ISOLATION/PURGE VALVE ANALYSIS

FOR

18"-1200 BUTTERFLY VALVE

Project Site Perry Nuclear Power Station

Units 1 & 2

Customer Cleveland Electric Illuminating Co.

Engineer Gilbert/Commonwealth

Specification No. SP-641-4549-00 Rev. XII

Original Purchase Order P-1248-K/SP-641 Amendment 7Z

Original Pratt Job No. D-45608 (D-0086-10)

Valve Tag Nos. RNP-6-1M14-F190, RNP-6-1M14-F200

RNP-6-2M14-F190, RNP-6-2M14-F200

General Arrangement Drawings C-6021 Rev. 2

Cross Section Drawing C-6020 Rev. 0

Prepared by: Rao N. Kaza

Date: 9-6-83

Reviewed by: [illegible]

Date: Sept. 7, 1983

Certified by: J.V.B[illegible]

Date: 9-21-83

JOHN V. BALLUN
39860
REGISTERED
PROFESSIONAL
ENGINEER
OF
ILLINOIS

CONTENTS

I. Introduction

This investigation has been made in response to a request by the customer/engineer for evaluation of containment isolation/purge valves during a faulted condition arising from a loss of coolant accident (LOCA).

The analysis of the structural and operational adequacy of the valve assembly under such conditions is based principally upon containment pressure versus time data, system response (delay) time, piping geometry upstream of the valve, valve orientation and direction of valve closure.

The above data as furnished by the customer/engineer forms the basis for the analysis. Worst case conditions have been applied in the absence of definitive input.

II. Considerations

The NRC guidelines for demonstration of operability in purge and vent valves dated 9/27/79, have been incorporated in this evaluation as follows:

A.1. Valve closure time during a LOCA will be less than <u>or</u> equal to the no-flow time demonstrated during shop tests, since fluid dynamic effects tend to close a butterfly valve. Valve closure rate vs. time is based on a sinusoidal function.

2. Flow direction through valve contributing to highest torque; namely, flow toward the hub side of disc if asymmetric, is used in this analysis. Pressure on upstream side of valve as furnished by customer/engineer is utilized in calculations. Downstream pressure vs. LOCA time was assumed to be worst case.

3. Worst case is determined as a single valve closure of the inside containment valve, with the outside containment valve fixed at the fully open position.

4. Containment back pressure will have no effect on cylinder operation since the same back pressure will also be present at the inlet side of the cylinder and differential pressure will be the same during operation.

5. Purge valves supplied by Henry Pratt Company do not normally include accumulators. Accumulators, when used, are for opening the valve rather than closing.

6. Torque limiting devices apply only to electric motor operators which were not furnished with purge valves evaluated in this report.

7&8. Drawing <u>or</u> written description of valve orientation with respect to piping immediately upstream, as well as direction of valve closure, is furnished by customer/engineer. In this report worst case conditions have been considered; 90° elbow (upstream) oriented 90° out of plane with respect to valve shaft, and leading edge of disc closing toward outer wall of elbow. Effects of downstream piping on system back pressure have been covered in paragraph A.2 (above).

The Pratt purge valve analysis program was developed for indicated LOCA conditions using existing Pratt model test data. During 1982, Pratt undertook additional model testing to consider alternate valve/piping configurations, such as elbows immediately and two diameters upstream of the valve with valve shaft "out-of-plane" with respect to elbows, flow from flat and arch side of disc, clockwise and counter-clockwise disc closure, and disc diameter to thickness ratios. The dynamic torques determined by the model tests were in all cases lower than calculated by the Pratt purge valve analysis program.

B. This analysis consists of a static analysis of the valve components indicating if the stress levels under combined seismic and LOCA conditions are less than allowable stresses and/or 0.40 x yield strength for shear (non-Code components) in Table 1 of the materials used.

A valve operator evaluation is presented based on the operator manufacturer's rating versus the calculated LOCA-induced fluid dynamic torques.

C. Sealing integrity can be evaluated as follows: Decontamination chemicals have very little effect on EPT and stainless steel seats. Molded EPT seats are generically known to have a cumulative radiation resistance of 1×10^{8} RADS at a maximum incidence temperature of 350°F. It is recommended that seats are visually inspected every 18 months and be replaced periodically as required.

Valve at outside ambient temperatures below 0°F, if not properly adjusted, may have leakage due to thermal contraction of the elastomer; however, during a LOCA, the valve internal temperature would be expected to be higher than ambient which tends to increase sealing capability after closure. The presence of debris <u>or</u> damage to the seats would obviously impair sealing.

This report includes a sizing factor which is a function of the valve Reynolds number and the 5" model Reynolds number. The sizing factor for the 18" valve is equal to 1.176 i.e. it increases the torque by approximately 17.6%. In response to questions on the earliest purge valve reports regarding the applicability of a sizing factor, Pratt developed a sizing factor based on the relationship between air foil lift coefficients and Reynolds number. Pratt has applied the sizing factor to all purge valve reports furnished for over two years.

III. Method of Analysis

Determination of the structural and operational adequacy of the valve assembly is based on the calculation of LOCA-induced torque, valve stress analysis and operator evaluation.

A. Torque calculation

The torque of any open butterfly valve is the summation of fluid dynamic torque and bearing friction torque at any given disc angle.

Bearing friction torque is calculated from the following equation:

$$T_B = P \times A \times U \times \frac{d}{2}$$

where

P =pressure differential, psi

A = projected disc area normal to flow, in^2

U = bearing coefficient of friction

d = shaft diameter, in.

Fluid dynamic torque is calculated from the following equations:

For subsonic flow

$$\left[R_{CR} \geq \frac{P_1}{P_2} > 1.07 \text{ (approx.)} \right]$$

$$T_D = D^3 \times C_{T1} \times P_2 \times \sqrt{\frac{K}{1.4}} \times F_{RE}$$

For sonic flow

$$\left[\frac{P_1}{P_2} \geq R_{CR} \right]$$

$$T_D = D^3 \times C_{T2} \times P_2 \times \sqrt{\frac{K}{1.4}} \times F_{RE} \qquad (F_{RE} \geq 1)$$

Where

T_D = fluid dynamic torque, in-lbs.

F'_{RE} = Reynold number factor

R_{CR} = critical pressure ratio, (f (α))

P_1 = upstream static pressure at flow condition, psia

P_2 = downstream static pressure at flow condition, psia

D = disc diameter, in.

C_{T1} = subsonic torque coefficient

C_{T2} = sonic torque coefficient

K = isentropic gas exponent ($\simeq$ 1.2 for air/steam mix)

α = disc angle, such that 90° = fully open; 0° = fully closed

Note that C_{T1} and C_{T2} are a function of disc angle, an exponential function of pressure ratio, and are adjusted to a 5" test model using a function of Reynolds number.

Torque coefficients and exponential factors are derived from analysis of experimental test data and correlated with analytically predicted behavior of airfoils in compressible media.

Empirical and analytical findings confirm that subsonic and sonic flow conditions across the valve disc have an unequal and opposite effect on dynamic torque. Specifically, increases in upstream pressure in the subsonic range result in higher torque values, while increasing P_1 in the sonic range results in lower torques. Therefore, the point of greatest concern is the condition of initial sonic flow, which occurs at a critical pressure ratio.

The effect of valve closure during the transition from subsonic to sonic flow is to greatly amplify the resulting torques. In fact, the maximum dynamic torque occurs when initial sonic flow occurs coincident with a disc angle of 72° (symmetric) or 68° (asymmetric) from the fully closed position.

The following computer output summarizes calculation data and torque results for valve opening angles of 90° to 0°.

D-45608(D0086-10)   TORQUE TABLE 1   7 / 18 / 83

JOB:GILBERT/PERRY
SAT.STEAM/AIR MIXTURE WITH 1.4 LBS STEAM PER 1-LBS AIR
SPEC.GR.= .738255   MOL.WT.= 21.3872   KAPA(ISENT.EXP.)= 1.19775   R= 72.1972
GAS CONSTANT-CALC.
SONIC SPEED(MOVING MIXTR.)= 1190.42   FEET/SEC AT 100   DEG.

MAX.TORQUE INCLUDES SIZE EFFECT(REYNOLDS NO.ETC)APPX. X 1.17639   FOR 18   IN
CH BASIC LINE I.D.
ALL PRESSURES USED:STATIC(TAP)PRESS.-ABSOLUTE;P2 INCL.RECOVERY PRESS.
(TORQUE)CALC'S VALIDITY:P1/P2>1.07;

VALVE TYPE: 18"-1200   CLASS 150
DISC SIZE: 15.7   INCHES   OFFSET ASYMMETRIC DISC
SHAFT DIA.: 2.25   INCHES
BEARING TYPE: BRONZE
SEATING FACTOR: 15
INLET PRESS.VAR.MAX.: 26.7   PSIA
MAX.ANG.FLOW RATE: 56596.4   CFM; 95450.8   SCFM; 5247.2   LB/MIN
CRIT.SONIC FLOW-90DG: 5723.17   LB/MIN AT 17.962   INLET PSIA
VALVE INLET DENSITY: 9.27126E-02   LB/FT^3-MIN. 9.27125E-02   LB/FT^3-MAX.
SYSTEM CONDITIONS:
PIPE IN-PIPE-OUT -AND- AIR/STEAM MIXTURE SERVICE @ 100 DEG.F
MINIMUM 0.75 DIAM. PIPE DOWNSTREAM FROM CENT.LINE SHAFT.

P1 ABS. PRESSURE(ADJ.)FOLLOWS TIME/PRESS.TRANSIENT CURVE.

--5 IN.MODEL EQUIV.VALUES						------ACTUAL SIZE VALUES-----				
ANGLE APPRX.	P1 PSIA	P2 PSIA	DELP PSI	PRESS. RATIO	FLOW (SCFM)	FLOW (LB/MIN)	TD ----INCHLBS	TB+TH ----	TD-TB-TH	TIME(LOCA) SEC.
90	26.70	14.75	11.95	.552 CR	95450	5247	5922	385	5536	1.00
85	26.70	14.75	11.95	.552	115184	6332	6134	399	5734	1.35
80	26.70	14.75	11.95	.552	112384	6178	6248	407	5841	1.68
75	26.70	14.75	11.95	.552	106326	5845	11312	736	10575	2.00
70	26.70	14.75	11.95	.552 CR	92506	5085	12145	791	11354	2.18
68	26.70	14.75	11.95	.553 CR	96199	5288	12233	796	11436	2.29
65	26.70	14.75	11.95	.552	80771	4440	10725	698	10027	2.53
60	26.70	14.75	11.95	.552	67774	3725	6769	440	6327	2.73
55	26.70	14.75	11.95	.552	56581	3110	5091	348	4743	2.88
50	26.70	14.75	11.95	.552	46088	2533	3478	390	3088	2.97
45	26.70	14.75	11.95	.552	47493	2610	2860	429	2431	3.00
40	26.70	14.75	11.95	.552	33928	1865	2101	464	1636	3.03
35	26.70	14.75	11.95	.552	22204	1220	1234	497	737	3.12
30	26.70	14.75	11.95	.552	16439	903	733	525	208	3.27
25	26.70	14.75	11.95	.552	12376	680	518	549	-31	3.47
20	26.70	14.75	11.95	.552	7356	404	400	570	-169	3.71
15	26.70	14.75	11.95	.552	3840	211	247	586	-338	4.00
10	26.70	14.75	11.95	.552	1794	98	187	597	-409	4.32
5	26.70	14.75	11.95	.552	467	25	153	604	-451	4.65
0	26.70	14.70	12.00	.551	0	0	5513	653	4859	5.00

SEATING + BEARING + HUB SEAL TORQUE (M/M)=   5513 IN-LBS @ 0 DEG.

B. Valve Stress Analysis

The Pratt butterfly valve furnished was specifically designed for the requirements of the original order which did not include specific LOCA conditions.

The valve stress analysis consists of two major sections: 1) the body analysis, and 2) all other components.

The body is analyzed per rules and equations given in paragraph NB-3545 of Section III of the ASME Boiler and Pressure Vessel Code. The other components are analyzed per a basic strength of materials type of approach. For each component of interest, tensile and shear stress levels are calculated. They are then combined using the formula:

$$S_{max} = \frac{1}{2}(T_1+T_2) + \frac{1}{2}\sqrt{(T_1+T_2)^2 + 4(S_1+S_2)^2}$$

where

S_{max} = maximum combined stress, psi

T_1 = direct tensile stress, psi

T_2 = tensile stress due to bending, psi

S_1 = direct shear stress, psi

S_2 = shear stress due to torsion, psi

The calculated maximum valve torque resulting from LOCA conditions is used in the seismic stress analysis, attachment #2, along with "G" loads per design specification. The calculated stress values are compared to code allowables and .40 x yield strength for non-code shear.

C. Operator Evaluation

Model: Bettis T312B-SR2

Rating: 47,000 in-lbs at full open and closed positions only

26,000 in-lbs at 45^{o} (minimum rating)

Maximum valve torque: 12,233 in-lbs

The maximum torque generated during a LOCA induces reactive forces in the load carrying components of the actuator. Since the LOCA induced torque calculated in this analysis is lower than the absorption rating of the operator, it is concluded that the Bettis models furnished are structurally suitable to withstand combined LOCA and seismic loads as defined in this analysis.

IV. CONCLUSION:

The calculated stresses of the valve components for combined seismic and LOCA conditions as shown in Table 1 of Attachment 2 are less than allowable stresses and/or 0.40 x yield strength for shear (non-Code components).

It is concluded that the valve structure and the valve actuator are capable of withstanding combined seismic and LOCA-induced loads based on the calculated torques developed in the analysis.

ATTACHMENT 1A

PRATT PROPOSAL LETTER

PRATT

HENRY PRATT COMPANY

401 SOUTH HIGHLAND AVENUE · AURORA, ILLINOIS 60507

December 29, 1982

Perry Nuclear Power Plant
The Cleveland Electric Illuminating Co.
P.O. Box 97
Perry, Ohio 44081

Attention: Mr. W. W. Nix

Subject: Cleveland Electric Illuminating Co.
Perry Nuclear Plant
Containment Purge Valves
Original Pratt Order No. D0086
Item 10 - 18" Tag M14-F190
Item 8 - 42" Tag M14-F040
Inquiry No. 101982-201-U dated 10/19/82

Gentlemen:

This letter includes a proposal for performing analyses on the subject valves similar to that furnished to other plants. This proposal covers the 18" and 42" valves only based on Steve Lemmo's 11/5/82 telephone call.

This proposal includes:

1. Aerodynamic torque calculations will be performed based based on the following considerations:

   A. Containment Pressure and/or Temperature - Time curves furnished by you.

   B. Maximum delay times from LOCA to initiation of valve rotation and no load valve closing time to be furnished by you.

   C. A valve sizing factor will be included (model vs. actual).

   D. It will be assumed that first incidence of sonic flow coincides with the critical valve disc angle as a worst case condition. This supports the presence of out of plane elbows immediately upstream of the valve with worst case valve flow direction and worst case valve closing direction from the full open to the closed position.

2. The calculated aerodynamic torque will be compared to the manufacturer's rating for the actuator.

3. A static stress analysis will be performed for valve components affected by the calculated aerodynamic torque loadings in combination with pressure and seismic loads. Code allowables will be used except that 0.4 times yield strength will be used as the allowable for non-Code components in shear.

4. Where the actuator rating or valve component allowables are exceeded based on closure from the full open position, the maximum valve opening angle required to reduce the calculated torque and/or component stresses to allowable levels will be provided. The static analysis referenced in 3. above will be based on the maximum opening angle.

5. This proposal is based on our current analysis program and test data. It does not include any additional testing.

6. This proposal does not include the design or furnishing of any modifications to increase the maximum opening angle.

7. Our response to NRC's criteria for demonstrating operability of purge valves dated 9/27/79 will be included in the analysis.

8. The cost of performing this analysis will be [redacted] for the 18" valves and [redacted] for the 42" valves. The terms of payment will be net 30 days after shipment of the analysis report(s). This proposal is valid for thirty (30) days.

9. The completion of this analysis is projected to be twenty (20) weeks after receipt and entry of purchase order and current availability of engineering schedule.

10. This proposal incorporates the "Terms and Conditions for Nuclear Analysis Orders" attached. If you have any objections to any of the terms, advise us in writing within fifteen days from the date of this proposal. If we do not hear from you as stated above, we assume you agree to these terms and will disregard contrary terms in your printed purchase order.

We hope you will find this proposal responsive to your needs. If we can be of any additional assistance in this matter, please advise.

Very truly yours,

HENRY PRATT COMPANY

Glenn Beane

Glenn L. Beane
Manager, Application Engineering

GLB/np
Enclosure
CC: A. K. Wilson

Proposal No: X51-7051P

TERMS AND CONDITIONS FOR NUCLEAR ANALYSIS ORDERS

LIMITATION OF LIABILITY

Henry Pratt Company (hereinafter "Seller") shall not be liable for loss of profit, special, incidental, indirect or consequential damage or loss. Any claim hereunder arising from, or pertaining to, services provided hereunder whether based on contract or tort shall not exceed the price quoted herein.

WARRANTY

Since the actual conditions in a loss of cooling accident (LOCA) are unknown, Seller's analysis only covers conditions as specified by the buyer. This proposal is for investigative analysis only and does not guarantee or warrant the adequacy of the equipment as originally furnished when subjected to conditions currently specified nor does it extend the original warranty.

INDEMNIFICATION-NUCLEAR INCIDENT

Buyer or the Owner of the nuclear facility will furnish nuclear liability protection in accordance with Section 170 of the Atomic Energy Act (42 U.S.C. Section 2210) and applicable regulations of the Nuclear Regulatory Commission. Should this system of protection be repealed or changed, Buyer or the Owner of the nuclear facility will maintain in effect during the period of operation of the plant, liability protection which would not result in a material impairment of the protection afforded to the Seller under the existing system.

Buyer waives any claim it might have against the Seller because of damage to, loss of, or loss of use of Buyer's property at the site of the nuclear facility resulting from nuclear energy hazards or nuclear incidents.

Buyer will indemnify the Seller and save it harmless from any loss or damage resulting from nuclear energy hazards or nuclear incidents on the site of the nuclear facility.

The foregoing waiver and indemnification provisions will apply to the full extent permitted by law and regardless of fault.

For purposes of these provisions the following definitions shall apply: "Nuclear energy hazards" shall mean the hazardous properties of nuclear material. "Hazardous properties" shall include radioactive, toxic, or explosive properties of nuclear material. "Nuclear material" shall include source material, special nuclear material or by-product material as those are defined in the Atomic Energy Act (42 U.S.C. Section 2014). "Nuclear incident" shall have the meaning given that term in the Atomic Energy Act (42 U.S.C. Section 2014(o)).

12/21/82

ATTACHMENT 1B

CUSTOMER/ENGINEER RESPONSE FOR PROPOSAL CONFIRMATION

THE CLEVELAND ELECTRIC ILLUMINATING COMPANY
P.O. BOX 97 ■ PERRY, OHIO 44081 ■ TELEPHONE (216) 259-3737 ■ ADDRESS-10 CENTER ROAD
Serving The Best Location in the Nation
PERRY NUCLEAR POWER PLANT

April 8, 1983

HENRY PRATT COMPANY
Creative Engineering for Fluid Systems
401 South Highland Avenue
Aurora, Illinois 60507

APPLICATION ENGR.
APR 12 1983

Attention: Glenn Beane

Subject: P- 1248/SP-641

Reference: (a) Inquiry #101982-301-U dated 10/19/82
(b) Henry Pratt's quote dated 12/29/82

Gentlemen:

Based on numerous phone conversations, please confirm the proposal will include the following:

1. Aerodynamic torque calculations will be performed based on the following considerations:

   a. Containment Pressure and Temperature versus Time curves furnished by the Owners (and attached) shall be used for the analysis.
   b. Maximum delay times from LOCA to <u>initiation</u> of valve rotation and no load valve closing time to be <u>furnished</u> by the Owners.
      1. The delay time from LOCA to initiation of valve rotation shall be one (1) second.
      2. The no load closing time for the valves shall be the times recorded during the Henry Pratt Company shop tests.
   c. A valve sizing factor will be included (model vs. actual), and include an explanation of how the "valve sizing factor" was determined.
   d. It will be assumed that first incidence of sonic flow coincides with the critical valve disc angle as a worst case condition. This supports the presence of out of plane elbows immediately upstream of the valve with worst case valve flow direction and worst case valve closing direction from the full open tothe closed position. That is,the Supplier shall explain their parameters for considering the effect of pipe bends or partially closed valves upstream of the subject valves in an aerodynamic torque calculation. The Supplier shall also explain how the parameters used for analysis are applicable to, or, are more conservative than Perry's unique system design.

2. The calculated aerodynamic torque will be compared to the manufacture's rating for the actuator.

3. A static stress analysis will be performed for valve components affected by the calculated aerodynamic torque loadings in combination with pressure and seismic loads. The seismic loads used for the static stress analysis

shall be from the curves furnished with ECN 10301-641-1 previously sent. Code allowables will be used except that 0.4 times yield strength will be used as the allowable for non-Code components in shear.

4. Where the actuator rating or valve component allowables are exceeded based on closure from the full open position, the maximum valve opening angle required to reduce the calculated torque and/or component stresses to allowable levels will be provided. The static analysis referenced in 3. above will be based on the maximum opening angle.

5. This proposal is based on our current analysis program and test data. It does not include any additional testing.

6. This proposal does not include the design or furnishing of any modifications to increase the maximum opening angle of the valves previously agreed up to and including Spec. Rev. X

7. Our response to NRC's criteria for demonstrating operability of purge valves dated 9/27/79 will be included in the analysis.

8. The cost of performing this analysis will be [redacted] for the 18" valves and [redacted] for the 42" valves. The terms of payment will be net 30 days after receipt of the analysis report (s). This proposal is valid for thirty (30) days.

9. The completion of this analysis is projected to be twenty (20) weeks or sooner after receipt and entry of purchase order and current availability of engineering schedule.

10 This proposal incorporated the "Terms and Conditions for Nuclear Analysis Orders" attached. If you have any objections to any of the terms, advise us in writing within fifteen days from the date of this proposal. If we do not hear from you as stated above, we assume you agree to these terms and will disregard contrary terms in your printed purchase order.

Sincerely,

Norman G. Dillen

Norman G. Dillen
Perry Project Services Dept.

NGD:dms

cc: J. Barron
L. Wynn
D. Brockett
A. Vild
File

ATTACHMENT 1C

PRATT'S RESPONSE TO CUSTOMER/ENGINEER'S RESPONSE

PRATT

HENRY PRATT COMPANY

401 SOUTH HIGHLAND AVENUE · AURORA, ILLINOIS 60507

April 14, 1983

Perry Nuclear Power Plant
The Cleveland Electric Illuminating Co.
P.O. Box 97
Perry, Ohio 44081

Attention: Mr. Norman G. Dillen
Perry Project Services Dept.

Subject: P-1248/SP-641
X-51-7051P/D0086

Reference: a) Inquiry No. 101982-301-U dated 10/19/82
b) Henry Pratt's quote dated 12/29/82
c) Your letter to Glenn Beane dated 4/8/83

Gentlemen:

Thank you for your referenced letter regarding our proposal to furnish purge valve analyses for the 18" and 42" valves.

Item 1.c. The valve sizing factor used is a function of the ratio of the valve Reynolds Number and the model Reynolds Number. The factor used will be included in the report. V + Basis for factor.

d. Pratt has conducted additional model testing to consider alternate valve/piping configurations including elbows immediately and two diameters upstream of the valve with valve shaft "out of plane" with respect to elbows, flow from flat and arch side of disc, clockwise and counter clockwise disc closure and disc diameter to thickness ratios. The torques determined by the model tests were in all cases lower than calculated by the analysis program and existing data base.

Pratt has not conducted specific model tests with partially open upstream valves but believes they would tend to lower torques because of the pressure drop across the upstream valve. If model tests with partially open upstream valves or other unique configuration are desired, these may be quoted at additional costs.

Item 3. The new seismic loads can be used in the analysis. It should be noted that these loads may indicate the need for structural modifications which are not included in this proposal.

Item 4. This proposal is for investigative analysis only and does not guarantee or warrant the adequacy of the equipment as originally furnished when subjected to conditions currently identified nor does it extend the original warranty. Accordingly this proposal does not include furnishing any redesign or modifications. Such items may be quoted at additional cost. Please note that operability assurance per your inquiry 020783-305-2 is being quoted separately.

Please consider furnishing your purchase order for this analysis this month as our current Engineering schedule would permit completion in twelve weeks if order is received by April 30, 1983.

Very truly yours,

HENRY PRATT COMPANY

Glenn Beane

Glenn L. Beane
Manager, Application
Engineering

GLB/np

CC: A. K. Wilson
J. R. Holstrom

ATTACHMENT 1D

CUSTOMER/ENGINEER'S P.O. & NECESSARY INFORMATION

EQUIPMENT PURCHASE AGREEMENT

Perry Nuclear Power Plant
Title PURGE ISOLATION VALVES
Agreement No. P-1243-K/SP-041
Amendment No. 72

THIS AGREEMENT between THE CLEVELAND ELECTRIC ILLUMINATING COMPANY ("CEI"), an Ohio corporation, DUQUESNE LIGHT COMPANY, A Pennsylvania corporation, OHIO EDISON COMPANY, an Ohio corporation, and its subsidiary, PENNSYLVANIA POWER COMPANY and TOLEDO EDISON COMPANY, an Ohio corporation, owners in common of undivided interests in the PERRY NUCLEAR POWER PLANT UNIT NO. 1 and UNIT NO. 2 PROJECT (the "OWNERS"), and HENRY PRATT COMPANY of Aurora, Illinois

(the "SUPPLIER"), is hereby amended effective as of the 23rd day of May, 1983.

