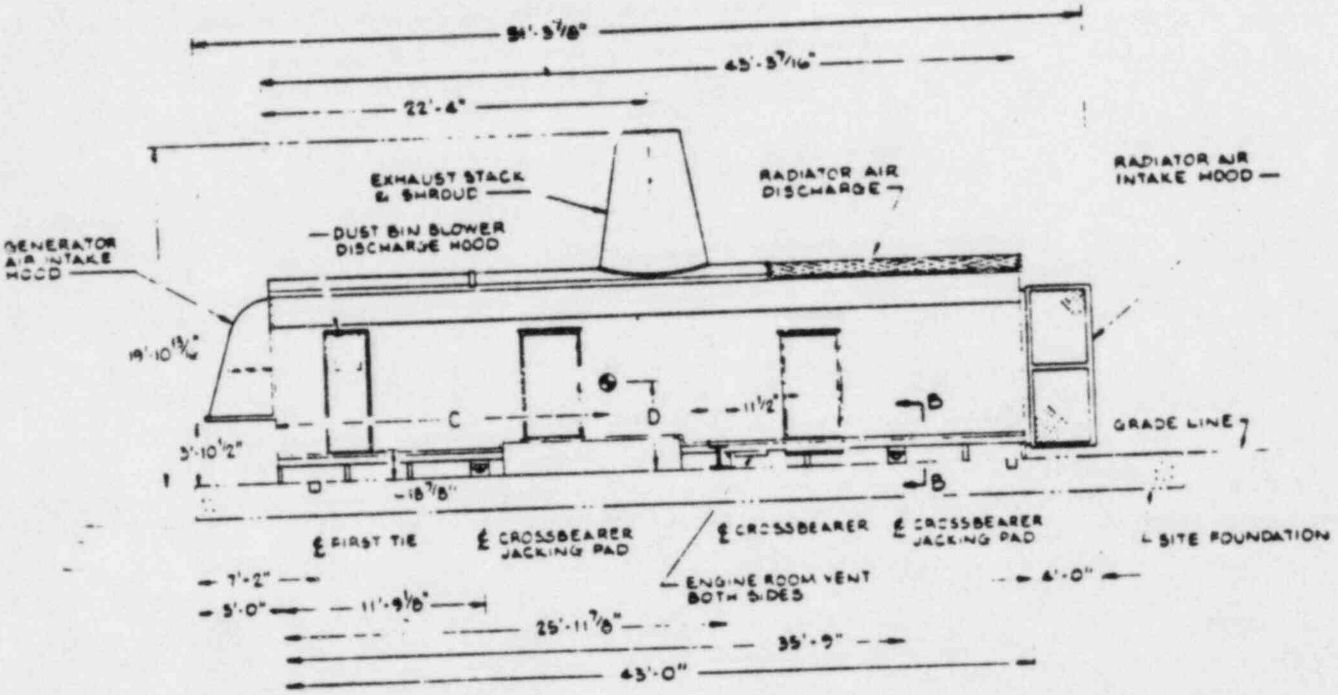

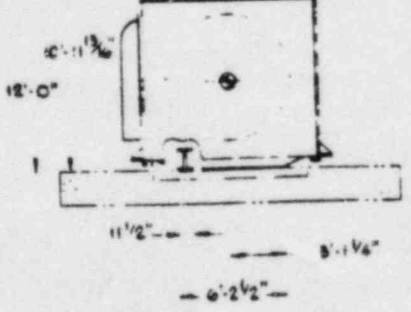

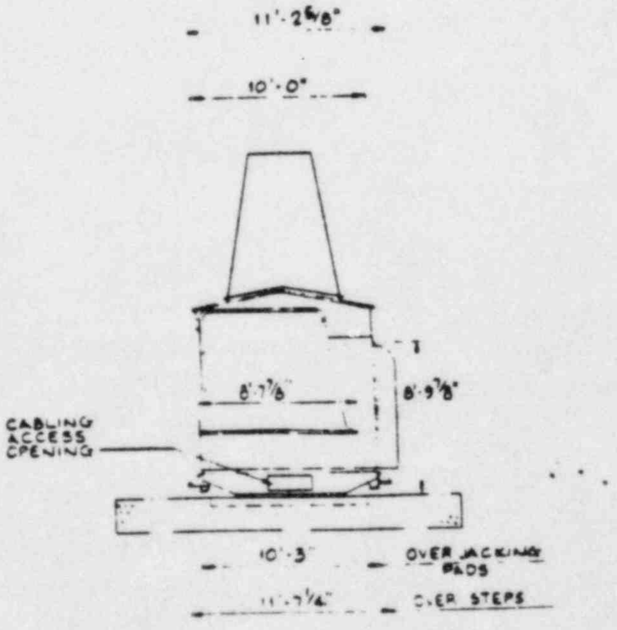

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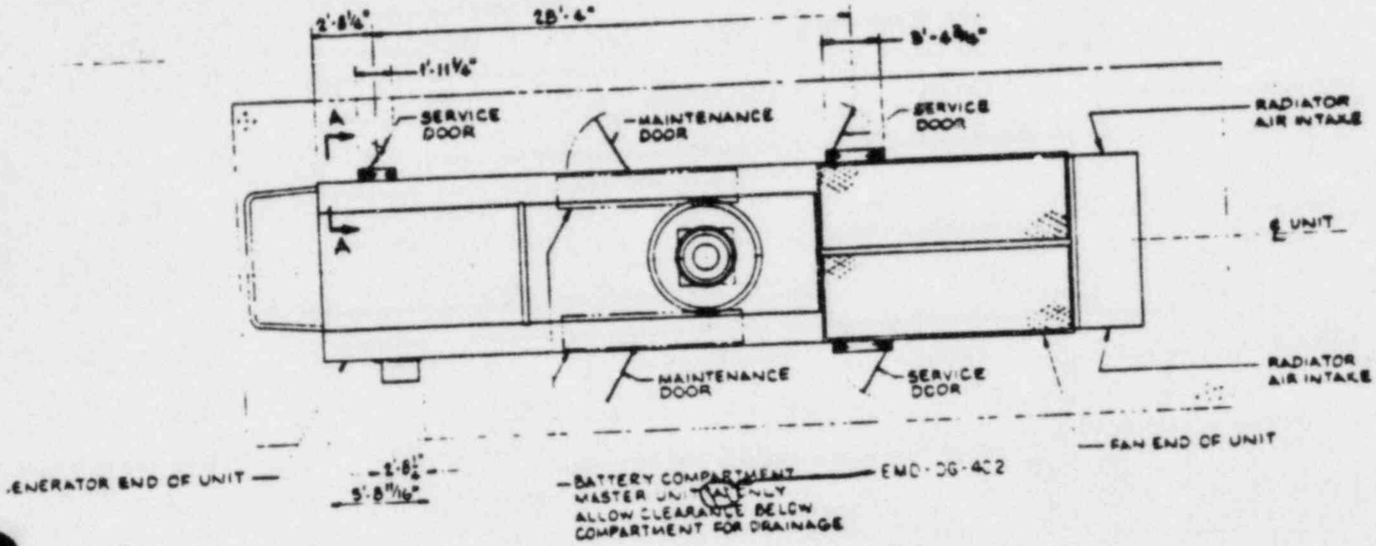
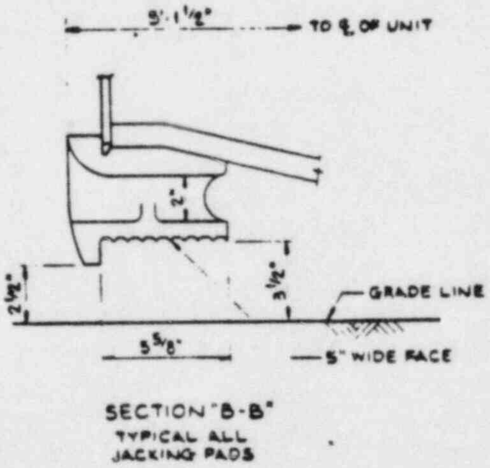
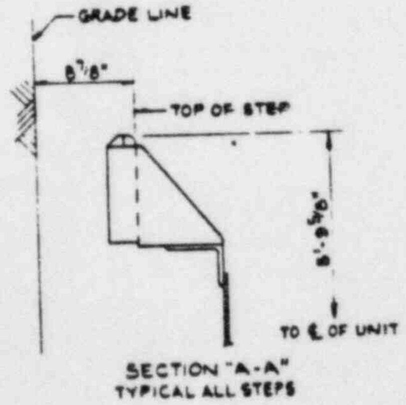




SECTION "B-B"
TYPICAL ALL JACKING PADS



SECTION "A-A"
TYPICAL ALL STEPS

	MASTER POWER UNIT "A"	POWER UNIT "B"
CENTER OF GRAVITY - C	19'-7"	19'-7½"
CENTER OF GRAVITY - D	5'-0½"	5'-2¼"
WEIGHT LESS SUPPLIES	128,300	124,600
WEIGHT WITH SUPPLIES	136,000	132,300

MODEL MP-45
MFR: ELECTRO-MOTIVE DIVISION
GENERAL MOTOR CORPORATION
LA GRANGE, ILLINOIS

NOTE:

EXHAUST STACK, STACK SHROUD, RADIATOR AIR INTAKE HOOD ENGINE GENERATOR AIR INTAKE HOOD, DUST BIN BLOWER DISCHARGE HOOD & STEPS ARE INSTALLED AFTER POWER UNIT IS IN PLACE ON SITE FOUNDATION.

Form GQ 3.08.1 Rev. 2

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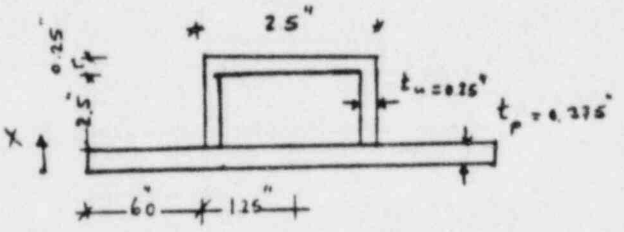

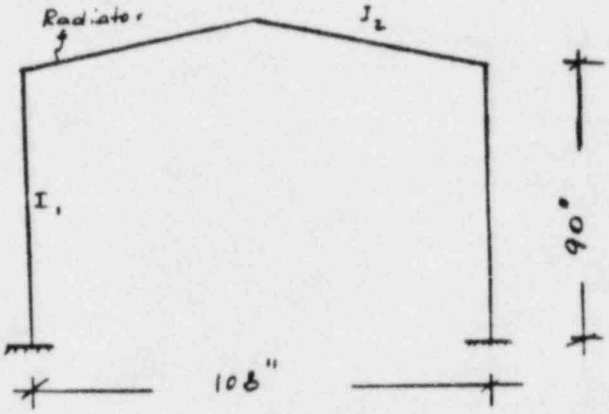

$$A = 14.5 \times 0.375 + 3 \times 2.5 \times 0.25$$

$$= 5.437 + 1.875 = 7.3 \ In^2$$

$$\bar{X} \cdot A = 5.437 \times \frac{0.375}{2} + 2.0 \times 0.75 \times (2.5 + 0.375 - \frac{0.25}{2}) + 2 \times 2.5 \times 0.25 \times (\frac{2.5}{2} + 0.375)$$

$$= 1.019 + 1.375 + 2.03 = 4.424 \ In^3$$

$$\bar{X} = \frac{4.424}{7.3} = 0.606"$$

$$I_1 = 14.5 \times \frac{(0.375)^3}{12} + 5.437 \left(0.606 - \frac{0.375}{2}\right)^2 + 2 \times 0.25 \times \frac{(2.5)^3}{12} + 2 \times 0.625 \times (1.019)^2$$

$$+ 2.0 \times \frac{(0.25)^3}{12} + 0.5 \times (2.144)^2$$

...

$$= 0.0637 + 0.9522 + 0.6510 + 1.2979 + 0.0026 + 2.2984$$

$$I_1 = 5.2658 \ In^4$$

$$C_{max} = 2.5 + 0.375 - 0.606 = 2.269 \ In$$

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Calculation of the Eigenfrequency As a Portal Frame

from Ref. 1. Page 221

$$f_1 = \frac{1}{2\pi}\left[\frac{12\sum J_i E_i}{L^3(M+0.37\sum M_i)}\right]^{1/2}$$

$$M = \frac{4500}{760} \times 24 \cdot \frac{1}{386.4} = \frac{1421}{386.4}$$

$$= 3.677$$

$$M_i = 7.3 \times 0.283 \times 90 \cdot \frac{1}{386.4} = 0.4812$$

$$f_1 = \frac{1}{2\pi}\left[\frac{12 \times 2 \times 5.3366 \times 29 \times 10^6}{(90)^3(3.677 + 2 \times 0.37 \times 0.4812)}\right]^{1/2}$$

$$= \frac{1}{2\pi}\left[\frac{3.714.3 \times 10^6}{729 \times 10^3 \times 4.033}\right]^{1/2}$$

$$f_1 = 5.65 \text{ Hz}.$$

$$\frac{W}{2} = \frac{1421}{2} = 710.6\ \#$$

The additional parts will be considered by an increase of the weight of radiators from 7105# to 800 # per column

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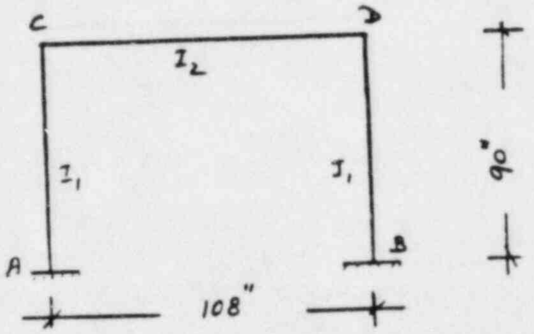
<u>Analysis of Stresses as a Frame</u>
(See Reference 2: Beton-Kalender 1970, Pages 501 & 502)

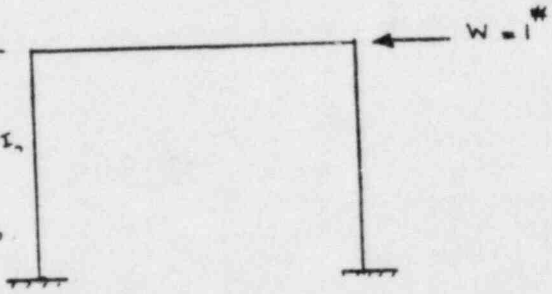
$$K = \frac{I_2}{I_1}\,\frac{h}{l} = \frac{10}{1}\,\frac{90}{108} = 8.33$$



$$M_A = -M_B = \frac{1.0 \times 90}{2} \times \frac{3 \times 8.33 + 1}{6 \times 8.33 + 1}$$

$$= 45 \times \frac{25.99}{50.98} = 22.94 \text{ \#-In}$$

$$M_C = -M_D = 45 \times \frac{24.99}{50.98} = 22.05 \text{ \#-In}$$



$$W = 1.600 \times 1.5 \times 0.55 = 1320 \text{ \#-In}$$

$$M_A = -M_B = 22.94 \times 1320 = 30280 \text{ \#-In}$$

$$V_A = \frac{3 \times 1320 \times 90 \times 8.33}{108(50.98)} = 539 \text{ \#}$$

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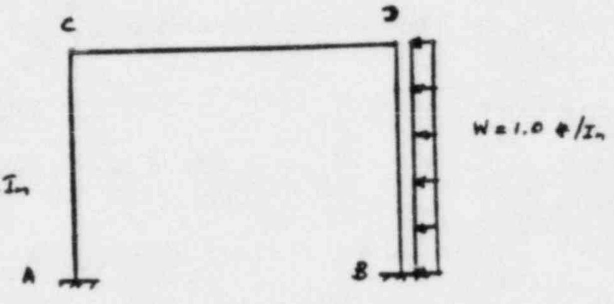


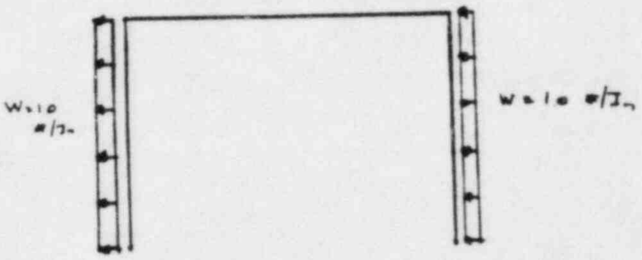
$$M_A = \frac{1.0 \times (90)^2}{24}\left(\frac{5 \times 8.33 + 9}{8.33 + 2} - \frac{12 \times 8.33}{6 \times 8.33 + 1.0}\right)$$

$$= 337.5(4.903 - 1.961) = 992.9 \text{ \#-In}$$

$$M_B = \frac{1.0 (90)^2}{24}\left(12 - \frac{5 \times 8.33 + 9}{8.33 + 2} - \frac{12 \times 8.33}{6 \times 8.33 + 1.0}\right)$$

$$= 337.5(12 - 4.903 - 1.961) = 1733.4 \text{ \#-In.}$$

$$V_A = -V_B = \frac{1.0 \times 8100 \times 8.33}{108 \times 50.98} = 12.25 \text{ \#}$$



$$M_A = M_B = 1733 + 993 = 2726 \text{ \#-In}$$

$$V_A = -V_B = 25 \text{ \#}$$

$$W = (1.875 + 24 \times 0.375)\, 0.283 = 3.077 \text{ \#/In}$$

$$M_A = M_B = 2726 \times 3.077 \times 1.5 \times .055 = 6920 \text{ \#-In.}$$

$$V_A = -V_B = 25 \times 3.077 \times 1.5 \times 0.55 = 63.5 \text{ \#}$$

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Axial Force N:

$N_{seismic} = (1+1.5+0.37)(3.077 \times 90+800) = 1674$ #

Total Forces Due to Dead Weight & Seismic:

$M = 30280 + 6920 = 37200$ #-In

$V = 1674 + 539 + 63.5 = 2277$ #

$\sigma_{seismic} = \frac{2277}{7.30} + \frac{37200}{5.2658} \times 2.269$

$= 312 + 16029 = 16341$ psi

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Stresses Due to Wind Load

$P = q_z I_w^2 C_{PL} + 0.25\ q_z I_w^2$

$q_z = 27$ psf.

$I_w = 1.07$

$C_{PL} \approx 1.0$

$P = 27 \times (1.07)^2 \times 1.0 + 0.25 \times 27 \times (1.07)^2$

$= 30.9 + 7.73 = 38.63$ psf

$A = 1.0 \times 1.0 = 1.0$ fsq

$P = 38.63 \times 1.0 \times \frac{1}{12} = 3.22$ #/In

$M_{wind} = 3.22 \times \frac{(90)^2}{2} = 13041$ #-In.

$\sigma_{wind} = \frac{13041}{5.3356} \times 2359 = 5764$

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Total Stress:

$$\sigma_{Total} = \sigma_{Seismic} + \sigma_{Wind}$$

$$= 16754 + 5764$$

$$= 22518 < 0.6 \times 1.3 \cdot f_y = 0.78 \times 36000 = 28000 \text{ psi}$$

Allowable Stress per AISC Manual of Steel construction 8th edition.

B.9

B.9 MISCELLANEOUS COMPONENTS

Calcs. For LOG NO 080318(

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☒ Safety-Related ☐ Non-Safety-Related

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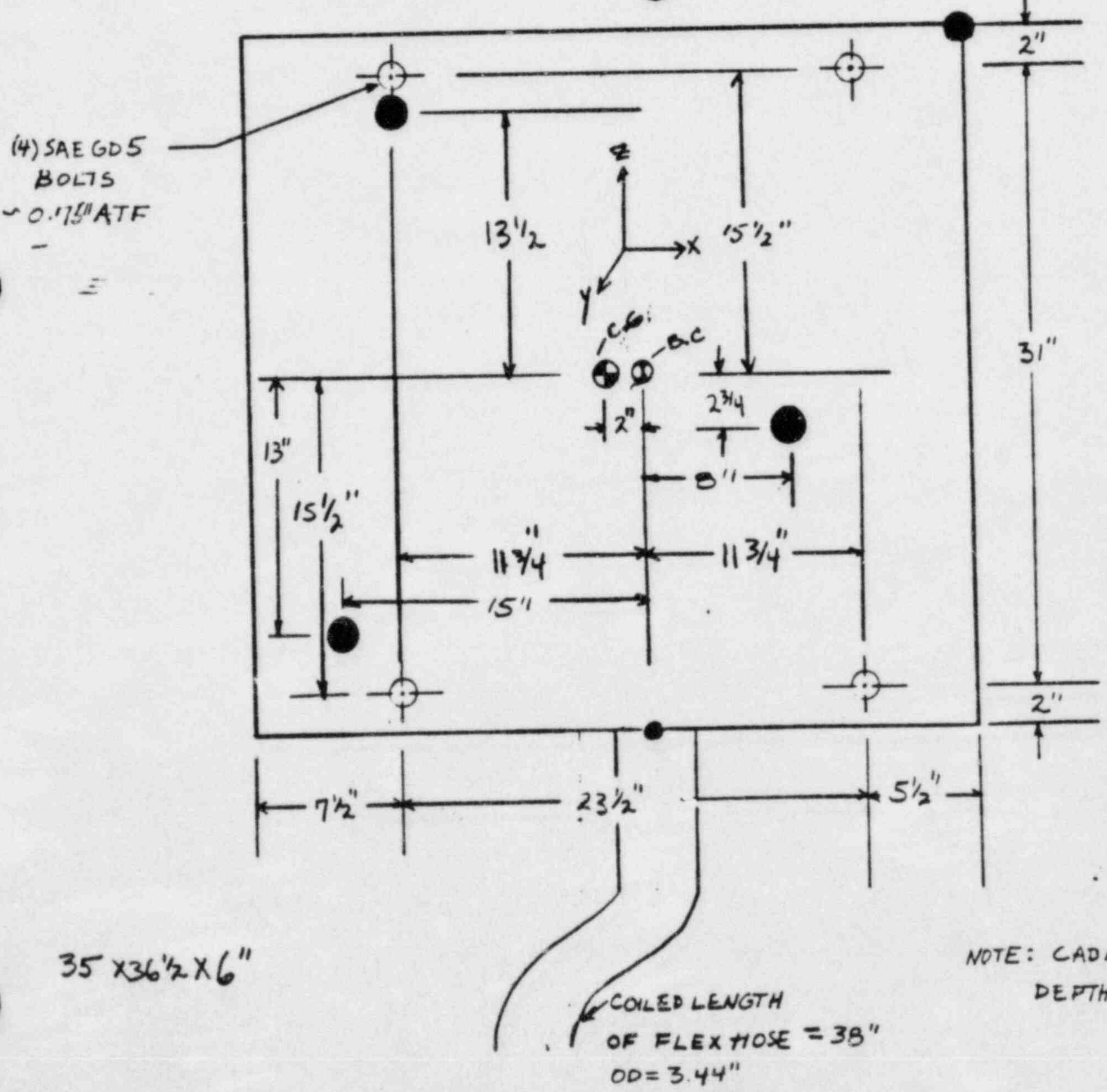
ENGINE CONTROL CABINET
(LOG NO 0803180100)

ESTIMATED CONCENTRATED LOAD
● 125 LBS
● 30 LBS
● 35 LBS



35 X 36½ X 6"

NOTE: CABINET DEPTH = 6"

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Calcs. For

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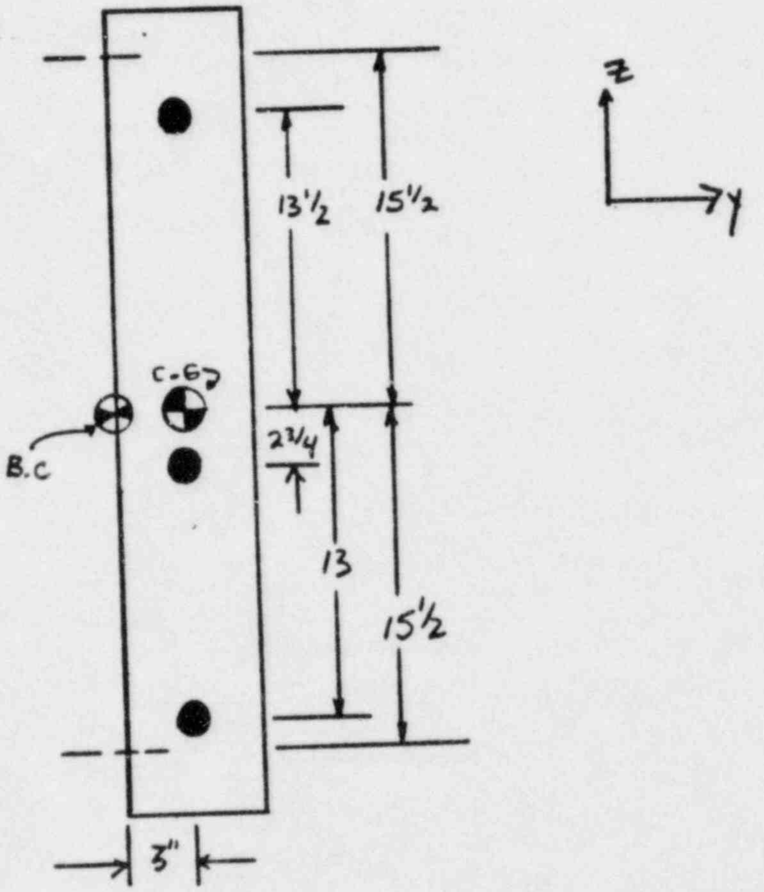

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USING PHOTOGRAPHS 51, 52, 53 AND FIELD GATHERED INFORMATION

BASED ON HOFFMAN ENCLOSURES NEMA TYPE 12 SINGLE DOOR ENCLOSE SIZE 36"x36"x8" CORRESPONDS TO 102 LBS
(CATALOGUE NO A-363608LP)

BASED ON HOFFMAN PANELS NEMA TYPE 12 PANEL SIZE 33x33 FOR ENCLOSURE SIZE 36x36
EQUALS 37 LBS (ENCLOSURE + PANEL = 102+37 = 139 LBS)

ANNACONDA FLEX HOSE MEASURED AT 3.44" OD FOR AN EXTENDED COIL LENGTH OF 38" AT BOTTOM

FOR 4" FLEX (CATALOGUE SC (2ND EDITION))

$$\frac{100\text{ LBS}}{25\text{ ft}} = 4\text{ LBS/FT} = 0.33\text{ LB/IN}$$

$$\therefore \text{TOTAL WEIGHT} = .33\ \frac{\text{LB}}{\text{IN}} \times 38\text{ IN} \cong 13\text{ LBS}$$

(3) ANNACONDA FLEX HOSES APPROXIMATELY 3" FOR TOTAL OF 180 IN (CONSERVATIVE) AT TOP RIGHT HAND CORNER OF ENCLOSURE

$$\frac{73\text{ LBS}}{25\text{ FT}_{\text{1050}}} = 2.92\ \frac{\text{LBS}}{\text{FT}} = 0.243\ \frac{\text{LB}}{\text{IN}} \times 180\text{ IN} = 44\text{ LBS}$$

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- CONCENTRATED LOADS TRANSFERRED TO BOLT CENTROID

$F_X = 125\text{ LBS}(2.8g) = 350 \text{ LBS}$

$F_Y = 125\text{ LBS}(2.8g) = 350 \text{ LBS}$

$F_Z = 125\text{ LBS}(7.1g) = 887.5 \text{ LBS}$

$M_X = F_Y(z) + F_Z(y) = 350(2.75) + 887.5(3) = 3625 \text{ IN-LBS}$

$M_Y = F_X(z) + F_Z(x) = 350(2.75) + 887.5(8) = 8062.5 \text{ IN-LBS}$

$M_Z = F_X(y) + F_Y(x) = 350(3.0) + 350(8) = 3850 \text{ IN-LBS}$

- CONCENTRATED LOADS TRANSFERRED TO BOLT CENTROID

$F_X = 30\text{ LBS}(2.8g) = 84 \text{ LBS}$

$F_Y = 30\text{ LBS}(2.8g) = 84 \text{ LBS}$

$F_Z = 30\text{ LBS}(7.1g) = 213 \text{ LBS}$

$M_X = 84(13.5) + 213(3) = 1773 \text{ IN-LBS}$

$M_Y = 84(13.5) + 213(11.75) = 3636.75 \text{ IN-LBS}$

$M_Z = 84(3.0) + 84(11.75) = 1239.0 \text{ IN-LBS}$

- CONCENTRATED LOADS TRANSFERRED TO BOLT CENTROID

$F_X = 35\text{ LBS}(2.8g) = 100.8$

$F_Y = 35\text{ LBS}(2.8g) = 100.8$

$F_Z = 35\text{ LBS}(7.1g) = 248.5$

$M_X = 100.8(13.0) + 248.5(3.0) = 2055.9 \text{ IN-LBS}$

$M_Y = 100.8(13.0) + 248.5(15.0) = 5037.9 \text{ IN-LBS}$

$M_Z = 100.8(3.0) + 100.8(15.0) = 1814.4 \text{ IN-LBS}$

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Calcs. For

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ENCLOSURE PLUS PANEL LOADS TRANSFERRED TO BOLT CENTROID

FX = 139 LBS (2.8g) = 389.2
FY = 139 LBS (2.8g) = 389.2
FZ = 139 LBS (7.1g) = 986.9
MX = F(y) = 986.9 (3) = 2960.7 IN-LBS
MY = FZ(x) = 986.9 (2) = 1973.8 IN-LBS
MZ = Fx(y) + Fy(x) = 389.2(3) + 389.2(2) = 1946 IN-LBS

NOZZLE LOADING

- DUE TO THE FLEXIBLE NATURE OF THE ANNACONDA FLEX HOSE CONNECTIONS THE NOZZLE LOADS ARE ASSUMED NEGLIBLE. THE ONLY LOAD CONSIDERED WILL BE THE WEIGHT OF FLEX HOSE:

(HOSE RUNNING OUT BOTTOM)
FZ = Wt + Wt(a_z) = 13 + 13(7.1) = 105.3 LBS
FX = FY = Wt(a_x) = 13 (2.8) = 36.4 LBS
$M_x = M_y$ = 36.4 (17.5) = 637 IN-LBS
MZ = 0

((3) HOSES RUNNING AT TOP RIGHT HAND CORNER

FZ = 44 + 44 (7.1g) = 356.4 LBS
FX = FY = 44(2.8) = 123.2 LBS
MX = Fy(z) = 123.2 (17.5) = 2156
MY = Fx(z) + Fz(x) = 123.2 (17.5) + 356.4(17.25) = 8303.9 IN-LBS
MZ = Fy(x) = 123.2 (17.25) = 2125.2 IN-LBS

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TOTAL LOADS (SEISMIC +NOZZLE)

$F_X = 350 + 84 + 100.8 + 389.2 + 36.4 + 123.2 = 1083.6$ LBS
$F_Y = 350 + 84 + 100.8 + 389.2 + 36.4 + 123.2 = 1083.6$ LBS
$F_Z = 887.5 + 213 + 248.5 + 986.9 + 105.3 + 356.4 = 2797.6$ LBS
$M_X = 3625 + 1773 + 2055.9 + 2960.7 + 637 + 8303.9 = 19{,}355.5$ IN LBS
$M_Y = 8062.5 + 3636.75 + 5037.9 + 1973.8 + 637 + 8303.9 = 27{,}652$ IN LBS
$M_Z = 3850 + 1239.0 + 1814.9 + 1946 + 2125.2 = 10{,}975$ IN LBS

BOLT AREA = $A = \frac{\pi}{4} (.5 \text{ IN})^2 = 0.196 \text{ IN}^2$ (based on the gross Area)

TOTAL BOLT AREA = $\Sigma A = 0.196 \text{ IN}^2 \times 4 = .784 \text{ IN}^2$

$I_{ZZ} = 4 (0.196 \text{ IN}^2)(11.75 \text{ IN})^2 = 108.241 \text{ IN}^4$

$I_{XX} = 4 (0.196 \text{ IN}^2)(15.5 \text{ IN})^2 = 188.356 \text{ IN}^4$

$J = I_{ZZ} + I_{XX} = 296.597 \text{ IN}^4$

$C_{X\,MAX} = 11.75$ IN

$C_{Z\,MAX} = 15.5$ IN

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TENSILE STRESS

$$\sigma_t = \frac{F_y}{\Sigma A} + \frac{M_x C_{z\,max}}{I_{xx}} + \frac{M_Z C_{x\,max}}{I_{zz}}$$

$$= \frac{1083.6}{0.784} + \frac{19,355.5(15.5)}{188.356} + \frac{10,975(11.75)}{108.241}$$

$$= 4171 \text{ PSI}$$

SHEAR STRESS

$$\tau_z = \frac{F_z}{\Sigma A} + \frac{M_x C_{z\,max}}{J} = \frac{2797.6}{0.784} + \frac{19,355.5(15.5)}{296.597}$$

$$= 4580 \text{ PSI}$$

$$\tau_x = \frac{F_x}{\Sigma A} + \frac{M_Z C_{x\,max}}{J} = \frac{1083.6}{0.784} + \frac{10,975(11.75)}{296.597}$$

$$= 1817 \text{ PSI}$$

$$\tau_{max} = \left[\tau_x^2 + \tau_z^2\right]^{1/2} = \left[(4580)^2 + (1817)^2\right]^{1/2} = 4927.3 \text{ PSI}$$

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ALLOWABLE STRESS

From ASME B+PV Code, 1977 Edition with summer 79 addendum, section III, article 2460 of Appendix XVII;

$F_t = 0.3Su = 0.3(^{*}120{,}000\ psi) = 36{,}000\ psi$ (service level A+B)

$F_v = 0.124Su = 0.124(^{*}120{,}000\ psi) = 14{,}880\ psi$ (")

$f_t = 4171\ psi$

$f_v = 4927\ psi$

The following Condition must be met:

$$\frac{f_t^2}{F_t^2} + \frac{f_v^2}{F_v^2} \leq 1.00$$

$$\frac{4171^2}{36{,}000^2} + \frac{4927^2}{14{,}880^2} = 0.123 \leq 1.00$$ the bolts are adequate.

* Note – From ASME B+PV CODE, 1977 Edition, section III, Table I-7.3 of Appendix I, SA-449 steel.

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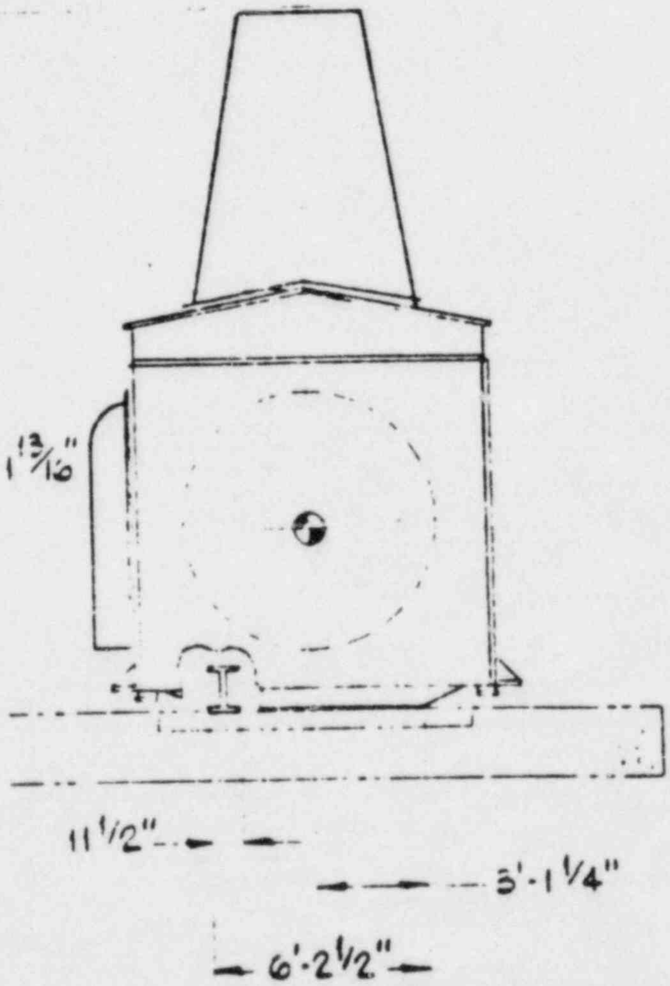
Prepared by Dail Wright Date 5/17/84
Reviewed by Date
Approved by Date

Engine Skid Assy.

Log No. 0803010000



The engine, generator and all of the components are mounted on a large framework or skid assembley. The skid is constructed of large I beams with steel plating bolted and welded to the top forming a walkway. The Engine and Generator are bolted to the skid. Because of this the skid becomes a very rigid foundation for the assembly.

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Cross Sectional Properties of The W-Section

The following dimensions were obtained via a field measurement ; (see Phone memo)

section height ≅ 16 in

flange thickness ≅ 1 in

using this information along with an 11½ in flange width (see cover page, from Shoreham Architectural drawing) the section was determined to be a W16x96.

Method of Analysis

Because of the unpredictable nature of the ground motion a conservative approach will be taken to show the adequacy of the skid structure. The engine skid is resting on a bed of railroad ties. Due to the unpredictable nature of the ground under the ties the skid will be assumed to be simply supported at the ends and checked for deflections. This condition would occur if all of the railroad ties except the two end supports would sink and not carry any load.

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Equipment Weight

Engine Weight = 43,091 Lbs (From Engine Owners manual)

Generator weight = 18,100 Lbs (" ")

Radiator weight = 9,000 Lbs (Phone Memo in radiator anal.)

Total Weight of unit = 136,000 Lbs (LILCO Shoreham Dwg. No F50272-1, Rev. 1, 4-25-84)

Seismic Loading

A static coefficient analysis will be performed in accordance with IEEE-344-1975. A static coefficient of 1.5 will be used to account for multi modal excitations and cross coupling.

Seismic "g" level = ENV[2%OBE, 3%SSE](1.5)

The stress in the I-Beams will be analyzed using a superposition method. The stress due to the uniform loads will be added to that caused by the concentrated loads (Engine, Generator, Radiator) and compared with the appropriate allowable.

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Uniform Load Stress

Weight Enclosure + The Skid Assy. = Total - Engine - Generator - Radiator

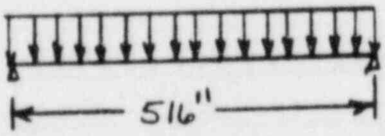
Weight = 136,000 - 9000 - 18,100 - 43,091 = 65,809 Lbs

Weight / beam = 65,809 / 2 Beams = 32,905 Lbs

ω = 32,905 Lbs / 516 in = 64 Lbs/in

$\omega_{stat+seis}$ = 64 Lbs/in + $(\overset{*}{0}.38g\text{'s})(1.5)(2/3)$ 64 Lbs/in = 88 Lbs/in

Analytical Model



$$M_{Max} = \frac{wL^2}{8} = \frac{(88)(516)^2}{8}$$

$$M_{max} = 2{,}928{,}816 \text{ in-lbs}$$

$I = 1360 \text{ in}^4$
W16×96

$$\sigma = \frac{Mc}{I} = \frac{(2{,}928{,}816)(16/2)}{1360}$$

$$\sigma = 17{,}228 \text{ psi}$$

*Note - Vertical Acceleration 2/3 Horizontal. From Shoreham Base spectrum, Env. [2% OBE, 3% SSE]

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Radiator Load Stress

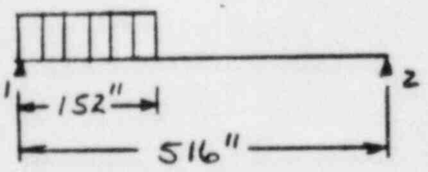
Radiator Weight = 9000 Lbs

Weight/Beam = 9000 Lbs/2 Beams = 4500 Lbs/Beam

$\omega = 4{,}500 \text{ Lbs} / {}^{*}152 \text{ in} = 30 \text{ Lbs/in}$

$\omega_{\substack{\text{static}\\ \text{+seismic}}} = 30 \text{ Lbs/in} + 30 \text{ Lbs/in}(0.38)(1.5)(2/3) = 41.4 \text{ Lbs/in}$

Analytical Model



$(a<c)\ R_1 = \frac{\omega b}{2L}(2c+b)$

$(x = a + \frac{R_1}{\omega})\ M_{max} = R1\left(a + \frac{R_1}{2\omega}\right)$

(the above relationships are from Design of Welded Structures, Blodgett)

a=0, b=152", C=364"
L=516 in

$R_1 = [(41.4 \text{ Lbs/in})(152)/2(516)](2\times 364 + 152) = 5366 \text{ Lbs}$

$M = 5{,}366\left(0 + \frac{5366}{2(41.4)}\right) = 347{,}753 \text{ psi}$

$\sigma = \frac{Mc}{I} = \frac{(347{,}753)(8)}{1360} = 2046 \text{ psi}$

* 152 in is the approximate length of the radiator. This was obtained through field measurements. See cooling system for further information.

Engine Load Stress

Engine Weight = 43,091 Lbs

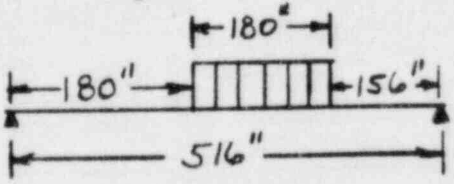
weight/Beam = 43,091 Lbs / 2 beams = 21,546 Lbs/Beam

Engine length ≅ 180 in (from (S+L) EMD-008028, Rev.00)

$\omega = 21{,}546\ \text{Lbs}/180\ \text{in} = 119.7\ \text{Lbs/in}$

$\omega_{\text{seismic}} = 119.7\ \text{Lbs/in} + 119.7(0.38)(1.5)(2/3) = 165.2\ \text{Lbs/in}$

Analytical Model



GEN. side

$$R_1 = \frac{Wb}{2L}(2c+b)$$

$$M_{max} = R_1\left(a + \frac{R_1}{2\omega}\right)$$

$$R_1 = \frac{(165.2)(180)}{2(516)}(2(180)+156)$$

$R_1 = 14{,}868$ Lbs

$$M = 14{,}868\left(180 + \frac{14868}{2(165.2)}\right) = 3{,}345{,}300\ \text{in-lbs}$$

$$\sigma = \frac{Mc}{I} = \frac{(3{,}345{,}300)(8)}{1360\ \text{in}^4} = 19{,}678\ \text{in-lbs}$$

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Generator Load Stress

Generator Weight = 18,100 Lbs (from Owners Manual)

Weight/Beam = 18,100 Lbs/2 Beams = 9,050 Lbs/Beam

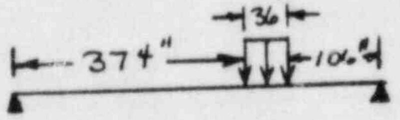
Generator length $\cong$ 36 in

ω = 9050/36 = 251 Lbs/in

ω_{seismic} = 251 Lbs/in + 251 Lbs/in (0.38)(1.5)(2/3) =

ω = 346 lbs/in

Analytical Model



$$R_1 = \frac{\omega b}{2L}(2c+b)$$

$$R_1 = \frac{(346)(36)}{2(516)}(2(374)+36) \qquad M_{max} = R_1\left(a + \frac{R_1}{2\omega}\right)$$

R_1 = 9463 Lbs

$$M = 9463\left(106 + \frac{9463}{2(346)}\right) = 1,132,443 \text{ in-lbs}$$

$$\sigma = \frac{Mc}{I} = \frac{(1,132,443)(8)}{1360} = 6,661 \text{ psi}$$

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Summary:

Uniform Load Contribution - 17,228 psi

Radiator Load Contribution - 2,046 psi

Engine Load Contribution - 19,678 psi

Generator Load Contribution - 6661 psi

Total Stress 45,613 psi

Material - A-36 steel

S_y = 36,000 psi

S_u = 58,000 psi (ASME Appendix I, table I-7.1)

$S_a = 0.66 S_y (1.6) = 38,400$ psi (for SSE loading, AISC)

The actual tensile stress calculated above is a very conservative estimate. All of the loading was assumed to act on the I-beams, No credit was taken for the rigidity of the equipment, peak × 1.5 levels of acceleration were used and the span was assumed to be supported only at the ends.

If we remove some of the conservative we can get a better, but stil very conservative approximation of the stress in the member.

Using Seismic SSE ZPA levels of 0.2g's (assuming the system is rigid) we have:

Verticle Acceleration = $0.2g's(2/3) = 0.133g's$

uniform $\omega = 64(1.133g's) = 73$ Lbs/in

Radiator $\omega = 30(1.133g's) = 34$ Lbs/in

Engine $\omega = 119.7(1.133g's) = 136$ Lbs/in

Generator $\omega = 251(1.133g's) = 284$ Lbs/in

$$M_{uniform} = (73)(516)^2/8 = 2,429,586$$

$$R_{1\,radiator} = [(30)(152)/2(516)][2(364)+152)] = 3888 \text{ Lbs}$$

$$M_{radiator} = 3888\text{Lbs}\left(\frac{3888}{2(41.4)}\right) = 182,602 \text{ in-lbs}$$

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$$R_{1\,\text{Engine}} = [(136)(180)/2(516)][2(180)+156)] = 12{,}240\text{ Lbs}$$

$$M_{\text{Engine}} = 12{,}240\left(180 + \frac{12240}{2(165.2)}\right) = 2656643\text{ in-lbs}$$

$$R_{1\,\text{Generator}} = [(284)(36)/2(516)][2(374)+36] = 7767\text{ Lbs}$$

$$M_{\text{Generator}} = 7767(106 + 7767/2(346)] = 910479\text{ in-lbs}$$

Estimated Stress

$$\sigma_a = \frac{\Sigma M c}{I} = \frac{(2429586 + 182{,}602 + 2656643 + 910479)(8)}{1360\text{ in}^4}$$

$$\sigma_a = 36{,}349\text{ psi}$$

$$\sigma_a = 36{,}349 < S_a = 38{,}400\text{psi}$$

Shear Stress

$\tau_r = F_r/A = 136{,}000\text{ Lbs}/28.2\text{ in}^2 = 4{,}823\text{ psi}$ (negligable)

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Conclusions:

The W-sections located below the structure have been shown, by a very conservative method, to maintain their structural integrity under a seismic SSE loading condition. The actual stress found in the ~~La~~LaSalle (ceco) Engine skid was;

$$\sigma_a = 8821 \text{ psi (from (S+L) EMD-008028)}$$

Noting the the general arrangement of the skid is similar we would expect, through a detailed analysis, to stress levels on the same or slightly higher order. The skid will remain structurally intact.

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MEMORANDUM OF
TELEPHONE CONVERSATION

Date: 5/14/84

Time: 10:30

Person Called: Charlie Gangone (Name) of L.I.L. Co. (Company)

Person Calling: David Wright (Name) of S&L (Company)

Project: Shoreham Project No. 6995-00

Subject Discussed: Dimensions for the Engine/Generator W-Section

Summary of Discussion, Decisions and Commitments:

Charlie stated that the beam was approximately 16" high with a flange thickness of approximately 1".

DW/eg

cc J. Sinnappan - 30
C. Gangone - LILCo
AEM/APD/DMW - 30

David Wright
Signature

File: ~~CQD-014117 - 30~~

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Prepared by David Wright Date 5/17/84
Reviewed by Date
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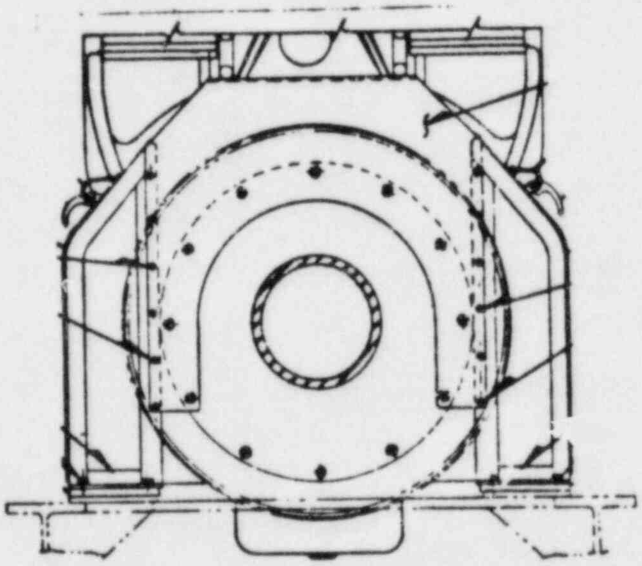
Flywheel Gaurds Left, Right

Log Numbers 0803020501, 502



The flywheel gaurds found on the Shoreham engines are identical to those found on the LaSalle (CE.Co.) Engines. The model number of each are EMB-8361631 and EMD-8361632. Since it has already been shown that the Owners group spectrum used in the analysis to qualify the other gaurd bounds the shoreham spectrum, all of the results will apply and can be used to show the adequacy of the equipment.

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Results Of Previous Analysis

The maximum stress was found to occur in the bolts. This value was

$\sigma_p = 3795$ psi

(found in (S+L) CQD file no. CQD-010330 Rev.00)

∴ This was well below the allowable stress of 23,000psi (SA-449).

Conclusion: The Gaurd will not come loose and damage any safety related Equipment.

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Prepared by Dal Wright Date 5/17/84
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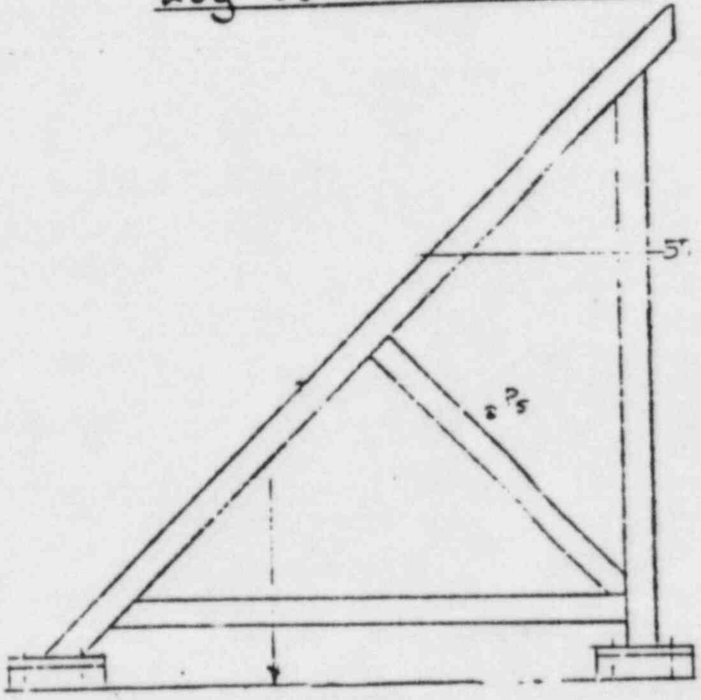
Accessory Rack

Log No. 0803030000



The accessory rack modeled in the finite element analysis of the La Salle Diesels ((S+L) EMD-008028 Rev.00 dated 4-19-77) is similar in size and construction to the Shoreham unit. Because of this similarity a comparison of the two will be made and adjustments done to account for any weight differences.

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Similarity of MaJor Comp.

The majority of the weight found on the Accy. racks comes from the Lube oil cooler, Lube oil filter and water expansion tank. A comparison of these item's will be made.

	LaSalle	Shoreham
Lube Oil cooler	① 1190 Lbs	② 2000 Lbs
Lube Oil Filter	② Identical	② Identical
Water Expansion tank	① 1500 Lbs	② 3897 Lbs

It should be pointed out that the shoreham weights are very conservative estimates of the actual weight.

Notes: ① Found in (S+L) EMD-008028 Rev.00, 4-19-77

② Found in this report under the appropriate item.

TITLE: MAXIMUM STRESSES

DATE: 2/11/77
FILE: W.O. 68683

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
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DIESEL GENERATOR SET -

S No.	Structure Region	Element No.	Node No.	Stress PSI
1	Generator Stator	329	1033-1086-1047	2262.
2	Generator Rotor	52	1128-1129	2016.
3	Generator Brg. Brackets	396	1225-1241	4748.
4	Exciter Stator	1023	847-855	415.
5	Accessory Rack Frame	489	334-345	5786.
6	Lube Oil Filter	529	523-527-531	5553.
7	Amot Thermostat	581	612-641-615	1320.
8	Water Expansion Tank	664	735-731-739	3235.
9	Heat Exchanger	713	241-240-239-245	20783.
10	Jacket Water Piping	777	444-443-449-451	12621.
11	Accessory Rack Piping	676	659-660	1058.
12	Base	329	135-136	8821.

AIR-START SKID-MOUNTED ASSEMBLY -

1	Air Receivers	16	30-31	107.
2	Base, Compressors and Air Dryers	26	24-34	1037.

SHEET II-3

FORM NO. 51-SERV.

ENGINEERING DEPARTMENT

STEWART & STEVENSON

TITLE: AX. BOLT LOADS

DATE: 2/25/77 FILE: 68683

PERFORMED BY: SHEKAR

DIESEL GENERATOR SET

SLNO	LOCATION OF BOLT	SIZE IN	STRESS AREA IN^2	TENSILE LOAD LBS.	SHEAR LOAD LBS.
1	FOUNDATION BOLT	1½	1.4041	13991.0	7197.0
2	EXCITER TO BASE CONNECTION BOLT	1¼	0.9684	341.0	291.0
3	GENERATOR TO BASE CONNECTION BOLT	1½	1.4041	16589.0	4033.0
4	GEN. STATOR TO BRG. BRACKET CONN. BOLT	1	0.6051	1273.0	4132.0
5	AIR TURNING BOX CONNECTION BOLT	½	0.1416	227.0	1045.0
6	ENGINE TO BASE CONNECTION BOLTS	1	0.6051	14368.0	10151.0
7	ACC. RACK FRAME TO BASE CONN. BOLT	7/8	0.4612	1661.0	644.0
8	WATER EXPANSION TANK CONN. BOLT	7/8	0.4612	452.0	350.0
9	LUBE OIL FILTER CONNECTION BOLT	¾	0.334	926.0	[illegible]
10	LUBE OIL COOLER CONNECTION BOLT	½	0.1416	2125.0	[illegible]

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Total LaSalle weight = 2690 Lbs

Total Shoreham weight = 5897 Lbs (estimated)

The shoreham equipment is $\frac{5897}{2690}$ = 2.19 times heavier.

since all of the mounting positions and the ecentricities are very similar a direct increase in the previous levels of stress will be done.

Maximum Accessor Rack Frame Stress (Previous) = 5786 psi

Maximum Predicted Accy. Rack frame stress for Shoreham = 5786(2.19) = 12,671 psi

Sa = 13.7 ksi (for SA-36 steel per ASME B+PV CODE, 77 edition, Section III, table I-7.1 of Appendix I)

Sa > σ_{max} so the frame is adequate

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Maximum Base Connection Bolt Loading (Previous) = 1661 Lbs (tension), 664 Lbs (shear)

Maximum Base connection Loading Predicted for Shoreham = 1661 Lbs (2.19) = 3638 Lbs (tension)

664 Lbs (2.19) = 1454 Lbs (shear)

Maximum Predicted Bolt Stress

Bolt Size – 7/8" SAE Gr. 5, Stress Area = 0.4612in

$\sigma_t = F_t/A = 3638 \text{ Lbs}/0.4612 \text{ in}^2 = 7888 \text{ psi}$

$\tau_v = F_v/A = 1454 \text{ Lbs}/0.4612 \text{ in}^2 = 3153 \text{ psi}$

$$\sigma_p = \frac{7888}{2} + \sqrt{\left(\frac{7888}{2}\right)^2 + 3153^2} = 8993 \text{ psi}$$

Sa = 23 ksi (for SA-449 bolting material per ASME B+PV code; 77 edition; Section III; Table I-7.3 of Appendix I.)

$S_a > \sigma_p$ so the bolts are adequate.

Conclusions: The Accy. rack has been conservatively shown to withstand a seismic event.

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C.1 ELECTRICAL EQUIPMENT PANELS

APPENDIX C.1a

MODAL AND RESPONSE SPECTRA
FINITE ELEMENT ANALYSIS OF AUXILIARY
AND DIESEL GENERATOR SWITCHGEAR
CUBLICLE INSTRUMENT PANELS

Purpose:

To objective of this analysis is to determine the dynamic characteristics of the door panels of the switchgear cubicles 1 and 1A using finite element modelling techniques and subsequently determine the stresses and deflections.

Conclusion:

The Auxiliary Switchgear Cubicle door panel is too flexible in its original configuration to adequately support the equipment mounted there during the specified seismic event. Reinforcing the panel with L2x2x1/4" angle stock stiffens the panel so that its dynamic characteristics are similar to those of the Diesel Generator Switchgear Cubicle door panels. The reinforced Auxiliary Switchgear Cubicle door panel has a lower first natural frequency than the Diesel Generator Switchgear Cubicle panel and supports more and heavier instruments. The reinforced Auxiliary Switchgear Cubicle panel was therefore used as a conservative bounding model for calculation of the maximum displacement and panel stress for all panels due to an SSE event. The maximum stress in the panel was calculated to be 990 lb/in^2 and the maximum stress in a reinforcing beam was 1222 lb/in^2. The maximum instrument deflection was 0.033". These values are well below allowable levels. The reinforced Auxiliary Switchgear Cubicle panel and the Diesel Generator Switchgear Cubicle panels are therefore qualified for the specified seismic events.

References

1. Field sketch prepared by I. T. Kisisel, attached as Figures 3 and 4.
2. Long Island Lighting Company, Drawing Number H-8113-1 Revision 1 4/25/84.
3. Sargent & Lundy Structural Analysis Program SLSAP, Program No. 097.130-6.6.

Introduction:

The instrument panels of the Auxiliary and Diesel Generator Switchgear Cubicles support electrical instruments that are necessary for the operation of the diesel generators. These generators must function during and after seismic events. The panels must therefore support the instruments during and after seismic excitation and not amplify the motion to an unacceptable level. The maximum displacement of the panel and panel stress must remain at safe levels. These conditions will be shown to be satisfied by finite element analysis.

Method:

Electrical controls for the four diesel generators at Shoreham are mounted on five panels. Four identical panels are mounted on Diesel Generator Switchgear Cubicles individually dedicated to one of the diesel generators and the fifth panel is mounted on the Auxiliary Switchgear Cubicle. Two finite element models were constructed to model both panel configurations. Figures 1 and 2. The numbered locations in Figures 1 and 2 indicate the locations of the instruments whose mass and mass center distance from the center plane of the panel (eccentricity) were included in the model. These values are shown in tables 1 and 2. The instrument eccentricities, panel measurements and panel configurations were obtained from field measurements. (Reference 1, Figures 3 and 4). Dimensions not shown in those figures were obtained from reference 2.

The sides of the panels are stiffened by a section of plate that was rolled to form a stiffening rib. The top and bottom of the diesel generator switchgear cubicle panel are stiffened by a one inch section of plate bent to a 90 degree angle to the panel. The structural effects of these stiffeners were modelled by beams whose properties are representative of these sections. (Section 2)

Both panels are hinged on one side and restrained by bolts on the other. The panel models were completely constrained at the bolt locations and only rotationally free about the hinge axis at the hinge locations. The panels were modelled as structural steel. Panel stress due to seismic excitation was calculated by the response spectrum method. The horizontal response spectrum for the Shoreham SSE with 3% damping is shown in Figure 5. The vertical response spectrum is formed by decreasing the acceleration values of the horizontal spectrum by one third. These spectra were closely enveloped and combined with the first ten modes of the more flexible panel to calculate stress and displacement due to dynamic excitation. The contribution of modes higher than the tenth was included by performing a pseudo static analysis for the ZPA acceleration of these spectra and combining these results with the response spectra analysis.

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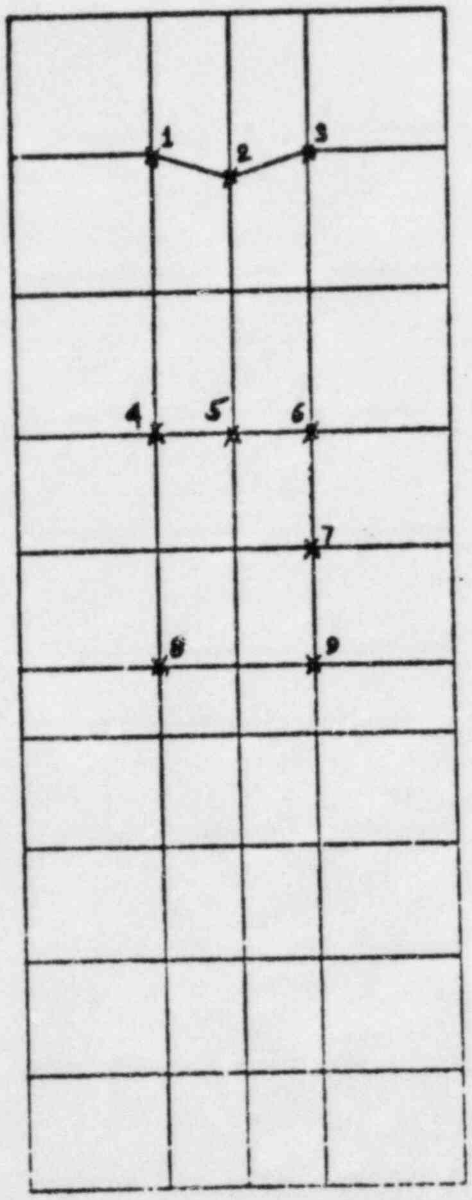

Numerals Indicate
Instrument Locations
(Refer to Table-1)

Figure 1 - Finite Element Model of
Generator Switchgear Cubicle
door panel

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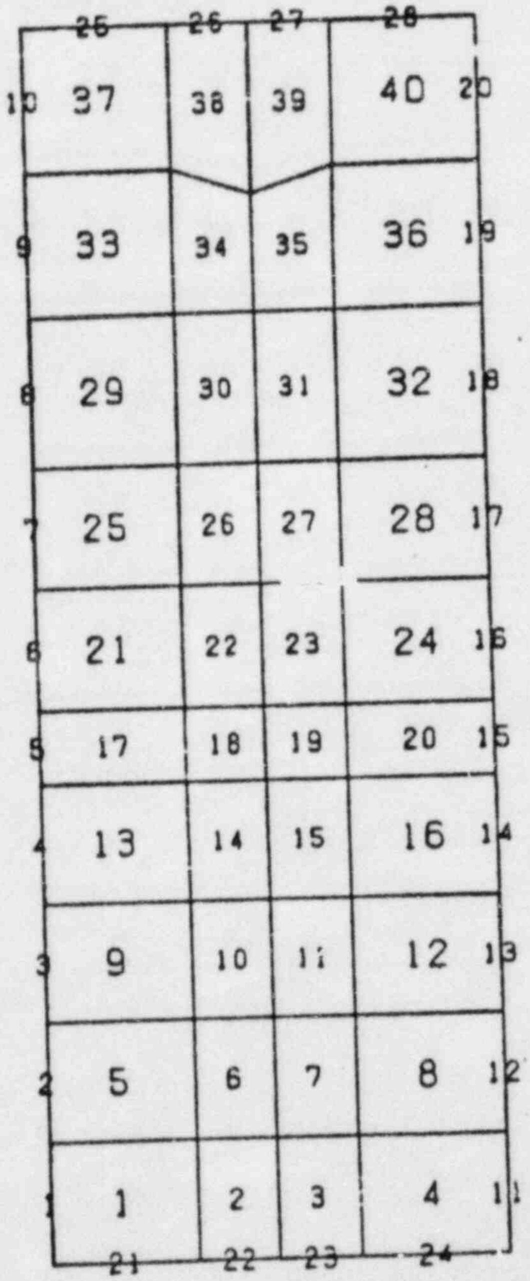

Figure 1.a - Element Numbers for Generator
Switchgear Cubicle door panel

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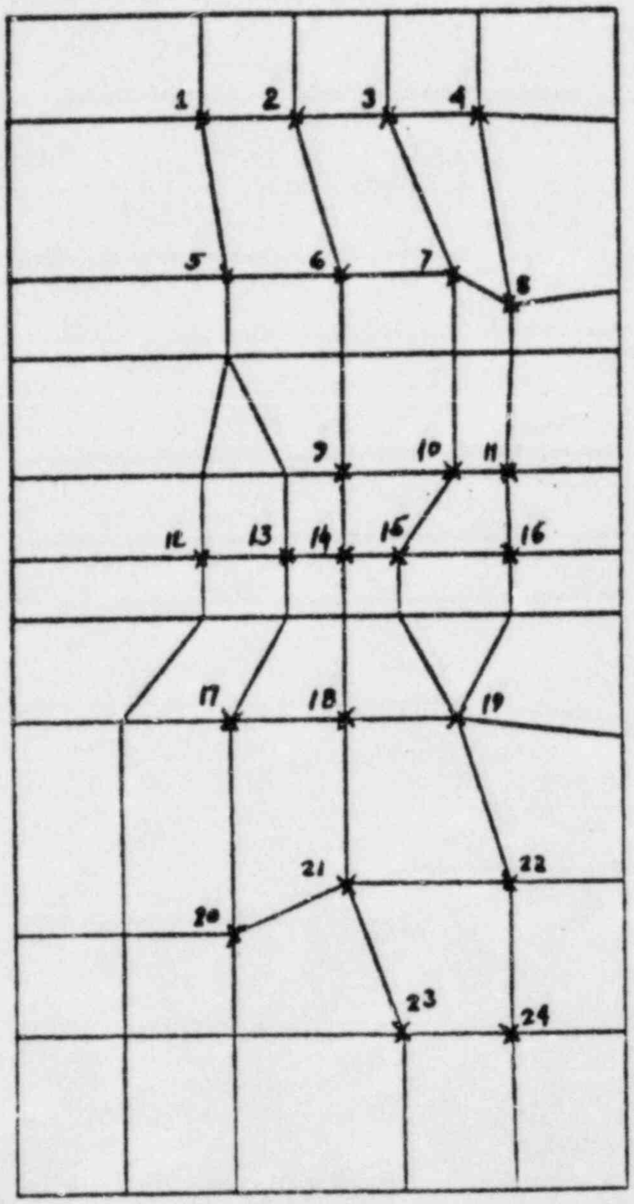

Numerals Indicate
Instrument Locations
(Refer to Table-2)

Figure 2 - Finite Element Model of Auxiliary Switchgear Cubicle 1A door panel

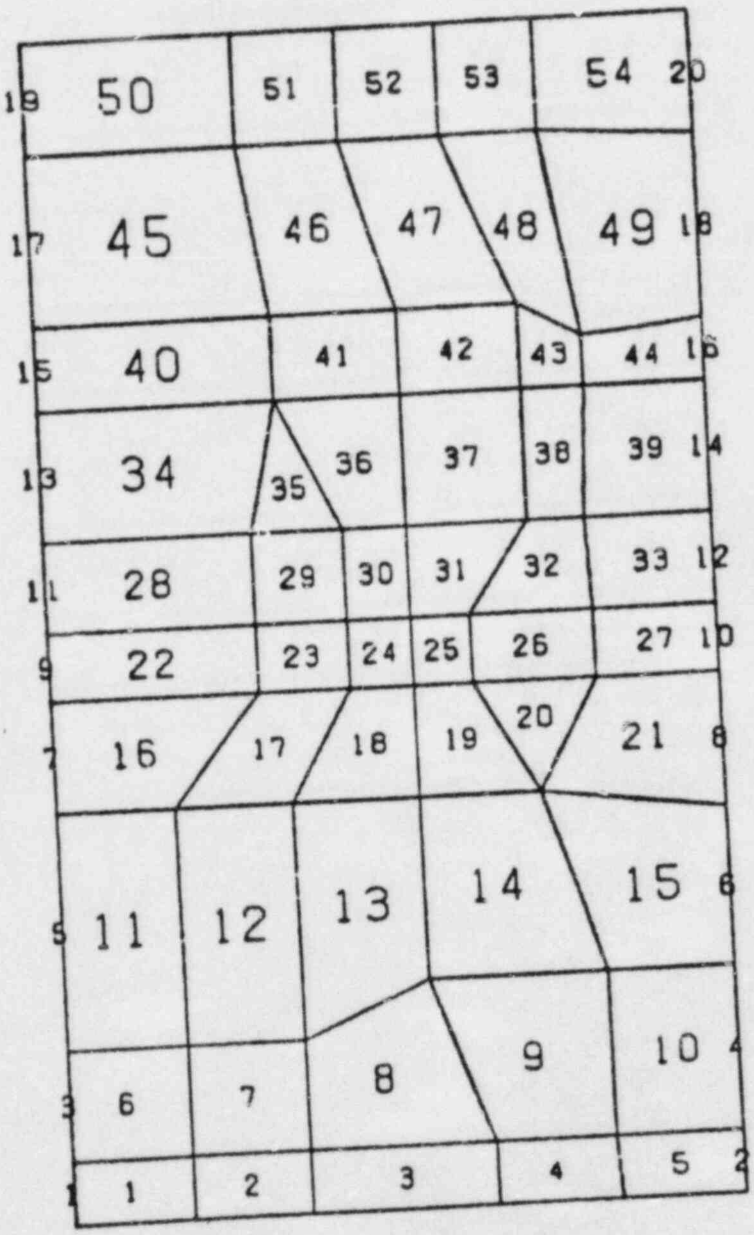

Figure 2.a - Element Numbers for Switchgear Cubicle 1A door panel

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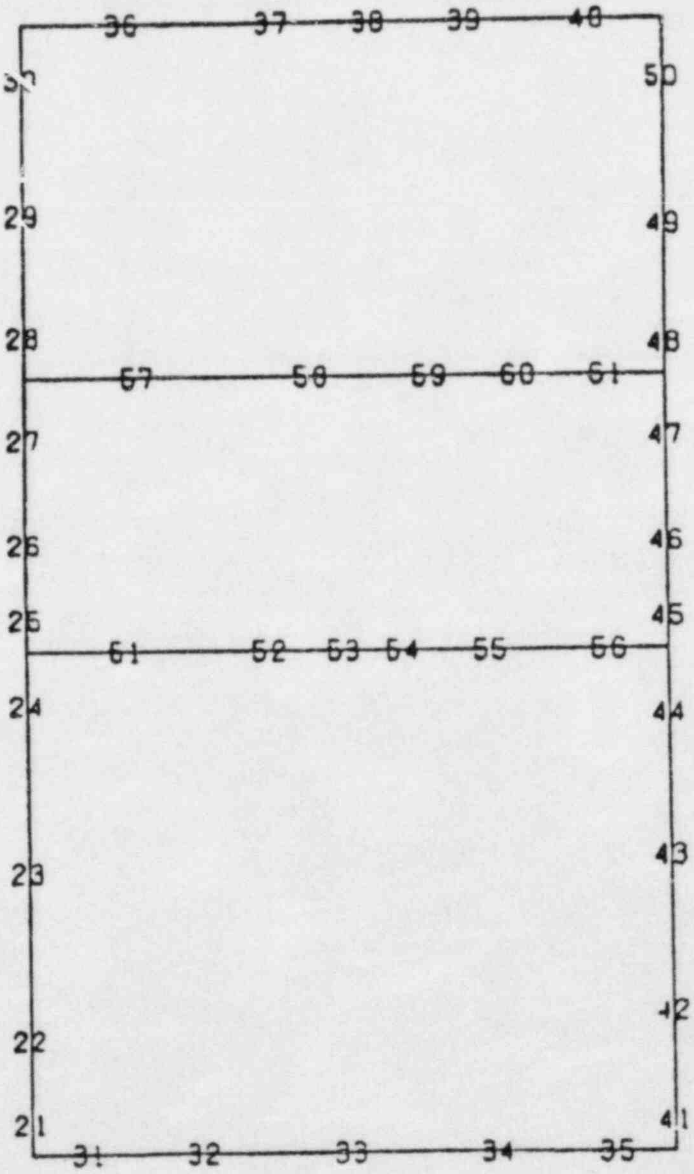

Figure 2.b – Proposed Reinforcing Member Element Numbers for Switchgear Cubicle 1A door panel

Location # in Figure 1	Instrument	Weight (lb)	Mass (slugs/12)	Eccentricity (in.)
1	Leading Vars Relay 55 Westinghouse CW 289B988A18A	10	.0258799	3
2	Reverse Power Relay 67 Westinghouse CW 289B988A09	12	.0310559	3
3	Overvoltage Relay Westinghouse CV5 1875512A	10	.0258799	3
4	AMM-WM-VAR Switch Allis Chalmer Type Z10 14-174-60 3-501	5	.01294	3.5
5	CB Control Switch Allis Chalmer Type 210	5	.01294	1.5
6	Voltmeter and freq. Switch Allis Chalmer Type 210	5	.01294	2.5
7	Governor Control Switch Allis Chalmer Type 210	5	.01294	1.0
8	Selector Switch Allis Chalmer Type 210	7	.0181159	3.5
9	Voltage Control Switch Allis Chalmer Type 210	5	.01294	1.0

Table 1: Instruments included in the model of the generator switchgear cubicle (Fig. 1)

