Attachment 8 to
NRC-94-0050

HOPPER AND ASSOCIATES
ENGINEERS

DECO - USNRC DOCKET NO. 50-341

FERMI 2 CCHVAC DUCTING SYSTEMS

CONCERN ITEM NO. 12

EVALUATION CALCULATIONS

Prepared for:	Detroit Edison Company Nuclear Engineering Fermi 2 6400 North Dixie Highway Newport, MI 48166
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May 1994

Revision 1, May 1994

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ENGINEERS

HA-05/94-0819
Revision 1

CALCULATION SHEET

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1.1 PROBLEM STATEMENT

On March 3, 1994, the U.S.N.R.C. submitted Docket No. 50-341 to Detroit Edison Company. This Docket consists of a request for additional information on Fermi 2 CCHVAC system design and operation. In particular, there are several concerns regarding the structural integrity calculations, HA-05/89-686 and HA-09/89-696, performed by Hopper & Associates.

This package is prepared in order to respond to one of the comments posed by the NRC regarding HA-05/89-686. Namely,

"The structural integrity and air tightness of the transverse joints was not evaluated"

1.2 INVESTIGATION APPROACH

The as-built configuration of transverse joints used in systems analyzed in HA-05/89-686 will be reviewed. The loads acting on these joints will be established from computer runs made by Hopper & Associates. Based on these loads, the structural integrity of the transverse joints will be evaluated. Also, the possibility of flange leakage will be considered, and fatigue capability will be discussed.

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1.3 RESULT SUMMARY

Based upon calculations performed in this package it was determined that transverse joints in systems 2848-3, 4316-1, 4316-6 and 4316-7 are structurally adequate to withstand Site Specific Earthquake loading and operational loading, without exceeding Code allowable stresses and forces, and without compromising air tightness.

SYSTEM DESCRIPTION

Four ducting systems were evaluated in HA-05/89-686;

2848-3

This system is located in the auxiliary Building at an elevation of 690'-3". The system consists of 18" diameter round ducts.

4316-1

This system is located in the auxiliary Building at elevation 690'-6". The system consists of 14"x14" square ducts.

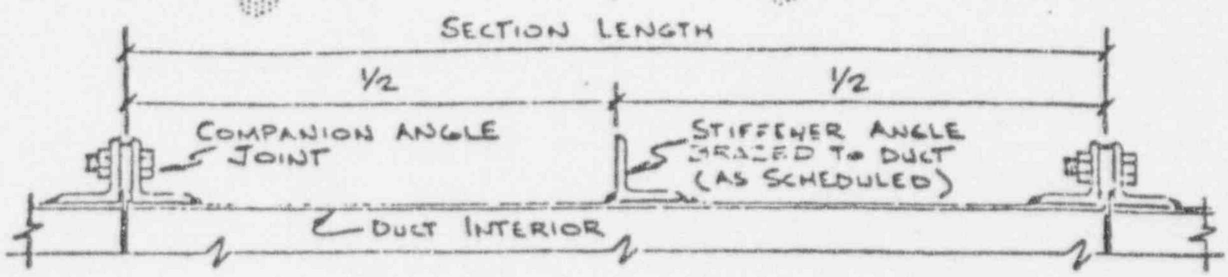
4316-6

This system is located in the auxiliary Building at elevation 690'-8". The system consists of 16"x16" square ducts and 17.5" diameter round ducts.

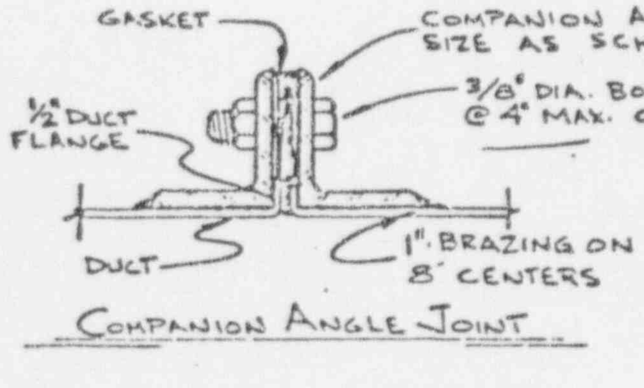
4316-7

This system is located in the auxiliary Building at elevation 690'-3". The system consists of 16"x16" square ducts and 18" diameter round ducts.

2.1 TRANSVERSE JOINT CONFIGURATION [Ref 1]



TYPICAL CROSS-SECTION



COMPANION ANGLE JOINT

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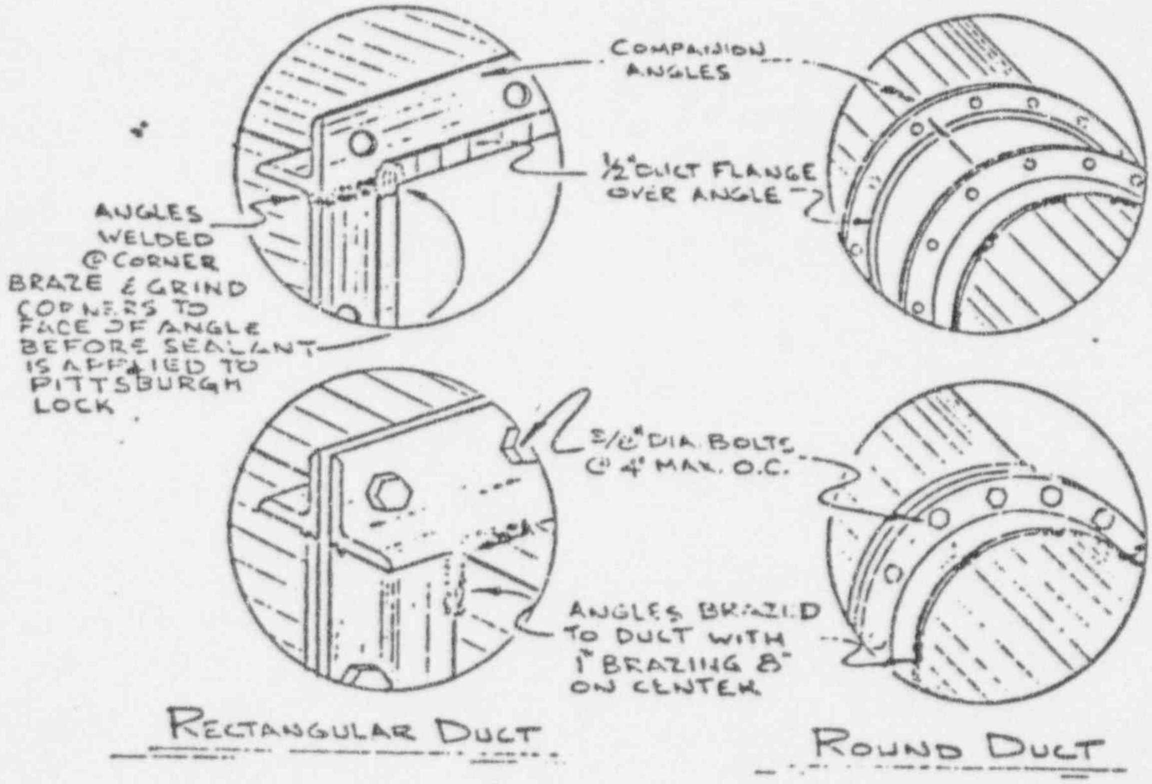


The angle sizes used are,

RECTANGULAR

L $1\frac{1}{2}" \times 1\frac{1}{2}" \times \frac{1}{4}"$

ROUND

L $1\frac{1}{2} \times 1\frac{1}{2}" \times \frac{1}{8}"$

2.2 MATERIALS

The materials used in construction are,

SHEET METAL	A-527 (G-90)	[Ref 1]
STIFFNER	A-575 (M-1020)	[Ref 1]
BOLTS	A-307 (assumed)	
GASKET	SCE 4-1 NEOPRENE CLOSED CELL SPONGE	[Ref 1]
WELD (< 13 gage)	FILLET TYPE, CLASS E CuSi (CONFORMING TO AWS-A5.6)	[Ref 1]
WELD (STEEL TO STEEL)	CLASS E7018 (CONFORMING TO AWS-A5.1)	