12. PRICE, Page 7

The SUPPLIER agrees to receive as full compensation for the performance of this Agreement within the times fixed, the sum of --00/100 Dollars ($).
The obligation of the OWNERS to pay said compensation shall be several and not joint and will be limited to each OWNER'S percentage of undivided ownership in the Perry Nuclear Power Plant Unit No. 1 and Unit No. 2 Project, such percentage being 31.11% in the case of The Cleveland Electric Illuminating Company, 13.74% in the case of Duquesne Light Company, 30% in the case of Ohio Edison Company, 5.24% in the case of Pennsylvania Power Company and 19.91% in the case of Toledo Edison Company.

The Tax Commissioner for the State of Ohio issued Direct Payment Permit No. 98-001843 to the OWNERS on October 18, 1973, and the sales and use taxes applicable to materials and property purchased by the permit holder subsequent thereto are being paid directly to the Treasurer of the State by the OWNERS.

Payments are to be made at the times and in the manner provided in Article 13.

The agreement price through amendment no. 6 was $.

SCHEDULE B

SPECIAL OR EXPLANATORY PROVISIONS

1. Scope--Article 1

A. This Amendment covers the cost to furnish purge valve analysis for the 18" and 42" valves.

1. Aerodynamic torque calculations will be performed based on the following considerations:

a. Containment Pressure and Temperature versus Time curves furnished by the Owners shall be used for the analysis.
b. Maximum delay times from LOCA to initiation of valve rotation and no load valve closing time to be furnished by the Owners.
   1. The delay time from LOCA to initiation of valve rotation shall be one (1) second.
   2. The no load closing time for the valves shall be the times recorded during the Henry Pratt Company shop tests.
c. The valve sizing factor used is a function of the ratio of the valve Reynolds Number and the model Reynolds Number.

Further, the Supplier agrees to:

1. State reason/justification for adding the valve sizing factor. If it was a request by the NRC, state who asked for it and when.
2. Show the developed torques from the calculation with the sizing factor and then show the method of determining the developed torques without the sizing factor. Also, the Supplier is to furnish how much of a safety factor is added.
3. State whether the Supplier feels the sizing factor is applicable and why.

d. It is understood the Supplier has conducted additional model testing to consider alternate valve/piping configurations including elbows immediately and two diameters upstream of the valve with valve shaft "out of plane" with respect to elbows, flow from flat and arch side of disc, clockwise and counter clockwise disc closure and disc diameter to thickness ratios. It is understood the torques determined by the model tests were in all cases lower than calculated by the analysis program and existing data base.

Model tests to date do not include partially open upstream valves or Owner responsible designs. This option (should the Owners so desire) is available at additional cost.

2. The calculated aerodynamic torque will be compared to the manufacturer's rating for the actuator.

3. A static stress analysis will be performed for valve components affected by the calculated aerodynamic torque loadings in combination with pressure and seismic loads. The seismic loads used for the static stress analysis shall be from the curves furnished with ECN 10301-641-1 previously sent. Code allowables will be used except that 0.4 times yield strenght will be used as the allowable for non-Code components in shear.

4. Where the actuator rating or valve component allowables are exceeded based on closure from the full open position, the maximum valve opening angle required to reduce the calculated torque and/or component stresses to allowable levels will be provided. The static analysis referenced in 3. above will be based on the maximum opening angle.

   This analysis does not guarantee or warrant the adequacy fo the equipment when subjected to specifications not previously known or agreed to by the Supplier. The associated cost for any rework (ie, redesign and/or modifications) is not included in the Amendment.

5. Supplier's response to NRC's criteria for demonstrating operability or purge valves dated 9/27/79 shall be included in the analysis.

II. Price--Article 12

A. The cost for performing the analysis will be [redacted] for the 18" valves and [redacted] for the 42" valves. The terms of payment will be net 30 days after receipt of report.

III. Commencement and Completion of Work

Start Work May 31, 1983 June 20, 1983

Completion date Week of August 15, 1983 Sept. 12, 1983

Monthly, or more often if appropriate, the Supplier shall submit a progress report to the Owners (c/o CEI). If any phase of the work is not on schedule, the Supplier shall state what remedies have been instituted to correct the situation.

IV. The following conditions also apply:

1. LIMITATION OF LIABILITY

The Supplier shall not be liable for loss of profit, special, incidental, indirect or consequential damage or loss. Any claim hereunder arising from, or pertaining to, services provided hereunder whether based on contract or tort shall not exceed the price quoted herein.

2. WARRANTY

Since the actual conditions in a loss of cooling accident (LOCA) are unknown, Supplier's analysis only covers conditions as specified by the Owners. This Amendment is for investigative analysis only and does not guarantee or warrant the adequacy of the equipment than otherwise specified nor does it extend the original warranty.

3. INDEMNIFICATION-NUCLEAR INCIDENT

The Owners of the nuclear facility will furnish nuclear liability protection in accordance with Section 170 of the Atomic Energy Act (42 U.S.C. Section 2210) and applicable regulations of the Nuclear Regulatory Commission. Should this system of protection be repealed or changed, the Owners of the nuclear facility will maintain in effect during the period of operation of the plant, liability protection which would not result in a material impairment of the protection afforded to the Supplier under the existing system.

The Owners waive any claim it might have against the Supplier because of damage to, loss of, or loss of use of Owners' property at the site of the nuclear facility resulting from nuclear energy hazards or nuclear incidents.

The Owners will indemify the Supplier and save it harmless from any loss or damage resulting from nuclear energy hazards or nuclear incidents on the site of the nuclear facility.

The foregoing waiver and indemnification provisions will apply to the full extent permitted by law and regardless of fault.

For purposes of these provisions the following definitions shall apply: "Nuclear energy hazards" shall mean the hazardous properties of nuclear material. "Hazardous properties" shall include radioactive, toxic, or explosive properties of nuclear material. "Nuclear material" shall include source material, special nuclear material or by-product material as those are defined in the Atomic Energy Act (42 U.S.C. Section 2014). "Nuclear incident" shall have the meaning given that term in the Atomic Energy Act (42 U.S.C. Section 2014(o)).

V. Appointment of Representatives, Powers and Functions--Article 2

A. Under Article 2, Subitem (a), Page 2 delete R.H. McNeal, Manager, Purchasing Department and substitute henceforth R.L. Farrell, Manaer, Perry Project Services Department. Also, delete G.W. Groscup, Manager, Nuclear Engineering Department and substitute henceforth F.R. Stead, Manager, Nuclear Engineering Department.

The Agreement is hereby ratified and confirmed in all other respect than herein above set forth.

THE CLEVELAND ELECTRIC ILLUMINATING COMPANY, for itself and as Agent for DUQUESNE LIGHT COMPANY, OHIO EDISON COMPANY, PENNSYLVANIA POWER COMPANY and TOLEDO EDISON COMPANY

Dated 6/7, 1983

By ______________________
Manager, Perry Project Services Department

HENRY PRATT COMPANY

Dated ______________, 19

By ______________________

Title ______________________

THE CLEVELAND ELECTRIC ILLUMINATING COMPANY

P.O. BOX 97 ■ PERRY, OHIO 44081 ■ TELEPHONE (216) 259-3737 ■ ADDRESS-10 CENTER ROAD

Serving The Best Location in the Nation
PERRY NUCLEAR POWER PLANT

July 26, 1983
PY/SO-641-18851

Henry Pratt Company
401 South Highland Avenue
Aurora, Illinois 60507

Attention: Mr. John Holstrum

RE: Purge and Vent Valve Analysis
SP-641 P-1248

Dear John:

The purpose of this letter is to document our telephone conversation of July 22, 1983.

Two major items were discussed during the course of the afternoon.

1. GAI will clarify the 0 to 10 second interval for the Containment Pressure/Temperature versus Time Curves

2. The Cleveland Electric Illuminating Company will accept a static stress analysis using 3g's on two axes and 4g's on the remaining axis as requested by the Henry Pratt Company.

If you have any questions in these matters please call.

Very truly yours,

D.R. Brockett
D. R. Brockett
Responsible Engineer SP-641

E. J. Adams
E. J. Adams
Senior Engineer

DRB:EJA/cab

cc: L. Wynn
T. Rockwell GAI
S. Lemmo GAI
B. Rosch GAI
S. Litchfield
K. Matheny
NDS 3.1
NDS 31.9 SP-641
SO/DC

APPLICATION ENGRG.

MAY 1 9 1983

Gilbert/Commonwealth engineers and consultants

GILBERT ASSOCIATES, INC., P. O. Box 1498, Reading, PA 19603/Tel. 215-775-2600/Cable Gilasoc/Telex 836-431

May 13, 1983

Henry Pratt Company
401 South Highland Avenue
Aurora, Illinois 60507

Attention: Mr. Glenn Beane

Re: The Cleveland Electric Illuminating Co.
Perry Nuclear Power Plant
Purge and Vent Valve Operability Analysis

Gentlemen:

This letter is in response to your Rao Kaza's request for a more concise description of the pressure time relation inside the containment vessel during a LOCA. The following information is to be used in the operability analysis you are presently performing.

TIME	TEMP °F	PRESSURE (PSIG)
0-10 sec.	100	4.3
10 sec.-100 sec.	125	7.3
100 sec.-16.7 min.	150	10.3
16.7 min.-30 min.	165	7.3
30 min.-80 min.	180	10.3
80 min.-3 hrs.	185	12.0
3 hrs.-18 hrs.	160	6.3
18 hrs.-10 days	120	3.8
10 days-100 days	90	0.0
100 days-180 days	90	0.0

If you have any questions, please call.

Very truly yours,

S. M. Lemmo

S. M. Lemmo
Building Service Engineer

J. S. Holton

J. S. Holton
Lead Building Service Engineer

M. Plica for

P. B. Gudikunst
Project Manager

SML/JSH/PBG/pam
cc: PO/DC(Attn:SP-641) - 2
P. B. Gudikunst - 2
M. Plica
R. J. Sheldon
F. C. Rosch
D. Brockett
L. Wynn
J. M. Lastovka
J. S. Holton
D. S. Maulick
P. L. Bunker
J. Hollstrum

525 Lancaster Avenue, Reading, PA Morgantown Road, Green Hills, Reading, PA 215 775 2600
209 East Washington Avenue, Jackson, MI 517 788 3000; 80 Pine Street, New York, NY 212 482 8480

APPLICATION ENGINEERING

JUL 8 1983

THE CLEVELAND ELECTRIC ILLUMINATING COMPANY

P.O. BOX 97 ■ PERRY, OHIO 44081 ■ TELEPHONE (216) 259-3737 ■ ADDRESS-10 CENTER ROAD

Serving The Best Location in the Nation
PERRY NUCLEAR POWER PLANT

July 6, 1983
PY/SO-641-18501

Henry Pratt Company
401 S. Highland Avenue
Aurora, Illinois 60507

Attention: Mr. Glenn Beane

RE: SP-641
Purge and Vent Valve
Operability Analysis

Dear Glenn:

The purpose of this letter is to transmit Floor Response Spectra Curves necessary to perform the Purge and Vent Valve Operability Analysis (P-1248-K). Pratt is instructed to use these curves in lieu of any Floor Response Spectra Curves received prior to this date.

These curves will be incorporated into SP-641 at a later date.

Very truly yours,

D. R. Brockett

D. R. Brockett
Responsible Engineer SP-641

E. J. Adams

E. J. Adams
Senior Engineer

DRB:EJA/cab

cc: J. Lastovka
J. Eppich
L. Wynn
N. Dillen
A. Vild
P. Hess
S. Lemmo - GAI
T. Rockwell - GAI
NDS File 31.9 SP-641
SO/DC

New Loads Response Spectra: Fig. 1 through 16 with the following:

Fig. 1 of 20, Loading Combination 1, OBE + SRV, Vertical, El. 630'-0" and 644'-0", 2% damping, Fig. EQ-630/644-1-V, Rev. 0, 6-7-82

Fig. 2 of 20, Loading Combination 1, OBE + SRV, Horizontal, El. 644'-0", 2% damping, Fig. EQ-644-1-H, Rev. 0, 6-7-82

Fig. 3 of 20, Loading Combination 4, SSE + SRV + CHUG, Horizontal, El. 644'-0", 3% damping, Fig. EQ-644-4-H, Rev. 0, 6-7-82

Fig. 4 of 20, Loading Combination 4, SSE + SRV + CHUG, Vertical, El. 664'-0", 3% damping, Fig. EQ-644-4-V, Rev. 0, 6-7-82

Fig. 5 of 20, Loading Combination 4, SSE + SRV + CHUG, Horizontal, El. 664'-0", 3% damping, Fig. EQ-644-4-H, Rev. 0, 6-7-82

Fig. 6 of 20, Loading Combination 4, SSE + SRV + CHUG, Vertical, El. 644'-0", 3% damping, Fig. EQ-664-4-V, Rev. 0, 6-7-82

Fig. 7 of 20, Loading Combination 1, OBE + SRV, Horizontal, El, 664'-0", 2% damping, Fig. EQ-664-1-H, Rev. 0, 6-7-82

Fig. 8 of 20, Loading Combination 1, OBE + SRV, Vertical, El. 664'-0", 2% damping, Fig. EQ-664-1-V, Rev. 0, 6-7-82

Fig. 9 of 20, Floor Response Spectra, SSE + SRV + CHUG, Vertical, El. 693'-0", 3% damping, Rev. 2, 5-2-83

Fig. 10 of 20, Floor Response Spectra, OBE + SRV, Horizontal, El. 693'-0", 2% damping, Rev. 2, 5-2-83

Fig. 11 of 20, Floor Response Spectra, OBE + SRV, Vertical, El. 693'-0", 2% damping, Rev. 2, 5-2-83

Fig. 12 of 20, Floor Response Spectra, SSE + SRV + CHUG, Horizontal, El. 693'-0", 3% damping, Rev. 2, 5-2-83

RECEIVED

JUN 23 1983

Fig. 13 of 20, Floor Response Spectra, SSE + SRV + CHUG, Vertical
El. 674'-0", 3% damping, Rev. 2, 5-2-83

Fig. 14 of 20, Floor Response Spectra, SSE + SRV + CHUG, Horizontal,
El. 674'-0", 3% damping, Rev. 2, 5-2-83

Fig. 15 of 20, Floor Response Spectra, OBE + SRV, Horizontal,
El. 674'-0", 2% damping, Rev. 2, 5-2-83

Fig. 16 of 20, Floor Response Spectra, OBE + SRV, Vertical,
El. 674'-0", 2% damping, Rev. 2, 5-2-83

Fig. 17 of 20, Floor Response Spectra, SSE + SRV + CHUG, Vertical,
El. 701'-0", 3% damping, Rev. 0, 5-5-83

Fig. 18 of 20, Floor Response Spectra, SSE + SRV + CHUG, Horizontal,
El. 701'-0", 3% damping, Rev. 0, 5-5-83

Fig. 19 of 20, Floor Response Spectra, OBE + SRV, Vertical,
El. 701'-0", 2% damping, Rev. 0, 5-5-83

Fig. 20 of 20, Floor Response Spectra, OBE + SRV, Horizontal,
El. 701'-0", 2% damping, Rev. 0, 5-5-83

NO. 340-1210 DIETZGEN GRAPH PAPER
SEMI-LOGARITHMIC
2 CYCLES X 10 DIVISIONS PER INCH

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MADE IN U.S.A.

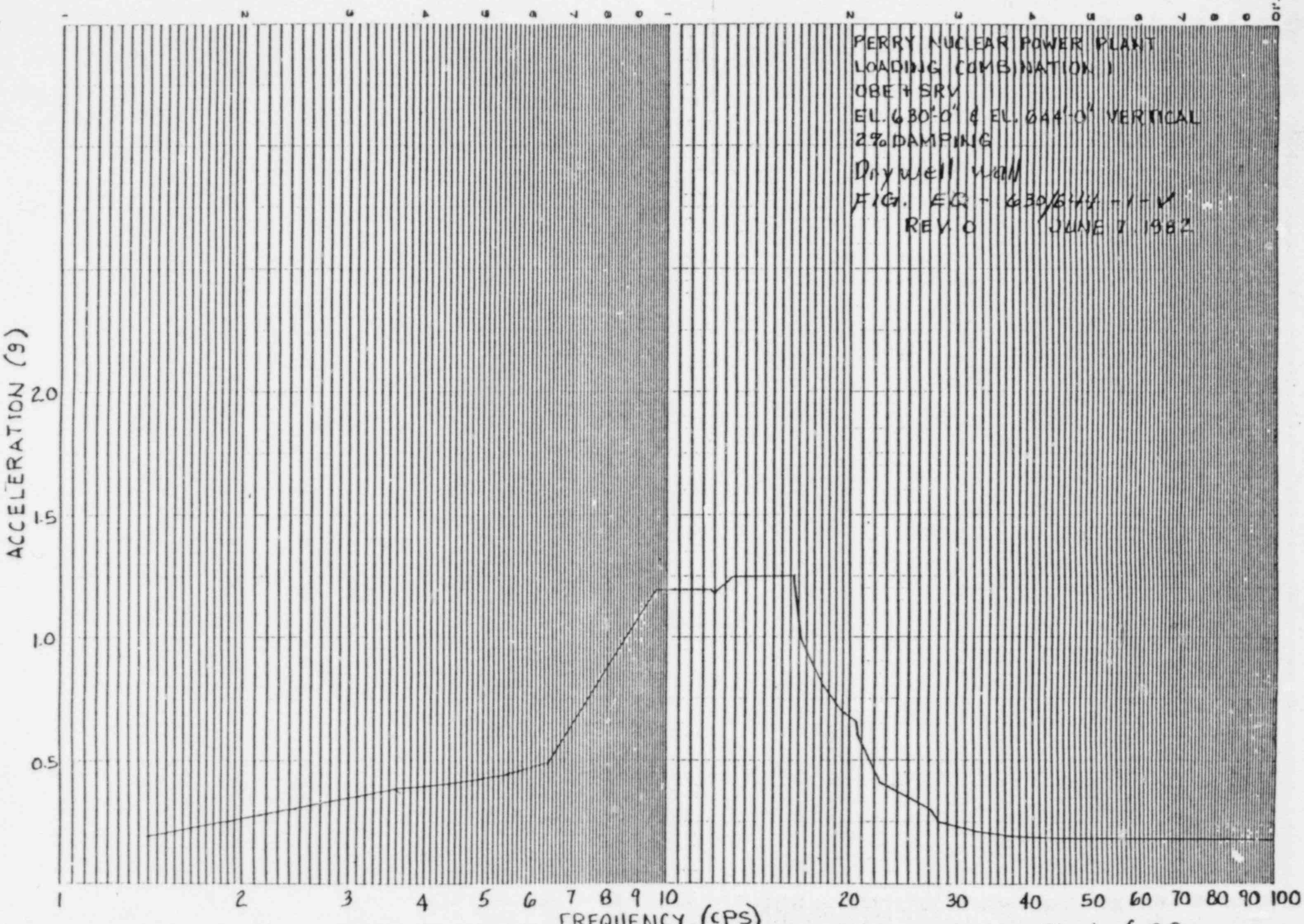

Fig. 1 of 20

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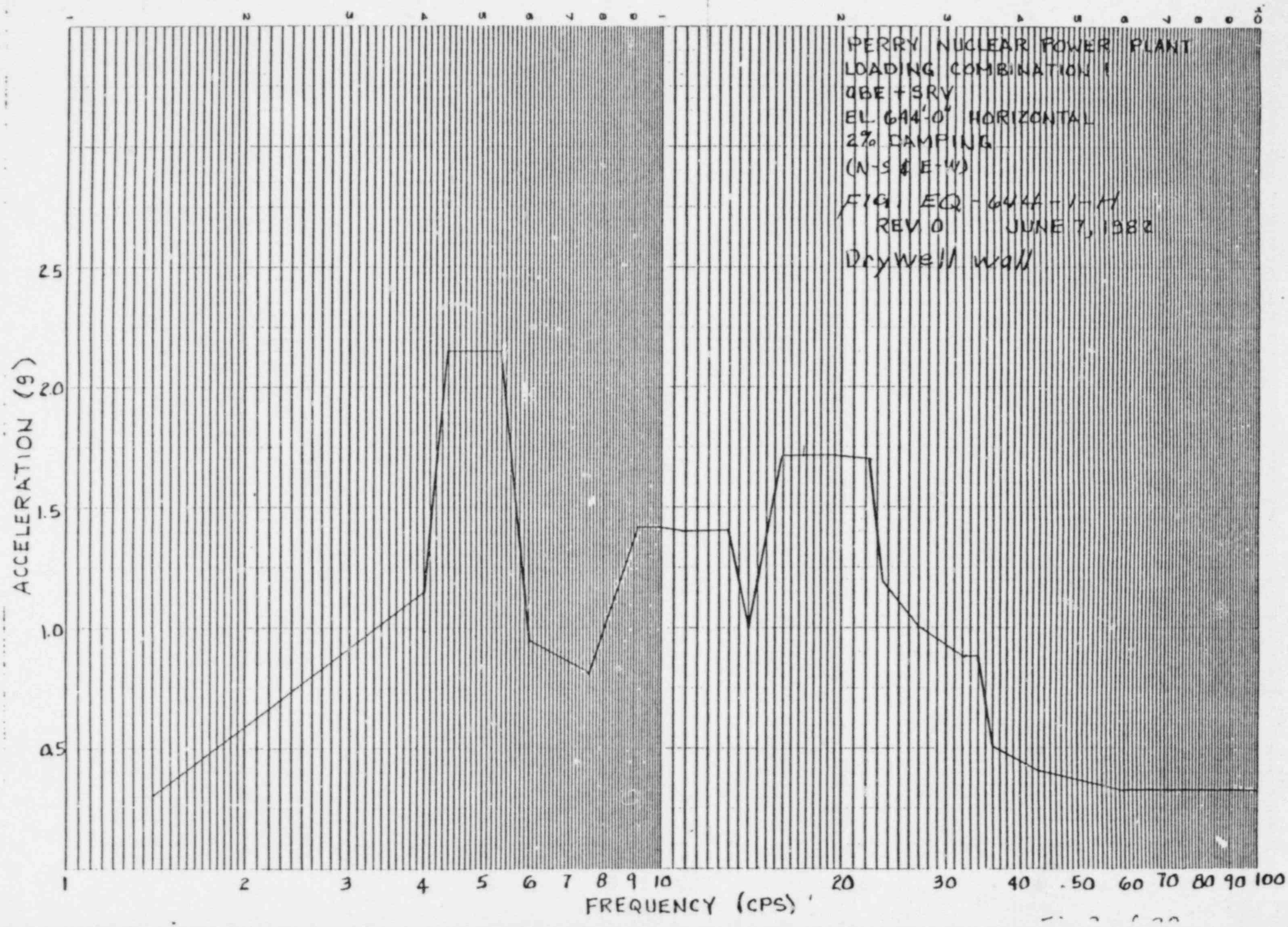
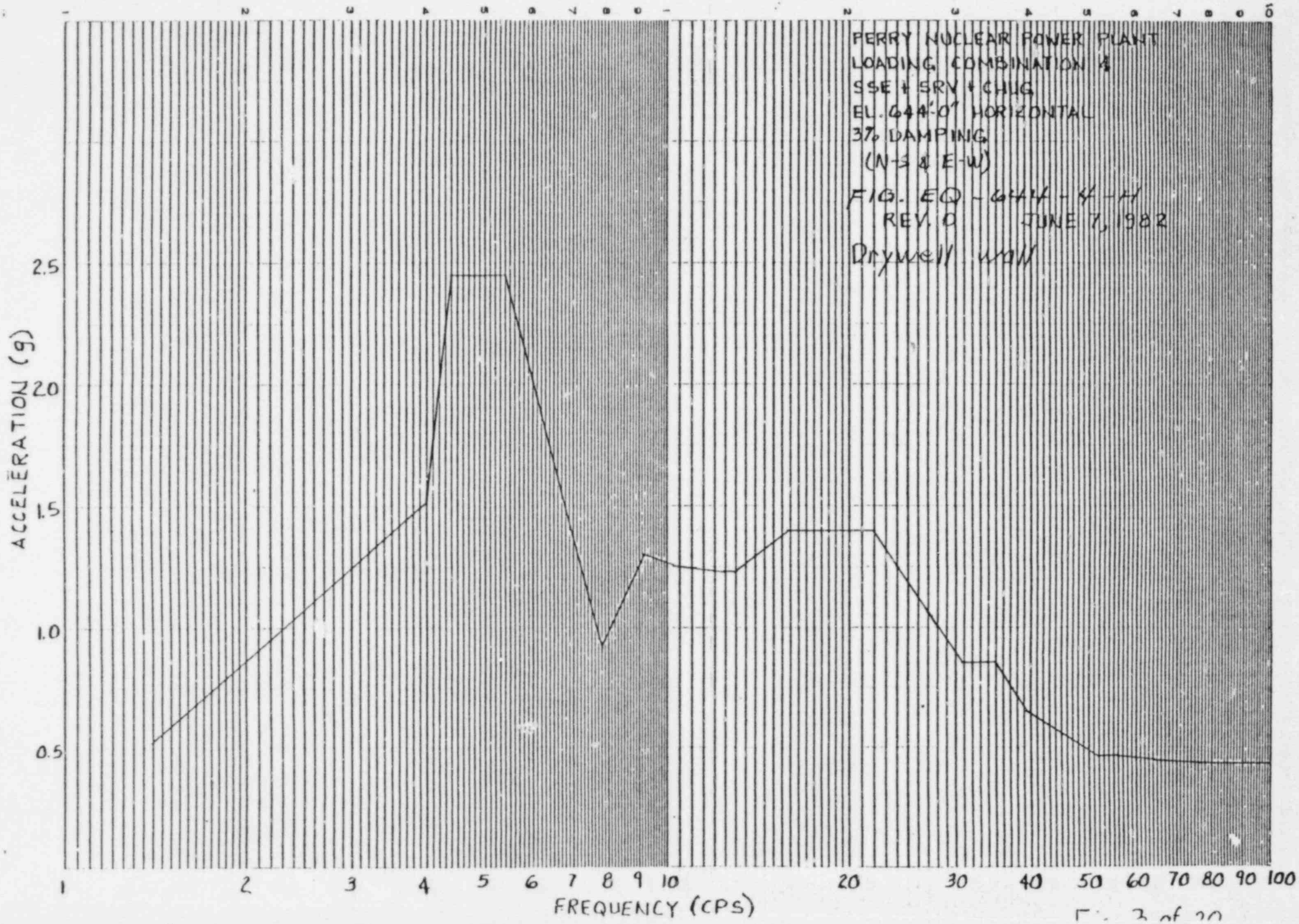
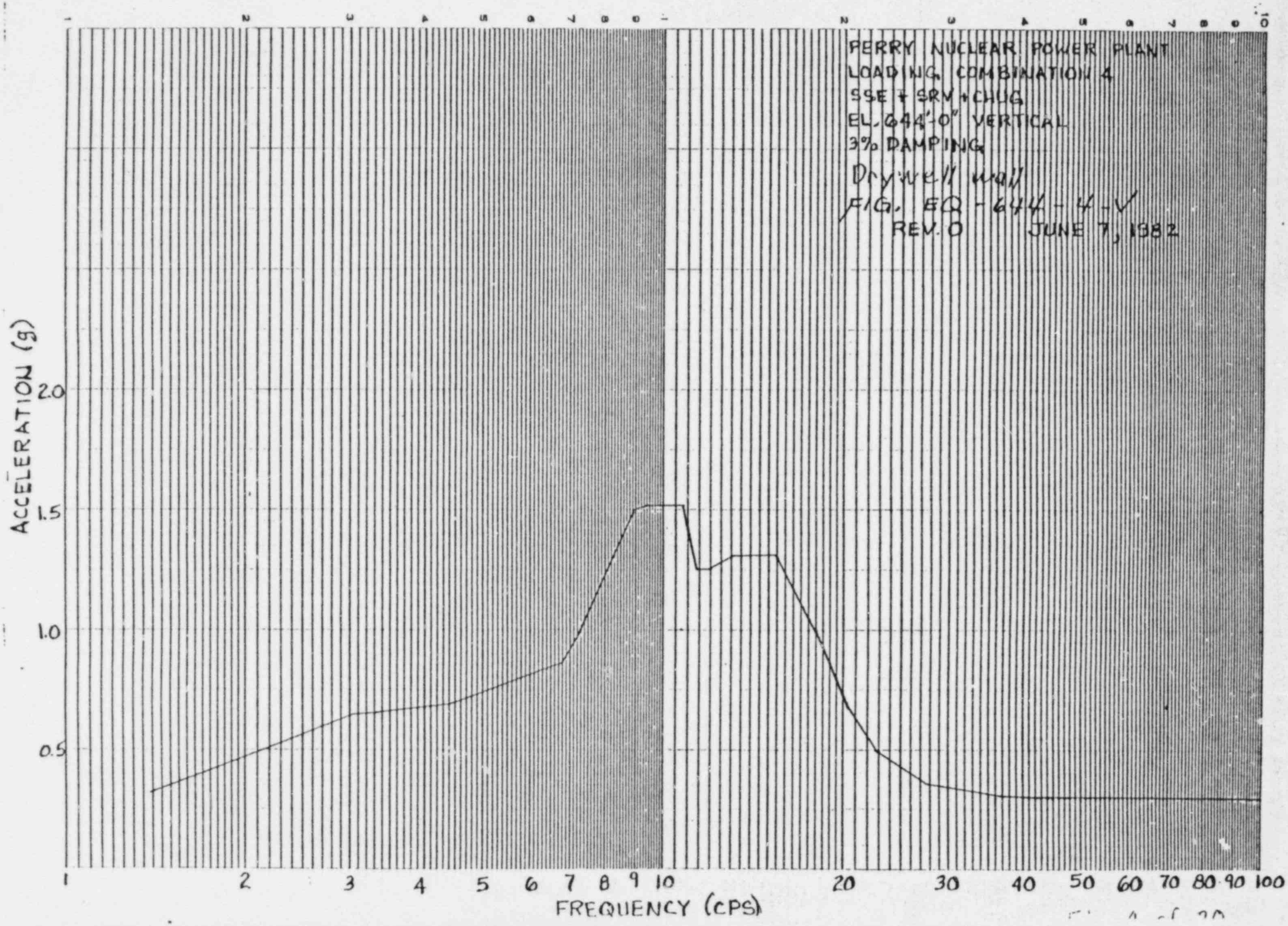