Location # in Figure 2	Instrument	Weight (lb)	Mass (slugs/12)	Eccentricity (in.)
1	Frequency meter Westinghouse S/M291B	5	.01294	1.25
2	AC Voltmeter	5	.01294	2.5
3	AC Ammeter	5	.01294	1.25
4	Kilowatt/Kilovar Meter	5	.01294	1.25
5	Differential Overcurrent Relays	12	.0310559	3.0
6	Differential Overcurrent Relays	12	.0310559	3.0
7	Differential Overcurrent Relays	12	.0310559	3.0
8	Annunciator Reset Pushbutton & others	12	.0310559	3.0
9	Auxiliary Unit Switch & others	6	.015528	3.0
10	Recepticle	2	.005176	1.0
11	Single Pole Switch	1	.002588	1.0
12	Voltage Control Switch	5	.01294	2.5
13	Governor Control Switch	7	.0181159	3.5
14	Lockout Relay	7	.0181159	3.5
15	Relay (not in use)	7	.0181159	3.5
16	Switch (not in use)	5	.01294	1.0
.7	Under Voltage Relays	10	.0258799	3.0
18	Under Voltage Relays	10	.0258799	3.0
19	Under Voltage Relays	10	.0258799	3.0
20	Watthour Meters	26	.0672878	3.0
21	Watthour Meters	26	.0672878	3.0
22	Terminal boards	15	.0388199	1.25
23	Phase shift transformer	20	.0517958	1.5
24	Frequency transducer	30	.0776398	2.5

Table 2: Instruments included in the model of Auxiliary Switchgear Cubicle 1A (Figure 2)

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Approved by Date

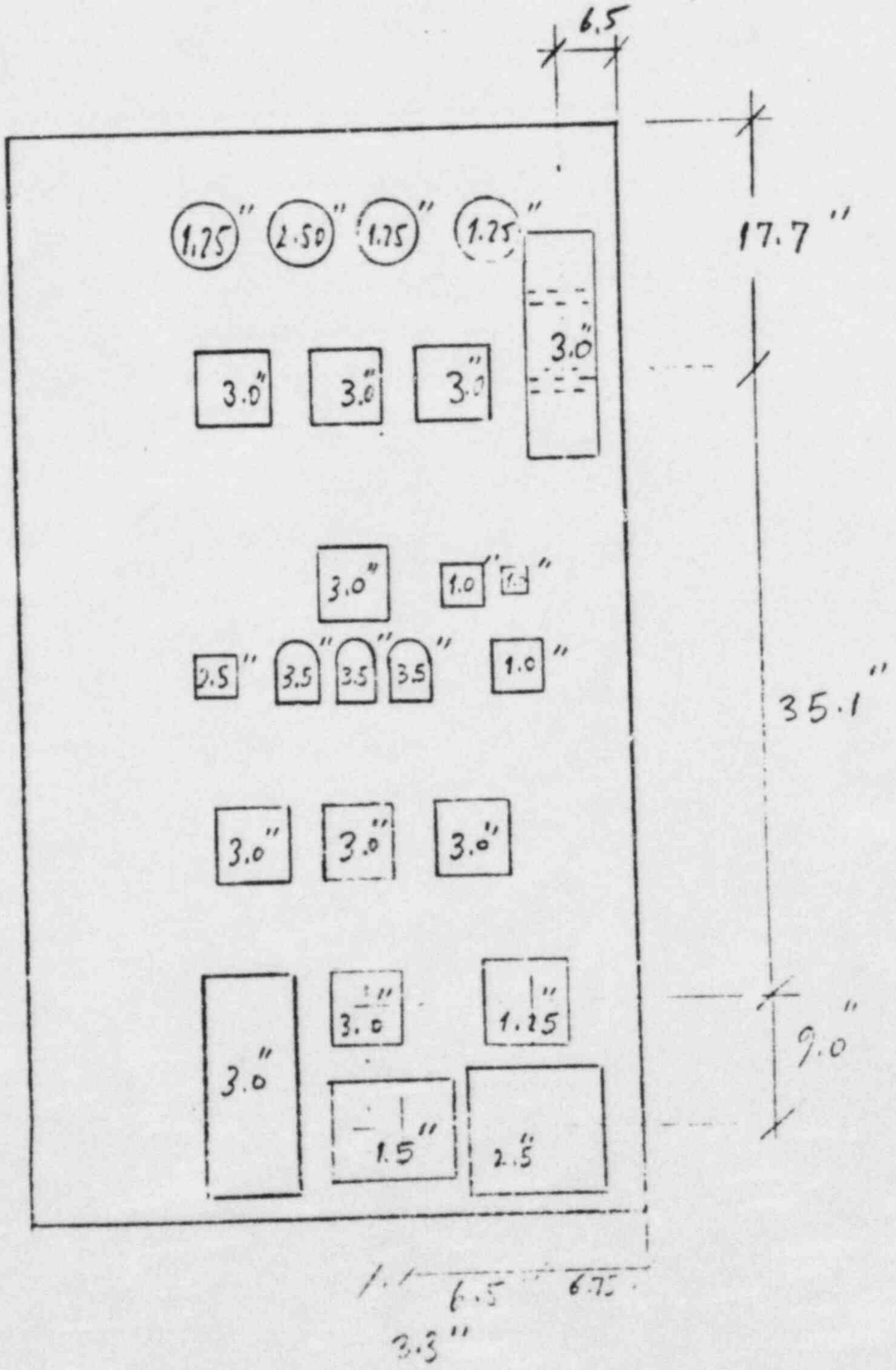

Figure 3 - Cubicle 1A Door, Instrument Eccentricities
(out of plane, in inches, marked on location)

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Project		Reviewed by	Date
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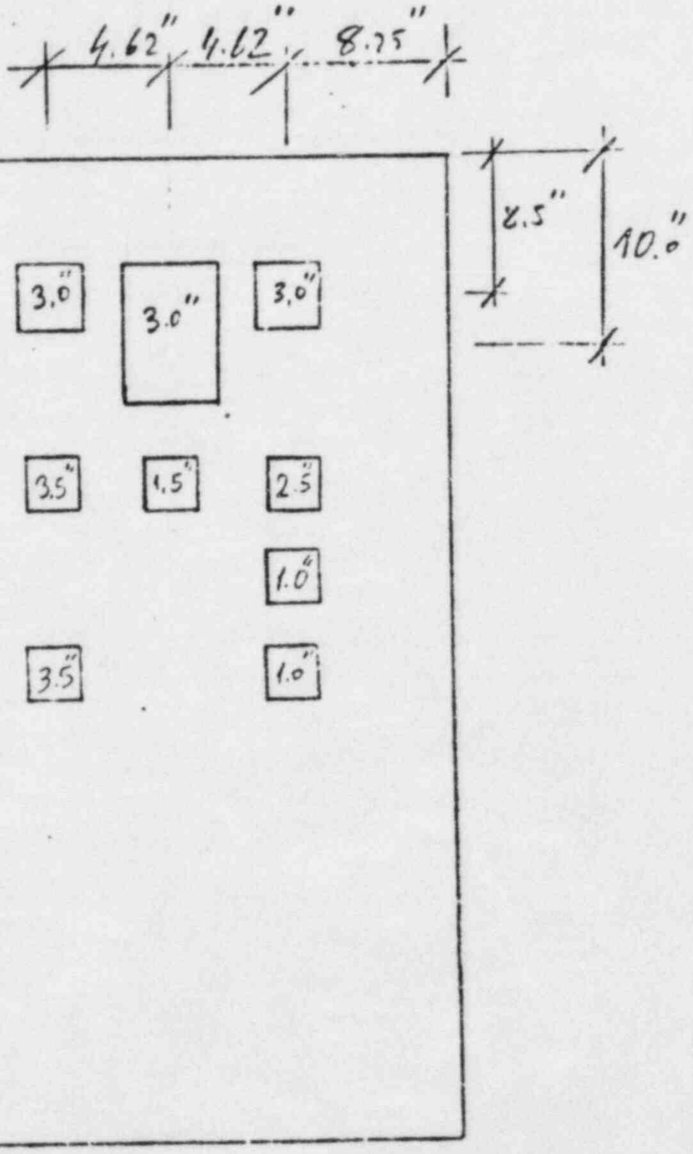

Figure 4 - Cubicle 1 , Door Panel, Instrument Eccentricities
(out of plane, in inches, marked on location)

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PROJECT NO. 6995-00
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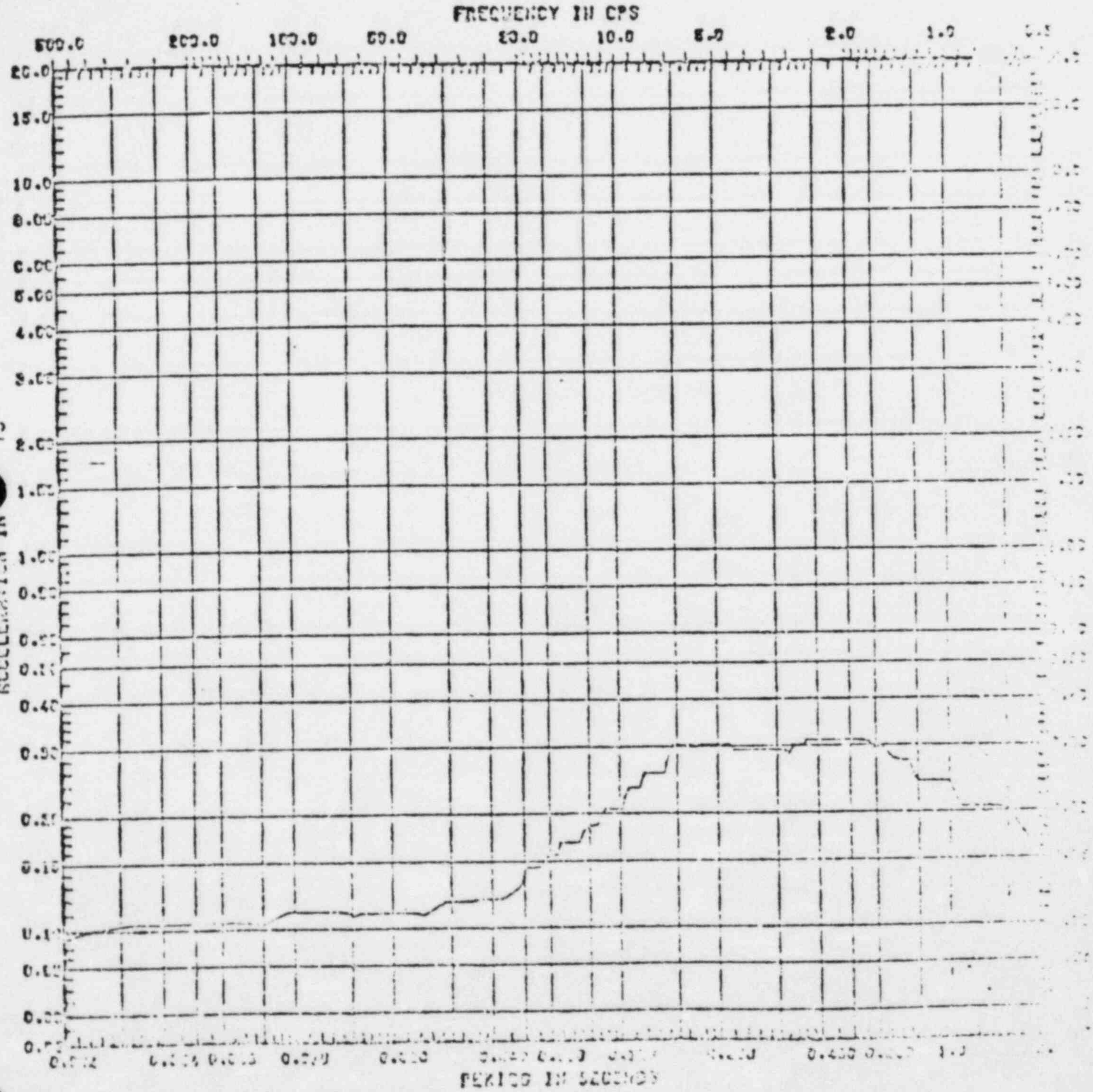
PROJ. NO. 6995-00
PAGE C.1.10

FIG. 5



LOADING:SSE(2%) EDGS
MODE BASE

SPECTRA NO. SRC 2%
ELEVATION GROUND

Results:

Modal analyses were performed to extract the first ten natural frequencies of the Auxiliary Switchgear Cubicle panel (Run ID S98SH) and the Diesel Generator Switchgear Cubicle panel (Run ID S99SH) using SLSAP (Reference 3). The lowest natural frequency of the Auxiliary Switchgear Cubicle panel was 3.352 Hz. This panel had nine natural frequencies below 25 Hz. The lowest natural frequency of the Diesel Generator Switchgear Cubicle panel was 13.34 Hz. Three natural frequencies were found below 25 Hz. The corresponding modes are shown in Figures 6, 7 and 8.

The Auxiliary Switchgear Cubicle panel model was reinforced by simulating L2x2x1/4 angle attached to the front of the panel along the sides, top, across the panel at points 6 inches above the bottom, 20.5 inches below the top, and 36.5 inches below the top. The section of the model below the bottom cross brace was neglected for modelling economy since no instruments are mounted there. The reinforced panel had a lowest natural frequency of 12.48 Hz. (Run ID S02SH). Three natural frequencies were calculated below 25 Hz. The first five modes of the reinforced panel are illustrated in Figures 9,10,11,12 and 13. The reinforced Auxiliary Switchgear Cubicle panel is slightly more flexible, larger, and supports heavier instruments than the Diesel Generator Cubicle panel. On this basis, the reinforced Auxiliary Switchgear Cubicle panel was chosen to obtain the maximum seismic response spectra for panel mounted instruments and for the response spectrum analysis to determine the maximum panel stress.

The response spectrum analysis of the Auxiliary Switchgear Cubicle panel was performed using the first ten modes of the reinforced auxiliary switchgear cubicle panel. The frequency range of these modes was 12.4Hz to 41.94Hz. The maximum stress in the panel was 990 lb/in^2 in the lower section of the panel. (Stress computation included as section 3) The maximum stress in a reinforcing beam was 1222 lb/in^2 in the vertical reinforcement on the bolted side of the panel below the lower bolt. The maximum plate deflection of .033" occurs in the lower section of the panel at location 21 of Figure 2.

A table listing the computer runs made in connection with these analyses and copies of the computer output in either microfiche or hard copy form is included in Section 4.

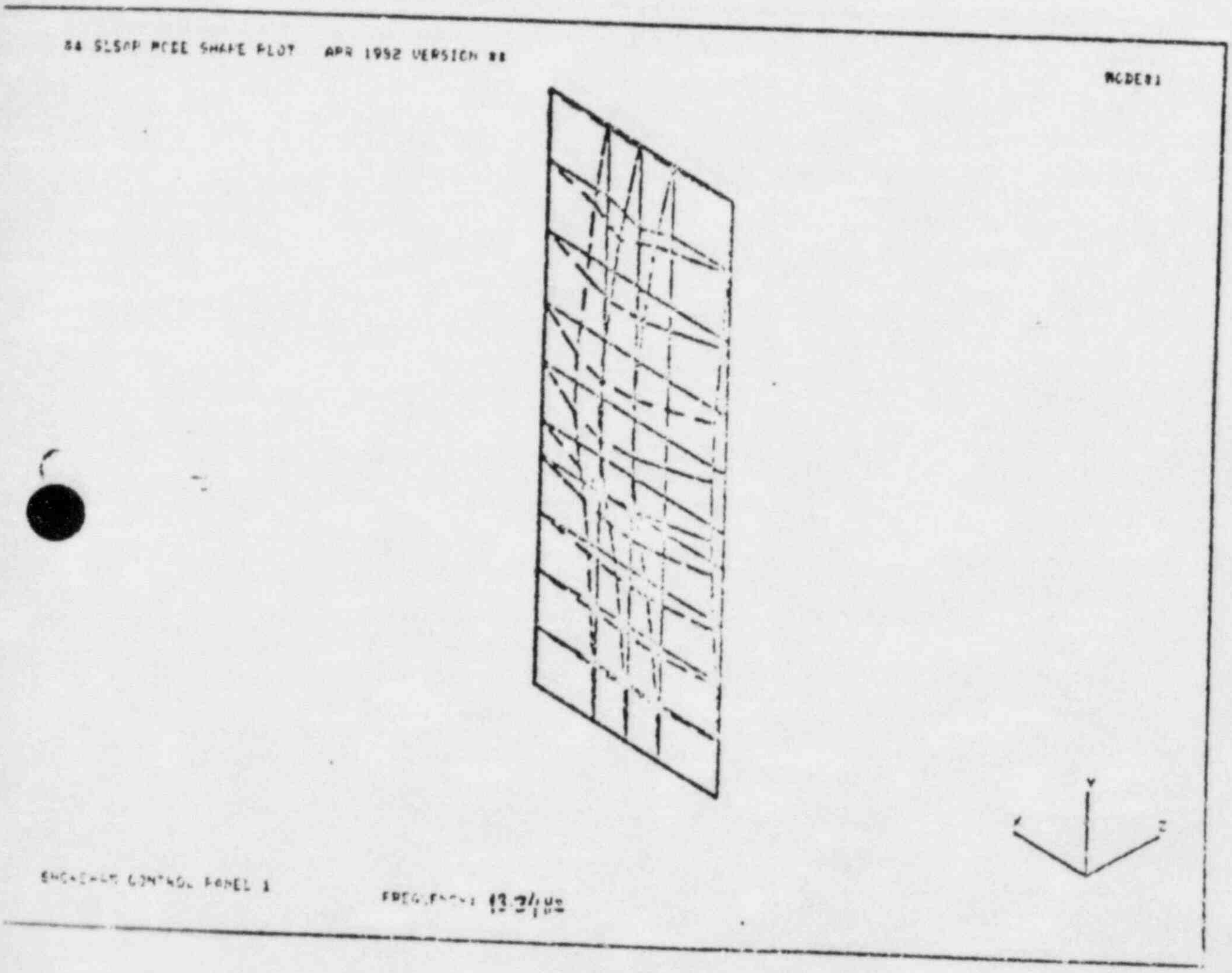


Figure 6 - First Mode of the Diesel Generator Switchgear Cubicle Panel

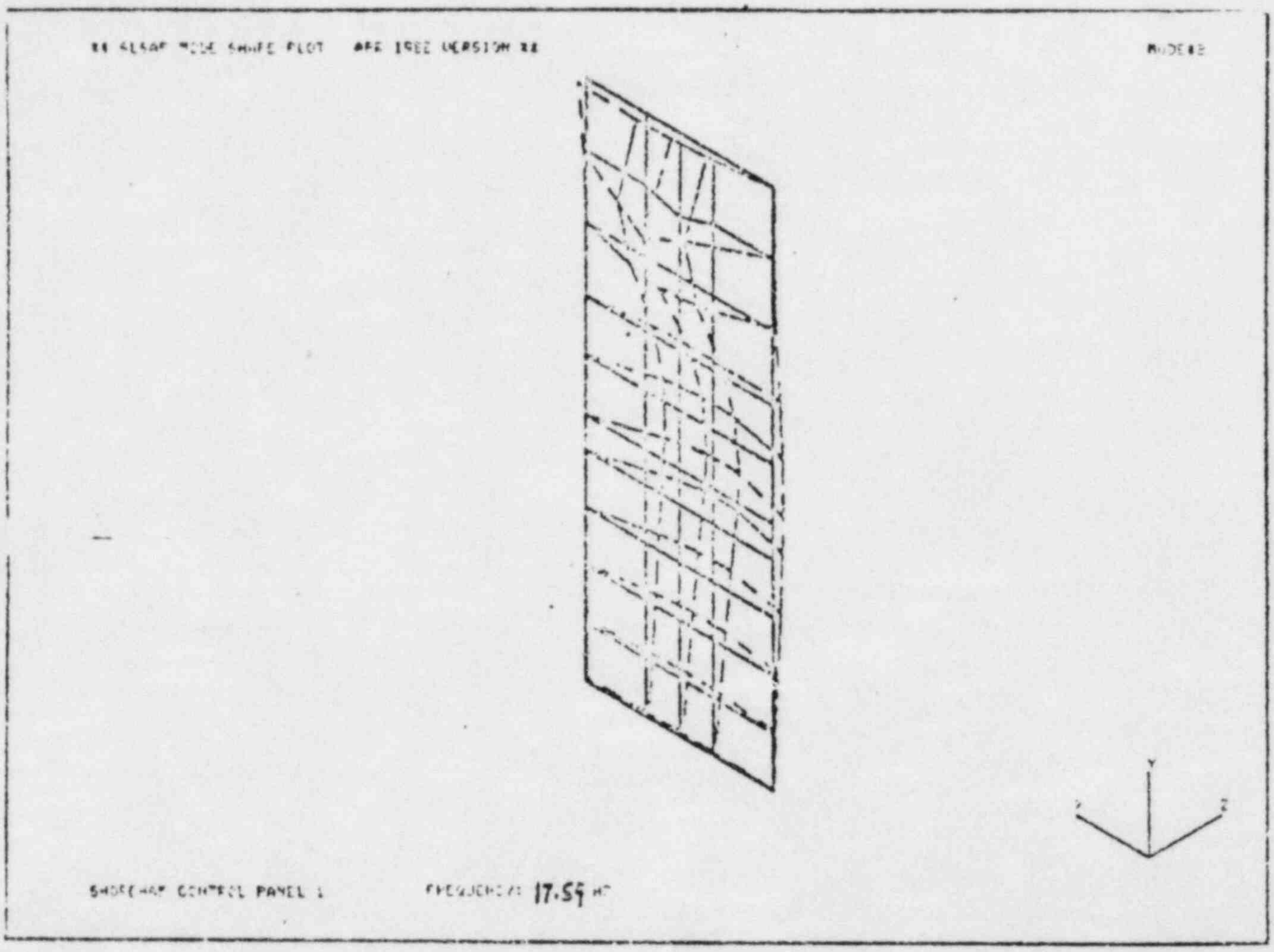


Figure 7 - Second Mode of the Diesel Generator Switchgear Cubicle Panel

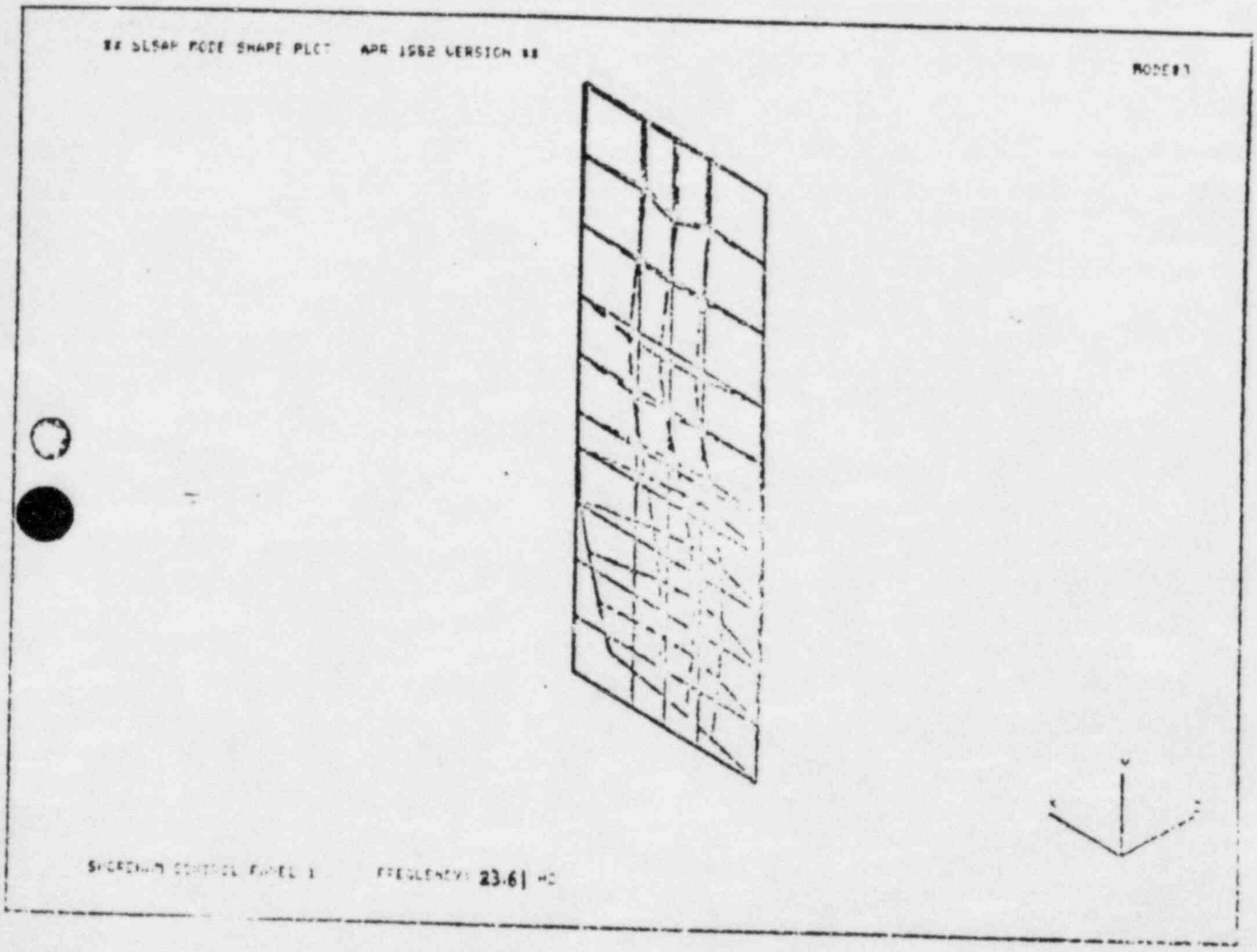


Figure 8 - Third Mode of the Diesel Generator Switchgear Cubicle Panel

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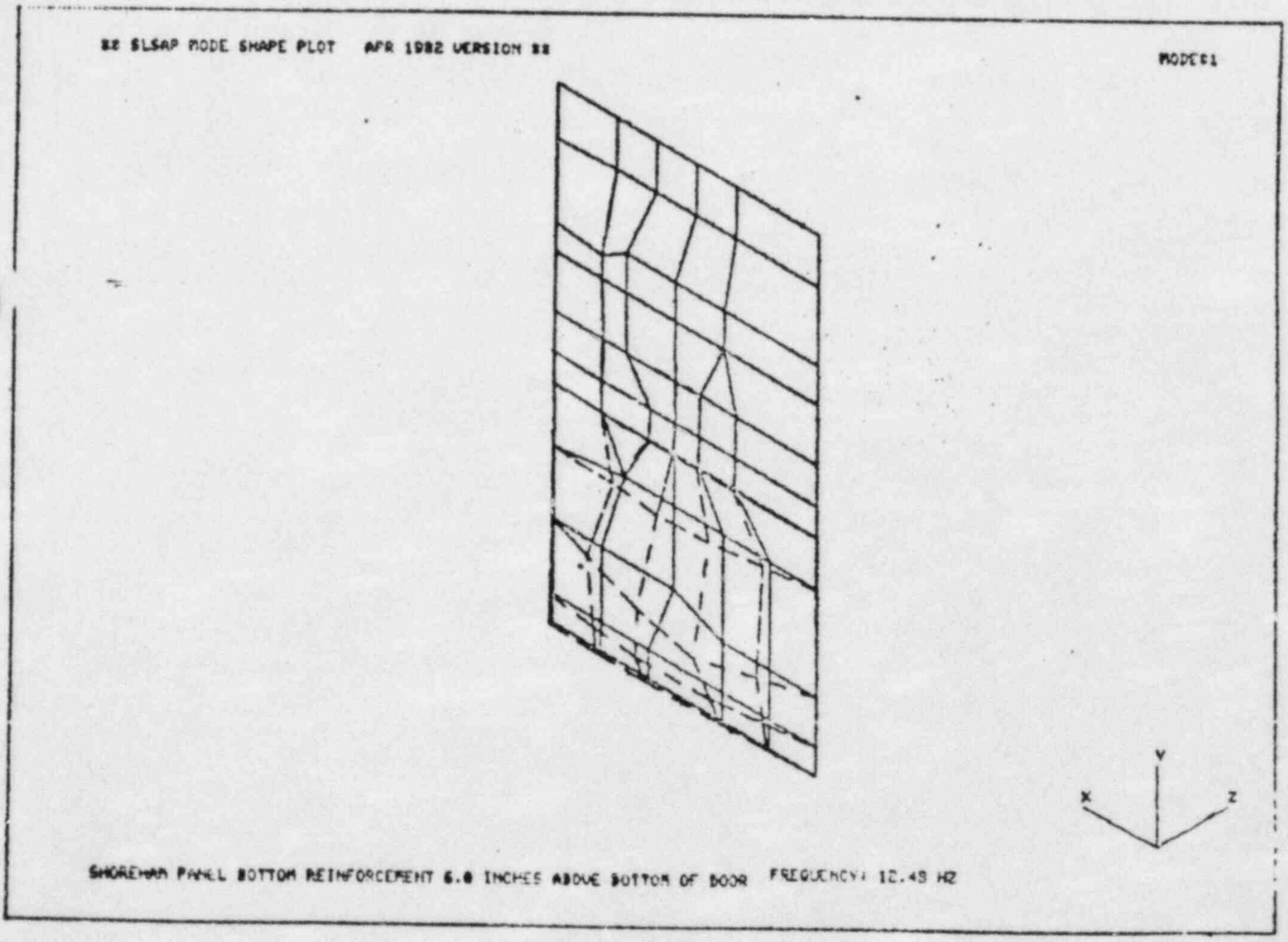
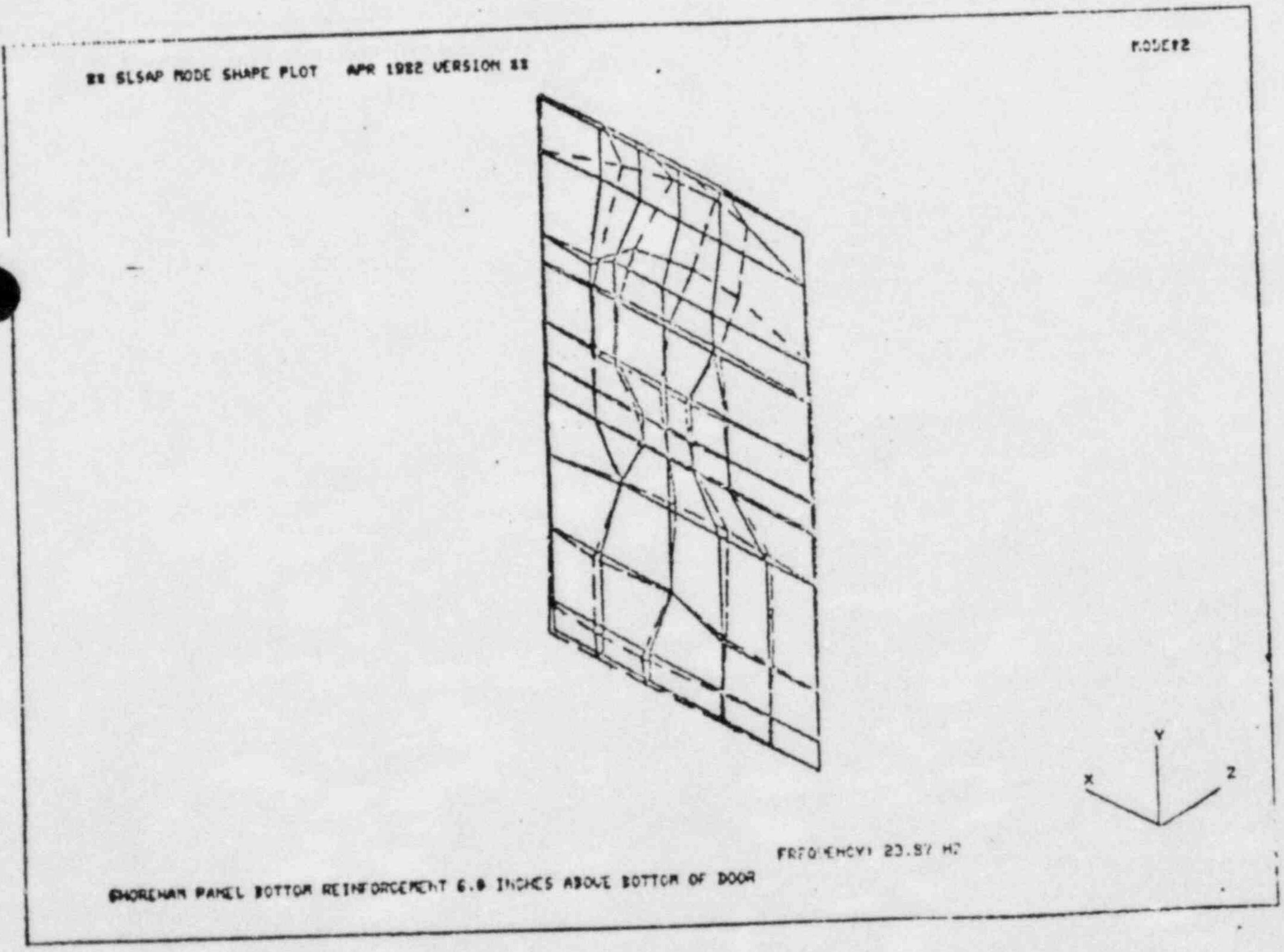
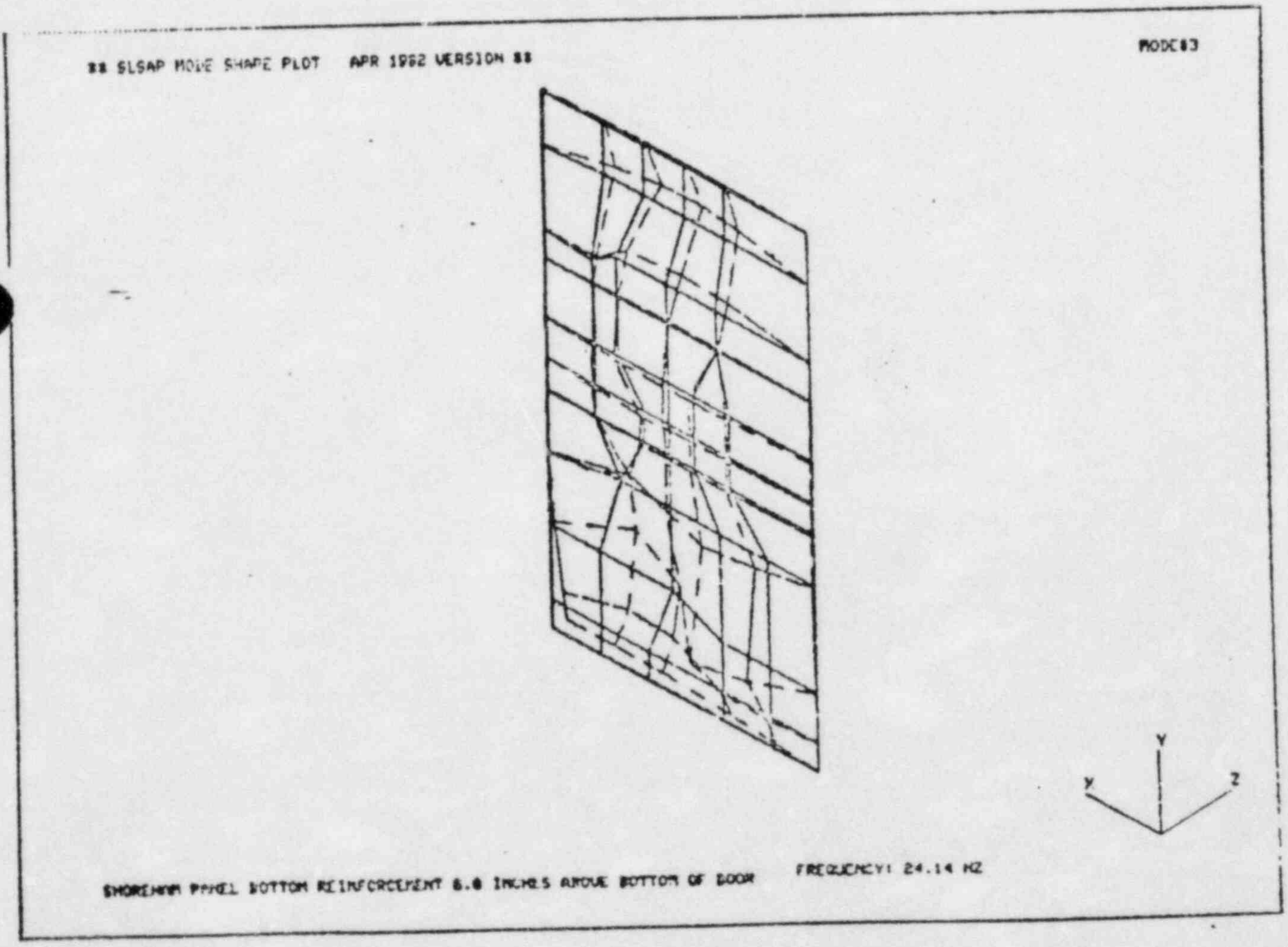


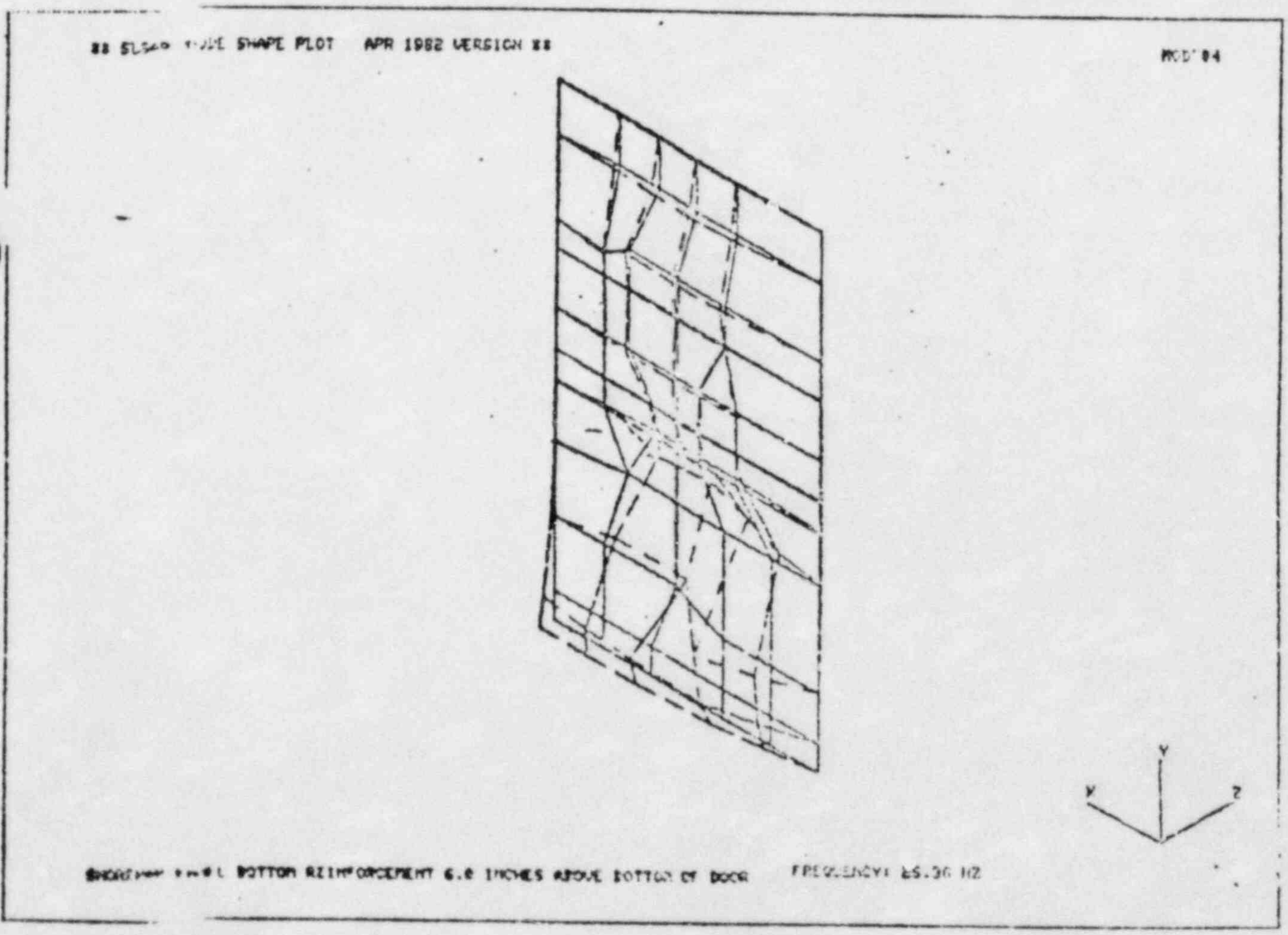
Figure 9 - First Mode of the Reinforced Auxiliary Switchgear Cubicle Panel



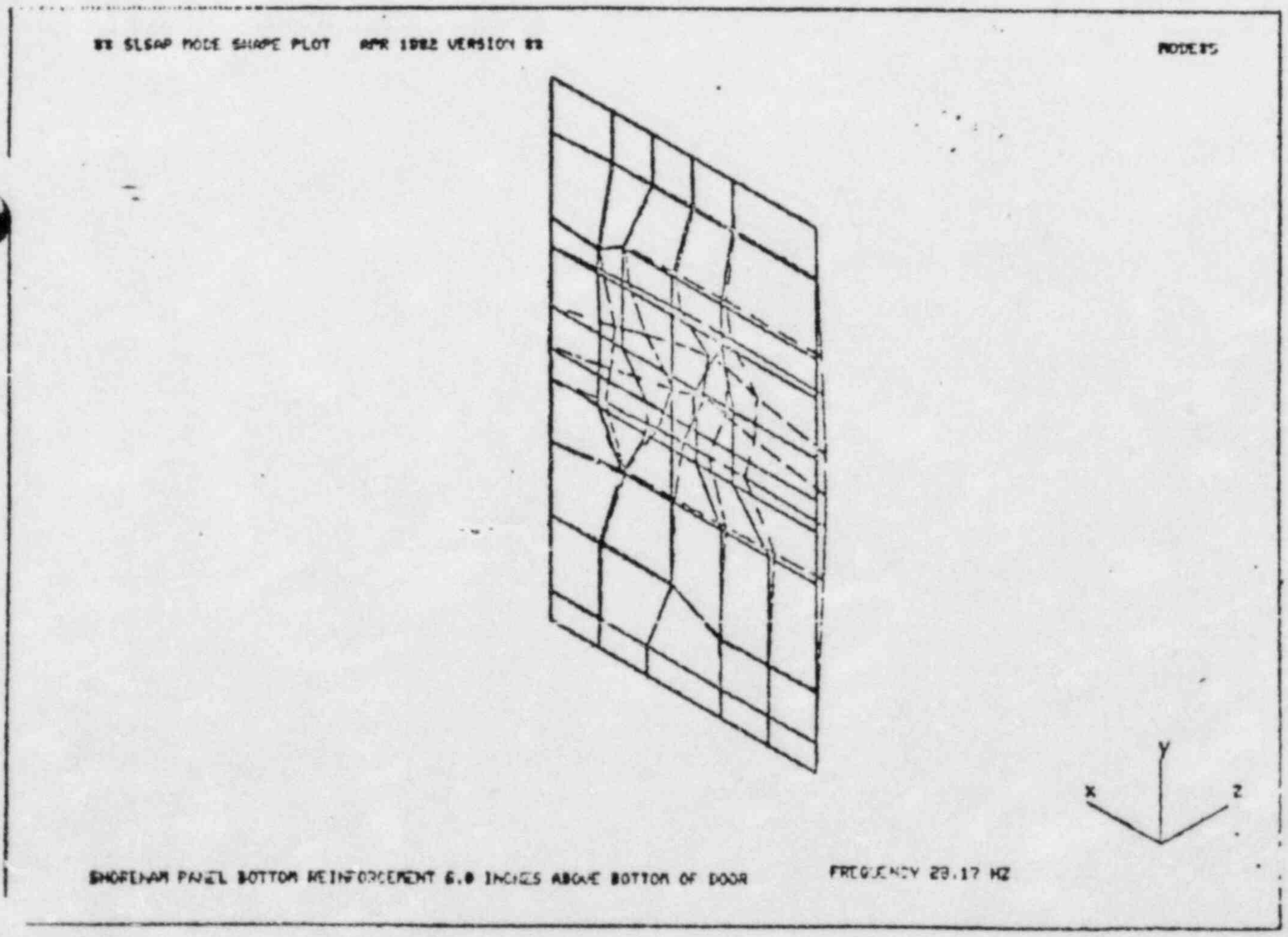
gure 10 - Second Mode of the Reinforced Auxiliary Switchgear Cubicle Panel



gure 11 - Third Mode of the Reinforced Auxiliary Switchgear Cubicle Panel



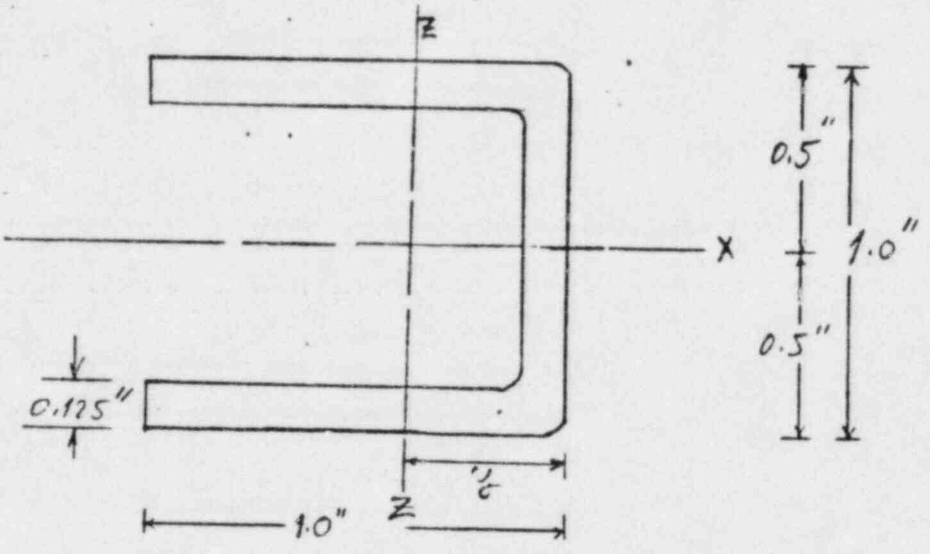
re 12 - Fourth Mode of the Reinforced Auxiliary Switchgear Cubicle Panel



gure 13 - Fifth Mode of the Reinforced Auxiliary Switchgear Cubicle Panel

Section 2: Panel Stiffener Properties

1) Side Stiffeners



Area = $.125x(1+1+.75) = .34375\ in^2$

Shear area, X direction = $.125x(1+1) = .25\ in^2$

Shear area, Z direction = $.125x1 = .125\ in^2$

Torsional inertia, $J = \frac{.125}{3}^{3} x\ (1+1+.75) = .0017904\ in^4$

Flexural Inertia about the XX axis

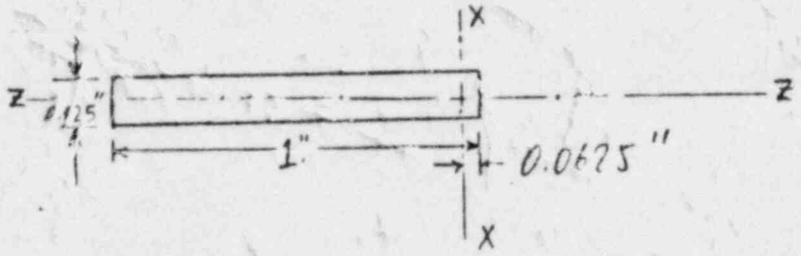
$$I_{xx} = 2\ (\frac{1x.125^3}{12} + .125x1x(.4375)^2) + \frac{.125x(.75)^3}{12} = .524\ in^4$$

Flexural Inertia about the Z-Z axis

$$y = \frac{2x(.125x1x.5)+.75x.125x.0625}{.125x2.75} = .3806818\ in.$$

$$I_{zz} = 2(\frac{.125(1)^3}{12} + .125x1(.5-.3806818)^2) + \frac{.75(.125)^3}{12} + .75x.125x\ (.3806818-.625)^2 = .0340058\ in^4$$

2) Top and bottom stiffeners of the generator switchgear cubicle



Area = .125 x 1 = .125 in^2

$J = \frac{(.125)^3\ 1}{3} = .000651\ in^4$

Flexural Inertia about the X-X axis, I_{xx}

$I_{xx} = \frac{(.125)\ 1^3}{12} + .125 \times 1 \times (.4375)^2 = .0343424\ in^4$

Flexural Inertia about the ZZ axis, I_{zz}

$I_{zz} = \frac{1(.125)^{-3}}{12} = .0001628\ in^4$

Section 3: Plate and Beam Stress Computation

Run ID: S05 SH

1) Stress in rolled edge of the cabinet, beam section number 1

Moments of inertia: about axis2, $I_2 = .053$ in^4

about axis3, $I_3 = .034$ in^4

Distance to outer fiber: bending about axis2, $C_2 = .5$ in

bending about axis3, $C_3 = .38$ in

or $C_3 = .62$ in

Axis 2 bending: $C_2/I_2 = 9.434$ in^{-3}

Axis 3 bending (take maximum distance) $C_3/I_3 = 18.235$ in^{-3}

Area = .344 in^2

Maximum Moments $M_2 = 47.05$ in lb at element #6

axial force, P=1.929 lb

$M_3 = 1.195$ in lb element #16

axial force P = 4.89

Maximum stress from M_2 element #6

$$\sigma = \frac{Mc}{I} + {}^{P}/\text{Area} = 448.02 \text{ lb/in}^2$$

2) Stress in reinforcing L2x2x1/4 beams.

Moment of Inertia $I = .348$ in^4
$C = .592$ in
or $C = 1.408$ in
Take maximum distance, $C/I = 4.046$ in^{-3}
Area $A = .938$ in^2

Maximum Moment

$M_2 = 301$ in/lb element 43

$P = 3.897$ lb

$$\sigma = \frac{Mc}{I} + {}^{P}/A = 1222 \text{ lb/in}^2$$

3) Stress in panel plates

$$\sigma = \frac{6M}{t^2} + S \qquad {}^{6}/t^2 = \frac{6}{(.125)^2} = 384 \text{ in}^{-2}$$

3) Cont'd

$$\sigma_{max} = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\frac{(\sigma_x - \sigma_y)^2}{2} + \tau^2}$$

element 8 M_{xx} = 1.849 lbin/in M_{yy} = 2.488 lbin/in

S_{xx} = 3.973 lb/in^2 S_{yy} = .6294 lb/in^2

S_{xy} = 1.147 lb/in^2

σ_x = 713.989 lb/in^2 σ_y = 956.02 lb/in^2 τ = 1.147 lb/in^2

σ_{max} = 956.03 lb/in^2

element 13 M_{xx} = 2.305 lbin/in M_{yy} = 2.403 lbin/in

S_{xx} = .4651 lb/in^2 S_{yy} = 2.726 lb/in^2

S_{xy} = .6629 lb/in^2

σ_x = 885.585 lb/in^2 σ_y = 925.478 lb/in^2

τ = .6629 lb/in^2

σ_{max} = 925.489 lb/in^2

element 20 M_{xx} = 2.57 lbin/in M_{yy} = .631 lbin/in

S_{xx} = 2.796 lb/in^2 S_{yy} = 2.206 lb/in^2

S_{xy} = 2.974 lb/in^2

σ_x = 989.68 lb/in^2 σ_y = 244.51 lb/in^2

τ = 2.974 lb/in^2

σ_{max} = 989.69 lb/in^2

Section 4: COMPUTER OUTPUT

Run ID	Date	Description	Fiche Original No.
S98SH	5/9/84	Auxiliary Switchgear Cubicle panel, modal analysis	Hard copy
S99SH	5/10/84	Diesel Generator Switchgear Cubicle panel, modal analysis	001
S02SH	5/11/84	Auxiliary Switchgear Cubicle panel, reinforced panel modal analysis	001
S05SH	5/17/84	Auxiliary Switchgear Cubicle panel, response spectrum analysis	001

APPENDIX C.1(b)

TIME HISTORY ANALYSIS OF MODIFIED SWITCHGEAR

CUBICLE 1A DOOR PANEL

Purpose:

The objective of this analysis is to use the base time history provided by LILCo in conjunction with the finite element model of the selected panel (Cubicle 1A door panel in switchgear housing) and develop elevated response spectra at instrument locations.

Introduction:

Using the available documents together with the on-site walkdown results, three categories of panels are identified as the most flexible structural components that may potentially result in high levels of elevated response spectra. These categories are:

- i) Switchgear, cubicle 1A door panel
- ii) Switchgear cubicles 1, 2, 3 & 4 door panels (identical)
- iii) Generator and governor control panels (located in unit housings)

Considering support framing, edge fixity conditions and panel, sub-panel plate thickness, weight distribution and overall dimensions; the panels mentioned in the third category above are excluded from detailed analyses presently due to their relatively high rigidity. Panels mentioned in the first two categories are much more flexible and were subjected to detailed dynamic analyses for comparison purposes as mentioned earlier. Detailed finite element analyses performed on cubicle 1A door panel and cubicle 1 door panel indicated that the cubicle 1A door panel was the most flexible structure in spite of the additional framing and therefore, was selected for the generation of the elevated response spectra through time history analysis.

Analysis

Same finite element model that was used for eigenvalue extraction is also used for the time history analysis of the cubicle 1A door panel. The time history record supplied by LILCo is converted to various load cases by multiplying the acceleration ordinates with the following factors:

			Factor
0BE:			
	Horizontal	-	1
	Vertical	-	2/3
SSE:			
	Horizontal	-	2
	Vertical	-	4/3

SLSAP structural analysis program with the NDYN = 4 option is used for analysis.

Two sets of analyses were performed, one for OBE loading and the other for SSE loading. In both cases, ground motion (acceleration time histories) are applied simultaneously in two horizontal and and vertical directions.

Wilson Theta integration algarithm is applied for the analyses. Two percent damping ratio for OBE and three percent damping ratio for SSE load cases were selected. Damping matrix coefficients for the global mass and stiffness matrices are computed as follows:

$$\lambda_i = \frac{\alpha}{wi} + \frac{wi}{2}$$

where:

λ_i = Damping ratio

α = Damping coefficient for global mass matrix

β = Damping coefficient for global stiffness matrix

Wi = Natural frequency

For Wilson Theta algarithm, uniform damping ratio is used.

Therefore, for OBE loading, the following are given:

OBE: $\lambda_{1,2}$ = 0.02 (Uniform damping ratio)

W1 = 78.42 rad/sec. (12.48 cps)

W2 = 177.00 rad/sec. (28.17 cps)

$$- \alpha = \frac{2 \times 78.42 \times 177.00}{(78.42)^2 - (177.00)^2} (0.02 \times 78.42 - 0.02 \times 177.00)$$

α = 2.174

$$\beta = \frac{2}{(78.42)^2 - (177.00)^2} (0.02 \times 78.42 - 0.02 \times 177.00)$$

β = 0.000157

and, for SSE loading:

SSE: $\lambda_{1,2}$ = 0.03

W1 = 78.42 rad/sec. (12.48 cps)

W2 = 117.00 rad/sec. (28.17 cps)

and using same equations:

α = 3.261

β = 0.000235

Selected integration time step was 0.004 sec. This step will guarantee at least an eight point representation of 30 Hz signal which is the upper bound frequency of interest per provided input data.

Response acceleration time histories at each node of the model are saved on permanent disk files as follows:

- OBE load case save file: ITK*SH00BE23.
- SSE load case save file: ITK*SH0SSE23.

Maximum absolute acceleration responses at each node are tabulated in the following pages for OBE and SSE respectively.

```
ACCELERATION MAXIMUM ABSOLUTE VALUES  — OBE
                         (UNIT . g)
NODE   TIME       X-        TIME       Y-        TIME       Z-        TIME       XX-       TIME       YY-       TIME   ZZ-
      (SEC.)
  1  .4128+01  .9954-01  .4128+01  .6636-01  .4128+01  .9954-01  0000      .0000     .5444+01  .1319-02  0000   0000
  2  .4128+01  .9957-01  .4128+01  .6638-01  .4128+01  .9920-01  .5444+01  .4003-02  .5444+01  .1171-02  0000   0000
  3  .4128+01  .9958-01  .4128+01  .6639-01  .4120+01  .9679-01  .5404+01  .7348-02  .6354+01  .8432-03  0000   0000
  4  .4128+01  .9960-01  .4128+01  .6640-01  .4120+01  .9798-01  .5448+01  .7629-02  .8884+01  .3026-02  0000   0000
  5  .4128+01  .9960-01  .4128+01  .6639-01  .4128+01  .9747-01  .5448+01  .4026-02  .5404+01  .3779-03  0000   0000
  6  .4128+01  .9960-01  .4128+01  .6638-01  .4128+01  .9707-01  .4192+01  .1090-02  .5448+01  .5054-03  0000   0000
  7  .4128+01  .9957-01  .4128+01  .6637-01  .4128+01  .9942-01  .3188+01  .2601-02  .5404+01  .2637-02  0000   0000
  8  .4128+01  .9958-01  .4128+01  .6638-01  .4128+01  .1004+00  .5444+01  .5458-02  .5444+01  .6460-02  0000   0000
  9  .4128+01  .9955-01  .4128+01  .6639-01  .4136+01  .1032+00  .5404+01  .1378-01  .5444+01  .2527-02  0000   0000
 10  .4128+01  .9960-01  .4128+01  .6639-01  .4136+01  .1052+00  .5404+01  .1326-01  .5408+01  .2248-02  0000   0000
 11  .4128+01  .9961-01  .4128+01  .6638-01  .4124+01  .1011+00  .5448+01  .5263-02  .5448+01  .4593-02  0000   0000
 12  .4128+01  .9961-01  .4128+01  .6638-01  .4128+01  .9772-01  .4192+01  .1098-02  .5448+01  .1909-02  0000   0000
 13  .4128+01  .9959-01  .4128+01  .6636-01  .4128+01  .9872-01  .3168+01  .5729-03  .5444+01  .4352-02  0000   0000
 14  .4128+01  .9960-01  .4128+01  .6638-01  .4132+01  .1019+00  .5448+01  .2948-02  .5444+01  .1134-01  0000   0000
 15  .4128+01  .9960-01  .4128+01  .6639-01  .5488+01  .1416+00  .5408+01  .7613-02  .5404+01  .1028-01  0000   0000
 16  .4128+01  .9961-01  .4128+01  .6640-01  .5404+01  .1698+00  .5488+01  .3225-02  .5408+01  .4853-02  0000   0000
 17  .4128+01  .9960-01  .4128+01  .6638-01  .4132+01  .1094+00  .5756+01  .1466-02  .5404+01  .1187-01  0000   0000
 18  .4128+01  .9960-01  .4128+01  .6637-01  .4128+01  .9308-01  .4192+01  .8220-03  .5448+01  .3273-02  0000   0000
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 20  .4128+01  .9961-01  .4128+01  .6630-01  .4136+01  .9862-01  .5404+01  .5199-02  .5370+01  .5257-02  0000   0000
 21  .4128+01  .9961-01  .4128+01  .6639-01  .4140+01  .1125+00  .5404+01  .1088-01  .5210+01  .4663-02  0000   0000
 22  .4128+01  .9960-01  .4128+01  .6640-01  .7116+01  .1187+00  .5404+01  .1322-01  .7240+01  .1820-02  0000   0000
 23  .4128+01  .9959-01  .4128+01  .6639-01  .4128+01  .1051+00  .5404+01  .6667-02  .5488+01  .5904-02  0000   0000
 24  .4128+01  .9954-01  .4128+01  .6636-01  .4128+01  .9954-01  0000      0000      0000      0000      0000   0000
 25  .4128+01  .9961-01  .4128+01  .6639-01  .4128+01  .9570-01  .3228+01  .2673-03  .5440+01  .6570-03  0000   0000
 26  .4128+01  .9951-01  .4128+01  .6640-01  .4116+01  .9687-01  .4956+01  .4048-02  .4952+01  .2566-03  0000   0000
 27  .4128+01  .9961-01  .4128+01  .6640-01  .4116+01  .9672-01  .4896+01  .5667-02  .3168+01  .7321-03  0000   0000
 28  .4128+01  .9960-01  .4128+01  .6640-01  .4116+01  .9617-01  .4856+01  .6042-02  .3168+01  .5421-03  0000   0000
 29  .4128+01  .9960-01  .4128+01  .6640-01  .4116+01  .9510-01  .4896+01  .5611-02  .3168+01  .8341-03  0000   0000
 30  .4128+01  .9960-01  .4128+01  .6629-01  .4128+01  .9691-01  .4956+01  .2659-02  .3168+01  .1242-02  0000   0000
 31  .4128+01  .9960-01  .4128+01  .6637-01  .4128+01  .9949-01  .3224+01  .7354-04  .3168+01  .1380-02  0000   0000
 32  .4128+01  .9961-01  .4128+01  .6639-01  .4128+01  .9566-01  .5412+01  .1351-03  .3172+01  .9766-03  0000   0000
 33  .4128+01  .9961-01  .4128+01  .6640-01  .4116+01  .1007+00  .4952+01  .3443-02  .4992+01  .1971-02  0000   0000
 34  .4128 01  .9961-01  .4128+01  .6640-01  .4120+01  .1034+00  .4992+01  .7046-02  .3192+01  .2003-02  0000   0000
 35  .4128+01  .9961-01  .4128+01  .6640-01  .4120+01  .1039+00  .4997+01  .7941-02  .3248+01  .9679-03  0000   0000
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 38  .4128+01  .9960-01  .4128+01  .6637-01  .4128+01  .9949-01  .2524+01  .2026-04  .3168+01  .1482-02  0000   0000
 39  .4128+01  .9961-01  .4128+01  .6638-01  .4128+01  .9601-01  .3168+01  .2911-03  .3172+01  .8535-03  0000   0000
 40  .4128+01  .9961-01  .4128+01  .6640-01  .4116+01  .1011+00  .3172+01  .1967-02  .4992+01  .2439-02  0000   0000
 41  .4128+01  .9961-01  .4128+01  .6640-01  .4120+01  .1047+00  .3172+01  .3622-02  .4996+01  .2161-02  0000   0000
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 47  .4128+01  .9950      .4128+01  .6640-01  .4128+01  .9849-01  .6912+01  .3329-02  .5412+01  .2222-03  0000   0000
 48  .4128+01  .9960-01  .4128+01  .6640-01  .4128+01  .9903-01  .6912+01  .3864-02  .3228+01  .4009-03  0000   0000
 49  .4128+01  .9959-01  .4128+01  .6639-01  .4128+01  .9934-01  .6912+01  .2626-02  .3228+01  .7960-03  0000   0000
 50  .4128+01  .9958-01  .4128+01  .6638-01  .4128+01  .9863-01  .6912+01  .1762-02  .3228+01  .9237-03  0000   0000
 51  .4128+01  .9956-01  .4128+01  .6637-01  .4128+01  .9954-01  .3224+01  .6419-04  .3228+01  .9934-03  0000   0000
 52  .4128+01  .9960-01  .4128+01  .6638-01  .4128+01  .9793-01  .3188+01  .6865-03  .6908+01  .1430-02  0000   0000
 53  .4128+01  .9960-01  .4128+01  .6640-01  .4128+01  .1085+00  .6912+01  .5960-02  .6912+01  .2676-02  0000   0000
```