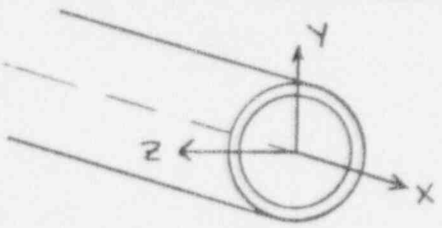
3.1 ASSUMPTIONS

1. Flanges are rigid for bolt tension analysis
2. Gasket material is rigid for flange leakage analysis
3. Bolting material is A307
4. Material A527 & A575, have the same mechanical strength properties as carbon steel A-36.
5. The AISI Code governs the design of ductwork because of its applicability to thin gage sheet metal. [Ref 15, Pg 11-3-1]

3.2 LOADS

The loads due to Deadweight are obtained from computer runs performed in HA-05/89-686 [Ref 2]. The seismic induced forces are from HA-05/89-686, obtained from a Site Specific Earthquake at 2% damping. The Hopper & Associates computer results are based on galvanized metal duct sheet thickness. To estimate the forces and moments that will result based on a bare metal thickness, all the forces reported in HA-05/89-686 are multiplied by a conservative factor of 1.1. This factor accounts for the frequency shift due to the added flexibility. The loads are summarized in Tables 3.2.1-4. The design pressure in all the systems is 22 inch H_2O or 0.795 psi.

DUCT SYSTEM 2848-3, 18"ϕ		CRITICAL LOCATION: ELEMENT 30 NODE 30
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	∅	730
F_y	-121	20
F_z	∅	840
M_x	-4100	1380
M_y	∅	52730
M_z	-874	410



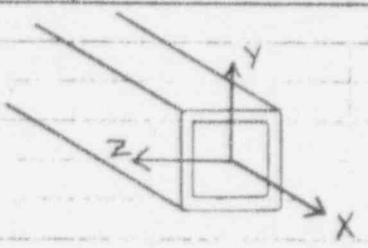
FORCES : LBS

MOMENTS: IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.1 LOADS FOR SYSTEM 2848-1

DUCT SYSTEM 4316-1, 14"□		CRITICAL LOCATION: ELEMENT 21 NODE 22
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	Ø	1050
F_y	-39	37
F_z	Ø	1856
M_x	-144	40
M_y	Ø	63260
M_z	-317	170



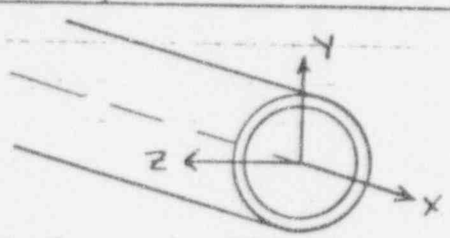
FORCES; LBS

MOMENTS; IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.2 LOADS FOR SYSTEM 4316-1

DUCT SYSTEM 4316-6, 17.5"∅		CRITICAL LOCATION: ELEMENT 50 NODE 47
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	∅	495
F_y	-303	∅
F_z	∅	302
M_x	∅	∅
M_y	∅	9750
M_z	605	∅



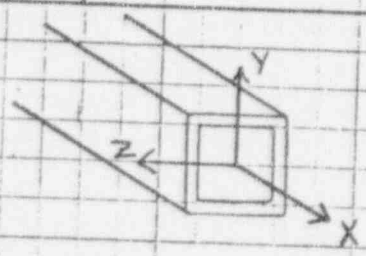
FORCES : LBS

MOMENTS: IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.3 LOADS FOR SYSTEM 4316-6

DUCT SYSTEM 4316-6, 16"☐		CRITICAL LOCATION: ELEMENT 27 NODE 13
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	∅	625
F_y	-62	39
F_z	∅	302
M_x	-309	210
M_y	15	10870
M_z	58	140



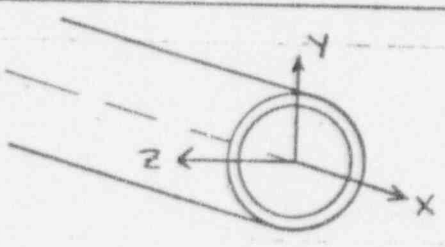
FORCES: LBS

MOMENTS: IN-LBS

*Values include 1.1 frequency adjustment factor

TABLE 3.2.3 CONT'D

DUCT SYSTEM 4316-7, 18" ϕ		CRITICAL LOCATION: ELEMENT NODE
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	ϕ	1428
F_y	29	29
F_z	0	468
M_x	-331	230
M_y	6	72210
M_z	-598	260



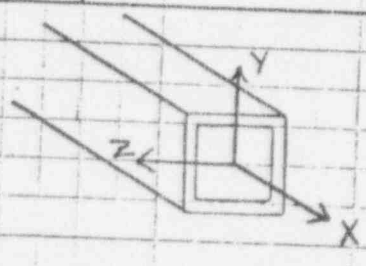
FORCES : LBS

MOMENTS : IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.4 LOADS FOR SYSTEM 4316-7

DUCT SYSTEM 4316-7, 16" □		CRITICAL LOCATION: ELEMENT NODE
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	∅	877
F_y	-39	∅
F_z	∅	681
M_x	-20	∅
M_y	28	67,700
M_z	-180	∅



FORCES: LBS

MOMENTS: IN-LBS

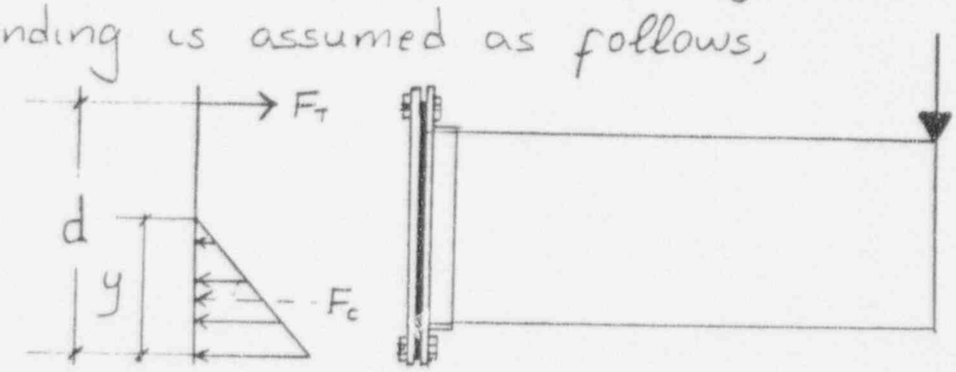
* Values include 1.1 frequency adjustment factor