PERRY NUCLEAR POWER PLANT
LOADING COMBINATION 4
SSE + SRV + CHUG
EL. 644'-0" HORIZONTAL
3% DAMPING
(N-S & E-W)
FIG. EQ-644-4-H
REV. 0 JUNE 7, 1982
Drywell wall
ACCELERATION (g)
2.5
2.0
1.5
1.0
0.5
1
2
3
4
5
6
7
8
9
10
20
30
40
50
60
70
80
90
100
FREQUENCY (CPS)


PERRY NUCLEAR POWER PLANT
LOADING COMBINATION 4
SSE + SRV + CHUG
EL. 644'-0" VERTICAL
3% DAMPING
Drywell wall
FIG. EQ-644-4-V
REV. 0 JUNE 7, 1982
ACCELERATION (g)
2.0
1.5
1.0
0.5
1
2
3
4
5
6
7
8
9
10
20
30
40
50
60
70
80
90
100
FREQUENCY (CPS)

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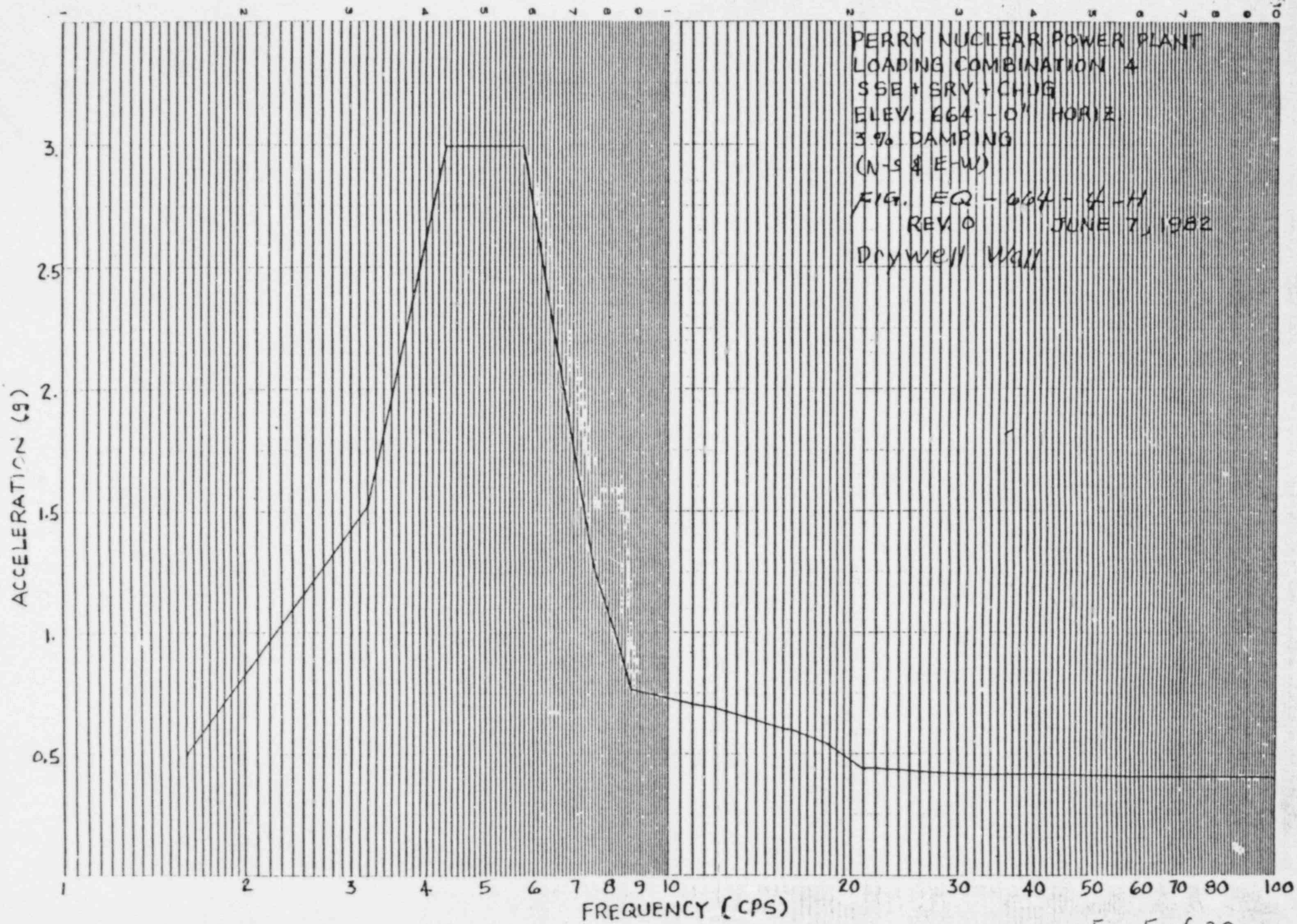

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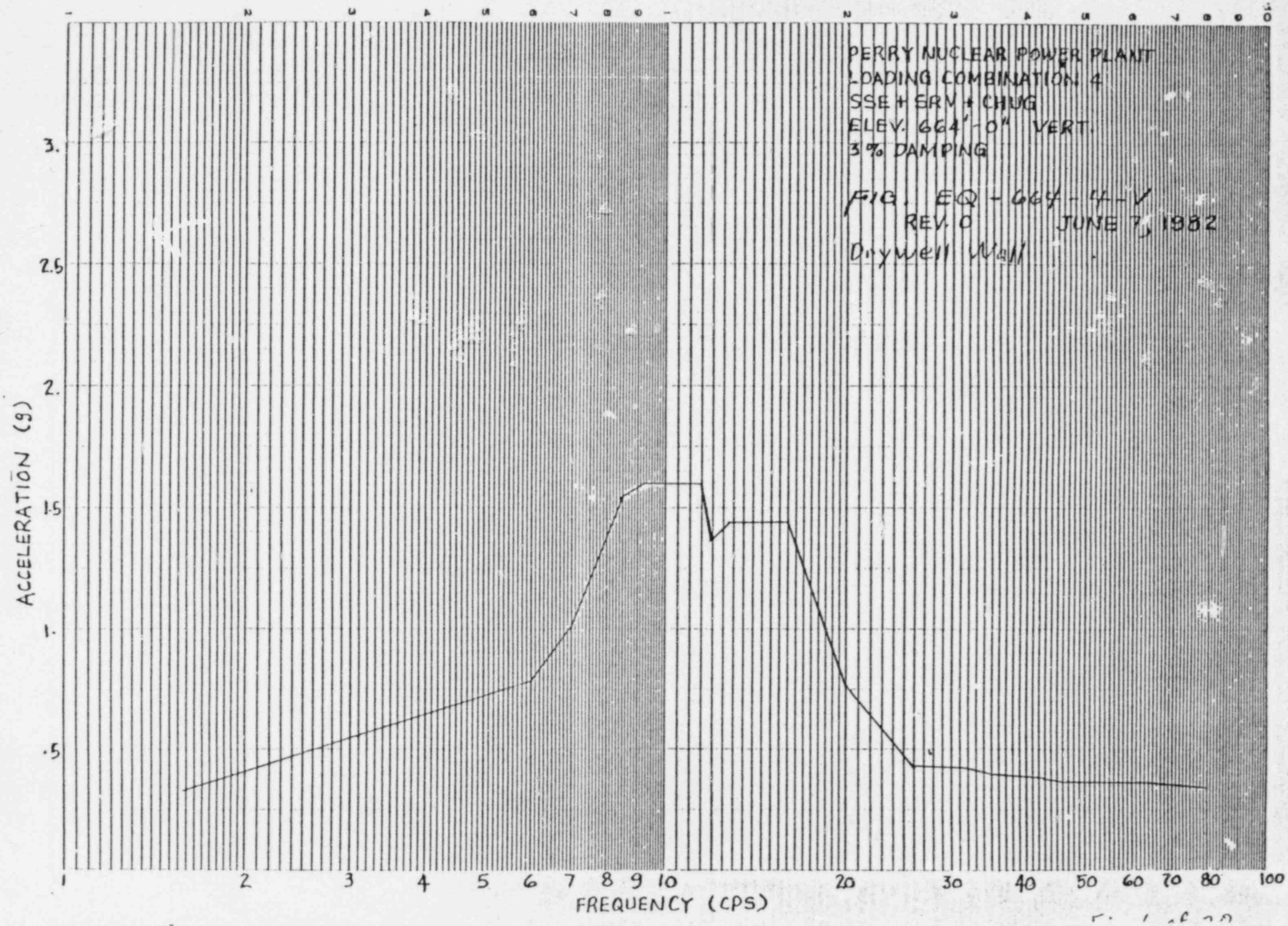


NO. 340-L210 DIETZGEN GRAPH PAPER
SEMI-LOGARITHMIC
2 CYCLES X 10 DIVISIONS PER INCH

DIETZGEN CORPORATION
MADE IN U.S.A.

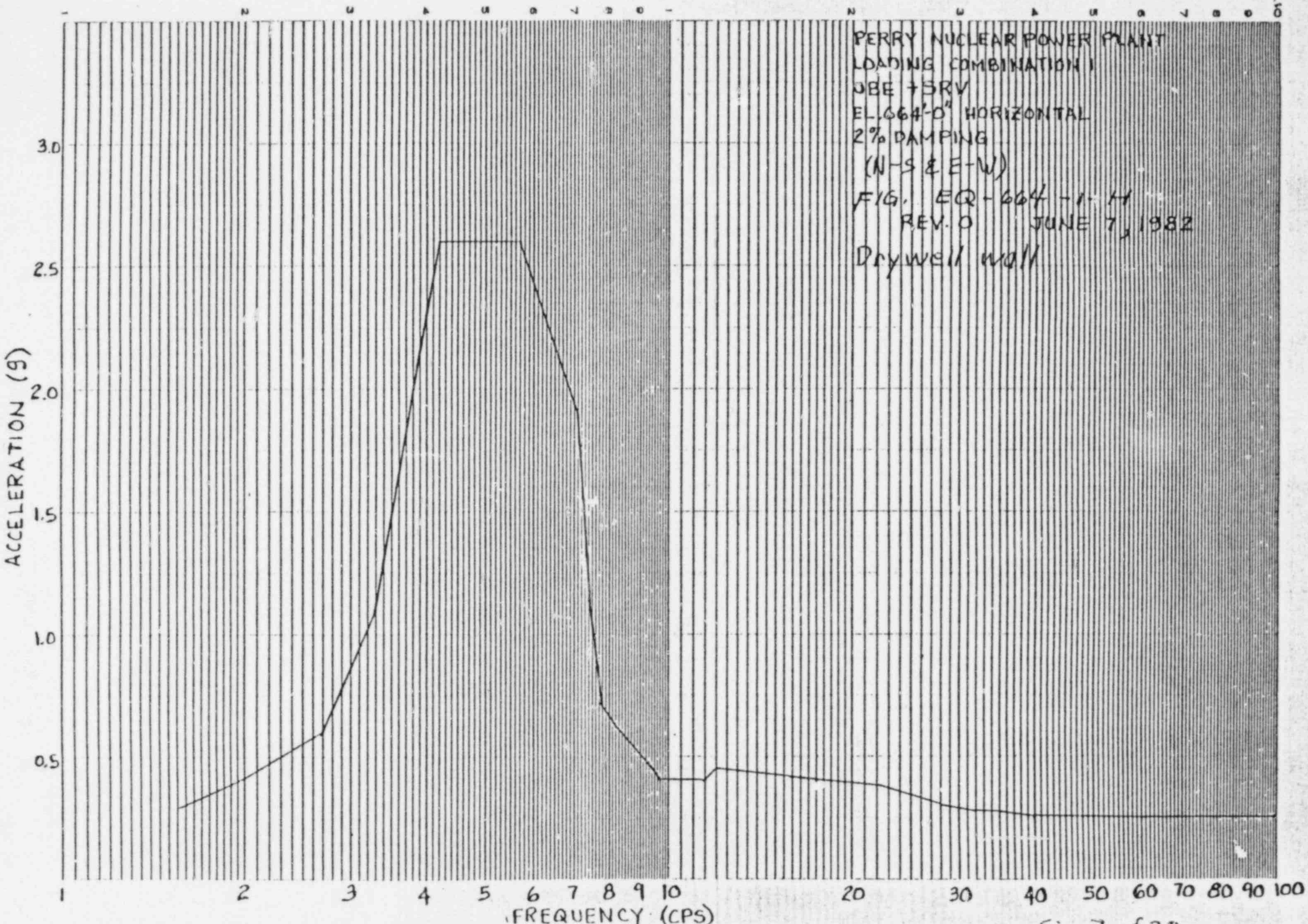


Fig. 7 of 20

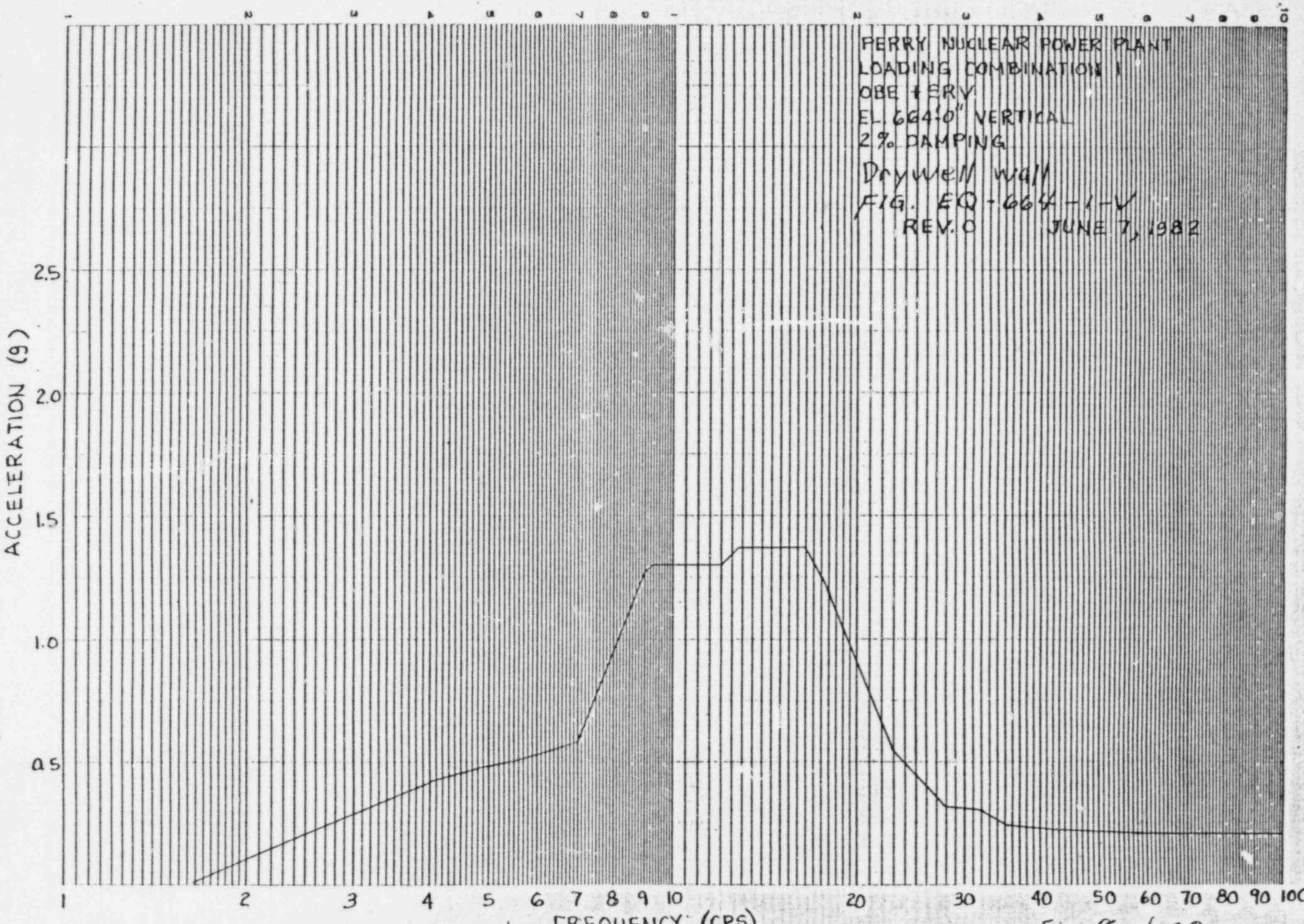
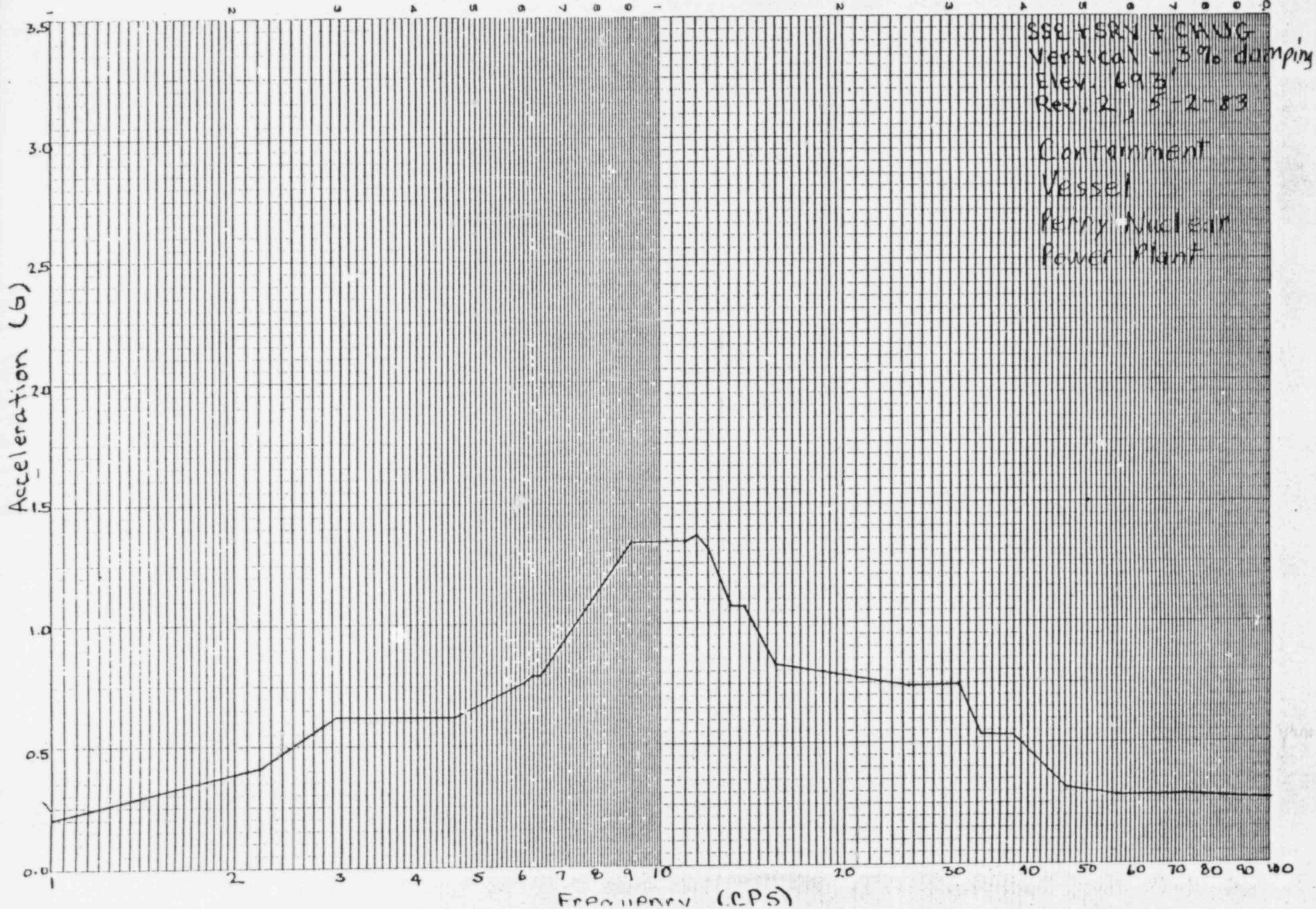

PERRY NUCLEAR POWER PLANT
LOADING COMBINATION I
OBE + SRV
EL. 664'-0" VERTICAL
2% DAMPING
Drywell wall
FIG. EQ-664-1-V
REV. 0
JUNE 7, 1982
ACCELERATION (g)
2.5
2.0
1.5
1.0
0.5
FREQUENCY (CPS)
1
2
3
4
5
6
7
8
9
10
20
30
40
50
60
70
80
90
100