SARGENT LUNDY

```
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TIME ELAPSED FROM THE BEGINING OF TIME MARCHING SOLUTION IS        .411+03
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ENTIRE TIME HISTORY AT EACH NODE
SAVED ON FILE :

ITK * SH00BE23.

(NTRAN FILE)

CALC NO. COD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.29

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                         (UNIT : g)

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  7   .4128+01  .1981+00  .4128+01  .1327+00  .4128+01  .1989+00  .3168+01  .4659-03  .5444+01  .4155-02  .0000     .0000
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SARGENT & LUNDY

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TIME ELAPSED FROM THE BEGINING OF TIME MARCHING SOLUTION IS        .411+03
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ENTIRE TIME HISTORY AT EACH NODE

SAVED ON FILE :

ITK * SHØSSE23.

(NTRAN FILE)

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.30

The NTRAN files are processed by program DATATRAN (S&L Program No.: DAT097193100) with option THXTRK to extract the saved time history records at selected nodes appropriately formatted for response spectra generation.

Acceleration time history records at nodes 15, 16 and 60 are extracted in all three directions for further use. These nodes represent the maximum response points on the structure and decision is based on the evaluation of maximum response values as well as the mode shape plots presented earlier.

The elevated response spectra curves are developed through the use of program RSG (S&L Program No.: RSG097050000) with Section 4.2 options. Extracted time history records are input individually along each direction and response spectra curves are obtained. A total of 80 separate periods are used for the computation. These periods include the standard 76 distinct periods suggested by NRC. Plots of the individual spectra are presented in the following pages for reference. Each curve is identified per load case, node number and direction. Also provided are two time history plots at node 16 for OBE and SSE load cases in Z direction (max. response direction) as examples of generated time histories.

Response spectra analysis results at various nodes indicate that no amplification in X and Y direction takes place. In other words, along the horizontal direction in the plane of panel, the ground response spectra is preserved as well as in the vertical direction in the same plane. Only significant amplification is observed in the horizontal direction, perpendicular to the plane of the panel (Z-direction). Furthermore, response at node 16 in this direction is the most severe with the exception of the frequency range 20 - 30 Hz. In this frequency range, maximum response is observed at the node point 60 along the Z-direction.

As mentioned previously, envelope elevated response spectra covers all these locations with their maximum responses at any frequency range and can be gereralized for any equipment location conservatively.

The following table provides a list of all the computer Run ID's and dates of the applicable time history analysis and response spectra generation. Microfiche copies of all these runs are included at the end of this report for reference (one each, for CQD library and for LILCo copies).

COMPUTER RUN LIST

Run ID.	Date	Subject	Fiche Orig. No.
SIKP	5/12/84	OBE Time History Analysis	1
SIKX	5/14/84	SSE Time History Analysis	1
SIKA	5/14/84	OBE Node 15-X Response Spectra Gen.	1
SIKB	5/14/84	OBE Node 15-Y Response Spectra Gen.	1
SIKC	5/14/84	OBE Node 15-Z Response Spectra Gen.	1
SIKD	5/14/84	OBE Node 16-X Response Spectra Gen.	1
SIKE	5/14/84	OBE Node 16-Y Response Spectra Gen.	1
SIKF	5/14/84	OBE Node 16-Z Response Spectra Gen.	1
SIKG	5/14/84	OBE Node 60-X Response Spectra Gen.	1
SIKH	5/14/84	OBE Node 60-Y Response Spectra Gen.	1
SIKI	5/14/84	OBE Node 60-Z Response Spectra Gen.	1
SIK0	5/15/84	SSE Node 16-X Response Spectra Gen.	1
SIKP	5/15/84	SSE Node 16-Y Response Spectra Gen.	1
SIKQ	5/15/84	SSE Node 16-Z Response Spectra Gen.	1
SIKS	5/15/84	SSE Node 60-Z Response Spectra Gen.	1

Conclusion:

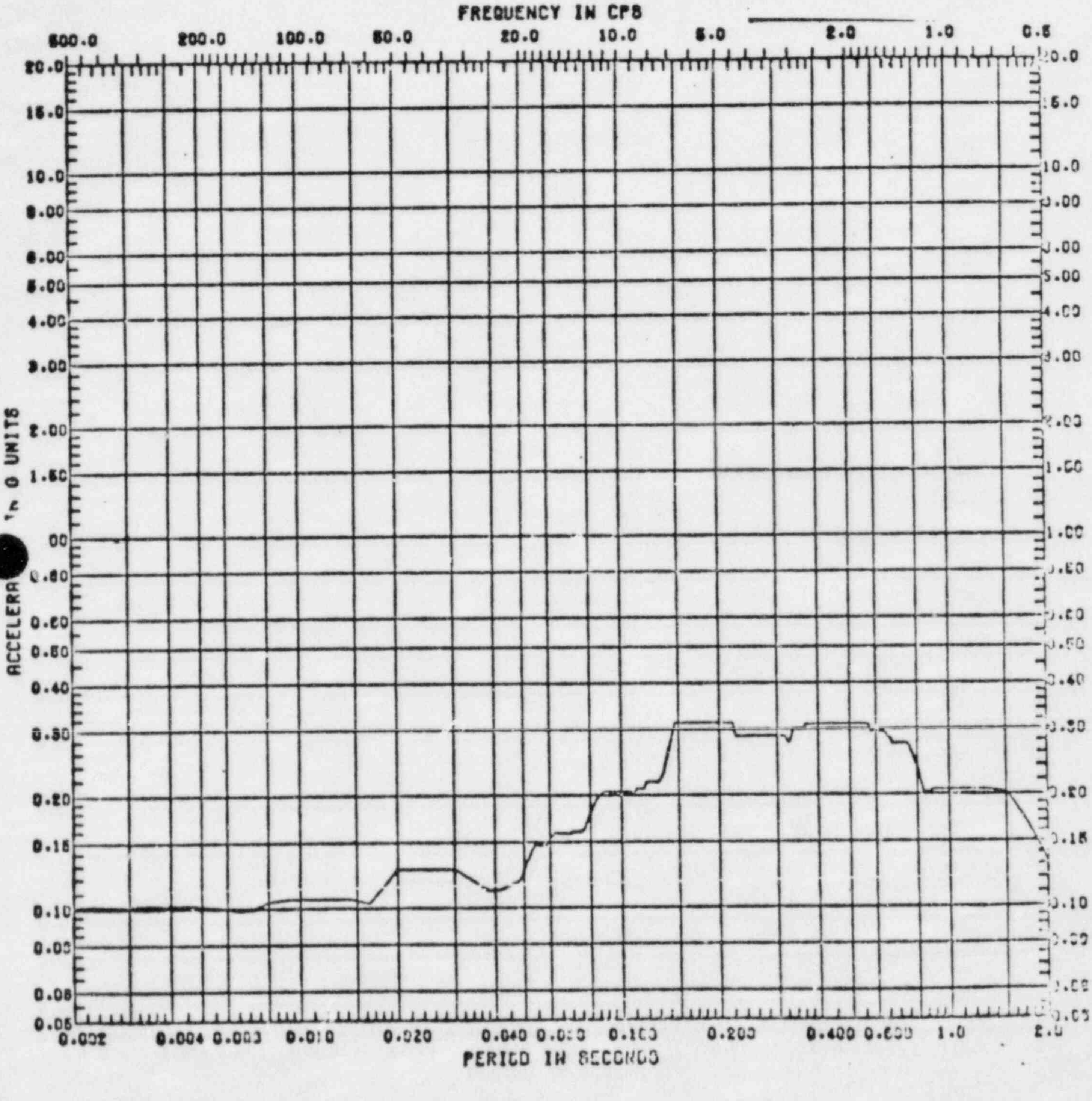
Using direct integration procedure along with the finite element model, elevated response spectra curves for OBE and SSE loads are obtained in vertical and horizontal directions. These spectra represent the envelope of the maximum response locations on the panels for each load case along corresponding directions.

SARGENT & LUNDY
ENGINEERS

14 MAY 84
SIKA

CALC NO. CQD- 014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.020
PAGE

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.35



LOADING:OBE(2%) EDG3
NODE 16-X
DIRECTION HOR ANGLE N/A
SPECTRA NO. OBE 2%
ELEVATION PANEL
LOCATION EDG3

SARGENT & LUNDY
ENGINEERS

14 MAY 84
SIKB

CALC NO. CQD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
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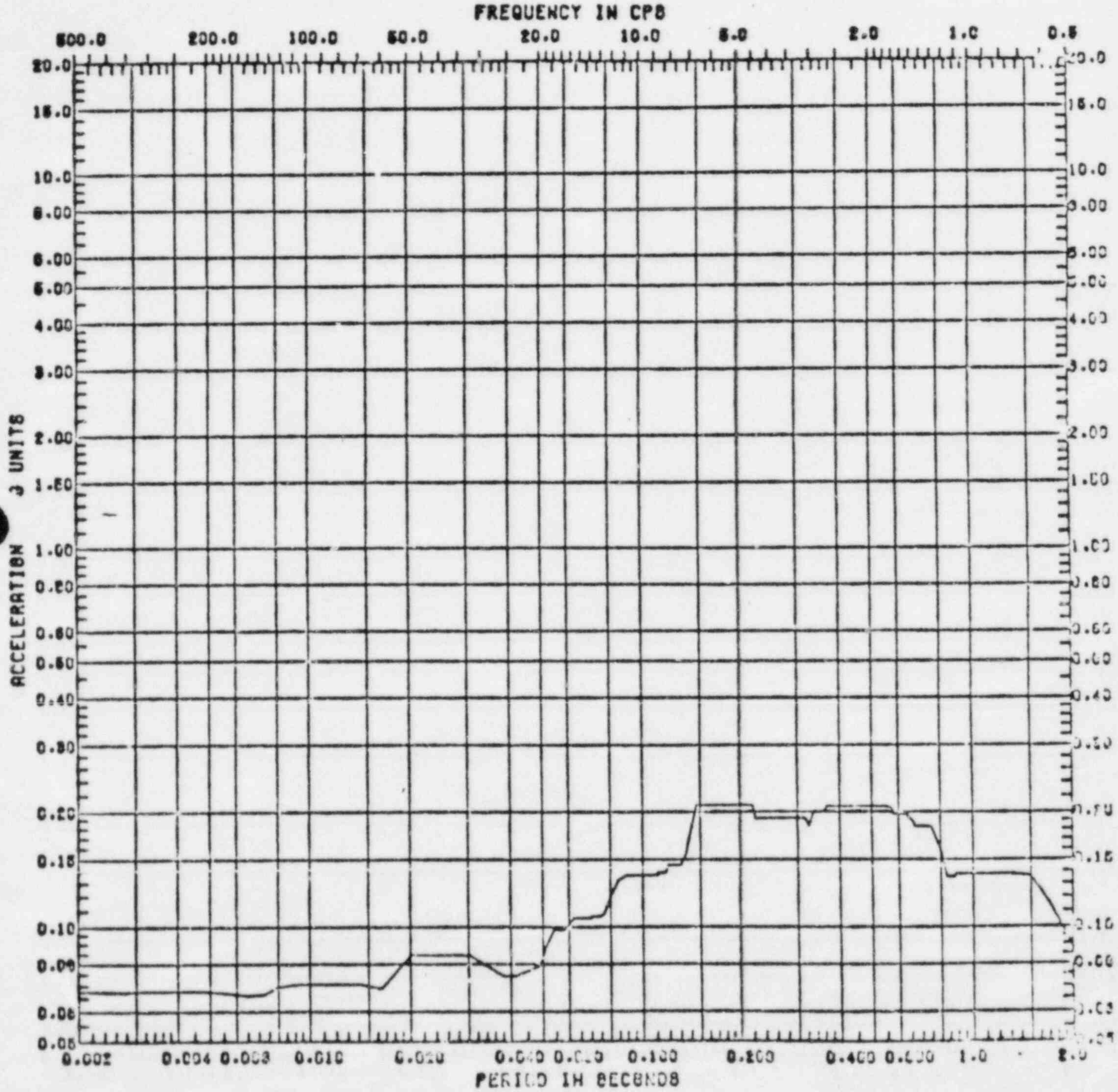
CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.36



LOADING:OBE(2%) EDGS
NODE 15-Y
DIRECTION VER ANGLE N/A

SPECTRA NO. OBE 2%
ELEVATION PANEL
LOCATION EDGS

1

SARGENT & LUNDY
ENGINEERS

14 MAY 84
SIKC

CALC NO. COD-014046
PROJECT SHOREHAM-1
REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.020
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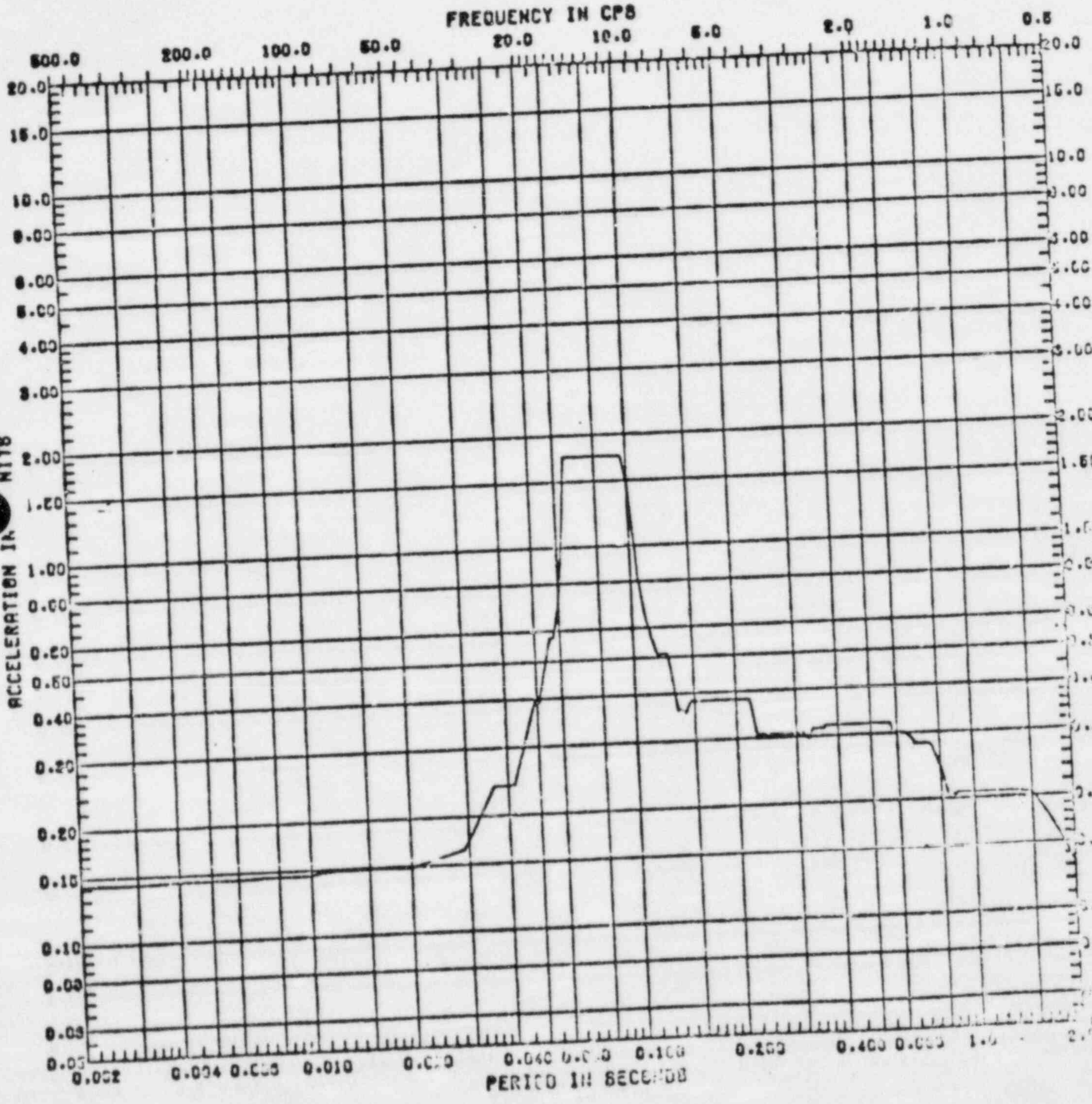
CALC NO. COD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C-1-37



LOADING:OBE(2%) EDGS
NODE 15-Z
DIRECTION HOR
ANGLE N/A
SPECTRA NO. OBE 2%
ELEVATION
PANEL
LOCATION EDGS

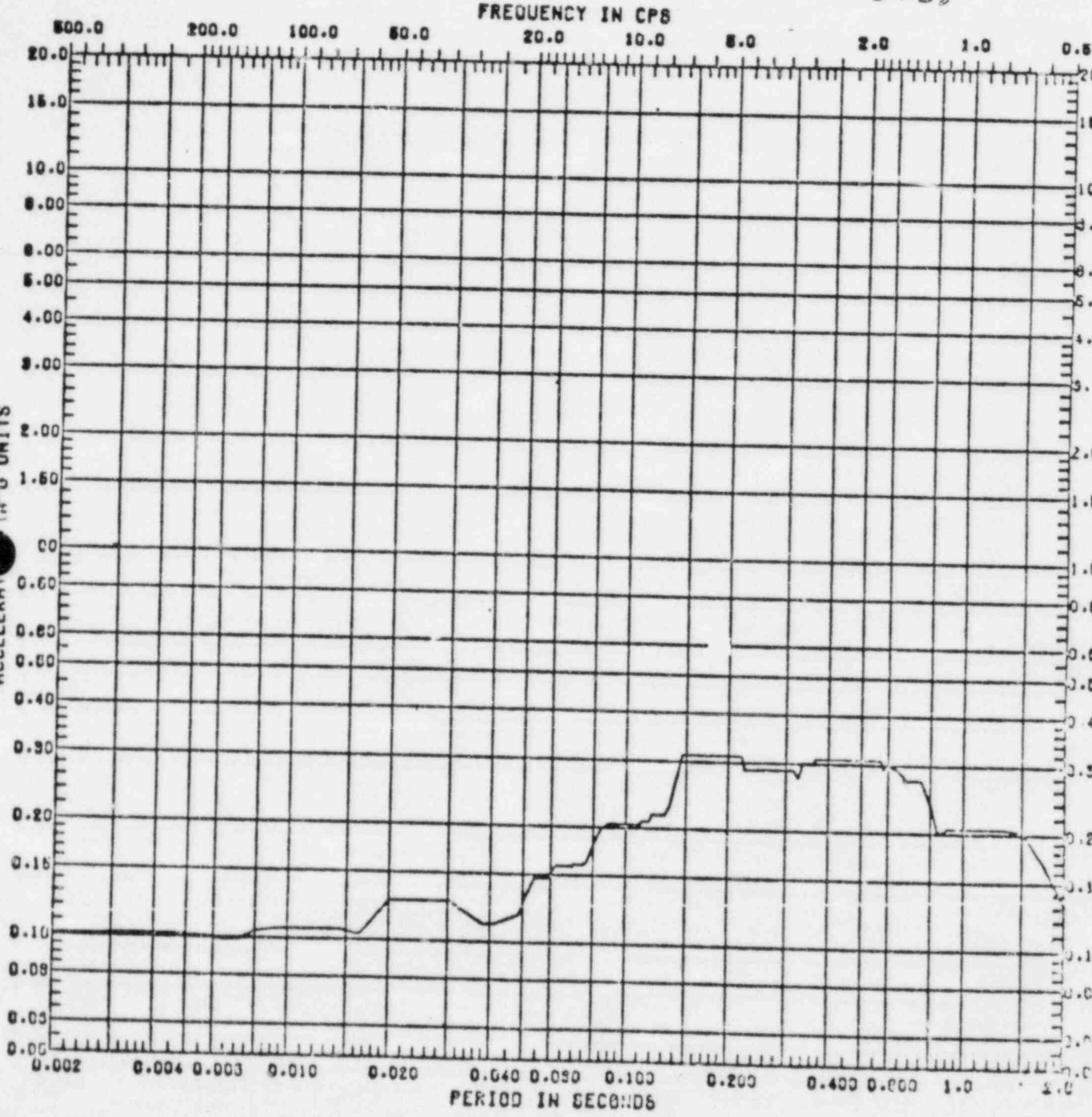
1

SARGENT & LUNDY
ENGINEERS

14 MAY 84
SIKD

CALC NO. CQD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.020
PAGE

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.38



LOADING:SSE(2%) EDOS
NODE 16-X
DIRECTION HOR
ANGLE N/A

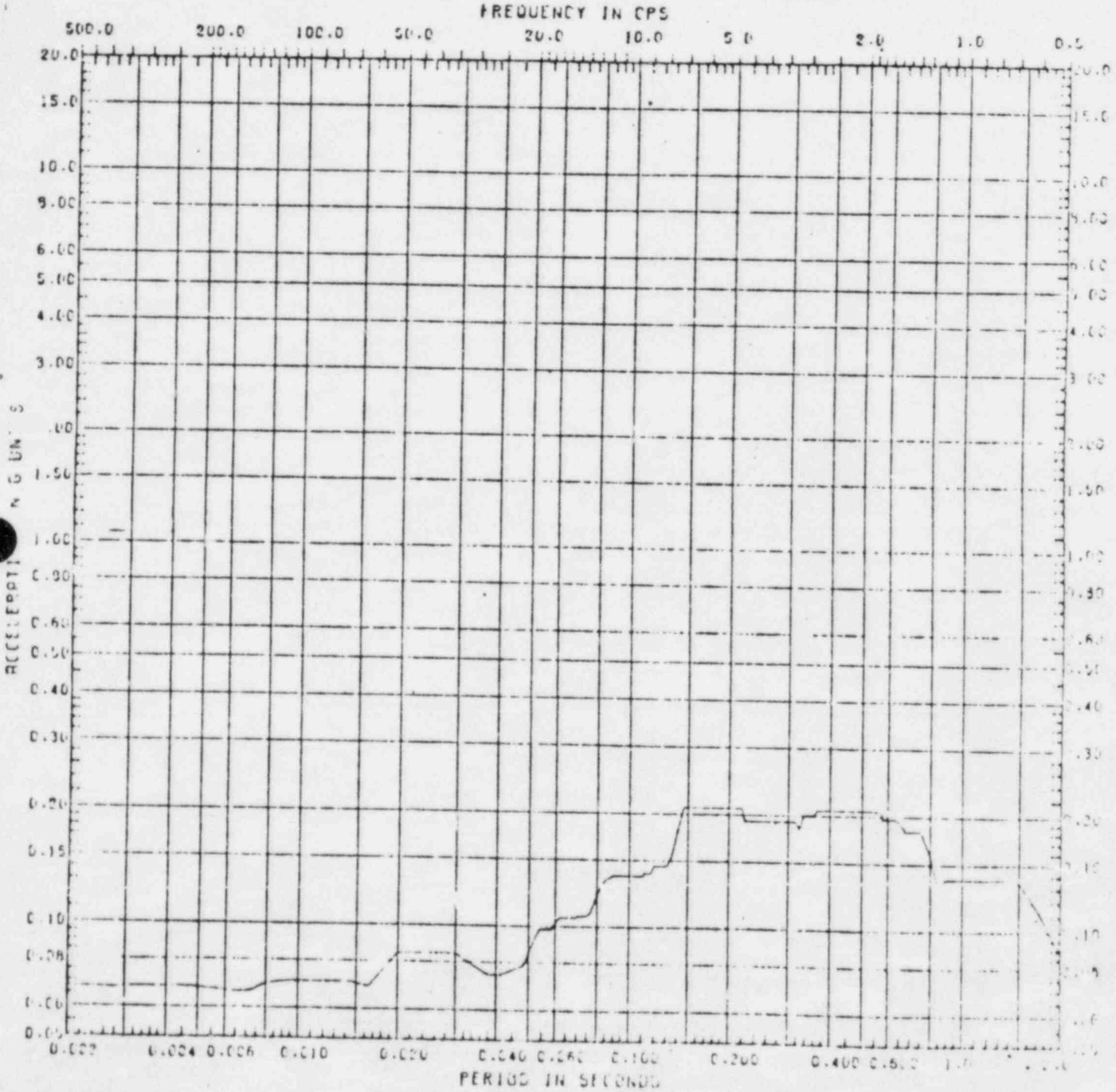
SPECTRA NO. SSE 2%
ELEVATION PANEL
LOCATION EDOS

SARGENT & LUNDY
ENGINEERS

14 MAY 84
SIKE

CALC NO. COD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.020
PAGE

CALC NO. COD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.39



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NODE 16-Y
DIRECTION VER
ANGLE N/A
SPECTRA NO. OBE 2%
ELEVATION
PANEL
LOCATION EDGS

SARGENT & LUNDY
ENGINEERS

14 MAY 84
SIKF

CALC NO. CQD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH S.DE
DAMPING 0.020
PAGE

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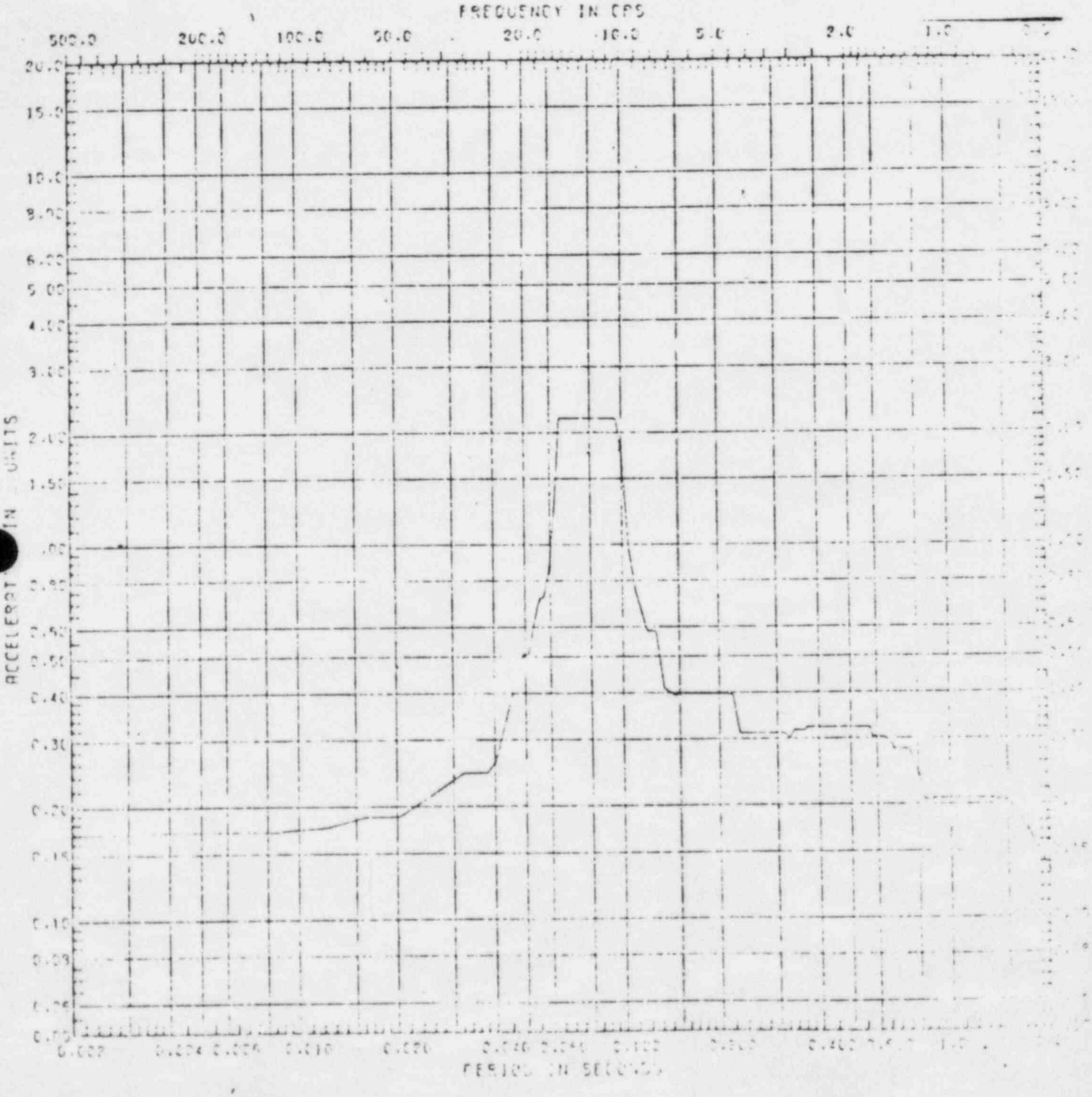
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.40



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DIRECTION N2S [illegible] N/A

SPECTRA NO. OBE 2%
ELEVATION [illegible]
LOCATION [illegible]

SARGENT & LUNDY
ENGINEERS

14 MAY 84
SIKG

CALC NO. COD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.020
PAGE

CALC NO. COD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
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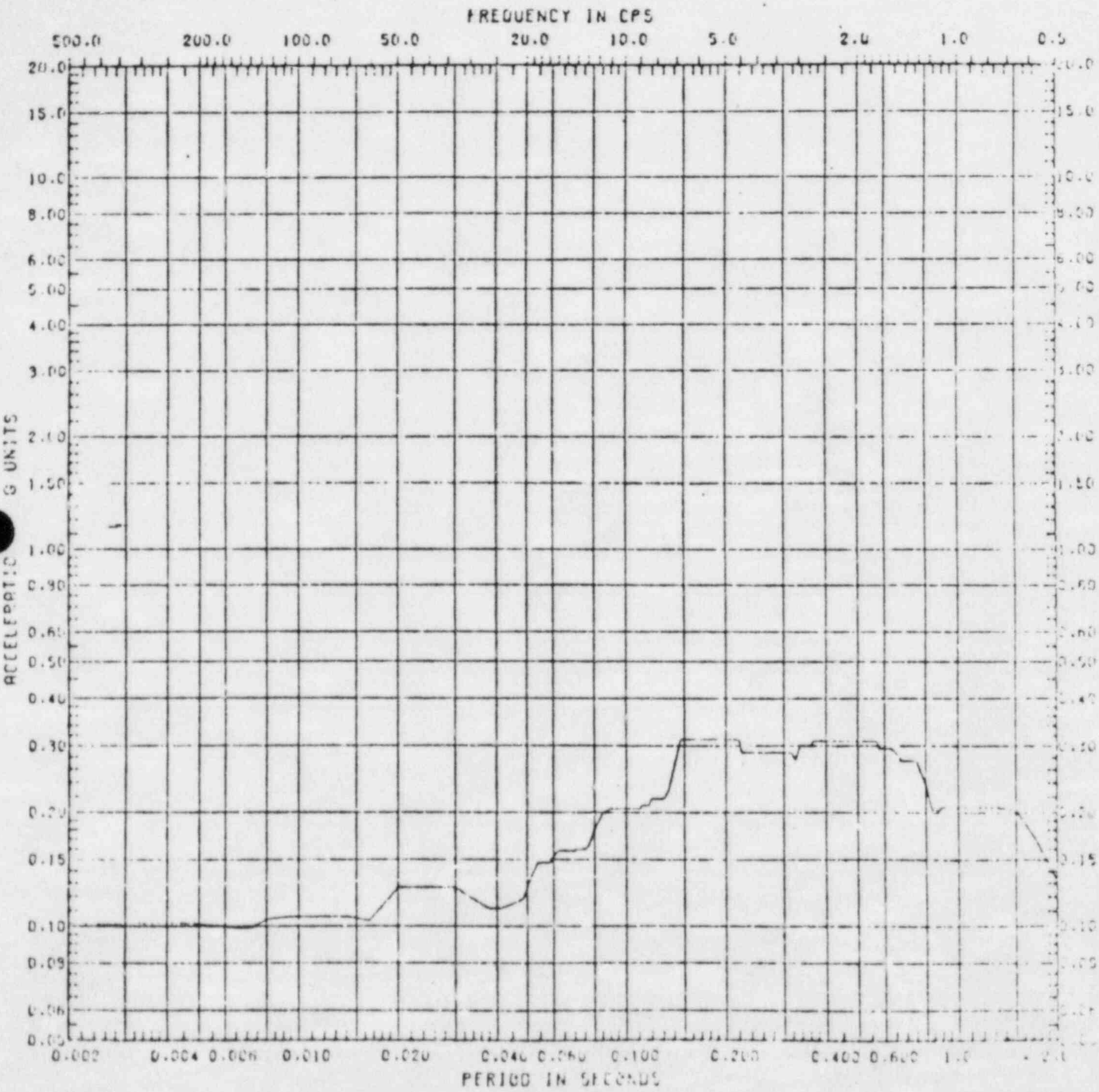

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SPECTRA NO. OBE 2%
NODE 60-X
ELEVATION PANEL
DIRECTION HOR
ANGLE N/A
LOCATION EDGS

SARGENT & LUNDY
ENGINEERS

14 MAY 84
SIKH

CALC NO. COD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
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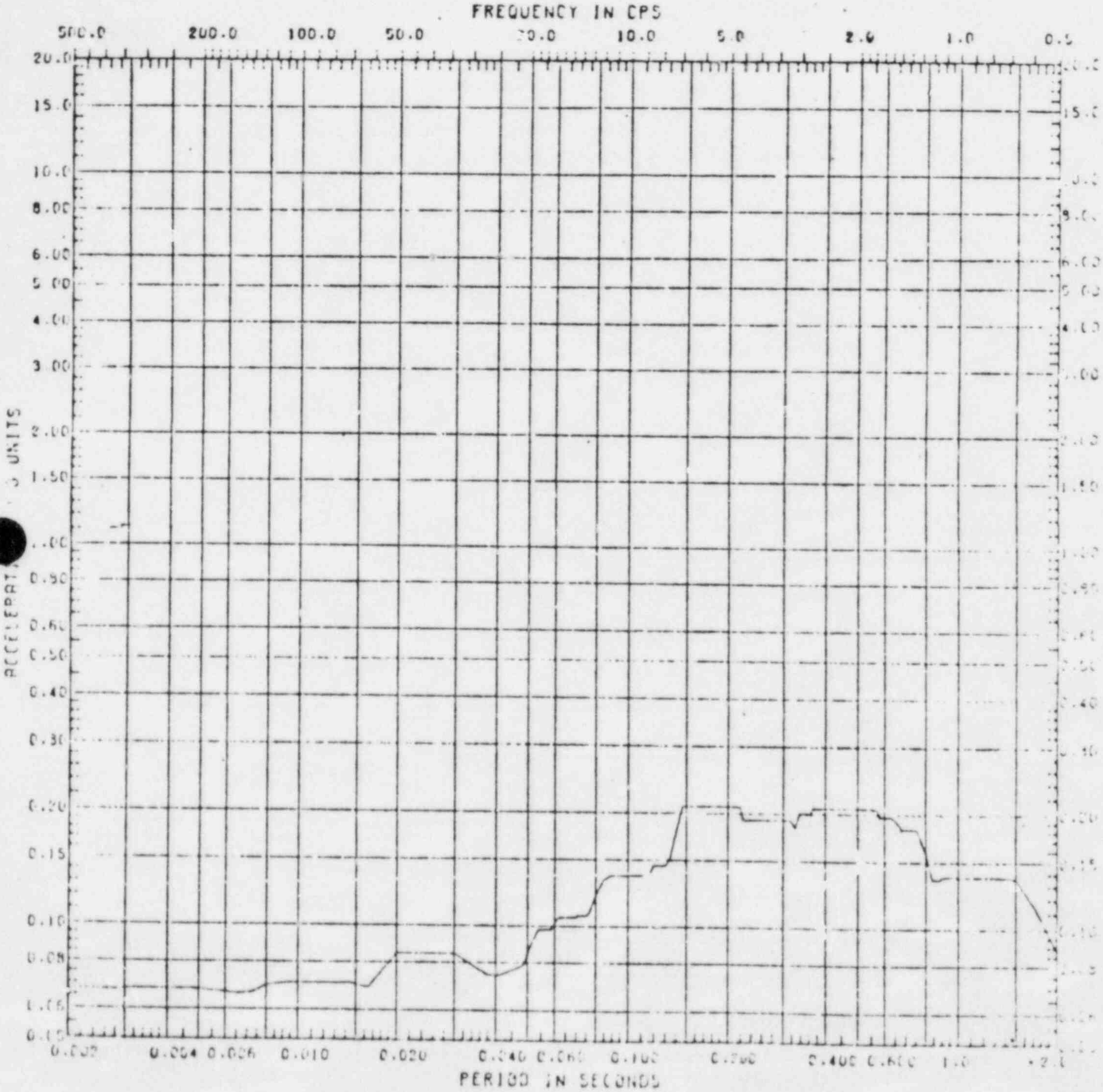

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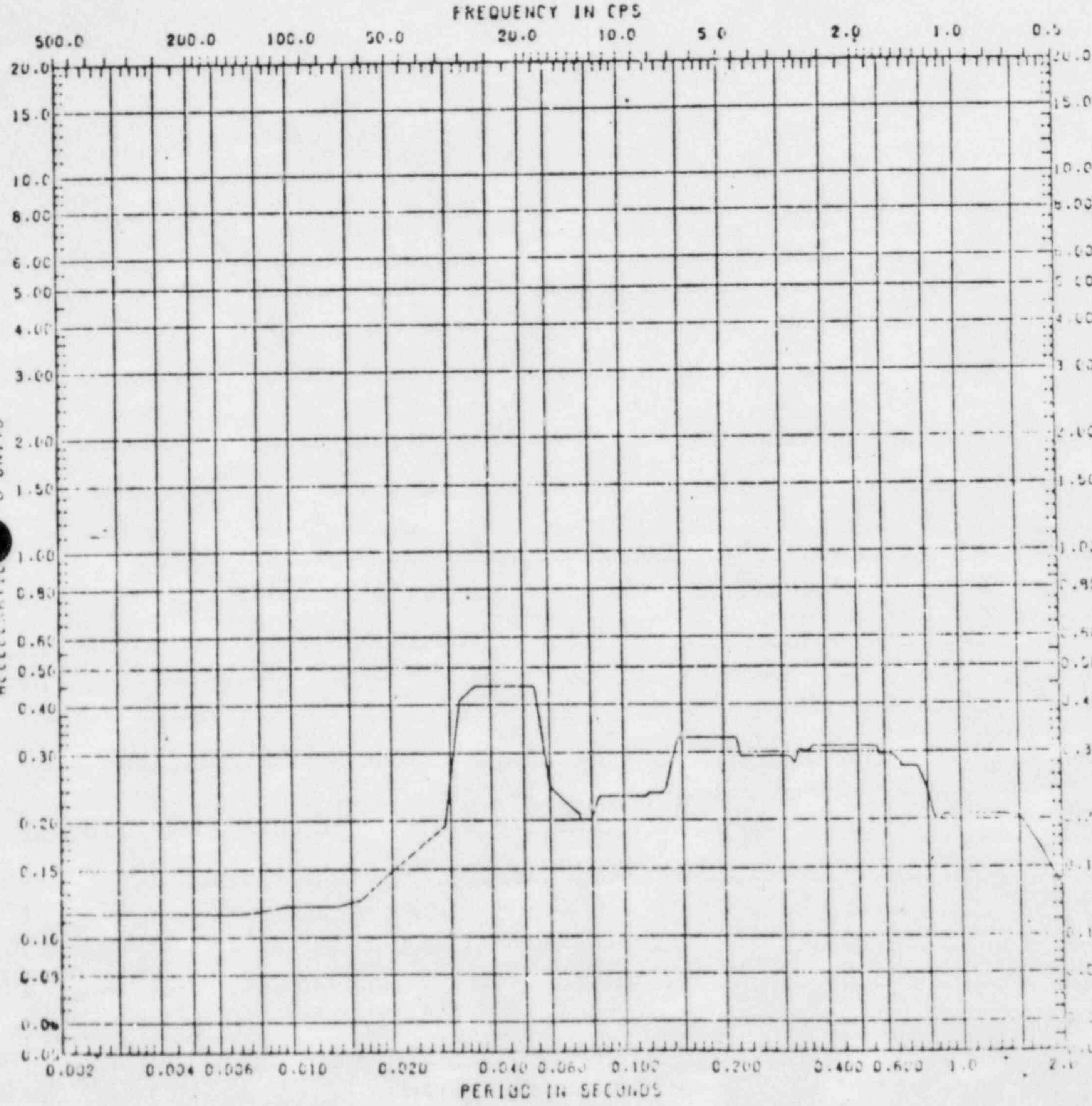
NODE 60-Y | ELEVATION PANEL
DIRECTION VER ANGLE N/A | LOCATION EDGS

SARGENT & LUNDY ENGINEERS

14 MAY 84
SIKI

CALC NO. CQD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.020
PAGE

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.43



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NODE 60-Z
DIRECTION HOR
ANGLE N/A
SPECTRA NO. OBE 2%
ELEVATION
PANEL
LOCATION EDGS

SARGENT & LUNDY
ENGINEERS

15 MAY 84
SIKB

CALC NO. CQD- 014046
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PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
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CALC NO. CQD-014046
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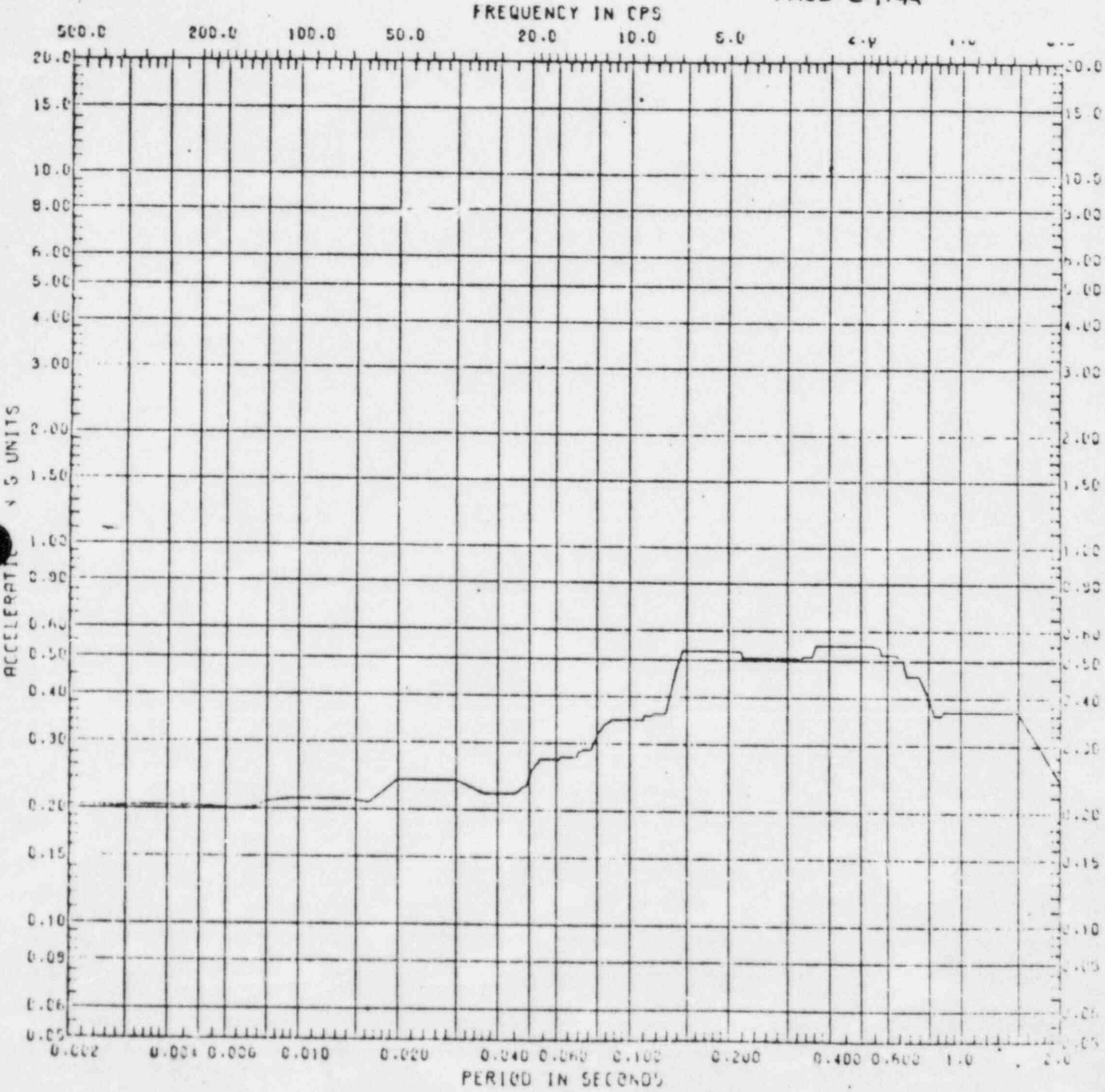
PROJ. NO. 6995-00
PAGE C.1.44



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NODE 16-X
DIRECTION HOR ANGLE N/A

SPECTRA NO. SSE 3%
ELEVATION PANEL
LOCATION EDGS

SARGENT & LUNDY
ENGINEERS

15 MAY 84
SIKP

CALC NO. CQD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.030
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CALC NO. CQD-014046
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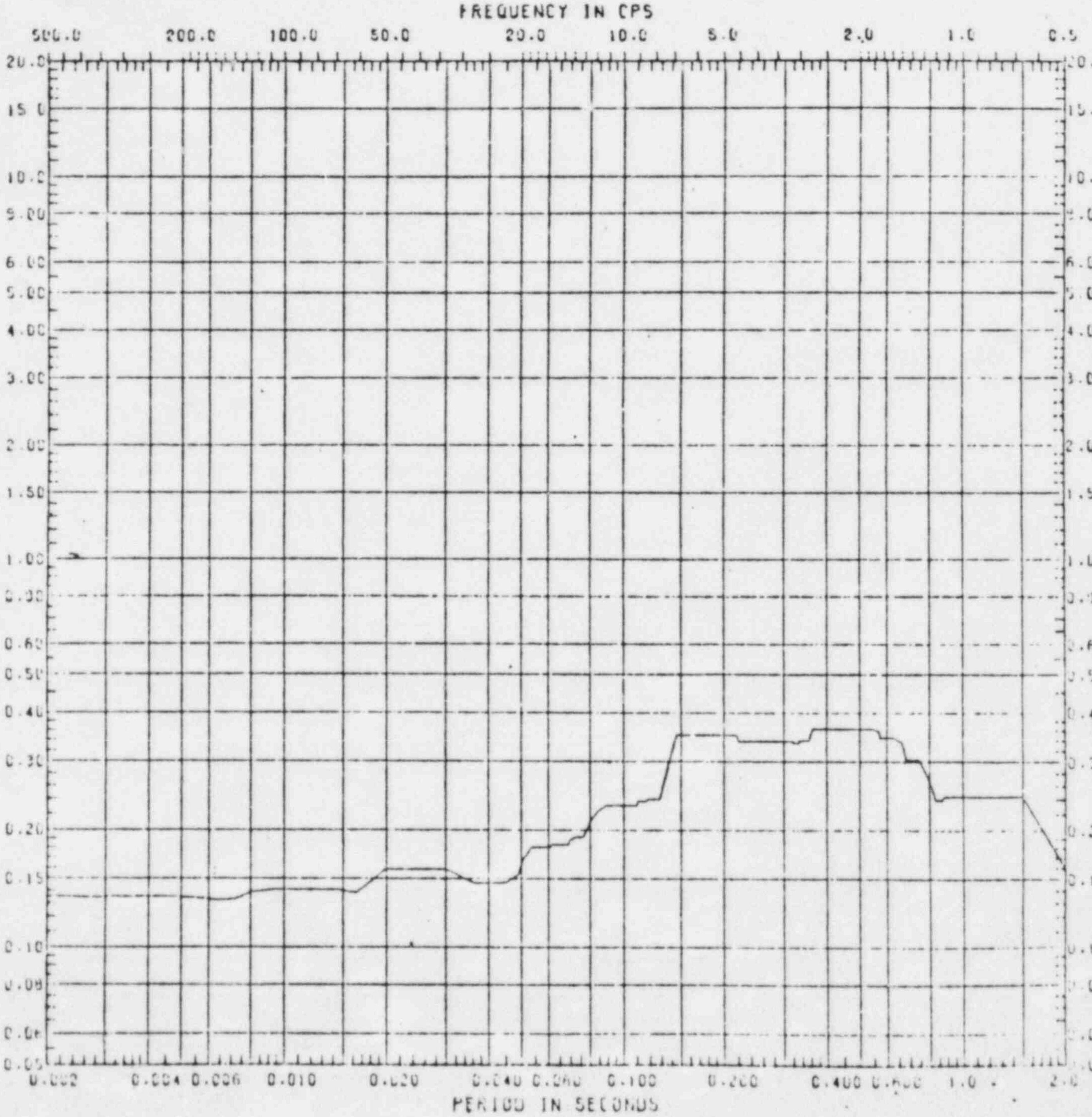

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SPECTRA NO. SSE 3%

NODE 16-Y

ELEVATION PANEL

DIRECTION VER ANGLE N/A

LOCATION EDGS

SARGENT & LUNDY
ENGINEERS

15 MAY 84
SIKO

CALC NO. COD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.030
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CALC NO. CQD-014046
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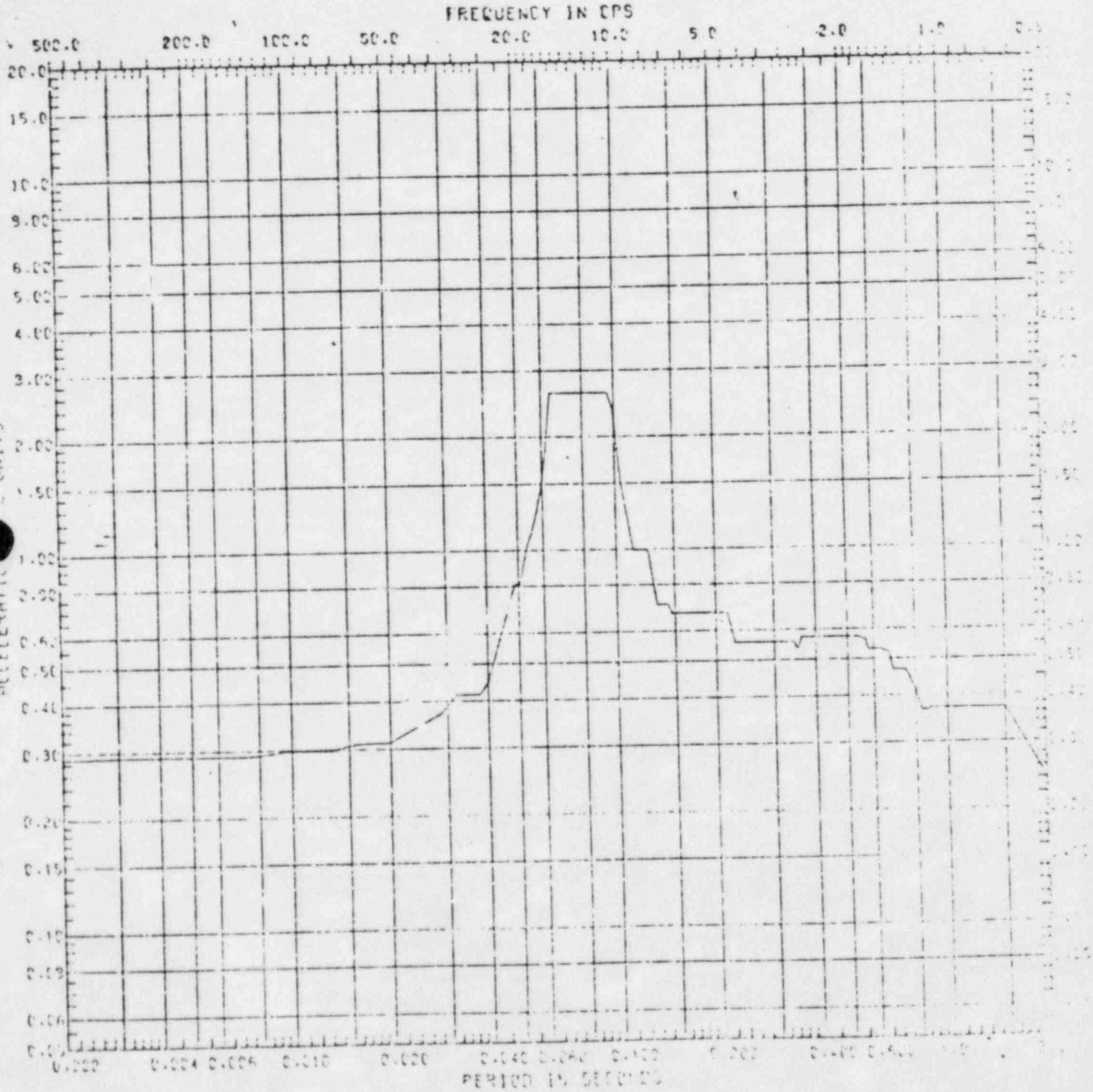

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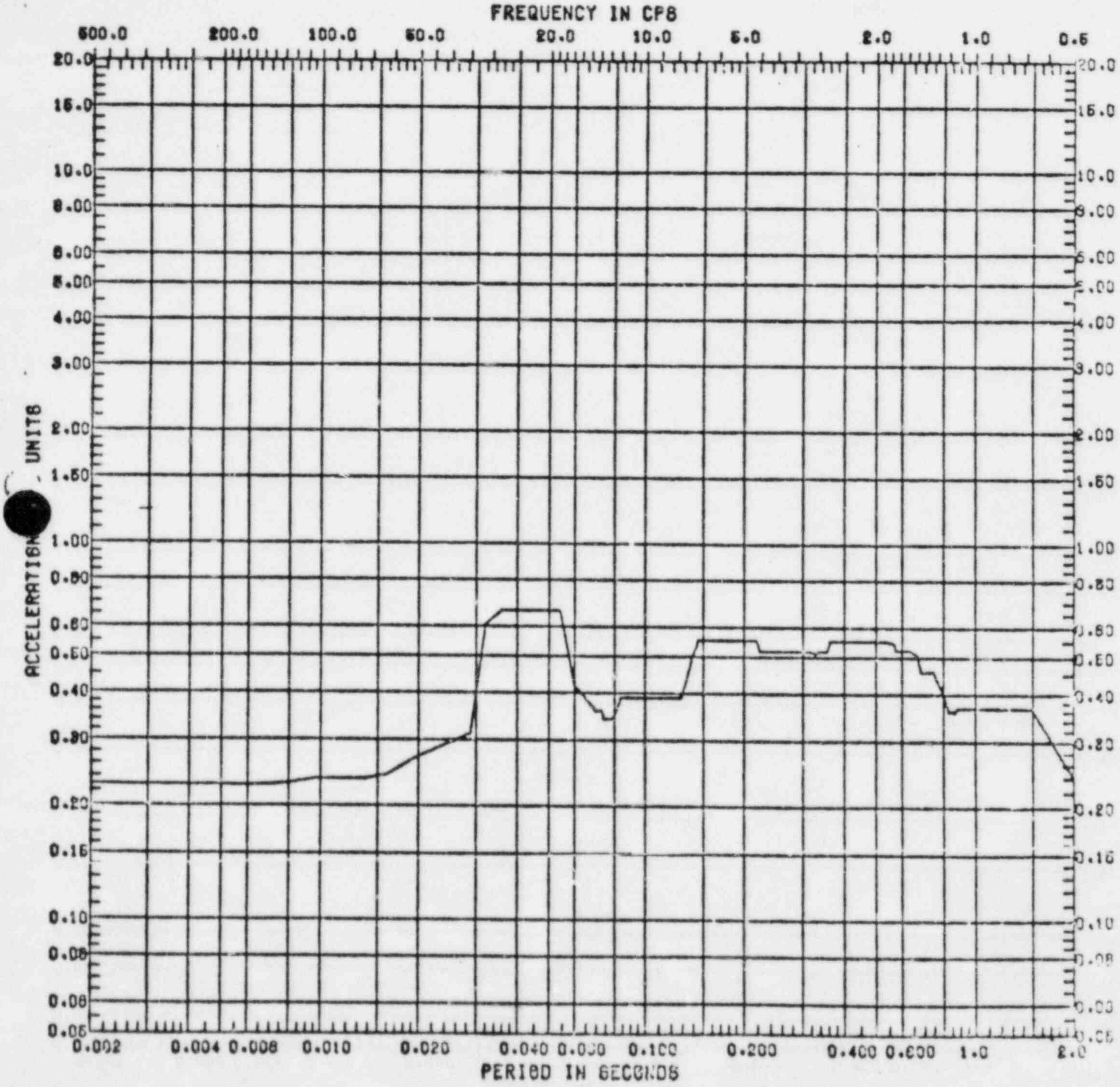
NODE 16-7
DIRECTION HOR ANGLE N/A
SPECTRA NO. SSE 3%
ELEVATION
PANEL
LOCATION EDG

SARGENT & LUNDY
ENGINEERS

15 MAY 84
SIKS

CALC NO. CQD-014046
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.030
PAGE

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE C.1.47



LOADING:SSE(3%) EDGC
NODE 80-Z
DIRECTION HOR ANGLE N/A
SPECTRA NO. SSE 3%
ELEVATION PANEL
LOCATION EDGS

CALC NO. CQD-014046
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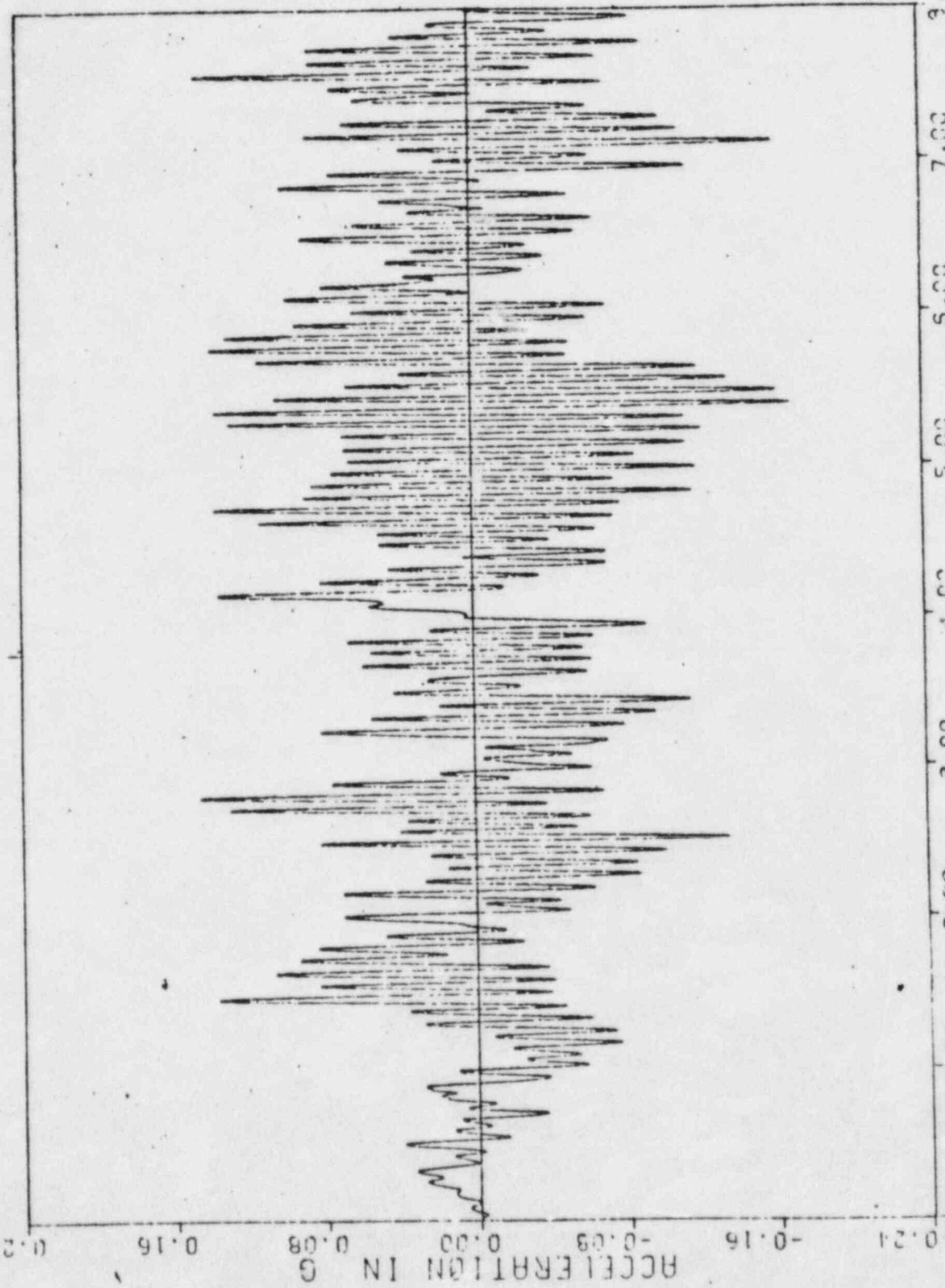
14 MAY 84



15-Z OBE 2% PANEL EDGS HOR N/A

13 MAY 84

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PROJ. NO. 6995-00
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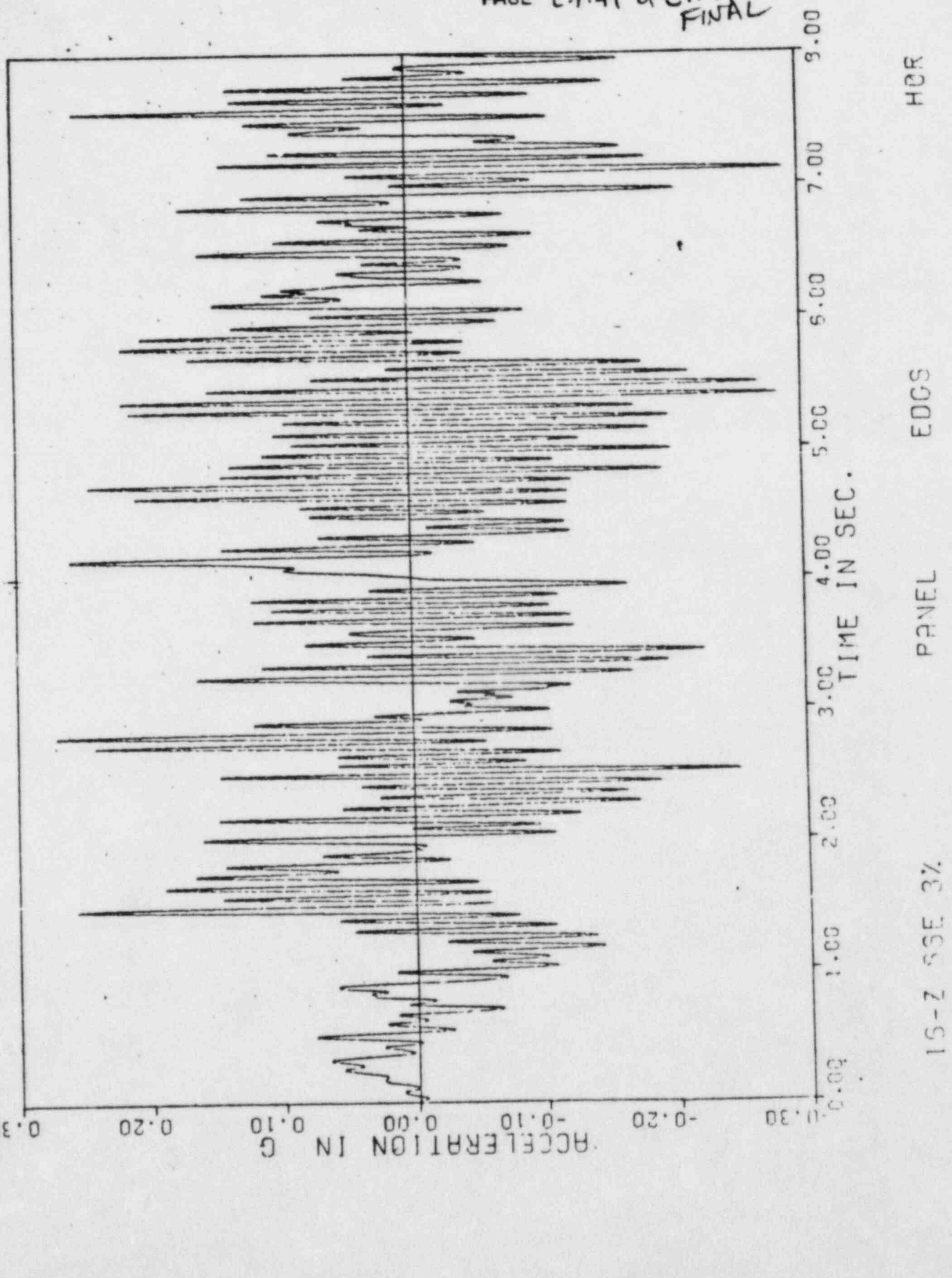
N/A



C.2

C.2 CONTROL INSTRUMENTS

Control Instrumentation

The electrical control and switchgear equipment in Shoreham's MP-45 diesel generator units are made up of both nuclear and commercial grade components. Some of these components are qualified by the use of existing seismic test reports. For the remainder of the components, the findings in the Lawrence Livermore National Laboratory Report NUREG/CR-2405, UCRL-15407, Subsystem Fragility (S.M.R.P. Phase I) was used to establish an operability risk level for component failure due to seismic loading. To further supplement the data from the above mentioned report, an analysis of the mounting screws for some of the electrical components has also been performed. This will ensure that the item will remain in place following an SSE level earthquake.

The static exciter has also been included in this section of the report. Although NUREG/CR-2405 doesn't list exciters as a generic category, it does list transformers, relays and other components that are used to make up the exciters circuitry. A list of the control instruments and a seismic reference for each was used to identify the specific components found in the system.

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Proj. No: [illegible]
Page [illegible] Of

ELECTRICAL DEVICES

GENERATOR & GOVERNOR CONTROL PANEL

Item No.	Reference Page	Description	Manufacturer	Model	Seismic Reference
1.		Voltage Regulator (VR)	Vickers	35-D870-2 S#61460-A33	NUREG CR-2405
2.		Gov. Peaking Load Cont. Box (EG)	Woodward Gov.	EMD-031025 8270-342	"
3.		Gov. Cont. Box (EGA)	Woodward Gov.	EMD-031025 8270-090	"
4.		Incremental Load Relay (ILR)	Vapor Corp.	36530082-01 EMD-8370794	"
5.		Low Load Relay (LLR)	Vapor	36530082-01 EMD-8370794	"
6		Breaker Aux. Relay (52W)	Vapor	36530082-01 EMD-8370794	"
7.		Exciter Contactor (EXC)	SQ-D Class 8504	Type EQ1281-62 (EMD-8288736)	"
8.		Adj. Resistor (Volt Range)	Ohmite	0960B	"
9.		Immersion Heater Contactor	Cutler Hammer	A10DNY1,Part A10D-1 Contact#624-2 EMD-8305991	"
10.		Relay Generator Voltage (GV)	Vapor	EMD-8263337	"
11.		Excitation Relay (ER)	Vapor	EMD-8263337	"
12.		Breaker Aux. Relay (52VX)	SQ-D	EMD-8269705 (Class 7001)	"
13.		Idle Run & Excitation Relay (IRE)	SQ-D	EMD8269705 (Class 7001)	"
		Alternating Overcurrent Relay (40 OC)	SQ-D	EMD-8267274	"
15.		Transformer-Anti-hunt (TBF-AH)		EMD-8267568	"

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Page [illegible]

ELECTRICAL DEVICES

Item No.	Reference Page	Description	Manufacturer	Model	Reference
		Fuses	Bussman	Non-6, AGC1 Non-15 EMD-8004758	EDGS, LaSalle
17.		Heater Overload Switch (OL)		EMD-8382961	NUREG CR-2405
18.		Field Flashing Cutout Relay (FFCO)	Westinghouse	Type SV Model 1876094	"

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Rev: 00 Date: 6/11/84
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ELECTRICAL DEVICES

SEQUENCE CONTROL PANEL

No.	Reference Page	Description	Manufacturer	Model	Seismic Reference
1.		Sequence Annun. Relay (SEQ)	MERCOR	EMD-8380774	NUREG CR-2405
2.		Engine Annun. Relay (EN)	MERCOR	EMD-8398985	"
3.		Engine Temp. Annun. Relay (ET)	MERCOR	EMD-8380774	"
4.		Low Temperature Annun. (LTRA)	MERCOR	EMD-8380774	"
5.		Normal Lock-out Relay (NLO)	Vapor	EMD-8370706	"
6.		Man Bearing Relay (MBX)	Vapor Corp.	EMD-8370794 Type 36530082-C1	"
		Excitation Relay (ERY)	Square-D	Type CO-5E (EMD-8253244)	"
8.		Excitation Relay (ERX)	Square-D	Type CO-5E EMD-8253244	"
9.		Oil Alarm Relay (OAD)	Square-D	Type CO-5E (EMD-8253242)	"
10.		Excitation Relay (ERS)	SQ-D	Type CO-5E EMD-8253244	"
11.		Zero Engine Speed Relay (ZSR)	Square-D	Type CO-5D (EMD-8253241)	"
12.		Pinion Failure Relay (PF)	Square-D	Type CO-5E (EMD-8253242)	".
13.		Generator Breaker Annun. Relay (GB)	MERCOR	EMD-8380774	"
14.		Speed Sensing Panel (SSP)	EMD	EMD-8398973	"
		Breaker Relay (52V)	Vapor	EMD-8365353	"
16.		Automatic Relay (AR)	Vapor	EMD-8419429	"

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ELECTRICAL DEVICES

	Reference Page	Description	Manufacturer	Model	Seismic Reference
17.		Unit Start Relay (USA)	Vapor	EMD-8370706	NUREG CR-2405
18.		Unit Shutdown Relay (UST)	AGASTAT	2422PG15A2 (EMD-8365352)	"
19.		Low Temperature Relay (LTR)	Vapor	EMD-8263337	"
20.		Starting Relay Aux. (STRA)	Vapor	EMD-8357415	"
21.		Main Bearing Relay (MBR)	Vapor	EMD-8265353	"
22.		Unit Start Relay (USR)	SQ-D	EMD-8365353	"
23.		Pinion Failure Switch (PFS)		EMD-8414178	"
24.		Selenium Rectifier (CR)		EMD-8365446	"
		Starting Contactor (ST)	SQ-D	EMD-8268542	"
26.		Starting Contactor (STZ)	Allis Chalmer	EMD-8398382	"
27.		Field Flashing Contactor (FFC)	SQ-D	EMD-8284295	"
28.		Alternator Overcurrent Relay (40-OCX)	SQ-D	EMD-8253242 Type CO-5E	"
29.		Overspeed Trip Limit Relay (OTL)	SQ-D	Class 7001 EMD-8263337	"
30.		Engine Speed Relay (ESR)	SQ-D	Class 7001 EMD-8269705	"
31.		Exciter Time Delay Delay (ETD)	SQ-D	EMD-8233242 Type CO-5E	"
32.		Unit Start Relay (USD)	AGASTAT	EMD-8398347	"
		Unit Vary Relay (UR)	Vapor	EMD-8370794 36530082-01	"
34.		Overspeed Trip Relay (OTT)	AGASTAT	Model-1072PHC696 EMD-8398347	"
35.		Fuses	Busman FPE	Non-10 [illegible]350	EDGS, [illegible]

ELECTRICAL DEVICES

DEADLINE CONTROL PANEL

Item No.	Reference Page	Description	Manufacturer	Model	Seismic Reference
1.		Deadline Overload Current Relay (SC1)	Westinghouse	EMD-8398310 Syle-292B400A12 Type SC1	NUREG CR-2405
2.		Deadline Overload Current Relay (SC2)	Westinghouse	EMD-8398310 Style292B400A12 Type SC1	"
3.		Rectifiers (CR8&CR9) (CR)		EMD-8365446	"
4.		Master Aux. Relay (MRX)	Vapor Corp.	EMD-8370794 Type 86530082-01	"
5.		Overload Timer Relay (OLT)	AGASTAT	EMD-8398311	"
		Oil Alarm Aux. Relay (OADX)	Vapor Corp.	Type36530082-01 EMD-8370794	"
7.		Master Relay (MR)	Vapor	EMD-8370706	"
8.		Deadline Relay (DLU)	Vapor	EMD-8365353	"
9.		Deadline Relay (DLT)	Vapor	EMD-8365353	"
10.		Resynch Relay (RSU)	Vapor	EMD-8370706	"

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ELECTRICAL DEVICES

[illegible]G CONTROL PANEL

tem No.	Ref. Page	Description		Manufacturer	Model	Seismic Reference
1		Starting Stepping Switch	(STS)		EMD-8295944	NUREG CR-2405
2		Breaker Relay Auxiliary	(52VM)	SQ-D	Type CO-5E EMD-8253242	"
3		Starting Relay	(STX)	SQ-D	Type CO-5E EMD-8253242	"
4		Starting Relay	([illegible])	SQ-D	Type CO-5E EMD-8253244	"
5		Starting Relay	(STD)	SQ-D	Type CO-5E EMD-8253242	"
6		Fuel Transfer Annun Relay	(FT)	MERCOR	EMD-8380774	"
7		Enginer Speed/Fuel Transfer Relay	(ESF)	VAPOR	EMD-8370794 36530082-01	"
8		Starting Relay	(STR)	SQ-D	Type CO-5E EMD-8253242	"
9		Starting Relay Aux	(STYA)	VAPOR	EMD-8227936	"
0		Rectifier	(CR37)		EMD-8263244	"
1		Battery Switch			EMD-8365961	"
2		Starting Resistor	(RE-ST)	DYNAMIC CORP	EMD-8399521	"
3		Fuses			EMD-8004555 (10A) EMD-8004549 (15A) EMD-8004548 (30A) EMD-8267501 (70A) EMD-8127052 (400A)	"

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ELECTRICAL DEVICES

LINE SEQUENCE CONTROL PANEL

ITEM NO.	REF. PAGE	DESCRIPTION		MANUFACTURER	MODEL	Seismic Reference
1		Motor Driven Rheostat	(MRG)	Woodward	Part #8270-010 EMD-8366578	NUREG CR-2405
2		Deadline Stepping Switch	(DLS)		EMD-8295944	"
3		Plant Start Relay	(PSR)	SQ-D	EMD-8253244 Type CO-5E	"
4		Resynch Relay	(RST)	SQ-D	EMD-8253245 Type CO-5E	"
5		Deadline Relay	(DLX)	SQ-D	EMD-8253242 Type CO-5E	"
6		Deadline Breaker Close Relay	(DLBC)	SQ-D	EMD-8253242 (Class 7001)	"
7		Resynch Relay	(RS)	SQ-D	EMD-8269705	"
		Resynch Relay	(RSR)	SQ-D	EMD-8297116	"
9		Load Breaker Relay	(52WL)	Vapor	EMD-8370794 (36530082-01)	"
10		System Breaker Relay	(52WS)	Vapor	EMD-8370794 (36530082-01)	"
11		Deadline Relay	(DLB)	Vapor	EMD-8370794 (36530082-01)	"
12		Deadline Relay	(DLZ)	Vapor	EMD-8370794 (36530082-01)	"
13		Deadline Relay	(DLR)	Vapor	FMD-8398823	"

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ELECTRICAL DEVICES

AUXILIARY SWITCHGEAR CUBICLE 1A

No.	Ref. Page	Description	Manufacturer	Style Model	Seismic Reference
1.		Frequency Meter	Westinghouse	S1M291B (KX-241) 199A10	Byron Seismic CQD-012397
2.		AC Ammeter	Westinghouse	KA-241	Zimmer Seismic EMD-000495
3.		AC Voltmeter	Westinghouse	S1M281B (KA-241) 460A12	Zimmer Seismic EMD-000495
4.		Kilowatt/Kilovar Meter	Westinghouse	S1M409C (KP-241) 711A63	Zimmer Seismic EMD-000495
5.		Differential Overcurrent Relays	Westinghouse	Type Co-11 289B094A16A	CQD-007509 & CQD-003648
6.		Watthour Meter	General Electric	DSW-64-701x3G646 DSW-66-700x71G7	NUREG CR-2405
	–	Under Voltage Relay	Westinghouse	Type (CV-2) 1875508A	"
8.		Rotary Lockout Relay	Allis Chalmer	Type 210 14-174-716-501	"
9.		Annunciator Reset Push Button	Square-D	Type TD Class 9001	EDGS, LaSalle
10.		Governor Control Switch	Allis Chalmer	Type 210 25-104-023-802	NUREG CR-2405
11.		Voltage Control Switch	Cutler Hammer		"
12.		Peaking Swith		14961280-50-004	"
13.		Start/Stop Switch	Cutler Hammer		"
14.		Phase Shift Transformers	Westinghouse	K5 1133788	"
15.		Frequency Transducer	Westinghouse	VC841	"
16.		Impulse Generator	General Electric	Type D-41	"
		Lamps	General Electric	00857283	"
.		Circuit Breaker for Strip Heater and Light			"
19.		Under Voltage Relay	Square-D	Class 7001, Type M08269703	"

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ELECTRICAL DEVICES

AUXILIARY SWITCHGEAR CUBICLE 1A, INTERIOR

[illegible]em No.	Ref. Page	Description	Manufacturer	Style Model	Seismic Reference
1.		Fuse Compt. (Primary)			NUREG CR-2405
2.		Cont. Power XFMR.			"
3.		Terminal Blocks			"
4.		Fuses (Secondary Bus)			"
5.		Switch DC Control			"

ELECTRICAL DEVICES

GENERATOR SWITCHGEAR CUBICLES 1, 2, 3 and 4

m No.	Ref. Page	Description	Manufacturer	Style Model	Seismic Reference
1.		Overvoltage Relay 59	Westinghouse	CV5-1875512A	NUREG CR-2405
2.		Reverse Power Relay 67	Westinghouse	CW 289B988A09	"
3.		Leading Vars Relay 55	Westinghouse	CW 289B988A18A	"
4.		CB Control Switch	Allis Chalmer	Type 210 14-137-631-501	"
5.		AMM-WM-VAR Switch	Allis Chalmer	Type 210 14-174-603-501	"
6.		Voltmeter & Freq. Switch	Allis Chalmer	Type 210 14-174-604-501	"
7.		Stop-Start Switch	Cutler Hammer		"
8.		Governor Control Switch	Allis Chalmer	Type 210 14-189-333-501	"
9.		Manual Switch	Allis Chalmer	Type 210 14-187-254-501	"
.		Voltage Control Switch	Allis Chalmer	Type 210 14-189-176-501	"
11.		Relay		42R099566	"
12.		Fuses	Bussman	FR6 1/4 A	"

GENERATOR SWITCHGEAR CUBICLES 1, 2, 3 and 4 - INTERIOR

1.		Circuit Breaker	Allis Chalmer	Type MA350B 1200A, 4160V 350 MVA@4160V 200 MVA@2300V	NUREG CR-2405

ELECTRICAL DEVICES

TRANSFORMER PANEL

em No.	Ref. Page	Description	Manufacturer	Style Model	Seismic Reference
1.		Current Transformer (CT) (CT1, CT2, CT3, CT7) (CT5, CT6, CT 4)		Type MCT1820DD EMD-8376438 EMD-8376437	NUREG CR-2405
2.		Potential Transformers (PT) PTA, PTB, PTC	Allis Chalmers	EMD-8365597	"

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ELECTRICAL DEVICES

ELECT. COMPONENTS IN ENGINE COMPARTMENT

No.	Ref. Page	Description	Manufacturer	Style Model	Seismic Reference
1.		Engine Oil Temp. (TIO)	G.M.	EMD-8031666	NUREG CR-2405
2.		Shutter Cont. Relay (TRA)	Square-D	EMD-8253242	"
3.		Shutter Cont. Relay (TRB)	Square-D	EMD-8253246	"
4.		Engine Temp. Switch (ESTD) Delay Relay	Square-D	EMD-8253246	"
5.		Engine Water Temp. (ETS) SW	ASCo	EMD-8323900	"
6.		Low Oil Temp. SW (LOTS)	Square-D	BCW-353	"
7.		Shutter Cont. Temp. (TB) SW	Square-D	BCW-352	"
8.		Imm. Htr. Temp. Cont. SW (TC)	Square-D	BCW-259	"
9.		Engine Water Temp. (TIW)	G.M	EMD-8031666	"

Calc. No: CQD-[illegible]
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ELECTRICAL DEVICES

ENGINE CONTROL CABINET

n No.	Ref. Page	Description	Manufacturer	Style Model	Seismic Reference
1.		Timer Relay (T)	Agastat	EMD-8379659 (S#W4134291)	NUREG CR-2405
2.		Fuel Transfer Relay(FTC1)			"
3.		Fuel Transfer Relay(FTC2)			"
4.		Rectifier (CR1)	International Rectifier	EMD-8365446	"
5.		Low Eng. Water Press. Switch (LWS)	Square-D	ACW5S40 (EMD-8358933)	"
6.		Main Bearing Oil Press. Switch (MB1)	Square-D	ACW5S40 (EMD-8358933)	"
7.		Main Bearing Oil Press. Switch (MB2)	Square-D	ACW5S41 (EMD-8322641)	"
8.		SW Immer. Heater			"
9.		Start/Stop Switch			"
		SW Fuel Transfer			"
11.		Main Bearing Oil Press. Switch (MB1)	American Standard Controls	EMD-8287507	"
12.		Main Bearing Oil Press. Switch (MB2)	ASCo	EMD-8291840	"

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5/21/84

LILCo - Shoreham Plant 6995-00

Similarity Analysis

Subject Matter

AGASTAT Time Delay Relay
Model No. 1072PHC696

History

Relay is from the old "2400" series design. This series has been replaced by the "7000" series which is in use today at LaSalle and other plants. The LaSalle seismic requirements envelope the requirements at Shoreham.

Similarity

2400 & 7000 Series

Parameter	Comparison
1) Mounting Bracket	No change
2) Overall Sized Weight	No significant change
3) Coil & Contacts	Similar

Vendor Test Report - E7012/7022

LaSalle Seismic Qualification - CQD-000845

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SARGENT & LUNDY

MEMORANDUM OF
TELEPHONE CONVERSATION

Date: 5-21-84

Time: 3:27 P.M. CST

Person Called: Don Alexander (Name) of AGASTAT (201)964-4400 (Company)

Person Calling: R. Moery (Name) of S&L (Company)

Project: LILCo Shoreham Project No. 6995-00

Subject Discussed: 1072PHC696 Time Delay Relay

Summary of Discussion, Decisions and Commitments:

The C696 denotes EMD division of GM as customer. This part "1072PHC696" is part of the old 2400 series. It was replaced by the current 7000 series. The mtg. bracket and dimensions are exactly the same on the 7000 series. Seismic qualification is covered by Agastat Report #E7012/E7022.

cc:

R. J. Moery
Signature

File:

Electrical Component Risk Evaluation

Typically, electrical components used in safety related applications in nuclear generating stations are seismically qualified by test where the Test Response Spectrum (TRS) envelopes the Required Response Spectrum (RRS). In risk studies, a mean or median fragility level of equipment, and the variability about this value, is established. Since fragility testing is rarely conducted for neclear power plant equipment, it is difficult to establish fragility levels above the qualification level. The subsystem fragility report uses a data base of military shock tests of similar, off-the-shelf equipment, to develop fragility descriptons for several generic categories of commercial grade equipment. A curve representing the frequency of failure, P, to a certain acceleration, A, is defined as fragility curve. The fragility curve for any component and its uncertainty can be expressed in terms of the best estimate of the median value of the fragility parameter to which the fragility curve is referenced. Use of a lognormal distribution enables easy development and expression of these curves and their uncertainty. With perfect knowledge, (i.e, only accounting for the random variability, B_R), the frequency of failure, $P_{f(a)}$, for a given acceleration 'A, is obtained from:

$$P_{f(a)} = \emptyset\left(\frac{\ln(A/\check{A})}{B_R}\right) \quad \text{———(1)}$$

(see page 2-11 of NUREG/CR-2405)

in which $\emptyset$ (.) is the standard Gaussian cumulative function, $\check{A}$ is the "best estimate" of the median acceleration capacity, and B_R is the logarithmic standard deviation associated with the underlying randomness of the capacity. If uncertainty is taken into account then:

$$P_{f(a)} = \emptyset\left(\frac{\ln(A/\check{A})}{B_C}\right) \quad \text{———(2)}$$

(see page 2-8 of NUREG/CR-2405)

in which $B_C = (B_R^2 + B_C^2)^{½}$ and B_u is the logarithmic standard deviation associated with the uncertainty of the capacity.

The electrical components in the control panels and switchgear assembly can be grouped under certian "generic category" (see Table I). A risk assessment is performed on all of these generic categories and a 90%, 95% and 99% confidence level is calculated for the acceleration capacity (see Table II). Response factor is defined as the ratio of the computed or synthesized test response to the actual response of the component. In most designs, the calculations, or the parameters used in synthesizing tests, are biased on the conservative side of the response factor will, in general, be greater than 1.0 and hence, the 90%, 95% and 99% 'g' confidence levels computed in Table II are conservative values.

The maximum ground acceleration level for the electrical control and switchgear assemblies was found to be 0.45 g's at any frequency (see attached plot of Shoreham's SSE at 5% damping). Comparing this value with the 90%, 95% and 99% confidence values, we can clearly see that the electrical control and switchgear components have a 90% confidence on the seismic requirements for these components. The predominant failure modes observed in all electrical and control equipment are relay chatter and breaker trip. Neither of these failure modes results, in all cases, in failure of the equipment to perform its intended function. They are, however, functional failures which must be addressed by the system analyst. Relay chatter is a functional failure mode that is self-correcting after the vibratory earthquake motion ceases. By neglecting the relay chatter, the confidence level for all electrical control and switchgear components increases from 90% to 99% to remain functional after experiencing an SSE level earthquake.