TABLE 3.2.4 CONT'D

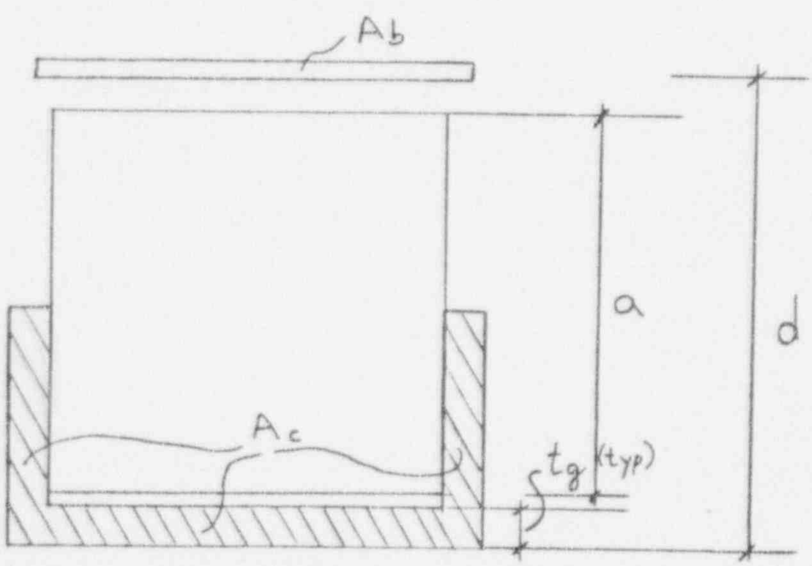
3.3 ANALYTICAL IDEALIZATION

3.3.1 BENDING

The load distribution on the flanges due to bending is assumed as follows,



RECTANGULAR SECTION



A_b = Total bolt area

A_c = Compressive area

t_g = Gasket width

a = Duct height or width

d = Distance from bolt to extreme compression

By finding the centroid y of this section, the the area in compression can be determined. The area being compressed is the gasket material, therefore the bolt area has to be transformed by the ratio,

$$\frac{E_s}{E_c}$$

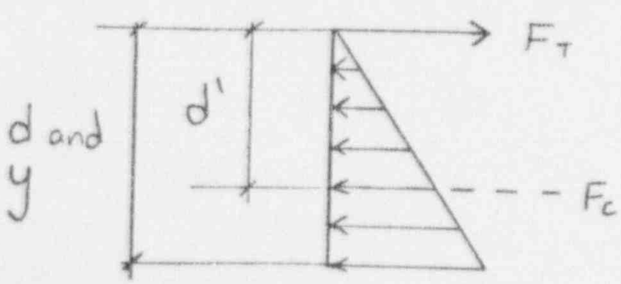
where,

$E_s = 29 \times 10^6$ psi

$E_c = 1 \times 10^3$ psi, modulus of elasticity for gasket material, Chloroprene [Ref 3, Pg 214]

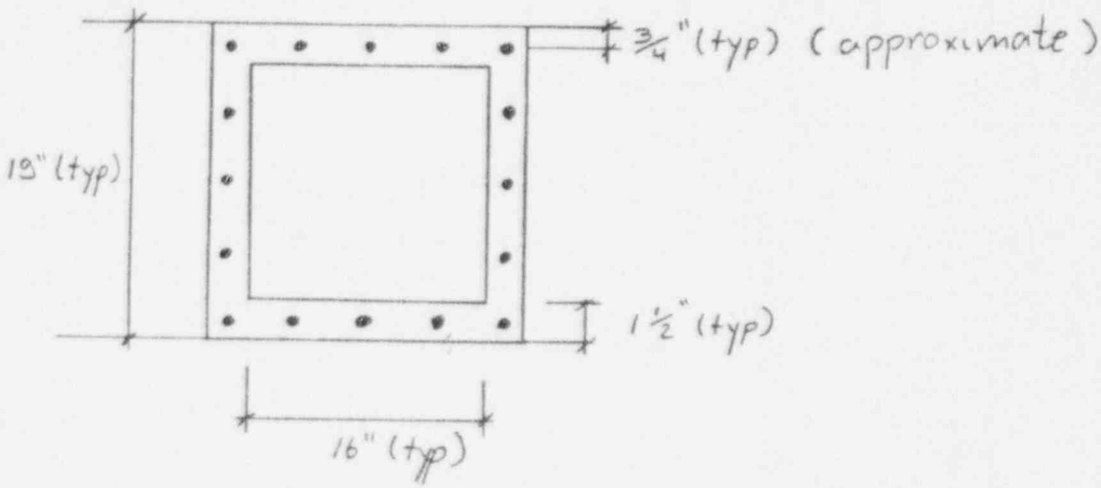
$$\frac{E_s}{E_c} = \frac{29 \times 10^6}{1 \times 10^3} = 29{,}000$$

Because the ratio is so large, the centroid would lie near the tension bolts. Thus the moment arm is minimized by



$$d' = d - \frac{y}{3}$$

Thus, for 16" x 16" duct and L 1½" x 1½" x ¼"



$$d' = \left[\left(16" + 2 \times (1\tfrac{1}{2}")\right) - \tfrac{3}{4}"\right] \frac{2}{3} = 12.17"$$

Similarly for the 14" x 14" duct

$d' = 10.83"$

The maximum tension is determined from,

$$F_T = \frac{M}{nd'}$$

M = bending moment
n = # of bolts in tension

Tension on one bolt,

16" x 16" $\quad F_{T_m} = \dfrac{M}{5\,(12.17)} = .0165\ M$

14" x 14" $\quad F_{T_m} = \dfrac{M}{4\,(10.83)} = .0231\ M$

ROUND SECTION

The tensile loading on a bolt can be estimated using the following approximation,

$$A_{bt} = \pi C t'_b$$

where

A_{bt} = Total bolt area, including compression

C = Bolt circle diameter

t'_b = Equivalent strip width

$$Z = \frac{\pi C^2 t'_b}{4} \qquad Z = \text{section modulus}$$

$$F'_T = \frac{M}{Z} A_{bt} \qquad F'_T = \text{Tension force on full section}$$

Substituting for Abt and Z, we get

$$F'_T = \frac{M}{\frac{\pi C^2 t'_b}{4}} (\pi C t'_b) = \frac{4M}{C}$$

Because the maximum bolt spacing for duct flanges is 4" o.c., the number of bolts in the circular duct connection is

18" ϕ,

$$n = \frac{\pi C}{4} = \frac{\pi(19.5")}{4} = 16 \text{ bolts}$$

17.5" ϕ

$$n = \frac{\pi(19")}{4} = 15 \text{ bolts}$$

Thus, the tension in one bolt is computed from:

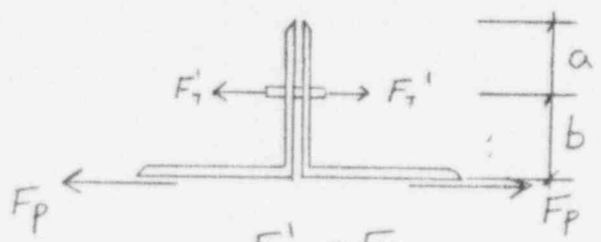
$$F_T = \frac{4M}{nC}$$

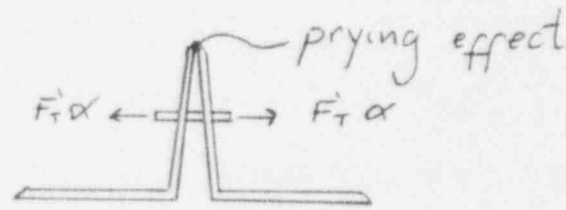
18" ϕ $\quad F_{T_m} = .0129\,M$

17.5 ϕ $\quad F_{T_m} = .0141\,M$

3.3.2 PRESSURE

Internal pressure will introduce longitudinal forces, which have to be taken by the bolts.





$$F_T' = F_p$$

If prying effect due to pressure loading is considered, the increase in bolt tension can be found from

$$F_{T_p} = \frac{F_T'\alpha}{n}$$

where,

$$\alpha = \left(1 + \frac{100bd^2 - 18wt^2/n}{70ad^2 + 21\frac{wt^2}{n}}\right) \qquad [\text{Ref } 4, \text{Pg } 251]$$

b = distance from edge of attachment to bolt

d = bolt diameter

a = distance from bolts in tension to tension edge

w = width of plate

n = number of bolts

t = thickness of plate

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Calculating α for various duct configurations,

DUCT	b	d	a	w	n	t	α
16"x16"	3/4"	3/8"	3/4"	19"	5	1/4"	1.51
14"x14"	3/4"	3/8"	3/4"	17"	4	1/4"	1.44
18" ϕ	3/4"	3/8"	3/4"	30.6"	8	1/8"	2.10
17.5" ϕ	3/4"	3/8"	3/4"	29.8"	7	1/8"	2.06

For circular ducts, 'w' is conservatively estimated as half the circumferential bolt circle length.