Fig. 8 of 20

2 CYCLES X 10 DIVISIONS PER INCH

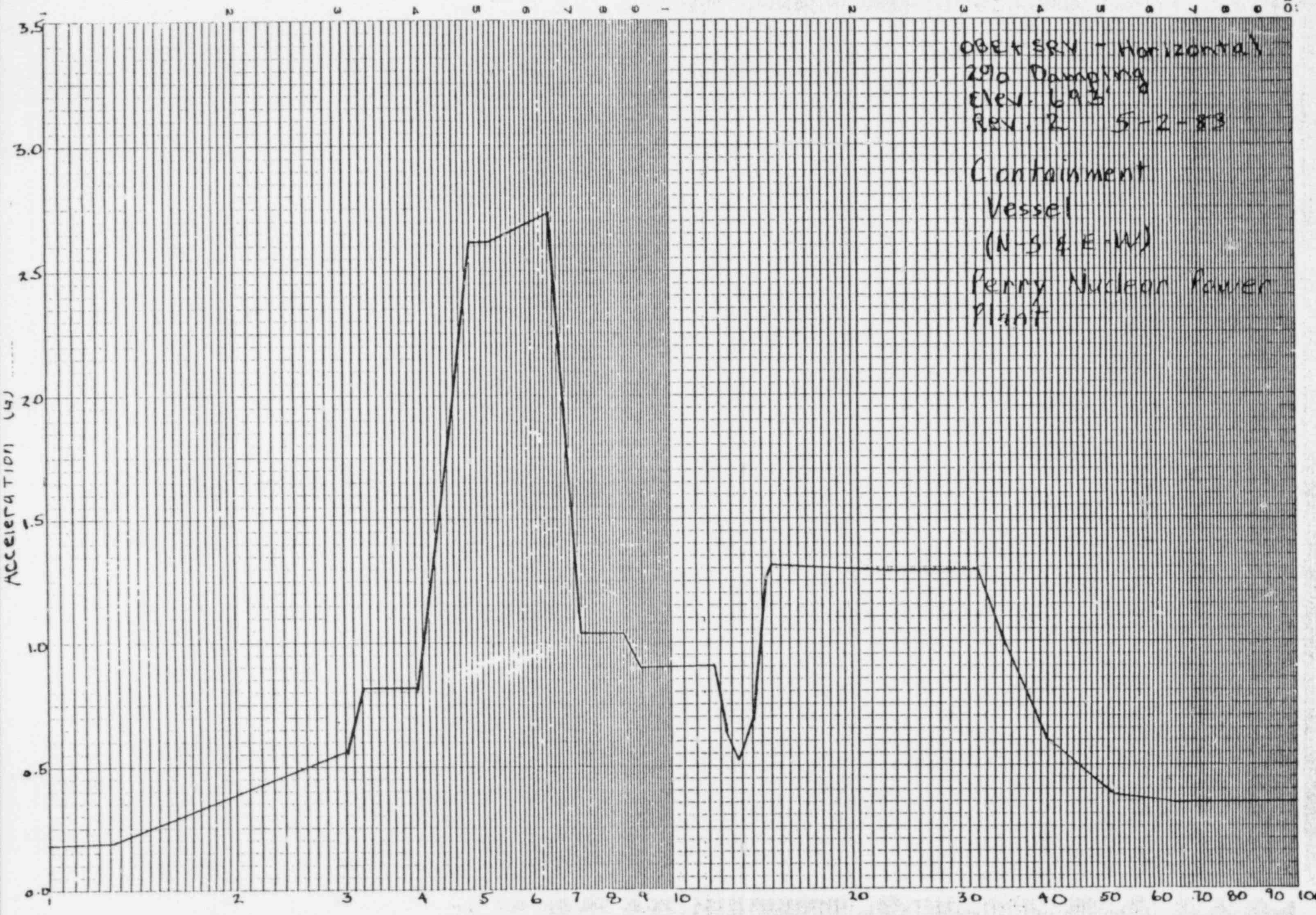
Fig. 9 of 20

Floor Response Spectra



Floor Response Spectra

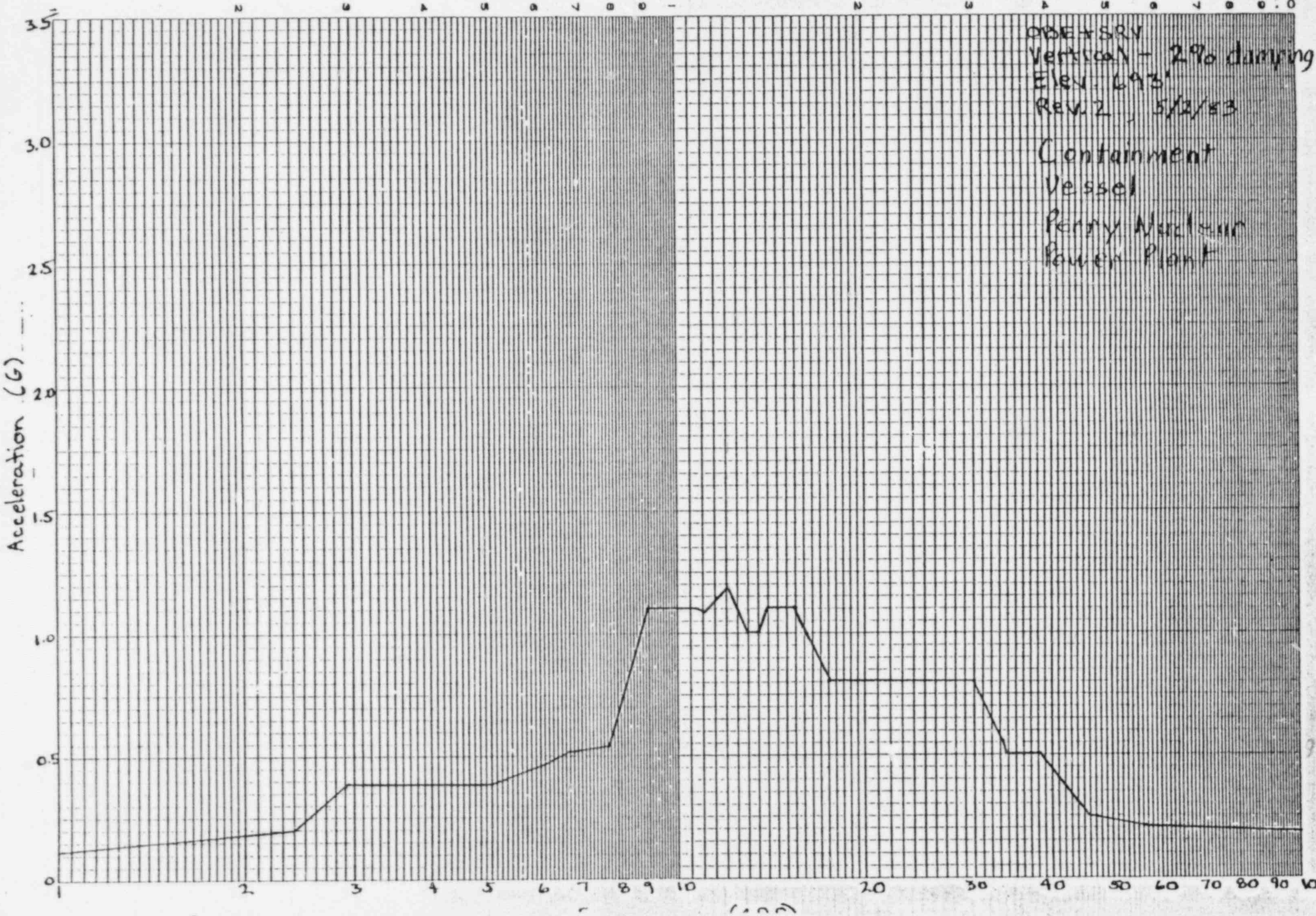
Fig. 10 of 20



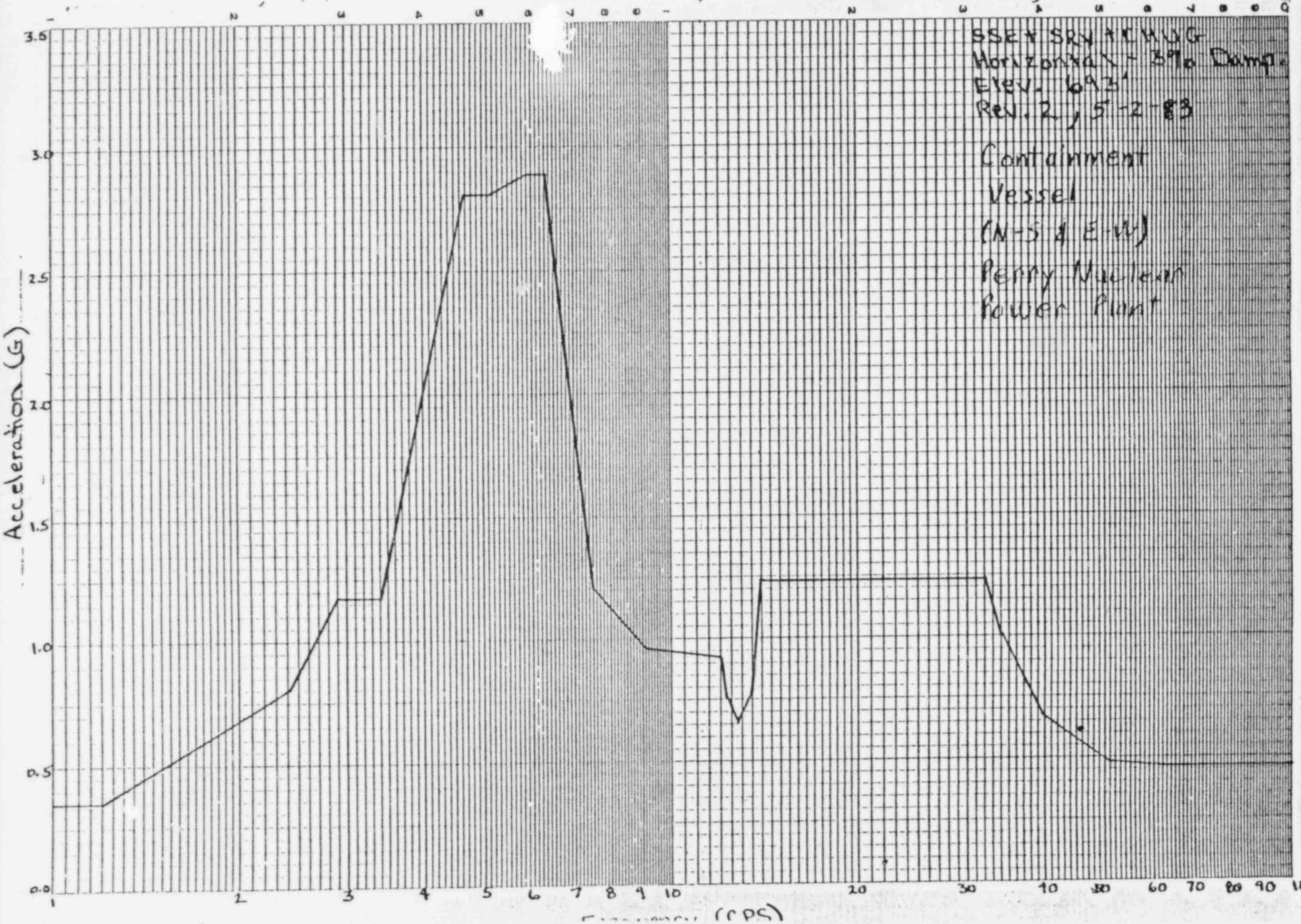
SEMI-LOGARITHMIC
2 CYCLES X 10 DIVISIONS PER INCH