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Table 1

GENERIC CATEGORY	SPECIFIC COMPONENT	FAILURE MODE	FUNDAMENTAL FREQUENCY Hz	FRAGILITY PARAMETER	DAMPING % OF CRITICAL	MEDIAN CAPACITY	[illegible] STD. DEVIATIONS			
							COMPOSITE β_c	RANDOM β_R	UNCERTAINTY β_U	[illegible]
Control Panels and Racks	Generic	Breaker Trip	5-10	Spectral Acceleration	5	7.7 g	0.73	0.4	0.61	6
Control Panels and Racks	Generic	Structural	5-10	Spectral Acceleration	5	14.6 g	0.8	0.4	0.69	6
Relay Cabinets	Generic	Relay Chatter	5-10	Spectral Acceleration	5	2.07 g	1.45	0.5	1.37	6
Relay Cabinets	Generic	Relay Trip	5-10	Spectral Acceleration	5	7.7 g	0.73	0.4	0.61	6
Relay Cabinets	Generic	Structural	5-10	Spectral Acceleration	5	14.6 g	0.8	0.4	0.69	6
Motor Control Centers	Generic	Relay Chatter	5-10	Spectral Acceleration	5	2.07	1.45	0.5	1.37	6
Motor Control Centers	Generic	Breaker Trip	5-10	Spectral Acceleration	5	7.7 g	0.73	0.4	0.61	6
Motor Control Centers	Generic	Structural	5-10	Spectral Acceleration	5	14.6 g	0.8	0.4	0.69	6
Breaker Panels	Generic	Breaker Trip	5-10	Spectral Acceleration	5	7.7 g	0.73	0.4	0.61	6
Breaker Panels	Generic	Structural	5-10	Spectral Acceleration	5	14.6	0.8	0.4	0.69	6
Emergency D.C. Power Units	Batteries	Case Cracking & Plate Failure	8	Spectral Acceleration	5	4.2 g	0.16	0.1	0.12	6
Switch Gear	4160 & 480 Volt Units	Relay Chatter	5-10	Spectral Acceleration	5	2.07 g	1.46	0.5	1.37	6
Switch Gear	4160 & 480 Volt Units	Breaker Trip	5-10	Spectral Acceleration	5	7.7 g	0.73	0.4	0.61	6
Switch Gear	4160 & 480 Volt Units	Structural	5-10	Spectral Acceleration	5	14.6 g	0.8	0.4	0.69	6
Transformers	Generic	Structural	5-10	Spectral Acceleration	5	10.7 g	0.21	0.1	0.18	2
Local Instruments & Transmitters	Generic	Electrical Function	Rigid	Zero Period Acceleration	NA	37.8 g	0.32	0.2	0.25	6
Instrument Panels & Racks	Generic	Relay Chatter	5-10	Spectral Acceleration	5	2.07 g	1.46	0.5	1.37	6
Instrument Panels & Racks	Generic	Breaker Trip	5-10	Spectral Acceleration	5	- 7.7 g	0.73	0.4	0.61	6
Emergency A.C. Power Units	Generator Control Panel	Relay Chatter	30	Spectral Acceleration	5	0.95 g	0.24	0.15	0.19	6
Emergency A.C. [illegible]er Units	Engine Control Panel	Failed Relay	11	Spectral Acceleration	5	2.0 g	0.25	0.15	• 0.20	6
Control Panels and Racks	Generic	Relay Chatter	5[illegible]	Spectral Acceleration	5	2.07 g	1.46	0.5	[illegible]	6

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Calcs For
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Client L.I.L.CO.
Project Shoreham
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Prepared by | Date
Reviewed by | Date
Approved by | Date

Generic Catagory	Specific Component	Failure Mode	fn (Hz)	Median Capacity	Acceleration level [illegible] confidence		
					90%	95%	[illegible]
Inst. Panels and Racks	Generic	Relay Chatter	5-10	2.07g's	0.310g's	0.186g's	0.067g's
Inst. Panels and Racks	Generic	Breaker Trip	5-10	7.7g's	2.98g's	2.308g's	1.385g's
Control Panels and Racks	Generic	Relay Chatter	5-10	2.07g's	0.31g's	0.186g's	0.067g's
Control Panels and Racks	Generic	Breaker Trip	5-10	7.7g's	2.98g's	2.308g's	1.385g's
Control Panels and Racks	Generic	Structural	5-10	14.6g's	5.16g's	3.9g's	2.227g's
Relay Cabinets	Generic	Relay Chatter	5-10	2.07g's	0.31g's	0.186g's	0.067g's
Relay Cabinets	Generic	Relay Trip	5-10	7.7g's	2.98g's	2.308g's	1.385g's
Relay Cabinets	Generic	Structural	5-10	14.6g's	5.16g's	3.9g's	2.227g's

Table II. Cont'd

SARGENT & LUNDY
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Client LIL Co.
Project Shoreham
Proj. No. 6995-00 Equip. No.

Prepared by		Date
Reviewed by		Date
Approved by		Date

Generic Catagory	Specific Component	Failure Mode	fn (Hz)	Median Capacity	Acceleration level at confidence level		
					90%	95%	99%
Emergency A.C. Power units	Generator Control Panel	Relay chatter	30	0.95g's	0.695g's	0.64g's	0.54g's
Emergency A.C. Power Units	Engine Control Panel	Failed Relay	11	2.0g's	1.445g's	1.324g's	1.111g's
Emergency D.C. Power units	Batteries	case + plate failure	8	4.2g's	3.411g's	3.225g's	2.883g's
Switch Gear	4160 and 480 volt units	Relay chatter	5-10	2.07g's	0.310g's	0.186g's	0.067g's
Switch Gear	4160 and 480 volt units	Breaker Trip	5-10	7.7g's	2.98g's	2.308g's	1.385g's
Switch Gear	4260 and 480 volt units	Structural	5-10	14.6g's	5.16g's	3.90g's	2.227g's
Transformers	Generic	Structural	5-10	10.7g's	8.14g's	7.566g's	6.532g's
Local Inst. and Transmitters	Generic	Electrical Function	Rigid	37.8g's	24.8g's	22.29g's	17.819g's

Table II.

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Generic Catagory	Specific Component	Failure Mode	fn (Hz)	Median Capacity	Acceleration level at Confidence level		
					90%	95%	
Motor Control Centers	Generic	Relay Chatter	5-10	2.07g's	0.31g's	0.186g's	0.067g's
Motor Control Centers	Generic	Breaker Trip	5-10	7.7g's	2.98g's	2.308g's	1.385g's
Motor Control Centers	Generic	Structural	5-10	14.6g's	5.16g's	3.9g's	2.227g's
Breaker Panels	Generic	Breaker Trip	5-10	7.7g's	2.98g's	2.308g's	1.385g's
Breaker Panels	Generic	Structural	5-10	14.6g's	5.16g's	3.9g's	2.227g's

Table II. Cont'd

SARGENT & LUNDY
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CALC NO. CQD-
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.050
PAGE

Calc. No: CQD-67646
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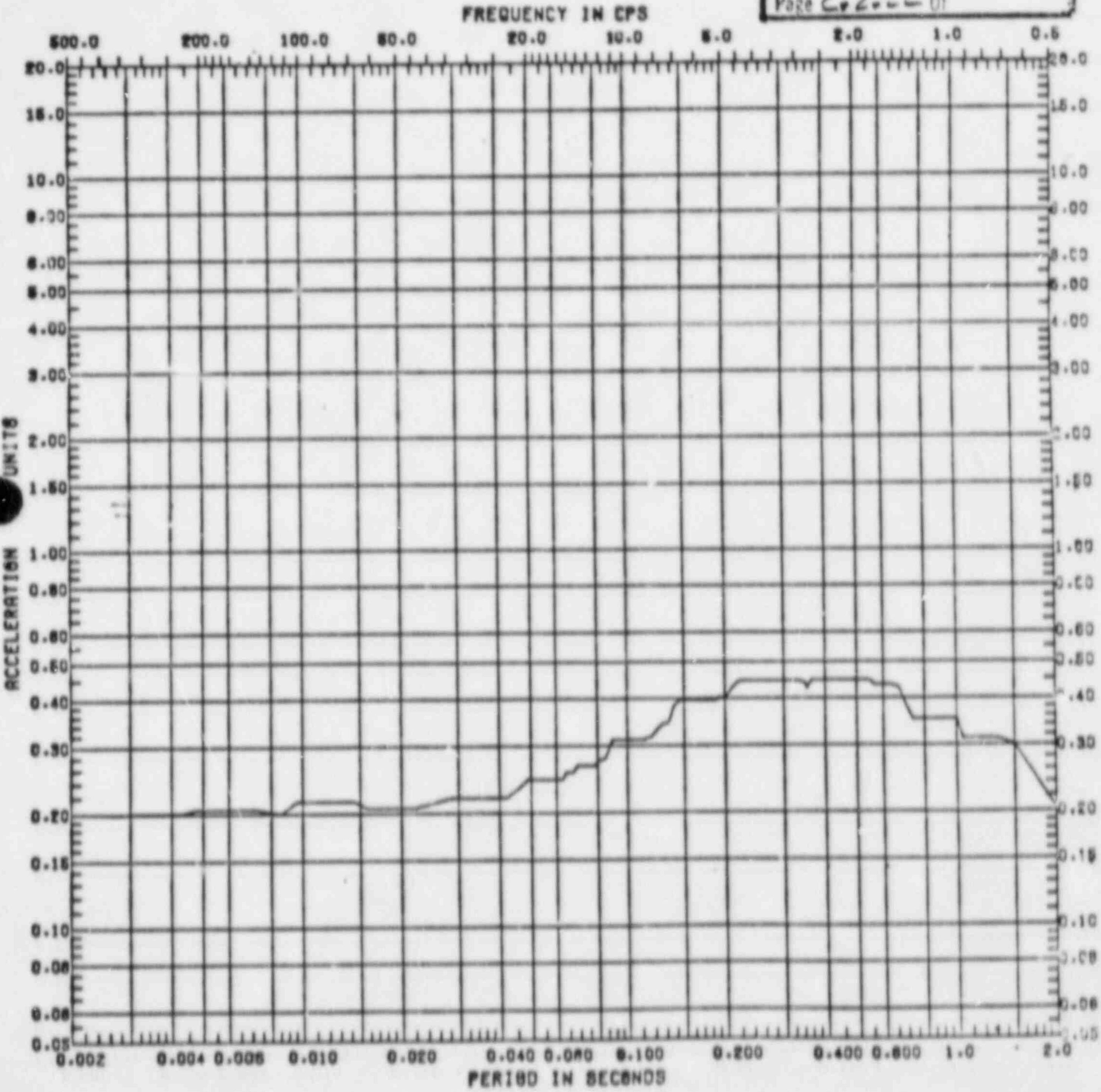

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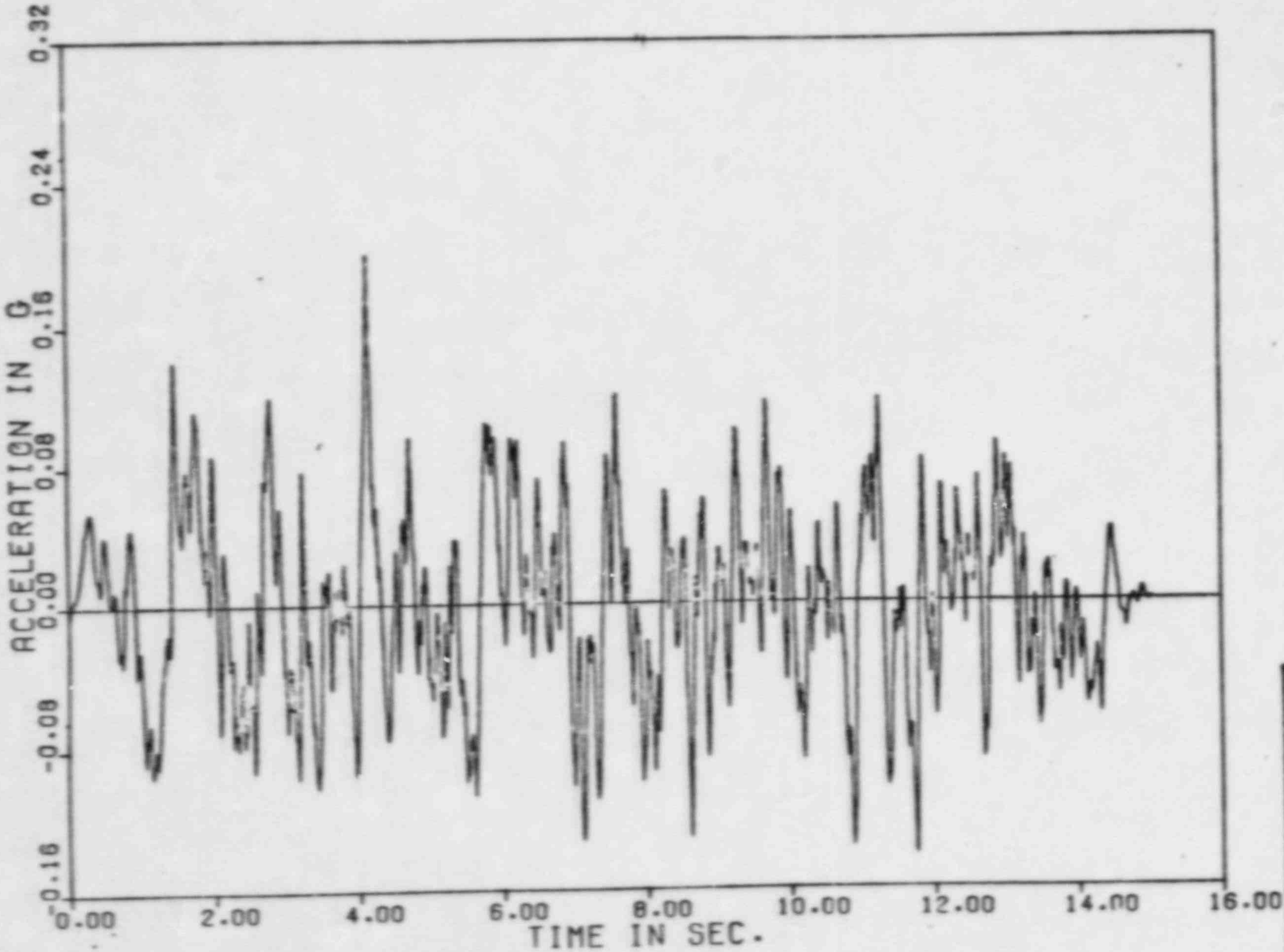
NODE BASE
1 DIRECTION HOR ANGLE N/A

SPECTRA NO. SSE 5%
ELEVATION GROUND
LOCATION EDG5

01 JUN 84



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BASE SSE 5% GROUND EDGS HOR N/A

SARGENT & LUNDY
ENGINEERS

01 JUN 84
SIKB

CALC NO. CQD-
PROJECT SHOREHAM-1 REV 0
PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING 0.050
PAOC

Calc. No: CQD-017040 Date: 3/1/84
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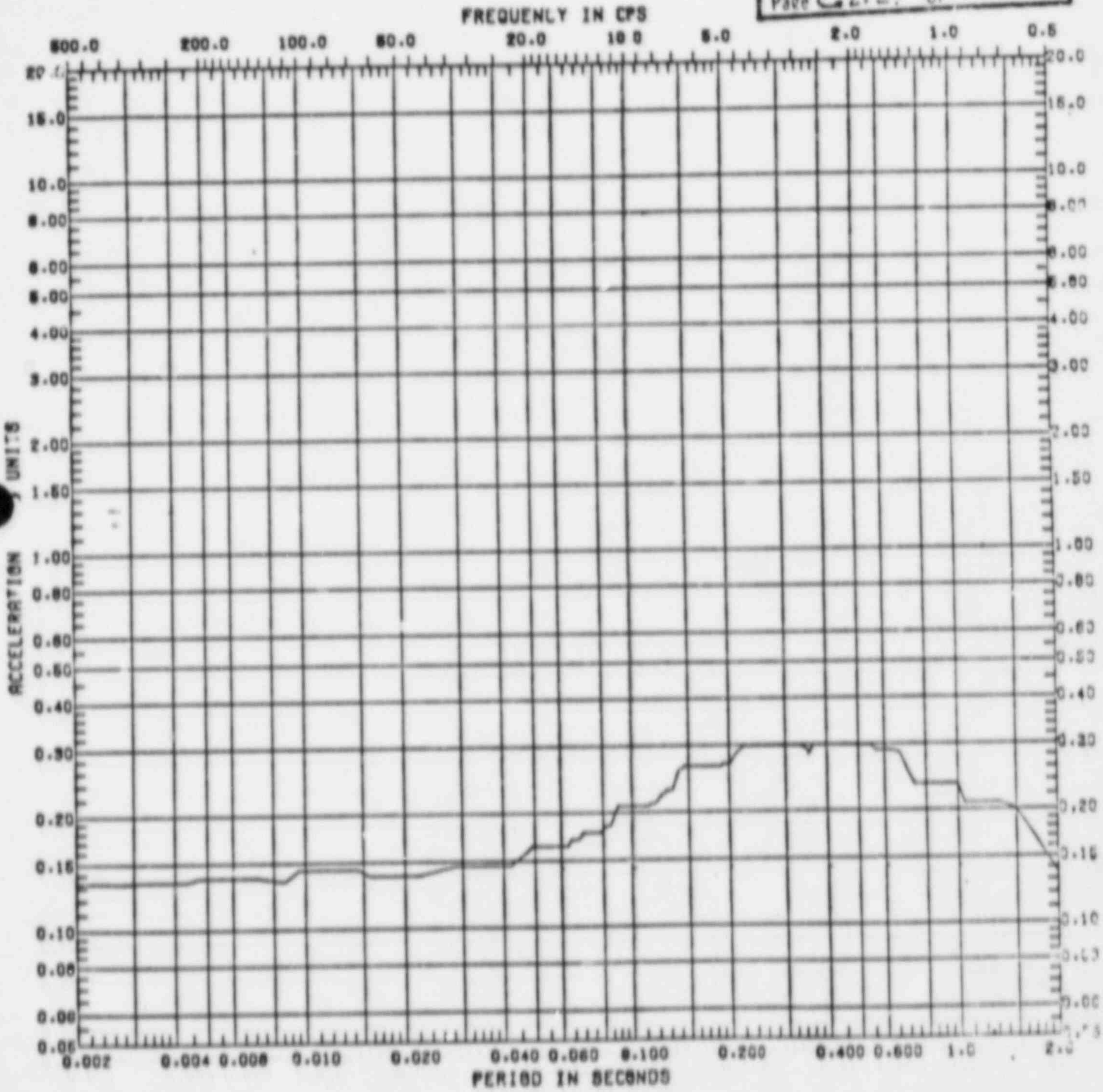
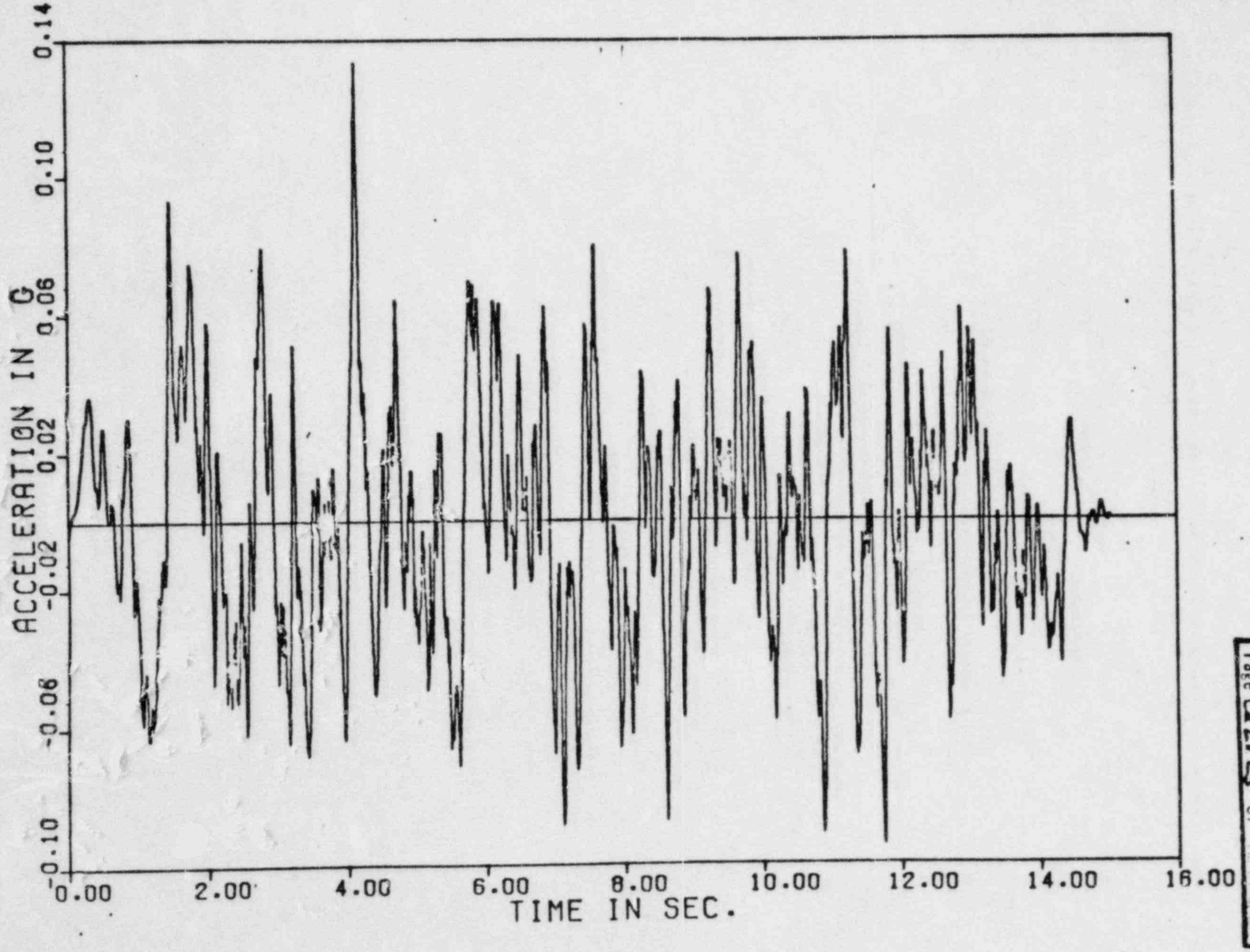


LOADING: SSE(5%) EDGS
NODE BASE
DIRECTION VER ANGLE N/A

SPECTRA NO. SSE 5%
ELEVATION GROUND
LOCATION EDGS

1

B
JUN 84



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BASE SSE 5% GROUND EDGS VER N/A

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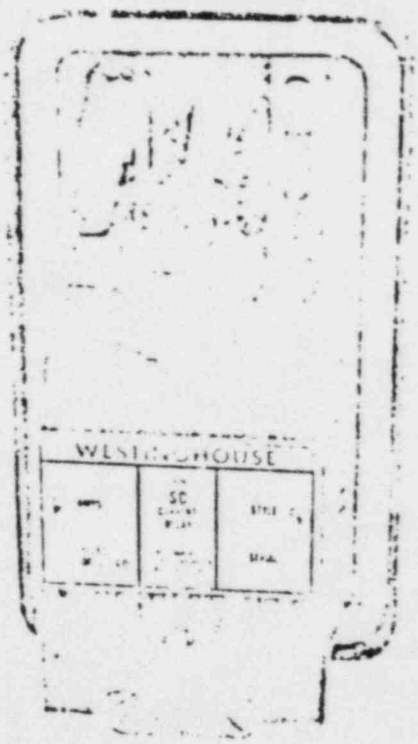
WESTINGHOUSE TYPE SC-1, SC-2 AND SV RELAYS



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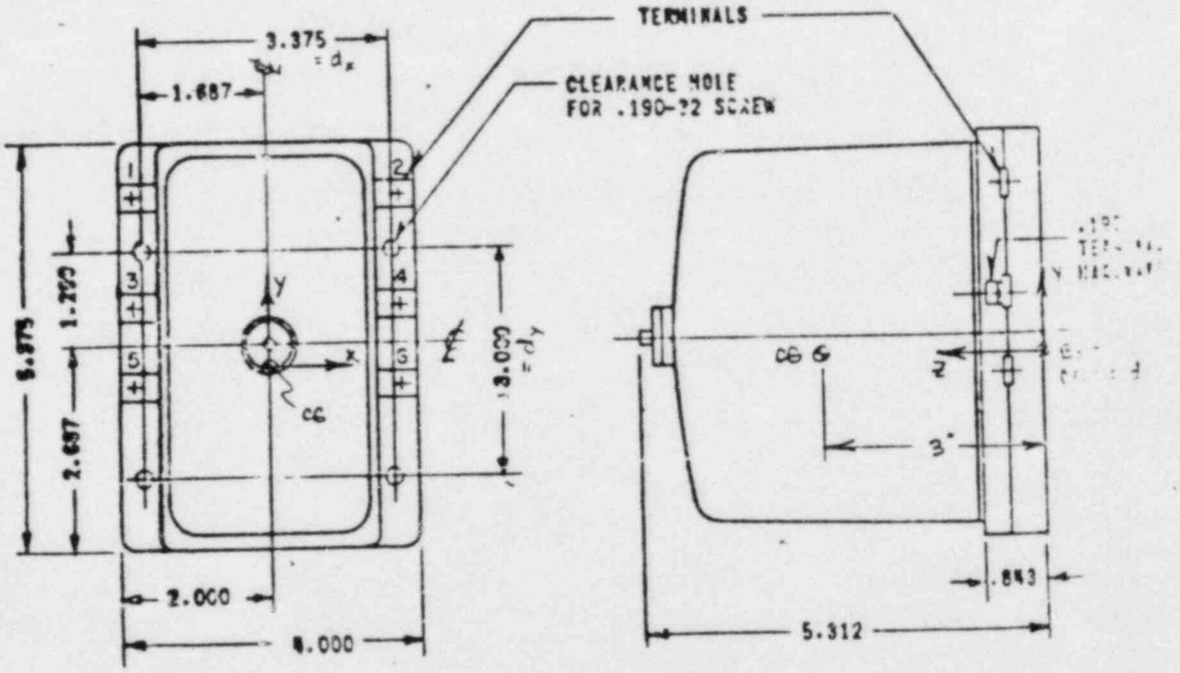
ASSUMPTIONS

1) For conservalism, assume relays are flexible in all 3-directions. 1.5 times peak response spectra values will be used for seismic accelerations. (Shoreham elevated response spectra will be used.
2) The front-connected case mounting pattern will be analyzed to envelope all case types, since it has the weakest mounting pattern.
3) All relay types are assumed to weigh 15 lbs or less; a weight of 15 lb. acting at a CG 3" out from the bolt centroid will be used to analyze the front-connected case.
4) SSE seismic accelerations will be used for qualification.



FRONT CONNECTED CASE All dimensions are in inches

Mounting screws are 0.19" ϕ $A_{bolt} = 0.0199 \text{ in}^2$ (tensile area)
$\Sigma A = 4(.0199) = .0796 \text{ in}^2$

DEAD WEIGHT LOADS

$F_x = F_z = 0$ $M_x = (15)(3) = 45$ in-lb
$F_y = .15$ lb ↓ $M_y = M_z = 0$

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SEISMIC LOADS

Seismic coefficients : $a_x = a_z = 1.5(2.6) = 3.9\,g$ $a_y = 1.5(0.36) = 0.54\,g$

$F_x = F_z = (15)(3.0) = 58.5$ lb
$F_y = (15)(0.54) = 8.1$ lb
$M_x = (8.1)(3) = 24.3$ in-lb
$M_y = (58.5)(3) = 175.5$ in-lb
$M_z = 0$

TOTAL LOADS

$F_x = F_z = 58.5$ lb
$F_y = 15 + 8.1 = 23.1$ lb ↓
$M_x = 45 + 24.3 = 69.3$ in-lb
$M_y = 175.5$ in-lb
$M_z = 0$

SCREW STRESSES

$$\sigma_z = \frac{F_z}{\Sigma A} + \frac{M_x}{2 d_y A_{crit}} + \frac{M_y}{2 d_x A_{crit}}$$

$$= \frac{58.5}{.0736} + \frac{69.3}{2(3)(.0100)} + \frac{175.5}{2(3.375)(.0177)} = 2622 \text{ psi}$$

$$\tau = \frac{\sqrt{F_x^2 + F_z^2}}{\Sigma A} = \frac{\sqrt{58.5^2 + 58.5^2}}{.0736} = 790 \text{ psi}$$

$$\sigma_{max} = \frac{\sigma_z}{2} + \sqrt{\left(\frac{\sigma_z}{2}\right)^2 + \tau^2} = 2842 \text{ psi}$$

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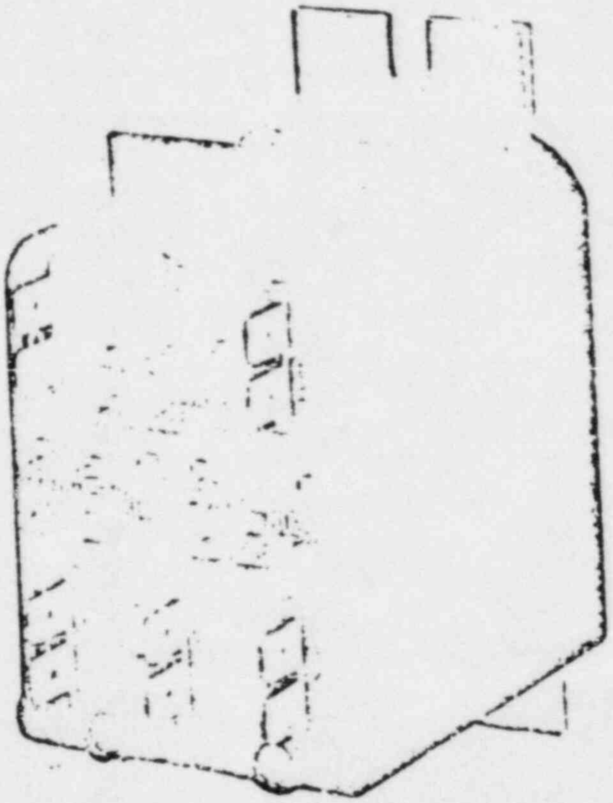
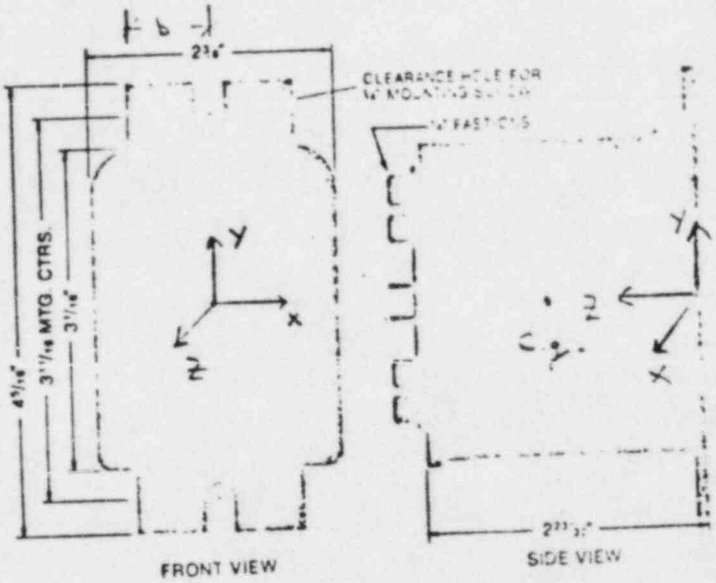
Vapor Corp Relay





Weight .... 15 oz

Assumptions for Analysis

1) The relay will be considered rigid, therefore the acceleration used will be 1.5 times the maximum peak ZPA values for the horizontal and vertical directions.

2) From the geometry of the relay, the center of gravity will be assumed to be in line with the mounting screws centroidal axis x, and y axis and midway between the front and base of the relay.

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3) Mounting screw material will be assumed to be commercial grade A-36 steel

Geometric Properties.

The eccentricities from the center of gravity to the mounting screw pattern centroid is as follows

$$x = 0 \quad y = 0 \quad z = \tfrac{1}{2}(2^{23}/32) = 1.36 \text{ in.}$$

Seismic Coefficients

ZPA accelerations (Envelope of OBE/SSE conditions)

$g_v = 0.15$ Vertical $g_h = 0.3$ horizontal

Screw Loads at pattern centroid

Deadweight Loads

$$F_y = P = 15 \text{ oz use } 1 \text{ lb.}$$
$$M_x = F_y z = 1(1.36) = 1.36 \text{ in-lbs.}$$

Seismic Loads

$$F_x' = F_z' = P(g_h \ast 1.5)$$
$$= 1(.3 \ast 1.5) = .45 \text{ lbs}$$

$$F_y' = P(g_v \ast 1.5)$$
$$= 1(.15 \ast 1.5) = .225 \text{ lbs}$$

$$M_x' = F_y' z + F_z' y$$
$$= .225(1.36) + 0$$
$$= .306 \text{ in-lbs}$$

Form GO-3.06.1 Rev. 2

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$M_y' : F_x' z + F_z' x$
$= .45(1.36) + 0$
$: .612$ in-lbs

$M_z' : F_y' x + F_z' y$
$= 0$

<u>Total loads</u>

$F_x = .45$ lbs
$F_y = 1 + .225 = 1.225$ lbs
$F_z : .45$ lbs
$M_x = 1.36 + .306 = 1.666$ in-lbs
$M_y : .612$ in-lbs
$M_z - 0$

<u>Screw Stresses</u>

Moments can be resolved into equivalent forces:

M_x moment $= F_1 d$
$F_1 = M_x/d = 1.666/(3\,{}^{11}/_{16}) = .45$ lbs

M_y moment $= F_2 b$
$F_2 = M_y / b = .612 / .25$ (b distance is distance from screw centroid to bracket edge (assumed))
$= 2.448$ lbs

Screw Tensile Stress $A : .0317\ in^2$

$$\sigma_T = F_z/2A + (F_1 + F_2)/A = \frac{.45}{2(.0317)} + \frac{.45 + 2.448}{.0317} = 95.5\ psi$$

Screw Shear Stress

τ = only 1 screw can resist the Fy force
$= F_y/A + F_x/2A = 1.225/.0317 + .45/2(.0317) = 45.7$ psi

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Calcs. For	
☐ Safety–Related	☐ Non–Safety–Related

Calc. No. CSD-214046	
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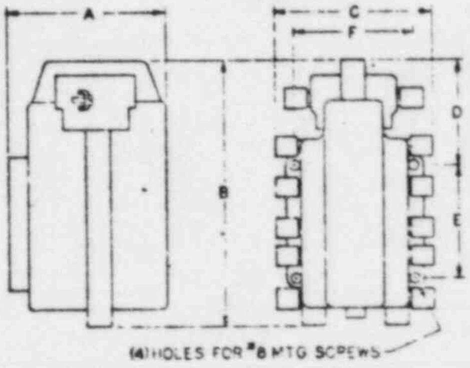
G.V & DRL Relays - Manufactured by Square D - Model 8501 & 7001

APPROXIMATE DIMENSIONS AND SHIPPING WEIGHTS



(4) HOLES FOR #8 MT'G SCREWS

Figure 1

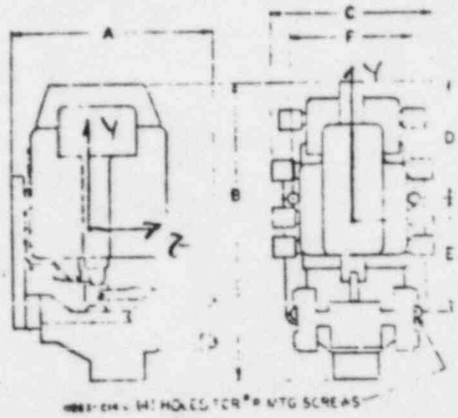
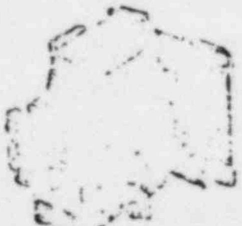


Figure 2

Class	Type	Figure Number	Dimensions — Inches						Weight (Lbs.)
			A	B	C	D	E	F	
	DO-02, -20, -22	1	3 5/32	3 7/32	3	57/64	2 1/8	2 1/4	1 1/2
	DO-40, -42	1	3 5/32	3 33/64	3	1 13/64	2 1/8	2 1/4	1 1/2
8501	DO-44, -60, -62, -64, -80, -82	1	3 5/32	5 1/64	3	2 1/64	2 1/8	2 1/4	2
	DDO-22, -42	2	3 17/32	5 7/8	3 1/16	2 13/64	2 1/8	2 1/4	2
	DEO-22, -42	2	3 13/16	5 9/16	3 1/4	2 13/64	2 1/8	2 1/4	2

Class 7001
Type DO-42



Class 8501
Type DO-22

Assumptions

1) It is assumed that the equipment is flexible in all directions therefore the peak [illegible] multiplied by 1.5 will be used in the [illegible] of seismic Loads
2) The CG of the unit is assumed along the bolt centroid and at coordinates X = 0, Y = .5, Z = 2" from the bolt centroid.
3) screw material is assumed as A-36 [illegible]

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SARGENT & LUNDY
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Calcs. For

☐ Safety-Related ☐ Non-Safety-Related

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Client
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Prepared by Date
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Accelerations (Envelope of OBE & SSE)

horizontal = 2.6 g's
Vertical = .36 g's

Weight Loads (Weights & dimensions of the largest relays shown in previous page are used)

$F_y = 2$ lbs
$M_x = F_y z = 2 \times 2 = 4$ in-lbs

Seismic Loads

$F_x = F_z = 2 \times 2.6 \times 1.5 = 7.8$ lbs
$F_y = 2 \times .36 \times 1.5 = 1.08$ lbs
$M_x = F_y z + F_z y = (1.08 \times 2.0) + (7.8 \times .5) = 6$ in-lbs
$M_y = F_x z = 7.8 \times 2 = 15.6$ in-lbs
$M_z = F_x y = 7.8 \times .5 = 3.9$ in-lbs

TOTAL LOADS

$F_x = F_z = 7.8$ lbs
$F_y = 4 + 1.08 = 5.08$ lbs
$M_x = 4 + 6 = 10$ in-lbs
$M_y = 15.6$ in-lbs
$M_z = 3.9$ in-lbs

Bolt Stresses 4 # 8 screws - Area = .0139 in^2

Tensile :

due to F_z: $\sigma = \frac{F_z}{4A_b} = \frac{7.8}{4 \times .0139} = 140$ psi

Due to M_x (M_x analyzed as a couple based on assumption that only 2 screws resist the total moment.)

$10 = F \times d$ or $10 = F \times 2.125 \rightarrow F = 4.705$ lbs

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Calcs. For
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$\sigma = \frac{4.705}{.0139} = 339$ psi

due to My (My analyzed as a couple based on assumption that only 2 bolts resist total moment.

$15.6 = Fd$ or $15.6 = F \times 2.25 \rightarrow F = 6.93$ lbs

$\sigma = \frac{6.93}{.0139} = 499$ psi

TOTAL TENSILE = 140 + 339 + 499 = 978 PSI

shear = due to $(F_x \, \& \, F_y)$

$\tau = \frac{(7.8^2 + 5.08^2)^{1/2}}{4 \times .0139} = 167$ psi

due to Mz: (MZ is analyzed as a couple based on assumption that only 2 diagonal screws resist the total moment)

$3.9 = Fd$ or $3.9 = F \times (2.125^2 + 2.25^2)^{\frac{1}{2}} \rightarrow F = 1.26$ lbs

$\tau = \frac{1.26}{.0139} = 91$ psi

TOTAL SHEAR = 167 + 91 = 258 psi

The above calculated stresses are very small therefore the screws are considered adequate

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Calcs. For

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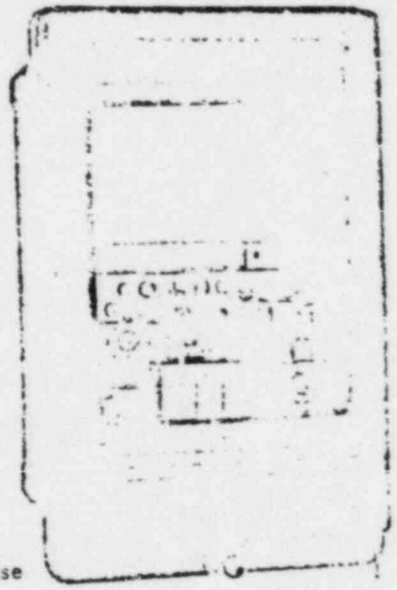


FT-11 Case

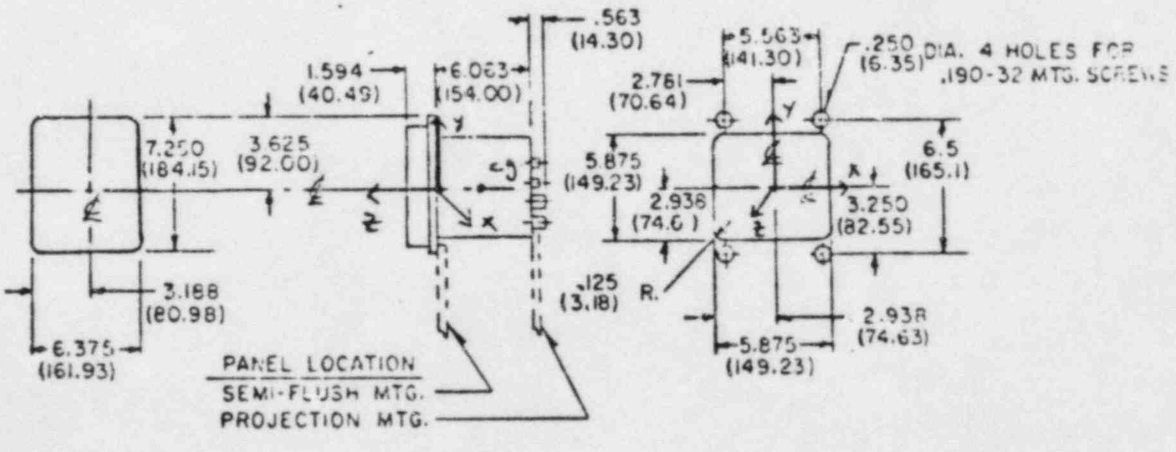
Case Size	Weight: Lbs. (kg)	
	Net	Shipping
* FT-11	8 (3.6)	11 (5)

Westinghouse Relays

TYPE CW POWER RELAY

TYPE CV RELAYS

*weight includes relay weight



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Assumptions For Analysis

1) The relay will be considered flexible, therefore the accelerations used will 1.5 times the maximum peak values for the horizontal and vertical directions.

2) From the geometry of the relay, the center of gravity will be assumed to be in line with the mounting screw pattern centroid along the x and y axis

3) The relay will be considered to be semi-flush mounted.

4) The center of gravity along the z axis will be assumed to be half the distance between the semi-flush mounting and the back of the relay.

5) Mounting screw material will be assumed to be of commercial grade A-36 steel

6) Case Type FT-11 is assumed to be the worst case for screw analysis and will be analyzed.

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Calcs. For

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Geometric parameters

The mounting screw pattern is symmetrical and uses 4 - #10-32 mounting screws. (.190 in dia)

The center of gravity eccentricities from the mounting screw centroidal axis is as follows.

$x = 0 \quad y = 0 \quad z = \frac{1}{2}(6.063 + .563) = 3.312$ in

Seismic Coefficients

Peak accelerations (Envelope of OBE/SSE conditions)

$gh = 2.6\,g$ horizontal $\quad gv = 0.38\,g$ vertical

Screw Loads at pattern centroid

Deadweight Loads : $F_y = 8$ lbs

$$M_x = F_y z + F_z y$$
$$= 8(3.312) + 0 = 26.504 \text{ in-lbs}$$

Seismic Loads : $F_x' = F_z' = F_y(gh * 1.5)$

$$= 8(2.6 * 1.5) = 31.2 \text{ lbs}$$

$$F_y' = F_y(gv * 1.5)$$
$$= 8(0.38 * 1.5) = 4.56 \text{ lbs}$$

$$M_x' = F_y' z + F_z' y$$
$$= 4.56(3.313) + 0 = 15.107 \text{ in-lbs}$$

$$M_y' = F_x' z + F_z' x$$
$$= 31.2(3.312) + 0 = \text{[illegible]}$$

Form OO-3.08.1 Rev. 2

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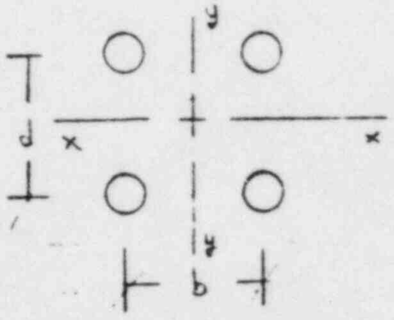
$M_z' = F_x' y + F_y' x$
$= 0$

Total loads:

$F_x'' = 31.2$ lbs
$F_y'' = 8 + 4.56 = 12.56$ lbs
$F_z'' = 31.2$ lbs
$M_x'' = 26.504 + 15.107 = 41.611$ in-lbs.
$M_y'' = 102.27$ in-lbs
$M_z'' = 0$

Screw Stresses

Moments can be resolved into equivalent Forces:



$b = 5.563$ in $d = 6.5$ in.

M_x'' moment $= F_1 d$
$F_1 = M_x''/d$
$= 41.611/6.5 = 6.4$ lbs

M_y'' moment $= F_2 b$
$F_2 = M_y''/b$
$= 102.37/5.563 = 18.57$ lbs

Screw tensile Stress

Area of Screw = 0.0199 in^2

$\sigma_{Tension} = F_z/4A + \frac{F_1 + F_2}{A}$
$= 31.2/4(.0199) + (6.4 + 18.57)/.0199$
$= 1647$ psi

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ENGINEERS
CHICAGO

Calcs. For

☐ Safety–Related ☐ Non–Safety–Related

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$$\tau_s = \left(F_x^2 + F_y^2\right)^{1/2} / 4A$$

$$= (31.2^2 + 12.56^2)^{1/2} / 4(.0199) = 422.5 \text{ psi}$$

Form GO 3.08.1 Rev. 2

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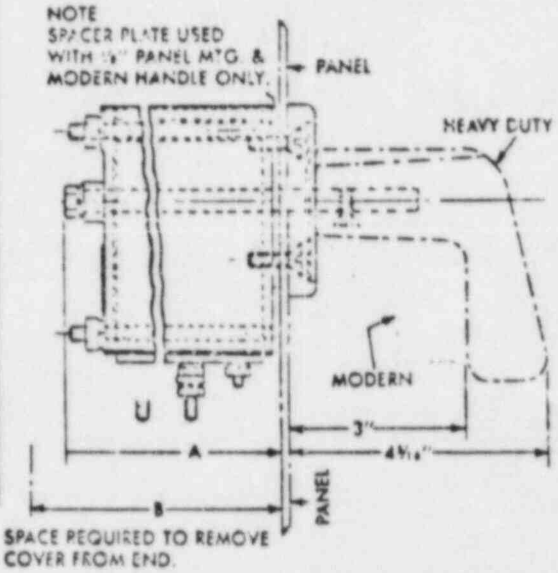
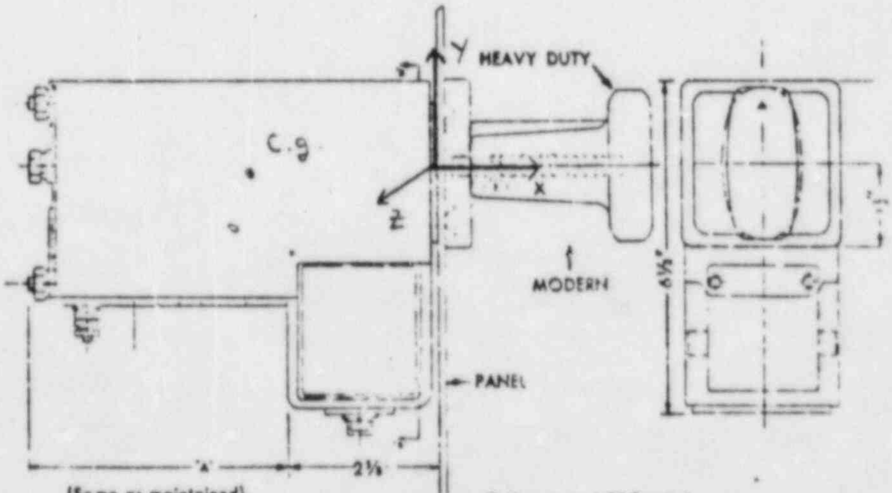
Allis-Chalmers Type 210 Control Switches

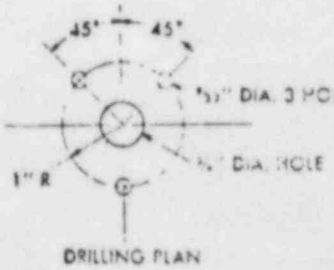
STANDARD DUTY 210				
	Maintained		Spring Return	
No. of Stages	Dim. A (in.)	Dim. B (in.)	Dim. A (in.)	Dim. B (in.)
2	3⅝	7¼	4⅝	7¼
3	4⁷⁄₁₆	8⅞	5¼	8⅞
4	5¼	10½	6⁵⁄₁₆	10½
5	6¹⁄₁₆	12⅝	6⅞	12⅝
6	6⅞	13¾	7¹¹⁄₁₆	13¾
7	7¹¹⁄₁₆	15⅜	8½	15⅜
8	8½	17	9⁵⁄₁₆	17
9	9⁵⁄₁₆	18⅝	10⅛	18⅝
10	10½	20¼	10¹⁵⁄₁₆	20¼
11	10¹⁵⁄₁₆	21⅞	11¾	21⅞
12	11¾	23½	12⁹⁄₁₆	23½
13	12⁹⁄₁₆	25⅛	13⅜	25⅛
14	13⅜	26¾		



Outline of standard and mill duty 210 switches.



Coil operated 210 switch.



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Client		Prepared by	Date
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Assumptions For Analysis

1) The control switches will be considered flexible, therefore the accelerations used will be 1.5 times the maximum peak values for the horizontal and vertical directions.

2) Since the switch can be in both horizontal and vertical service the envelope of the [illegible] vertical and horizontal response spectras will be used.

3) From the data provided the coil operated 210 switch with the largest dimensions will be used.

4) Screw material will be assumed to be of commercial grade A-36 steel.

5) Due to the low weight of the switch the center of gravity, for the switch will be assummed to lie along the mounting screws centroid axis (y-z directions)

6) The center of gravity of the switch along the x-axis will be assumed to be at approximately the mid point of the unit.

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CHICAGO

Calcs. For

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Geometric properties

Mounting screw holes are 9/32" dia holes
Mounting screws will be approximately 1/4" dia [illegible]

The switch c.g. along the x-axis is calculated to be

$\simeq (A + 2\,5/8 + 4\,5/16)/2 = (13.375 + 2.625 + 4.3125) \div 2 = 10.15$ in
from end

or $\simeq 3.20$ in from mounting screws.

Therefore eccentricities from mounting screw centroid are
$= \quad x = 3.20 \quad y = 0 \quad z = 0$

Seismic Coefficients

Peak accelerations (envelope of [illegible] conditions)
$g_h = 2.6g$ Horizontal $\quad g_v = 0.35g$ vertical $\quad$ [2.6g will be used]

Mounting screw loads at pattern centroid.

Dead weight loads $\quad$ For horizontal mounting (worst case)

$F_y = P = 5$ lbs
$F_x = F_z = 0$
$M_z = F_y x + F_x y$
$= 5(3.20) = 16.0$ in-lbs
$M_x = M_y = 0$

Seismic loads

$F_x' = F_z' = Pg(1.5) = 5(2.6)(1.5) = 19.5$ lbs
$F_y' = Pg(1.5) = 19.5$ lbs

$M_x' = 0$

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$M_y' : F_x' z \; F \; F_z' x$

$= 0 + 19.5(3.20) = 62.40$ in-lbs

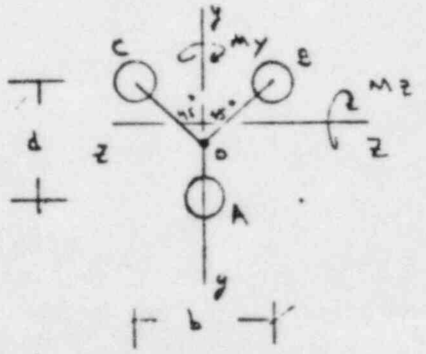
$M_z' = F_y' x + F_x' y$
$= 19.5(3.20) + 0 = 62.40$ in-lbs

Total loads

$F_x = 19.5$ lbs
$F_y = 5 + 19.5 = 24.5$ lbs
$F_z = 19.5$ lbs
$M_x = 0$
$M_y = 62.40$ in-lbs
$M_z = 16.0 + 62.40 = 78.40$ in-lbs

Screw Stresses

Moments can be resolved into equivalent Forces



radius from point 0 = 1 inch
∴ d = 1 + 1 cos 45 = 1.707 in

b = 2 (1 cos 45) = 1.414 in

Moment My is assummed to be resisted by screws E and C

$M_y = F b \quad F_T = M_y / b = 62.40 / 1.414 = 44.13$ lbs

Moment Mz is assummed to be resisted by screw A and C only
$M_z = F d \quad F_T = M_z / d =$ [illegible]5.40/1.[illegible]7 = 45.[illegible]

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Calcs. For

☐ Safety-Related ☐ Non-Safety-Related

Calc. No. CSD-61042

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Client

Project

Proj. No. 6945-00 Equip. No.

Prepared by Date

Reviewed by Date

Approved by Date

Screw tensile stress

Area A of 1/4" screw = .0217 in²

Tensile stress $\sigma_T = \frac{F_X}{3A} + \frac{F_T}{A}$

$= \frac{19.5}{3(.0217)} + \frac{42.5 + 45.0}{.0217}$ = 2045 psi

Shear Stress $\tau = (F_Y^2 + F_Z^2)^{1/2} / 3A$

$= (24.5^2 + 19.5^2)^{1/2} / 3(.0217)$ = 329 psi

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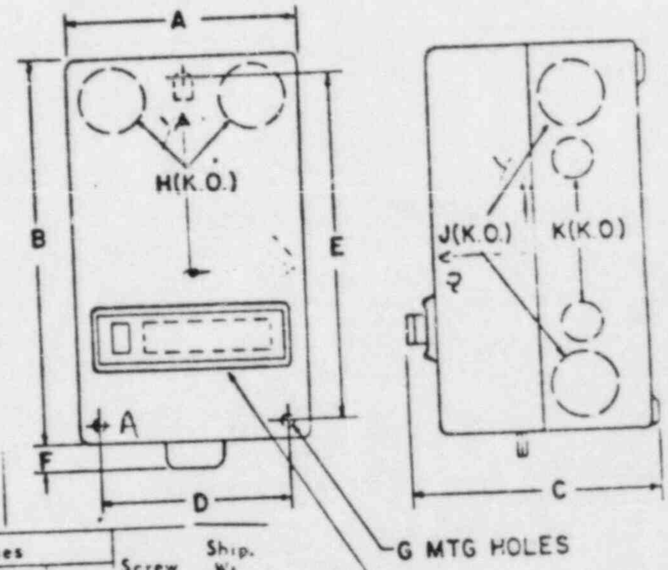
Client	
Project SHNPP	
Proj. No. 6995-00	Equip. No.

Prepared by Jon Papadopoulos	Date
Reviewed by	Date
Approved by	Date

Immersion Heater Contactor — Cutler-Hammer Mod'l A10DNY1

Catalog Number
A10DNY1 0210
A10DNY1 0610

NEMA 1 ENCLOSED – SIZES 0 THRU 4



Dimensions In Inches						Screw Size G	Ship. Wt. Lbs.
Wide A	High B	Deep C	Mounting		F		
			D	E			
11 3/8	21	8 1/4	9 1/2	17	5/8	3/8-16	42

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Assumptions:

1) It is assumed that the equipment is flexible in all 3-directions and peak accelerations multiplied by a factor of 1.5 [illegible] in calculating the seismic loads
2) It is assumed that the CG of the equipment is along the bolt centroid and at z = 4" from the bolt centroid
3) It is assumed that the [illegible] enclosure and for conservatism the data [illegible] to the [illegible]
4) Assume screw material to be A-36 or equivalent

Bolt centroid (from Bolt A) is at x = 9.5/2 = 4.75
z = 0 ; y = 17/3 = 5.67"

Accelerations (Envelope of OBE & SSE)
Horizontal = 2.6 g's
Vertical = .38 g's

Loads for Bolt requirements (at bolt centroid)

Weight Loads: Fy = 42 lbs
Mx = Fyz = 42x4 = 168 in-lbs

Seismic Loads
Fx = Fz = 42x(2.6x1.5) = 164 lbs
Fy = 42x(.38x1.5) = 24 lbs
Mx = Fyz = 24x4 = 9.6 in-lbs
My = Fxz = 164x4 = 656 in-lbs
Mz = 0

TOTAL LOADS: Fx = Fz = 164 lbs, Fy = 42+24 = 66 lbs
Mx = 168+9.6 = 254 in-lbs My = 656 in-lbs

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ENGINEERS
CHICAGO

Calcs. For

Y Safety-Related ☐ Non-Safety-Related

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Reviewed by Date
Approved by Date

Bolt stresses

3 - $3/8$" ϕ Bolts Bolt Area = .078 in^2

Tensile:

due to Fz: $\frac{Fz}{3A_b} = \frac{164}{3 \times .078}$ = 701 psi

due to Mx: (assume that only 2 Bolts resist total moment

M_x = 264 in-lbs = F × d or 264 = F × 17 → F = 15.53 lbs/bolt

$\sigma = F/A_b = \frac{15.53}{.078}$ = 199 psi

Due to My: (assume 2 Bolts resist total moment)

F = 656/9.5 = 69 lbs/Bolt

$\sigma = \frac{69}{.078}$ = 885 psi

TOTAL TENSILE = 701 + 199 + 885 = 1785 psi

shear stress:

$\tau = \frac{(Fz^2 + Fy^2)^{1/2}}{A_b} = \frac{(164^2 + [illegible]^2)^{1/2}}{3 \times .078}$ = 708 psi

Allowables (AISC for A-36)

Tensile = .44 Su = .44 × 58 = 25.5 ksi
Shear = .23 Su = .23 × 58 = 13.3 ksi
for combined Tension & Shear = .44 Su = .44 × 58 = 25.5 ksi

The stresses are very small and below the allowables. Therefore the screws are adequate

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SARGENT & LUNDY
ENGINEERS
CHICAGO

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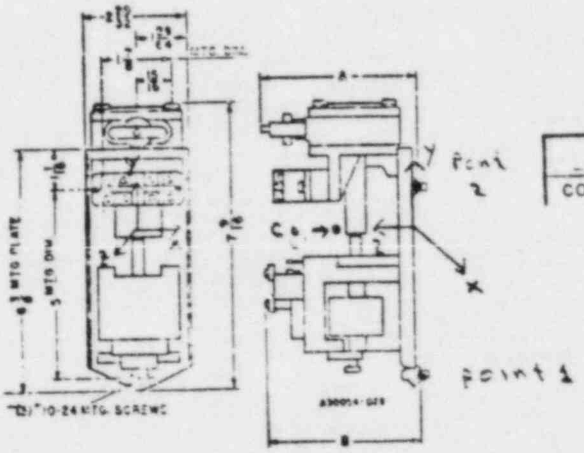
Square D Company Timing Relays

Type CO-5E, CO-21E



APPROXIMATE DIMENSIONS
AND
SHIPPING WEIGHTS

Type	A	B
CO1, 2, 5, 21, 22, 25 D & E	4⅛	4½

Types BO and CO
Weight — 5 Lbs.

Assumptions for analysis:

1) The relay will be considered Flexible, therefore the acceleration used will be 1.5 times the maximum peak values for the horizontal and vertical directions.

2) From the geometry of the relay the center of gravity will be assumed to be at the bolt pattern centroid and a distance half the depth of the relay from the relay base.

4) Screw material will be assumed to be of commercial grade A-36 Steel.

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Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

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Client LILCO
Project Shoreham
Proj. No. 6995-00 Equip. No.

Prepared by Date
Reviewed by Date
Approved by Date

Geometric parameters

Bolt pattern centroid is up ⅔ the distance between points 1 and 2 from point 1

The center of gravity of the relay is ½ E or ½ (4.25)
∴ the center of gravity eccentricities from the screw pattern centroid are x = 0 y = 0 z = 2.125 in.

Seismic Coefficients

Peak accelerations (Envelope of OBE/SSE conditions)

g_h = 2.6 g Horizontal, g_v = 0.38 g vertical

Screw loads at pattern centroid

Deadweight Loads: $F_y = P = 5$ lbs
$M_x = F_y z + F_z y = 5(2.125) = 10.625$ in. lbs

Seismic Loads

$F_x' = F_z' = P(g_h * 1.5)$
$= 5(2.6 * 1.5) = 19.5$ lbs

$F_y' = P(g_v * 1.5)$
$= 5(0.38 * 1.5) = 2.85$ lbs

$M_x' = F_y' z + F_z y$
$= 2.85(2.125) = 6.056$ in. lbs

$M_y' = F_x' z + F_z x$
$= 19.5(2.125) = 41.438$ in. lbs

$M_z = F_y' x + F_z' y$
$= 0$ in. lbs

Form GO-3.08.1 Rev. 2

SARGENT & LUNDY
ENGINEERS
CHICAGO

Calcs. For

☒ Safety-Related ☐ Non-Safety-Related

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Client LILco
Project Shoreham
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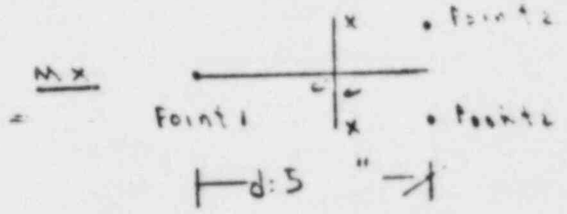
Total Loads: $F_x = 19.5$ lbs
$F_y = 5 + 2.85 = 7.85$ lbs
$F_z = 19.5$ lbs
$M_x = 10.625 + 6.026 = 16.651$ in lbs
$M_y = 41.438$ in-lbs
$M_z = 0$ in-lbs

Screws Stresses

Moments can be resolved into equivalent forces

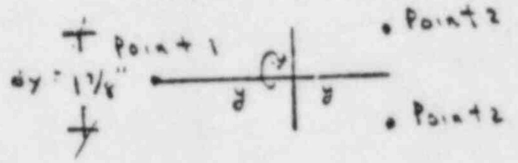
Mx



Total moment Mx will be assumed to be resisted by the screw at Point 1 and 1 screw at point 2

F tension point 2 = M_x/d
= 16.651/5
= 3.34 lbs

My



Total moment My will be assumed to be resisted by the screws at points 2 only

F Tension Point 2 = M_y/d'
= 41.438/1.875
= 22.1 lbs

Screw Tensile stress

A = Screw area = 0.0174 in² for #10-[illegible] screws

$$\sigma_{Tension} = \left[\frac{F_z}{3A} + \left[\frac{F_{tension\ point\ 1} + F_{tension\ point\ 2}}{A}\right]\right]$$

$$= \left[\frac{19.5}{3(.0174)} + \frac{[3.34 + 22.1]}{.0174}\right] = 1836 \text{ psi}$$

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τ shear $= (F_y^2 + F_x^2)^{1/2} / 3A$

$= (7.85^2 + 19.5^2)^{1/2} / 3(.0174)$

$= 402.7$ psi

Screws for relay are acceptable.

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Client LILCo
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Prepared by Bruce M. Long Date 5-31-84