The longitudinal tension forces arise from membrane stresses in the duct panel. For circular ducts, this force is easily determined from,

$$F_p = P A_p = P\left(\frac{\pi \cdot D^2}{4}\right)$$

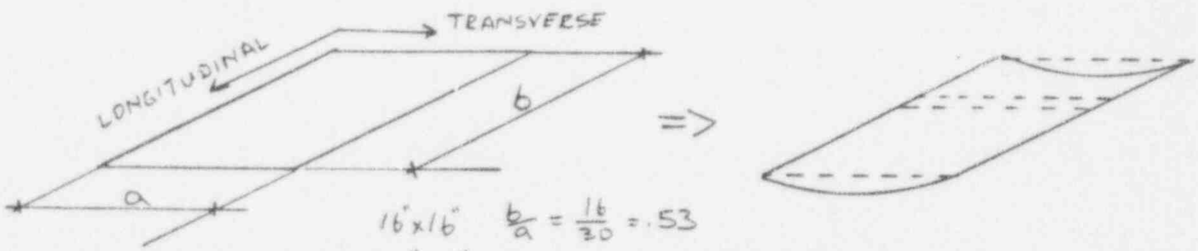
where,

D = diameter of the duct

18" ϕ $\quad F_p = \frac{(.795\,psi)(\pi\,(18)^2)}{4} = 203\ lb$

17.5 ϕ $\quad F_p = \frac{(.795\,psi)(\pi\,(17.5)^2)}{4} = 192\ lb$

For rectangular ducts obtaining, longitudinal forces is a much more complex task. From [Ref 14, Pg 422], plates with aspect ratio $b/a < \frac{2}{3}$ can be considered infinitely long, and therefore can be analyzed as a plate bending into a cylindrical surface.



16"x16" $b/a = \frac{16}{30} = .53$

14"x14" $b/a = \frac{14}{30} = .47$

From above it can be seen that only transverse tensile stresses will be significant. Indeed in most literature the longitudinal tensile stress is not discussed. However, a reasonable estimate of stress can be obtained by calculating the deflection at the center of the plate. From the deflection, the strain and the stress can be estimated.

From [Ref 5, Pg 480], for a simple supported plate,

$$\frac{qb^4}{Et^4}$$

$b = 16''$ or $14''$

$q = .795$ psi

$t = .0478''$

<u>14"x14"</u>

$$\frac{(.795)}{29\times10^6}\cdot\left(\frac{14''}{.0478}\right)^4 = 202$$

<u>16"x16"</u>

$$\frac{(.795)}{29\times10^6}\left(\frac{16}{.0478}\right)^4 = 345$$

The deflection at the center of the plate is obtained from

$$\frac{y}{t}_{14''} = 2.03$$

$$\frac{y}{t}_{16''} = 2.55 \quad \text{(extrapolated)}$$

Thus,

$$y_{14''} = (.0478)(2.03) = .097 \text{ inch}$$

$$y_{16''} = (.0478)(2.55) = .122 \text{ inch}$$

The transverse tensile stress at the center of the plate is,

$$\left(\frac{\sigma_d b^2}{E t^2}\right)_{14''} = 10.9$$

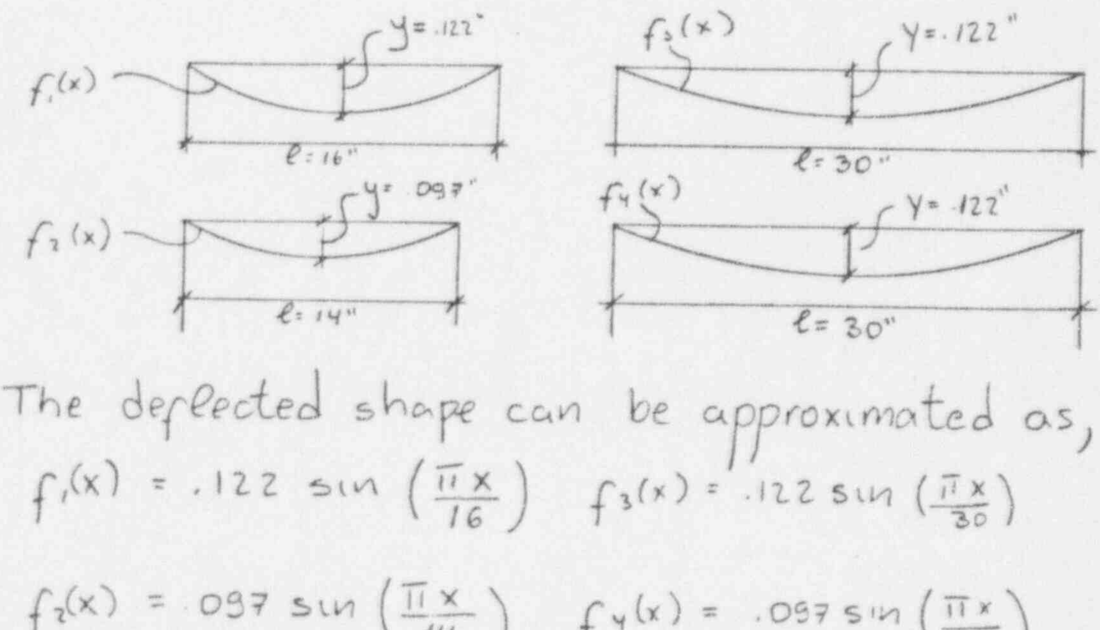
$$\left(\frac{\sigma_d b^2}{E t^2}\right)_{16''} = 18 \quad \text{(extrapolated)}$$

or

$$\sigma_{d\,14''} = \frac{10.9\,(29\times10^6)(.0478)^2}{14^2} = 3685 \text{ psi}$$

$$\sigma_{d\,16''} = \frac{18\,(29\times10^6)(.0478)^2}{16^2} = 4659 \text{ psi}$$

The longitudinal stress can be estimated using the following assumption,



The deflected shape can be approximated as,

$$f_1(x) = .122 \sin\left(\frac{\pi x}{16}\right) \qquad f_3(x) = .122 \sin\left(\frac{\pi x}{30}\right)$$

$$f_2(x) = .097 \sin\left(\frac{\pi x}{14}\right) \qquad f_4(x) = .097 \sin\left(\frac{\pi x}{30}\right)$$

The length of the arc can be found from,

$$L = \int_0^{\ell} \sqrt{1 + \left(\frac{d}{dx} f(x)\right)^2}\, dx$$

The strain can be computed from,

$$\varepsilon = \frac{L - \ell}{\ell} = \frac{\Delta L}{\ell}$$

So, for

$\ell = 16''$

$$L = \int_0^{16} \sqrt{1 + \left(\frac{d}{dx}\left(.122 \sin\left(\frac{\pi x}{16}\right)\right)\right)^2}\, dx$$

$$\varepsilon = 1.434 \times 10^{-4}$$

similarly,

$\ell = 30''$

$\varepsilon_{16''} = 4.08 \times 10^{-5}$ $\qquad$ $\varepsilon_{14''} = 2.579 \times 10^{-5}$

$\ell = 14''$

$$\varepsilon = 1.184 \times 10^{-4}$$

The ratio's are,

$$\frac{\varepsilon_{16''}}{\varepsilon_{30''}} = \frac{1.434 \times 10^{-4}}{4.08 \times 10^{-5}} = 3.515$$

$$\frac{\varepsilon_{14''}}{\varepsilon_{30''}} = \frac{1.184 \times 10^{-4}}{2.579 \times 10^{-5}} = 4.59$$

Knowing the ratio of the strains for the transverse and longitudinal directions, the longitudinal stress can be estimated by,

$$(\sigma_{d\,long})_{16"} = \frac{4659}{3.515} = 1326 \text{ psi}$$

$$(\sigma_{d\,long})_{14"} = \frac{3685}{4.59} = 803 \text{ psi}$$

The total tensile force due to pressure is obtained from,

$$F_P = (\sigma_{d\,long})\, A_{duct}$$

16"x16" $F_P = (1326)(.0478" \times 16" \times 4) = 4056$ lb

14"x14" $F_P = (803)(.0478 \times 14 \times 4) = 2150$ lb

The tension per bolt, including prying, for every section is determined from,

$$F_{T_P} = \frac{F_P\,\alpha}{n} = \frac{F_T'\,\alpha}{n}$$

The results are summarized in Table 3.3.1.