Floor Response Spectra

Fig. 11 of 20

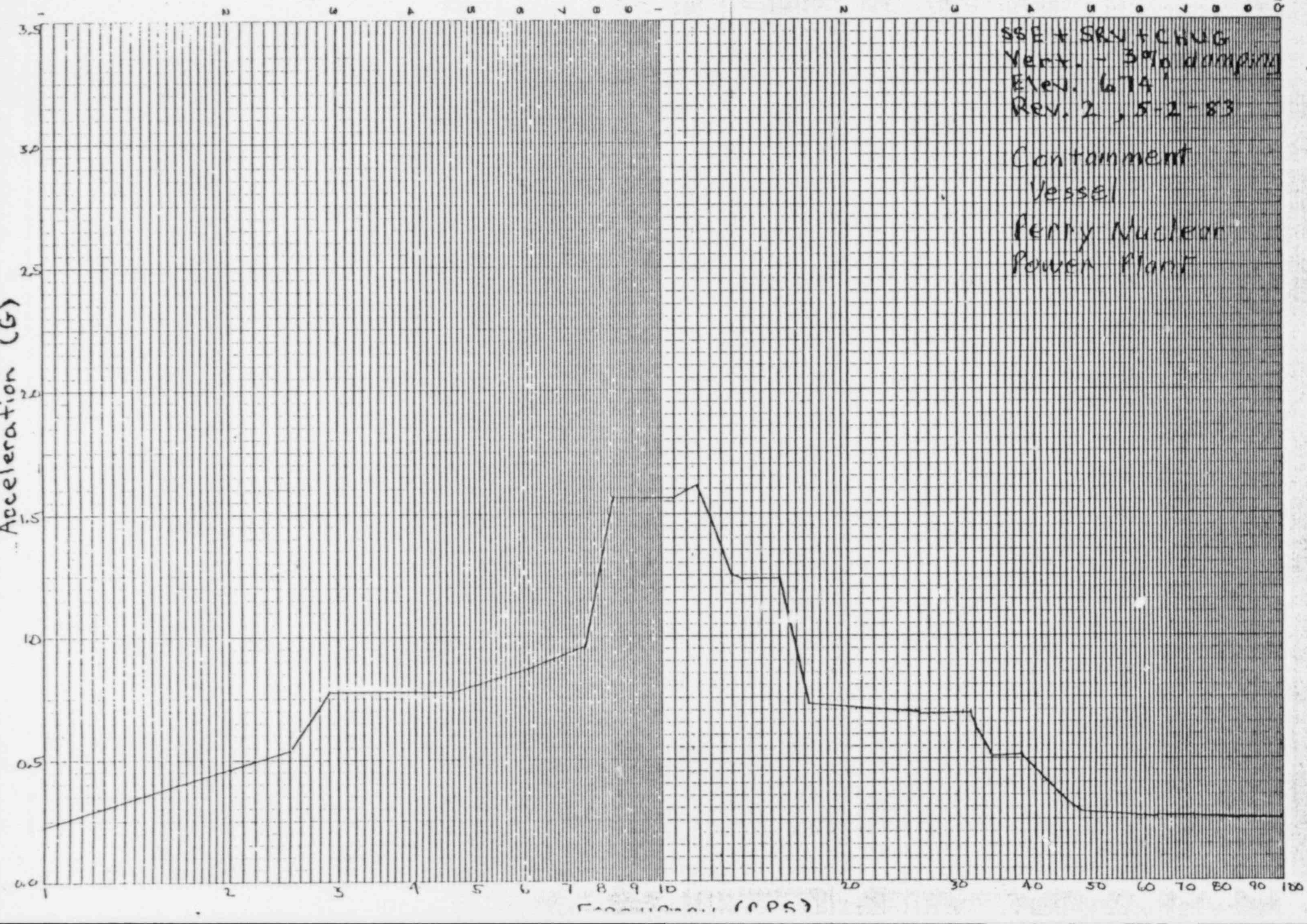


Floor Response Spectra Fig. 12 of 20



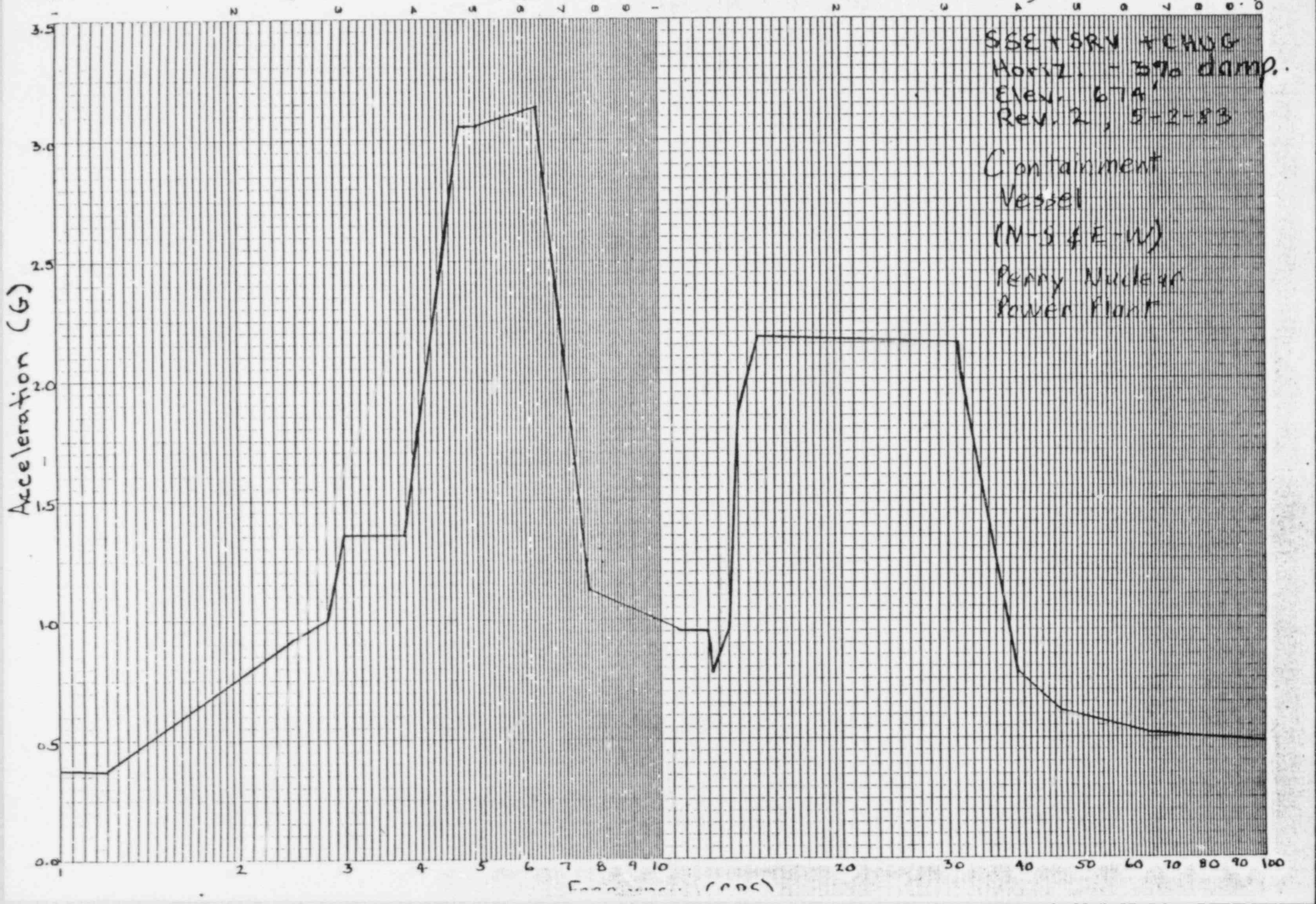
Floor Response Spectra

Fig. 13 of 20



Floor Response Spectra

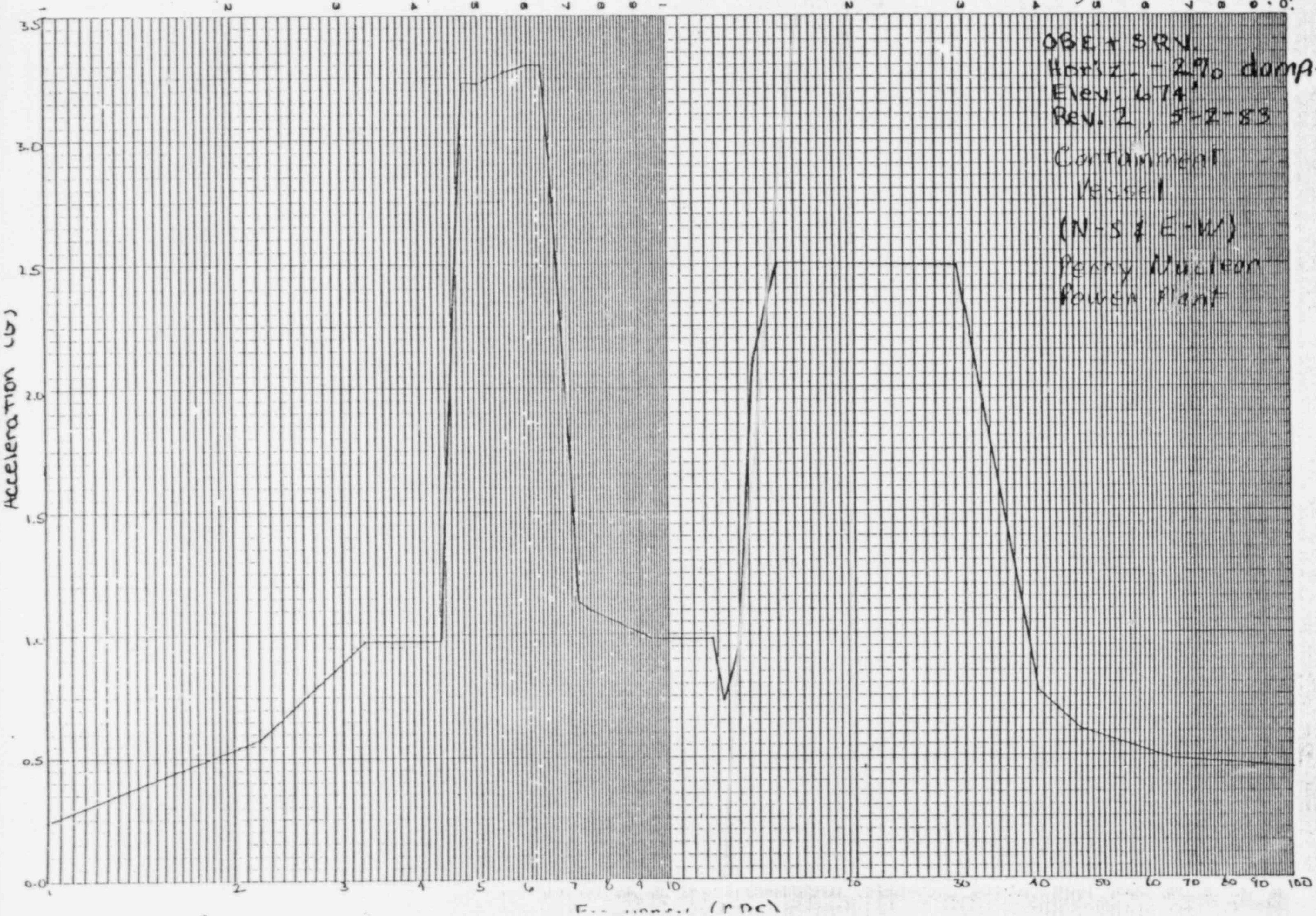
Fig. 14 of 20



2 CYCLES X 10 DIVISIONS PER INCH

Floor Response Spectra

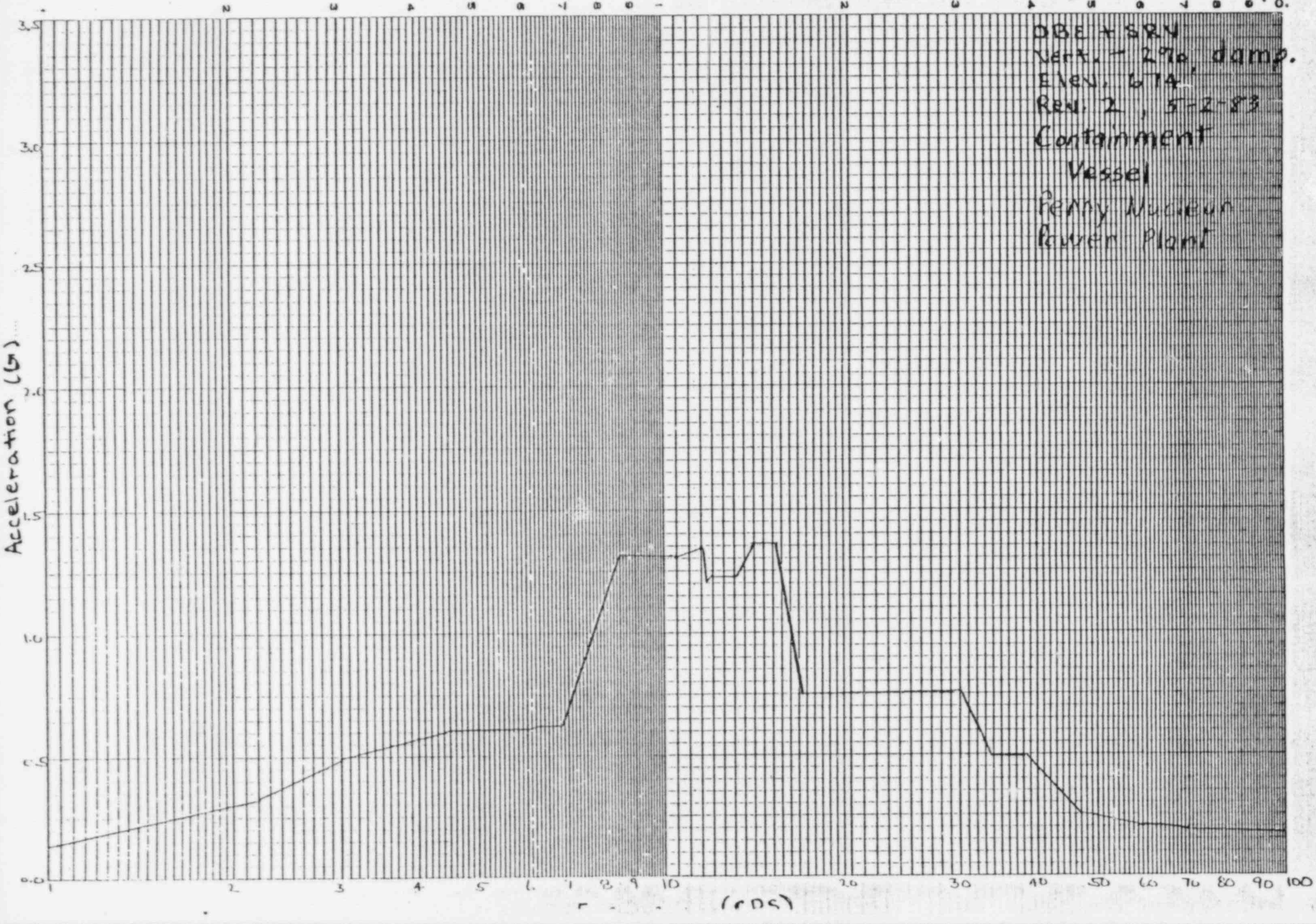
Fig. 15 of 20



2 CYCLES X 10 DIVISIONS PER INCH

Floor Response Spectra

Fig. 16 of 20



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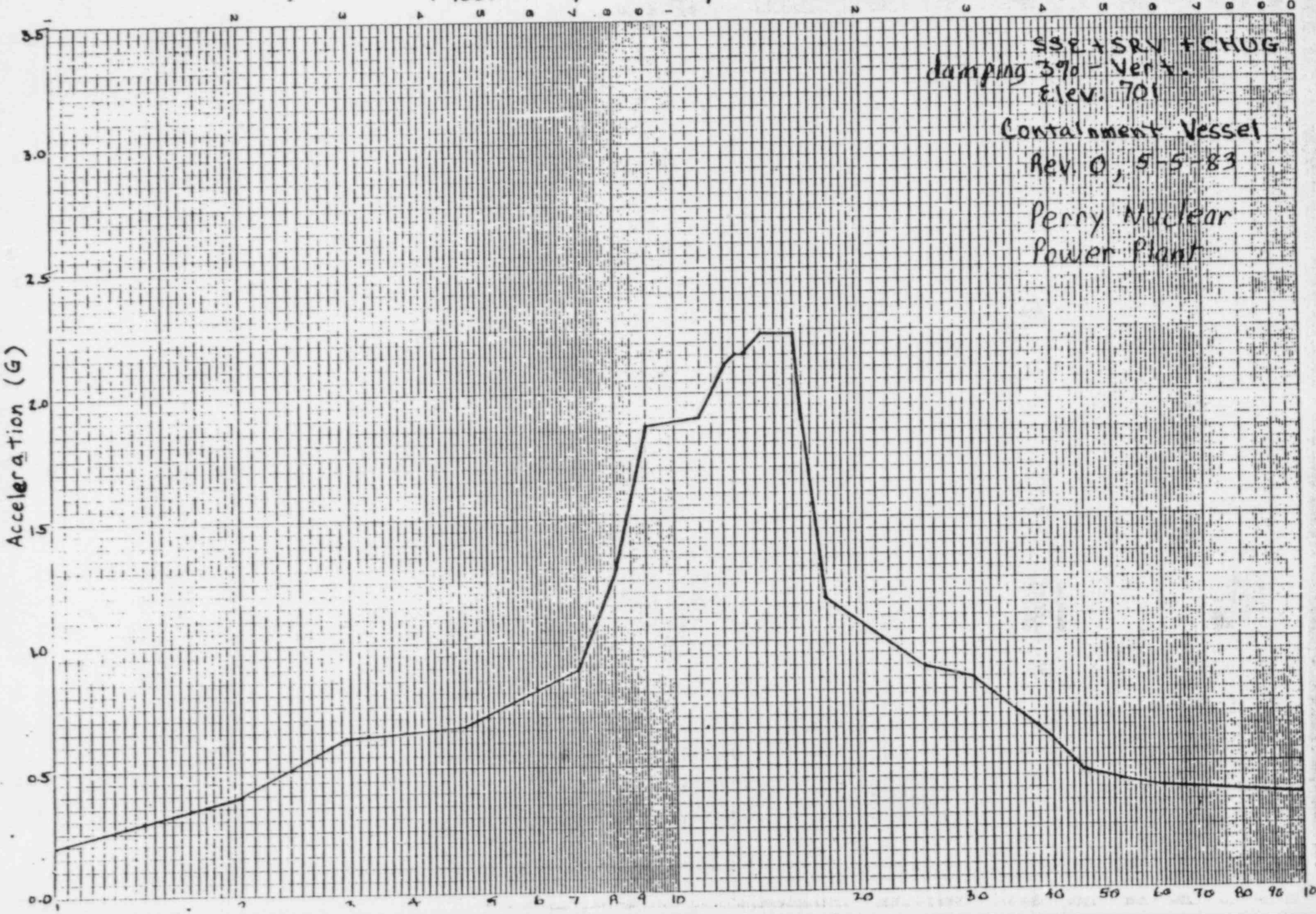
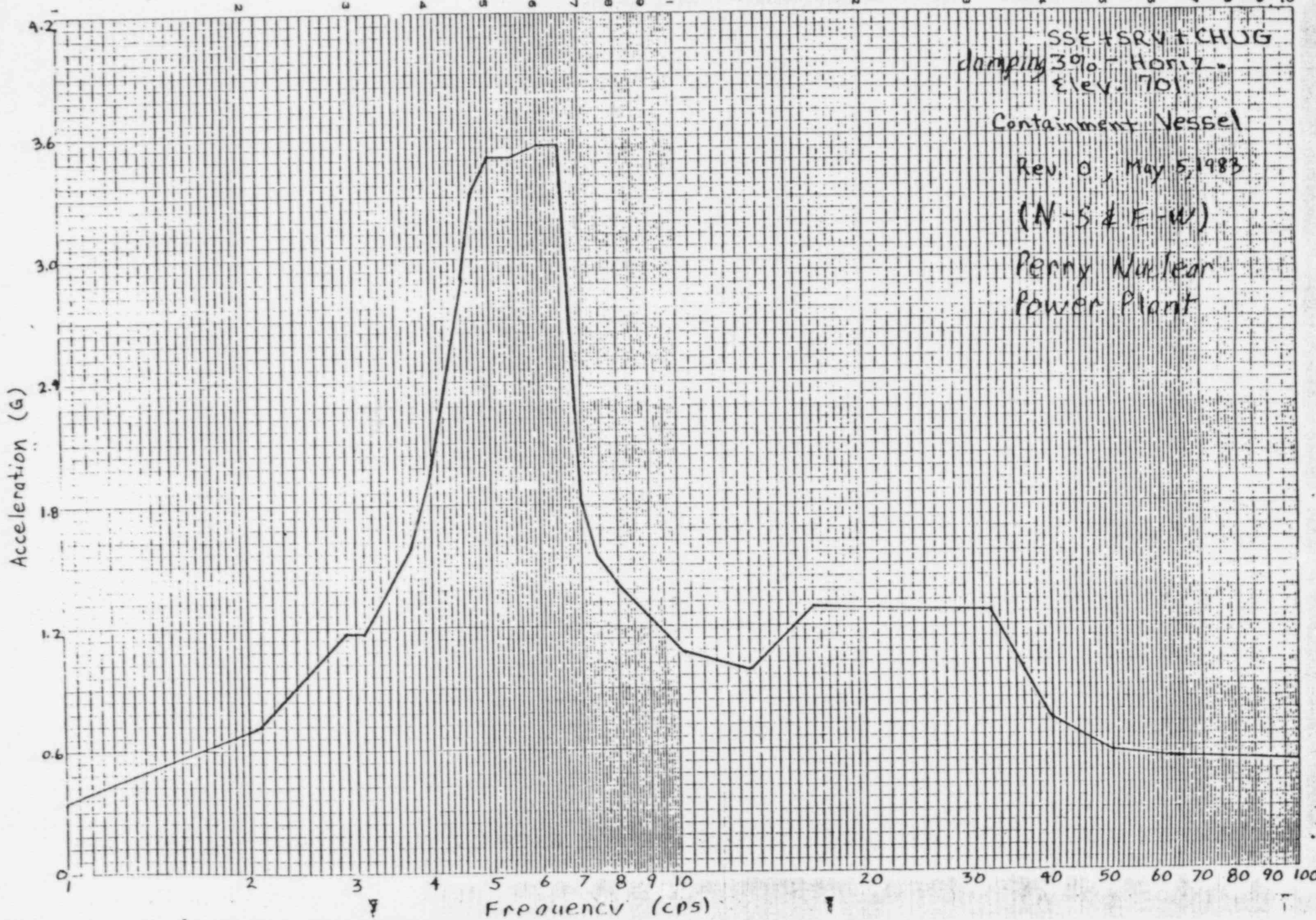
Floor Response Spectra

Fig. 17 of 20



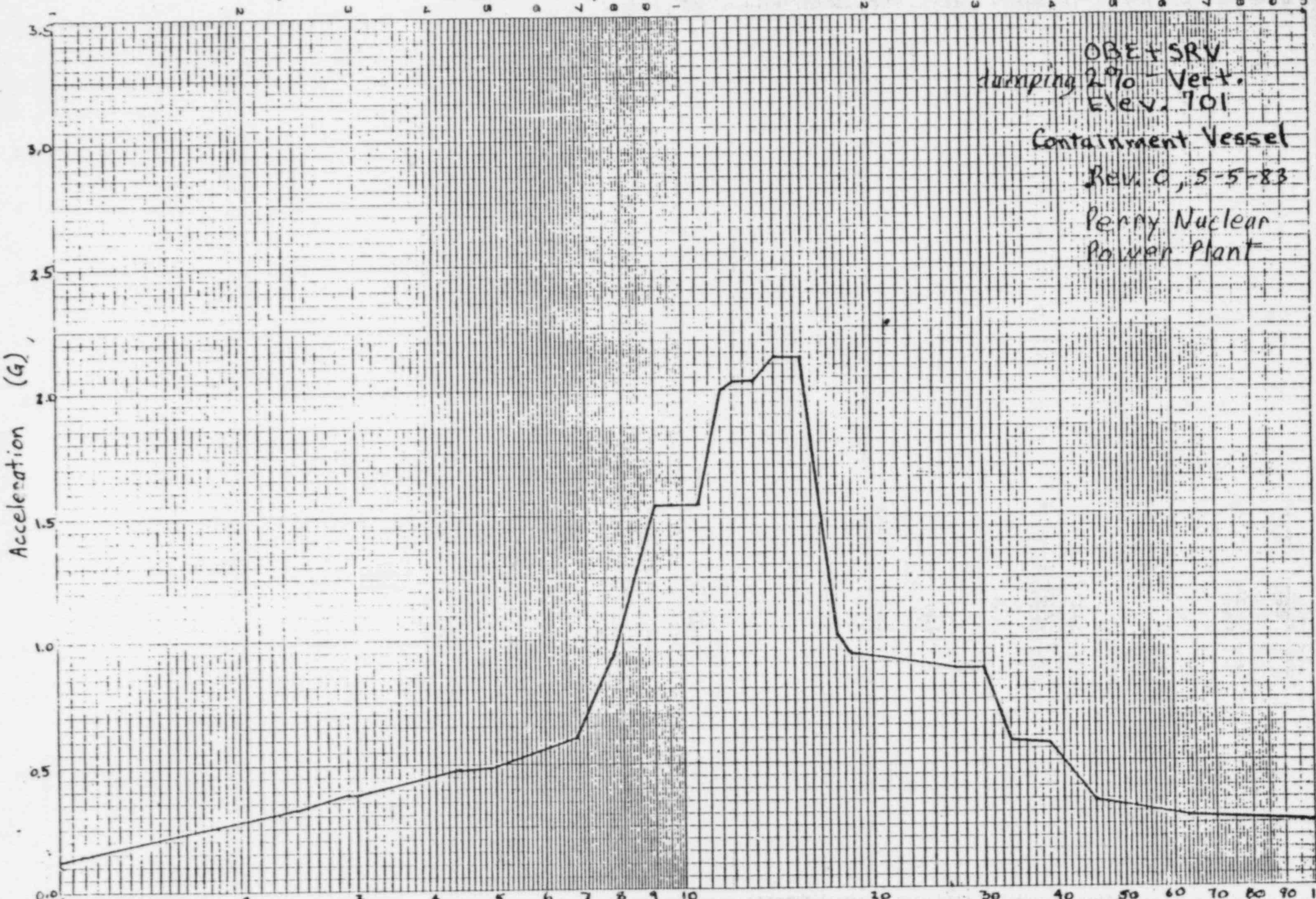
Floor Response Spectra

Fig. 18 of 20



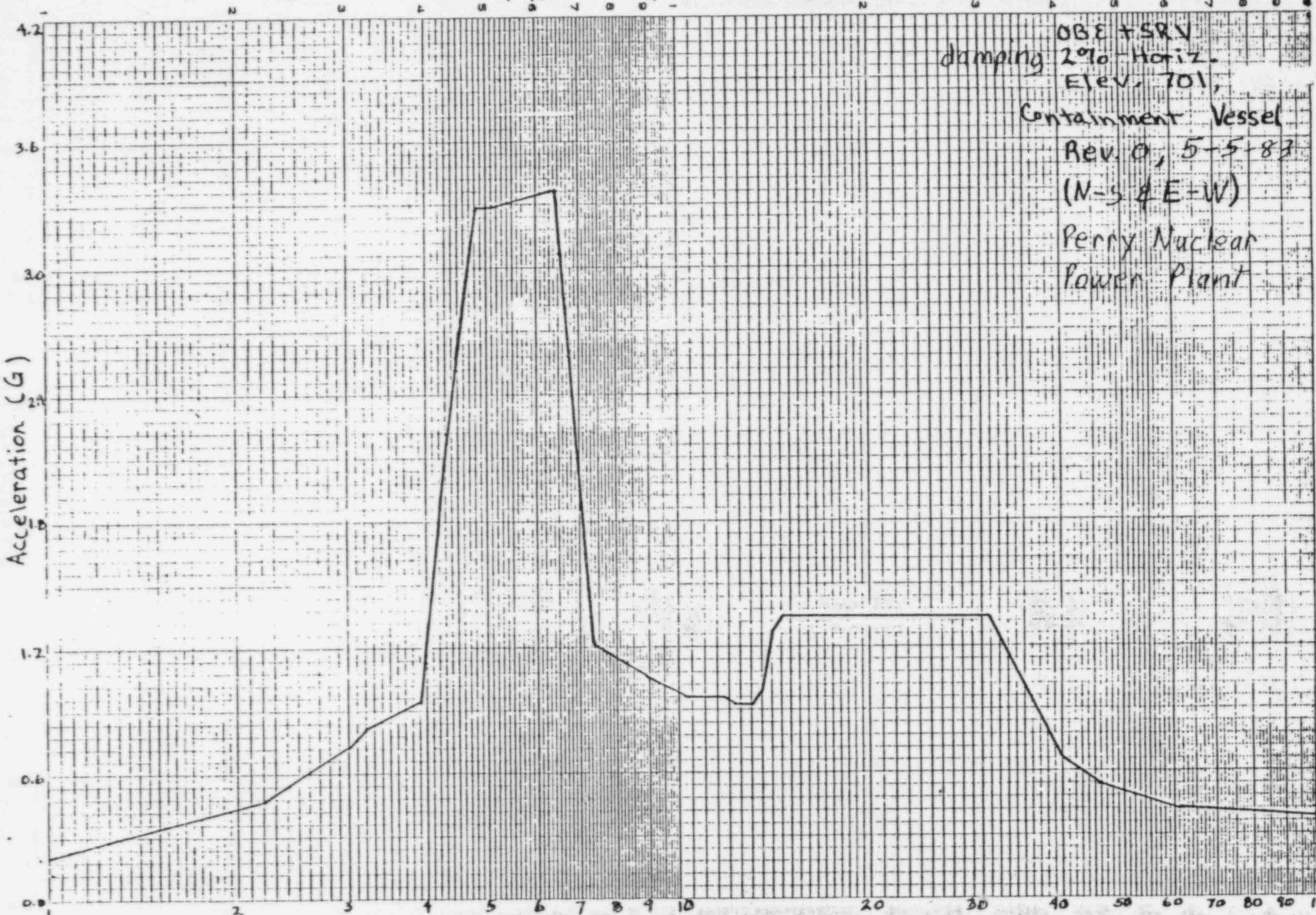
Floor Response Spectra

Fig. 19 of 20



Floor Response Spectra

Fig. 20 of 20



ATTACHMENT 2

Nuclear Purge Valve

Stress Analysis

SEISMIC ANALYSIS

FOR 18 INCH

NUCLEAR PURGE VALVE

TABLE OF CONTENTS

LIST OF FIGURES

NOMENCLATURE

The nomenclature for this analysis is based upon the nomenclature established in Paragraph NB-3534 of Section III of the ASME Boiler and Pressure Vessel Code. Where the nomenclature comes directly from the code, the reference paragraph or figure for that symbol is given with the definition. For symbols not defined in the code, the definition is that assigned by Henry Pratt Company for use in this analysis.

ANALYSIS NOMENCLATURE

A_f	Effective fluid pressure area based on fully corroded interior contour for calculating crotch primary membrane stress (NB-3545.1(a)), in^2
A_m	Metal area based on fully corroded interior contour effective in resisting fluid force on A_f (NB-3545.1 (a)), in^2
A_3	Tensile area of cover cap bolt, in^2
A_4	Shear area of cover cap bolt, in^2
A_5	Tensile area of trunnion bolt, in^2
A_6	Shear area of trunnion bolt, in^2
A_7	Tensile area of operator bolt, in^2
A_8	Shear area of operator bolt, in^2
B_1	Unsupported shaft length, in.
B_2	Bearing bore diameter, in.
B_3	Bonnet bolt tensile area, in^2
B_4	Bonnet bolt shear area, in^2
B_5	Bonnet body cross-sectional area, in^2
B_6	Top bonnet weld size, in.
B_7	Bottom bonnet weld size, in.
B_8	Distance to outer fiber of bonnet from shaft on y axis, in.
B_9	Distance to outer fiber of bonnet from shaft on x axis, in.
C	A factor depending upon the method of attachment of head, shell dimensions, and other items as listed in NC-3225.2, dimensionless (Fig. NC-3225.1 thru Fig. NC-3225.3)

C_b	Stress index for body bending secondary stress resulting from moment in connected pipe (NB-3545.2(b))
C_p	Stress index for body primary plus secondary stress, inside surface, resulting from internal pressure (NB-3545.2(a))
C_2	Stress index for thermal secondary membrane stress resulting from structural discontinuity
C_3	Stress index for maximum secondary membrane plus bending stress resulting from structural discontinuity
C_6	Product of Young's modulus and coefficient of linear thermal expansion, at 500°F, psi/°F (NB-3550)
C_7	Distance to outer fiber of disc for bending along the shaft, in.
C_8	Distance to outer fiber of disc for bending about the shaft, in.
C_9	Distance to outer fiber of flat plate of disc for bending of unsupported flat plate, in.
d	Inside diameter of body neck at crotch region (NB-3545.1(a)), in.
d_m	Inside diameter used as basis for determining body minimum wall thickness, (NB-3541), in.
D_1	Valve nominal diameter, in.
D_2	Shaft diameter, in.
D_3	Disc pin diameter, in.
D_4	Thrust collar outside diameter, in.
D_5	Spring pin diameter, in.
D_6	Cover cap bolt diameter, in.
D_7	Trunnion bolt diameter, in.

D_8	Operator bolt diameter, in.
D_9	Bonnet bolt diameter, in.
E	Modulus of elasticity, psi
F_b	Bending modulus of standard connected pipe, as given by Figs. NB-3545.2-4 and NB-3545.2-5, $in.^3$
F_d	1/2 x cross-sectional area of standard connected pipe, as given by Figs. NB-3545.2-2 and NB-3545.2-3, $in.^2$
F_N	Natural frequency of respective assembly, hertz
F_x	W_3g_x--Seismic force along x axis due to seismic acceleration acting on operator extended mass, pounds
F_y	W_3g_y--Seismic force along y axis due to seismic acceleration acting on operator extended mass, pounds
F_z	W_3g_z--Seismic force along z axis due to seismic acceleration acting on operator extended mass, pounds
g	Gravitational acceleration constant, inch-per-$second^2$
G_b	Valve body section bending modulus at crotch region (NB-3545.2(b)), in^3
G_d	Valve body section area at crotch region (NB-3545.2 (b)), in^2
G_t	Valve body section torsional modulus at crotch region (NB-3545.2(b)), in^3
g_x	Seismic acceleration constant along x axis
g_y	Seismic acceleration constant along y axis
g_z	Seismic acceleration constant along z axis
h_g	Gasket moment arm, equal to the radial distance from the center line of the bolts to the line of the gasket reaction (NC-3225), in.
H_2	Top trunnion bolt square, in.
H_3	Bottom trunnion bolt square, in.

ANALYSIS NOMENCLATURE

H_4	Bonnet bolt square, in.
H_5	Operator bolt square, in.
H_6	Bonnet bolt circle, in.
H_7	Operator bolt circle, in.
H_8	Bonnet height, in.
H_9	Actual body wall thickness, in.
I_1	Bonnet body moment of inertia about x axis, in^4
I_2	Bonnet body moment of inertia about y axis, in^4
I_3	Disc area moment of inertia for bending about the shaft, in^4
I_4	Disc area moment of inertia for bending along the shaft, in^4
I_5	Moment of inertia of valve body, in^4
I_6	Moment of inertia of shaft, in^4
I_7	Disc area moment of inertia for bending of unsupported flat plate, in^4
J_1	Distance to neutral bending axis for top trunnion bolt pattern along x axis, in.
J_2	Distance to neutral bending axis for top trunnion bolt pattern along y axis, in.
J_3	Distance to neutral bending axis for bonnet bolt pattern along x axis, in.
J_4	Distance to neutral bending axis for bonnet bolt pattern along y axis, in.
J_5	Distance to neutral bending axis for operator bolt pattern along x axis, in.
J_6	Distance to neutral bending axis for operator bolt pattern along y axis, in.
K	Spring constant

ANALYSIS NOMENCLATURE

K_1	Distance of bonnet leg from shaft centerline, in.
K_2	Thickness of disc above shaft, in.
K_3	Length along z axis to c.g. of bonnet plus adapter plate assembly, in.
K_4	Top trunnion width, in.
K_5	Top trunnion depth, in.
K_6	Height of top trunnion, in.
L_1	Valve body face-to-face dimension, in.
L_2	Thickness of operator housing under trunnion bolt, in.
L_3	Length of engagement of cover cap bolts in bottom trunnion, in.
L_4	Length of engagement of trunnion bolts in top trunnion, in.
L_5	Bearing length, in.
L_6	Length of structural disc hub welds, in.
L_7	Length of engagement of bonnet bolts in adapter plate, in.
L_8	Length of engagement of bonnet bolts in bonnet, in.
L_9	Length of engagement of stub shaft in disc, in.
m	Reciprocal of Poisson's ratio
M	Mass of component
Mx	$W_3(g_yZ_o+g_zY_o)$, operator extended mass seismic bending moment about the x axis, acting at the base of the operator, in-lbs.
My	$W_3(g_xZ_o+g_zX_o)$, operator extended mass seismic bending moment about the y axis, acting at the base of the operator, in-lbs.

ANALYSIS NOMENCLATURE

M_z	$W_3(g_xY_o+g_yX_o)$, operator extended mass seismic bending moment about the z axis, in-lbs.
$\overline{M_x}$	$M_x+F_yT_5$, operator extended mass seismic bending moment about the x axis, acting at the bottom of the adapter plate, in-lbs.
$\overline{M_y}$	$M_y+F_xT_5$, operator extended mass seismic bending moment about the y axis, acting at the bottom of the adapter plate, in-lbs.
$\overline{\overline{M_x}}$	$M_x+F_x(T_5+H_8)+g_xW_4K_3$, operator extended mass seismic bending moment about the x axis, acting at the base of the bonnet, in-lbs.
$\overline{\overline{M_y}}$	$M_y+F_x(T_5+H_8)+g_xW_4K_3$, operator extended mass seismic bending moment about the y axis, acting at the base of the bonnet, in-lbs.
M_8	Bending moment at joint of flat plate to disc hub, in-lbs.
N_a	Permissible number of complete start-up/shut-down cycles at hr/100°F/hr/hr fluid temperature change rate (NB-3545.3)
NA	Not applicable to the analysis of the system
N_1	Number of top disc pins
N_2	Number of operator bolts
N_3	Number of trunnion bolts
P_d	Design pressure, psi
P_r	Primary pressure rating, pound
P_s	Standard calculation pressure from Fig. NB-3545.1-1, psi
P_e	Largest value among Peb, Ped, Pet, psi
Peb	Secondary stress in crotch region of valve body caused by bending of connected standard pipe, calculated according to NB-3545.2(b), psi

ANALYSIS NOMENCLATURE

Ped	Secondary stress in crotch region of valve body caused by direct or axial load imposed by connected standard piping, calculated according to NB-3545.2 (b), psi
Pet	Secondary stress in crotch region of valve body caused by twisting of connected standard pipe, calculated according to NB-3545.2(b), psi
P_m	General primary membrane stress intensity at crotch region, calculated according to NB-3545.1(a), psi
Pm'	Primary membrane stress intensity in body wall, psi
Q_p	Sum of primary plus secondary stresses at crotch resulting from internal pressure, (NB-3545.2(a)), psi
Q_T	Thermal stress in crotch region resulting from 100°F/hr fluid temperature change rate, psi
Q_{T1}	Maximum thermal stress component caused by through wall temperature gradient associated with 100°F/hr fluid temperature change rate (NB-3545.2(c)), psi
Q_{T2}	Maximum thermal secondary membrane stress resulting from 100°F/hr fluid temperature change rate, psi
Q_{T3}	Maximum thermal secondary membrane plus bending stress resulting from structural discontinuity and 100°F/hr fluid temperature change rate, psi
r	Mean radius of body wall at crotch region (Fig. NB-3545.2(c)-1), in.
r_i	Inside radius of body at crotch region for calculating Q_p (NB-3545.2(a)), in.
r_2	Fillet radius of external surface at crotch (NB-3545.2(a)), in.
R_4	Disc radius, in.
R_5	Shaft radius, in.
R_m	Mean radius of body wall, in.
R_6	Radius to O-ring in cover cap, in.

ANALYSIS NOMENCLATURE

S	Assumed maximum stress in connected pipe for calculating P_e (NB-3545.2(b)), 30,000 psi
S_m	Design stress intensity, (NB-3533), psi
S_n	Sum of primary plus secondary stress intensities at crotch region resulting from 100°F/hr temperature change rate (NB-3545.2), psi
S_{p1}	Fatigue stress intensity at inside surface in crotch region resulting from 100°F/hr fluid temperature change rate (NB-3543.3), psi
S_{p2}	Fatigue stress intensity at outside surface in crotch region resulting from 100°F/hr fluid temperature change rate (NB-3545.3), psi

S(1) through S(74) are listed after the alphabetical section.

t_e	Minimum body wall thickness adjacent to crotch for calculating thermal stresses (Fig. NB-3545.2(c)-1), in.
t_m	Minimum body wall thickness as determined by NB-3541, in.
T_e	Maximum effective metal thickness in crotch region for calculating thermal stresses, (Fig. NB-3545.2 (c)-1), in.
ΔT_2	Maximum magnitude of the difference in average wall temperatures for walls of thicknesses t_e, T_e, resulting from 100°F/hr fluid temperature change rate, °F
T_1	Thickness of cover cap behind bolt head, in.
T_2	Thickness of shaft behind spring pin, in.
T_3	Thrust collar thickness, in.
T_4	Cover cap thickness, in.
T_5	Adapter plate thickness, in.