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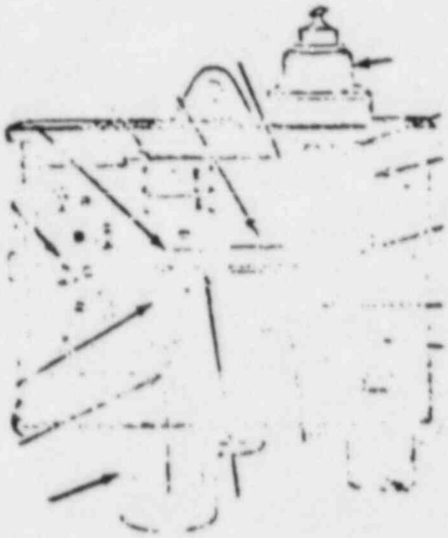
Thermostatic Controller

Log #0803101130



MF. : Barber Colman Co.
Model #: TP-222

Note: Morrison-Knudson Co. supplied LILCo with representative E.D.G.S. system drawings from a similar E.D.G.S. Morrison-Knudsen dwg. #6041-F04001, rev. B depicts this device,

The thermostatic controller on the Shoreham E.D.G.S's work in conjunction with the shutter actuator in regulating air flow across the E.D.G.S. radiator.

At this time we cannot verify the particular vendor/model for this device. The controller depicted above is per the Morrison-Knudsen supplied dwg.

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Prepared by
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Date
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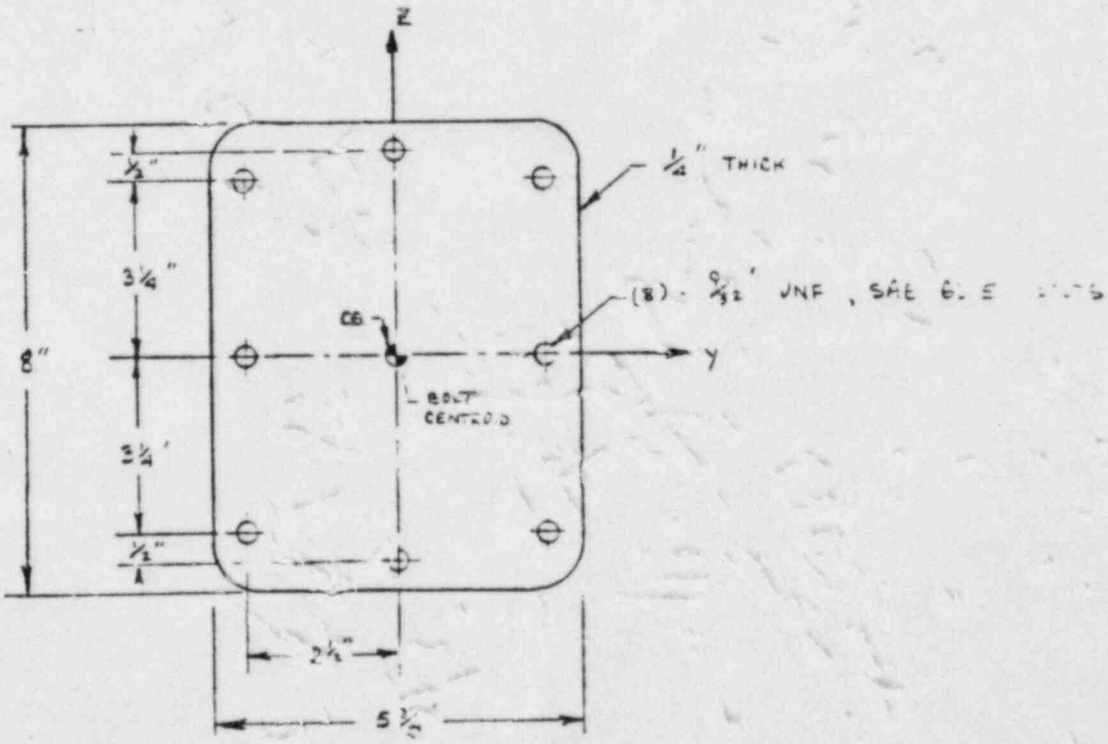
Date

CRANKING LOCK-OUT SWITCH MOUNTING

LOE NO 0803162005



Center of gravity (CG) is located approximately 3" out from mounting face plate above bolt centroid.

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BOLT PROPERTIES

$\text{AREA} = (9/32)^2 \frac{\pi}{4} = 0.062 \text{ IN}^2 \quad = A$

$\text{TOTAL BOLT AREA} = 8 \times 0.062 \text{ IN}^2 = 0.497 \text{ IN}^2 = \Sigma A$

INERTIA PROPERTIES

$I_{yy} = 4(0.062)(3.25)^2 + 2(0.062)(3.75)^2 = 4.363 \text{ IN}^4$

$I_{zz} = 6(0.062)(2.5)^2 = 2.325 \text{ IN}^4$

$J = I_{yy} + I_{zz} = 4.363 + 2.325 = 6.688 \text{ IN}^4$

$C_{y\,max} = 2.5 \text{ IN}$

$C_{z\,max} = 3.25 \text{ IN}$

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SEISMIC LOADS

FX = 10LBS (2.8) = 28 LBS
FY = 10LBS (2.8) = 28 LBS
FZ = 10LBS (7.1) = 71 LBS
MX = 0.0
MY = 71 (3) = 213 IN-LBS
MZ = 28(3) = 84 IN-LBS

ENGINE VIBRATION LOADS

ACCEL LEVELS PER "DATA SUPPLEMENT TO RESPONSE OF AN EMD 645 ENGINE..."

FX = 10 (.95) = 9.5 LBS
FY = 10 (.85) = 8.5 LBS
FZ = 10 (1.0) = 10 LBS
MX = 0.0
MY = 10 (3) = 30 INLBS
MZ = 8.5(3) = 25.5 IN LBS

TOTAL LOADS

FX = 28 + 9.5 = 37.5 LBS
FY = 28 + 8.5 = 36.5 LBS
FZ = 71 + 10 = 81 LBS
MX = 0.0
MY = 213 + 30 = 243 INLBS
MZ = 84 + 25.5 = 109.5 IN LBS

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TENSILE STRESS

$$\sigma_t = \frac{FX}{\Sigma A} + \frac{M_y C_z}{I_{yy}} + \frac{M_z C_y}{I_{zz}}$$

$$= \frac{37.5}{0.497} + \frac{243(3.75)}{4.363} + \frac{109.5\ (2.5)}{2.325} = 375\,\text{PSI}$$

SHEAR STRESS

$$\tau_y = \frac{F_y}{\Sigma A} + \cancelto{0}{\frac{M_x C_{y\,max}}{J}} = \frac{36.5}{.497} = 74\ \text{PSI}$$

$$\tau_z = \frac{F_z}{\Sigma A} + \cancelto{0}{\frac{M_x C_{z\,max}}{J}} = \frac{81}{.497} = 163\,\text{PS}$$

$$\tau_{max} = \tau_y + \tau_z = 237$$

PRINCIPAL STRESS

$$\frac{375}{2} + \sqrt{\left(\frac{375}{2}\right)^2 + (237)^2} = 490\,\text{PSI}$$

THE PRINCIPAL STRESS IS NEGLIBLE. THE ALLOWABLE STRESS FOR SAE GR 5 BOLTS (SA-449) IS 23,000 PSI FOR BOLTS < 1 INCH. ANY OPERATING LOADS WOULD BE INSIGNIFICANT AND NOT JUSTIFIED TO CALCULATE SINCE WE HAVE SHOWN SUCH LOW STRESSES ABOVE.

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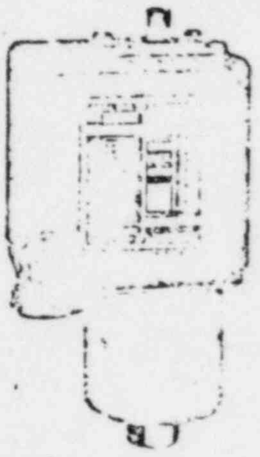
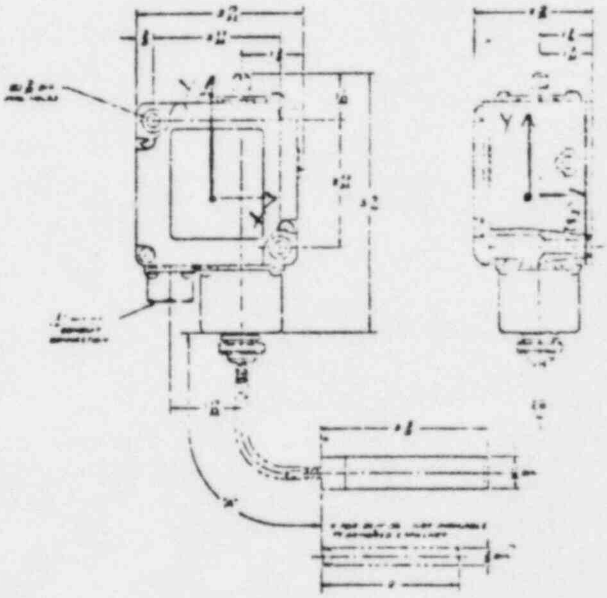
Pressure Switches - Square D - model ACW & BCW



Type ACW

Type BCW



Assumptions

1) It is assumed that the equipment is flexible in all directions. Therefore peak accelerations multiplied by a factor of 1.5 will be used in calculating the seismic loads
2) The CG of the unit is assumed above the bolt centroid and at approximately x=0, y=0 z= $1\frac{11}{16}$" with respect to [illegible]
3) Screw material is assumed to be A-36 steel
4) Weight of the equipment is assumed to be 5 lbs

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Calcs. For

[✓ Y] Safety-Related ☐ Non-Safety-Related

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Accelerations (Envelope of OBE & SSE)
horizontal = 2.6 g's
Vertical = .36 g's

Weight Loads

$F_y = 5$ lbs
$M_x = F_y Z = 5 \times 1.25 = 6.25$ lbs

Seismic Loads
$F_x = F_Z = 5 \times 2.6 \times 1.5 = 19.5$ lbs
$F_y = 5 \times .36 \times 1.5 = 2.7$ lbs
$M_x = F_y Z = 2.7 \times 1.25 = 3.375$ in-lbs
$M_y = F_x Z = 19.5 \times 1.25 = 24.375$ in-lbs
$M_Z = 0$

TOTAL LOADS
$F_x = F_Z = 19.5$ lbs
$F_y = 2.7 + 5 = 7.7$ lbs
$M_x = 6.25 + 3.375 = 9.625$ in-lbs
$M_y = 24.375$ in-lbs
$M_Z = 0$

Bolt Stresses (2 # 12 screws area = .024 in²)

Tensile:
Due to F_Z: $\sigma = \frac{19.5}{2 \times .024} = 405$ psi

due to M_x (M_x is analyzed as a couple)

$9.625 = Fd$ or $9.625 = F \times 2 \frac{23}{2} \rightarrow F = 3.54$

$\sigma = \frac{3.54}{.024} = 148$ psi

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Due to My (My is analyzed as a couple)

$24.375 = F \times d$ or $24.375 = F \times 2\frac{23}{32} \rightarrow F = 8.966$ lbs

$\sigma = \frac{8.966}{.024} = 374$ psi

TOTAL TENSILE = 406 + 148 + 374 = 928 psi

Shear $\frac{(F_x^2 + F_y^2)^{1/2}}{4 \times ib} = \frac{(19.5^2 + 7.7^2)^{1/2}}{2 \times} = \frac{20.965}{2 \times .024} = 437$ psi

The above calculated stresses are very small therefore the stresses are considered [illegible]

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Prepared by Jon Papadopoulos Date
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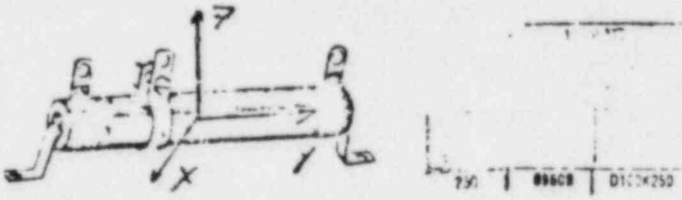
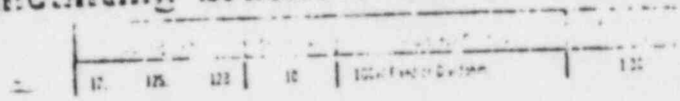
Adj. Resistor (Volt Range) - Ohmite model [illegible]



mounting brackets for tubular resistors



Assumptions

1) It is assumed that the equipment is rigid in all directions, therefore ZPA will be used in calculating the seismic [illegible]
2) It is assumed that the CG is at the geometric center of the equipment and @ x=0, y=0 z=8" with respect to the [illegible]
3) Assume that the mounting Bracket width is 0.5 in

Accelerations (Envelope of OBE & SSE)
Horizontal = .30 g's } use 0.5g horizontal & 0.[illegible] vertical
Vertical = .16 g's } in the calculations for [illegible]

Weight Loads: $F_y = 20$ lbs
$M_x = F_y Z = 20 \times 8 = 160$ in-lbs

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Seismic Loads: $F_x = F_z = 20 \times .5 = 10$ lbs
$F_y = 20 \times .3 = 6$ lbs
$M_x = F_y Z = 6 \times 8 = 48$ in-lbs
$M_y = F_x Z = 10 \times 8 = 80$ in-lbs
$M_z = 0$

Total Loads:
$F_x = F_z = 10$ lbs
$F_y = 20 + 6 = 26$ lbs
$M_x = 160 + 48 = 208$ in-lbs
$M_y = 80$ in-lbs
$M_z = 0$

BOLT STRESSES (2 #10 Screws - Area = .0174 in², Material [illegible]
- Tensile: due to Fz: $\sigma = \frac{10}{.0174} = 575$ psi

due to M_x: $M_x = Fd \Rightarrow 208 = F \times 73 \Rightarrow F = 208/73 = 2.85$

$\sigma = F/A = \frac{2.85}{.0174} = 164$ psi

due to My: $M_y/2 = Fd \Rightarrow F = (80/2)/.25 = 160$ lb

$\sigma = \frac{160}{.0174} = 9195$ psi

TOTAL TENSILE = 575 + 164 + 9195 = 9934 psi

$$\text{Shear} = \frac{(F_x^2 + F_y^2)^{1/2}}{2 A_b} = \frac{(10^2 + 26^2)^{1/2}}{2 \times .0174} = 800 \text{ psi}$$

Allowables: Tensile = $.44 S_u = .44 \times 58 = 25.5$ ksi
Shear = $.23 \times 58 = 13.3$ ksi
Combine Tension & Shear: $.44 S_y = 25.5$ ksi

The allowable stresses are higher than the calculated therefore the mounting screws are adequate.

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Client
Project SHOREHAM
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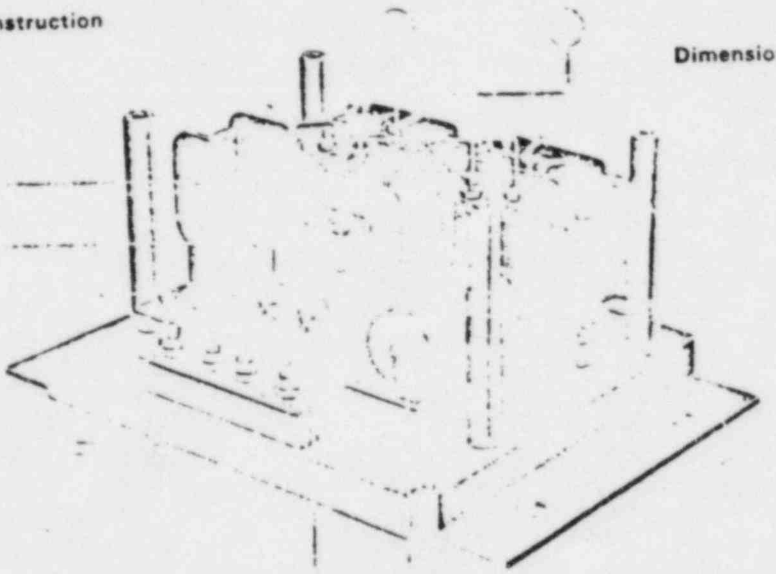
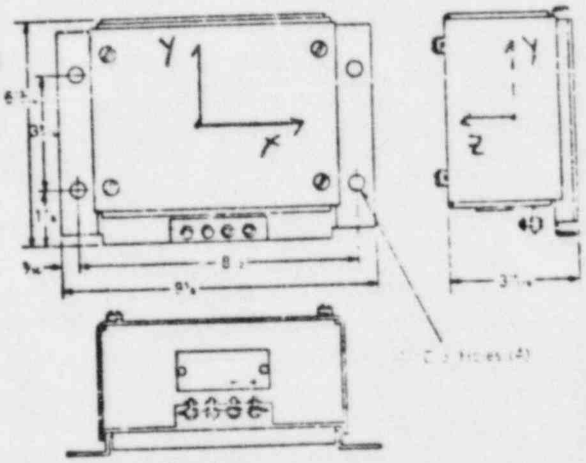
Prepared by Jim Papadopoulos Date
Reviewed by L.J Date
Approved by Date

Frequency Transducer – Westinghouse Model VC841

Construction



Dimensions



VC-841 Frequency Transducer

Net Weight .... 13 lbs.
Shipping Weight .... 18 lbs.

Assumptions:

1) It is assumed that the transducer is rigid in all directions therefore ZPA will be used for calculating seismic loads
2) Assume that the CG of the transducer is at its geometric center and at x=0, y=0, z=2" with respect to the bolt centroid

Accelerations (Envelope of OBE & SSE)

Horizontal: .30 g's use .50 g's in the calculations for conservatism
Vertical: .16 g's use .30 g's in the calculations for conservatism

Weight Loads: Fy: 18 lbs
$M_x = F_y Z = 18 \times 2 = 36$ in-lbs

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Seismic Loads: $Fx = FZ = 18 \times .5 = 9$ lbs

$Fy = 18 \times .3 = 5.4$ lbs

$Mx = FyZ = 5.4 \times 2 = 10.8$ in-lbs

$My = FxZ = 9 \times 2 = 18$ in-lbs

$MZ = 0$

TOTAL LOADS FOR BOLT REQUIREMENTS

$Fx = FZ = 9$ lbs

$Fy = 18 + 5.4 = 23.4$ lbs

$Mx = 36 + 10.8 = 46.8$ in-lbs

$My = 18$ in-lbs

$MZ = 0$

Bolt Stresses: (4 Bolts - 3/8" φ - Bolt area = .0773 [illegible]
Material A-36 (assumed)

Tensile: Due to FZ: $\sigma = \frac{FZ}{4 \times .0773} = \frac{9}{4 \times .0773} = 29$ psi

Due to M_x: (assume only 2 Bolts resist this moment)

$Mx = 46.8 = Fd$ or $46.8 = F * 3\frac{9}{16} \rightarrow F = 13.14$ lbs

$\sigma = \frac{13.14}{.0773} = 170$ psi

Due to My: (assume 2 Bolts resist this moment)

$My = 18 = F \times d$ or $18 = F \times 8.5 \rightarrow F = 2.12$ lbs

$\sigma = \frac{2.12}{.0773} = 27$ psi

TOTAL Tensile Stress = $29 + 170 + 27 = 226$ psi

Shear: $\frac{(Fx^2 + Fy^2)^{1/2}}{4 \times Ab} = \frac{(9^2 + 23.4^2)^{1/2}}{4 \times .0773} = 81$ psi

The above calculated stresses are very small (negligible) therefore the bolts are considered adequate.

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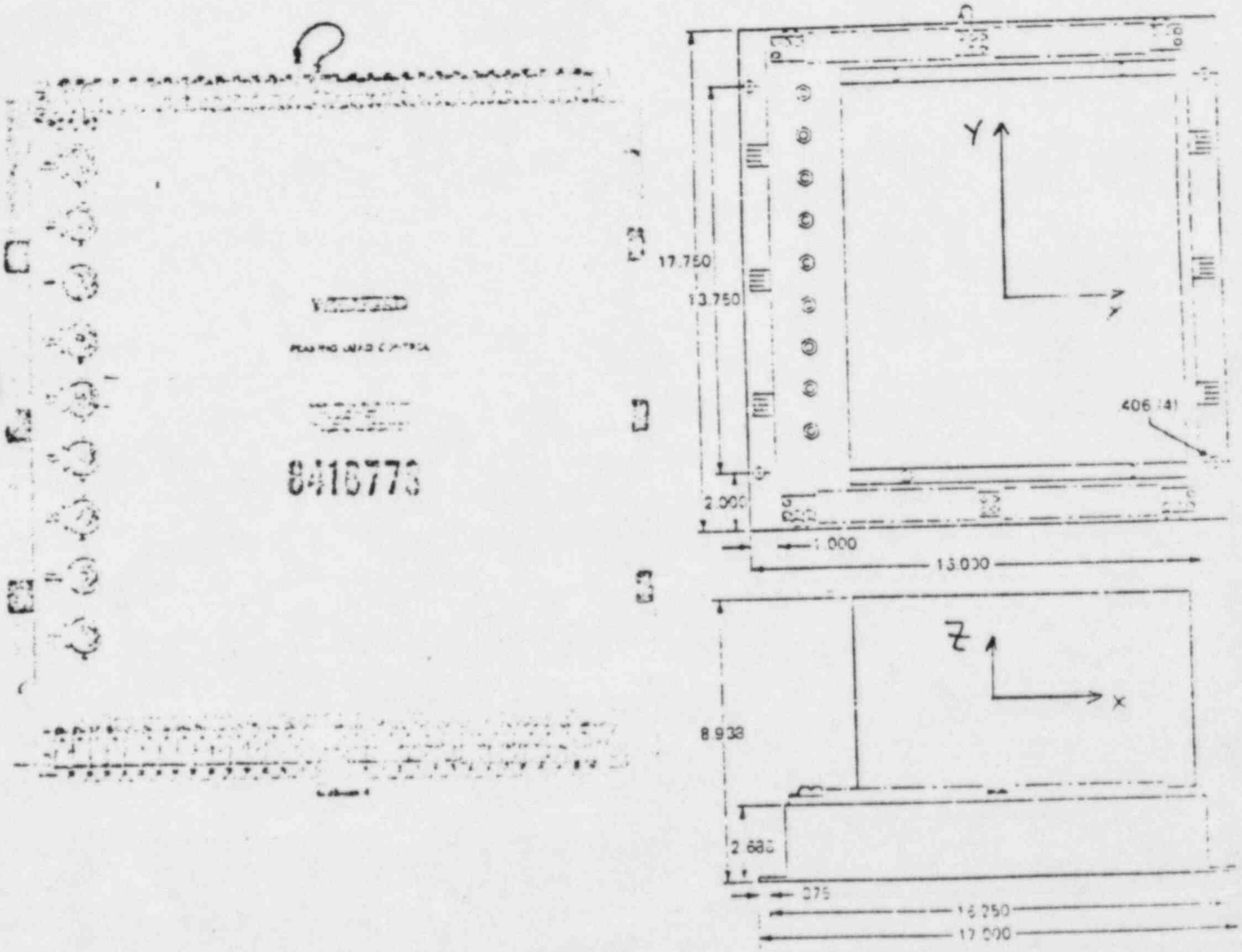
Gov. Peaking Load Cont Box - Woodward Gov. Mod. [illegible]

Outline Drawing, Peaking Load Control



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Assumptions:

1) It is assumed the equipment is rigid in all directions therefore ZPA will be used in calculating the seismic loads;
2) It is assumed that the weight of the unit is 60 lbs
3) The CG of the unit is assumed at geometric center and at coordinate x=0, y=0, Z=4.5", with respect to the [illegible]

Weight Loads:

Fy = 60 lbs
Mx = FyZ = 60x4.5 = 270 in-lbs

Accelerations (Envelope of OBE & SSE)

Horizontal: .30 g's use .50 g's in the calculation
Vertical = .16 g's use .30 g's in the calculation

Seismic Loads

Fx = Fz = 60x.5 = 30 lbs
Fy = 60x.3 = 18 lbs
Mx = FyZ = 18x4.5 = 81 in-lbs
My = FxZ = 30x4.5 = 135 in-lbs
MZ = 0

TOTAL LOADS FOR BOLT REQUIREMENTS

Fx = FZ = 30 lbs
Fy = 60+18 = 78 lbs
Mx = 270+81 = 351 in-lbs
My = 135 in-lbs
MZ = 0

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Bolt Stresses (4 - 1/2" Bolts Area ≈ .032 in²

Tensile:

Due to FZ: $\sigma = \frac{FZ}{4 \times A_b} = \frac{30}{4 \times .032} = 234$ Psi

Due to Mx: (Analyzed as a couple based on assumption that only 2 Bolts resist Total Moment

$351 = Fd$ or $351 = F \times 13.75 \rightarrow F = 25.52$ lbs

$\sigma = \frac{2552}{.032} = 798$ psi

Due to My (My analyzed as a couple based on assumption that total moment is resisted by only 2 Bolts)

$135 = Fd$ or $135 = F(16.25) \rightarrow F = 8.3$ lbs

$\sigma = \frac{8.3}{.032} = 259$ psi

TOTAL TENSILE = 234 + 798 + 259 = 1231 psi

$$\text{Shear} = \frac{(Fx^2 + Fy^2)^{1/2}}{4 \times Ab} = \frac{(30^2 + 78^2)^{1/2}}{4 \times .032} = 653 \text{ psi}$$

The above calculated stresses are very small therefore the bolts are considered adequate.

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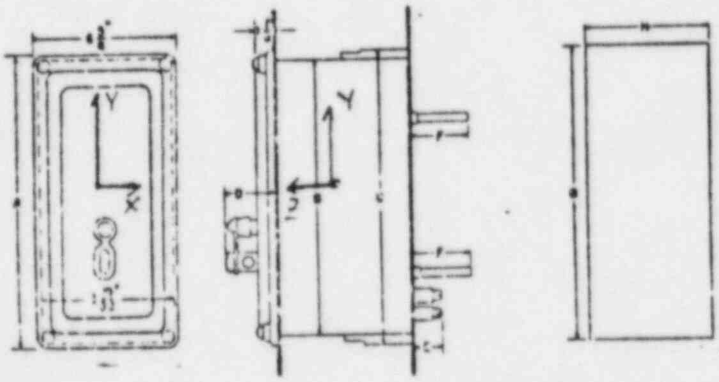
Watthour Meter - General Electric Model DS-U-64 & 65

DIMENSIONS



Types DS or DSW	Dimensions in Inches						Panel Cutout †	
	A	B	C	D (only DSM)	E	F Surface Mtg	G	H
DS-53, -65, -66	9⅛	8 [illegible]	8 11/16	1 11/16	⅜	3	5 [illegible]	5 [illegible]
DS-53, -55	9⅛	8 [illegible]	8 [illegible]	1 [illegible]	[illegible]	3	3 [illegible]	5 [illegible]
DS-64	16 [illegible]	15 [illegible]	15⅝	1 [illegible]	⅜	3	16	5 [illegible]
DS-54	16 [illegible]	15 [illegible]	15 [illegible]	1 [illegible]	⅜	3	16	5 [illegible]
DS-67, -69	16 [illegible]	15 [illegible]	15⅝	1 [illegible]	¼	3	16	5 [illegible]

	Fig No.	Approx Weight in Lb **		Overall Dimensions in Inches		
		Shipping	Net	Height	Width	Depth
2-stator	1	18	14	9 [illegible]	6 [illegible]	8
[illegible]	2	27	25	16 [illegible]	[illegible]	8
4-stator	2	23	26	16 [illegible]	6 [illegible]	8

Fig. 1. Mounting dimensions for switchboard watthour meters

Assumptions:

1) It is assumed that the equipment is rigid in all directions therefore ZPA will be used in calculating the seismic loads

2) The CG of the unit is assumed at its geometric center and at coordinates X=0, Y=0, Z=4" with respect to the bolt centroid.

3) It is assumed that the unit is supported at the 4 corners by 1/4" diameter bolts of A-36 material. This assumption is based on data from similar types of watthour meters which show that the [illegible] meters are supported by 8-1/4" bolts)

4) It is assumed that the equipment is a 4-Stator model therefore weight corresponding to that model will be used in the calculations.

5) It is assumed that the bolts are located at a distance =.25" from the edges

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Calcs. For

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Accelerations (Envelope of OBE & SSE)
Horizontal: .30 g's use .50 g's
Vertical: .16 g's use .30 g's

Weight Loads

$F_y = 28$ lbs
$M_x = F_y Z = 28 \times 4 = 11.2$ in-lbs

Seismic Loads:

$F_x = F_Z = 28 \times .5 = 14$ lbs
$F_y = 28 \times .3 = 8.4$ lbs
$M_x = F_y Z = 8.4 \times 4 = 33.6$ in-lbs
$M_y = F_x Z = 14 \times 4 = 56$ in-lbs
$M_Z = 0$

TOTAL LOADS FOR BOLT REQUIREMENT:

$F_x = F_Z = 14$ lbs
$F_y = 28 + 8.4 = 36.4$ lbs
$M_x = 112 + 33.6 = 145.6$ in-lbs
$M_y = 56$ in-lbs
$M_Z = 0$

Bolt stresses (4-$\frac{1}{4}$" φ. Bolts - Area = .032 in²)
Tensile:

Due to FZ: $\sigma = \frac{F_Z}{A} = \frac{14}{4 \times .032} = 109$ psi

Due to Mx: (Analyzed as a couple and [illegible] on the assumption that 2 bolts resist the Mx

$145.6 = F \times d$ or $145.6 = F \times (16\frac{5}{16} - .5) \rightarrow F = 9.2$ lbs
$\sigma = \frac{9.2}{.032} = 286$ psi

Due to My (Analyzed as a couple based on assumption that only 2 bolts resist the My)
$56 = F d$ or $56 = F \times (6\frac{?}{?} - .5) \rightarrow F = 9.14$ lbs

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$$\sigma = \frac{F}{A} = \frac{9.14}{.032} = 286 \text{ psi}$$

TOTAL TENSILE = $109 + (285 \times 2) =$ [illegible]

$$\text{Shear} = \frac{(F_x^2 + F_y^2)^{1/2}}{4 \times A_b} = \frac{(14^2 + 36.4^2)^{1/2}}{4 \times .032} = 305 \text{ psi}$$

The above calculated stresses are very low therefore the bolts are considered [illegible]

Form GQ-3.08.1 Rev. 2

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STATIC EXCITERS

LOG. NO. 0806080000

The static exciter transformers are mounted to a frame formed of 1 x 1 x 1/4 and 2 x 2 x 1/4 angles. A sketch of this frame is included. The structural integrity of this frame will be investigated. Stresses will be calculated for the critical angles as well as for the mounting bolts. Note that only structural integrity, not operability, can and will be shown based on the information source.

ANGLE STRESSES

It can be seen from the included sketch and photos obtained during field inspection that the static exciter transformers are bolted to the 2 x 2 x 1/4 angles at the top of the structure. These are considered to be the critical structural elements. For a conservative analysis, the 2 x 2 x 1/4 angle will be modeled as a simply supported beam of 77" span. One-half of the load due to the transformers will be applied to this beam. The beam is assumed flexible, so 1.5 times peak response spectra values (use elevated response spectra) will be applied horizontally and vertically. Based on the field inspection photos, each transformer is estimated to weigh 40 lb. and 5 transformers are mounted to the frame.

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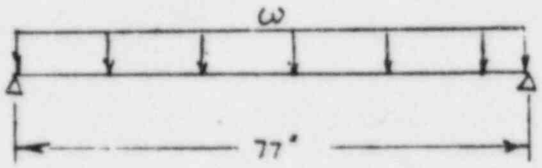
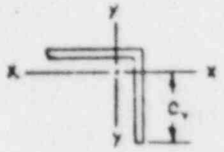


2 x 2 x 1/4 L



$I_x = I_y = 0.348$

$C_x = C_y = 2 - 0.592 = 1.408$

$S_x = S_y = \frac{0.348}{1.408} = 0.247$

$\omega_{angle} = \frac{3.19}{12} = 0.266$ lb/in

$\omega_{exciters} = \frac{1}{2}\left[\frac{5(40)}{77}\right] = 1.299$ lb/in

$\omega_{total} = 0.266 + 1.299 = 1.565$ lb/in

– Dead Weight Plus Seismic Accelerations:

(SSE) $a_{horiz} = 1.5(2.6) = 3.9\,g$

$a_{vert} = 1.0 + 1.5(0.36) = 1.54\,g$

$\omega_{horiz} = (3.9)(1.565) = 6.104$ lb/in

$\omega_{vert} = (1.54)(1.565) = 2.410$ lb/in

$$(M_{max})_y = \frac{\omega_{horiz}\,l^2}{8} = \frac{(6.104)(77)^2}{8} = 4524 \text{ in-lb}$$

$$(M_{max})_x = \frac{\omega_{vert}\,l^2}{8} = \frac{(2.410)(77)^2}{8} = 1786 \text{ in-lb}$$

$$\sigma_{max} = \frac{(M_{max})_x}{S_x} + \frac{(M_{max})_y}{S_y} = \frac{(1786 + 4524)}{0.247} = 25{,}547 \text{ psi}$$

Assuming angles are A-36 steel (S_y = 36 ksi)

$$\sigma_{allow} = \frac{4}{3}(0.6)(36{,}000) = 28{,}800 \text{ psi} \quad \text{(AISC Manual)}$$

Since $\sigma_{max} = 25{,}547$ psi $< \sigma_{allow} = 28{,}800$ psi, the structural steel angles are adequate to withstand applicable loadings

Form GC-3 08.1 Rev. 2

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Reviewed by	Date
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FRAME MOUNTING BOLT STRESSES

Total Weight :

- Transformers 5 × 40 = 200 lb
- Structural steel :

2×2×¼ ∠'s 3(77) + 2(25) = 281" length

$w = 0.226$ lb/in ⇒ $W = 0.226(281) = 63.5$ lb

1×1×¼ ∠'s 2(77) + 2(25) + 4(14) = 260" length

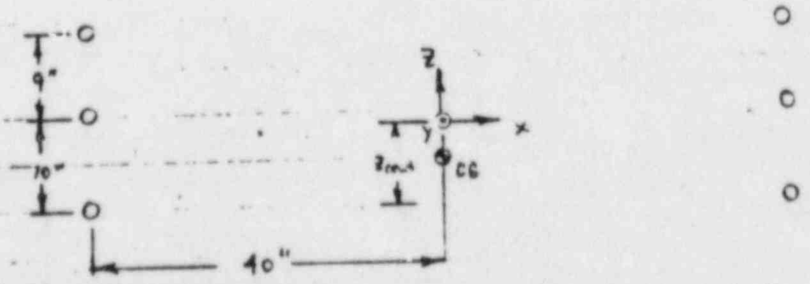
$\omega = \frac{1.49}{12} = 0.124$ lb/in ⇒ $W = 0.124(260) = 32.24$ lb

- Combined $W = 200 + 63.5 + 32.24 = 296$ lb

- To account for screens, cable and miscellaneous hardware, increase this weight 50% :

$W_{total} = 1.5(296) = 444$ lb

Based on the field inspection photos, the mounting bolts are assumed to be ½"φ. The mounting pattern is as shown below.



A 8" CG effect is assumed, based on the transformer eccentricities. Y_{CG} is assumed as 8"

$Z_{centroid} = \frac{2(10+19)}{6} = 9.67"$

$A_{bolt} = 0.142$ in² (tensile stress area) $\Sigma A = 6(0.142) = 0.852$ in²

$I_{xx} = 2(0.142)(9.67^2 + 0.33^2 + 9.33^2) = 51.3$ in⁴

$I_{zz} = 6(0.142)(40^2) = 1363$ in⁴

$J_{yy} = I_{xx} + I_{zz} = 1414$ in⁴

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Seismic Plus Dead Wt Loads :

As with the angle, the frame structure is assumed flexible

$F_x = F_z = (444)(3.9) = 1732$ lb
$F_y = (444)(1.54) = 683.8$ lb
$M_x = (1732)(8) + (683.8)(6) = 17{,}275$ in-lb
$M_y = (1732)(5) = 8660$ in-lb
$M_z = (1732)(8) = 13{,}856$ in-lb

$$\sigma_t = \frac{F_y}{A} + \frac{M_x C_z}{I_{xx}} + \frac{M_z C_x}{I_{zz}}$$

$$\sigma_t = \frac{683.8}{0.852} + \frac{(17{,}275)(9.67)}{51.3} + \frac{(13{,}856)(40)}{1363} = 4466 \text{ psi}$$

$$\tau = \frac{\sqrt{F_x^2 + F_z^2}}{A} + \frac{M_y\sqrt{C_x^2 + C_z^2}}{J_{yy}}$$

$$\tau = \frac{\sqrt{2(1732)^2}}{0.852} + \frac{8660\sqrt{9.67^2 + 40^2}}{1414} = 3127 \text{ psi}$$

$$\sigma_{max} = \frac{\sigma_t}{2} + \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2} = 6075 \text{ psi}$$

Assuming SA-449 bolts $\sigma_{allow} = 230$ ksi per ASME B&PVC Appendices, 1983, Table I-7.3

Since $\sigma_{max} = 6.075$ ksi $< \sigma_{allow} = 230$ ksi the frame mounting bolts are adequate to withstand dead weight and seismic loading

TRANSFORMER MOUNTING

Due to the limited time available for the Phase 2 report, not enough information could be obtained to properly analyze the transformer mountings. This has been judged to be a critical item for the structural integrity of the static exciters and will be investigated further in the Phase 3 report.

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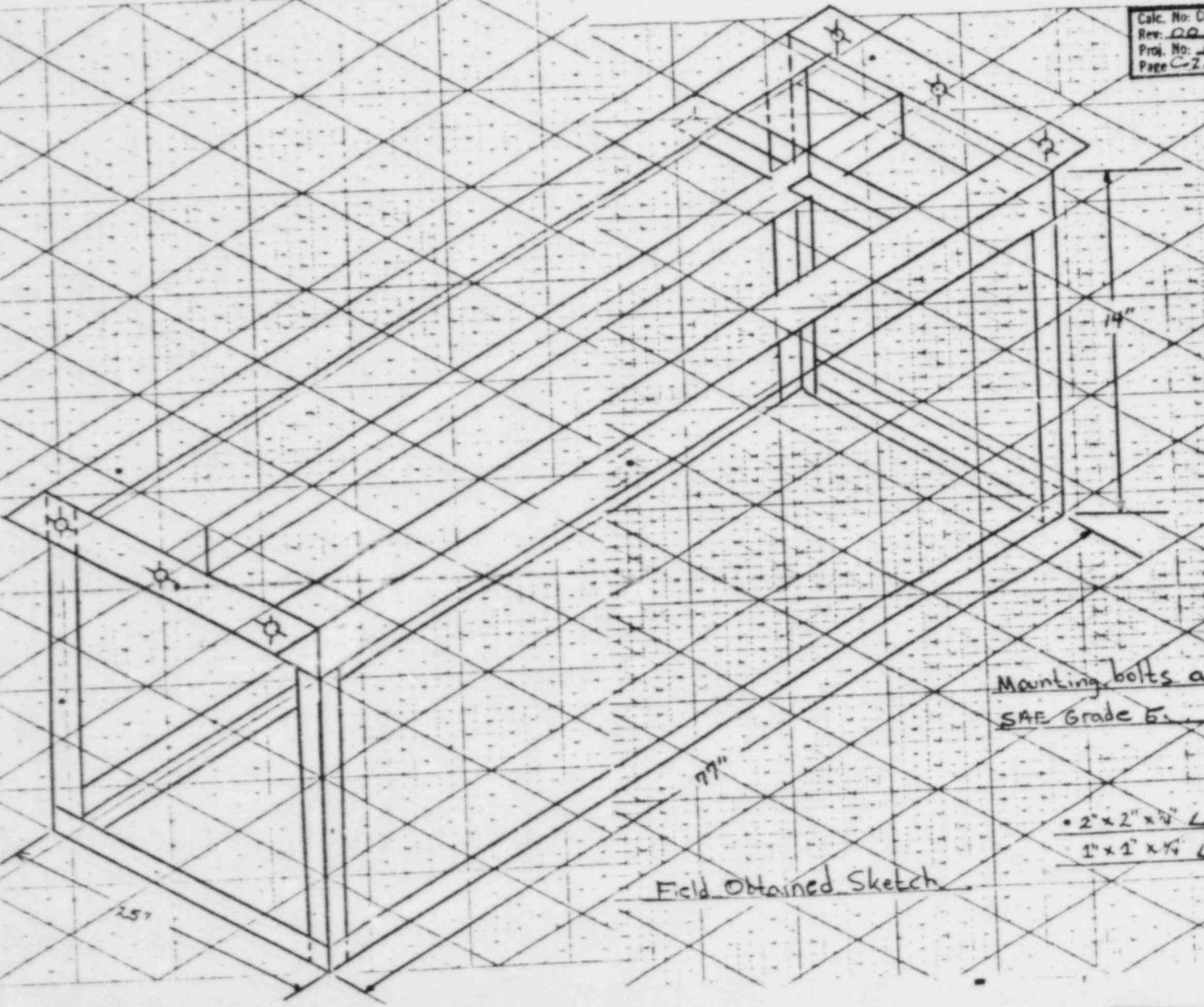
FINAL



Mounting bolts are SAE Grade 5.

- 2" x 2" x 1/4" L

1" x 1" x 1/4" L

Field Obtained Sketch

D.1

APPENDIX-D ENGINE OPERABILITY

ENGINE OPERABILITY

Purpose:

The purpose of this analysis is to present a summary of the seismic qualification for EMD 20-645E4 engine.

Conclusion:

It is concluded that the EMD 20-645E4 engine is capable of functioning properly during and following seismic disturbance.

Data and Assumption:

The data and assumptions in the analysis and tests are documented in References 1 and 2 and the EMD proprietary reports which are listed in References 1 & 2.

References:

1) Seismic Analysis of EMD 645 Engine" - Report No. E77-1 dated 3/14/77 - Electro-Motive Div. - General Motors Corporation
2) "Supplement Report on Seismic Analysis of EMD 645 Engines - Report No. E77-1A dated 7/18/84 - Electro-Motive Div. - General Motors Corporation
3) "LaSalle Div. 3 EMD 645-20 on Skid Component Resonant Frequency Search - In Situ Measurements - SWRI Doc. No. 06-6073 ISM20 April 1983
4) Shoreham EDG Enveloped Response Spectra - see pages A.1.4,6,8,10
5) Response of an EMD 645 Engine and Selected Accessories to Impulsive and Steady State Loads - SWRI Doc. #02-6073-001 December 1980

References: (Cont'd.)

6) Data Supplement to Response of an EMD 645 Engine and Selected Accessories to Impulsive and Steady State Loads - SWRI Transmittal Filed in CQD-012628, 2/22/84.
7) Effect of the Hawaiian Earthquake of April 26, 1973 upon the EMD Power Generating Sets Located in the Hawaiian Islands - EMD Report dated July 23, 1976
8) "Seismic Analysis for LaSalle Diesel Generator" - Stewart & Stevenson Services, Inc. - W.O. 68683 - Filed at S&L File #EMD-008028
9) Addenda to Ref. 8 - S&L File #EMD-011334
10) Revision to Ref. 8 & 9 - S&L File #CQD-010333

Introduction:

The diesel engine is a major component of the Emergency Diesel Generator System (EDGS). The function of the EDGS system is to produce electric power in the event of an emergency. The diesel engine drives a generator to produce the required level of electric power. The engine has to be operable during and following a seismic event. This qualification aspect is addressed in the following sections.

Qualification Methodology:

The qualification is based on combined testing and analysis as follows:

1) Underwater explosions were used under the direction of the U.S. Navy for shock testing of the running engines (Ref. 1,2). These tests were performed in accordance with the Military Specification Mil-S-901C (Navy). The acceleration levels in these tests ranged from 100 to 200 g's in the vertical direction and 45 to 120 g's in the horizontal direction. The navy high shock engine did include some components of higher strength material than those used in this Shoreham plant engine. The component of the Shoreham engine can therefore be considered qualified to at least 1/3rd of the above 'g' level. The factor of 1/3rd is based on the ratio of ultimate strengths of the materials in the two cases. Hence, our engine can be considered qualified for accelerations of 33g to 66g in the vertical direction and 15g to 40g in the horizontal direction; <u>OR</u> for a combined acc. of:

$$\text{Combined SRSS 'g'} = \sqrt{15^2 + 15^2 + 33^2} = 39.23g$$

The maximum required spectral 'g' loading (peak) for the Shoreham plant is .55g (Ref. 4). Moreover, as shown in Ref. 1, 2 & 3; the majority of the engine components have fundamental frequencies greater than [illegible] and all the components have fundamental frequencies greater than 8Hz-where the peak of the applicable response spectra (Ref. 4) ends. Hence, the highest 'g' level to which any engine component can be subjected under seismic loads will be less than g_{max} where:

Qualification Methodology:

1) Cont'd.

$$g_{max} = \underset{\text{Modal Participation}}{(1.5)}\ \underset{\text{Peak}}{(.55)}\ \underset{\text{3 Directions Absolute Sum}}{(3)} = 2.475\text{ g}$$

As discussed earlier, the actual least combined 'g' level in the test to which these components are qualified is 39 g. Hence, it is conservatively qualified. We have a factor of safety = $\frac{39.23}{2.475}$ = 15.85 in this qualification.

As discussed above, the accelerations as well as the duration of the Navy (MIL-S-901C) shock tests are much higher than those required for seismic qualification. Hence, the requirements of IEEE-344 Section 7.5 are fully satisfied.

Qualification Methodology: (Cont'd.)

2) An important source, further substantiating the seismic adequacy of the EMD engine, is the actual experience with 16 and 20 cylinder engines during an earthquake in Hawaii (April 26, 1973). Reference 7 documents satisfactory performance of EMD engines subjected to this earthquake which reached a magnitude level of 6.2 on the Richter scale.

3) An analysis is performed to calculate the natural frequencies and critical stresses in various engine components. This analysis is documented in Ref. 1 & 2. The analysis shows that the basic structure of the engine, the crankcase and the oil pan are rigid - having all frequencies greater than 33 Hz. A total of 59 components for the engine were analyzed in detail. This analysis shows that the actual 'g' level and the stresses in the components for a conservative response spectrum (Fig.1) with ZPA of 3g are well below the test 'g' levels and the allowable stresses for the material respectively. As seen from Figure 1, the amplification factors vary from 15$(=\frac{3}{.2})$ to 27$(=\frac{15}{.55})$ from the rigid to the flexible range. Hence, the conservatism is quite adequate.

Reference 3 (Table II-1 in Ref. 3) lists the lowest identifiable resonance frequencies from in-situ measurements. These data furhter confirm the analytically determined frequencies (Ref. 1, 2). All frequencies determined by this test report (Ref. 3) are greater than 8 Hz where the peak of the applicable response spectra ends; (Fig.2to[illegible]) and most frequencies are in the ZPA range.

Qualification Methodology: (Cont'd.)

4) Response of the engine and selected accessories to impulsive loads is measured in Ref. 5 by tests during operation of the engine. According to this study, the maximum shock spectra acceleration value at any frequency between 2 to 100 Hz during start-up is given by:

$$S(f) = 0.7e^{.379f}$$

where

f = frequency in Hz

S = acceleration in g

Now, at a frequency of 15 Hz corresponding to the running speed of 900 rpm of the engine, the shock loading during start-up is given by

$$S(f) = (.7)\ e^{(.0379)\ (15)} = 1.24g$$

Hence, if we take the worst case when the engine is simultaneously subjected to both start-up shock loading and the seismic shock loading, the maximum acceleration is given by:

$$g_{max} = 1.24 + 2.475 = 3.75g$$

Hence, in an extremely unlikely event of start-up combined with earthquake, we still have a factor of safety of $\frac{39}{3.75} = 10.5$.

Qualification Methodology:

5) A detailed finite element analysis of the engine assembly was performed (Ref. 8, 9, 10) utilizing ANSYS computer program. All critical components including the engine, the base, the foundation bolts, bearing, shaft, generator, and exciter were included in the finite element model. In this analysis it is shown that the stresses and deformations for the engine assembly are all within the code allowable limits. These reports qualified the engine including all mountings, foundation bolts and couplings for the input base response spectra given in Fig.6 to Fig.9. Since, as shown in these curves, the Shoreham base response spectra are enveloped by the spectra for which this qualification was done, the Shoreham engine including all its mountings, foundation bolts and couplings are adequately qualified by these reports. It may be noted that in Fig. 7 the slightly higher 'g' level in Shoreham spectrum is not critical since no component frequency is as low as 1 Hz.