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DUCT	α	F_T'	n	F_{TP}
16"x16"	1.51	4056 lb	16	383 lb
14"x14"	1.44	2150 lb	12	258 lb
18" ϕ	2.10	203 lb	16	27 lb
17.5" ϕ	2.06	192 lb	15	27 lb

TABLE 3.3.1. TENSILE FORCES ON BOLTS DUE TO PRESSURE

3.4 EVALUATION CONDITIONS AND CRITERIA

3.4.1 APPLICABLE CODES AND STANDARDS

AISI - Cold Formed Steel Design Manual, 1989. [Ref 6]

AWS - Structural Welding Code, Steel, 1983. [Ref 7]

AISC - Steel Construction Manual, 9th Edition. [Ref 8]

ANSI/ASME N509 - 1980 [Ref 9]

3.4.2 STRENGTH CRITERIA

The bolt stress allowables are based on the AISC or AISI specifications, depending on thickness of the material being connected.

If thinnest connected part $\geq 3/16"$ AISC

$< 3/16"$ AISI

The welds strength shall be checked against AISI and AWS D1.1 specifications. As discussed in AISI [Ref 6, Pg II-33] commentary, for connected material less than .15 inch in thickness, tearing of the sheet metal is the controlling failure mode.

Stiffener strength allowables are based on ANSI N509-1980, Parag. 5.10.3.3. Which specifies a maximum stress of 0.9 of yield stress for combined loads which include the Safe Shutdown Earthquake, and 0.6 of yield for operation.

3.4.3 DEFLECTION CRITERIA

allowable static deflection of stiffners is based on ANSI N509-1980, Parag. 5.10.3.4, which specifies for flange connections,

1/8" per foot of span, but no more than 3/4".

3.4.4 FLANGE LEAKAGE CRITERIA

The Codes do not specifically address this issue, however a conservative criteria can be developed from an assumption that flange leakage will occur when the bolt tightening stress is overcome by bending, tensile and pressure stresses, introduced on the bolt from external loading.

3.4.5 FATIGUE CRITERIA

Fatigue consideration is based on material properties and fatigue curves found in various literature.

4.1 SYSTEM 2848-3 - 18" φ

BOLT - CHECK

Tensile forces in bolts will arise from, Deadweight, Seismic and Pressure loading.

$$F_{T_{BOLT}} = F_{Tm} + F_{Tp} + F_{TT}$$

where,

F_{Tm} = Tension arising from bending forces

F_{Tp} = Tension arising from pressure forces

F_{TT} = Tension arising from tensile forces

From previous results,

$F_{Tm} = .0129\,M$ (Pg 20)

where

$M = M_{DW} + M_{SEIS}$

$M = 874 + 52730 + 410 = 54014$ in-lb

$F_{Tm} = .0129\,(54014) = 697$ lb

$F_{TT} = \frac{730}{16 \text{ bolts}} = 46$ lb

$F_{Tp} = 27$ lb (Pg 28)

$F_{T_{BOLT}} = 697 + 46 + 27 = 770$ lb

The stress on a bolt is,

$$A_{bolt} = \frac{\pi}{4}(3/8)^2 = .11 \text{ inch}^2$$

$$p_t = \frac{F_{TBOLT}}{A_{bolt}} = \frac{770}{.11} = 7000 \text{ psi}$$

From AISC, Table 1-A, the allowable tensile stress on A307 bolt is

$$P_t = 20,000 \text{ psi}$$

$\therefore$

$$\underline{p_t = 7000 < P_t = 20,000 \text{ psi} \quad O.K}$$

The shear on bolts developes from torsional moment, and shearing forces.

$$F_{v_m} = \frac{zM_x}{nC}$$

$$= \frac{4100 + 1380}{8 \times 19.5"} = 36 \text{ lb}$$

$$F_{v_v} = \frac{121 + 20 + 840}{16} = 62 \text{ lb}$$

$$F_{v\,bolt} = F_{v_m} + F_{v_v} = 36 + 62 = 98 \text{ lb}$$

$$p_v = \frac{98}{.11} = 891 \text{ psi}$$

$P_v = 10,000$ psi AISC, Table 1-D

$$\underline{891 < 10,000 \text{ psi} \quad OK}$$

FLANGE - CHECK

The duct flanges consist of companion angle shapes. For 2848-3 the angles are, $L1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$".

STRESSES

Stresses in flanges arise from internal pressure loading, transmitted in form of hoop and longitudinal stresses.

The pressure stress is determined from

$$p_H = \frac{qr}{t}$$

where,

$q = .795$ psi, internal pressure

$r = 18/2 = 9$"

$t = \frac{1}{8}$", angle leg thickness

$$p_H = \frac{.795(9.0)}{.125} = 58 \text{ psi} \ll .6(36000) = 21{,}600 \text{ psi}$$

There is a possibility that the angle ring may buckle. The critical buckling pressure is determined from,

$q' = \frac{3EI}{r^3}$ [Ref 5, Pg 679, Case 8]

$I = 2 \times .078$ inch^4 = .156 inch^4

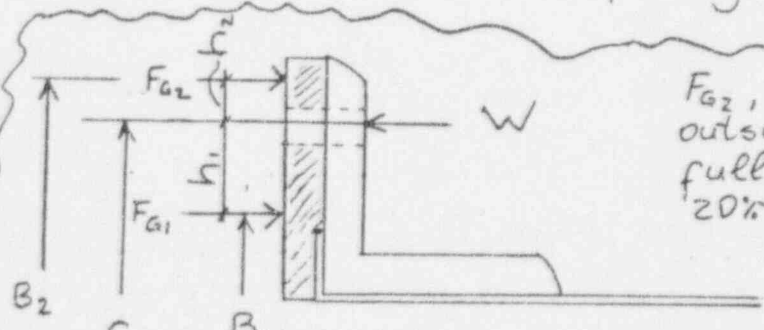
$r = 9.0$ inch

So,

$q' = 18{,}600$ psi >> .795 psi, design pressure

Therefore buckling should not occur in round flanges.

Tension in bolts will introduce stresses in the flange as shown,



F_{G2}, the gasket load outside the bolt is not fully effective. assume 20% effectiveness.

C = 19.5", bolt circle diameter

B_1 = 18.75", gasket center diameter (apprx.)
B_2 = 20.25"

$h = h_1 - \frac{h_2}{5} = \left(\frac{C - B_1}{2}\right) - \left(\frac{B_2 - C}{2(5)}\right) = .3$", moment arm

The tensile load on bolts can be conservatively averaged by

$$W = \frac{F_{TBOLT} \times n}{\pi C} \quad \text{per circumferential inch}$$

$$= \frac{770 \times 16}{\pi (19.5)} = 201 \text{ lb/inch}$$

The moment is,

$$M = Wh = 201 (.30) = 60.3 \text{ in-lb/inch}$$

The stress per inch of flange is

$$P_f = \frac{M t_f}{2 I} = \frac{60.3 (.125)}{2 \frac{(.125)^3 (1)}{12}} = 23,160 \text{ psi}$$

The allowable for earthquake induced loads, per ANSI N509-80, is

$$.9 (P_y) = .9 (36,000) = 32,400$$

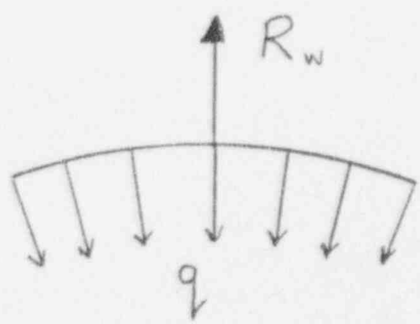
$$P_f = 23,160 < 32,400 \text{ psi} \therefore \text{OK}$$

DEFLECTION

Judging from the stresses produced by internal pressure, flange deflection will be small in round ducts.