ANALYSIS NOMENCLATURE

T_6	Thickness of bottom bonnet plate, in.
T_7	Thickness of top bonnet plate, in.
T_8	Maximum required operating torque for valve, in-lbs.
U_1	Area of bottom bonnet weld, in^2
U_2	Area of top bonnet weld, in^2
U_3	Shaft bearing coefficient of friction
U_4	Bearing friction torque due to pressure loading (shaft journal bearings)
U_5	Bearing friction torque due to pressure loading plus seismic loading (shaft journal bearings)
U_6	Thrust bearing friction torque
V_1	Distances to bolts in bolt pattern on adapter plate, in.
V_2	Distances to bolts in bolt pattern on adapter plate, in.
V_3	Distances to bolts in bolt pattern on adapter plate, in.
V_4	Distances to bolts in bolt pattern on adapter plate, in.
V_5	Distance to bolts in bolt pattern on bonnet, in.
V_6	Distance to bolts in bolt pattern on bonnet, in.
V_7	Distance to bolts in bolt pattern on bonnet, in.
V_8	Distance to bolts in bolt pattern on bonnet, in.
W	Total bolt load, pounds
W_1	Valve weight,pounds
W_2	Banjo weight,pounds
W_3	Operator weight, pounds
W_4	Bonnet and adapter plate assembly weight, pounds
W_6	Weld size of disc structural welds, in.

W_7	Weight of disc, pounds
W_8	Length of weld around perimeter of bonnet, in.
Xo	Eccentricity of center of gravity of operator extended mass along x axis, in.
Yo	Eccentricity of center of gravity of operator extended mass along y axis, in.
Zo	Eccentricity of center of gravity of operator extended mass along z axis, in.
Z_1	Bending section modulus of bottom bonnet welds, in^3
Z_2	Bending section modulus of top bonnet welds, in^3
Z_3	Torsional section modulus of bottom bonnet welds, in^3
Z_4	Torsional section modulus of top bonnet welds, in^3
Z_7	Distance to edge of disc hub, inches
Δy	Maximum deflection of component, inches

ANALYSIS NOMENCLATURE

S(1)	Combined bending stress in disc, psi
S(2)	Bending stress in disc due to bending along the shaft, psi
S(3)	Bending stress in disc due to bending about the shaft, psi
S(4)	Bending tensile stress in unsupported flat plate, psi
S(5)	Shear tear out of shaft through disc, psi
S(6)	Shear stress across structural hub welds of disc, psi
S(7)	Combined stress in shaft, psi
S(8)	Combined bending stress in shaft, psi
S(9)	Combined shear stress in shaft, psi
S(10)	Bending stress in shaft due to seismic and pressure loads along x axis, psi
S(11)	Bending stress in shaft due to seismic load along y axis, psi
S(12)	Torsional shear stress in top shaft due to operating torque, psi
S(13)	Direct shear stress in shaft due to seismic and pressure loads, psi
S(14)	Torsional shear stress at reduced disc pin cross-section, psi
S(15)	Shear stress across top disc pin due to operating torque, psi
S(16)	Bearing stress on top disc pin, psi
S(17)	Combined shear stress across bottom disc pin, psi
S(18)	Shear stress across bottom disc pin due to torsional load, psi
S(19)	Shear stress across bottom disc pin due to seismic load, psi

ANALYSIS NOMENCLATURE

S(20) Compressive stress on shaft bearing due to seismic and pressure loads, psi

S(21) Shear tear out of cover cap bolts through tapped holes in bottom trunnion, psi

S(22) Shear tear out of cover cap bolt head through bottom cover cap, psi

S(23) Combined stress in cover cap bolts, psi

S(24) Direct tensile stress in cover cap bolts, psi

S(25) Shear stress in cover cap bolts due to torsional loads, psi

S(26) Combined stress in cover cap, psi

S(27) Radial stress in cover cap, psi

S(28) Tangential stress in cover cap, psi

S(29) Shear stress in cover cap, psi

S(30) Bearing stress on thrust collar, psi

S(31) Shear load on thrust collar spring pin, pounds

S(32) Bearing stress of spring pin on thrust collar, psi

S(33) Shear tear out of spring pin through thrust collar, psi

S(34) Shear tear out of spring pin through bottom shaft, psi

S(35) Shear tear out of trunnion bolt through tapped hole in trunnion, psi

S(36) Bearing stress of trunnion bolt on tapped hole in trunnion, psi

S(37) Bearing stress of trunnion bolt on through hole in bonnet plate, psi

S(38) Shear tear out of trunnion bolt head through bonnet plate, psi

S(39) Combined stress in trunnion bolt, psi

S(40)	Direct tensile stress in trunnion bolt, psi
S(41)	Tensile stress in trunnion bolt due to bending moment, psi
S(42)	Direct shear stress in trunnion bolt, psi
S(43)	Shear stress in trunnion bolt due to torsional load, psi
S(44)	Shear tear out of operator bolt head through hole in bonnet, psi
S(45)	Bearing stress of operator bolt on through hole in bonnet, psi
S(46)	Combined stress in operator bolts, psi
S(47)	Direct tensile stress in operator bolts, psi
S(48)	Tensile stress in operator bolts due to bending moment, psi
S(49)	Direct shear stress in operator bolts, psi
S(50)	Shear stress in operator bolts due to torsional loads, psi
S(51)	Combined stress in bonnet body, psi
S(52)	Direct tensile stress in bonnet body, psi
S(53)	Tensile stress in bonnet body due to bending moment, psi
S(54)	Direct shear stress in bonnet body, psi
S(55)	Shear stress in bonnet body due to torsional load, psi
S(56)	Combined shear stress in bottom bonnet weld, psi
S(57)	Total tensile stress in bottom bonnet weld, psi
S(58)	Direct tensile stress in bottom bonnet weld, psi
S(59)	Tensile stress in bottom bonnet weld due to bending moment, psi

S(60)	Total shear stress in bottom bonnet weld, psi
S(61)	Direct shear stress in bottom bonnet weld, psi
S(62)	Shear stress in bottom bonnet weld due to torsional load, psi
S(63)	Combined shear stress in top bonnet weld, psi
S(64)	Total tensile stress in top bonnet weld, psi
S(65)	Direct tensile stress in top bonnet weld, psi
S(66)	Tensile stress in top bonnet weld due to bending moment, psi
S(67)	Total shear stress in top bonnet weld, psi
S(68)	Direct shear stress in top bonnet weld, psi
S(69)	Shear stress in top bonnet weld due to torsional load, psi
S(70)	Combined stress in trunnion body, psi
S(71)	Direct tensile stress in trunnion body, psi
S(72)	Tensile stress in trunnion body due to bending moment, psi
S(73)	Direct shear stress in trunnion body, psi
S(74)	Shear stress in trunnion body due to torsional load, psi

SUMMARY TABLE INTRODUCTION

In the following pages, the pertinent data for the butterfly valve stress analysis is tabulated in three categories:

1. Stress Levels for Valve Components
2. Natural Frequencies of Components
3. Valve Dimensional Data

In Table 1, Stress Levels for Valve Components, the following data is tabulated:

Component Name

Code Reference (when applicable)

Stress Level Name and Symbol

Analysis Reference Page

Material Specification

Actual Stress Level

Allowable Stress Level

The material specifications are taken from Section II of the code when applicable. Allowable stress levels are Sm for tensile stresses and .6 Sm for shear stresses. The allowable levels are the same whether the calculated stress is a combined stress or results from a single load condition. Sm is the design stress intensity value as defined in Appendix I, Tables I-7.1 of Section III of the code.

In Table 2, Natural Frequencies of Valve Components, the following data is tabulated:

Summary Table Introduction

Component Name

Natural Frequency Symbol

Analysis Reference Page

Component Material

Natural Frequency

In Table 3, Valve Dimensional Data, the values for the pertinent valve dimensions and parameters are given.

Pages 20-26, Stress Level Summary sheets, Frequency Analysis Summary sheets, and Valve Dimensional Data sheets have been assembled at the beginning of the report submittal. They are located directly behind the design review record for the corresponding production order.

TABLE 1 STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVEL PSI
Body	NB-3541.1	Primary membrane stress in crotch region	P_m	35	ASME SA-516 Gr.70	1063	Sm 17500
		Primary membrane	$P_{m'}$	36	ASME SA-516 Gr.70	1307	Sm 17500
	NB-3545.2	Primary plus secondary stress due to internal pressure	Q_p	36	ASME SA-516 Gr.70	5345	Sm 17500
	NB-3545.2	Pipe reaction stress			ASME SA-516 Gr.70		1.5 Sm 26250
		Axial Load	P_{ed}	36		1393	
		Bending Load	P_{eb}	36		2718	
		Torsional Load	P_{et}	36		2718	
	NB-3545.2	Thermal secondary stress	Q_t	38	ASME SA-516 Gr.70	1193	Sm 17500
	NB-3545.2	Primary plus secondary stress	S_n	38	ASME SA-516 Gr.70	8448	3Sm 52500
	NB-3545.3	Normal duty fatigue stress $N_a \geq 2000$	S_p	38	ASME SA-516 Gr.70	6488	Sm 65000
Disc	NB-3546.2	Combined bending stress in disc	S(1)	39	ASME SA-516 Gr.70	4551	1.5 Sm 26250
	NB-3546.2	Bending tensile in unsupported flat plate	S(4)	41	ASME SA-516 Gr.70	2296	Sm 17500
	NB-3546.2	Shear tear out of shaft through disc	S(5)	41	ASME SA-516 Gr.70	5511	.6 Sm 10500
		Shear stress across disc hub welds	S(6)	41		3126	.6 Sm 7220

TABLE 1 STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVEL PSI
Shafts	NB-3546.3	Combined stress in shaft	S(7)	42	ASME SA-564 Type 630, Cond. H-1150	24411	Sm 33700
	NB-3546.3	Torsional shear stress at reduced pin cross-section	S(14)	43	ASME SA-564 Type 630, Cond. H-1150	7820	.6 Sm 20220
Disc Pin	NB-3546.3	Shear stress in top pin	S(15)	44	ASME SA-320 Gr.B8M	6298	.6 Sm 8160
	NB-3546.3	Bearing stress on top pin	S(16)	44	ASME SA-320 Gr.B8M	3387	Sm 13600
	NB-3546.3	Combined shear stress in bottom pin	S(17)	44	ASME SA-320, Gr.B8M	3297	.6 Sm 8160
Shaft Bearing		Compressive stress on shaft bearing	S(20)	45	ASTM B-438, Gr.1	3002	Sm 4000
Cover Cap	NB-3546.1	Shear tear out of cover cap bolts thru tapped holes in bottom trunnion	S(21)	46	ASME SA-516 Gr.70	2276	.6 Sm 10500
	NB-3546.1	Shear tear out of cover cap bolt head thru cover cap	S(22)	46	ASME SA-516 Gr.70	456	.6 Sm 10500
	NB-3546.1	Combined stress in cover cap bolts	S(23)	46	ASME SA-193 Gr.B7	9936	Sm 25000
		Combined stress in cover cap	S(26)	46	ASME SA-516 Gr.70	7337	Sm 17500
Thrust Bearing		Bearing stress on thrust collar	S(30)	49	SAE-660	808	Sm 8800

TABLE 1 STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVEL PSI
Thrust Bearing (Cont.)		Shear load on thrust collar spring pin	S(31)	49	AISI-420	1947	Pm 5400#
		Bearing stress of spring pin on thrust collar	S(32)	49	SAE-660	5774	Sm 8800
		Shear tear out of spring pin thru bottom shaft	S(34)	49	ASME SA-564 Type 630, Cond. H-1150	2971	.6 Sm 20220
Operator Mounting		Shear tear out of trunnion bolt thru tapped hole in trunnion	S(35)	50	ASME SA-516 Gr.70	4333	.6 Sm 10500
		Bearing stress of trunnion bolt on tapped hole	S(36)	50	ASME SA-516 Gr.70	6160	Sm 17500
		Bearing stress of trunnion bolt on thru hole in bonnet	S(37)	50	ASTM A-36	6160	Sm 12600
		Shear tear out of trunnion bolt head thru bonnet	S(38)	50	ASTM A-36	2348	.6 Sm 7560
		Combined stress in trunnion bolt	S(39)	52	SAE Gr.8	26464	Sm 30000
		Shear tear out of operator bolt head thru hole in bonnet	S(44)	52	ASTM A-36	677	.6 Sm 7560
		Bearing stress of operator bolt on hole in bonnet	S(45)	52	ASTM A-36	3641	Sm 12600

TABLE 1 STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVEL PSI
Operator Mounting (Cont.)		Combined stress in operator bolt	S(46)	52	SAE Gr.8	7274	Sm 30000
		Combined stress in bonnet body	S(51)	55	ASTM A-36	9403	Sm 12600
		Combined shear stress in bottom bonnet welds	S(56)	56		1882	.6 Sm 7200
		Combined shear stress in top bonnet welds	S(63)	56		1349	.6 Sm 7200
		Combined stress in trunnion body	S(70)	57	ASME SA-516 Gr.70	1840	Sm 17500

Table 2 NATURAL FREQUENCIES OF VALVE COMPONENTS

Component Name	Natural Frequency Symbol	Ref. Page	Material	Natural Frequency (Hertz)
Body	F_{N1}	58	ASME SA-516 Gr.70	25339
Banjo	F_{N2}	59	ASME SA-564 Type 630 Cond. H-1150	2705
Cover Cap	F_{N3}	59	ASME SA-516 Gr.70	724
Bonnet	F_{N4}	60	ASTM A-36	266

TABLE 3

DIMENSIONAL DATA

Job Number: D-45608 (D-0086-10) Valve Size: 18"-1200

Operator Mounting: TEE BONNET Operator: T312B-SR2

A_f	33.25	C_3	.60	g	32.2
A_m	10.30	C_6	2.49	G_b	618.2
A_3	.142	C_7	2.72	G_d	142.17
A_4	.126	C_8	2.36	G_T	1236.4
A_5	.226	C_9	.50	g_x	3
A_6	.202	d	16.876	g_y	3
A_7	.462	d_m	16.876	g_z	4
A_8	.419	D_1	18	H_2	4.25
B_1	1.566	D_2	2.25	H_3	4.25
B_2	2.50	D_3	.685	H_4	N/A
B_3	N/A	D_4	3.125	H_5	N/A
B_4	N/A	D_5	.375	H_6	N/A
B_5	9.69	D_6	.50	H_7	7.5
B_6	.375	D_7	.625	H_8	7.0
B_7	.375	D_8	.875	H_9	2.354
B_8	4.5	D_9	N/A	I_1	157.63
B_9	3.0	E	30E6	I_2	22.57
C	.30	F_b	56	I_3	67.43
C_b	1.0	F_d	6.6	I_4	105.32
C_p	3.0	F_x	1341	I_5	6673
C_o	1.30	F_y	1341	I_6	1.258
C_2	.43	F_z	1788	I_7	.93

J_1	.625	M_z	10205	ΔT_2	1.80
J_2	.812	$\overline{M}_x$	24062	T_1	1.188
J_3	N/A	$\overline{M}_y$	19502	T_2	.141
J_4	N/A	$\overline{\overline{M}}_x$	24062	T_3	.75
J_5	N/A	$\overline{\overline{M}}_y$	19502	T_4	.374
J_6	N/A	M_8	4270	T_5	N/A
K_o	2.25	N_a	2000	T_6	.625
K_1	N/A	N_1	2	T_7	.625
K_2	1.0	N_2	4	T_8	12551
K_3	3.5	N_3	4	U_1	8.38
K_4	5.5	P_d	275	U_2	8.38
K_5	6.0	P_r	150	U_3	.25
K_6	1.875	P_s	285	V_1	N/A
L_1	6.0	Q_{T1}	1000	V_2	N/A
L_2	N/A	r	2.016	V_3	N/A
L_3	.437	r_i	8.438	V_4	N/A
L_4	.625	r_2	1.0	V_5	.875
L_5	4.0	R_4	7.85	V_6	3.745
L_6	36.7	R_5	1.125	V_7	.875
L_7	N/A	R_m	9.615	V_8	7.804
L_8	.625	R_6	2.25	W_1	679
L_9	3.84	S	30,000	W_2	188
m	3.5	t_c	1.467	W_3	447
M_x	14286	t_m	.48	W_4	37
M_y	9727	T_c	2.354	W_6	.50

W_7	158
W_8	N/A
X_o	2.53
Y_o	5.08
Z_o	3.88
Z_1	27.64
Z_2	9.65
Z_3	28.26
Z_4	0
Z_7	5.75

Standard Stress Report

for

NRS Butterfly Valve

with

Bonnet Mounted

Cylinder Operator

ANALYSIS INTRODUCTION

Described in the following pages is the analysis used in verifying the structural adequacy of the main elements of the NRS butterfly valve. The analysis is structured to comply with Paragraph NB-3550 of Section III of the ASME Boiler and Pressure Vessel Code (hereafter referred to as the code). In the analysis, the design rules for Class 1 valves are used, since the requirements for this class of valve is much more explicit than for either Class 2 or 3 design rules. The design rules for Class 2 and 3 are exceeded by the rules for Class 1 valves.

Valve components are analyzed under the assumption that the valve is either at maximum fluid dynamic torque or seating against the maximum design pressure. Analysis temperature is 300°F. Seismic accelerations are simultaneously applied in each of three mutually perpendicular directions.

Seismic loads are made an integral part of the analysis by the inclusion of the acceleration constants g_x, g_y, g_z. The symbols g_x, g_y, g_z represent accelerations in the x, y and z directions respectively. These directions are defined with respect to the valve body centered co-ordinate system as illustrated in Figure 1. Specifically, the x axis is along the pipe axis, the z axis is along the shaft axis, and the y axis is mutually perpendicular to the x and z axes, forming a right hand triad with them.

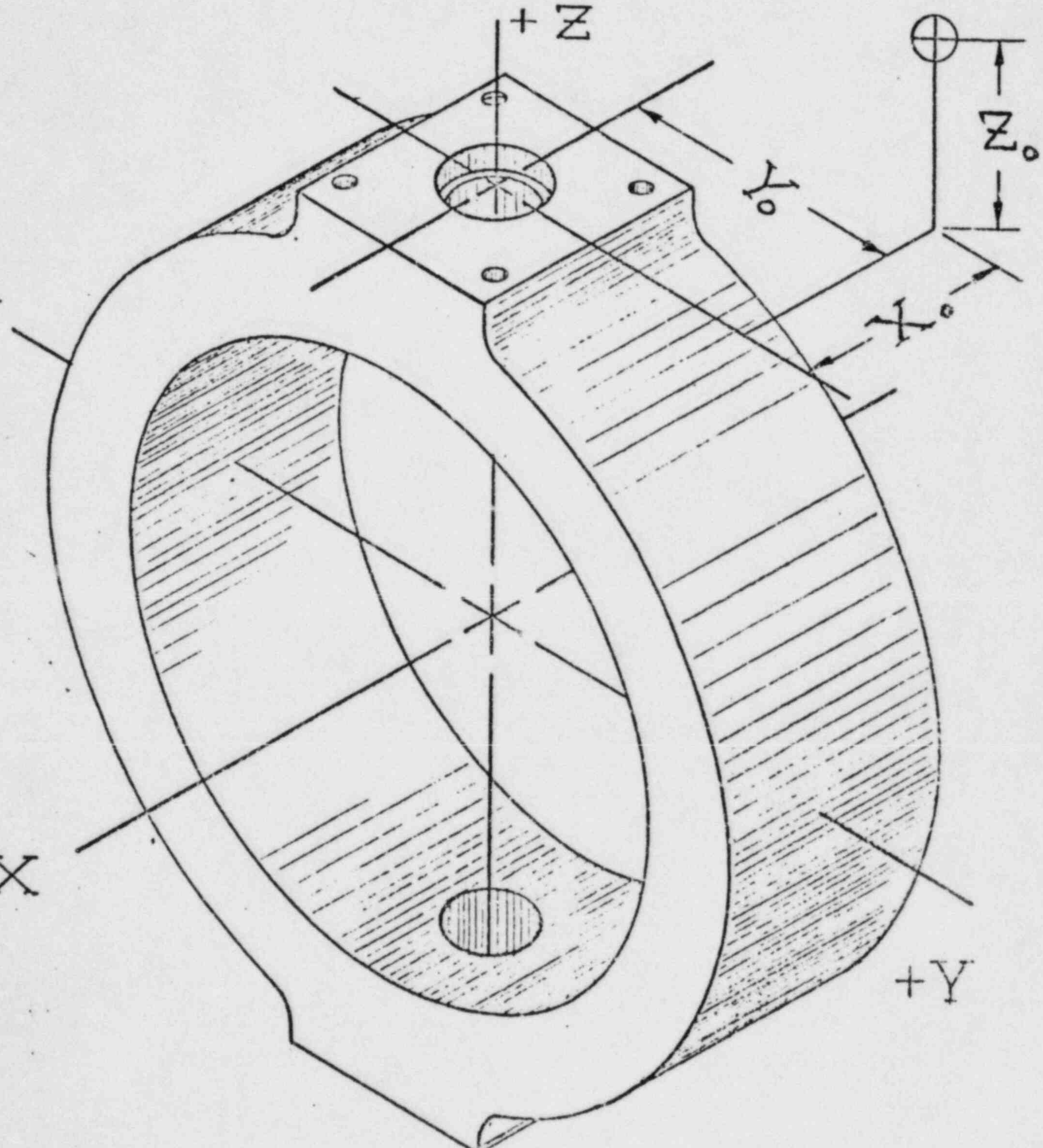


Figure 1 VALVE BODY SPATIAL ORIENTATION

Analysis Introduction

Valve orientation with respect to gravity is taken into account by adding the appropriate quantity to the seismic loads. The justification for doing this is that a gravitational load is completely equivalent to a 1g seismic load.

The analysis of each main element or sub-assembly of the butterfly valve is described separately in an appropriately titled section. In addition to containing sketches where appropriate, each section contains an explanation of the basis for each calculation. Where applicable, it also contains an interpretation of code requirements as they apply to the analysis.

Figure 2 is a cross-section view of the butterfly valve, and its associated components. Detailed sketches are provided throughout the report to clearly define the geometry.

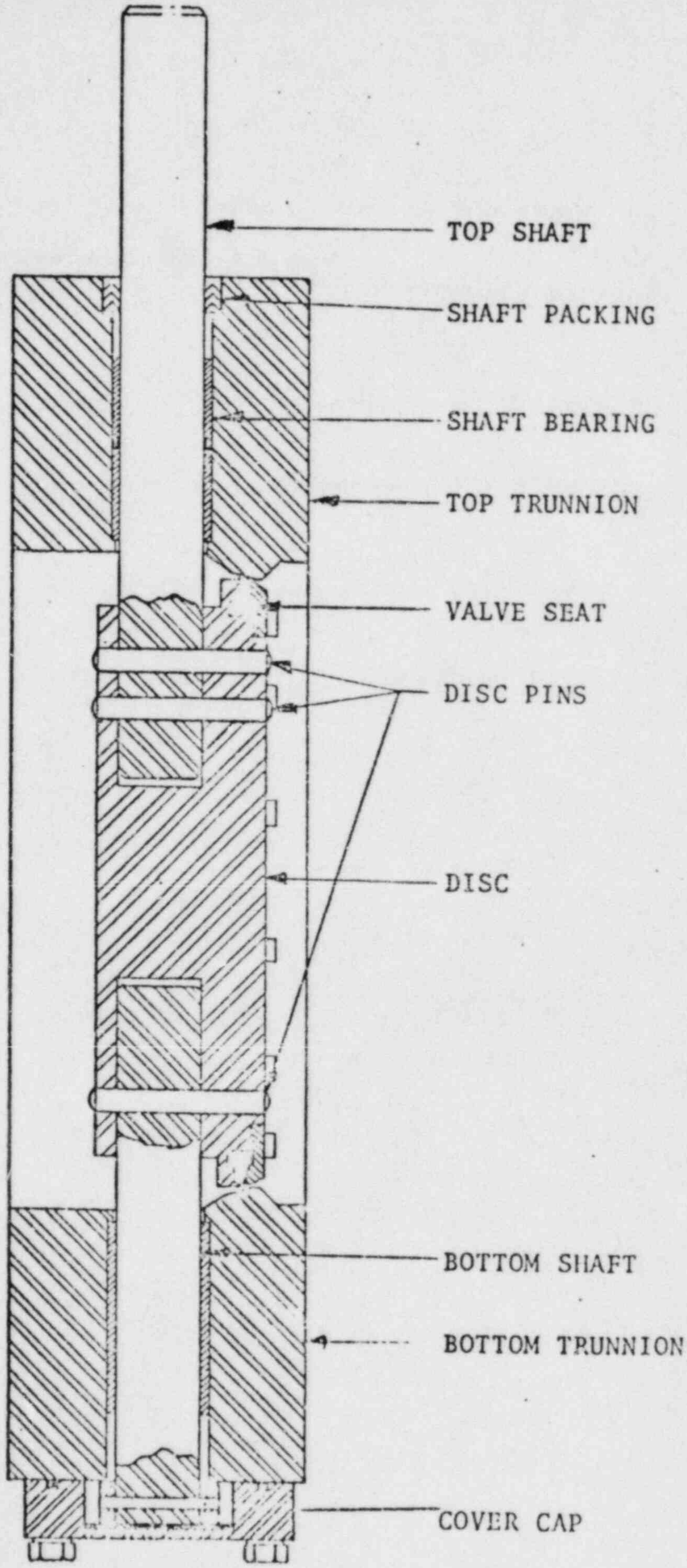


Figure 2 VALVE CROSS-SECTION

END CONNECTION ANALYSIS

The NRS butterfly valve is a uniflange design. Rather than having flanges that are external to and distinct from the body, the body shell is fabricated so that the end connections are machined directly into the body shell. The outside and inside diameter of the body shell conform to the requirements of the American National Standards Institute (ANSI) standard B16.5. The end connections, either flanged or weld end, also conform to this standard.