Based on Items 1 thru 5, we can consider Shoreham EMD 20-645E4 engine adequate - structurally and functionally - during and following an earthquake.

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PROJECT NO. -6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING - 0.050
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FIG. 1

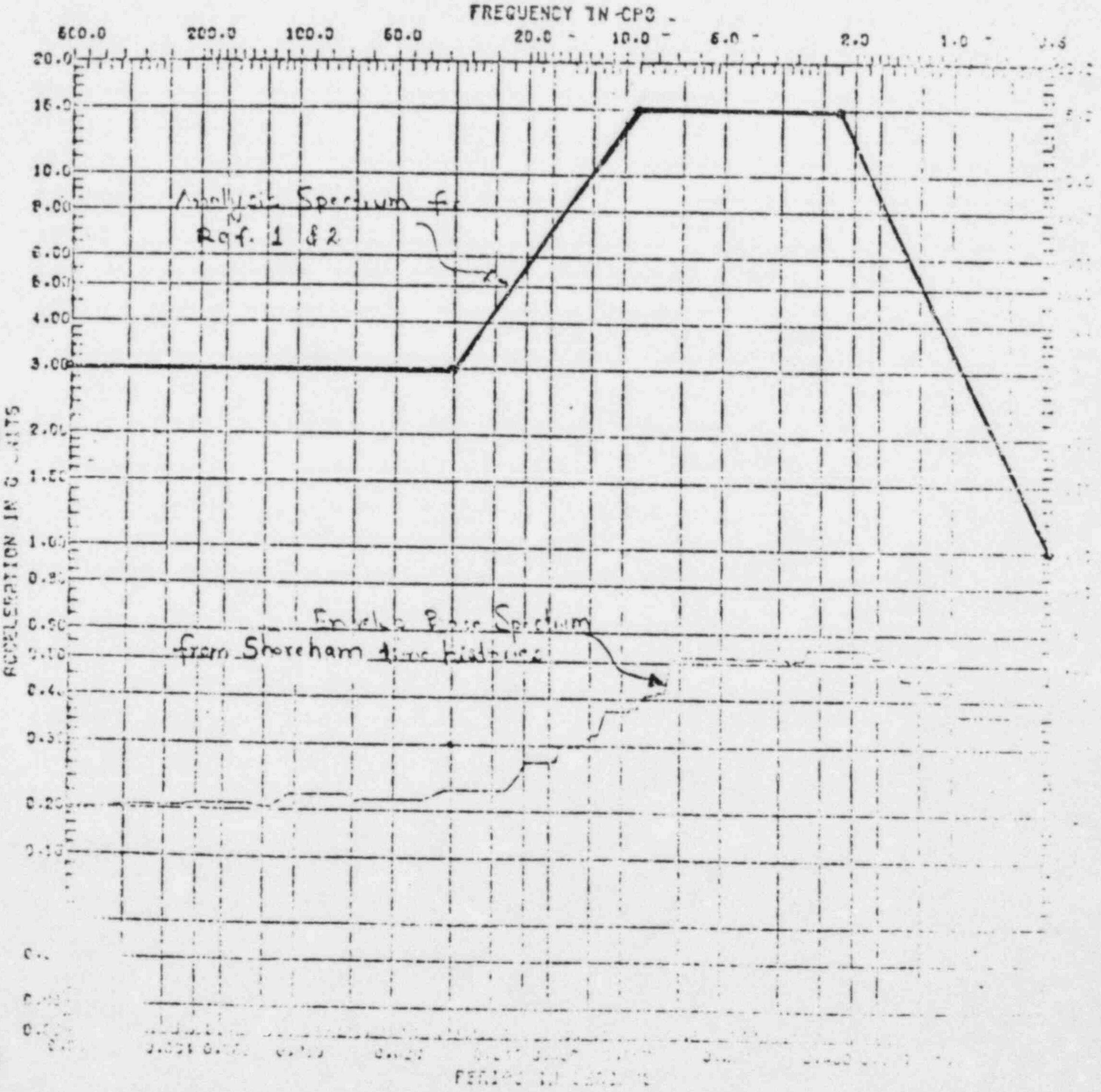

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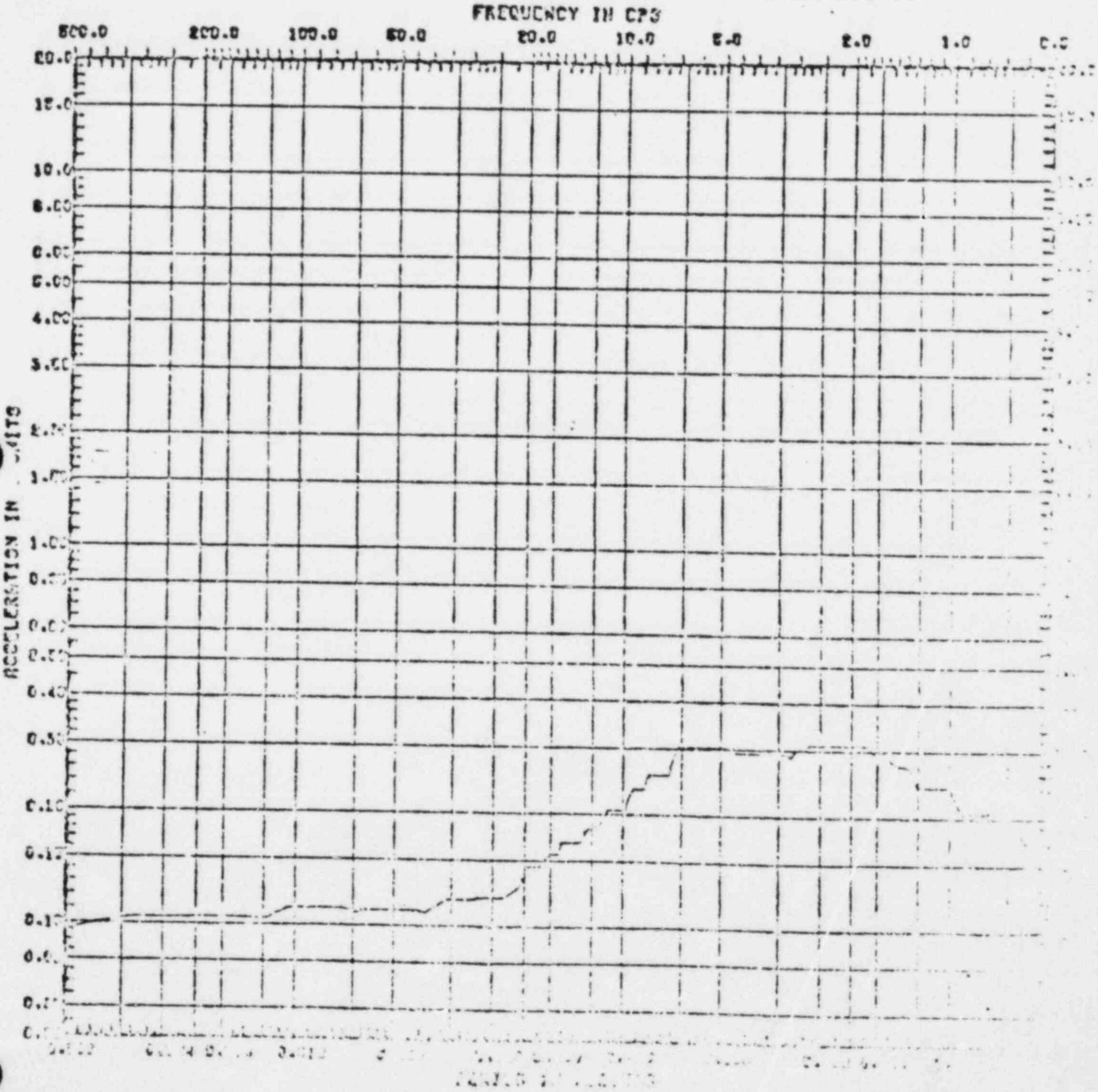
FIG. 2



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MODE CASE
DIRECTION N/A ANGLE N/A
ELEVATION [illegible]
LOCATION [illegible]

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ENGINEERS

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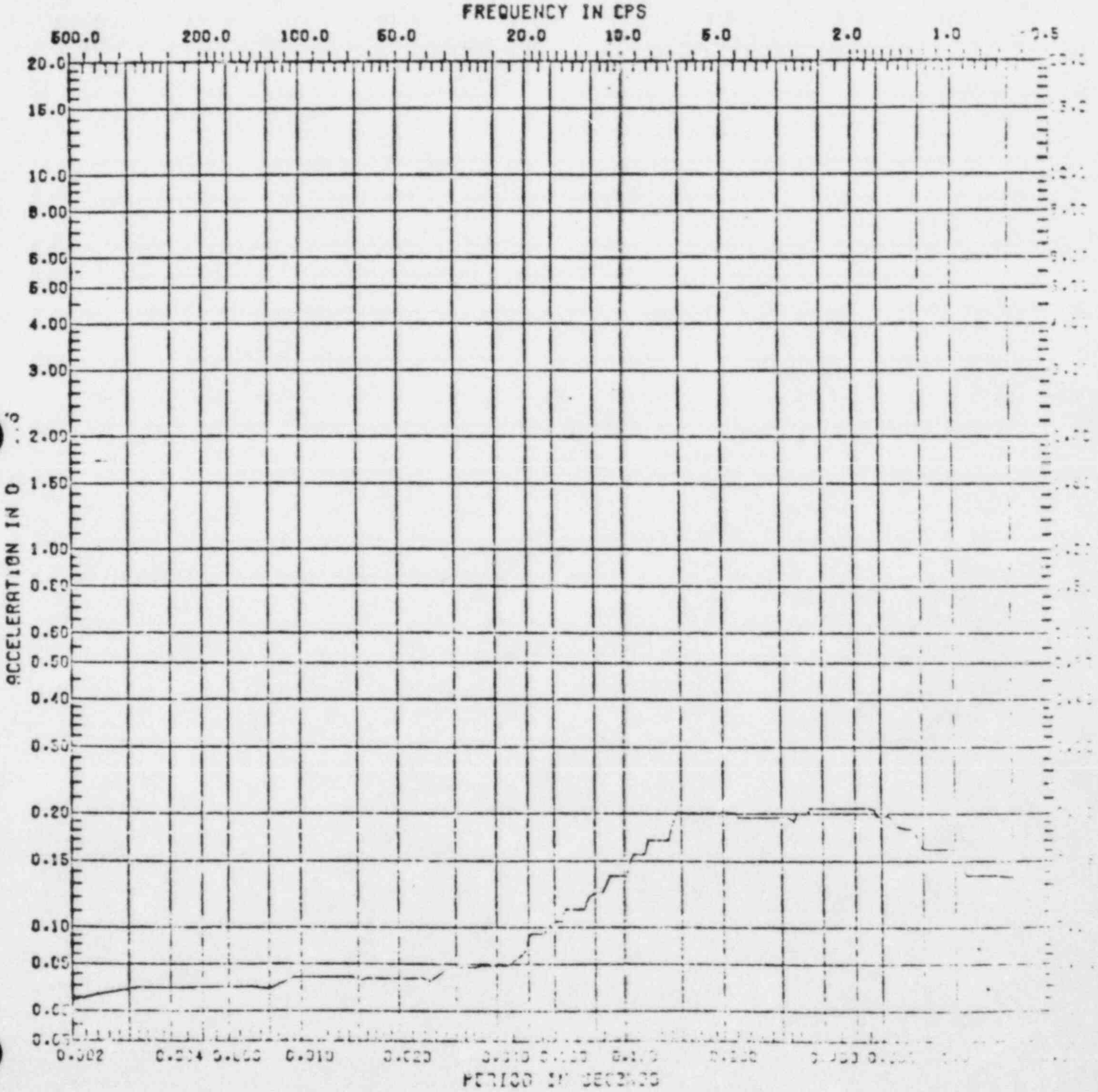
FIG. 3



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SPECTRA NO. OBE 2%

NODE BASE

ELEVATION GROUND

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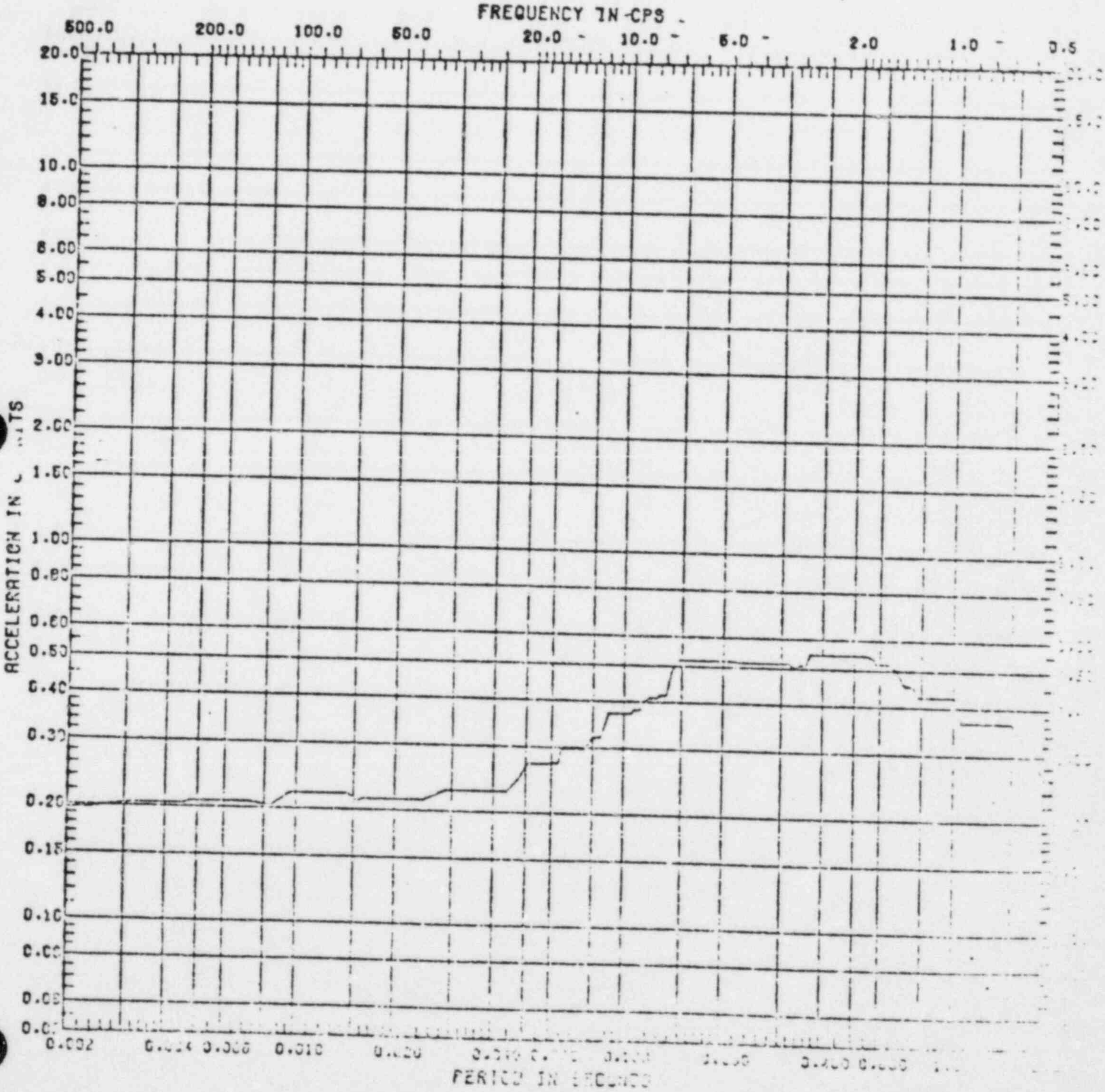
FIG. 4



LOADING:SSE(SX) ECCS
NODE BASE
DIRECTION HOR ANGLE N/A
SPECTRA NO. SSE SX
ELEVATION GROUND
LOCATION [illegible]

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PROJECT NO. 6955-00
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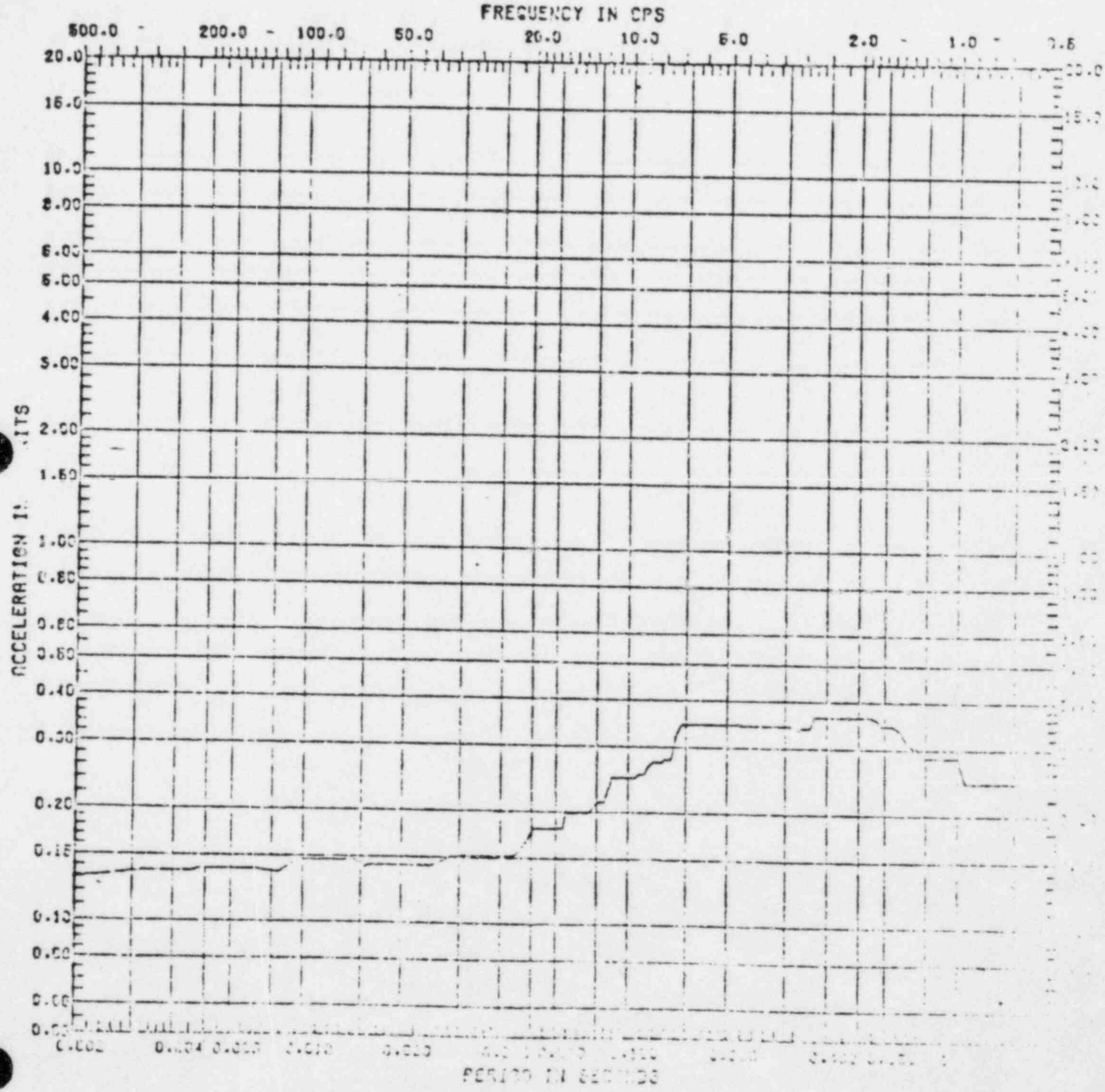
FIG. 5



LOADING:SSE(3%) EDGS SPECTRA NO. SSE 3%

NODE BASE ELEVATION GROUND

DIRECTION VER ANGLE N/A LOCATION EDG

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PEAKS WIDENED BY 20% ON EACH SIDE
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FIG. 6

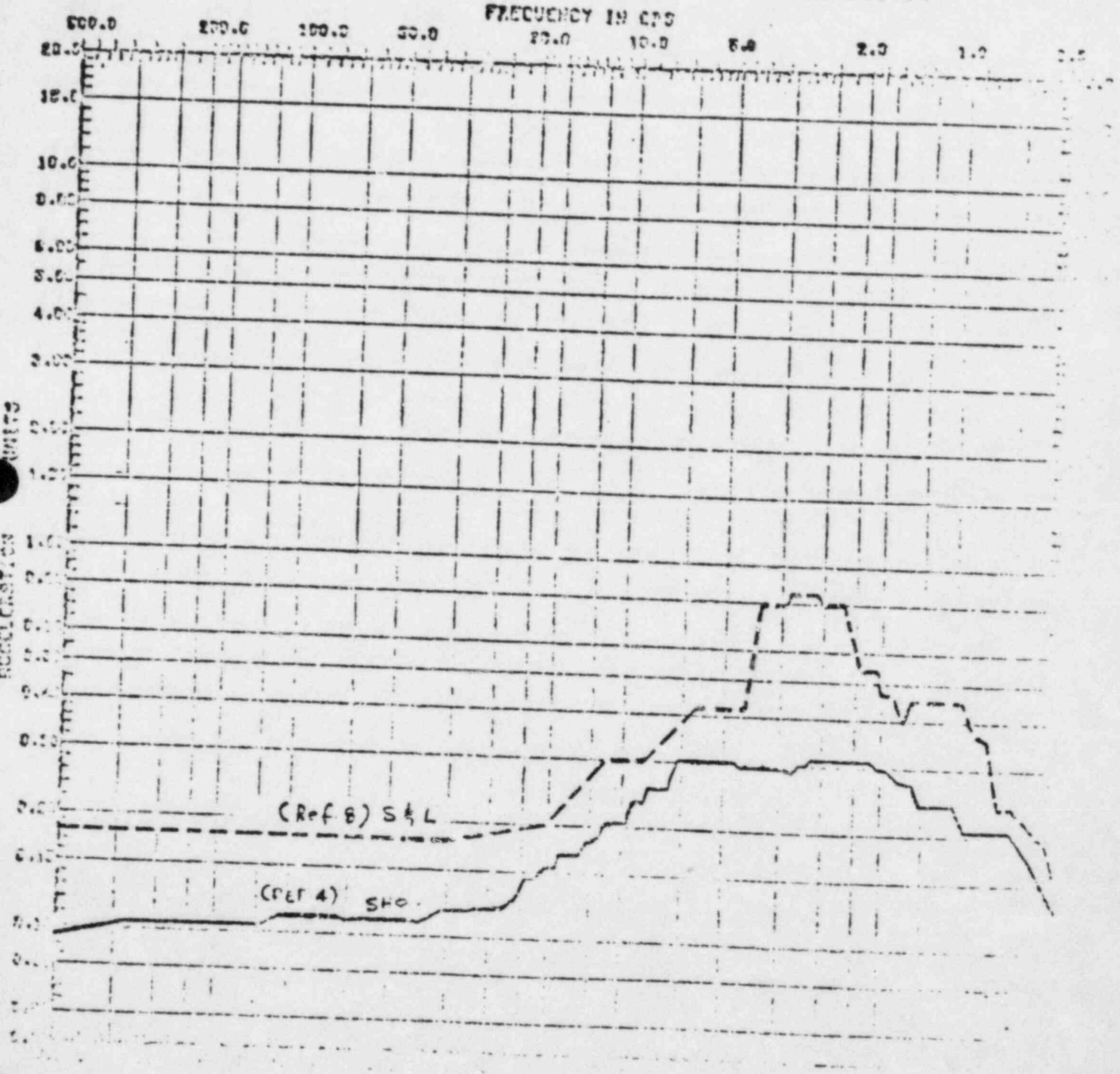

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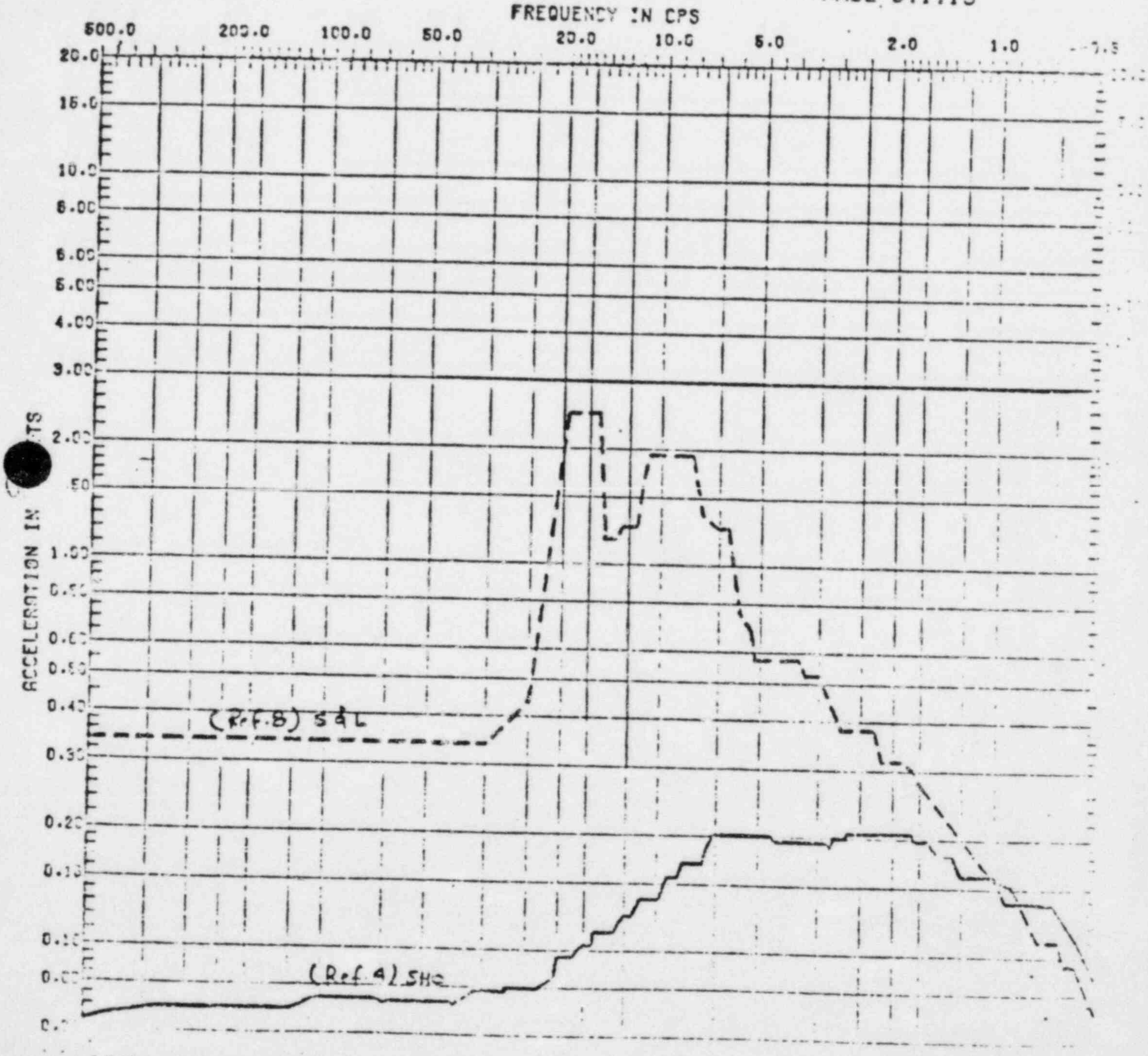
FIG. 7



[illegible]

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(11)

SARGENT & LUNDY
ENGINEERS

10 MAY 84
SIKC

CALC NO. COD-
PROJECT - SHOREHAM-1 - REV - 0
PROJECT NO. -6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
DAMPING -0.030
PAGE

CALC NO. CQD-014046
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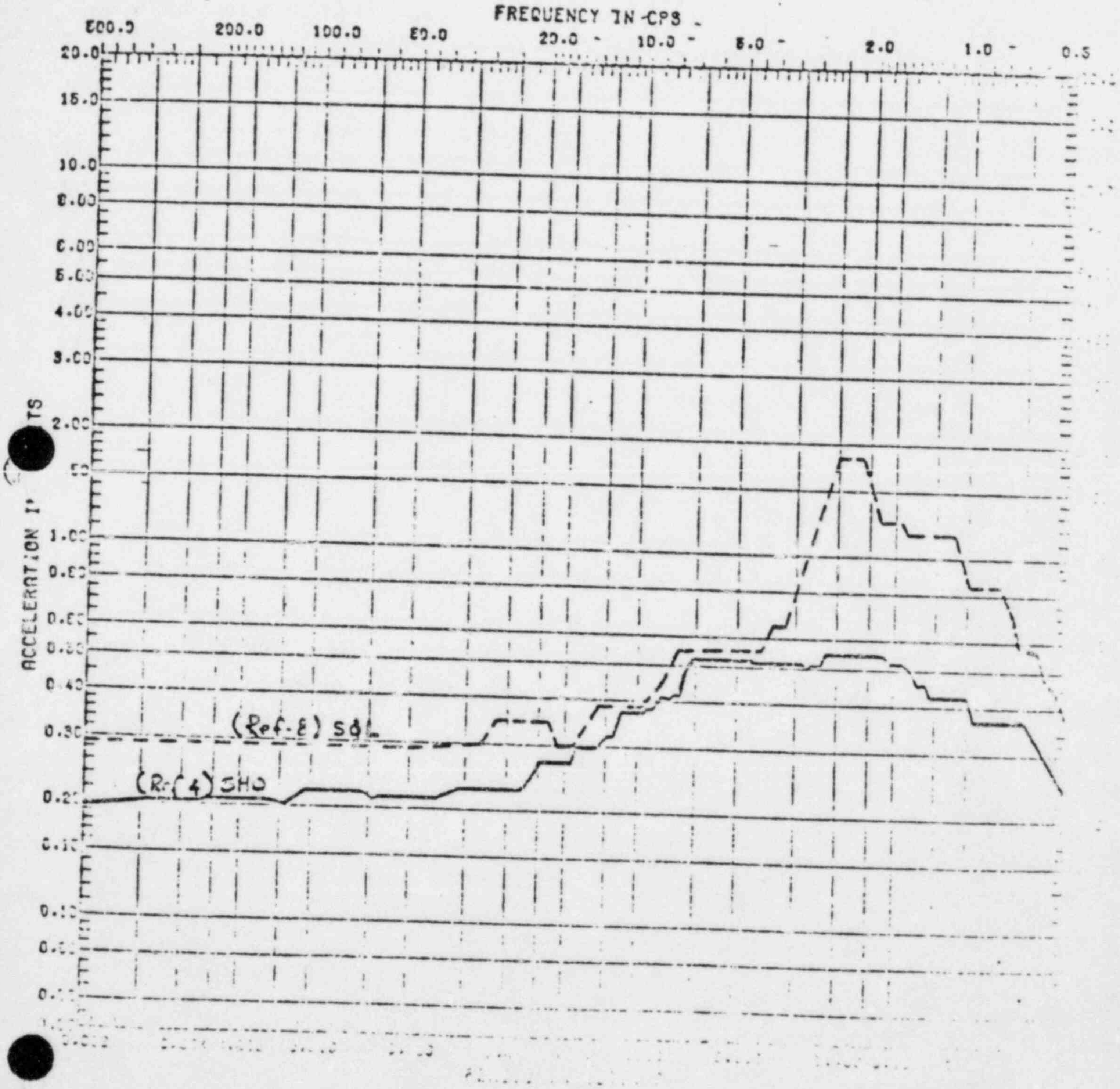
FIG. 8



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MODE
DIRECTION ...R
ANGLE N/A
ELEVATION
GROUND

SARGENT & LUNDY
ENGINEERS

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CALC NO. CQD-
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PROJECT NO. 6995-00
PEAKS WIDENED BY 20% ON EACH SIDE
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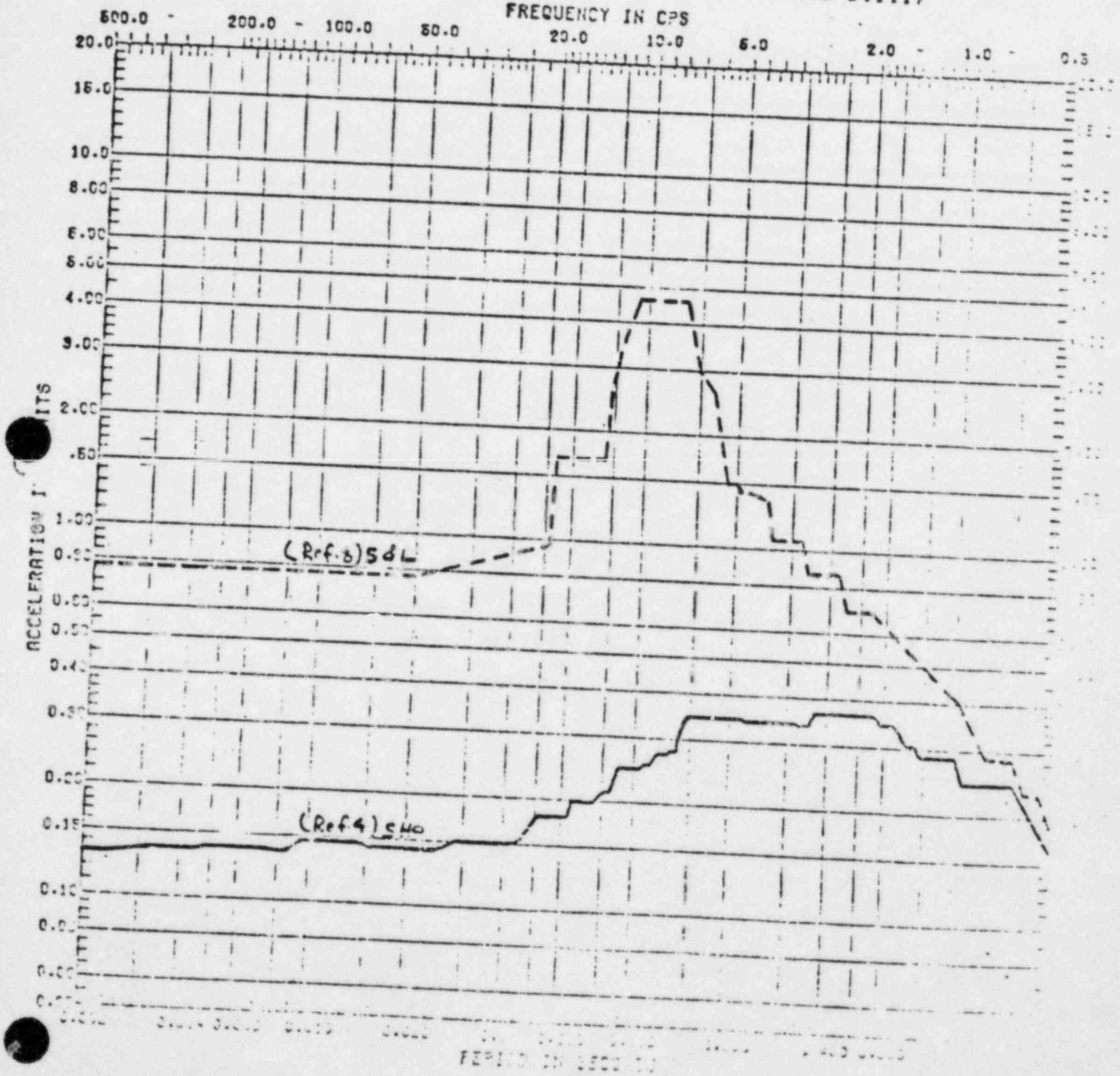
FIG. 9



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SPECTRA [illegible]
ELEVATION GROUND
LOCATION [illegible]

ENGINEERING REPORT
ELECTRO-MOTIVE DIVISION
GENERAL MOTORS CORPORATION

CALC NO. CQD-014046
REV. 00 DATE 6/1/84
PROJ. NO. 6995-00
PAGE D.1.18

REPORTED BY: J. Chen

DATE: 3/14/77

APPROVED BY: R. F. Hart

REPORT NO. E77-1

SECTION: Engine Design

SUBJECT: SEISMIC ANALYSIS OF EMD 645 ENGINE

In this report, all standard EMD 645 Roots blown and turbocharged engines are shown to be capable of functioning properly when subjected to the ground disturbance of an earthquake. This study is based upon the analysis of the individual components of the engine, as well as the engine as a whole. We drew upon the following sources of information:

a) Underwater explosions were used under the direction of the U.S. Navy for shock testing of running "high shock" engines, both Roots blown and turbocharged. (See the list of reference attached.) The "high shock engine" includes some components made of higher strength material than is used on our standard engine. The components of the standard engine are, therefore, qualified for a shock loading somewhat lower than the "high shock engine", on a pro rata basis, determined by the ultimate tensile strengths of the materials involved.

b) Actual experience with our turbocharged 16-and 20-cylinder engines during an earthquake in Hawaii has been documented in the attached report.

c) Calculations of natural frequency and of stresses have been compiled to supplement the above.

The turbocharged engine is more heavily stressed by seismic loads than the Roots blown engine, because of the heavy overhanging turbocharger mass. The twenty-cylinder 645 engine is more heavily stressed than the smaller 645 engines because of the larger mass and greater distance between supports. It follows that qualification of the twenty-cylinder turbocharged engine qualifies the full line of Roots blown and turbocharged 645 engines. Our attention has, therefore, been directed to the analysis of the twenty-cylinder turbocharged engine.

When an earthquake occurs, the ground moves in a random fashion as a function of time. The power of its [illegible] frequencies of less than [illegible]. In order [illegible] the following values [illegible]:

Horizontal acceleration [illegible]

Vertical acceleration: 1 g

Frequency: 1/3 Hz to 33 Hz

The first step in the analysis is to show that the basic structure of the engine, the crankcase and oil pan, may be treated as a rigid structure; that its fundamental natural frequency is above the highest ground loading frequency of 33 Hz. When this is established, it follows that the ground accelerations of 3 g's horizontal and 1 g vertical may be treated as a single "shock" loading of the same magnitude for the basic structure. Furthermore, the crankcase will transmit the random vibratory ground accelerations essentially unchanged as a "floor loading" to the engine components supported by it.

The analysis of each engine component supported by the rigid crankcase then depends upon the fundamental natural frequency of that component. If this natural frequency is above 33 Hz, the seismic loading on the component may be taken as a single shock load of 3 g's horizontally and 1 g vertically. If the fundamental natural frequency of the component falls below 33 Hz, the random vibratory "floor loading" on the component may again be replaced by a single shock load, but now amplified to take into account the effect of resonance. The amplification by which the 3 g horizontal and 1 g vertical floor load is multiplied is a function of the damping in the system. A conservative estimate of this damping is 3% of critical damping, and on this basis Figure 2 shows the amplification factor as a function of the fundamental natural frequency of the component.

The details of the analysis for each item follow:

Engine Crankcase and Oil Pan

As shown in figure 1 a right hand Cartesian coordinate system is used with the origin at the geometric center of the crankshaft. Positive directions on the 3 axes are as follows:

X, axially toward the accessory drive end; Y, vertically upward; Z, horizontally toward the right bank.

The fundamental natural frequency of the basic engine structure was calculated based upon the assumption that the structure was a simply supported, uniformly loaded prismatic beam with a span of 180". The following physical data were used for the computation:

Total engine weight 43,500#
Equivalent cross-sectional area [illegible].2 in.^{2}
Equivalent moment of inertia of section:

$I_z = 50,000 \text{ in.}^4$

$I_y = 34,300 \text{ in.}^4$

Center of gravity of engine

$\overline{X}$ = -10" (toward the generator end of engine)

$\overline{Y}$ = 15" (above the crankshaft centerline)

$\overline{Z}$ = 0

The fundamental natural frequency is 81.8 Hz. in the X-Y plane and [illegible].5 Hz. in the X-Z plane. These frequencies being about twice the highest possible seismic excitation frequency preclude the possibility of resonance of the engine as a whole.

The engine is mounted to the base with two 1"-8 bolts of GM 280 (SAE Grade 5) material at each of the four mounting pads. In addition, two 1" diameter dowels are used at each of the two rear mounting pads. It has been sufficient in this analysis to make the conservative assumption that all load is carried by the bolts. Using the superposition of the shock loads given above (3 "g's" in the horizontal direction and 1 "g" vertically) the maximum total stress in any bolt is [illegible] psi tension. This stress is satisfactory, being less than one-third of the minimum ultimate tensile strength of 120,000 psi and only 40% of the 92,000 psi minimum yield point.

Turbocharger

The following excerpt is from the 1966 High Shock Study, reference 5.

"In the elastic lattice mass point system analyzed, the turbocharger was considered as a single mass point overhung as a cantilever from the end of the diesel engine. At this point, which represents the center of mass of the entire turbocharger, a coordinate system was placed with the X direction parallel to the rotor axis, the Y direction vertical, and the Z direction horizontal in the lateral direction. The total weight of the turbocharger of 1075 pounds was suspended at this mass point 8.485 in. out from the turbocharger drive gear housing on a cantilevered support having a section moment of inertia about the Z-axis of 1900 in.^{4}, a section moment of inertia about the Y-axis of 4190 in.^{4}, an area in the Y-Z plane of 36.3 in.^{2}, and a modulus of elasticity of 28 X 10^6 P.S.I. Polar moments about the X, Y, and Z axes were given as [illegible], 894, and 795 in-lb-sec.^{2} respectively. The turbocharger was considered as a six degree of freedom system and linear and torsional accelerations were computed about these axes. The shock accelerations for which the turbocharger was evaluated are those due to an elastic vibration response of the engine and turbocharger to an impact load. These accelerations [illegible] during [illegible] oscillations of [illegible] which [illegible] to. For the described [illegible]

Following maximum accelerations were given:

Direct	Linear, G	Freq., Hz	Torsional, rad/sec.^{2}	Freq., Hz
X	127.42	221.73	2331.44	191.04
Y	162.85	148.10	1393.46	221.73
Z	84.36	221.73	3099.58	221.73

As the entire turbocharger was considered to be a point mass concentrated as its center of gravity in the shock computations, the above accelerations are those experienced at the turbocharger center of gravity. These accelerations act simultaneously and are a result of an external shock to the entire diesel engine in the vertical Y direction. They represent the total response of the turbocharger due to both its own elasticity and to that of the entire diesel engine. Computations for external shocks in the X and Z directions did not result in as great a response as that of the Y direction.

In analyzing the components of the turbocharger, these accelerations were directly applied to the center of gravity of the components involved, and the inertia loads which were taken to act through the center of gravity calculated. These inertia loads which were based on component weights of the T-16 turbocharger were used to determine the loading on the various turbocharger components, and from these loadings the stresses and deflections of the components were calculated."

The above study shows that the natural frequency of the turbocharger is far higher than the range of seismic excitations. It, therefore, qualifies as a rigid component, and is rigidly mounted to ground. The equivalent single shock loadings of an earthquake will, therefore, be 3 "g's" in the horizontal direction and one "g" vertically. The effect of earthquake induced force on the turbocharger will be negligible inasmuch as these seismic loadings are only a very small fraction of the actual high shock loads applied to the structure.

Fuel Filter

The fuel filter is mounted on the accessory [illegible] of the engine [illegible] two 1/2-20 UNF bolts of GM [illegible] (SAE Grade [illegible]) material. [illegible] structure between the fuel filter mounting and ground is taken to be infinitely rigid. The calculated natural frequency of the fuel filter assembly is 17.2 Hz. Since this falls within the range of possible seismic excitation the effective single shock load will have to involve the "floor loading" times the approprate amplification factor in order to take into account the possibility of resonance associated with the prolonged earthquake. From Fig. 2, it is seen that the amplification is equivalent to 2.2 giving a load of 6.5 g's axially. Inasmuch as the center of gravity of the filter assembly is 7 inches below the mounting bolts, it is the axial component of floor loading which is relevant and the total effective single shock load will be the 6.5 g's. The highest bolt loading will occur when this equivalent shock loading acts in the positive X direction, pulling the filter assembly away from the engine. In order to determine the allowable loading on the bolts in "g's", we observe that each fastener may be safely loaded to its proportionate limit or "proof strength" or 13,600 lbs. Since the centerline of the bolts is 11/16" below the upper edge of the mounting flange the moment capability of the mounting is 18,700 inch lbs. for the top bolts. The allowable shock load is found to be 97 g's, compared with the maximum seismic shock loading of 6.5 g's.

At the maximum loading of 6.5 g's, the bending moment at the governing section of the bracket casting is calculated to be 1200 in-lb and the section modulus to be 0.215 in.^{3}. Hence, the induced bending stress at the section of the bracket is 5,600 psi, less than 1/2 of the fatigue strength of 14,000 psi for the material, EMS40, of which the bracket is made.

Crankshaft System

The lowest fundamental frequency of the crankshaft system will involve that portion at the generator end of the crankshaft which overhangs the rear main bearing and supports the flywheel, ring gear, and approximately one-half of the weight of the generator rotor. This frequency is calculated to be 86.5 Hz., well above the seismic range, based upon the following assumptions.

a) The portion of the crankshaft in question may be simulated by a 7.5" uniform diameter round shaft acting as a 16.3" long cantilever beam, built-in at the rear main bearing and receiving a concentrated vertical load at its free end. The flexible coupling at the end of the [illegible] essentially zero moment at this location.

b) The natural frequency may be calculated by the superposition of two mass systems acting on the cantilever beam: first, a concentrated load of 4,170 lbs. at the free end of the beam and second, the uniformly distributed weight of the beam itself.

Power Assembly

The primary moving parts of the power assembly are the piston, piston carrier, piston pin and connecting rod. In an operating engine the forces on these components are predominantly those attributable to gas pressure, while the inertia forces provide only a minor input. Therefore, dynamic effects due to seismic excitation will be insignificant as far as induced stresses are concerned. The natural frequency is calculated on the basis of two springs in series. One spring represents the bending of the crankshaft between adjacent main bearing supports and the other spring is the connecting rod itself. The effective mass has been taken as the complete piston assembly plus the entire connecting rod. Based on this assumption, the calculated natural frequency is 415 Hz, far above the seismic range. Hence, resonance due to seismic excitation cannot occur.

Exhaust Valve Gear

The exhaust valve gear consists of two independent sets of three identical springs each. Each set of three springs is spanned by a valve bridge assembly which serves to actuate the valves simultaneously. The fundamental mode of this set of three springs is one of pure translation parallel to the cylinder centerline, which is calculated as 29.94 Hz. For a single valve and spring the fundamental natural frequency is 31.9 Hz.

The exhaust valve is not a 'spring-mass" free vibration system It is held in a preloaded configuration by a spring exerting a force of 117 lb. upon the valve. Seismic loading cannot exceed the 'ground loads" specified above and will result in a force of less than [illegible] for the single valve assembly weight component tending to unseat [illegible] valve.

Overspeed Trip Assembly

The fundamental natural frequency of the overspeed trip assembly is calculated as 23.7 Hz. Although this is within the range of possible earthquake frequencies, the resonance cannot occur, since the overspeed trip assembly is not acting as a spring-mass free vibration system.

As specified in Engineering Test Instruction [illegible] is set at 10% above the normal maximum engine speed. This [illegible] the overspeed tri [illegible]ill not be activated by any [illegible] load [illegible] is less than the [illegible]ivalent of 990 R.P.M. With this setting, [illegible] will have a net preload of 41 lbs. on the overspeed trip pawl at the 900 RPM operating speed. This 41 lb. force would have to be overcome by seismic forces for unseating to occur. The maximum seismic load possible for the system of 1.64 lb. is only 5.2 lb., based on a maximum acceleration of 3 g's horizontal and 1 g vertical. Hence unseating cannot occur and a "spring-mass" system cannot be formed.

Other Engine Components

The attached table lists the additional engine components which have been analyzed for natural frequencies as well as for the dynamic strength of the components and of their mounting systems. These components include the water pumps, heat exchangers, governor, lube oil pump, scavenging pump, lube oil strainer, and turbocharger oil filter. The fundamental frequency for each component was calculated in each coordinate direction. In the table the static weight of each component and the definition of the bolting system attaching the components to the engine are also given for reference.

It is noted that all fundamental frequencies in this table are higher than the frequency range produced by earthquakes. Therefore, resonant excitation due to seismic shocks for any component listed in the table will not occur.

The bolts evaluated for strength in the table are those that hold the components to the accessory drive housing or to the camshaft drive housing. The adequacy of the bolts that hold these housings rigidly to the crankcase has been qualified in the computer analysis for the high shock engine, reference #2.

The structural castings of each of these components are qualified for seismic loading by the following interpretation of the actual Navy high-shock test results, as follows:

a) The Navy high shock components were of EMS43 ductile iron having an ultimate tensile strength of 60,000 psi. Each of these components was subjected to numerous shock loads in random horizontal directions, the least of these loads being 45 g's, and no failures occurred.

b) The components for our standard engines are made of EMS11 cast iron, having an ultimate tensile strength of 20,000 psi. To simulate seismic loading for these rigid components, the highest possible horizontal shock load to be sustained will be 3 g's.

We conclude that the standard engine components can withstand at least 15 g's without failing, whereas the maximum seismic load will be 3 g's. Expressed in terms of stress, it is seen that the highest seismic induced stress will be no greater than 4,000 psi, safely below the 20,000 psi ultimate tensile strength.

Summary

In the conclusion of this report, it should be pointed out that the high shock engine was developed in accordance with the Military Specification Mil-S-901C and was tested based on a test procedure approved by the Naval Shipyard. In that test procedure, the severity and number of underwater explosive shocks; the instrumentation and the engine operating conditions during the test were all specified. The test results which were documented in the listed references have shown that the instrumentation registered the dynamic inputs ranging from 100 to 200 g's in the vertical direction and 45 to 120 g's in the horizontal direction, depending on the particular explosion and on where the instrumentation was mounted. Inspections were made during and after the testing and it was noted that there was no significant damage, nor any malfunctions. The results from these high shock tests were interpreted in a conservative manner for our standard engine with the conclusion that seismic loading would not affect the normal function of the standard engine.

In addition, actual operating experience with our standard 20-cylinder engines was experienced in a Hawaiian earthquake which was registered as a magnitude of 6.2 on the Richter Scale and is documented in the attached report. It is seen that those engines functioned satisfactorily through this major seismic distrubance.

In view of the analysis above it is concluded that our current standard 20-645 engine unquestionably qualifies for seismic loading. On this basis, the entire line of 645 engines is found to be qualified for seismic loading.

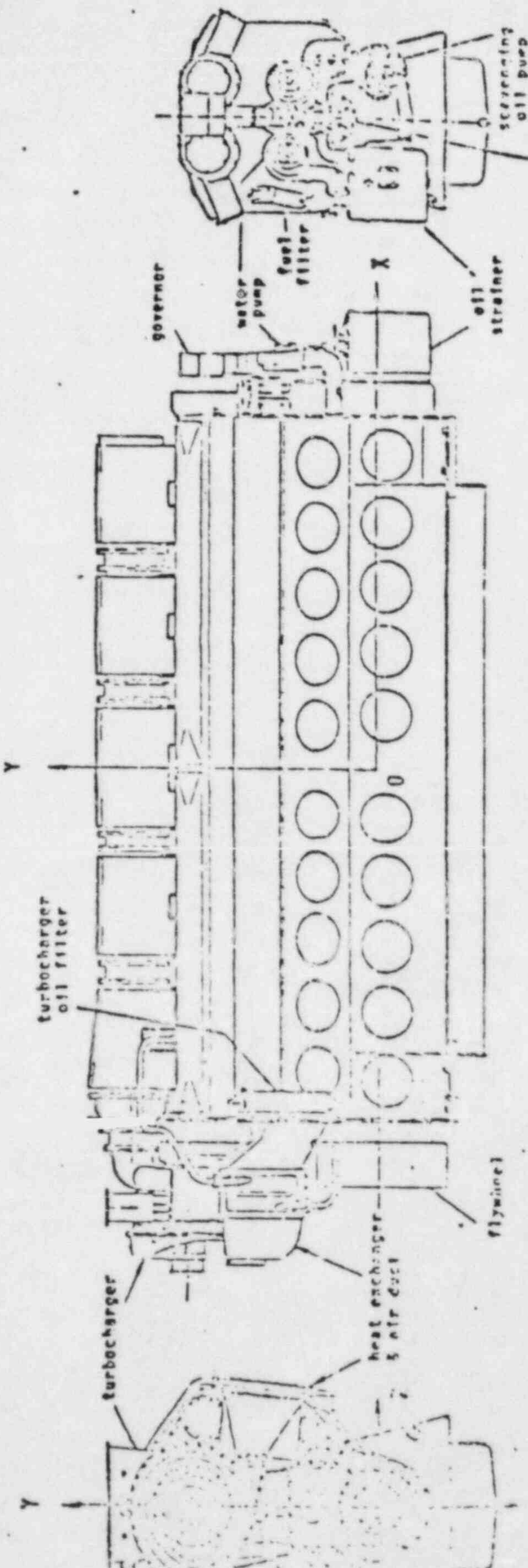
[illegible] shock [illegible]

Shock results are presented. Good documentation however substantiation of g levels & motion on the engine req.

TABLE
20-645E4 (Industrial) Engine Appendages Properties

Components	Overhung Weight (lbs.)	Degrees of Freedom	Fundamental Natural Frequency (Hz)			No. of Mounting Bolts & Bolt Materials	Dynamic Shock That Mounting Bolts Can Withstand
			X-axis	Y-axis	Z-axis		
Water Pump (Left Bank)	155	3	*	310.45	94.48	(6)1/2-20 UNF GM280	176 G's
Water Pump (Right Bank)	155	3	*	379.3	221.46	(6)1/2-20 UNF GM280	182 G's
Heat Exchanger [illegible]	282	3	*	148.09	77.1	(11)1/2-20 Case Side GM280	88 G's
Heat Exchanger (Right Bank)	282	3	*	148.09	77.1	(8)7/16-14 Turbo Side GM300	68 G's
[illegible]	100	3	188.79	*	101.22	(6)1/2-20 UNF GM280	18 G's
Lube Oil Pump	235	3	*	*	150.65	(10)1/2-20 UNF GM280	103 G's
[illegible]	70	3	212.86	223.93	223.93	(12)1/2-20,(2)1/2Studs GM280 SAE 1117	204 G's
[illegible]	273	3	94.48	158.07	101.22	(10)1/2-20 UNF GM280	121 G's
[illegible]	400	5	321.73	374.0	348.0	-----	---
[illegible]	48	3	170.35	--	170.35	(4)3/8-24 Studs SAE 1117	45.5 G's
[illegible]	98	6	--	221.75	221.75	-----	---

* [illegible] fundamental natural frequency greater than 450 Hz.



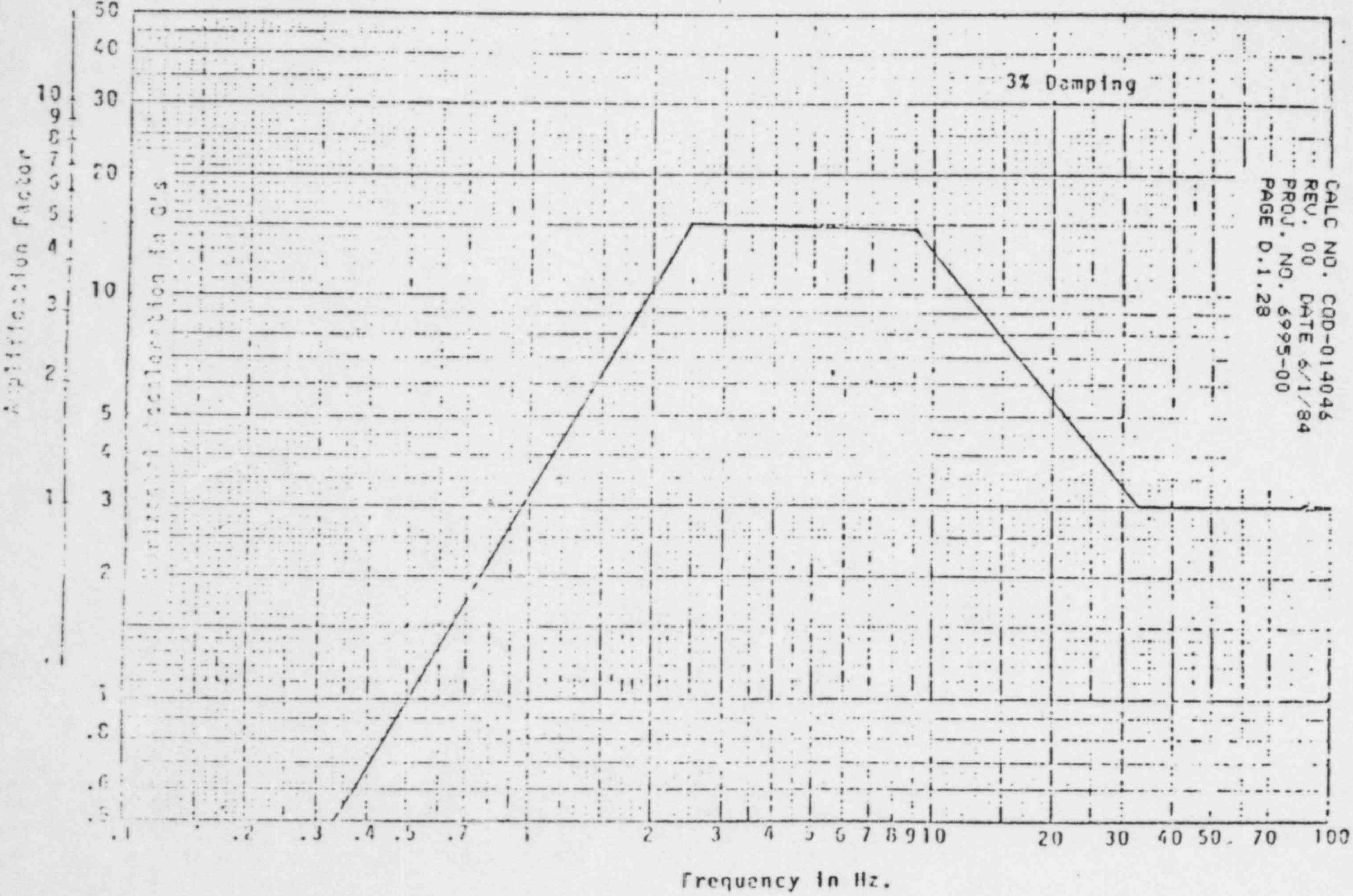
GENERATOR DRIVE END

ACC. DRIVE END

20-CYLINDER ENGINE APPENDAGES ARRANGEMENT

FIGURE 1

FIGURE 2



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(1) Shock Analysis of a General Motors Marine Diesel Engine, 16-567D Turbocharged U411-66-018, April, 1966, by Electric Boat Division, General Dynamics.

(2) Detailed results of computer output (8 volumes) including: ?

Input Data
Mode Analysis
Main Structure, Deflection and Stress Analysis
Engine Appendage Bolting Analysis
Acc. Drive Housing Stress Analysis
Cam Drive Housing Stress Analysis

(3) Heavyweight Shock Test of Electro-Motive Div., General Motors Corporation Main Propulsion Engine for LST 1179 Class Vessels (16-645E5N) U412-67-192, by General Dynamics, June, 1967.

(4) Instrumentation for Heavyweight Shock Test of Electro-Motive Div., General Motors Corporation Main Propulsion Engine for LST 1179 Class Vessels U412-67-292, by General Dynamics, July, 1967.

(5) Technical Memorandum #135 - High Shock Loading Evaluation of The EMD Diesel Engine Turbocharger by Electro-Motive Division, June, 1966. ?

(6) Calculations on High Shock Turbocharger Components #97594 by Electro-Motive Division, June, 1966. ?

(7) Navy High Shock Qualification Tests Model 16-645E5N Marine Engine by Electro-Motive Div., August, 1967.

(8) EMD File #503 - High Shock Dynamic Forces, 1956-1966. ✓

(9) Instrumentation for Heavyweight Shock Test of Electro-Motive Div., GMC 750 KW A.C. Diesel Generator Set for LST 1179 Class Vessels (12-645E5N) #U412-67-355, by General Dynamics, October, 1967.

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(12) Experimental Project Release File on High Shock Engine Components Development-Project 72-870-97594, Book #1, 2, and 3, 1965-1967.

(13) Engine Design Calculation File

ENGINEERING REPORT
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TED BY: M.B. Goetzke MBG

APPROVED BY: J. Chen Jc

DATE: July 18, 1983

REPORT NO.: E77-1A

SECTION: Engine Design

SUBJECT: SUPPLEMENT REPORT ON SEISMIC ANALYSIS OF EMD 645 ENGINE

This report supplements report E77-1, Seismic Analysis of EMD 645 Engine, dated 3/14/77.

As concluded in report E77-1, the EMD 645 series engine will function properly when subjected to a seismic disturbance. This has been thoroughly confirmed. However, even though many minor components were not analytically included in that report, they were qualified experimentally in high shock test as well as in rail service.

Since the issuance of report E77-1, it has been widely accepted, that EMD engines are considered capable of withstanding a seismic disturbance when used for nuclear power station stand-by application. Recently, Southwest Research Institute (SwRI) compiled a list of EMD engine systems that they felt could potentially impact the functionality of our engine if subjected to a severe seismic excitation (see letter to H.M. Arbuckle by C.F. Unrun of SwRI dated 10/ 6/82 on the subject "*Seismic Qualification of EMD 645 EDGS*"). This list included major and minor components of which all major and many minor components were analyzed in report E77-1. Therefore, SwRI would like to see the report updated. EMD has agreed to expand the analysis to cover several additional components and reflect the significant changes made since then. The SwRI list contains 53 items. To this list, EMD added 6 additional areas making a total of 59 items altogether, which will cover all the components for the 645 engine and make a complete report.

To update the report, some criteria and guidelines used in report E77-1 were followed for the analysis. These guidelines include the following:

1) *The 20-cylinder 645 engine is more heavily stressed than are the shorter 645 engines because of the larger mass and greater distance between supports. Thus, the qualification of the 20-cylinder engine components qualifies the full line of turbocharged engines.*

2) *The crankcase and oil pan are considered as a rigid structure. The analysis of each engine component supported by the case and pan either directly or indirectly is based on an assumption that the mounting system will transmit the random vibratory ground acceleration essentially unchanged as a floor loading to the engine components.*