WELD - CHECK

The flange angles are brazed to the sheet metal as was shown in section 2.2. The brazing is 1" long and is placed at 8" on center. This braze is effected only by internal pressure as shown,



The braze reaction force is approximated by,

$R_W = q\, d_f$

q = .795 psi

d = 60", tributary duct length, i.e distance between stiffeners

f = 8", length of duct taken by one braze.

$R_w = .795(60)(8) = 382$ lb

The allowable load is determined from AISI, Pg 1-70, which specifies that for a load transverse to the weld the allowable force is,

$R_n = t L F_u$, fillet weld

where

$t = .0478"$, thickness of sheet metal

$L = 1"$, length of braze

$F_u = 58{,}000$ psi, ultimate strength of sheet metal.

$R_n = (.0478)(1)(58{,}000) = 2773$ lb

Dividing by the safety factor

$R_a = \frac{R_n}{\Omega_w}$ $\Omega_w = 2.5$, welding safety factor

$R_a = \frac{2773}{2.5} = \underline{1110 \text{ lb} > R_w = 382 \text{ lb}}$ O.K.

FLANGE LEAKAGE - CHECK

As discussed previously, flange leakage is assumed to occur when the bolt tightening stresses are overcome by tension from various bending tensile and pressure stresses. In terms of forces, the leakage criteria is,

$$F_{mk} - F_m - F_t - F_p = 0$$

where,

$$F_{mk} = \frac{\sigma_{mk} n \pi D_b^2}{4}$$

D_b = 3/8", bolt diameter

n = 16, number of bolts

σ_{mk} = ½ of bolt yield strength or bolt stress that will cause 50% compression in gasket, whichever is smaller. This is a reasonable assumption.

= ½ (36,000) = 18.000 psi

or σ_{mk} equals stress needed to achieve 50% compression of the gasket.

To estimate stress needed to achieve 50% compression, the stiffness of the gasket needs to be calculated.

bolt hole area

$$\text{Area} = \frac{\pi}{4}(21^2 - 18^2) - 16\,\frac{\pi}{4}(4/8)^2$$
$$= 88.7 \ \text{inch}^2$$

[THE DIMENSIONS ARE ESTIMATED]

$K = \dfrac{EA}{L}$,

E = 100-3000 psi, Neoprene [Ref 3, Pg. 214]
use 1000 psi
A = 88.7 psi

$L = 1/8''$, gasket thickness (assumed)

$$K = \frac{(1000)(88.7)}{1/8''} = 7.09 \times 10^5 \ \text{lb/in}$$

Now,

$P = K\Delta$, $\Delta = 1/16''$, 50% compression

$$P = 7.09 \times 10^5 (1/16'') = 44{,}350 \ \text{lb}$$

Per bolt stress,

$$\sigma_{BOLT} = \frac{P}{n A_b} = \frac{44350}{16(.11)} = 25{,}200 \ \text{psi}$$

Considering the range in E of Neoprene, a

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more reliable value of $\frac{1}{2}\sigma_y$ is used. Thus,

$$F_{mk} = \frac{(18,000)(16)(\pi)(3/8)^2}{4}$$

$$= 31,800 \text{ lb}$$

$$F_m = \frac{4M}{C}$$ - from (Pg 19)

$M = 54,014$ in-lb, $M_{DW} + M_{SEIS}$

$$F_m = \frac{4(54,014)}{19.5} = 11,100 \text{ lb}$$

$F_t = 730$ lb

$F_p = 27$ lb $(16) = 432$ lb from (Pg 28)

$F_{mk} - F_m - F_t - F_p$

$31,800 - 11,100 - 730 - 432 =$

$19,540 > 0$ ∴ No Leakage

4.2 SYSTEM 4316-1 14" x 14" ⊞

BOLT - CHECK

The forces producing tension on the bolts are,

$$F_{TBOLT} = F_{Tm} + F_{T_T} + F_{Tp}$$

For the 14" x 14" square duct

$F_{Tm} = .0231M$ (Pg 18)

where

$M = M_{DW} + M_{SEIS}$

$= 317 + 63260 + 170 = 63{,}747$ in-lb

$F_{Tm} = .0231(63{,}747) = \underline{1473 \text{ lb}}$

$F_{T_T} = \dfrac{F_x}{n, \#bolts} = \dfrac{1090}{12} = \underline{91 \text{ lb}}$

$\underline{F_{Tp} = 258 \text{ lb}}$

$F_{TBOLT} = 1473 + 91 + 258 = 1822$ lb

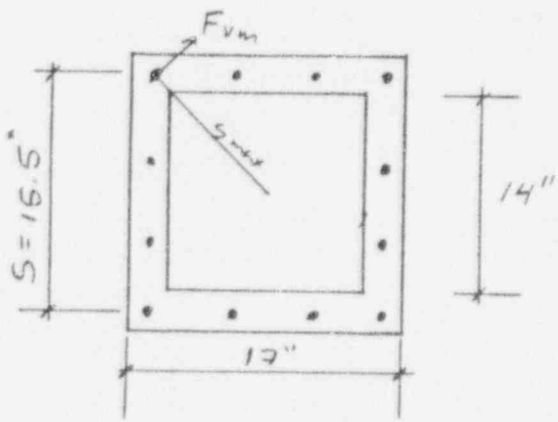
The tensile stress on the critical bolt is

$$p_t = \frac{F_{TBOLT}}{A_{bolt}} = \frac{1822}{.11 \text{ in}^2} = 16{,}570 < 20{,}000 \text{ psi}$$

∴ O.K.

The shear from torsional moment and shear forces is,

$$F_{vm} = \frac{M\, s_{max}}{\Sigma s^2}$$



$$\Sigma s^2 = 8\left(\frac{15.5}{2}\right)^2 + 4\left(\frac{\sqrt{2}\,15.5}{2}\right)^2 = 961 \text{ in}^2$$

$$F_{vm} = \frac{(144+40)\left(\frac{\sqrt{2}\,15.5}{2}\right)}{961} = 2 \text{ lb}$$

$$F_{vv} = \frac{39 + 37 + 1856}{12} = 161 \text{ lb}$$

$$F_{v_{BOLT}} = F_{vm} + F_{vv} = 161 + 2 = 163 \text{ lb}$$

The stress,

$$p_v = \frac{163}{.11} = 1482 \text{ psi} < 10{,}000 \text{ psi} \quad \text{O.K}$$

Assuming a linear tension and shear interaction,

$$\frac{p_t}{P_t} + \frac{p_v}{P_t} < 1$$

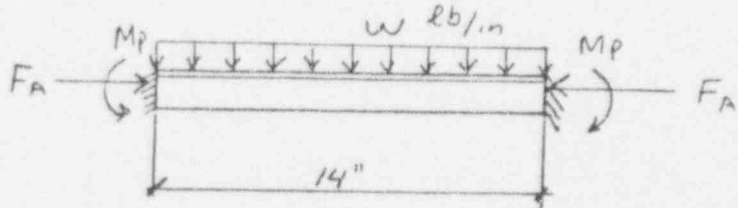
$$\frac{16,570}{20,000} + \frac{1482}{10,000} = 0.977 < 1 \quad O.K.$$

FLANGE - CHECK

For the 14"x14" duct in system 4316-1, the companion angles are made up of L 1½" x 1½" x ¼".