BODY ANALYSIS

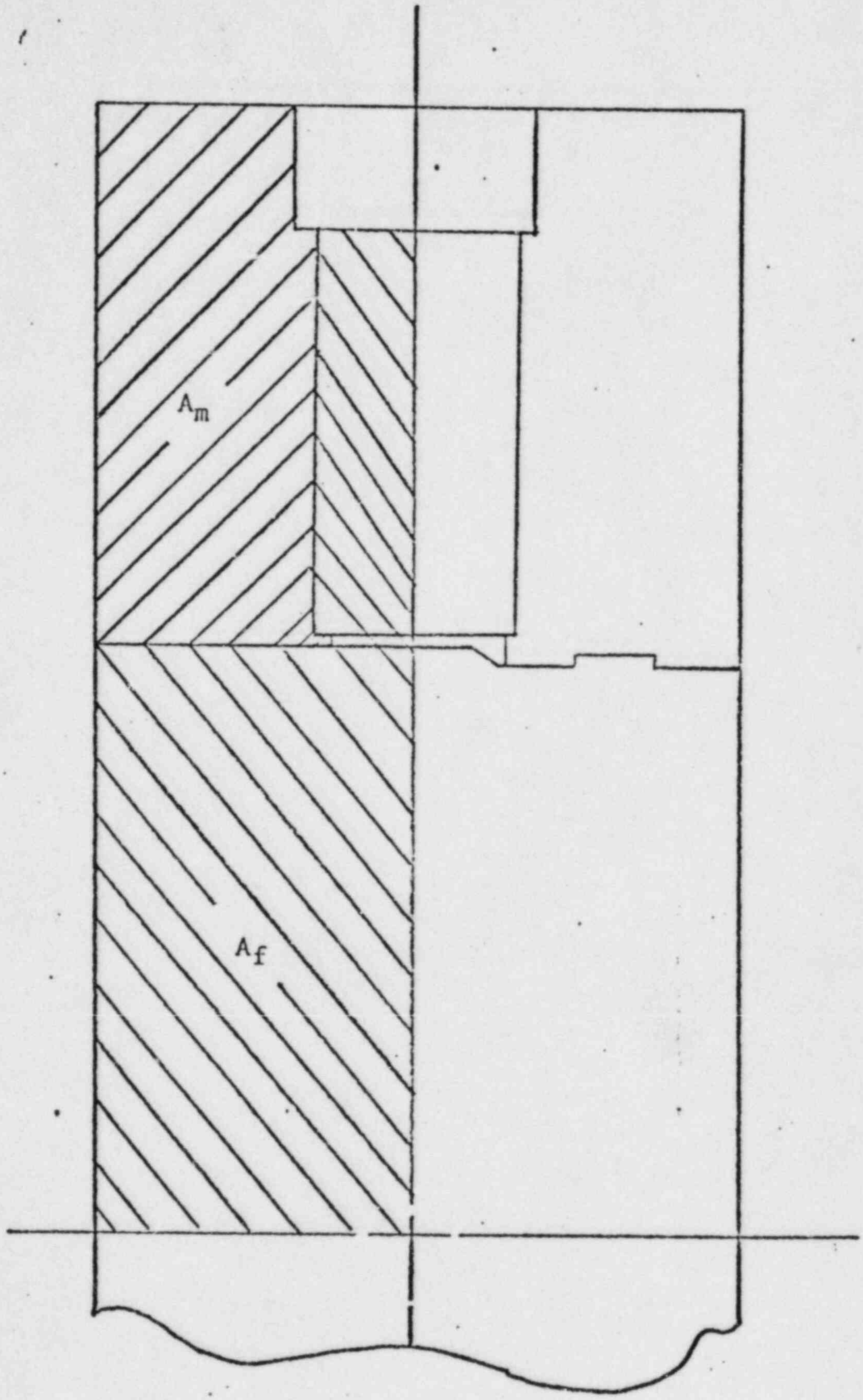
The body analysis consists of calculations as detailed in Paragraph NB-3540 of Section III of the Code. Paragraph NB-3540 is not highly oriented to butterfly valves as related to various design and shape rules. Therefore, certain of the design equations cannot be directly applied for butterfly valves. Where interpretation unique to the calculation is necessary, it is explained in the subsection containing that calculation description.

Figure 3 illustrates the essential features of the body geometry through the trunnion area of the valve. The symbols used to define specific dimensions are consistent with those used in the analysis and with the nomenclature used in the Code.

1. Minimum Body Wall Thickness

   Paragraph NB-3542 gives minimum body wall thickness requirements for standard pressure rated valves.

   The actual minimum wall thickness in the NRS valve occurs between the flange bolt holes and body bore.



PRESSURE-AREA ANALYSIS

BODY CROSS-SECTION

Figure 3

2. Body Shape Rules

The NRS valve meets the requirements of Paragraph NB-3544 of the code for body shape rules. The external fillet at trunnion to body intersection must be greater than thirty percent of the minimum body wall thickness.

3. Primary Membrane Stress Due to Internal Pressure

Paragraph NB-3545.1 defines the maximum allowable stress in the neck to flow passage junction. In a butterfly valve, this corresponds with the trunnion to body shell junction. Figure 3 shows the geometry through this section.

The code defines the stresses in this area using the pressure area method. As seen in Figure 3, certain code-defined dimensions are not applicable to this style of butterfly valve. For example, there is no radius at the crotch when seen in a view along the flow pattern, as the neck extends to the face of the body. To comply with the intent of the code, the areas A_f and A_m are interpreted as shown in the cross-section (Figure 3). Using these areas, the primary membrane stress is then calculated.

$$P_m = (A_F/A_m + .5)\ p_s$$

Body Analysis

As an alternate method of determining the primary membrane stress, an equivalent analysis for primary membrane stress is applied to an area away from the trunnions. In these areas, the metal area and fluid area are as shown in Figure 4. Since the depth of the metal area is equal to the depth of the fluid area, the ratio A_f/A_m is equivalent to the mean radius of the body over the thickness of the body shell, R_m/H_9. The primary membrane stress through this section is then:

$$P_{m'} = (R_m/H_9 + .5)\ p_s$$

4. Secondary Stresses

A. Body Primary plus secondary stress due to internal pressure: Paragraph NB-3545.2(a) of Section III of the code defines the formulas used in calculating this stress.

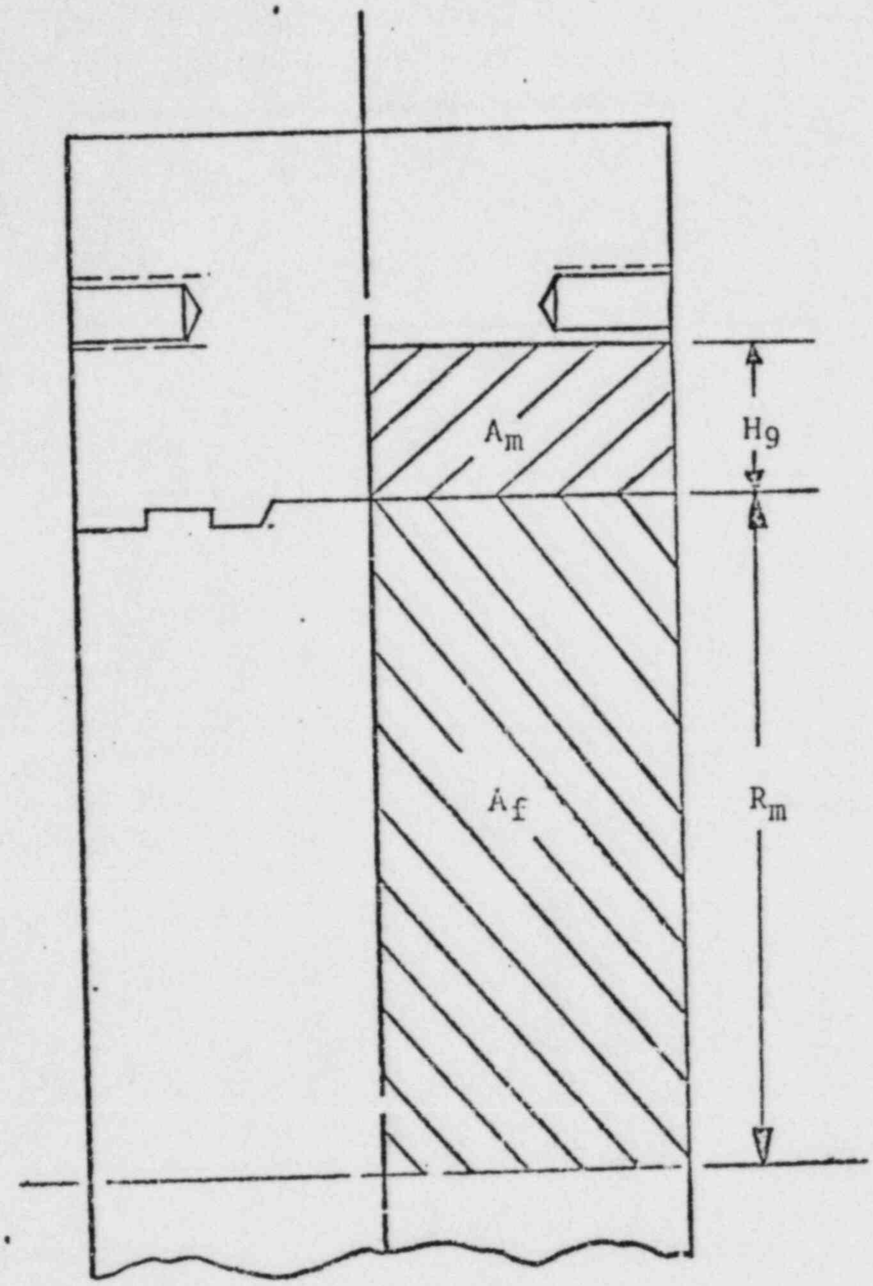
$$Q_p = C_p \left[\frac{r_i}{t_e} + .5 \right] p_s$$

B. Secondary stress due to pipe reaction: Paragraph NB-3545.2(b) gives the formulas for finding stress due to pipe reaction.

$$P_{ed} = \frac{F_d S}{G_d} \qquad \text{(Direct or Axial Load Effect)}$$

$$P_{eb} = \frac{C_b F_b S}{G_b} \qquad \text{(Bending Load Effect)}$$

$$P_{et} = \frac{2 F_b S}{G_t} \qquad \text{(Torsional Load Effect)}$$



PRESSURE AREA ANALYSIS

CROSS-SECTION IN BODY

Figure 4

C. Thermal secondary stress: Paragraph NB-3545.2(c) of Section III of the code gives formulas for determining the thermal secondary stresses in the pipe.

$$Q_T = Q_{T1} + Q_{T2}$$

Where

$$Q_{T2} = C_6 C_2 \Delta T_2$$

D. Primary plus secondary stresses: This calculation is per Paragraph NB-3545.2 and is simply the sum of the three previous secondary stresses.

$$S_n = Q_p + P_e + 2Q_{t2} \leq 3S_m$$

5. Valve Fatigue Requirements

Paragraph NB-3543.3 of Section III of the code defines requirements for normal duty valve fatigue.

The allowable stress level is found from Figure I-9.0. Since the number of cycles is unknown, a maximum value of 2,000 is assumed. The allowable stress can then be found from Figure I-9.1 for carbon steel. This then gives an allowable stress of 65,000 psi.

$$S_{p1} = 2/3\ Q_p + P_{eb}/2 + Q_{T3} + 1.3Q_{T1}$$

$$S_{p2} = .4\ Q_p + P_{eb} + 2Q_{T3}$$

Where:

$$Q_{T3} = C_6 C_3 \Delta T_2$$

DISC ANALYSIS

Section NB-3546.2 defines the design requirements of the valve disc. Both primary bending and primary membrane stress are mentioned in this section. For a flat plate such as the butterfly valve disc, membrane stress is not defined until the deflection of the disc reaches one-half the disc thickness. Since total deflection of the disc is much less than one-half the thickness, membrane stresses are not applicable to the analysis.

Figure 5 shows the disc for the NRS butterfly valves. The disc is designed to provide a structurally sound pressure retaining component while providing the least interference to the flow.

Primary Bending Stress

Due to the manner in which the disc is supported, the disc experiences bending both along the shaft axis and about the shaft axis. The combined bending stress is maximized at the disc center where the maximum moment occurs. The moment is a result of a uniform pressure load.

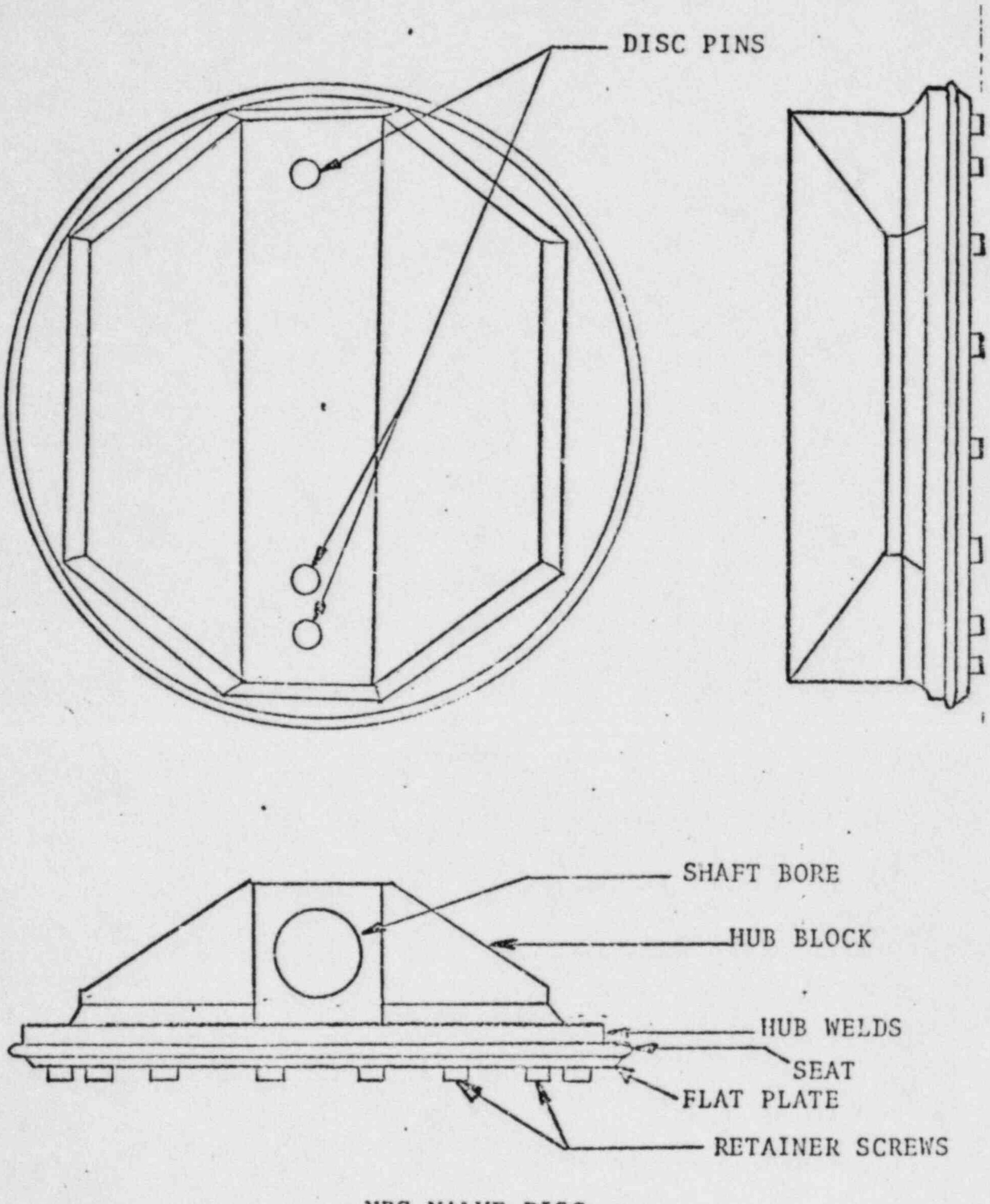
Combined bending stress in disc:

$$S(1) = (S(2)^2 + S(3)^2)^{1/2}$$

Where:

$$S(2) = \frac{.90413\ P_s R_4^3 C_7}{I_4}$$ = Bending stress due to moment along shaft axis, psi

$$S(3) = \frac{.6666\ P_s R_4^3 C_8}{I_3}$$ = Bending stress due to moment about shaft axis, psi



NRS VALVE DISC

Figure 5

Disc Analysis

Bending stress of unsupported flat plate:

$$S(4) = \frac{M_8 C_9}{I_7}$$

Shear Tear Out of Shaft

The disc is designed so the minimum thickness of material surrounding the shaft extension in the disc is above the shaft on the arch side. The loading is due to both seismic and pressure loads.

$$S(5) = \frac{\pi P_s R_4^2 + W_2 \sqrt{g_x^2 + g_y^2 + g_z^2}}{2L_9 (K_2 + \frac{D_2}{2}(1 - \text{SIN } 45^\circ))}$$ = Shear tear out shaft through disc, psi.

Shear Stress in Hub Welds

$$S(6) = \left[\left[\frac{\pi R_4^2 P_s}{W_6 L_6} + \frac{T_8}{Z_7 W_6 L_6} \right]^2 + \left[\frac{\pi R_4^2 .283 (g_y^2 + g_z^2)^{1/2}}{L_6 W_6} \right]^2 \right]^{1/2}$$

SHAFT ANALYSIS

The shaft is analyzed in accordance with Para NB-3546.3 of Section III of the Code. The shaft loading is a combination of seismic, pressure, and operating loads. Maximum torsional loading is either a combination of seating and bearing torque or bearing and dynamic torque. Columnar stress in not considered in the shaft loading due to its' negligible effect on the stress levels. Figure 2 shows the banjo assembly with the through shaft.

Shaft stresses due to pressure, seismic and operating loads:

$$S(7) = \frac{S(8)}{2} + \frac{(S(8)^2+4\ S(9)^2)^{1/2}}{2}$$

Where:

$S(8) = (S(10)^2+S(11)^2)^{1/2}$ = Combined bending stress, psi

$S(10) = \dfrac{(\pi R_4{}^2 p_s + W_2 g_x).25\ B_1 R_5}{\pi .25 R_5{}^4}$ = Bending tensile stress due to pressure and seismic loads along x axis, psi

$S(11) = \dfrac{.25 W_2 g_y\ B_1 R_5}{.25\ \pi\ R_5{}^4}$ = Bending tensile stress due to seismic loads along y axis, psi

$S(9) = (S(12)^2+S(13)^2)^{1/2}$ = Combined shear stress, psi

$S(12) = \dfrac{T_8 R_5}{.5 \pi R_5{}^4}$ = Torsional shear stress, psi

$S(13) = 1.333 \left[\dfrac{.5\pi R_4{}^2 p_s + .5 W_2 (g_x{}^2 + g_y{}^2)^{1/2}}{\pi R_5{}^2} \right]$ = Direct shear stress, psi

Also worthy of attention is the torsional shear stress at the reduced cross-section where the disc pin passes through the shaft.

Shaft Analysis

$$S(14) = S(12)\left[\frac{\frac{\pi R_5{}^4}{2}}{\frac{\pi R_5{}^4}{2} - \frac{D_2 D_3{}^3}{12} - \frac{D_3 D_2{}^3}{12}}\right]$$

DISC PIN ANALYSIS

As seen in Figure 2, there are two stub shafts to the disc pins. The top pins are subjected to torsional load as they transmit the operating torque. The bottom pin is subject to shear loads due to seismic and torsional loads.

Shear stress in top disc pin:

$$S(15) = \frac{T_8 - .5U_5}{2N_1R_5 .785D_3^{\,2}}$$

Bearing stress on top disc pin:

$$S(16) = \frac{T_8 - .5U_5}{2R_5K_2D_3N_1}$$

Combined shear stress - bottom disc pin:

$$S(17) = \left[S(18)^2 + S(19)^2\right]^{1/2}$$

Torsional shear stress in bottom disc pin:

$$S(18) = \frac{(.5U_5 + U_6)}{D_2 .785D_3^{\,2}}$$

Shear stress in bottom pin one to seismic accelerations + pressure on end of shaft:

$$S(19) = \frac{W_2g_z + \pi R_5^{\,2}P_0}{2(.785)D_3^{\,2}}$$

DISC PIN ANALYSIS

Where:

$$U_4 = .785(2R_4)^2 P_0 U_3 R_5$$

$$U_5 = U_4 + W_2 g_x U_3 R_5$$

$$U_6 = \left[W_2 g_z + \pi R_5^2 P_0 \right] .25(D_4 + D_2)/4$$

P_0 = Actual shut-off pressure

SHAFT BEARING ANALYSIS

The sleeve bearings in the trunnion (Figure 2) are subjected to both seismic and pressure loads.

$$S(20) = \frac{\pi p_d R_4{}^2 + W_2 (g_x{}^2 + g_y{}^2)^{1/2}}{2\ L_5 D_2} = \text{Compressive stress on shaft bearing, psi}$$

COVER CAP ANALYSIS

Figure 6 shows the bottom trunnion assembly, including the cover cap and cover cap bolts.

1. Cover cap bolt stresses:

The cover cap experiences loading from the weight of the banjo and from pressure loads. In determining stress levels, the bolts are assumed to share torsional and tensile loading equally.

Shear tear out of bolts through tapped holes in trunnion:

$$S(21) = \frac{W_2 \sqrt{g_x^2+g_y^2+g_z^2} + \pi P_s R_6^2}{4 L_3 \, 2.83 \, D_6}$$

Shear tear out of bolt heads through cover cap, psi:

$$S(22) = \frac{W_2 \sqrt{g_x^2+g_y^2+g_z^2} + \pi P_s R_6^2}{4 T_1 \, 5.2 \, D_6}$$

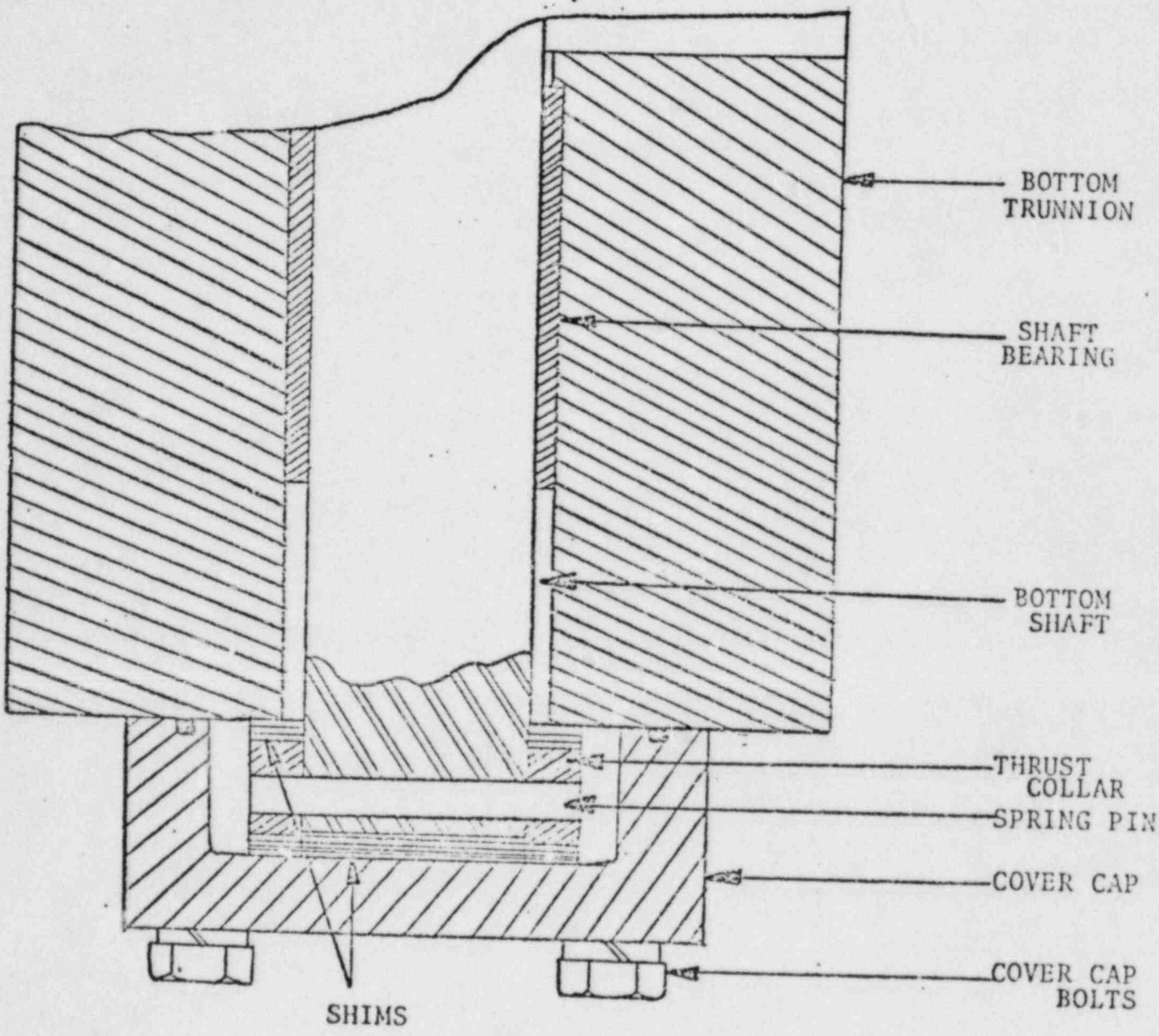
Combined stress in bolts, psi:

$$S(23) = \frac{S(25)}{2} + \frac{(S(25)^2 + 4S(24)^2)^{1/2}}{2}$$

Where:

$$S(24) = \frac{.25 W_2 \sqrt{g_x^2+g_y^2+g_z^2} \; (D_2 + .66 (D_4 - D_2))}{.707 H_3 \, 4 \, A_4}$$

= Shear Stress in Bolts Due to Torsional Load.