3) *The analysis of each engine component supported by the rigid crankcase depends upon the fundamental natural frequency of that component. If the natural frequency of the component is found to be greater than 33 Hz, a single shock load of 3 g's horizontal and 1 g vertical would be the maximum seismic loading that could be exerted on the components. Therefore, the component or system would be qualified as adequate since it was designed to sustain a lot higher shock load as proved in the high shock tests.*

When the component has a natural frequency lower than 33 Hz, which falls within the range of possible seismic excitation, additional analysis based on appropriate amplification factor to take into account the possibility of resonance associated with the prolonged earthquake will be performed specifically.

4) *In addition, there had been some design changes on some of the engine components since the issuance of report E77-1. This present report has taken that into account and reflects the latest changes which significantly effect the component or system natural frequency and the inertia loading due to weight change.*

The attached table summarizes the results for all the aforementioned 59 items. The table contains three columns, the first listing the item, the second stating the method of qualification based on either analytical results or experimental data, and the third listing comments and conclusions. Reviewing the results from the table, it can be seen that all engine components and systems can withstand a seismic disturbance without failure. Note that the bolting system of the components and systems with natural frequency higher than 33 Hz, were not analyzed individually because they are more than adequate for seismic qualification based on Navy high shock testing. During the Navy high shock testing, EMD engine encountered shock loads of at least 45 g's in both vertical and horizontal direction with no failure occurring.

In light of the information included in this report and EMD report E77-1, it can be concluded that EMD's 20-645 engine can withstand a seismic disturbance without affecting the functionality of the engine and, hence, so does the entire line of EMD 645 engines.

REFERENCES

1. EMD Report E77-1 (3/14/77). Seismic Analysis of EMD 645 Engine.

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2. EMD Calculation File.

3. FDI Report #A-7-81, For Power Systems, Division of Morrison Knudsen, August 24, 1981.

4. "Engine-Generator Shock Loads". Memo by K.D. Mels, February 11, 1974.

Item No.	Item Description	Method of Qualification	Comments and Conclusions
1.	Turbocharger	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
2.	Turbocharger oil filter	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
3.	Turbocharger filter oil supply manifold	EMD calculations, natural frequency f_n = 1004 Hz (see Ref. 2) and also qualified by Navy high shock test (see Ref. 1).	The oil supply manifold is located inside the cam drive housing and is rigidly mounte on the crankcase end plate. The natural frequency is so high that seismic resonance will never occur.
4.	Aftercooler (heat exchanger)	EMD calculations (see Ref. 1). The aftercooler natural frequency calculation was made with the aftercooler duct, Item 5, as a system. They were also qualified by Navy high shock test (see Ref. 1).	Both right and left bank applications previously qualified for seismic operation per Ref. 1.
5.	Aftercooler duct	See Item 4.	See Item 4.
6.	Aftercooler water discharge line	Power Systems calculations, natural frequency f_n = 217 Hz (see Ref. 3).	Natural frequency greater than 33 Hz; therefore, seismi resonance will never occur.

Item No.	Item Description	Method of Qualification	Comments and Conclusions
7.	Aftercooler water discharge "Y"-flange	EMD calculations, natural frequency as a system with pipe assembly f_n = 169 Hz (see Ref. 2).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
8.	Aftercooler water inlet line	Power Systems calculations, natural frequency f_n = 127 Hz (see Ref. 3).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
9.	Exhaust duct	EMD calculations and Navy high shock test (see Ref. 1).	The exhaust duct is part of the turbocharger. The natural frequency of the duct was previously calculated with the turbocharger assembly (see Ref. 1).
10.	Air starter motor	EMD calculations, natural frequency f_n = 464 Hz (see Ref. 2).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
11.	Air starter motor bracket	EMD calculations (the bracket is rigidly mounted on the oil pan) natural frequency for the bare bracket f_n > 1000 Hz. Natural frequency with motors in place f_n = 138 Hz.	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.

Item No.	Item Description	Method of Qualification	Comments and Conclusions
12.	Lube oil separator	EMD calculations: Items 12-15 calculated as a system, natural frequency f_n = 299 Hz.	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.
13.	Lube oil separator bracket	See Item 12.	See Item 12.
14.	Ejector end	See Item 12.	See Item 12.
15.	Ejector chamber	See Item 12.	See Item 12.
16.	Flanged flexible tube	EMD calculations, natural frequency f_n = 537 Hz (see Ref. 2).	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.
17.	Air line	EMD calculations, natural frequency f_n = 493 Hz (see Ref. 2).	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.
18.	Flywheel (ring gear and coupling disk)	EMD calculations, natural frequency f_n > 75 Hz (see Ref. 2). Also see Ref. 4.	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.
19.	Crankshaft gear	Qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1

Item No.	Item Description	Method of Qualification	Comments and Conclusions
20.	No. 1 idler gear	Qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
21.	Turbocharger drive gear	EMD calculations: gear and spring assembly calculated as a system, linear natural frequency f_n = 1000 Hz (see Ref. 2).	Natural frequency greater tha 33 Hz; therefore, seismic resonance will never occur.
22.	Spring assembly	See Item 21.	See Item 21.
23.	No. 2 idler gear	Qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
24.	Camshaft drive gear	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
25.	Exhaust chamber	The exhaust chamber is rigidly mounted to the top of the crankcase and was qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
26.	Exhaust chamber expansion joint	EMD calculations, natural frequency f_n > 3400 Hz (see Ref. 2) and also qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.

Item No.	Item Description	Method of Qualification	Comments and Conclusions
27.	Adapter	Qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
28.	Adapter to turbocharger (expansion joint)	Qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
29.	Engine ramps		Ramps have no effect on engi operation.
30.	Viscous damper (harmonic balancer)	EMD calculation (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
31.	Governor	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
32.	Governor drive	Qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
33.	Governor drive gear	Qualified by Navy High Shock Test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
34.	Injector control linkage	EMD calculations, natural frequency for the shaft f_n = 235 Hz, for the lever f_n = 122 Hz, and for the control rod f_n = 97 Hz (see Ref. 2).	All natural frequencies grea er than 33 Hz; therefore, seismic resonance will never occur.

Item No.	Item Description	Method of Qualification	Comments and Conclusions
35.	Fuel oil filter	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
36.	Scavenging oil pump	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
37.	Fuel pump	Navy High Shock tested (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
38.	Main lube oil and piston cooling pump	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
39.	Oil strainer	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
40.	Fuel oil manifolds	EMD calculations, natural frequency f_n = 121 Hz (see Ref. 2).	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.
41.	Pipe assembly, fuel pump to fuel oil manifold	EMD calculations, natural frequency f_n = 72 Hz (see Ref. 2).	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.

Item No.	Item Description	Method of Qualification	Comments and Conclusions
42.	Pipe assembly, flanged block to fuel manifold and valve assembly	EMD calculations, natural frequency f_n = 466 Hz (see Ref. 1).	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.
43.	Fuel oil crossover manifold	Power Systems calculations, natural frequency for the assembly f_n = 167 Hz (see Ref. 3).	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.
44.	Water pump	EMD calculations (see Ref. 1).	Previously qualified for seismic operation per Ref. 1
45.	Water inlet elbow	Power Systems calculations, natural frequency f_n = 167 Hz (see Ref. 3).	Previously qualified by Navy High Shock Test also (see Ref. 1).
46.	Engine crankcase pressure detector	Qualified by vendor.	Qualified for seismic operation by vendor.
47.	Pipe, strainer to scavenging pump	Power Systems calculations, natural frequency f_n = 167 Hz (see Ref. 3).	Natural frequency greater th 33 Hz; therefore, seismic resonance will never occur.

Item No.	Item Description	Method of Qualification	Comments and Conclusions
48.	Piston cooling elbow	The piston cooling elbow is rigidly mounted to the crankcase and cylinder liner. Experimental tests show that the natural frequency f_n = 1200 to 1400 Hz.	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
49.	Strainer elbow	EMD calculations, natural frequency f_n = 2556 Hz (see Ref. 2).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
50.	Shutdown solenoid	EMD calculations, natural frequency f_n = 395 Hz (see Ref. 2).	Rotary stroke solenoid, linear acceleration will not affect function; also natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
51.	Water outlet adapter	Power Systems calculations, natural frequency f_n > 67 Hz (see Ref. 3).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
52.	Overspeed trip housing and cover	EMD calculations, natural frequency for cover f_n = 163 Hz; natural frequency for housing and cover assembly f_n > 1000 Hz (see Ref. 2).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.

Item No.	Item Description	Method of Qualification	Comments and Conclusions
53.	Accessory drive housing	EMD calculations, natural frequency for the housing assembly, $f_n > 1000$ Hz (see Ref. 2).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
54.	Water manifold	EMD calculations, natural frequency $f_n = 562$ Hz (see Ref. 2).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
55.	Water jumpers	EMD calculations, natural frequency $f_n = 1620$ Hz (see Ref. 2).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
6.	Turbocharger lube oil soak back line	Power Systems calculations, natural frequency $f_n = 67$ Hz (see Ref. 3).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
7.	Crankcase top deck oil drain line	Power Systems calculations, natural frequency $f_n > 67$ Hz (see Ref. 3).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.
8.	Turbocharger soak back pump	Qualified by Navy High Shock test (see Ref. 1).	Previously qualified for seismic operation per Ref. 1.
9.	Water pipe assembly, between water pumps	EMD calculations, natural frequency $f_n = 119$ Hz (see Ref. 2).	Natural frequency greater than 33 Hz; therefore, seismic resonance will never occur.

Effect Of The Hawaiian Earthquake of April 26, 1973 Upon EMD Power Generating Sets Located In The Hawaiian Islands

By

Hugh A. Williams, Jr.

July 23, 1976

Electro-Motive Division
General Motors Corporation
LaGrange, Ill. [illegible]

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Effect of the Hawaiian Earthquake of April 26, 1973 Upon EMD Power Generating Sets Located in the Hawaiian Islands

Introduction

As a manufacturer of diesel engine generator sets, Electro-Motive Division of General Motors Corporation has been requested frequently to supply evidence that these sets can successfully withstand seismic shocks of the frequency and amplitude usually produced by earthquakes. Several approaches might be pursued to demonstrate the ability of these units to withstand seismic shock. One very effective method involves documentation of the effect of a major earthquake upon such units.

On April 26, 1973, a major earthquake occurred in the Hawaiian Islands. On that date, thirteen (13) power generating sets which had been manufactured by Electro-Motive Division were installed in three major utility networks on the islands of Hawaii, Maui, and Kauai. All of these thirteen units were subjected to the shocks and aftershocks produced by this earthquake.

Objectives

The objectives of this report are two-fold:

1. To describe the major earthquake which occurred on April 26, 1973 at 10:26:[illegible].6 Hawaiian standard time.
2. To describe the effect of this earthquake upon EMD power generating sets located on the islands of Hawaii, Maui, and Lauai.

Description of the April 26, 1973 Earthquake

The April 26, 1973 earthquake was of magnitude 6.2 on the Richter Scale, and its epicenter was located at latitude 19.9° North; longitude 155.1° West, near the northeast coast of Hawaii Island, near Honomu, approximately 10 miles north of Hilo.

A rather complete report on this earthquake has been prepared by the Seismic Engineering Branch of the U.S. Geological Survey, as compiled from [illegible] reports and [illegible]

Description of the April 26, 1973 Earthquake (Continued)

questionnaire card canvass. This account categorizes the damage wrought by this earthquake according to the relative seismic intensity, beginning with Intensity VIII on the island of Hawaii, and concluding with certain communities on Kauai Island in which the earthquake was not felt. A copy of this report is contained in the Appendix.

It is anticipated that additional information will be published by the U. S. Geological Survey relative to Strong-Motion Instrumental Results of the Earthquake of April 26, 1973 and aftershocks.

EMD Power Generating Sets Located in the Hawaiian Islands

At the time of the April 26, 1973 earthquake, there were thirteen (13) EMD -manufactured power generating sets installed in the Hawaiian Islands. Eleven (11) were MP-45 sets which include Model 20-645E4 engines and two (2) were S36GA sets which include Model 16-567 D4 engines.

Since the earthquake, three (3) additional EMD MP-45 generating sets have been installed on the Island of Hawaii. A complete list of EMD Power Generating Sets Located in the Hawaiian Islands is shown in Table 1. Of the total of sixteen (16) EMD generating sets, fourteen (14) sets are EMD Model MP-45 sets.

Description of EMD MP-45 Generating Set

General Motors Electro-Motive Division manufactures an MU Type power plant, designed for the specific purpose of providing AC power at 4160 volts, 3 phase, 60 cycle for peaking, spinning reserve, and base load operation; modified plants are capable of operation at lower voltage and frequency. The plant is automatic in operation, requiring only start and stop signal to go on and off the line. Normally, full load operation in peaking mode is attained approximately 3 minutes after a start signal is received depending on the number of units being used. If the plant is equipped for fast starts, full load operation [illegible] should only be used in [illegible] sible to control the power [illegible] panels in the switch gear [illegible].

Description of EMD MP-45 Generating Set

Each MU type power plant consists of from one to five MP type diesel driven generating units. An MP-45 generating unit consists of a skid mounted enclosure housing a 20-645E4 diesel engine directly coupled to a Model A20C synchronous generator, and the required engine supporting auxiliaries, such as oil coolers and filters, an engine cooling water system, an engine exhaust snubber, a fuel system, an engine control panel, and an electrical control cabinet. The generating unit outline is shown in Figure 1.

The control equipment necessary for completely automatic operation of the MU type power plant is housed in the switch gear station regardless of the number of generating units in the installation. An outline of the switchgear station is contained in Figure 2.

A typical site layout and typical foundation arrangement are contained in Figure 3.

With respect to the types of foundations utilized in the Hawaiian Islands, the three (3) MP-45 units on Kauai are on conventional railroad ties except for the switchgear station which is mounted on a reinforced concrete slab. Figure 4 depicts the three (3) MP-45 units and switchgear station of Kauai Electric Co., located at Port Allen. The two (2) S36GA units are not shown in this photograph.

All of the MP units and switchgear stations on the islands of Hawaii and Maui are mounted on reinforced one-foot thick concrete slabs, essentially as depicted in Figure 5, an actual construction drawing. The three (3) MP-45 unit site of the Hilo Electric Light Co., Ltd., located at Waimea (Kamuela) on the island of Hawaii is shown in Figure 6.

The reinforced concrete slabs are clearly shown in Figure 7, which is a photograph taken of the Maalaea, Maui site of Maui Electric Co., Ltd., following installation of the first of three MP45 units in 1971. Also shown is the switchgear station.

Effect of the Eart[illegible] of April 26, 19[illegible] Power Generating [illegible]

The following [illegible] ption of the effect of subject earth-quake upon EMD gen[illegible] tained from [illegible] and from Electro-M[illegible] was on Kauai, [illegible]

On the Island of Hawaii

Out of a total of five MP-45 units installed on the island of Hawaii in the Hilo Electric Light Company's system, two of the three units (#13 and #14) at Waimea were operating at 10:26 AM, April 26, 1973, each carrying 2,500 KW. When the earthquake hit, these two MP-45 units, along with a steam and a hydro unit were knocked off the line, creating a loss of about one-third of the system load capacity. The #13 and #14 Waimea units tripped, presumably on over-current relay.

At 10:36 AM, the electric company gave an "all" unit start signal. The #12 unit at Waimea and the #16 unit at Hilo came on line (fast start), carrying 2500 KW each. The #15 unit at Hilo had a high speed differential relay target showing and failed to come on line at that time, possibly due to the earthquake, since it was not on line during the earthquake. It was reset later.

At 11:25 AM, the #13 and #14 units at Waimea were reset. This was done by an electric company employee who was called out, since the units are unattended in the morning. Only the #14 unit was put on line at 2,500 KW, following a 10 minute idle period.

Several days after the earthquake, two EMD service engineers made a thorough inspection of the MU sites at Hilo and Waimea and could find no evidence of damage or shifting of the power units or switchgear. Furthermore, none of the slabs were cracked.

No repairs were needed on the MP45 units. Only the relays were checked and reset. All units were operated satisfactorily later. An electric company official also stated that there was no evidence of the units having moved on the reinforced concrete slabs.

On the Island of Maui

Three MP-45 units were installed on the Island of Maui in the Maui Electric Company's system at [illegible] at the time of the April 26, 1973 earthquake. Based [illegible] limited information, it is believed [illegible] operating.

When queried relative to the effect of the [illegible] upon the EMD generating [illegible] company [illegible] that a review of [illegible] sessions with operating [illegible]

On the Island of Kauai

On the Island of Kauai at the time of the April 26, 1973 earthquake, there were three MP-45units and two S36GA units installed in the Kauai Electric Company's system at Port Allen. One of the MP-45 units was shut down at the time. All other units were running, including the system steam plant.

An EMD field service engineer was present at the MU site at Port Allen and was inside one of the MP-45 units at the time of the earthquake. He was not aware of the earthquake until advised by the electric company personnel after he emerged from the unit. There were no reported problems with the MP-45 or S36GA units, either electrically or mechanically, as a result of the earthquake.

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Fig. 1 - Generating Unit Outline
Fig. 2 - Switchgear Outline
Fig. 3 - Typical Site Layout and Typical Foundation Arrangement
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Fig. 6 - MU Site at Waimea (Kamuela), Hawaii
Fig. 7 - MU Site at Maalaea, Maui

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Table 1 - EMD Power Generating Sets Located in the Hawaiian Islands

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Table 1

EMD Power Generating Sets

Located in the Hawaiian Islands

Location	Customer Unit No.	Delivery Date	Engine Model	Generator Model	EMD Gen. Set Model	Unit Rating-KW	
						Base Load	Peak
HAWAII - Hilo Electric Light Co., Ltd.							
Waimea-Kamuela	12	8-70	20-645E4	A20C	MP45	2500	2750
	13	2-72	20-645E4	A20C	MP45	2500	2750
	14	2-72	20-645E4	A20C	MP45	2500	2750
Kanoelehua-Hilo	15	6-72	20-645E4	A20C	MP45	2500	2750
	16	6-72	20-645E4	A20C	MP45	2500	2750
-		9-73	20-645E4	A20C	MP45	2500	2750
Kailua-Kona		3-74	20-645E4	A20C	MP45	2500	2750
		3-74	20-645E4	A20C	MP45	2500	2750
MAUI - Maui Electric., Ltd.							
Maalaea		10-71	20-645E4	A20C	MP45	2500	2750
		6-72	20-645E4	A20C	MP45	2500	2750
		7-72	20-645E4	A20C	MP45	2500	2750
KAUAI - Kauai Electric Co.							
Port Allen		3-64	16-567D4	A15C	S36GA	--	2000
		4-64	16-567D4	A15C	S36GA	--	2000
		5-68	20-645E4	A20C	MP45	2500	2750
		5-68	20-645E4	A20C	MP45	2500	2750
		5-68	20-645E4	A20C	MP45	2500	2750

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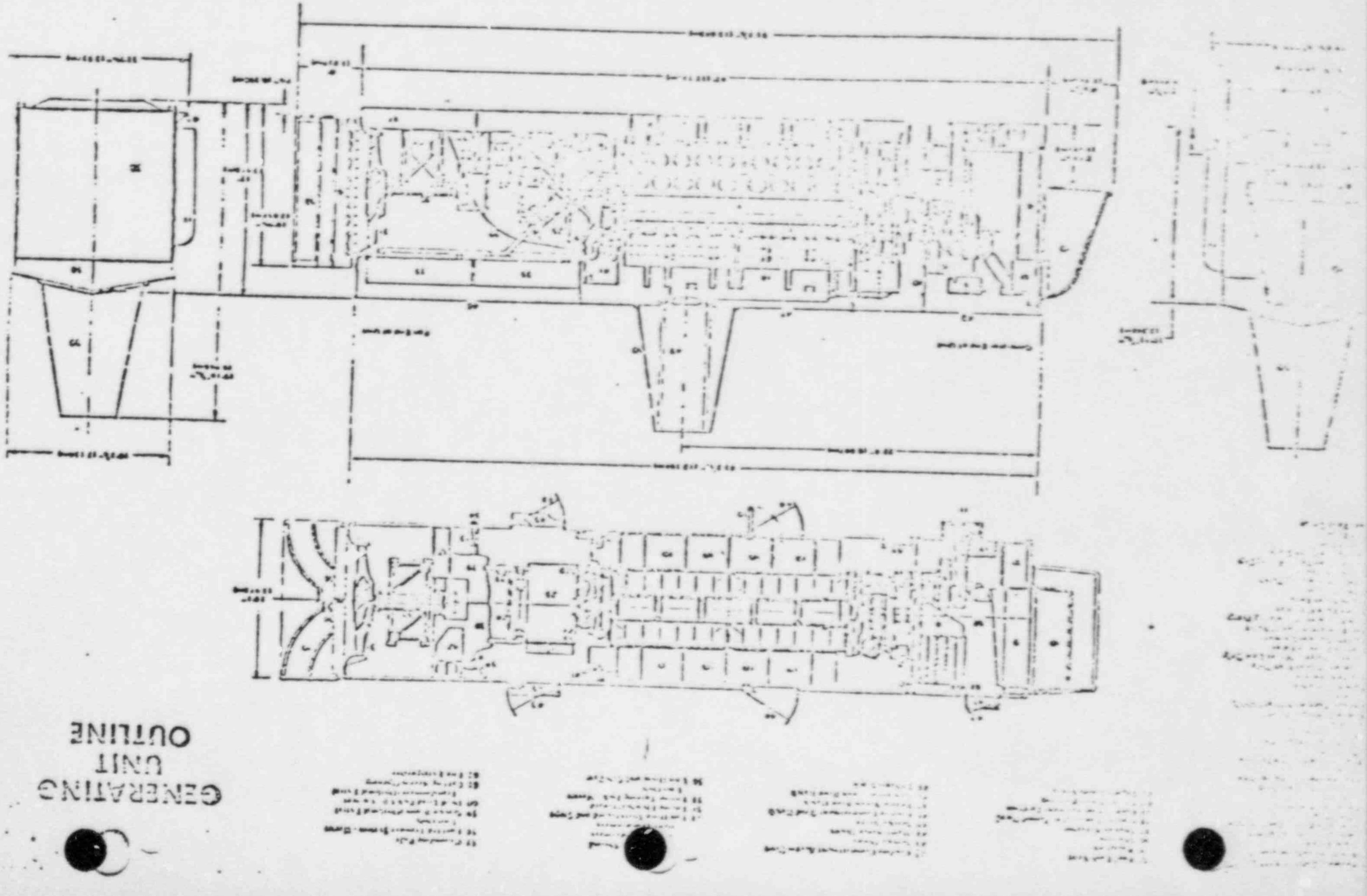
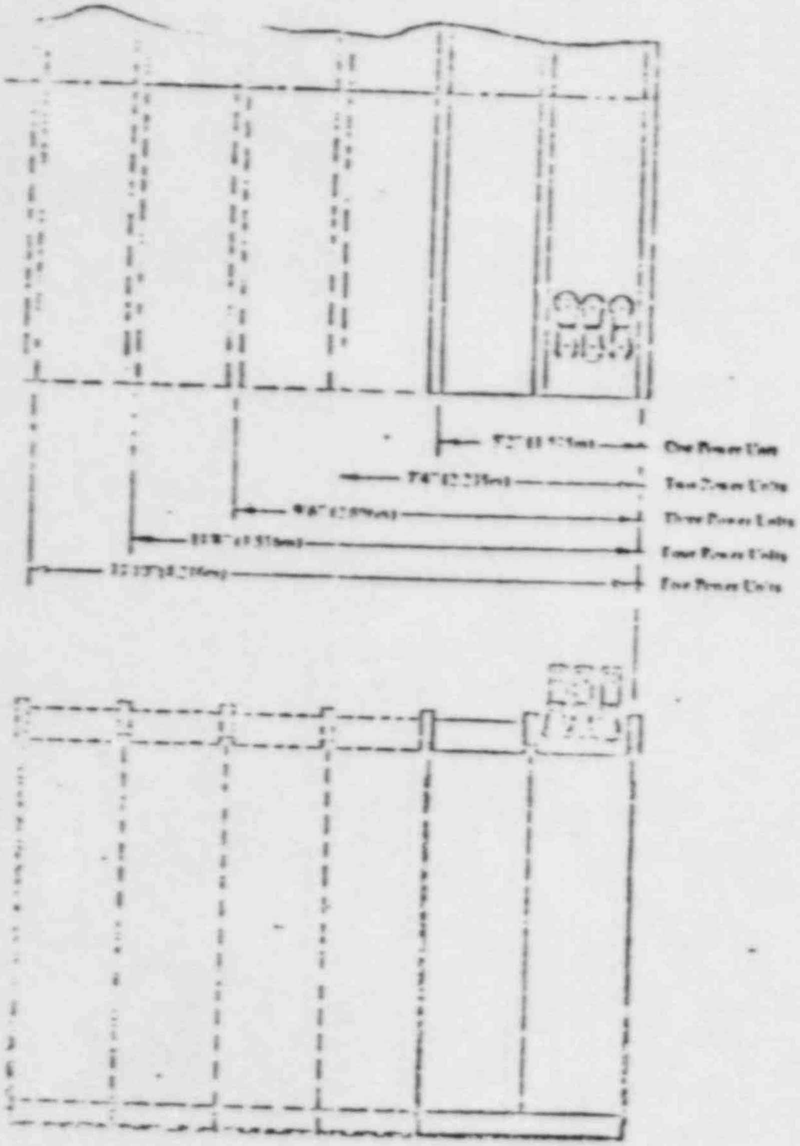
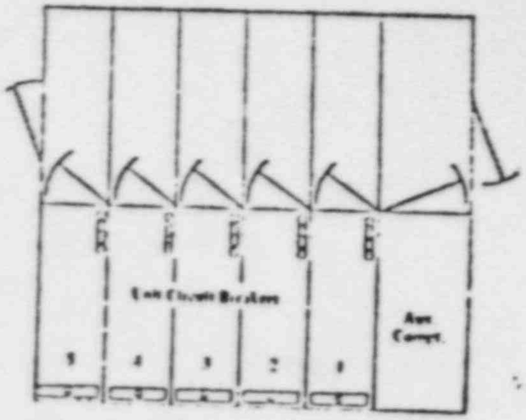


SWITCHGEAR JTLINE





Floor Plan – Five Unit Switchgear Station

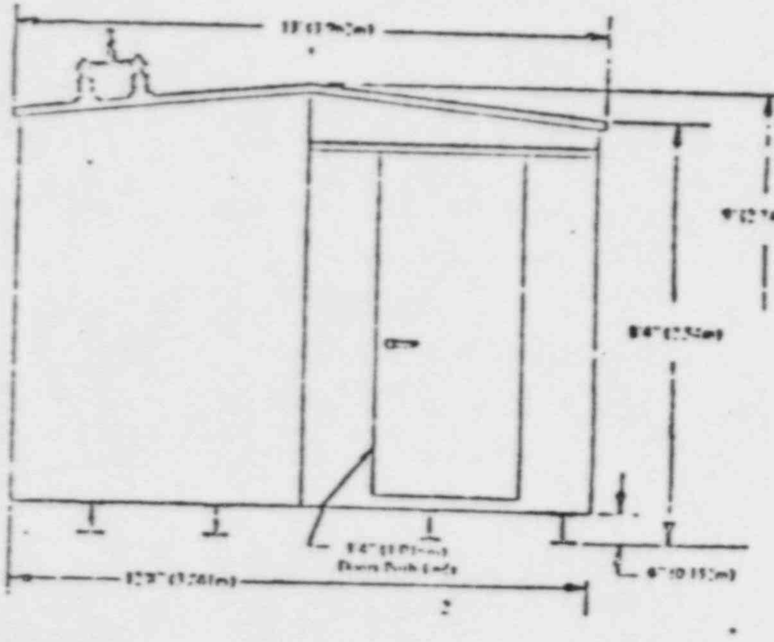


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Figure 2

APPENDIX

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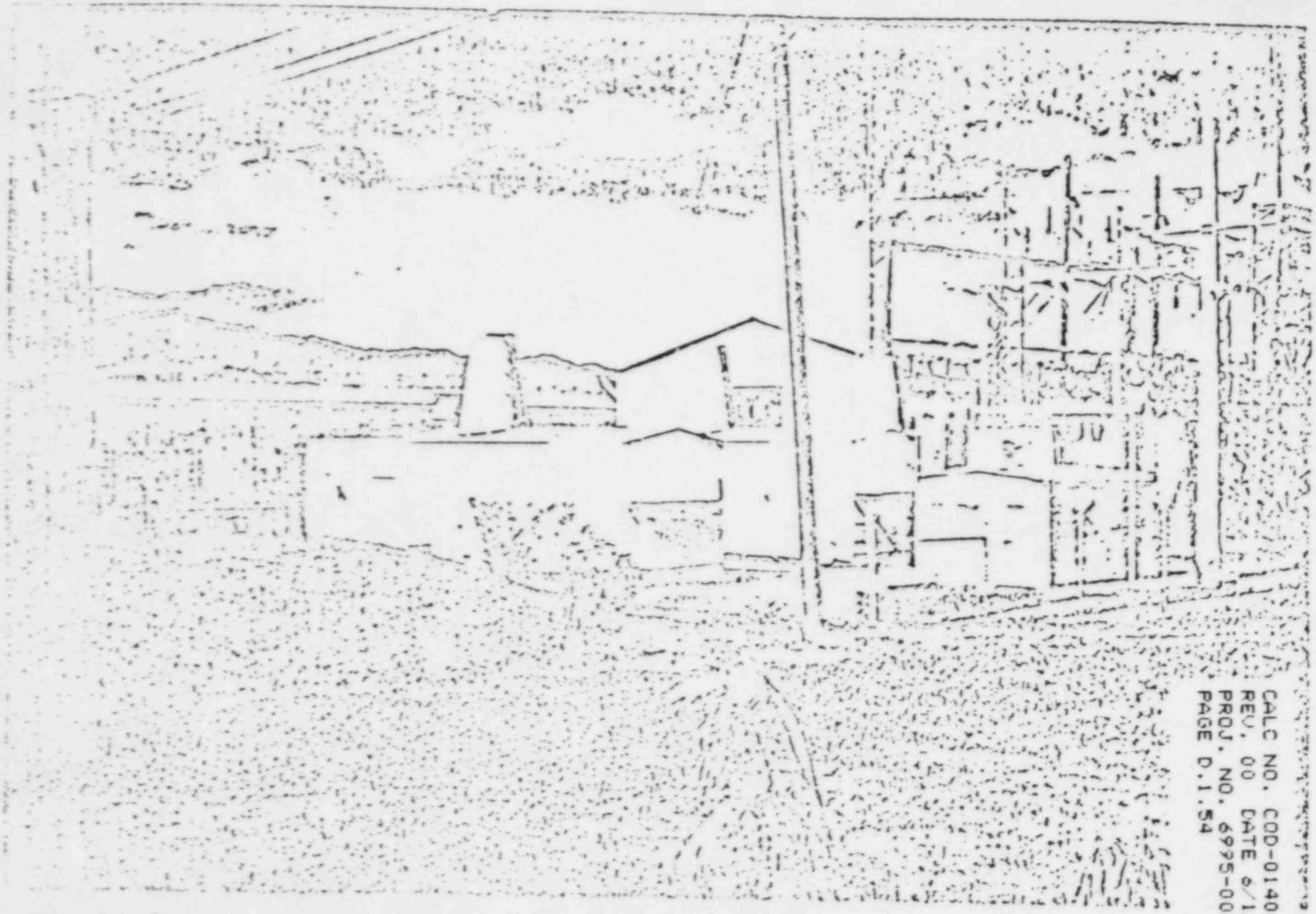

Figure 4 - [illegible] Site of Kauai Electric Co. at Port Allen, Kaui

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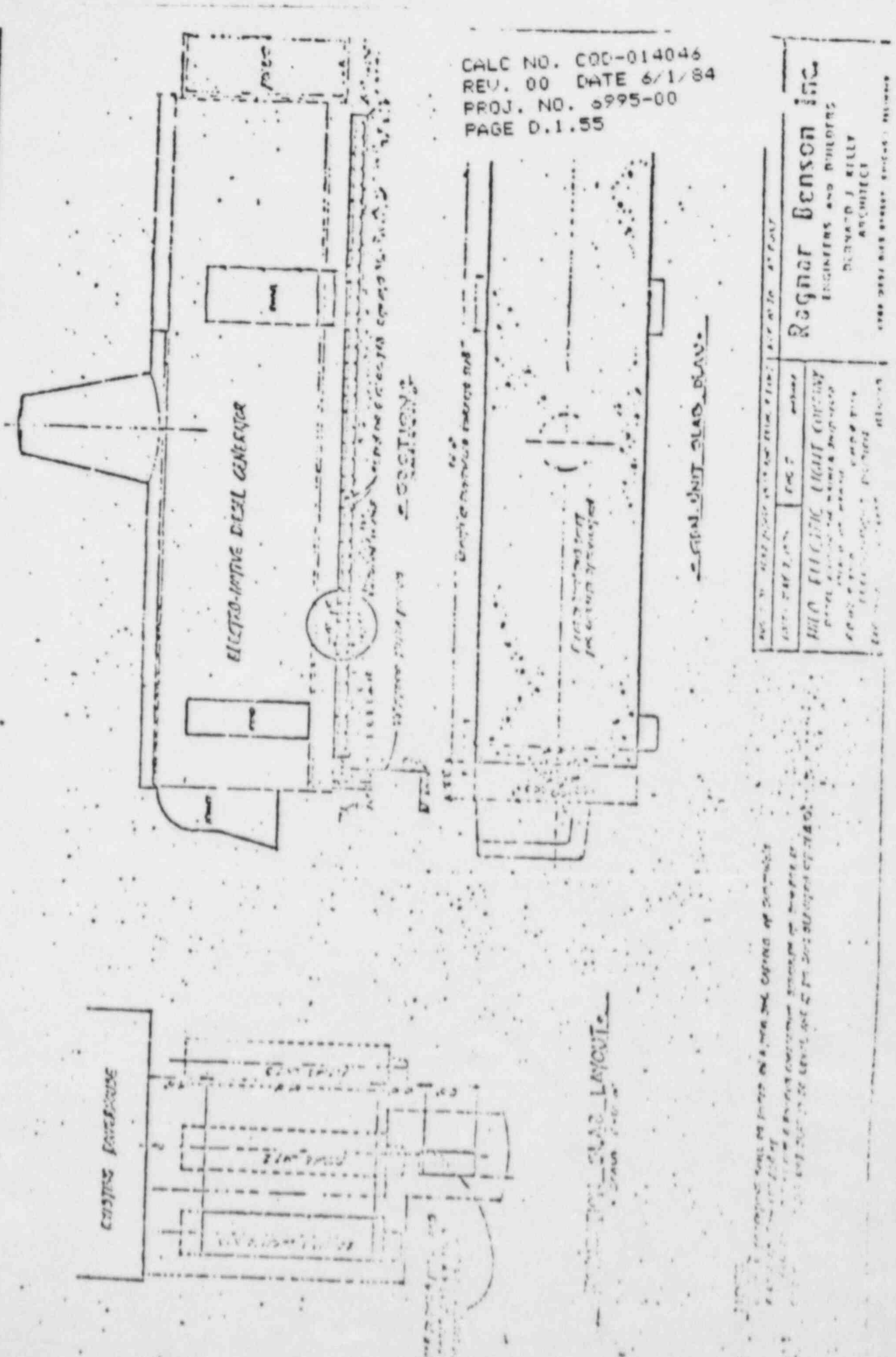

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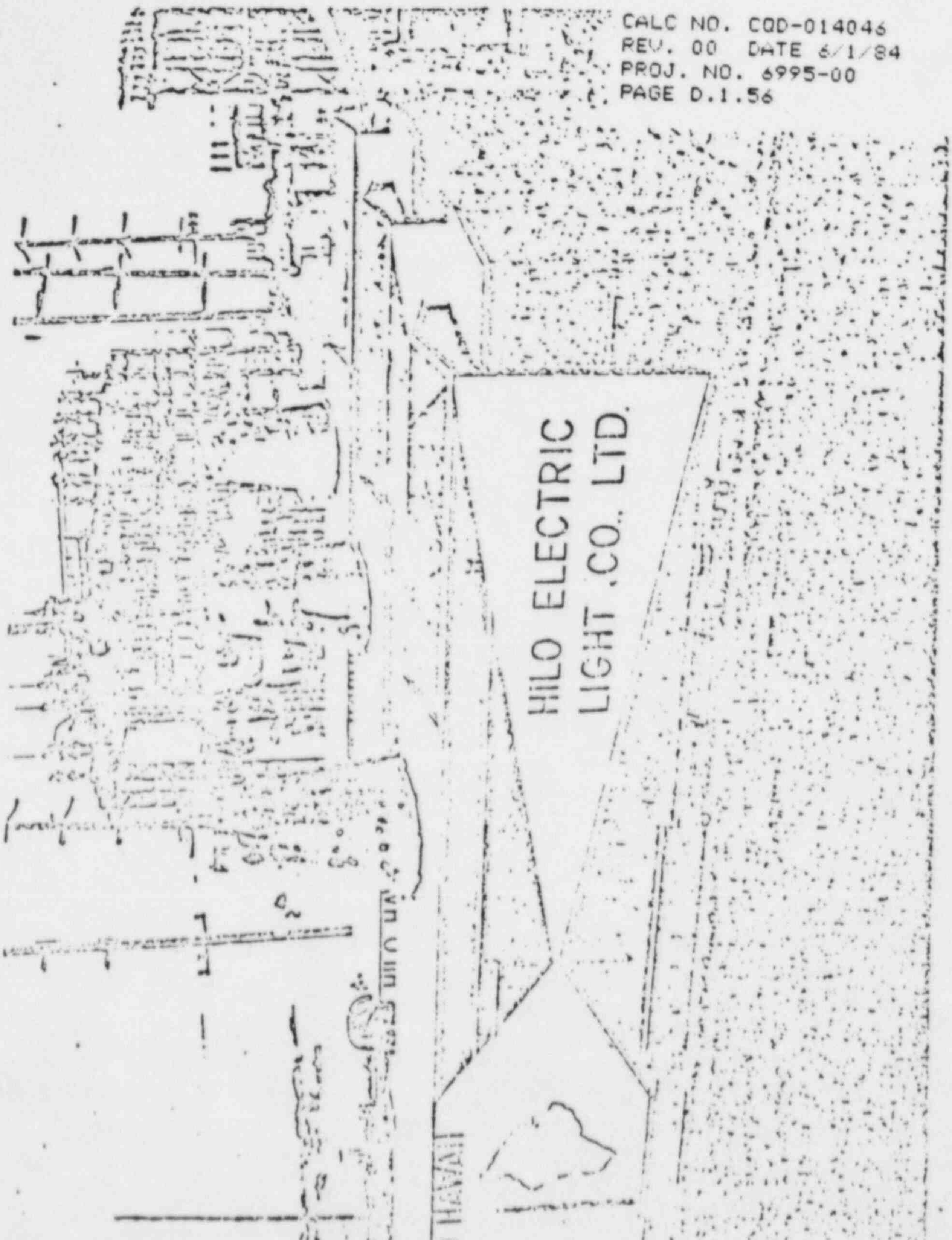


Figure 6 - Site of Hilo Electric Light Co., Ltd., at Waimea, Hawaii

Figure 7 - HB Site of Maui Electric Co., Ltd., at Maalaea, Maui

APPENDIX

REPORT ON HAWAIIAN EARTHQUAKE OF APRIL 26, 1973,
(AS COMPILED FROM PRESS REPORTS AND QUESTIONNAIRE CARD CANVASS)

26 April, 10:26:28.0 Hawaiian Standard Time. Magnitude 6.2 (Hawaiian Volcano Observatory, U.S. Geological Survey).

Epicenter 19.9° N., 155.1° W., near coast of northeast Hawaii Island, near Honomu about 16 km north of Hilo, at a depth of 50 km, HVO.

Felt over a very large area--from east coast of Hawaii Island northwest through the principal Hawaiian Islands of Maui, Kahoolawe, Lanai, Molokai, Oahu, and Kauai, a distance of approximately 595 km. Maximum intensity VIII. No fatalities occurred; at least 11 persons were injured. Damage was estimated at about $5.6 million. Damage to buildings, roads, water, gas, and power facilities caused authorities to declare a state of emergency over most of the northeast coastal areas of Hawaii Island.

The following was reported by the press: Through the afternoon of April 27, 200 homeowners had reported losses exceeding $700,000 over an area from Hilo northwest to Waimea, a distance of approximately 64 km. The Civil Defense Administration said 70 "Big Island" business firms, many of them losing display windows, had reported damage totaling $375,000, and that this figure was expected to rise. State facilities sustained at least $950,000 damage. Figures included at least $300,000 damage to four schools which were temporarily closed. Damage at Hilo's Pier 1 was estimated at $350,000 for necessary repairs alone. State roads sustained more than $100,000 in damage, the heaviest damage occurring along the [illegible]

26 April, 10:26:28.0--Continued.

Gulch areas where the Belt Highway was closed at three points during most of the day on the 26th, and limited to local one-way traffic on the 27th. County losses were estimated at $893,000. The biggest item was County roads with an estimated $600,000 in damages for immediate repairs--$300,000 damage to Wainaku and Kaiwiki county roads near Hilo; $15,000 in the Waimea area; $90,000 in the Hamakua district; $135,000 in North Hilo; and $60,000 elsewhere. Next highest figure of damage was the waterworks damage in the Kaiwiki, Kaiele, and Papaikou areas with 10 major problems costing $278,000 to repair. Several hundred residents north of Hilo remained without water throughout the 27th. Many powerlines were down between Hilo and Hakalau. Power was switched off between Pepeekeo and Hakalau to reduce shock hazard. Telephone company reported only one exchange was out of order, this at Kawailani. On the 27th, most electric, telephone, and gas services had been restored to all areas of the Island except for isolated outages in the Wainaku, Kaiwiki, Amauulu, and Kaumana areas. Lines between Pepeekeo and Hakalau which had been knocked down were back on poles. Telephone line service drops--lines from poles to individual homes--were knocked out in the Amauulu, Kaiwiki, and Wainaku areas but most were replaced by the 27th. Engineer for the Hilo Gas Company reported that all gas customers, except in isolated cases in Wainaku and Kaumana, were being served on the 27th. Seven major landslides occurred along the Hamakua Highway, closing off three gulches and completely [illegible]

26 April, 10:26:28.0--Continued.

Point Beach Park. The makai side of Waihau Lane in Puueo (Hilo) collapsed, damaging the roadway, chain-link fence, a telephone pole, and breaking a waterpipe which sent a geyser of water into the air. At Wainaku Overlook, police barricated the entrance to the overlook because of ever-widening cracks of 20 to 25 cm in the roadway. There were 46- to 68-meter-long cracks in the earth along Kaiwiki Road in Wainaku. At Hilo Harbor, Pier 1, a 366-meter-long concrete pier, was split from end to end by a 1.27- to 2.54-cm crack; two other piers sustained lesser damage. Downtown Hilo was ordered closed. All traffic came to a halt. Moving cars could not keep their course and bounced around. One man was pinned in the rubble of a collapsed building, the Typewriter Center. A home in Hilo's Puueo district reportedly collapsed. Many windows were broken. Traffic lights were knocked out. Plaster fell from many buildings to the sidewalk. All around the block bordered by Kamehameha Avenue, Haili Street, Keawe Street, and Kalakaua Street, many plate glass windows were shattered. There were no reports of major damage at the Hilo Airport, but cracks were observed in the airport restaurant walls. In the control tower, men were forced to hold onto a rack to keep their balance. The Malia Apartment Building (Hilo), a two-story cinder block building, was damaged. The outside walls on the two ends of the rectangular-shaped, 24-unit building were torn loose and cinder blocks fell. The top of the [illegible] a gaping hole in the ground. On the [illegible]