STRESS

Stresses in flanges arise from internal pressure loading, acting over tributary area.



self weight

$$w = (.795\,psi)(30") + .39 = 24.3 \ lb/in$$

$$M_p = \frac{wL^2}{12} = \frac{24.3(14)^2}{12} = 397 \ in\text{-}lb$$

The stress is,

$$p_b = \frac{M_p}{Z} + \frac{F_A}{A}$$

$$F_A = \frac{wL}{2} = \frac{24.3(14)}{2} = 170.1$$

where,

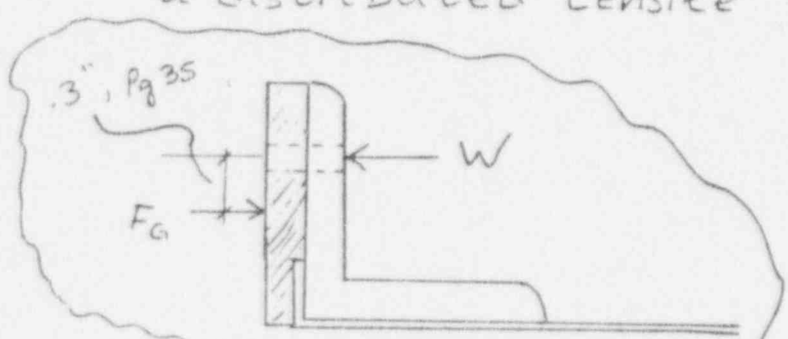
$Z = .262 \text{ in}^3$, section modulus for 2 L $1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{4}$"

$A = 2((1.5 + 1.25) \times .25) = 1.375 \text{ in}^2$

$$P_b = \frac{397}{.262} + \frac{170.1}{1.375} = 1640 \text{ psi} < 21{,}600 \text{ psi}$$

$\therefore$ OK

The stress on the flange from the bolts is estimated by assuming a distributed tensile load,



$$W = \frac{F_{T\,[illegible]}\, nt}{L_f}$$

$nt = 4$, # bolts taking tension

$L_f = 17"$, full length of flange

$$W = \frac{1822 \times 4}{17} = 429 \text{ lb/inch}$$

The moment per inch of flange,

$M = wh \quad h = .3''$

$= 429 \times .3'' = 129$ in-lb/inch

The stress is,

$$p_f = \frac{M t_f}{2I} = \frac{129\ (.25)}{2 \times \frac{(.25)^3(1)}{12}}$$

$p_f = 12{,}390$ psi $< 32{,}400$ psi $\therefore$ O.K.

DEFLECTION

For deflection, conservatively assume the ends of the flange are pinned,

$$\Delta_{max} = \frac{5wL^4}{384EI} \qquad I = .278\ in^4, \text{ for 2 angle sections}$$

$$\Delta_{max} = \frac{5\ (24.3)\ (14)^4}{384\ (29 \times 10^6)(.278)} = .0015 \text{ inch}$$

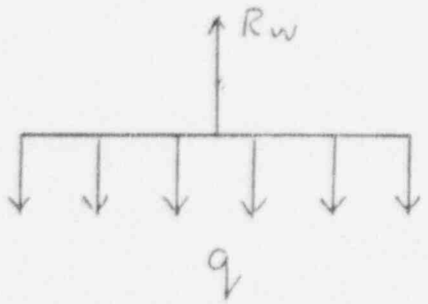
The allowable deflection is $\frac{1}{8}''$ per foot of span. Thus,

$\Delta_{all} = \frac{1}{8}\left(\frac{14}{12}\right) = .146$ inch $> .0015$ inch

$\therefore$ OK

WELD - CHECK

The flange angles are brazed to the sheet metal by 1" long brazes at 8 inches on center.



The weld reaction force "Rw" is determined from.

$R_w = q d f$

q = .795 psi, internal pressure
d = 30 inch, tributary length
f = 8 inch, length of duct taken by one weld

$R_w = (.795)(30)(8) = 191$ lb

The allowable l: d is calculated from,

$$R_n = t L F_u$$

$$= (.0478)(1)(58{,}000) = 2773 \text{ lb}$$

$$R_a = \frac{R_n}{\Omega_w} = \frac{2773}{2.5} = 1110 \text{ lb}$$

Thus,

$$R_a = 1110 \text{ lb} > R_w = 191 \text{ lb} \therefore \text{O.K.}$$

FLANGE LEAKAGE - CHECK

Flange leakage will occur when the following criteria is met,

$$F_{mu} - F_m - F_t - F_p = 0$$

where,

$$F_{mu} = \frac{\sigma_{mu} \, n_t \, \pi D_b^2}{4}$$

σ_{mu} = 18,000 psi, bolt make up stress

n_t = 12, # of bolts

D_b = 3/8", bolt diameter

$F_m = F_{TBOLT} \times n_t$ $n_t = 4$, # bolts taking tension

$F_{TBOLT} = 1822$ lb, for 14"x14" duct in system 4316-1

$F_p = F_{TP} \times n_t$

$F_{TP} = 258$ lb, for 14"x14" duct

Now,

$$F_{mk} = \frac{(18{,}000)(12)\,\pi\,(3/8)^2}{4} = 23{,}860 \text{ lb}$$

$F_m = 1822 \times 4 = 7288$ lb

$F_p = 258 \times 12 = 3096$ lb

$F_t = 1090$ lb

$23860 - 7288 - 3096 - 1090 = 12386 > 0$

$\therefore$ No leakage

4.3 SYSTEM 4316-6

4.3.1 17.5" ⌀

BOLT - CHECK

The forces producing tension in the bolts are,

$$F_{T\,BOLT} = F_{Tm} + F_{TT} + F_{TP}$$

For the 17.5" ⌀ duct,

$F_{Tm} = .0141M$ (Pg 20)

where,

$$M = M_{DW} + M_{SEIS}$$

$$= 605 + 9750 = 10{,}355 \text{ in-lb}$$

$$F_{TT} = \frac{F_x}{n,\ \#\text{ bolts}} = \frac{495}{15} = 33 \text{ lb}$$

$F_{TP} = 27$ lb (Pg 28)

$$F_{T\,BOLT} = .0141(10{,}355) + 33 + 27 = 206 \text{ lb}$$

The tensile stress on the critical bolt is,

$$pt = \frac{F_{T\,BOLT}}{A_{bolt}} = \frac{206}{.11} = 1873 \text{ psi} < 20{,}000 \text{ psi}$$

$\therefore$ OK

The shear from torsional moment and shear forces is,

$$F_{vm} = \frac{2M_x}{nC} = \frac{2(0)}{15(19)} = 0$$

$$F_{vv} = \frac{303+302}{15} = 40.4\,lb$$

$$F_{v\,BOLT} = F_{vm} + F_{vv} = 40.4\,lb$$

The stress,

$$p_v = \frac{40.4}{.11} = 367\,psi < 10{,}000\,psi \;\therefore OK$$

Because stresses are so low, no need to check interaction.

FLANGE - CHECK

For the 17.5" ϕ duct in system 4316-6, the companion angles are made up of L1½" x 1½" x 1/8"

STRESS

The stress due to pressure is determined from,

$$p_H = \frac{qr}{t_f}$$

$$p_H = \frac{.795\left(\frac{17.5}{2}\right)}{.125} = 56\,psi \ll 21{,}600\,psi$$

$\therefore$ O.K.

As was shown earlier, buckling is not a factor.

The stress on the flange from the tensile force on the bolts is estimated from,

$$W = \frac{F_{TBOLT} \times n}{\pi C} = \frac{206 \times 15}{\pi (19)} = 52\ lb/inch$$

The moment,

$$M = Wh = 52(.3) = 15.6\ in\text{-}lb/inch$$

The stress per inch of flange is,

$$p_f = \frac{M t_f}{2I} = \frac{15.6(.125)}{2\,\frac{(.125)^3(1)}{12}}$$

$$p_f = 5990\ psi < 32{,}400\ psi \ \therefore\ OK$$

DEFLECTION

As discussed previously, deflection of round sections is small.