BOTTOM TRUNNION AND THRUST BEARING ASSEMBLY

Figure 6

Cover Cap Analysis

$$S(25) = \frac{W_2 \sqrt{g_x^2+g_y^2+g_z^2} + \pi P_s R_6^2}{4 A_3}$$ = Tensile Stress in Bolts Due to Seismic And Pressure Loads, psi

2. Cover cap stresses:

The combined stress in the covercap is calculated using the following formulas:

$$S(26) = \frac{S(27) + S(28)}{2} + \frac{((S(27) + S(28))^2 + 4S(29)^2)^{\frac{1}{2}}}{2}$$

Where:

$$S(27) = \frac{3(.785 (D_4 + .25)^2 P_s + W_2 g_z)}{4 \pi T_4^2}$$ = Radial Stress

$$S(28) = \frac{3(.785(D_4 + .25)^2 P_s + W_2 g_z)}{4 \pi T_4^2 m}$$ = Tangential Stress

$$S(29) = \frac{.785 (D_4 + .25)^2 P_s + W_2 g_z}{\pi (D_4 + .25) T_4}$$ = Shear Stress

THRUST BEARING ANALYSIS

As seen in figure 6, the thrust bearing assembly is located in the bottom trunnion. It provides restraint for the banjo in the z direction, assuring that the disc edge remains correctly positioned to maintain optimum sealing. Formulas used to analyze the assembly are given below.

1. Bearing stress on thrust collar due to seismic and pressure loads:

$$S(30) = \frac{W_2 \sqrt{g_x^2+g_y^2+g_z^2} + \pi P_s R_5^2}{.785 (D_4^2-(D_2+.25)^2)}$$

2. Shear load on thrust collar spring pin due to seismic, pressure and torsional loads:

$$S(31) = \left[(W_2 g_z + \pi P_s R_5^2)^2 + \left(\frac{.25 W_2 g_z (D_2+.0833+.66 (D_4-D_2))}{R_5}\right)^2\right]^{1/2}$$

3. Bearing stress of spring pin on thrust collar:

$$S(32) = \frac{((W_2 g_z + \pi P_s R_5^2)^2 + (.25 W_2 g_z)^2)^{1/2}}{D_5 (D_4-D_2)}$$

4. Shear tear out of spring pin through bottom of shaft:

$$S(34) = \frac{W_2 g_z + \pi P_s R_5^2}{2D_2 T_2}$$

OPERATOR MOUNTING ANALYSIS

The operator mounting consists of the top trunnion, the bonnet, the operator housing, and the bolt connections. The elements of the assembly are shown in Figure 7.

1. Bolt stresses and localized stress due to bolt loads. The following assumptions are used in the development of the equations:

A. Torsional, direct shear, and direct tensile loads are shared equally by all bolts in the pattern.

B. Moments across the bolt pattern are opposed in such a way that the load in each bolt is proportional to its distance from the neutral bending axis.

(a) Shear tear out of trunnion bolt through tapped hole in top trunnion.

$$S(32) = \frac{\dfrac{F_z + W_4\sqrt{g_x{}^2+g_y{}^2+g_z{}^2}}{4} + \dfrac{\overline{\overline{M}}_x(J_2+H_2)}{2J_2{}^2+2(J_2+H_2)^2} + \dfrac{\overline{\overline{M}}_y(J_1+H_2)}{2J_1{}^2+2(J_1+H_2)^2}}{.9\pi\, L_4 D_7}$$

(b) Bearing stress on tapped holes in trunnion.

$$S(33) = \frac{\dfrac{M_z + T_8}{4(.707\, H_2)} + \dfrac{(F_x{}^2+F_y{}^2)^{1/2}}{4} + \dfrac{W_4\,(g_x{}^2+g_y{}^2)^{1/2}}{4}}{D_7 L_4}$$

(c) Bearing stress on through hole in bonnet.

$$S(34) = \frac{\dfrac{M_z + T_8}{4(.707\, H_2)} + \dfrac{(F_x{}^2+F_y{}^2)^{1/2}}{4} + \dfrac{W_4\,(g_x{}^2+g_y{}^2)^{1/2}}{4}}{D_7 T_6}$$

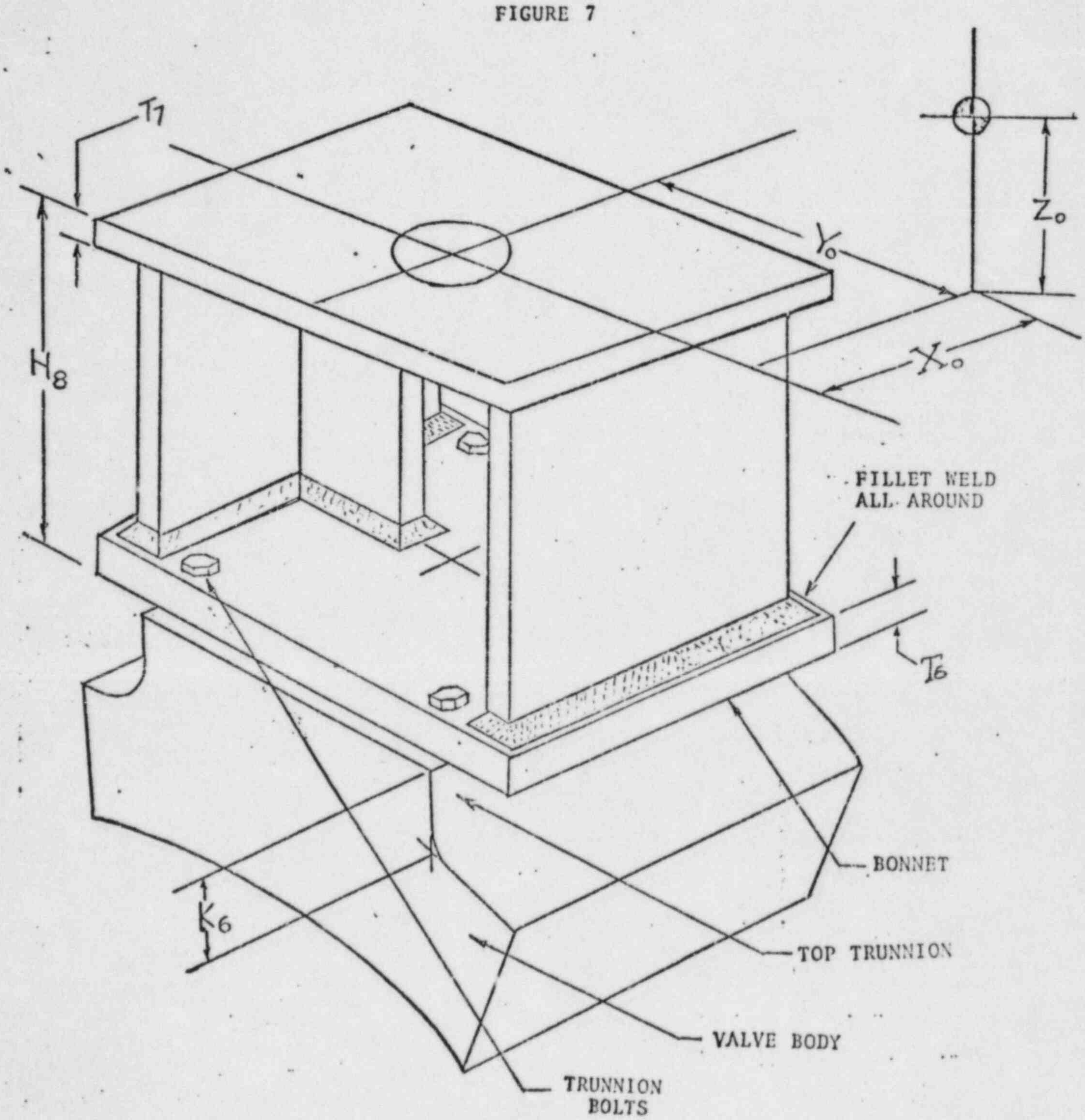
Operator Mounting Analysis (con't.)

(d) Shear tear out of trunnion bolt heads through bonnet.

$$S(35) = \frac{\dfrac{F_z + W_4 g_z}{4} + \dfrac{\overline{\overline{M}}_x (J_2 + H_2)}{2J_2^{\,2} + 2(J_2 + H_2)^2} + \dfrac{\overline{\overline{M}}_y (J_1 + H_2)}{2J_1^{\,2} + 2(J_1 + H_2)^2}}{5.2\, D_7 T_6}$$

TOP TRUNNION MOUNTING

FIGURE 7



e. Combined stress in trunnion bolts (See Fig. 8)

$$S(39) = \frac{S(40)+S(41)}{2} + \frac{((S(40)+S(41))^2+4(S(42)+S(43))^2)^{\frac{1}{2}}}{2}$$

Where

$$S(40) = \frac{F_z+W_4 g_z}{4A_5} = \text{Direct tensile stress, psi}$$

$$S(41) = \frac{\dfrac{\overline{M_x}(J_2+H_2)}{2J_2{}^2+2(J_2+H_2)^2} + \dfrac{\overline{M_y}(J_1+H_2)}{2J_1{}^2+2(J_1+H_2)^2}}{A_5}$$ = Tensile stress due to extended mass bending moment, psi

$$S(42) = \frac{(F_x{}^2+F_y{}^2)^{\frac{1}{2}} + W_4(g_x{}^2+g_y{}^2)^{\frac{1}{2}}}{4A_6}$$ = Direct shear stress, psi

$$S(43) = \frac{(M_z+T_8)}{(.707\ H_2)4A_6}$$ = Shear stress due to torsional load, psi

f. Shear tear out of operator bolt head through bonnet.

$$S(44) = \frac{\dfrac{F_z}{N_2} + \dfrac{M_x V_6}{2(V_5{}^2+V_6{}^2)} + \dfrac{V_8 M_y}{2(V_7{}^2+V_8{}^2)}}{5.2\ D_8 T_7}$$

g. Bearing stress on through holes in bonnet.

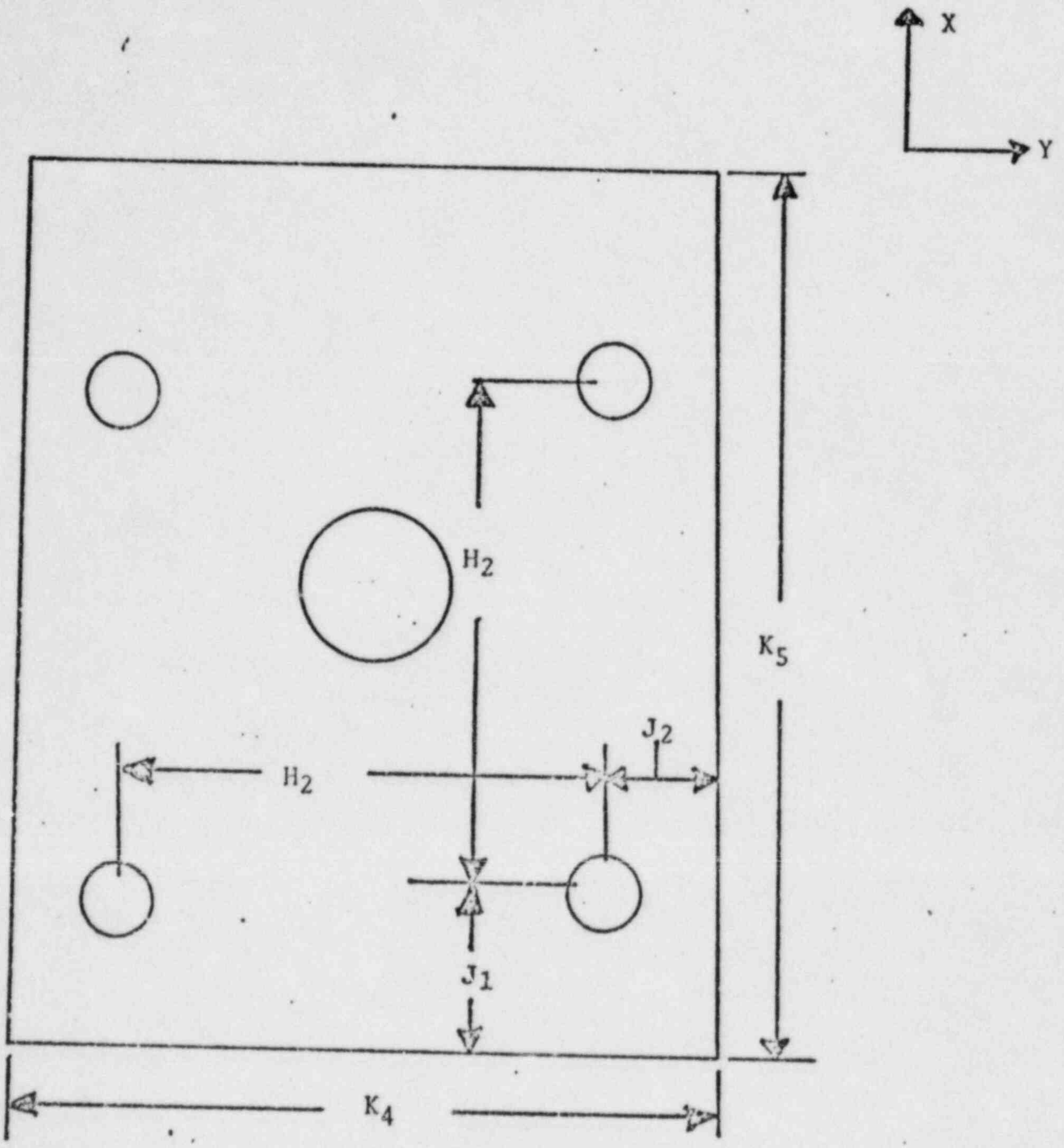
$$S(45) = \frac{\dfrac{M_z+T_8}{.5\ H_7 N_2} + \dfrac{(F_x{}^2+F_y{}^2)^{\frac{1}{2}}}{N_2}}{D_8 T_7}$$

h. Combined stress in operator bolts (See Fig. 9)

$$S(46) = \frac{S(47)+S(48)}{2} + \frac{((S(47)+S(48))^2+4(S(49)+S(50))^2)^{\frac{1}{2}}}{2}$$

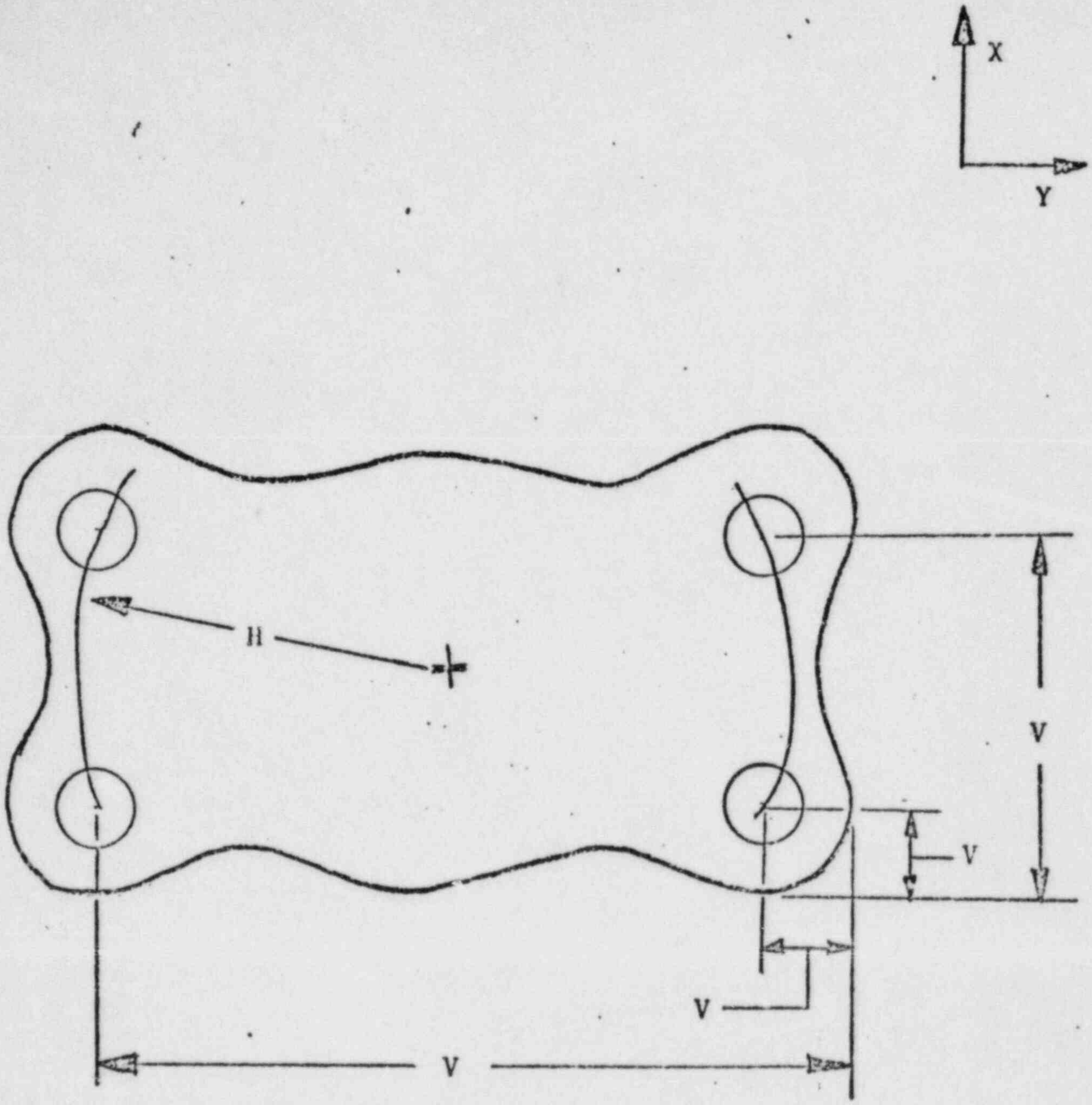
Where

$$S(47) = \frac{F_z}{N_2 A_7}$$



TOP TRUNNION BOLTING

Figure 8



OPERATOR BOLT PATTERN

Figure 9

$$S(48) = \frac{M_x V_6}{2(V_5{}^2+V_6{}^2)A_7} + \frac{M_y V_8}{2(V_7{}^2+V_8{}^2)A_7}$$

$$S(49) = \frac{(F_x{}^2+F_y{}^2)^{1/2}}{N_2 A_8}$$

$$S(50) = \frac{M_z+T_8}{.5\ H_7 N_2 A_8}$$

2. Bonnet Stresses

The bonnet stresses are calculated with the assumption that loading is through the bolt connections as previously defined.

a. The maximum combined stress in the bonnet was calculated using the following formulas:

$$S(51) = \frac{S(52)+S(53)}{2} + \frac{((S(52)+S(53))^2+4(S(54)+S(55))^2)^{1/2}}{2}$$

= Combined stress in bonnet legs

$S(52) = \dfrac{F_z+W_4 g_z}{B_5}$ = Direct tensile stress, psi

$S(53) = \dfrac{\overline{\overline{M}}_x B_8}{I_1} + \dfrac{\overline{\overline{M}}_y B_9}{I_2}$ = Tensile stress due to bending moment, psi

$S(54) = \dfrac{(F_x{}^2+F_y{}^2)^{1/2} + W_4(g_x{}^2+g_y{}^2)^{1/2}}{B_5}$ = Direct shear stress, psi

$S(55) = \dfrac{T}{\overline{K}_0} C_0$ = Shear stress in bonnet body due to torsional load, psi

where

T = Torque, in-lbs.
C_0 = Torsional constant for non-circular cross-section
K_0 = Function of cross-section, in.^{4}

b. The maximum combined shear stress in the bonnet mounting plate to body welds was calculated using the following formulas:

Operator Mounting Analysis

Bottom Bonnet Weld

$$S(56) = \frac{(S(57)+4S(58)^2)^{\frac{1}{2}}}{2}$$ = Combined shear stress in bottom weld, psi

Where

$$S(57) = S(59)+S(60)$$ = Total tensile stress, psi

$$S(59) = \frac{F_z+W_4 g_z}{U_1}$$ = Direct tensile stress, psi

$$S(60) = \frac{\overline{\overline{M}}_x}{Z_1} + \frac{\overline{\overline{M}}_y}{Z_2}$$ = Bending tensile stress

$$S(58) = S(61)+S(62)$$ = Total shear stress

$$S(61) = \frac{(F_x{}^2+F_y{}^2)^{\frac{1}{2}} + W_4(g_x{}^2+g_y{}^2)^{\frac{1}{2}}}{U_1}$$ = Direct shear stress, psi

$$S(62) = \frac{M_z+T_8}{Z_3}$$ = Torsional shear stress, psi

Top Bonnet Weld

$$S(63) = \frac{(S(64)^2+4S(65)^2)^{\frac{1}{2}}}{2}$$ = Combined shear stress in top bonnet weld

Where

$$S(64) = S(66)+S(67)$$ = Total tensile stress, psi

$$S(66) = \frac{F_z}{U_2}$$ = Direct tensile stress, psi

$$S(67) = \frac{M_x}{Z_1} + \frac{M_y}{Z_2}$$ = Bending tensile stress, psi

$$S(65) = S(68)+S(69)$$ = Direct shear stress, psi

Operator Mounting Analysis

$$S(68) = \frac{(F_x{}^2+F_y{}^2)^{1/2}}{U_2} = \text{Direct shear stress, psi}$$

$$S(69) = \frac{M_z+T_8}{Z_4} = \text{Torsional shear stress, psi}$$

c. Trunnion body stresses are calculated using the following assumptions:

1. Operator loading is through the bolt connections.
2. There is an equal and opposite reaction to the bolt loads at the body.

The combined stress in the trunnion body was calculated using the following formulas:

$$S(70) = \frac{S(71)+S(72)}{2} + \frac{((S(71)+S(72))^2+4(S(73)+S(74))^2)^{1/2}}{2}$$

Where

$$S(71) = \frac{F_z+W_4g_z}{K_4K_5-.785\ B_2{}^2} = \text{Direct tensile stress, psi}$$

$$S(72) = \frac{(\bar{\bar{M}}_x+F_yK_6).5K_4}{.0833K_5K_4{}^3-\frac{\pi B_2{}^4}{64}} + \frac{(\bar{\bar{M}}_y+F_xK_6).5K_5}{.0833K_4K_5{}^3-\frac{\pi B_2{}^4}{64}} = \text{Bending tensile stress, psi}$$

$$S(73) = \frac{(F_x{}^2+F_y{}^2)^{1/2} + W_4(g_x{}^2+g_y{}^2)^{1/2}}{K_4K_5-.785\ B_2{}^2} = \text{Direct shear stress, psi}$$

$$S(74) = \frac{(M_z+T_8).5(K_4{}^2+K_5{}^2)^{1/2}}{.0833(K_4K_5{}^3+K_5K_4{}^3)-\frac{\pi B_2{}^4}{32}} = \text{Torsional shear stress, psi}$$

FREQUENCY ANALYSIS

A. Introduction

To calculate the natural frequency of the various components of the NRS valve, a model system with a single degree of freedom is constructed. The individual components and groups of components are modeled and analyzed as restoring spring forces which act to oppose the respective weight forces they are subjected to. The static deflection of the component is calculated and is related to natural frequency as:

$$F_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

or

$$F_n = \frac{1}{2\pi}\sqrt{\frac{g}{\Delta y}}$$

or

$$F_n = \left(\frac{9.8}{\Delta y}\right)^{\frac{1}{2}}$$

The analysis details the equations and assumptions used in determining the natural frequencies listed in the summary table. Sketches are provided where appropriate.

B. Valve Body Assembly

The body shell, as seen in Figure 1, is assumed to experience loading due to the entire valve weight.

Natural Frequency of the body shell:

$$F_{N1} = \left(\frac{9.8}{\Delta y_1}\right)^{\frac{1}{2}}$$

Where

$\Delta y_1 = \dfrac{W_1 L_1^3}{48 \; E \; I_5}$ = Maximum deflection of body shell due to valve weight, in.

C. Banjo Assembly

Figure 2 shows the banjo assembly in the body. The natural frequency of the banjo assembly is calculated using the following:

$$F_{N2} = \left(\frac{9.8}{\Delta y_2}\right)^{\frac{1}{2}}$$

Where

$\Delta y_2 = \dfrac{W_7 B_1^3}{12 \; E \; I_6}$ = Maximum deflection of shaft, inches

D. Cover Cap Assembly

As seen in Figure 6, the cover cap supports the banjo. The natural frequency of the cover cap is calculated as follows:

$$F_{N3} = \left(\frac{9.8}{\Delta y_3}\right)^{\frac{1}{2}}$$

Where

$\Delta y_3 = \dfrac{3(m^2-1) \; W_2 \; (.5D_4+.125)^2}{16\pi E \; T_4^3 m^2}$ = Maximum deflection of cover cap

E. Bonnet Assembly

Figure 7 shows the top trunnion assembly. The following assumptions are made in calculating the bonnet natural frequency:

Frequency Analysis

1. The worst valve assembly mounting position is where the bending moment is predominant in producing deflection.
2. The bonnet is assumed fixed at the top trunnion.
3. The adapter plate is assumed to be integral with and have a cross-section the same as the component it mounts to.

Natural frequency of bonnet:

$$F_{n4} = \left(\frac{9.8}{\Delta y_4}\right)^{\frac{1}{2}}$$

Where

$$\Delta y_4 = \frac{W_3 H_8{}^3 + W_4 K_3{}^3}{3EI_1} + \frac{W_3 Z_o H_8{}^2}{2EI_1}$$

ATTACHMENT 3

GENERAL ARRANGEMENT

AND

CROSS-SECTION DRAWINGS

SEE

APERTURE

CARDS

*OVERSIZED DRAWINGS

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