26 April, 10:26:28.0--Continued.

River, an older plantation house collapsed. It was reported that 17 homes (no specific location given) were shaken off their foundations and that five collapsed completely. The Hawaiian Telephone Company building on Kinoole Street (Hilo) sustained structural damage but the extent was not known on April 27. Damage to homes on Halaulani Street (Hilo) was extensive. Most homes on Halaulani Street have rock foundations, which were shifted several centimeters from their original locations. Most rock walls around the foundations caved in.

INTENSITY VIII:

Haaheo (near Hilo). Press reported heavy damage to waterlines. Haaheo School was not expected to open until May 1 owing to a major water leak.

Hakalau. Felt by and frightened all in community (observers walking in new and old, two-story wood frame buildings). Windows, doors, and dishes rattled strongly; buildings creaked strongly. Loud earth noises. Ground cracked; landslides; water disturbed. Tombstones overturned. Electricity off; telephone out. TV fell off stand. "Nearly every home suffered loss. We live on a sugar plantation community. The water systems were poor. Damage to our pipelines was heavy, all required repair. Residents were without water for about 4 days. Even schools were closed for 2 days owing to shortage of water. It's back to normal now (May 4). Some people are still scared, even when a small shock occurs. We are thankful that no lives were lost and that no one was hurt, and that damage was not too high." [illegible]

26 April, 10:26:28.0. Intensity VIII--Continued.

Hilo. (Press) Subsidence reported at Kawaihae Harbor. Hilo's Pier 1 cracked, with damage estimated at $350,000; two other piers sustained lesser damage. Damage to Wainaku and Kaiwiki county roads (near Hilo), $300,000; Hamakua district, $90,000. Landslides and ground cracked. Waterworks damage in the Kaiwiki, Kaiele, and Papaikou areas estimated at $278,000. Electric, telephone, and gas services disrupted. Cesspools collapsed. One downtown building collapsed. Buildings cracked and shifted on foundations. Both ends of a two-story cinder block apartment building were shaken loose and cinder blocks fell. Federal building roped off because of wide cracks at all three levels. Hilo College buildings sustained structural damage. Damage at Central Fire Station estimated at $10,000; County Building, $5,000; Hilo Processing Corporation, $139,000; Mauna Kea Sugar Company, $260,000. At Hilo Union High School, plaster fell and some acoustical panels popped open in the school's newest building; one student was hit on the head by a flying panel. Plaster fell from many buildings to the sidewalk. Many plate glass windows cracked in downtown Hilo.

Hilo. Felt by and frightened all in community (observers walking and sitting on second floor of old, well-built, four-story cement building). Windows, doors, and dishes rattled some; building cracked severely. Loud earth noises. Trees and bushes shook; vehicles rocked. Ground cracked; landslides; water disturbed. Chimneys, tombstones, elevated water tanks, etc., cracked, twisted, and overturned. Hanging objects swung violently. Small

26 April, 10:26:28.0. Intensity VIII--Continued.

objects shifted, overturned, and fell. Furniture shifted and overturned. Plaster cracked, broke, and fell. Windows broke. Cement cracked. "Many residents on this island sustained considerable damage to kitchen goods. Cabinet and refrigerator doors were opened and food was lost along with glassware and dishes." Ground: Filled in, level.

Honomu. Felt by and frightened all in community (observers in various types of buildings--old, new, one- and six-story wood frame). Windows, doors, and dishes rattled; buildings creaked very loudly. Loud earth noises. Trees and bushes shook; vehicles rocked; woman driving truck was frightened. Ground cracked; many landslides along with fallen trees; water disturbed. Many tombstones twisted and overturned. Hanging objects swung violently north-south, and fell. Many small objects shifted, overturned, and fell. Furniture moved, some broke; freezer and refrigerator moved and doors opened. Windows cracked. Plaster cracked, broke, and fell. Concrete floor cracked. "Many items fell from my store shelves. It took me 3 days to clean up the mess. My TV and transistor also were damaged. Damage great." Ground: Rocky, gravelly, loose, but observer in level, sloping, and steep area.

Kaumana (about 8 km southwest of Hilo). Press reported heavy damage to water-lines. Some homes were shaken loose from foundations.

Ninole. Felt by and frightened all in community (observer walking in well-built, one-story wood frame building). Windows, doors, and dishes rattled; buil

26 April, 10:26:28.0. Intensity VIII--Continued.

creaked strongly. Loud earth noises. Trees and bushes shook; vehicles rocked. Ground cracked; landslides; water disturbed. Chimneys, tombstones, elevated water tanks, etc., cracked, twisted, and overturned. Hanging objects swung violently. Small objects shifted, overturned, and fell. Plaster cracked, broke, and fell. Damage moderate. Ground: Level, sloping.

Ookala. Felt by and awakened all in community; frightened many (observer standing in old, one-story wood frame building). Windows, doors, and dishes rattled. Faint earth noises. Trees and bushes shook; vehicles rocked. Ground cracked; landslides; water disturbed. Chimneys, tombstones, elevated water tanks, etc., twisted and overturned. Hanging objects swung moderately. Small objects shifted, overturned, and fell. Furniture shifted. Plaster cracked, broke, and fell. Some windows cracked. Some houses shifted slightly off foundations. Ground: Gravelly, level.

Papaaloa. Felt by, awakened, and frightened all in community; general panic (observer standing in old, one-story wood frame building). Windows, doors, and dishes rattled; building rumbled. Loud earth noises. Trees and bushes shook; vehicles rocked. Ground cracked; landslides; water disturbed. Chimneys, tombstones, water tanks cracked and overturned. Hanging objects swung violently. Small objects shifted, overturned, and fell. Furniture shifted and broke. Plaster cracked, broke, and fell. Windows cracked. Roof leaked. Damage great. Ground: Sloping.

26 April, 10:26:28.0. Intensity VIII--Continued.

Papaikou (about 6 km north of Hilo). Felt by, awakened, and frightened all in community. "Panic in a sense." Observer standing on first floor of old, two-story wood frame building. Windows, doors, and dishes rattled. Building swayed east-west. Loud earth noises. Trees and bushes shook and swayed; vehicles rocked. Ground cracked; landslides; water disturbed. Main waterline broke; cesspool caved in. All hollow tile concrete steps crumbled and fell to the ground. Hanging objects swung violently east-west. Small objects overturned and fell; all objects on shelves fell. Furniture shifted and overturned. Plaster broke and fell. Windows cracked. Foundation moved about 7.6 cm. "Many concrete buildings had ceiling and wall cracks. Wood frame structures moved from foundations." Damage great. Press reported some homes were shaken loose from foundations. Front door of one home completely off and back door damaged. One report to police stated a telephone pole was "hanging" and a water tank had collapsed. Heavy damage to waterlines. The most severe structural problems to schools occurred at the Kalanianaole School at Papaikou, one of the hardest hit areas. The roof dropped 7 to 10 cm and a ceiling warp appeared between the two principal sections of the main building. Ground: Red clay type of soil, level.

Pepeekeo. Felt by, awakened, and frightened all in community (observers walking, sitting, and standing in well-built, one-story concrete tile building). Windows, doors, and dishes rattled. Loud earth noises. Trees and bushes shook; vehicles rocked. Ground cracked; landslides; water disturbed. Chimneys, tombstones,

26 April, 10:26:28.0. Intensity VIII--Continued.

and water tanks shaken severely. Stone wall overturned. Hanging objects swung violently. Small objects shifted, overturned, and fell. Furniture shifted, overturned, and broke. Plaster cracked, broke, and fell. Building cracked and settled. Windows cracked. Damage moderate. Ground: Compact, level.

Intensity VII:

Honokaa. Felt by all and frightened many in community (observer working in one-story wood frame building). Windows, doors, and dishes rattled. Building swayed and seemed as though it would collapse. Loud earth noises. Vehicles rocked. Ground cracked; landslides; water disturbed. Some tombstones and elevated water tanks cracked and overturned. Hanging objects swung violently. Small objects shifted, overturned, and fell. Furniture shifted. Plaster cracked, broke, and fell in some buildings. Windows cracked in few buildings. Damage moderate. Ground: Compact, filled in, level. Press reported the State Building, opened 3 years ago, had extensive interior damage. At the Honokaa School, electricity and water were off for a while; steam pot plumbing dislodged; one building vacated temporarily owing to suspicious-looking cracks.

Kamuela. Felt by observer standing in well-built, one-story brick building. Loud earth noises. Ground cracked; landslides. Chimneys and tombstones cracked, twisted, and overturned.

Kapaau. Felt by and frightened all in community (observer [illegible]

26 April, 10:26:28.0. Intensity VII--Continued.

one-story wood frame building with concrete floor). Windows, doors, and dishes rattled; building creaked loudly. Loud earth noises. Houses shook; trees and bushes shook; vehicles rocked. Hanging objects swung moderately. Small objects shifted, overturned, and fell. Furniture shifted. Plaster cracked between ceiling and walls. Damage slight. Ground: Compact, level. Press reported damage at the Kalahikiola Church was estimated at $75,000.

Keakealani School. School building and water tank shifted on foundations (press).

Kohala. Press reported damage at the Kohala High School was estimated at $20,000.

Laupahoehoe. Felt by, awakened, and frightened all in community (observer sitting in old, well-built, one-story building). Windows rattled. Building swayed; wall swayed. Loud earth noises. Small objects fell--flower vases, books, and papers on shelf. Furniture shifted. At the Laupahoehoe School, press reported damage at the community-school library was estimated at $25,000. Nearly all of the new library ceiling panels fell and one of the minor columns was split. Also, there was much shattering of glass panels facing makai.

Mahukona Bridge (North Kohala district). Press reported damage to the bridge was estimated at $20,000.

North Kohala District. Miscellaneous damage to a number of [illegible] in the North Kohala district was estimated at $[illegible]000.

26 April, 10:26:28.0. Intensity VII--Continued.

Paahuau. Felt by and frightened all in community (observer standing in old, one-story wood frame building). Windows rattled. Loud earth noises. Trees and bushes shook; vehicles rocked. Landslides. Tombstones cracked, twisted, and overturned. Hanging objects swung violently. Small objects fell. Furniture shifted. Windows cracked. Ground: Rocky, sloping.

Pololu Valley Lookout (North Kohala district). Press reported damage at the lookout was estimated at $40,000.

Waimea (Hawaii Island; about 64 km northwest of Hilo). (Press) In the Waimea area, road damage was estimated at $15,000. At the Waimea School, damage was estimated at $25,000, owing to broken lenses and light fixtures. Stone chimney fell through the roof at the Waimea Steakhouse. Police reported glass shattered and other minor damage at various other places, including two schools.

Intensity VI:

Halaula. Frightened many in community (observer standing in old, one-story wood frame building). Windows, doors, and dishes rattled; large buildings creaked. Moderate earth noises. Trees and bushes shook; vehicles rocked. Landslides. Hanging objects swung moderately. Small objects shifted. Furniture shifted. Windows cracked. Damage slight. Ground: Level.

Hana. Felt by many in community. Landslides. "Near Hana Bay, some vases fell from shelves; dishes rattled. We were on west side of Haleakala Crater in a car and did not feel the shock. However, when we returned to Hana,

26 April, 10:26:28.0. Intensity VI--Continued.

we noticed old stone walls surrounding a pasture were crumbled and had fallen southeast--toward Hilo." Ground: Steep, east slope of Haleakala Crater.

Hana District (east Maui Island area). Press reported papaya trees were knocked down in the Hana district.

Hauula. Frightened many in community (observer sitting in old, fairly well-built, one-story brick building). Faint earth noises. Vehicles rocked. Underground pipeline broke. Ground: Compact, level.

Hawaii National Park. Felt by all and frightened many in community (observer walking in old, one-story wood frame building). Windows, doors, dishes, everything rattled. Seemed as if the ceiling would collapse. Faint earth noises. Vehicles rocked. Ground cracked. Small objects shifted. Ground: Compact, level. Press reported the famed Thurston Lava Tube and the eruption area were closed. Considerable rubble fell into the Tube, causing a blocked passageway. At the eruption site, cracks had widened significantly at the ledge overlooking Mauna Ulu vent eruption. Landslides occurred at Kilauea Caldera. Few pipes were shaken loose.

Keaau. Felt by and frightened all in community (observer walking in 10-year-old, one-story cement tile building). Windows and doors rattled; building creaked. Moderate earth noises. Trees and bushes shook; vehicles rocked. Hanging objects swung moderately. Small objects shifted, overturned, and fell. Plaster cracked. Ground: Compact, level.

26 April, 10:20:28.0. Intensity VI--Continued.

Kealakekua Bay. Rocks fell (press).

Kurtistown. Water tank almost overturned (press).

Maui Island (West Maui-Pali Highway). Press reported rocks fell on the West Maui-Pali Highway.

Mauna Kea Beach Hotel (South Kohala district). Much plaster fell (press).

Naalehu. Felt by and frightened all in community (observer walking in well-built, one-story wood frame building). Windows, doors, and dishes rattled loudly; building creaked very loudly. Loud earth noises. Trees and bushes shook; vehicles rocked. Hanging objects swung moderately. Small objects shifted and overturned. Furniture shifted. Damage slight. Ground: Compact, level.

Naniloa Surf Hotel. (Press) A couple of waterpipes broke, some leakage. Walls over entrance to Crown Room were cracked. Some guests ran outside. Damage very slight.

Paauilo. Felt by and frightened all in community (observer sorting mail in well-built, one-story building; general merchandise store). Windows, doors, and dishes rattled. Loud earth noises. Hanging objects swung violently. Small objects shifted and fell; glasses broke. Furniture shifted. Walls cracked. Plaster cracked, broke, and fell. Damage moderate. Ground: Level.

Pahoa. Felt by and frightened all in community (observer sitting in well-built, one-story hollow tile building). Windows, doors, and dishes rattled; building "groaned." Loud earth noises. Trees and bushes shook; vehicles rocked. Hanging objects swung moderately. Ground: Rocky, level.

26 April, 10:26:28.0. Intensity VI--Continued.

Parker Ranch (Parker Ranch Shopping Center). Damage to fallen merchandise was estimated at $500 (press).

Pohakuloa (about 43 km due west of Hilo). Press reported there was about a 0.6-cm crack running across Bradshaw Field, an old Army airstrip. There also was minor damage at the Post Exchange and Club; bottles fell.

Puainako. Much merchandise fell from store shelves (press).

Volcano. Felt by observer standing in well-built, one-story wood frame building with cement floor. Windows, doors, and dishes rattled; building creaked slightly. Faint earth noises. Water splashed from elevated water tank. Hanging objects swung. Ground: Rocky.

West Hawaii Island (Kohala, Kona, and Kau districts). Press reported there was no serious damage in these districts. Several roads were strewn with rocks but all major highways were passable. In Kona and Kau, hundreds of meters of old stone walls collapsed.

Intensity V: Aiea, Captain Cook, Haiku, Hawi (plaster cracked; damage slight), Hickam Air Force Base (Honolulu), Holualoa, Honaunau (some plaster cracked; damage slight), Honolulu, Kahului (plaster on joints opened), Kailua, Kailua Kona. Kaneohe and Kaneohe Marine Corps Air Station, Kaunakakai, Kaupo, Kealakekua, Kihei, Kualapuu, Kula, Lahaina, Makawao, Mountain View (ceiling panels in school loosened; press), Mount Haleakala Summit (Maui Island), Pahala, Puunene, Ulupalakua, Wahiawa, Wailuku, Waimanalo, and Wheeler Air Force Base (near Kunia).

26 April, 10:25:28.0--Continued.

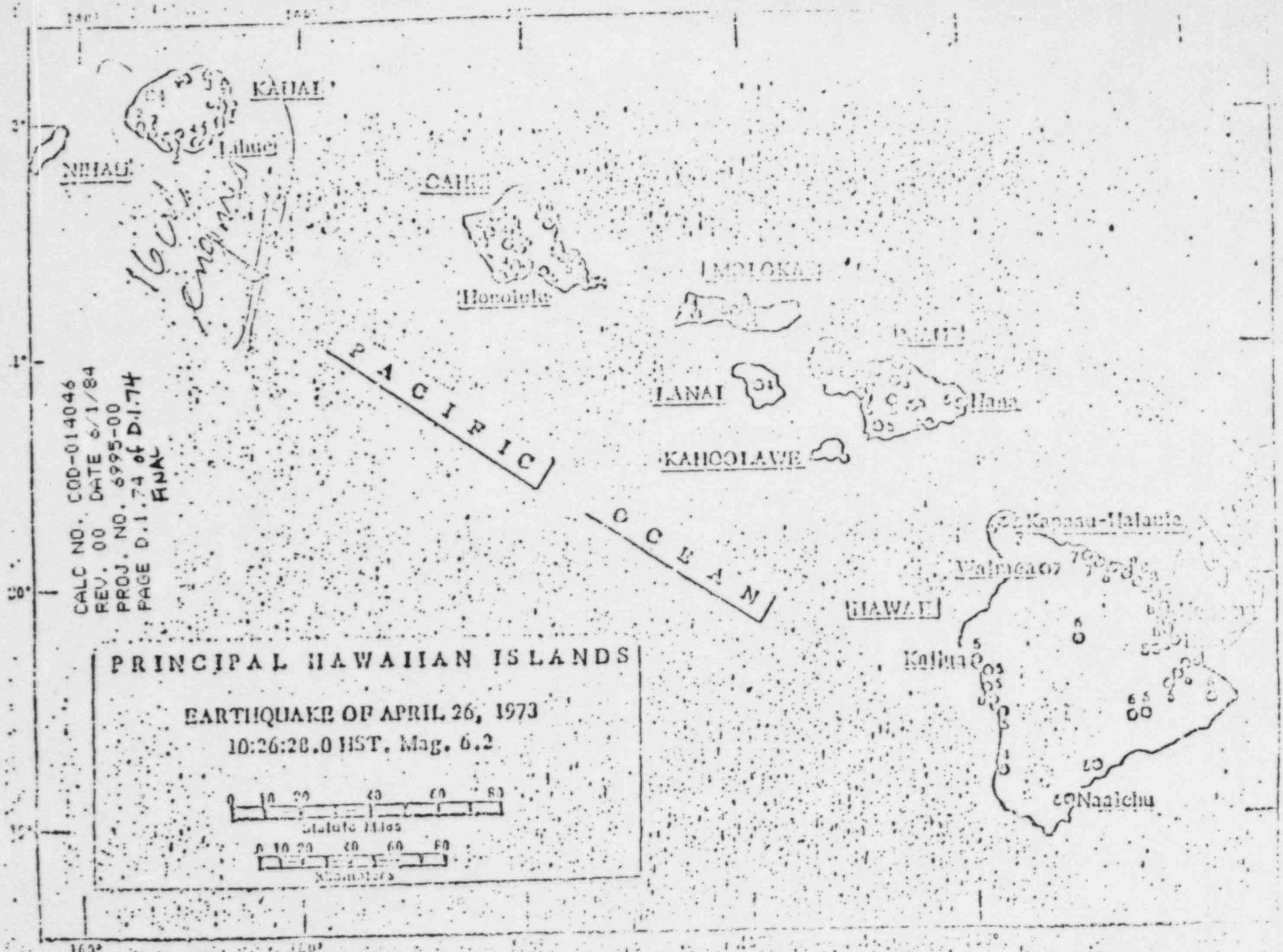
Intensity IV: Haliimaile, Haleiwa, Hanalei, Hoolehua, Kaaawa, Kahakuloa, Kahuku, Kalaupapa, Kokee Ranger Station (northwest coastal area of Kauai Island), Kula, Laie, Lanai City, Lihue, Makapuu Point (Coast Guard Light Station), Maunaloa, Mount Kaala (northwest Oahu Island), Pearl City, Puhi, Waialua, Waianae, and Waipahu.

Intensity I-III: Eleele, Ewa Beach, Hanamaulu, Hanapepe, Kapaa, Kekaha, Kilauea, Lawai, and Waimea (Kauai Island).

Reported not felt: Anahola, Kalaheo, Kaumakani, Kealia, Koloa, and Makaweli (all on Kauai Island).

- - - -

Seismic Engineering
U.S. Geological Survey
San Francisco, Ca. 94131
December 18, 1973



PRINCIPAL HAWAIIAN ISLANDS

EARTHQUAKE OF APRIL 26, 1973
10:26:28.0 HST, Mag. 6.2

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REV. 00 DATE 6/1/84
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1.3

F.1

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APPENDIX-F PHOTOGRAPHS

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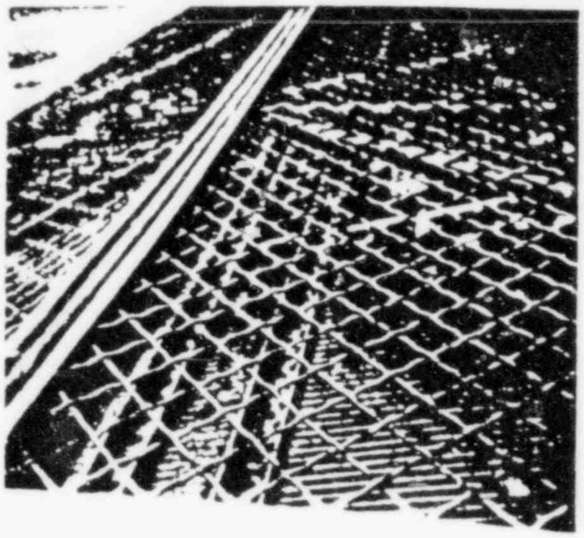
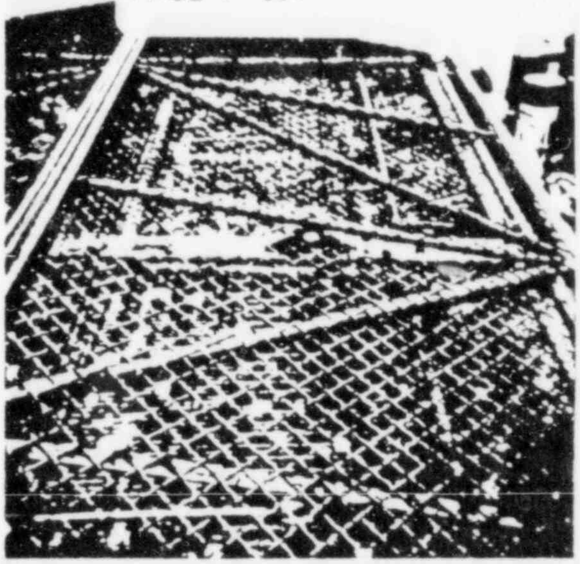
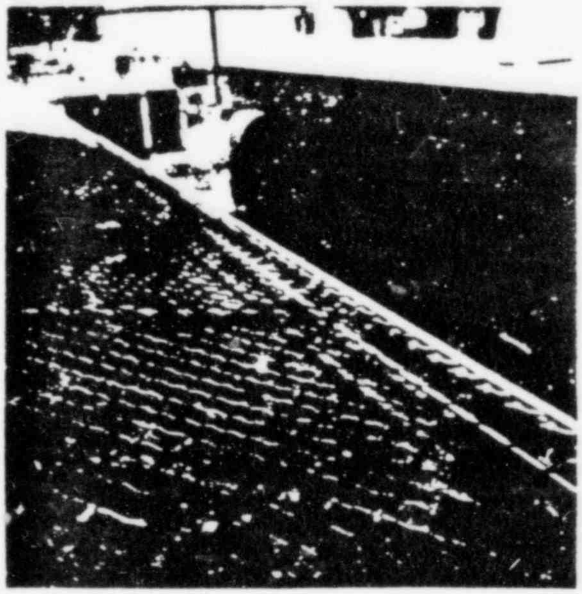


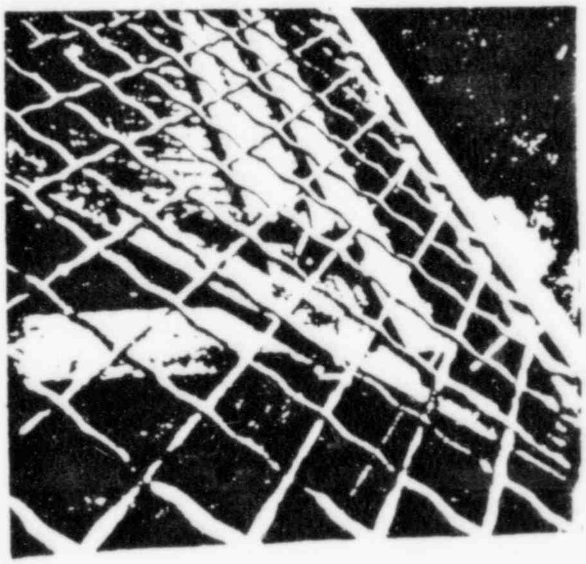
1. Radiator, External top view.



2.a. Radiator, External top view.



2.b



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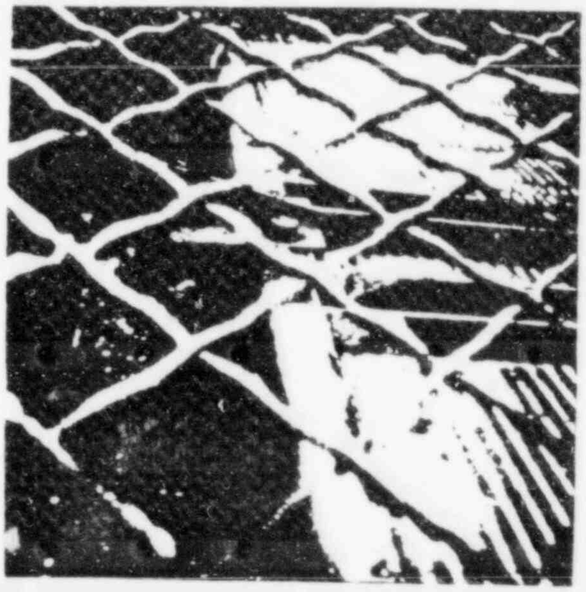
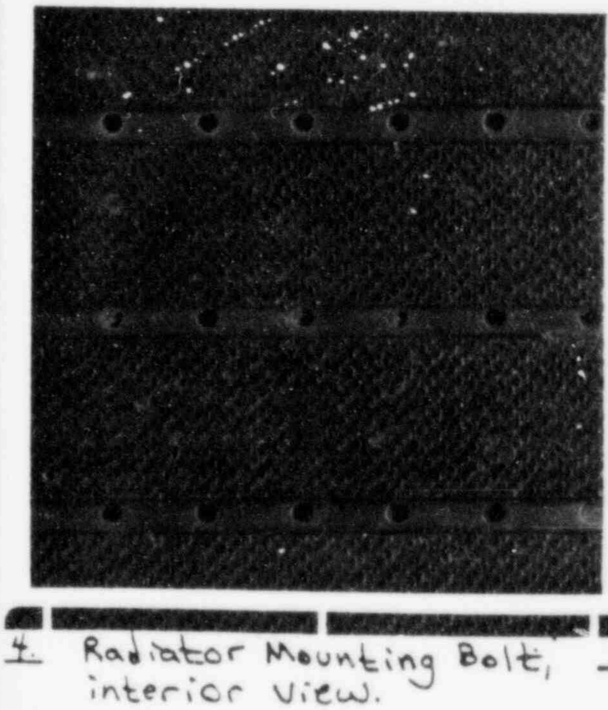


3. Radiator Bracket, External top view.



4. Radiator Mounting Bolt, interior view.

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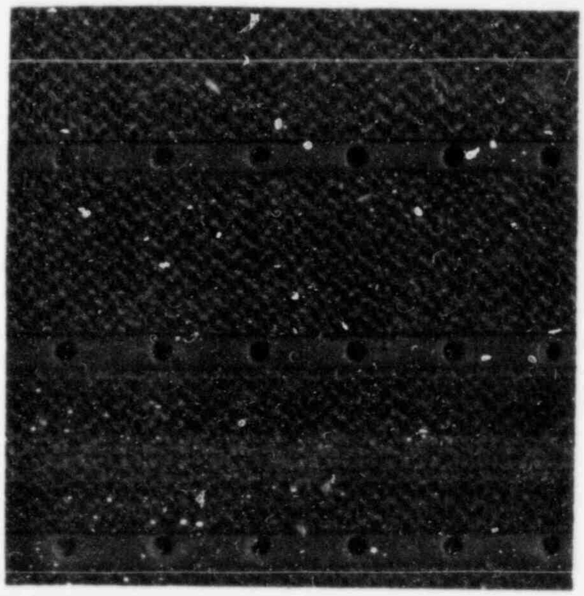
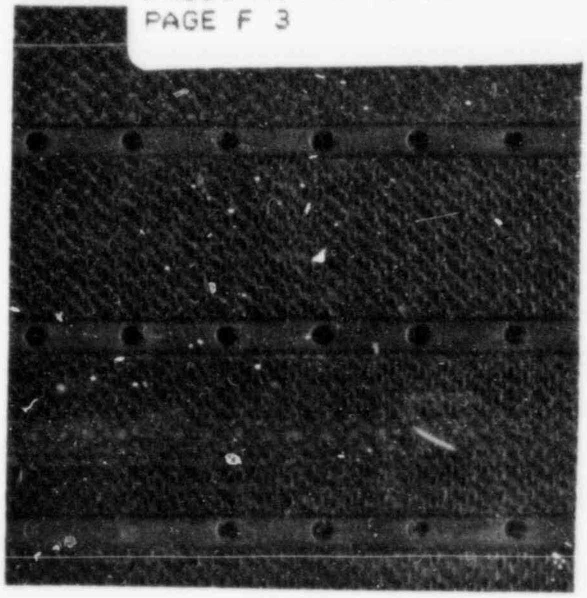
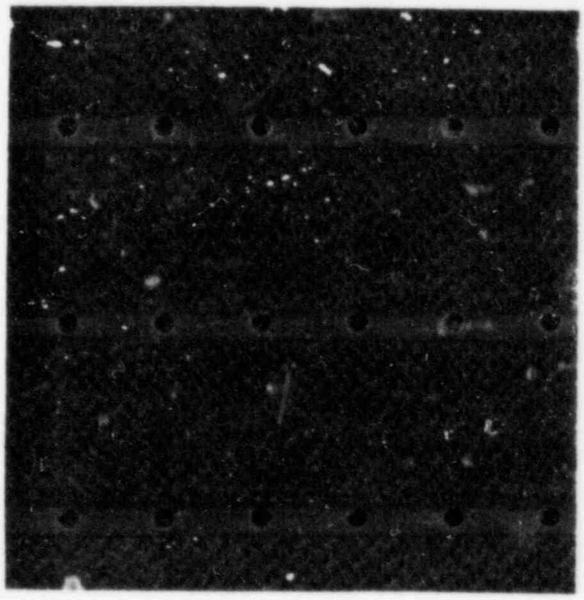


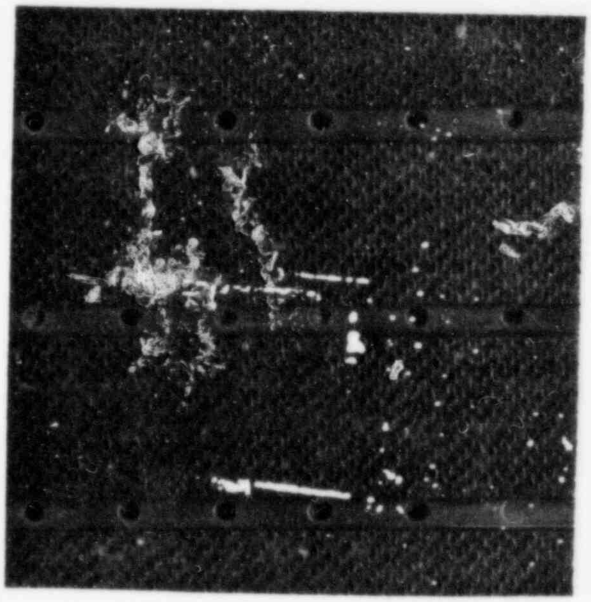
5 Piping, Radiator, Engine outlet to radiator



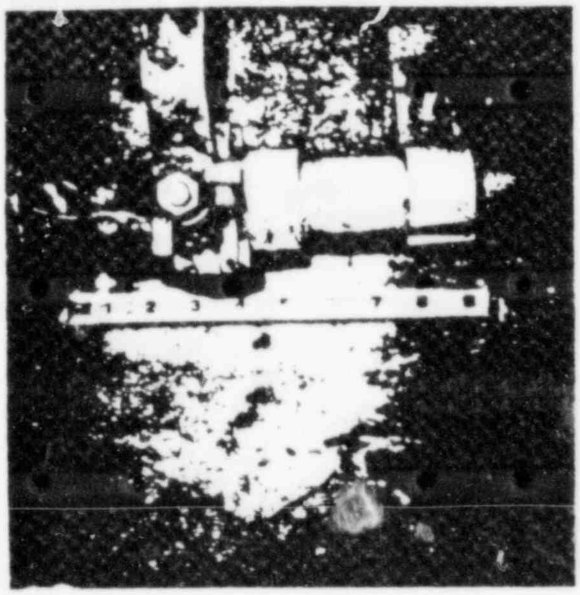
6. Piping, Engine outlet to Radiator



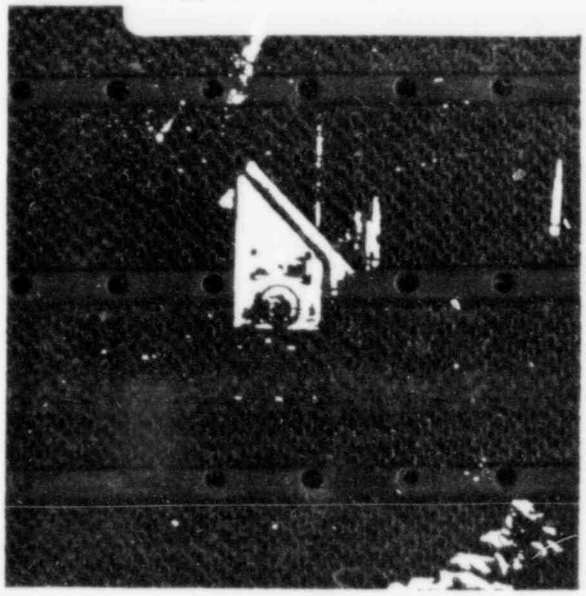
7 Piping, Radiator to Lube oil cooler



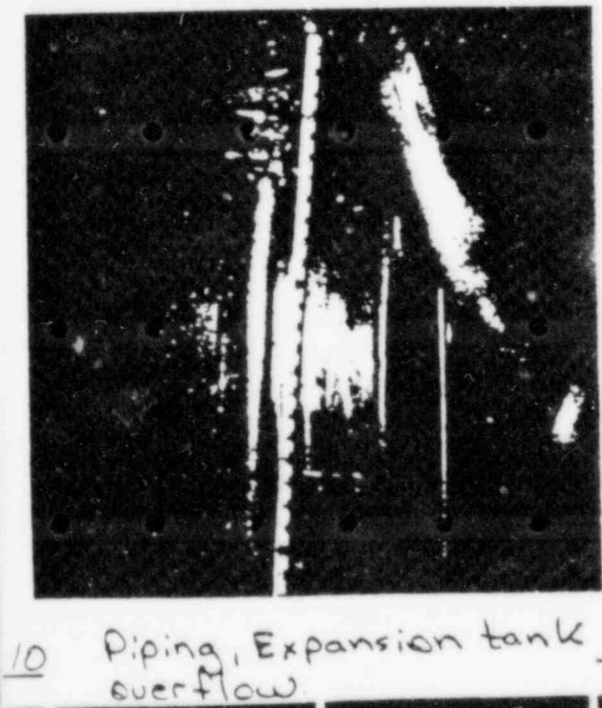
8 Drain Piping, Oil (with tape) and Water



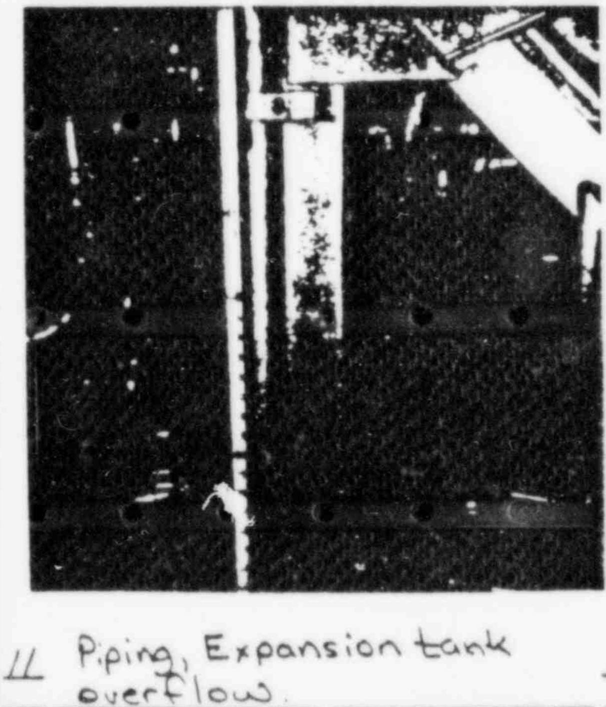
8.1 Cooling System Drain Valve



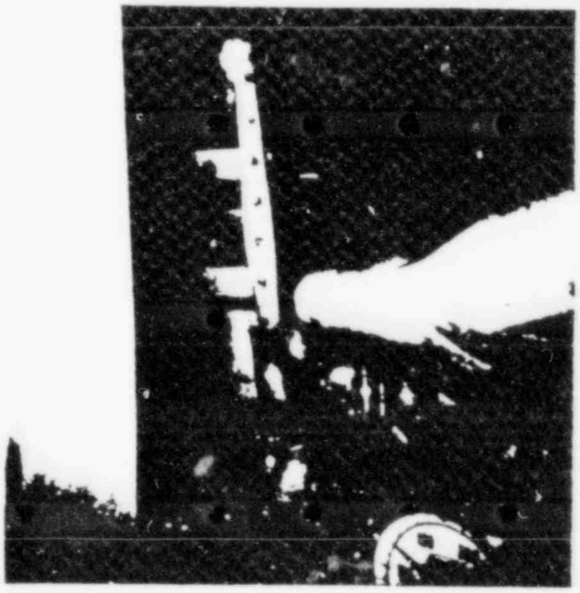
9 Immersion heater Piping



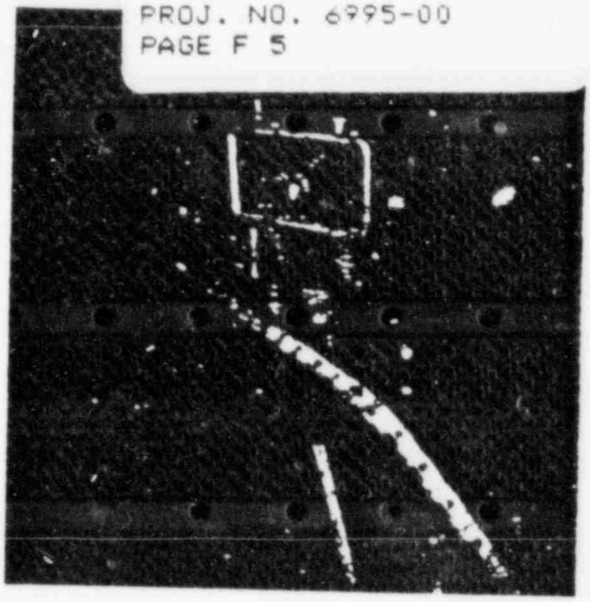
10 Piping, Expansion tank overflow



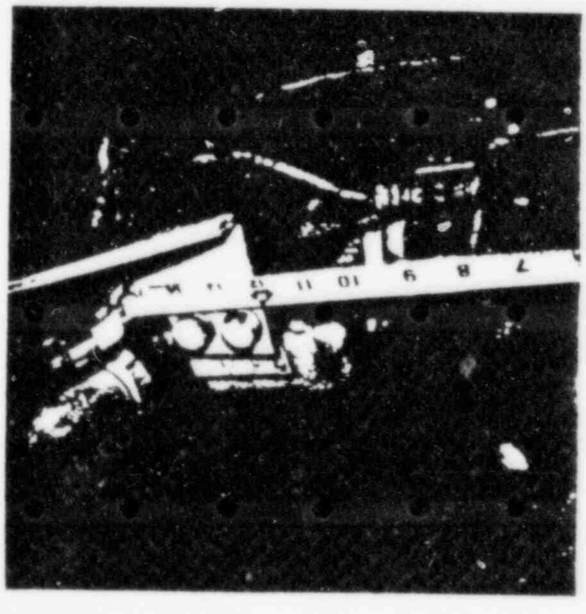
11 Piping, Expansion tank overflow



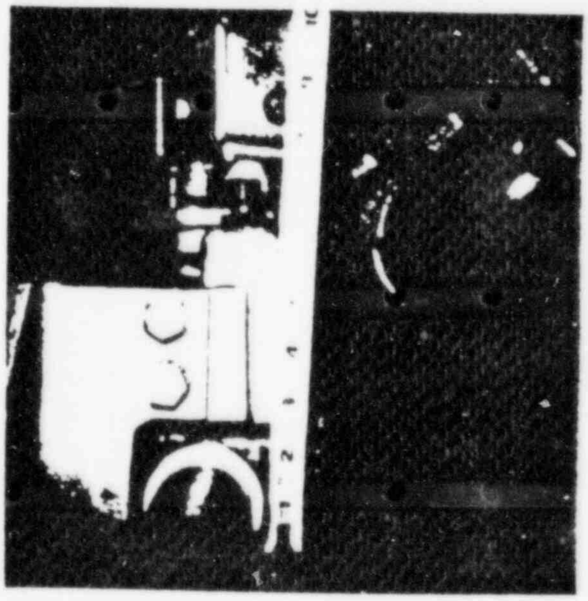
12 Piping, Expansion tank to water pumps (3½" section)



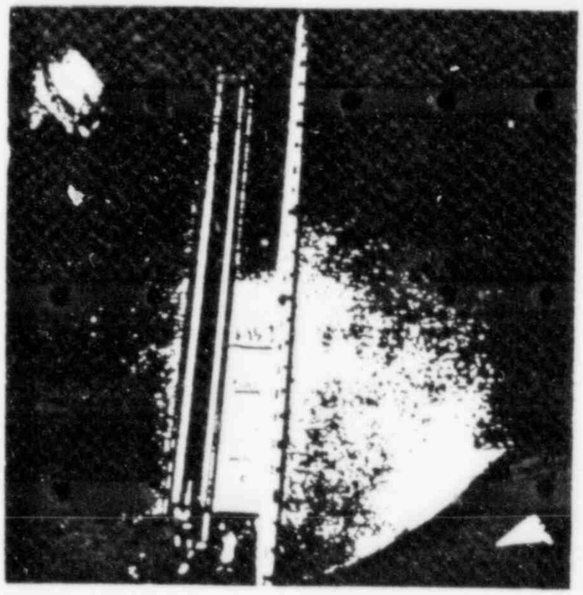
13 Temperature Control Manifold



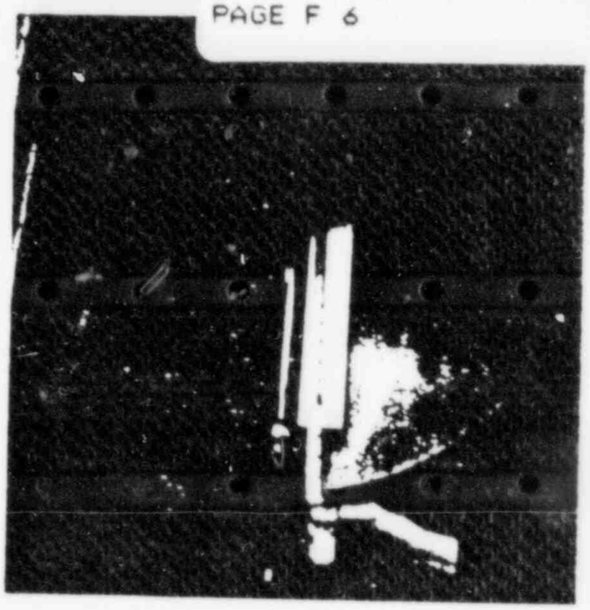
14 Temperature Control Manifold



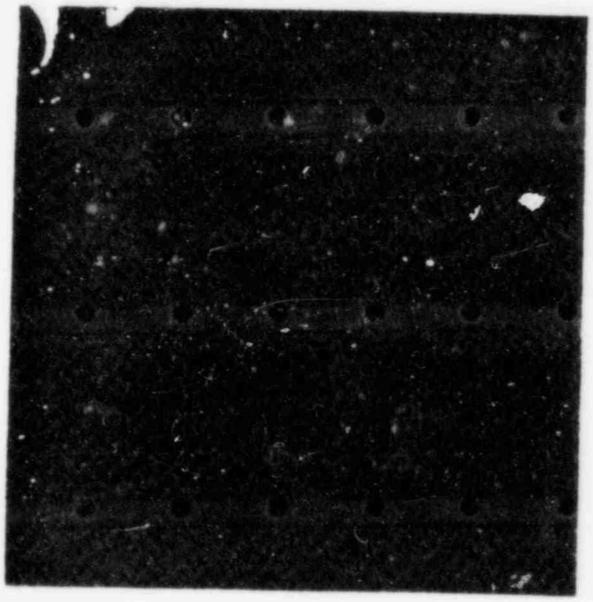
15 Temperature control Manifold.



16 water Expansion tank and Sight Glass



17 Water Expansion tank and Sight Glass



18 Fuel Cutoff Valve

19 Fuel Cutoff Valve

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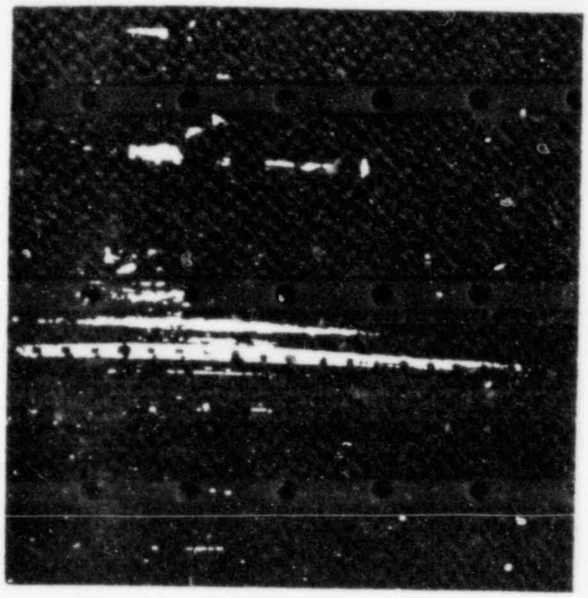
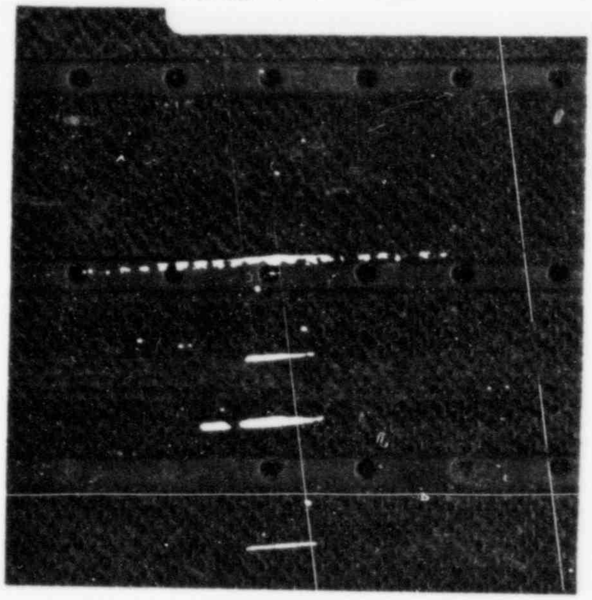
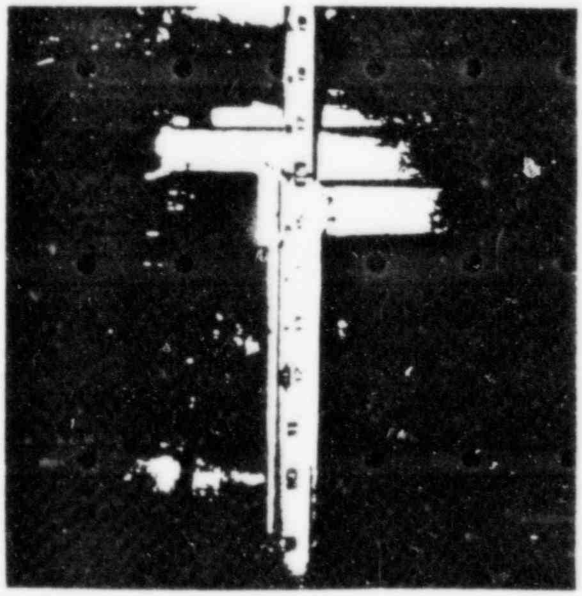


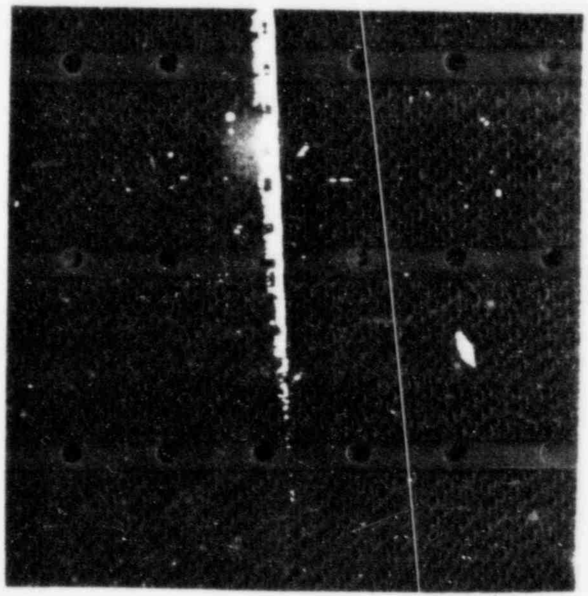
20 Fuel oil Transfer system



21 Fuel oil Transfer system



22 Fuel Oil Transfer system



23 Fuel Transfer Strainer (1 of 2 strainers TYP.)

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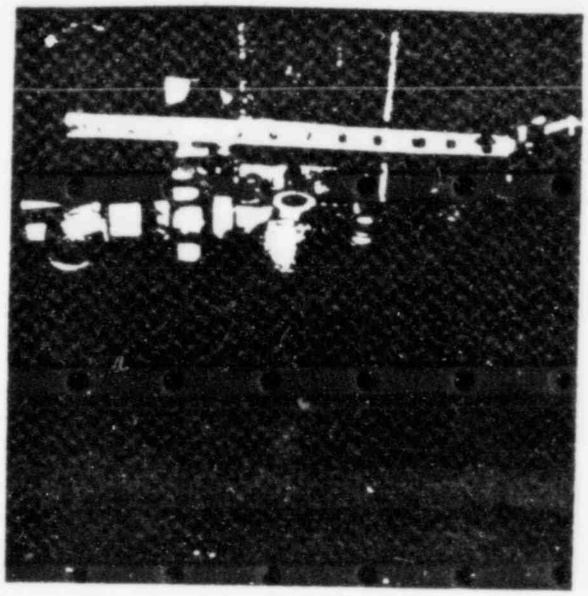
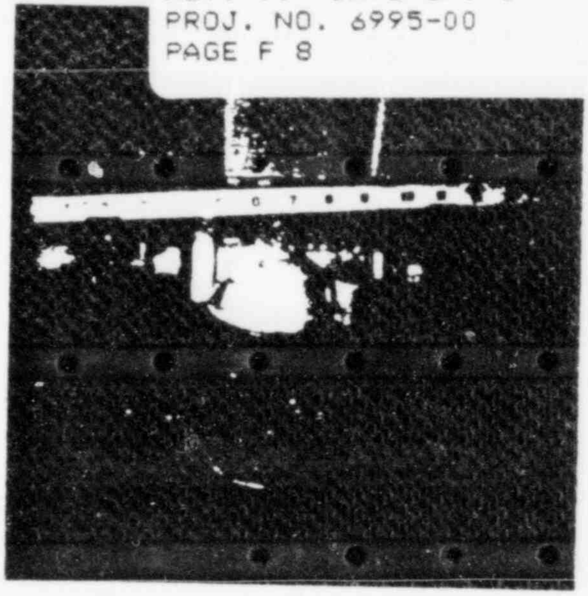
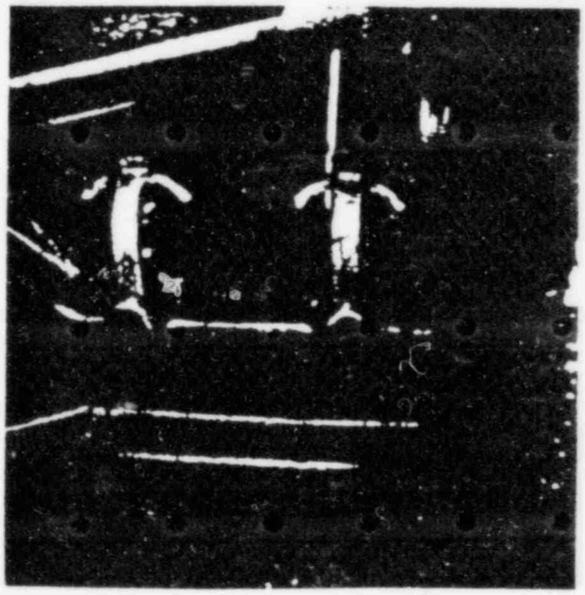


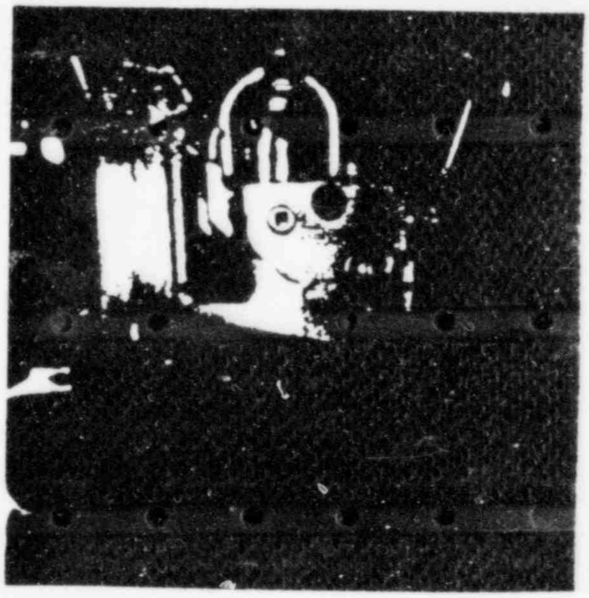
24 Fuel oil Suction strainer



25 Fuel oil Suction strainer

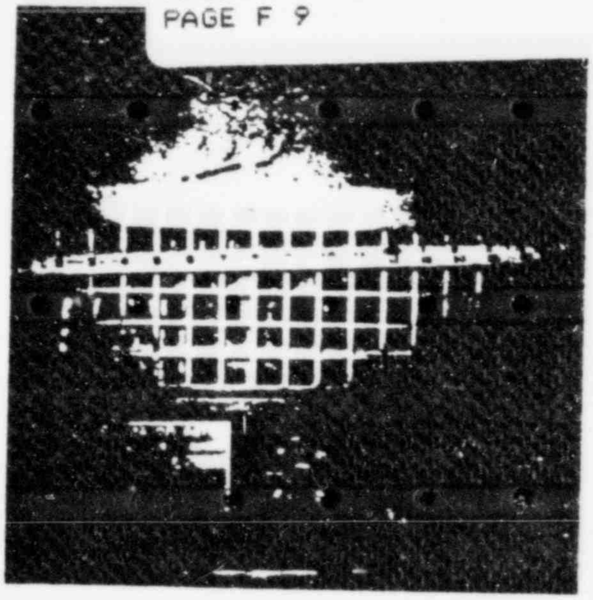


26 Fuel Oil Manifold

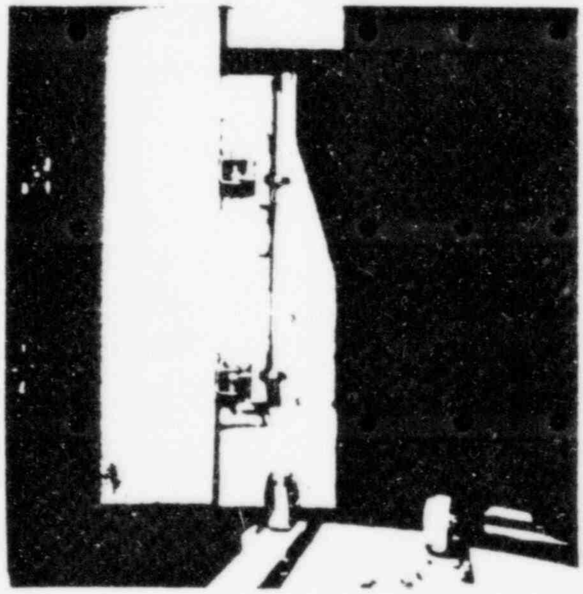


27 Fuel oil Manifold.

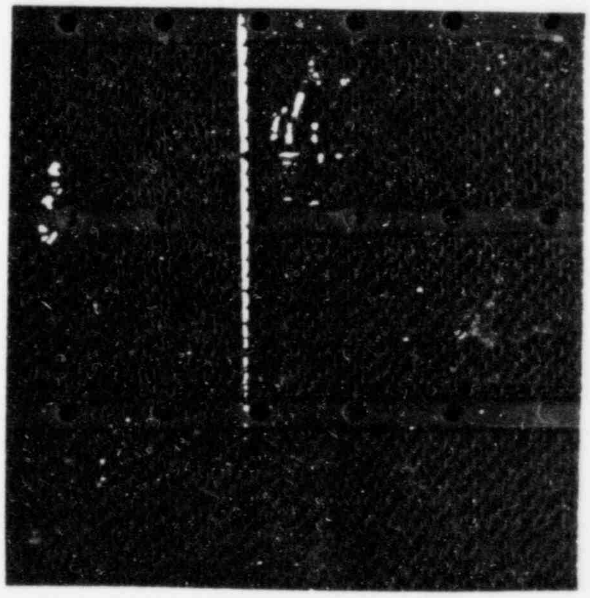
28 Piping, Fuel Oil



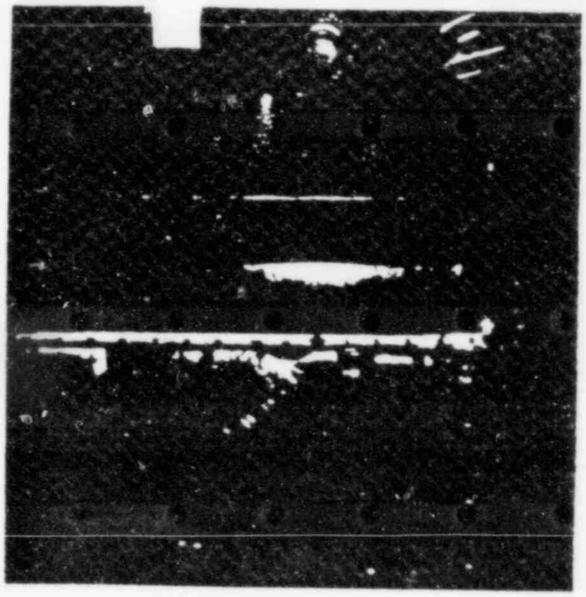
29 AIR Intake Filter



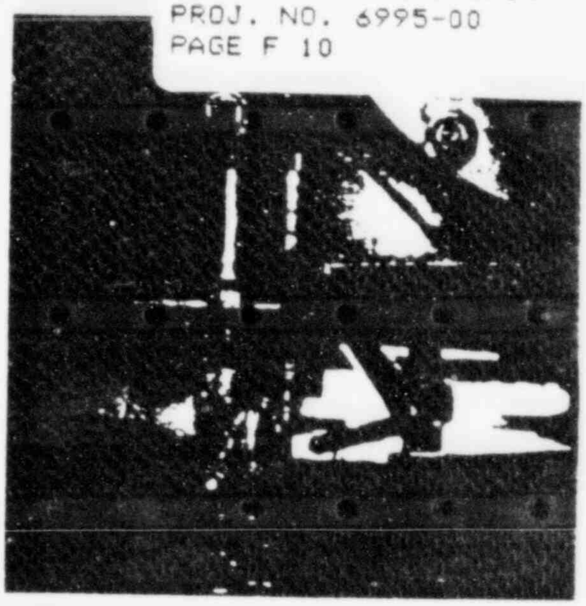
30 AIR Intake filter



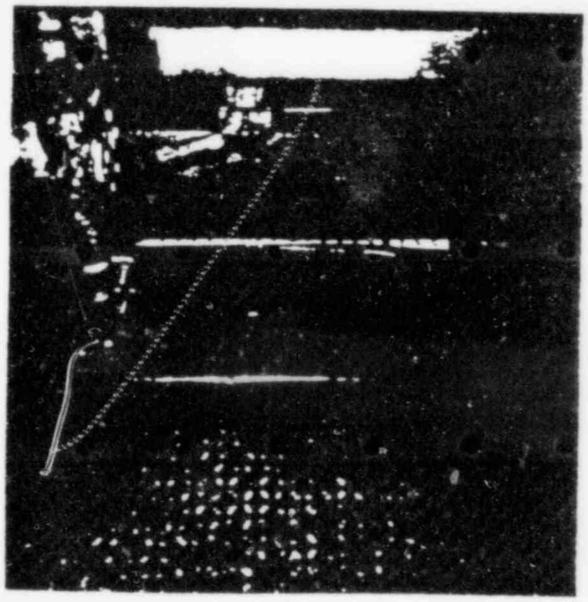
31 Turbocharger Filter (oil)



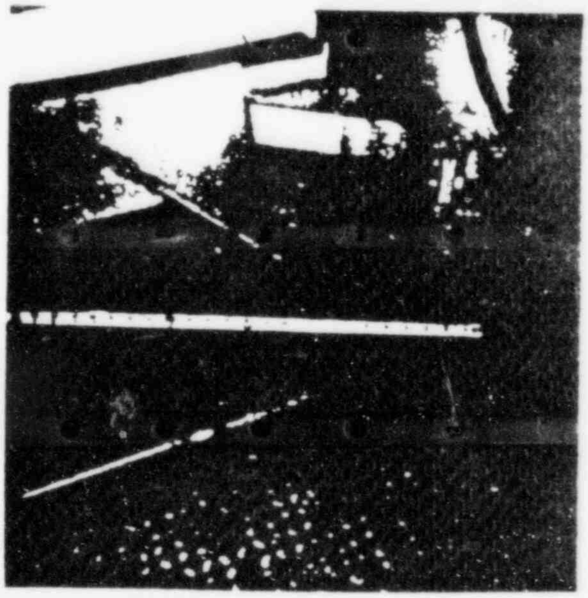
32 Lube Oil "Y" strainer and Piping.



33 Piping, "Y" strainer to soakback filter



34 Lube oil filter inlet From "Y" strainer



35 Piping, soakback filter to "Y" strainer

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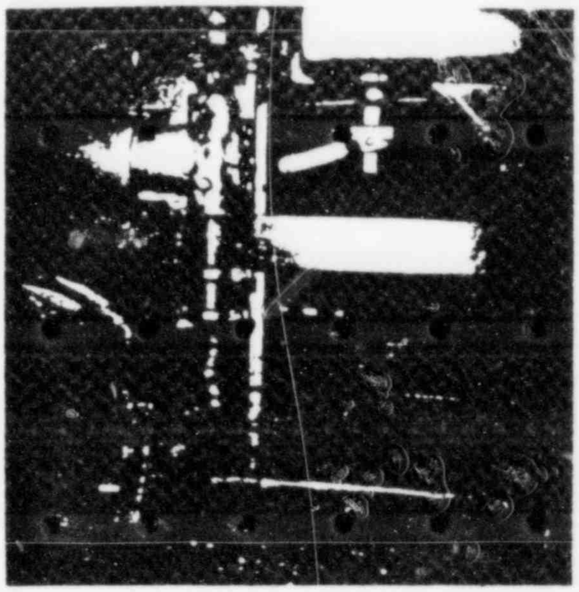
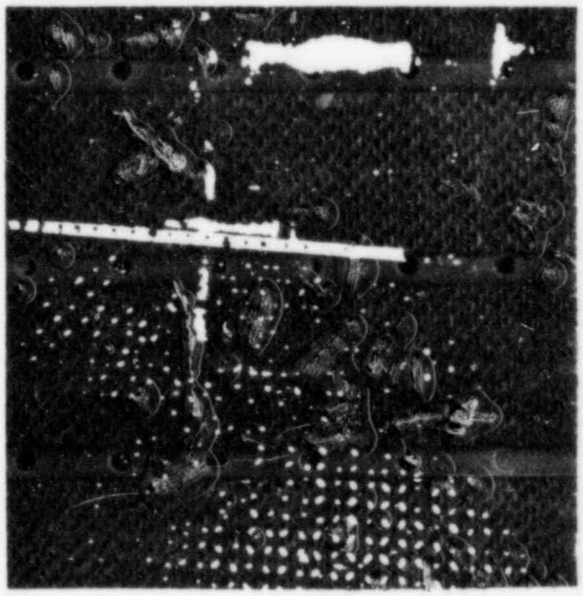


36 Piping, Lube oil "Y" strainer to soakback filter

37 Lube oil Circ. Pump to "Y" strainer

38 Piping, Lube oil Circ pump to "Y" strainer



39 Piping, Lube oil, Engine outlet to Circulating Pump

11

SARGENT & LUNDY ENGINEERS

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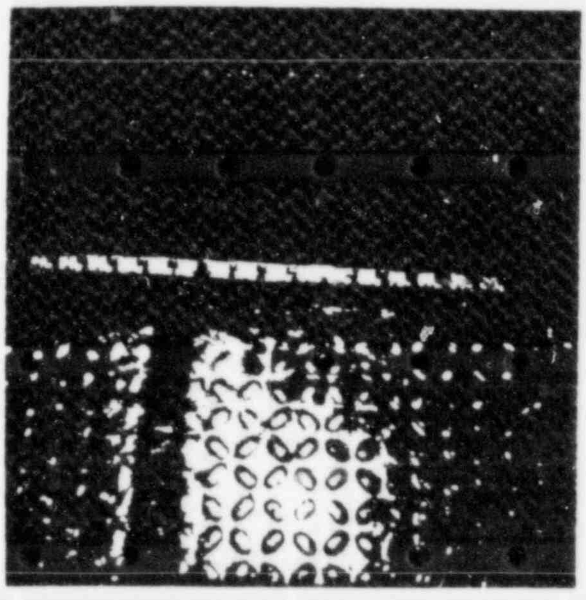
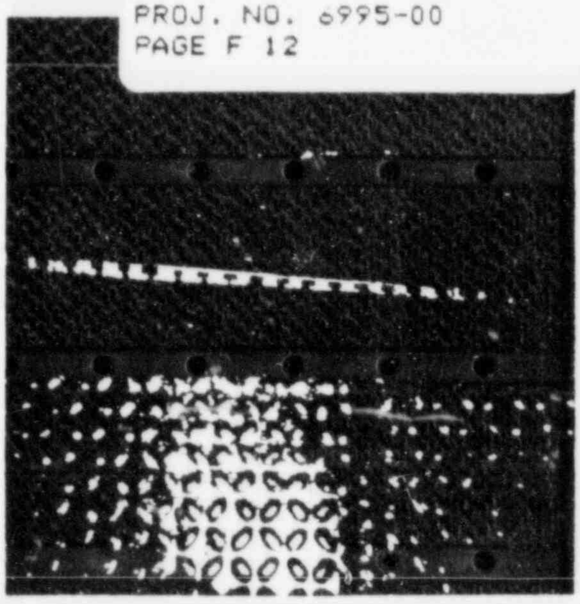
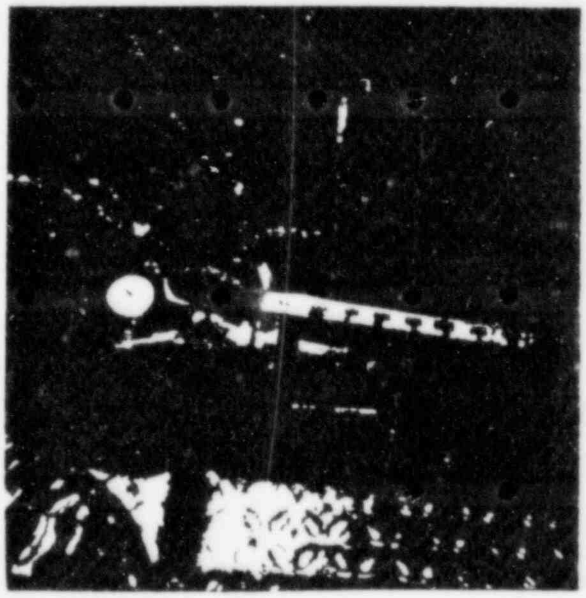


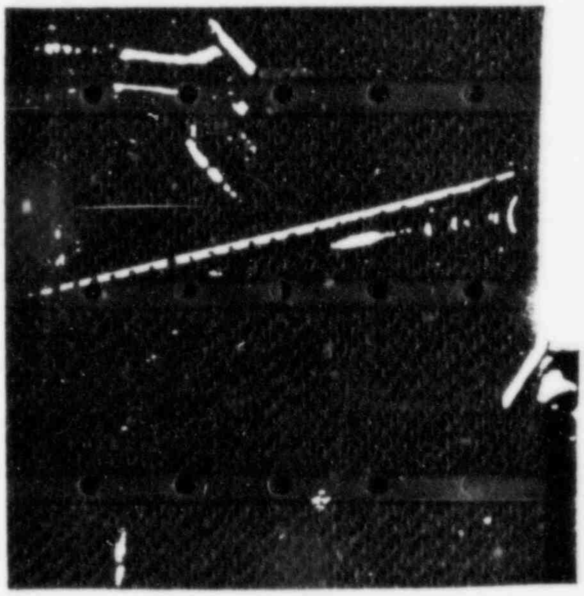
40 Lube Oil piping, Engine —
to circulating pump



41 Piping, Lube oil, engine —
outlet to circulating pump



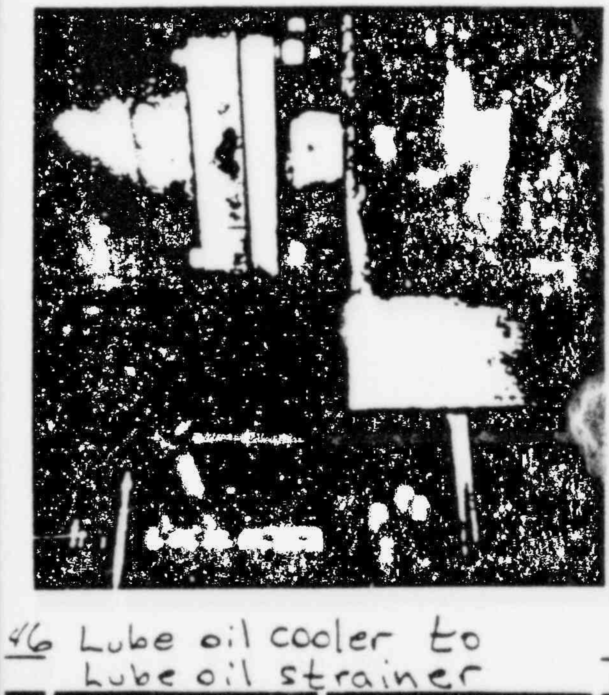
42 Lube oil Piping; engine —
to circulating pump



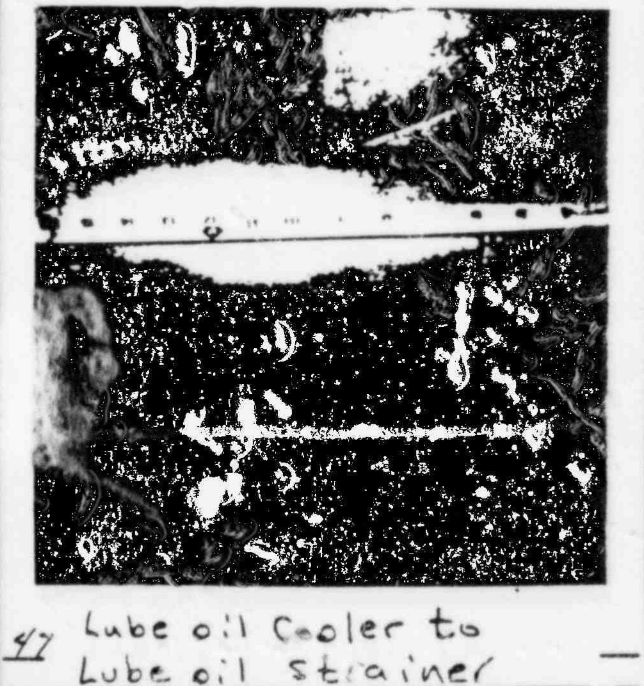
43 Piping, Lube oil filter —
to Lube oil scavenging pump

44 Lube oil filter to scavenging pump

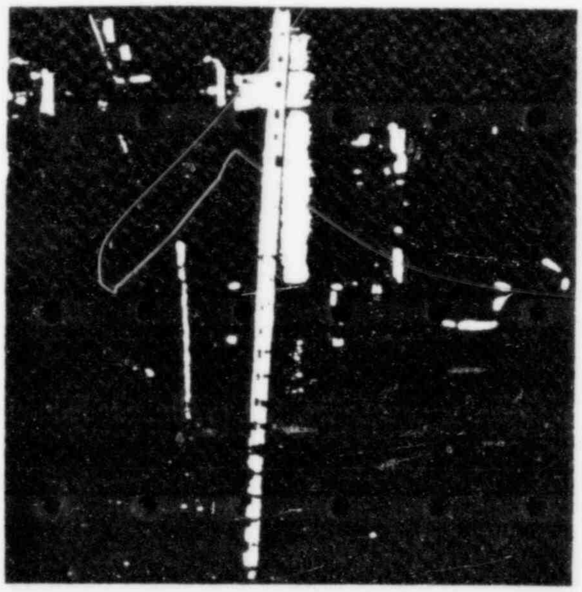
45 Piping, Lube oil cooler to Lube oil Filter



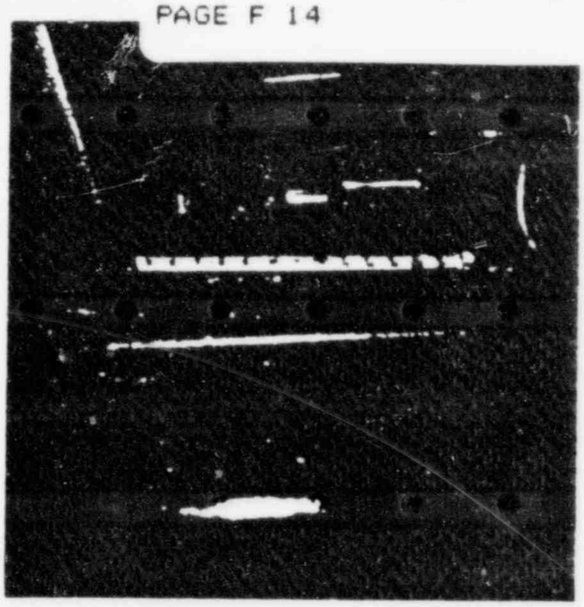
46 Lube oil cooler to Lube oil strainer



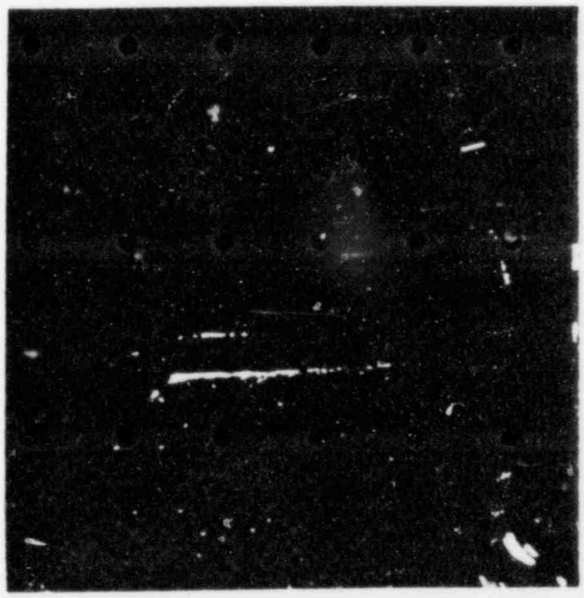
47 Lube oil Cooler to Lube oil strainer



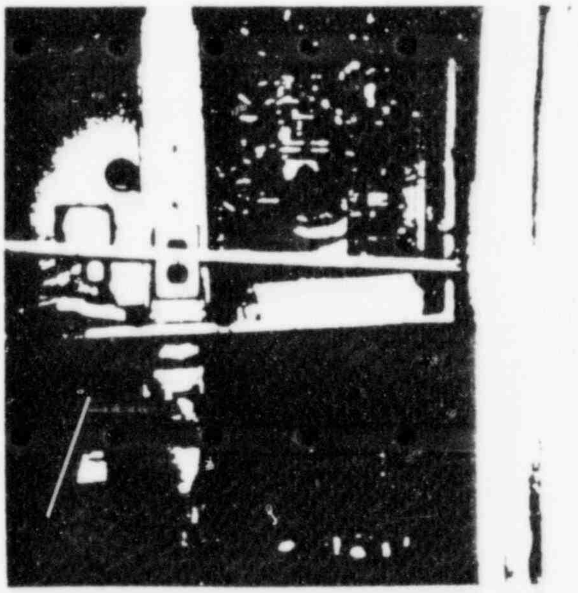
48 Lube oil Cooler to Lube oil Strainer —



49 Flex Connection, Lube oil cooler to strainer —



50 Lube oil Filler and Valve. —



51 Engine control Panel —

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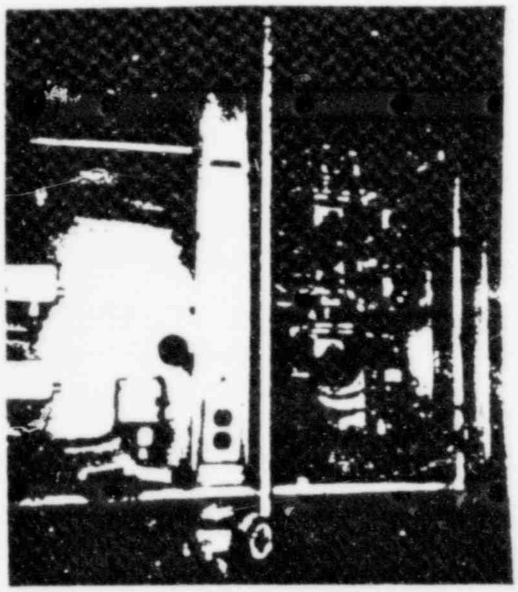
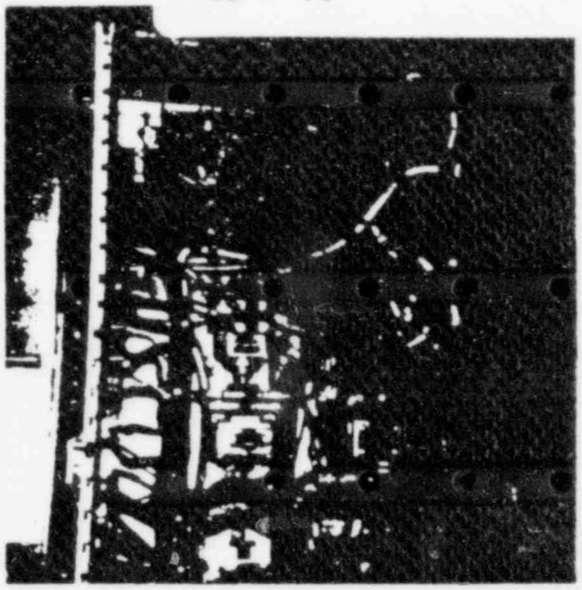
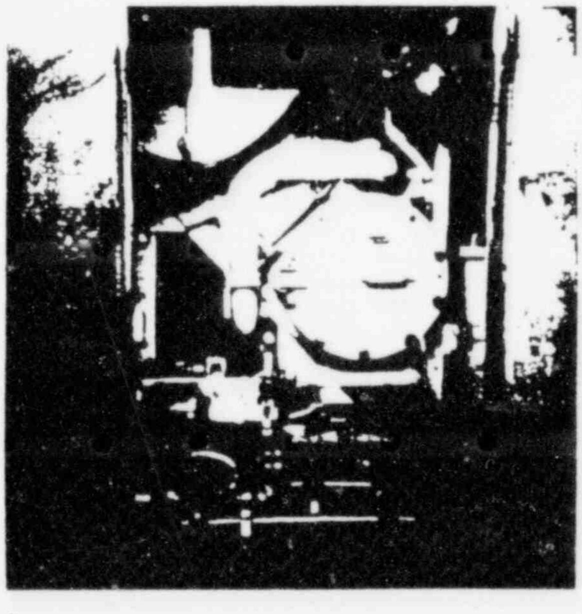


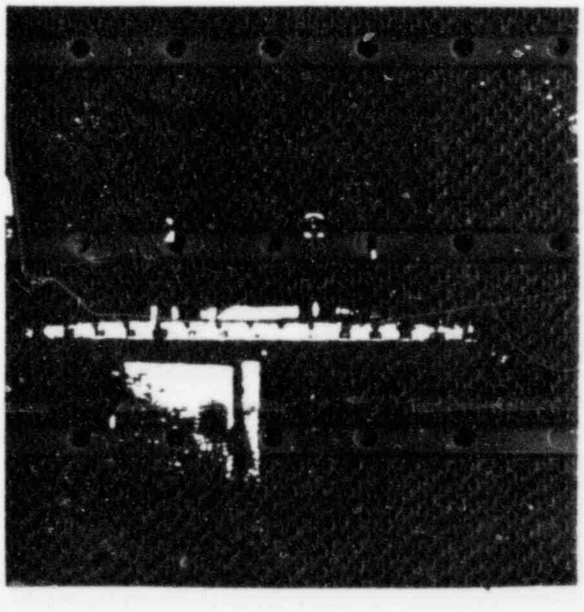
52 Engine Control Panel



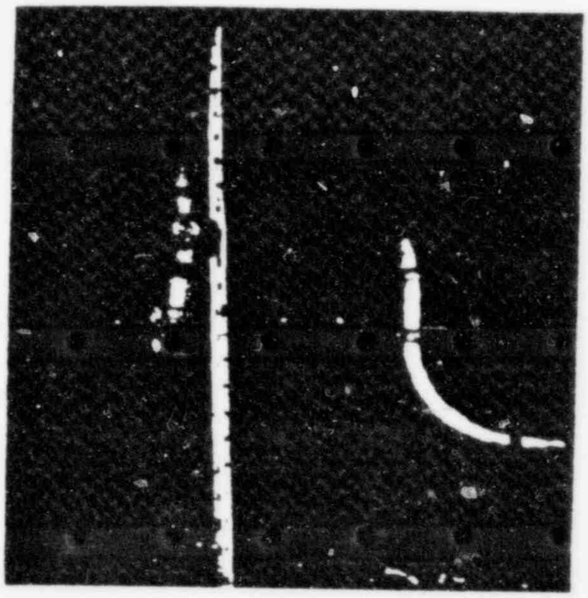
53 Engine Control Panel



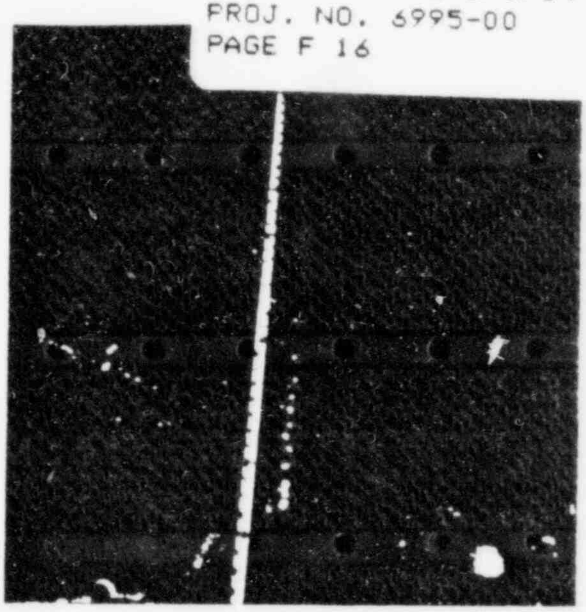
54 Accessory Rack, Lube oil cooler, filter and Pump



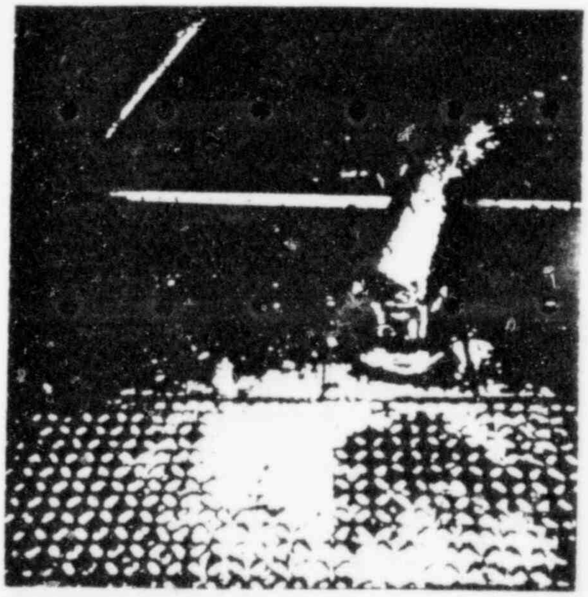
55 Engine outlet Piping and Adaptor



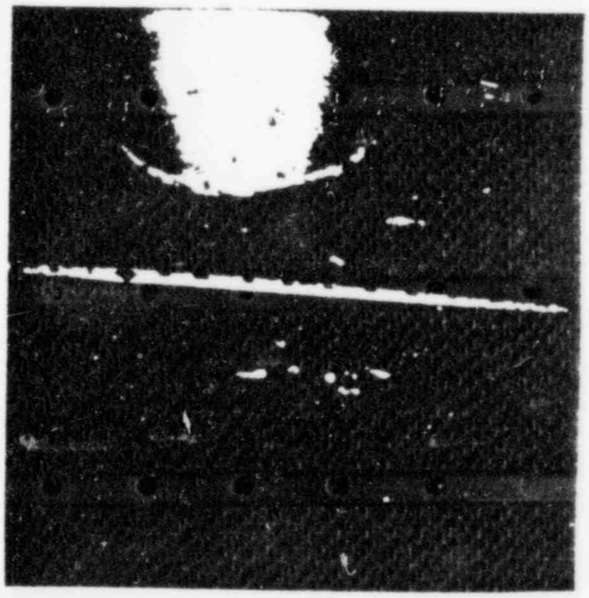
56 Engine Outlet Piping



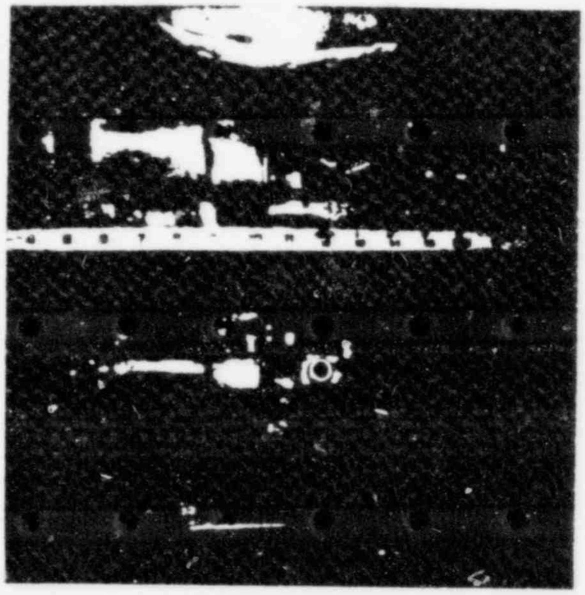
57 Cooling Vent piping, Expansion tank to radiator



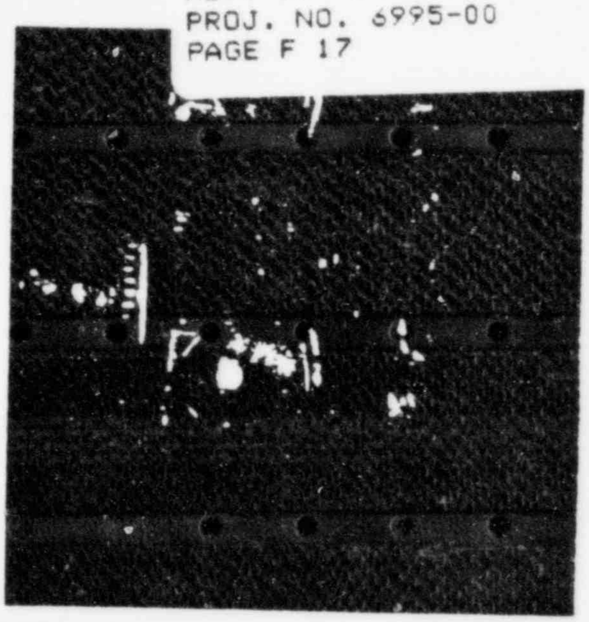
58 Immersion Heater



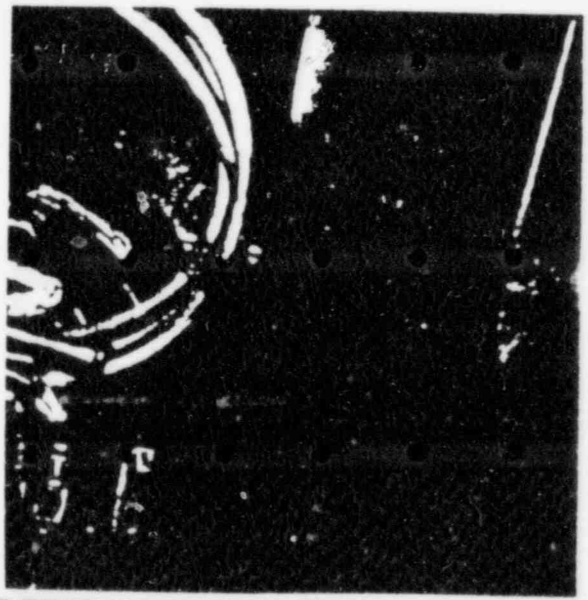
59 Electric Start Motors



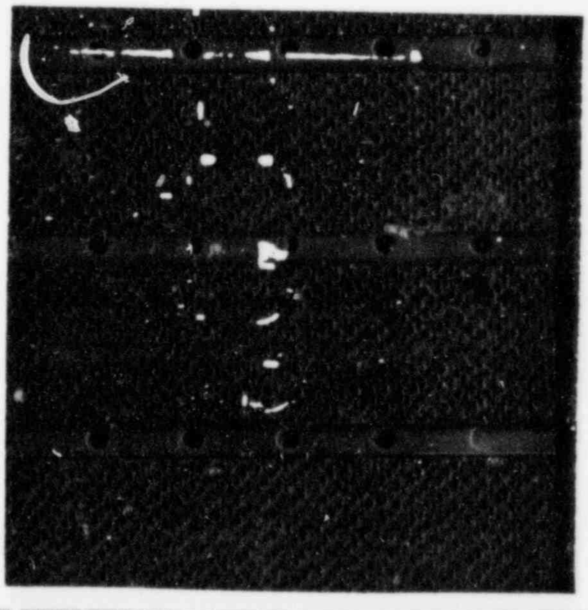
60 Electric Start Motors



61 Electric Start Motor Brackets

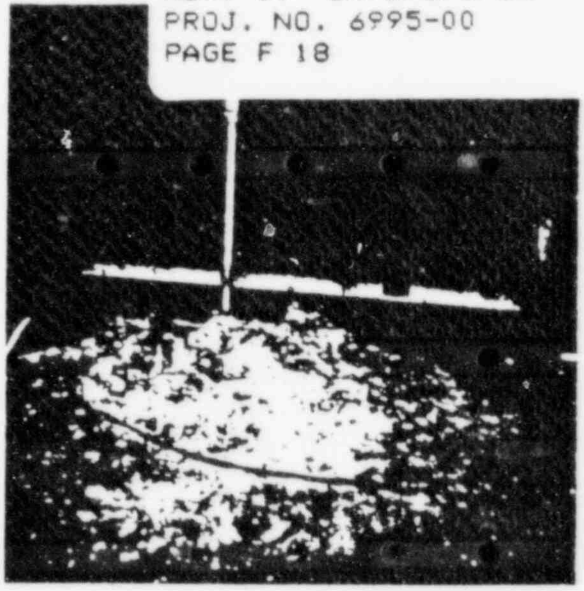


Static Exciter

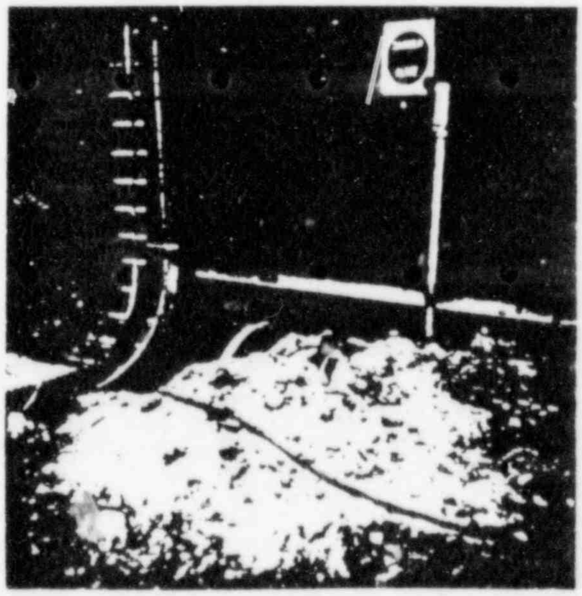


Static Exciter

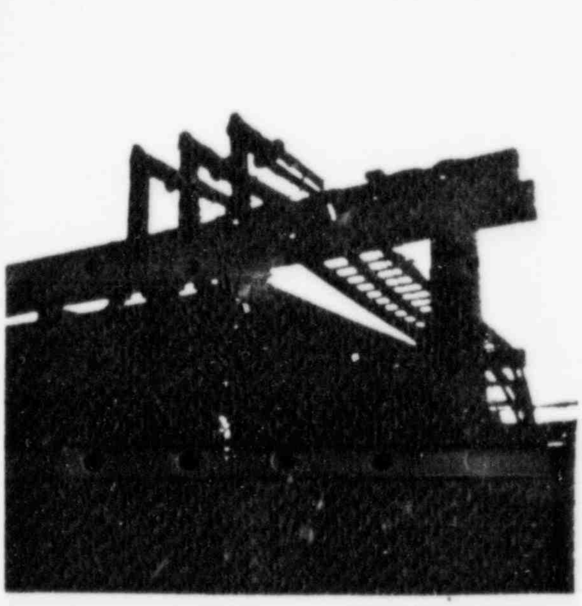
Common Switchgear Base



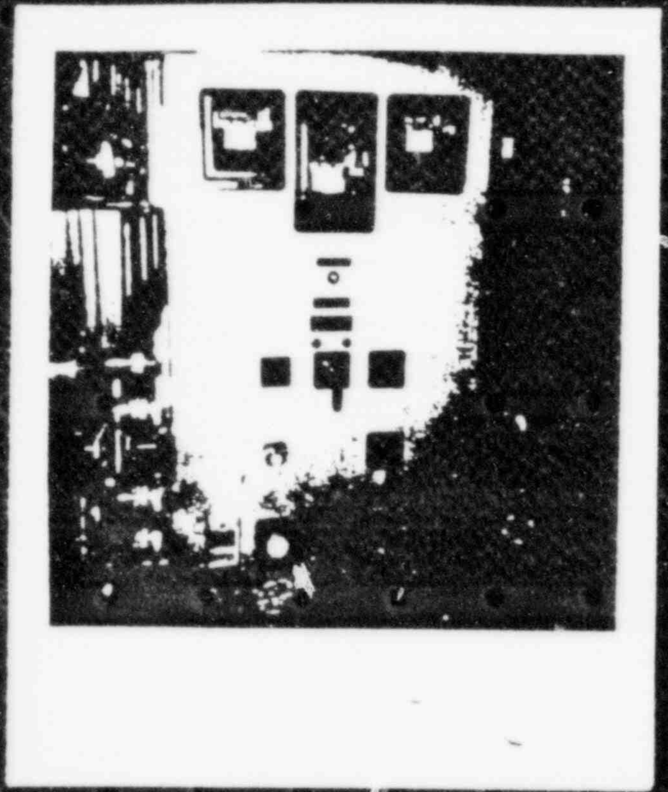
Common Switchgear Base

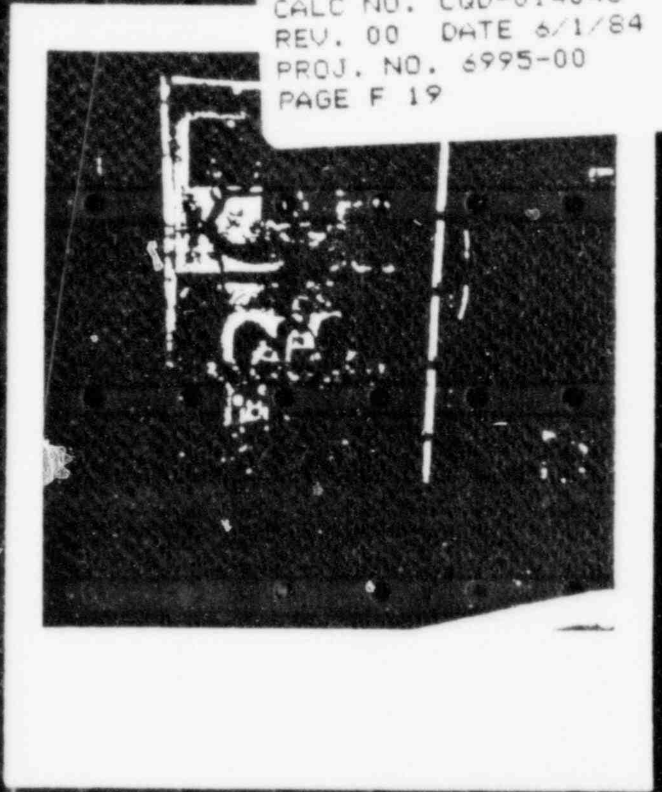


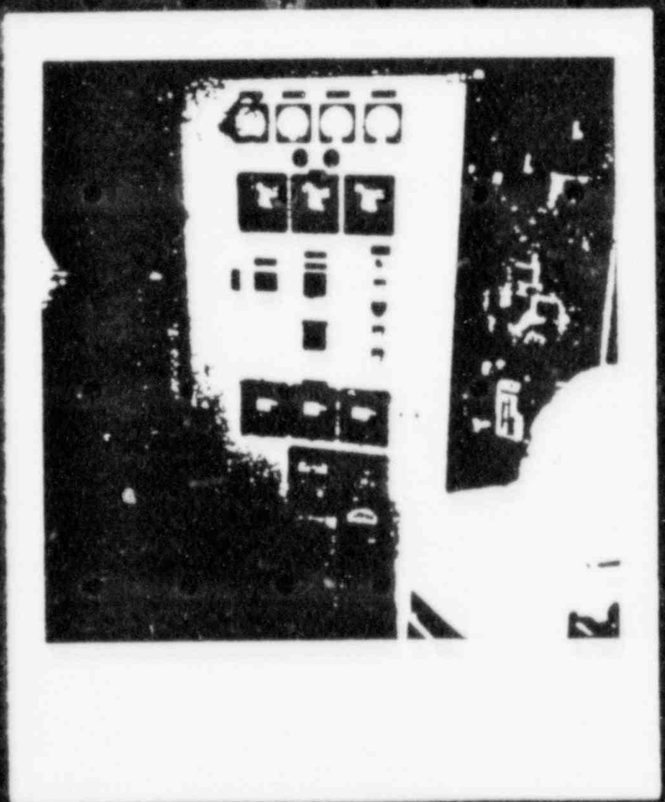
Common Switchgear substructure

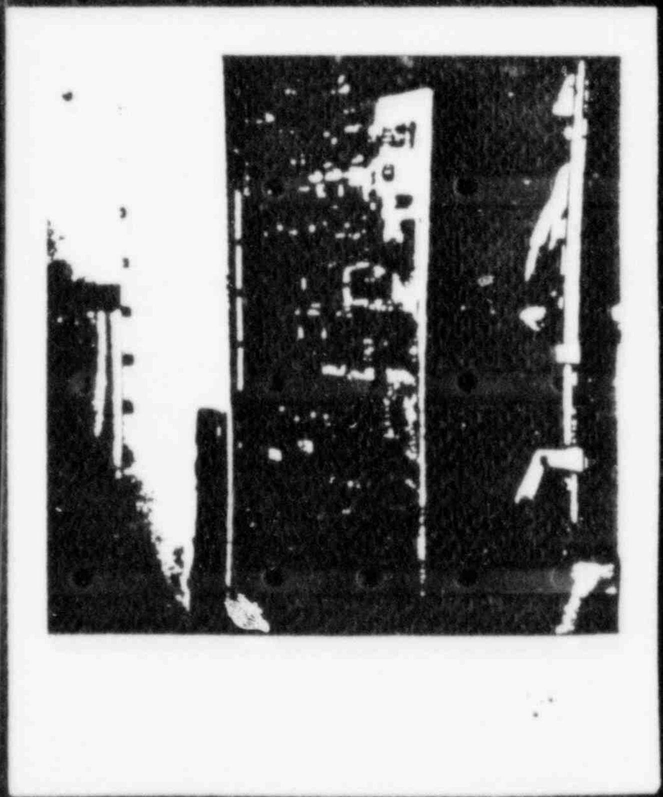


Common Switchgear External View









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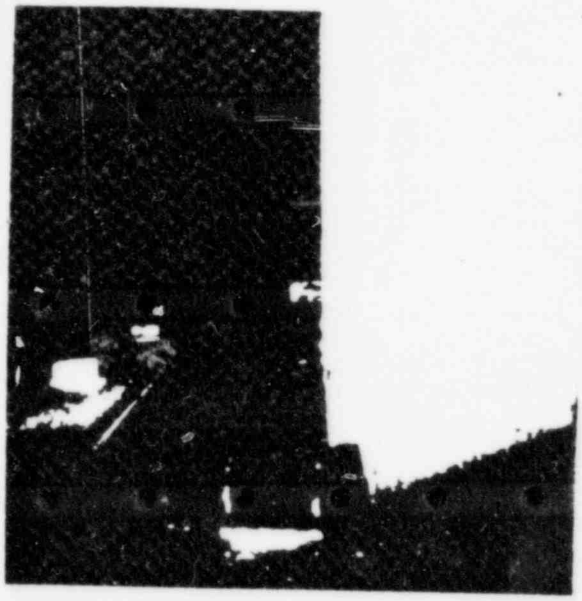
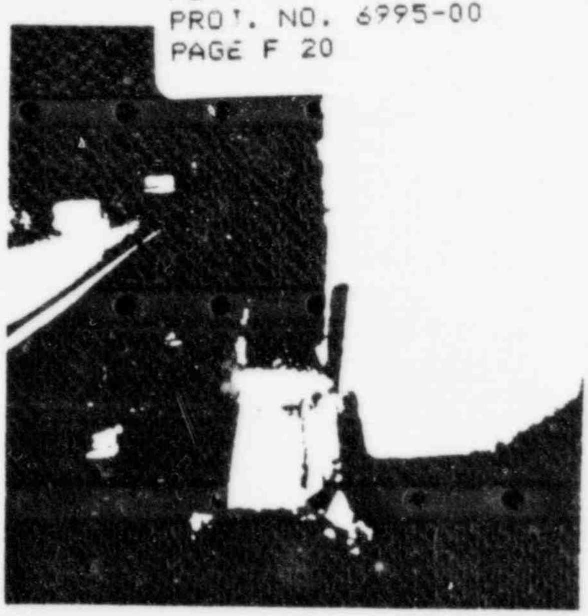
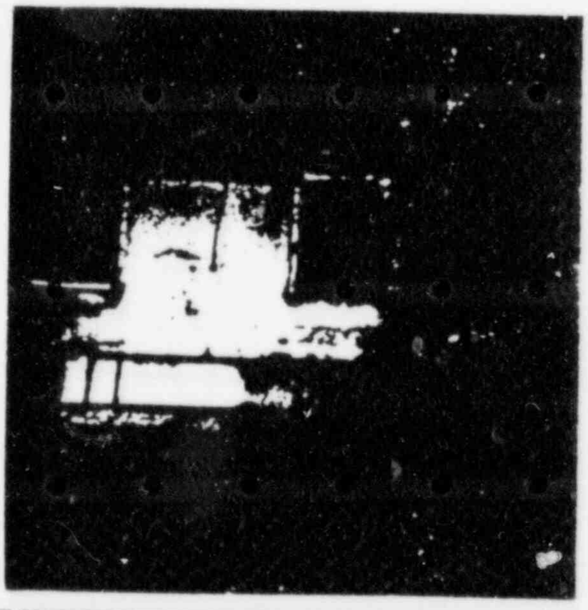


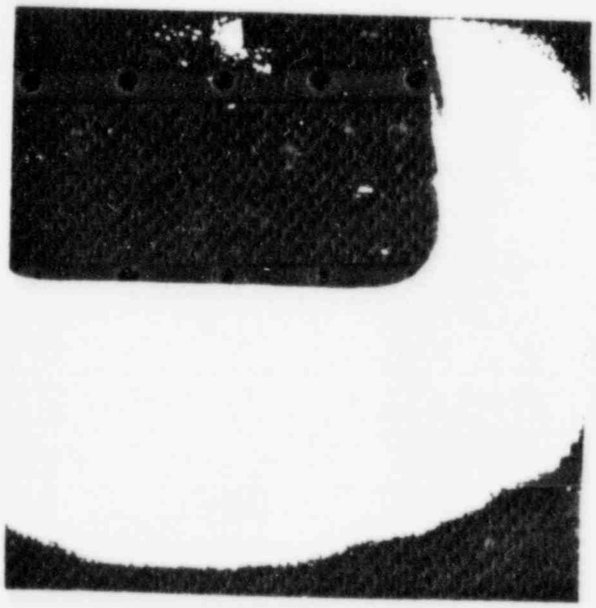
Generator Flex Duct



Generator Cooling System flex duct



Generator Cooling System Bearing support.



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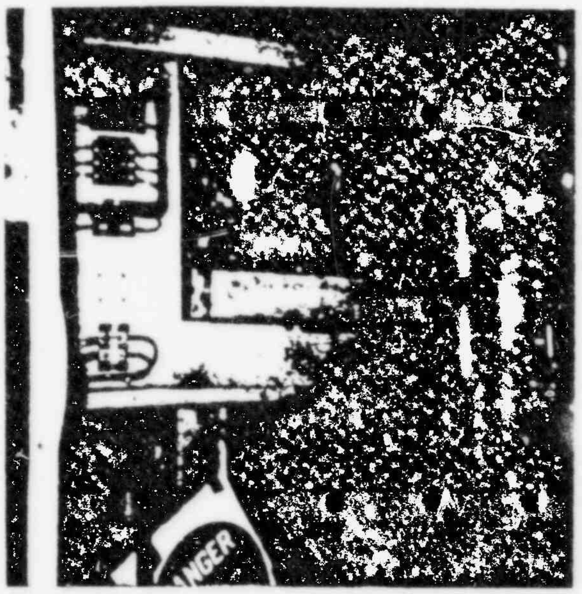
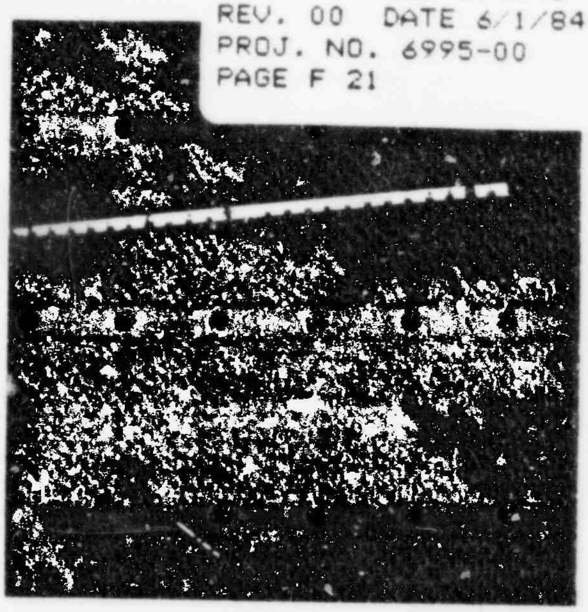
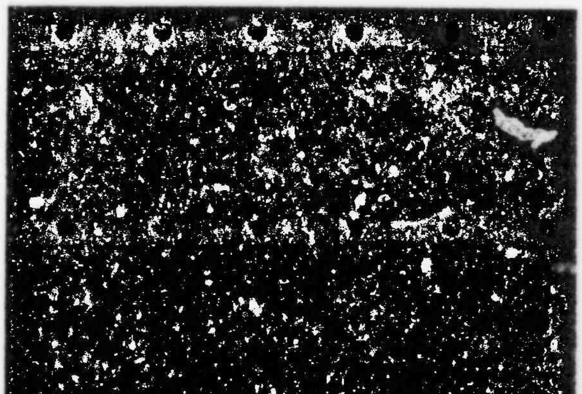
ENGINEERS



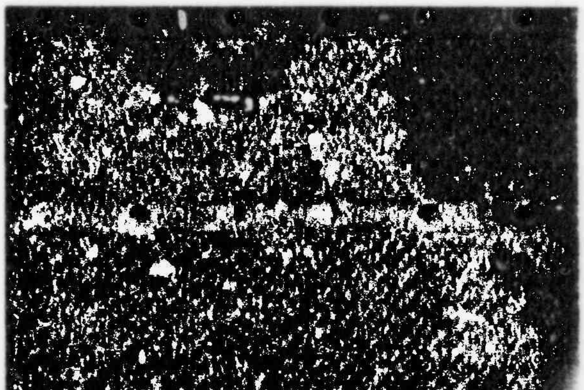
Common Switchgear



Radiator Fan drive shaft framework



Radiator Piping



Radiator Nozzles
TYP. of 4

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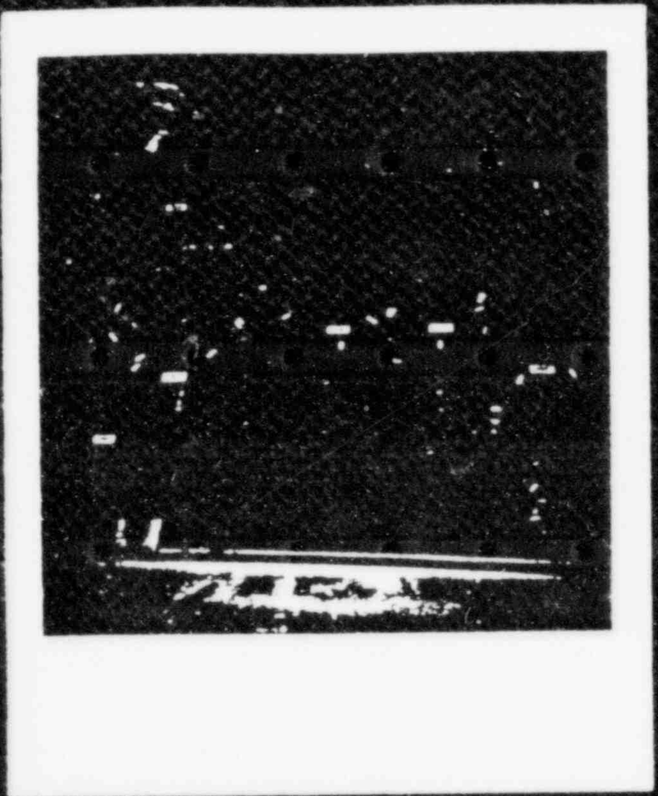

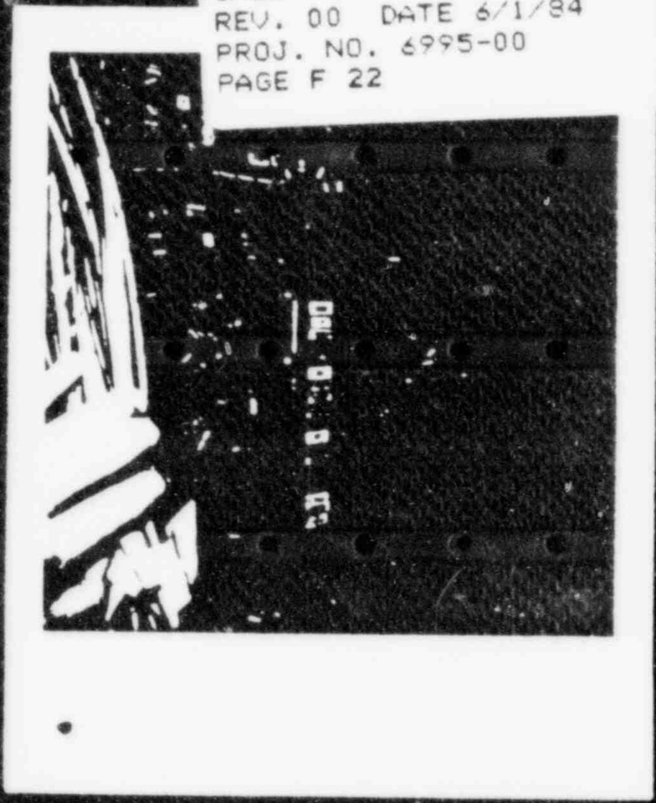

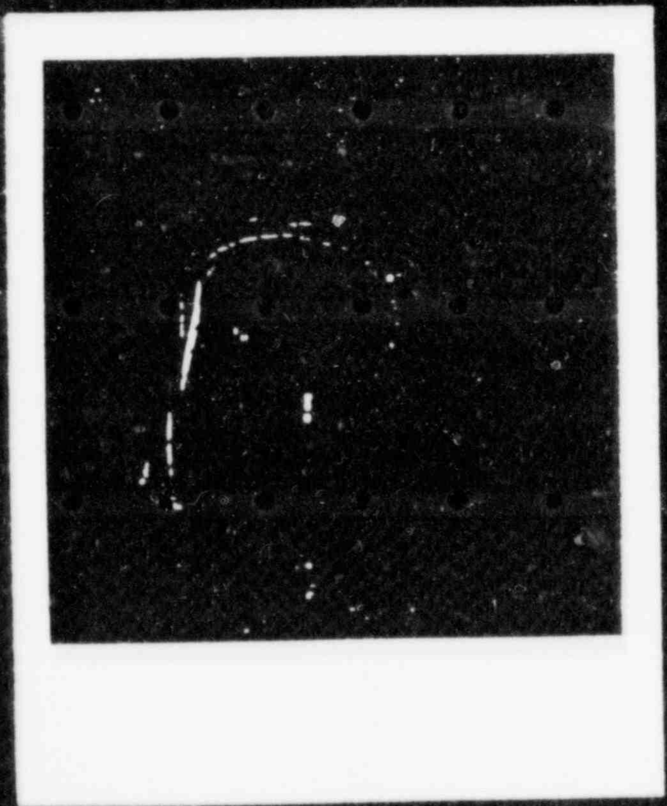

SARGENT & LUNDY
ENGINEERS

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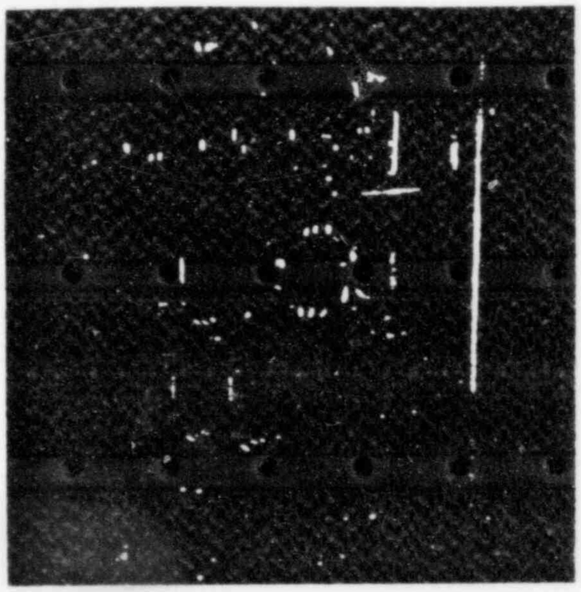
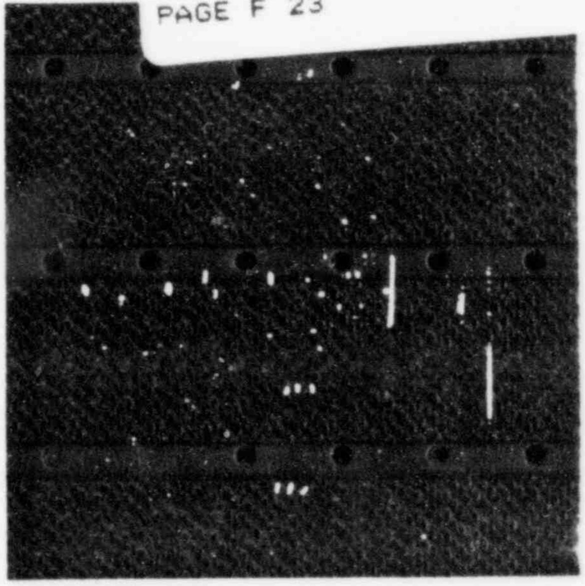
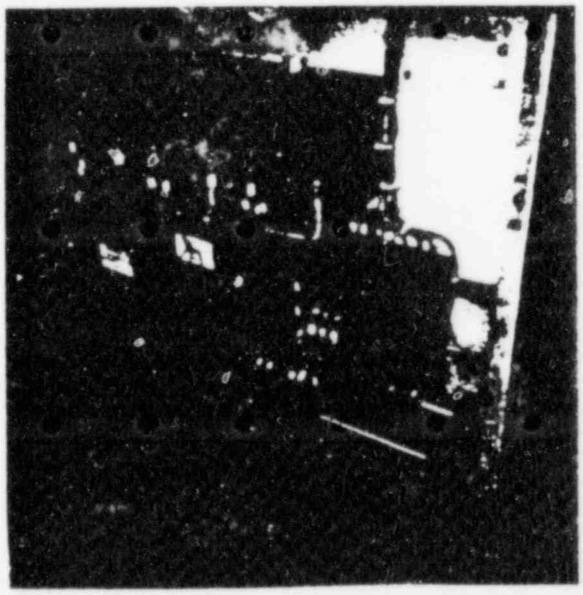
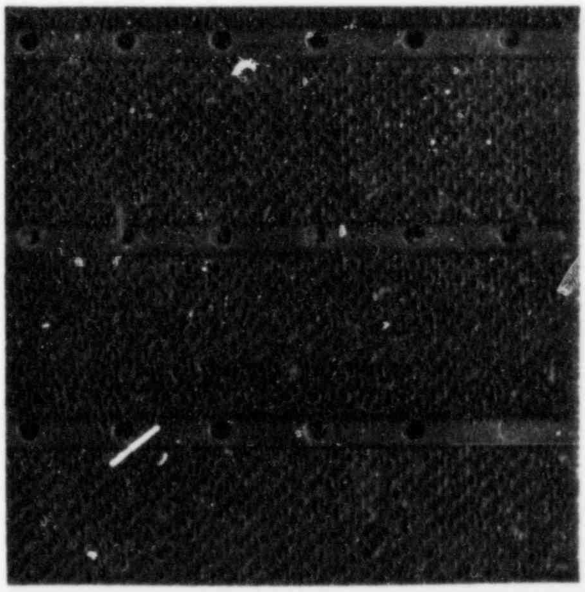




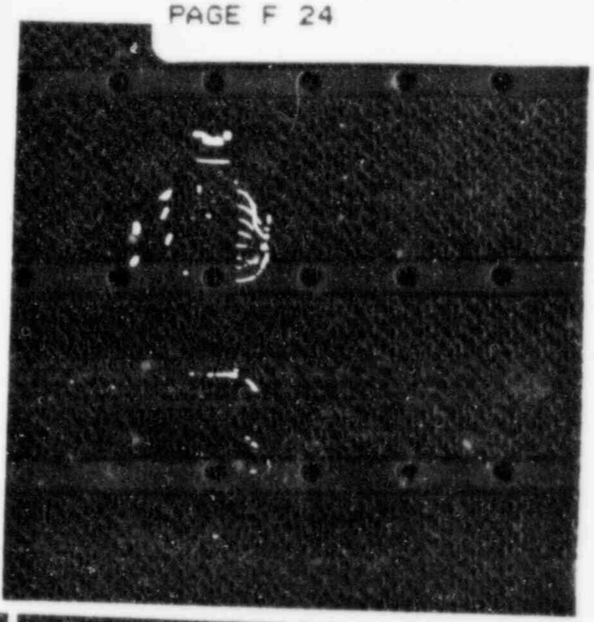


SARGENT & LUNDY
ENGINEERS
CHICAGO

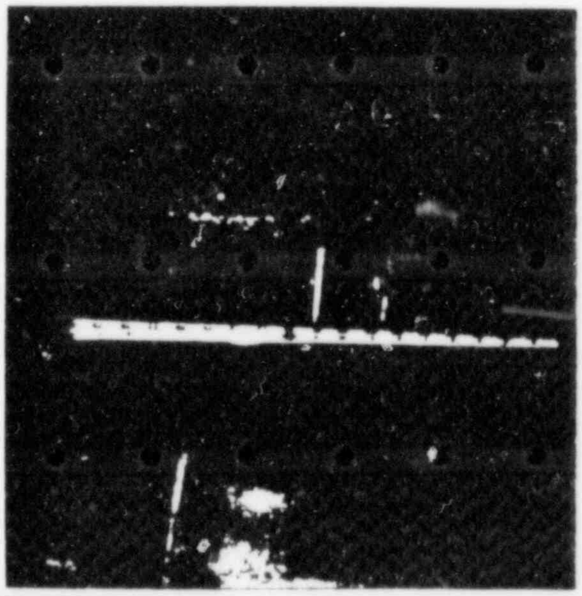


Radiator fan shaft and bearings

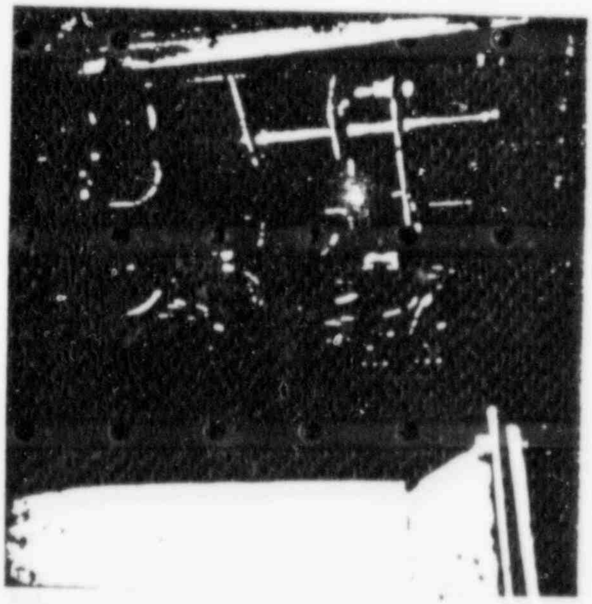
Static Exciter



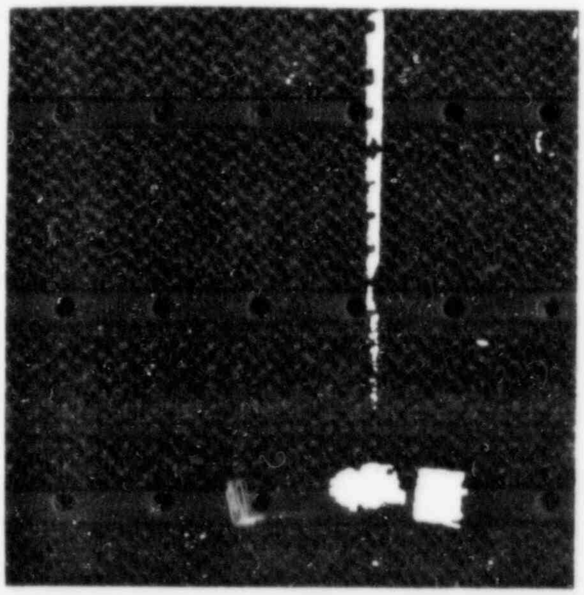
Static Exciter



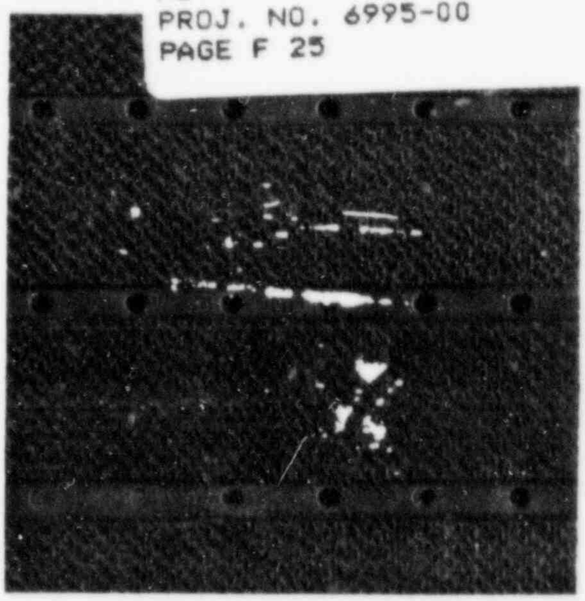
Static Exciter



Static Exciter

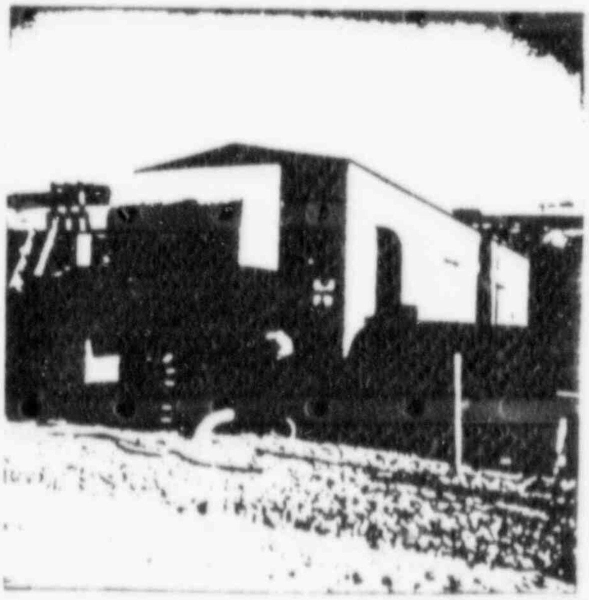


Heat Exchanger

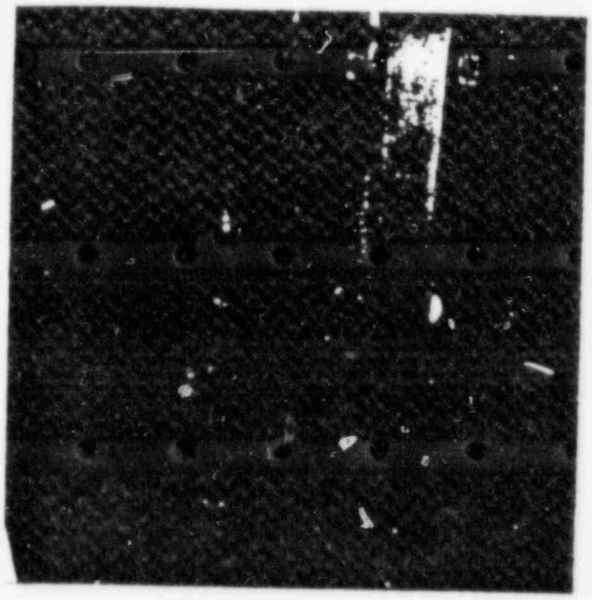


Lube oil Filler Line

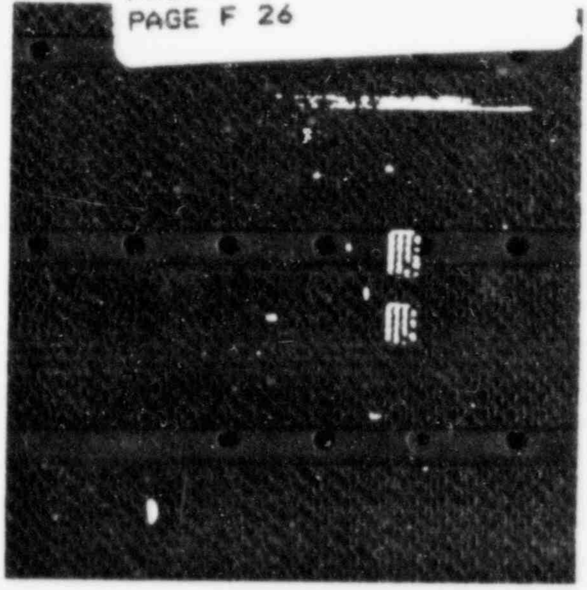
Fuel Oil Check Value



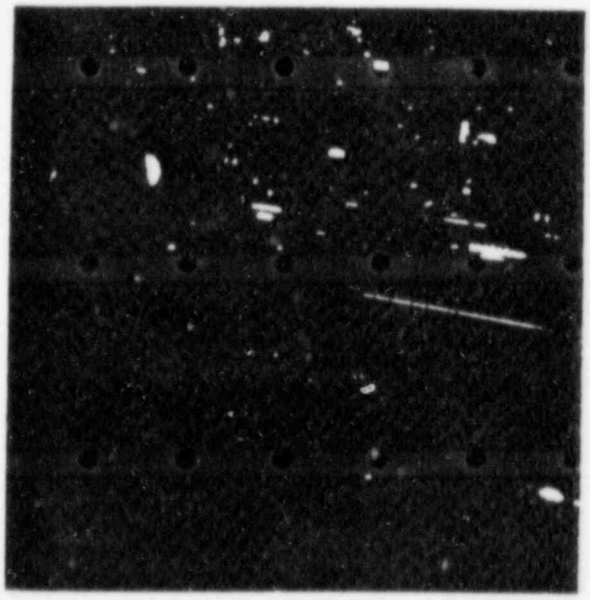
Engine Enclosure



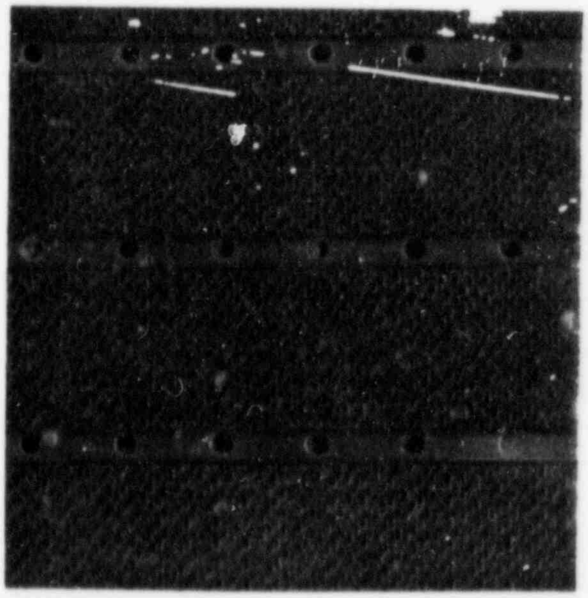
GENERATOR & GOVERNOR CONTROL PANEL
(MIDDLE TOP LEFT)



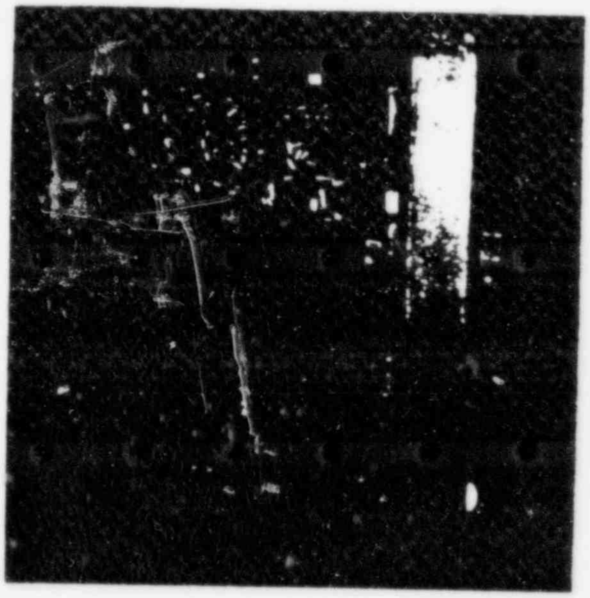
GENERATOR & GOVERNOR CONTROL PANEL
(TOP RIGHT)



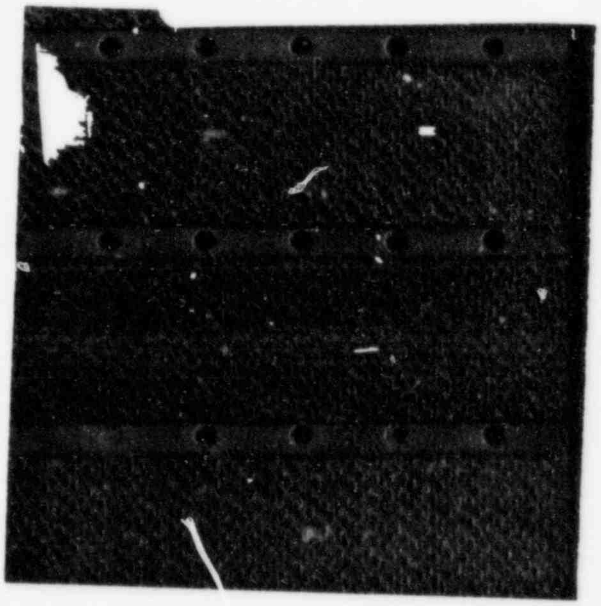
GENERATOR & GOVERNOR CONTROL PANEL
(MIDDLE RIGHT)



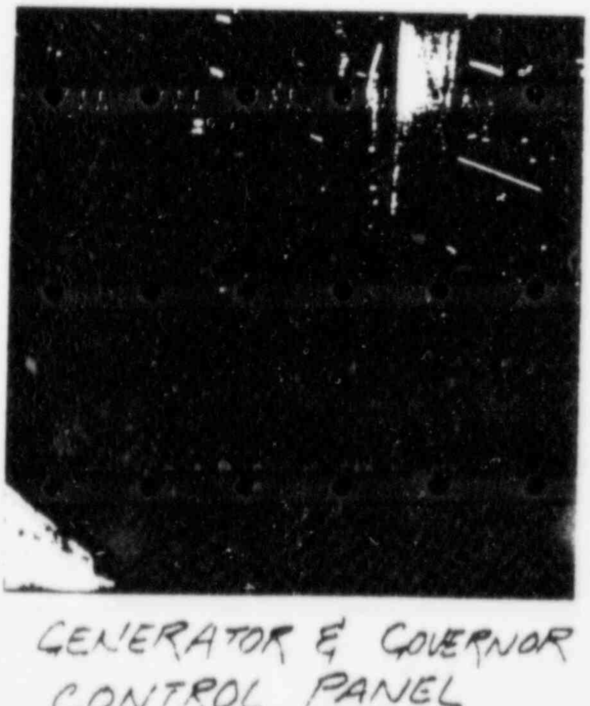
GENERATOR & GOVERNOR CONTROL PANEL
(BOTTOM RIGHT)



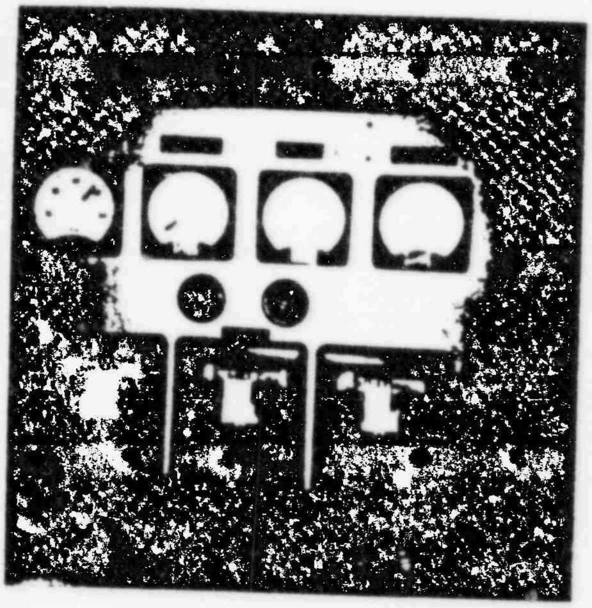
GEN. & GOVERNOR
CONTROL PANEL
(TOP LEFT)



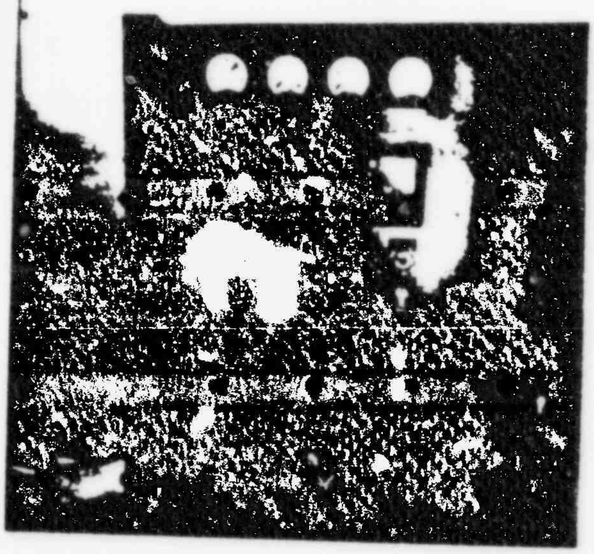
DOOR PANEL



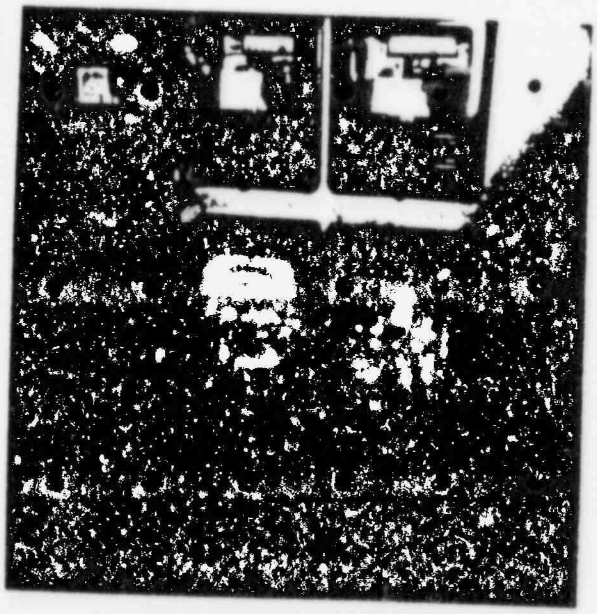
GENERATOR & GOVERNOR
CONTROL PANEL
(BOTTOM LEFT)



[illegible] 1A DOOR PANEL
(CLOSE UP TOP)



CUBICLE 1A DOOR PANEL



CUBICLE 1A DOOR PANEL
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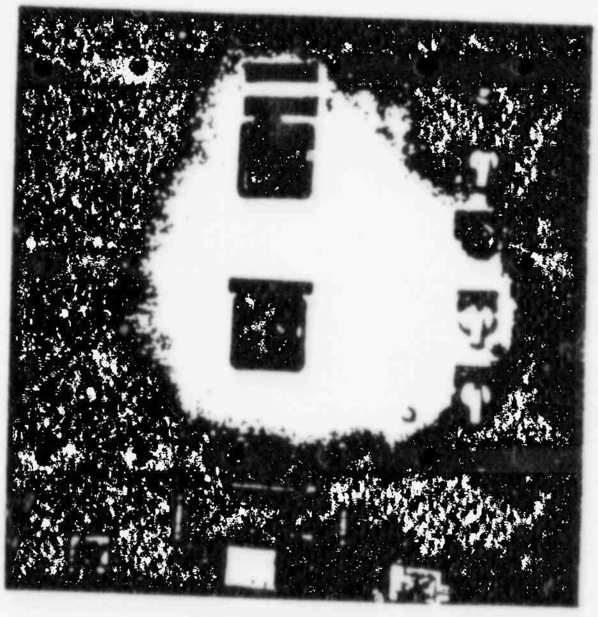

CUBICLE 1A DOOR PANEL
(CLOSE UP MIDDLE)

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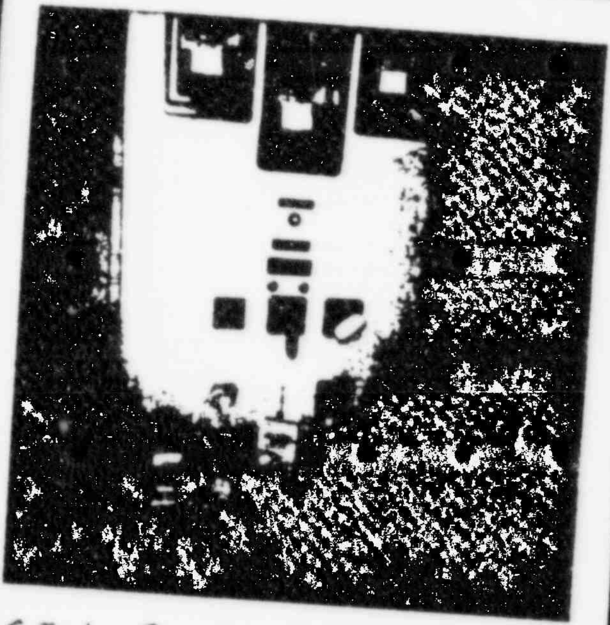
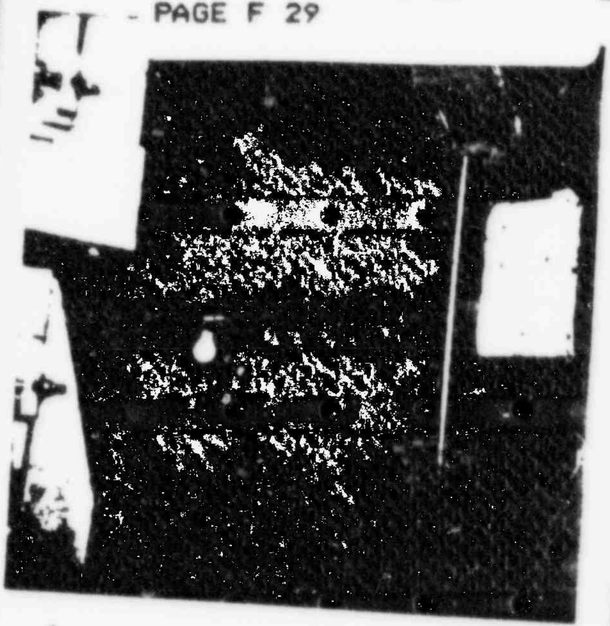


GEN. SWITCHGEAR
CUBICLE 1
(COMMON SWITCHGEAR)



AUXILIARY SWITCHGEAR