WELD - CHECK

The flange angles are brazed to the sheet metal by 1" long brazes at 8 inches on center.

The weld reaction force 'Rw' was found to be,

$R_w = 382$ lb (Pg 38)

The allowable is also the same as before,

$R_a = 1110$ lb

Thus,

$R_w = 382 \text{ lb} < 1110 \text{ lb} \therefore$ OK.

FLANGE LEAKAGE - CHECK

Flange leakage will occur when the following criteria is satisfied,

$$F_{mk} - F_m - F_t - F_p = 0$$

where,

$$F_{mk} = \frac{(18,000)(15)\pi(3/8)^2}{4} = 29,821 \text{ lb}$$

$$F_m = \frac{4(10,355 \text{ in-lb})}{19} = 2180 \text{ lb}$$

$$F_t = 495 \text{ lb}$$

$$F_p = 27 \times 15 = 405 \text{ lb}$$

So,

$$29,821 - 2180 - 495 - 405 = 26,741 \text{ lb} > 0$$

$\therefore$ No leakage

4.3.2 16" x 16" ⊞

BOLT - CHECK

The forces producing tension on the bolts are,

$$F_{T\,BOLT} = F_{Tm} + F_{TT} + F_{TP}$$

For the 16"x16" square duct,

$F_{Tm} = .0165M$ (Pg 18)

where,

$M = M_{DW} + M_{SEIS}$

$= 58 + 15 + 10870 + 140 = 11,083$ in-lb

$F_{Tm} = .0165\,(11,083) = 183$ lb

$F_{TT} = \dfrac{F_x}{n, \#\ \text{bolts}} = \dfrac{625}{16} = 39$ lb

$F_{TP} = 383$ lb (Pg 28)

$F_{TBOLT} = 183 + 39 + 383 = 605$ lb

The tensile stress on the critical bolt is;

$$pt = \frac{F_{TBOLT}}{A_{bolt}} = \frac{605}{.11} = 5500\ psi < 20,000\ psi$$

∴ O.K.

The shear from torsional bending and shear forces is,

$$F_{V_m} = \frac{M_{s\,max}}{\Sigma s^2}$$

$$\Sigma s^2 = 12\left(\frac{17.5}{2}\right)^2 + 4\left(\frac{\sqrt{2}\,17.5}{2}\right)^2 = 1531 \text{ in}^2$$

$$F_{V_m} = \frac{(309+210)\left(\sqrt{2}\,\frac{17.5}{2}\right)}{1531} \cong 5 \text{ lb}$$

$$F_{V_V} = \frac{62+39+302}{16} = 26 \text{ lb}$$

$$F_{V_{BOLT}} = F_{V_m} + F_{V_V} = 5 + 26 = 31 \text{ lb}$$

The stress on a bolt,

$$p_v = \frac{31}{.11} = 282 \text{ psi} < 10{,}000 \text{ psi} \quad \text{O.K.}$$

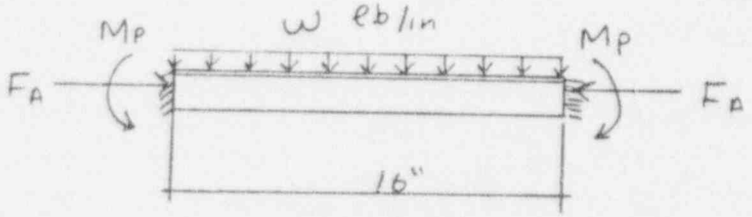
again, since tensile and shear stresses are very low, there is no need to check interaction.

FLANGE - CHECK

For the 16"x16" duct in system 4316-6, the companion angles are made up of L1½"x1½"x1/4".

STRESS

Stresses in flanges arise from internal pressure loading, acting over tributary area.



$$w = (.795)(30) + .39 = 24.3 \text{ lb/inch}$$

$$M_p = \frac{wL^2}{12} = \frac{24.3\,(16)^2}{12} = 519 \text{ in-lb}$$

$$F_A = \frac{wL}{2} = 195 \text{ lb}$$, reaction from the other flange angle

The stress is,

$$P_b = \frac{M_p}{Z} + \frac{F_A}{A}$$

where,

$Z = .262\ in^3$, section modulus

$A = 1.375\ in^2$

$$P_b = \frac{519}{.262} + \frac{195}{1.375} = 2130\ psi < 21{,}600\ psi$$

$\therefore$ OK

The stress on the flange from the bolts is calculated from,

$$W = \frac{F_{TBOLT} \times nt}{L_f}$$

$nt = 5$, # bolts taking tension

$L_f = 19''$, full length of flange

$$W = \frac{605 \times 5}{19} = 160\ lb/inch$$

The moment per inch of flange

$$M = Wh = 160(.3) = 48\ in\text{-}lb/inch$$

The stress is,

$$P_f = \frac{M t_f}{2 I_f} = \frac{48(.25)}{2\,\frac{(.25)^3(1)}{12}}$$

$$P_f = 4610\ psi < 32{,}400\ psi \quad O.K.$$

DEFLECTION

For deflection, conservatively assume the ends of the flange are pinned,

$$\Delta_{max} = \frac{5wL^4}{384EI}$$

$I = .278 \text{ in}^4$, for 2 angle sections

$$\Delta_{max} = \frac{5(24.3)(16)^4}{384(29\times10^6)(.278)} = .0026 \text{ inch}$$

The allowable deflection is,

$$\Delta_{all} = \frac{1}{8}\left(\frac{16}{12}\right) = .16 \text{ inch}$$

Thus,

$\Delta_{max} = .0026 \text{ inch} < .16 \text{ inch} \therefore$ O.K.

WELD - CHECK

As calculated previously the load acting on the weld is below the allowable,

$R_w = 191 \text{ lb} < R_a = 1110 \text{ lb} \therefore$ O.K.

FLANGE LEAKAGE - CHECK

Flange leakage will occur when the following criteria is satisfied,

$$F_{mk} - F_m - F_t - F_p = 0$$

where,

$$F_{mk} = \frac{(18000)(16)\pi(3/8)^2}{4} = 31,808 \text{ lb}$$

$$F_m = F_{TBOLT} \times n_t = 605 \times 5 = 3025 \text{ lb}$$

$$F_p = 383 \times 16 = 6128 \text{ lb}$$

$$F_t = 625 \text{ lb}$$

Thus,

$$31,808 - 3025 - 6128 - 625$$

$$= 22,030 > 0 \quad \therefore \text{ No leakage}$$

4.4 SYSTEM 4316-7

4.4.1 18" ϕ

BOLT - CHECK

The forces producing tension on the bolts are,

$$F_{TBOLT} = F_{Tm} + F_{TT} + F_{TP}$$

For the 18" ϕ duct,

$F_{Tm} = .0129M$ (Pg 20)

where,

$M = M_{OW} + M_{SEIS}$

$= 598 + 6 + 72210 + 260 = 73074$ in-lb

$F_{Tm} = .0129\,(73074) = 943$ lb

$F_{TT} = \dfrac{F_x}{n, \# \text{bolts}} = \dfrac{1428}{16} = 90$ lb

$F_{TP} = 27$ lb (Pg 28)

$F_{TBOLT} = 943 + 90 + 27 = 1060$ lb

The tensile stress on the critical bolt is,

$Pt = \dfrac{F_{TBOLT}}{A_{bolt}} = \dfrac{1060}{.11} = 9636\text{ psi} < 20{,}000\text{ psi}$

$\therefore$ OK

The shear from torsional bending and shear forces is,

$$F_{Vm} = \frac{2M_x}{nC} = \frac{2(331+230)}{16\ (19.5)} \cong 4\ lb$$

$$F_{Vv} = \frac{29+29+468}{16} = 33\ lb$$

$$F_{V BOLT} = 4+33 = 37\ lb$$

The stress on bolt,

$$p_v = \frac{37}{.11} = 337\ psi < 10{,}000\ psi \quad \therefore OK$$

Because stresses are low, no need to check interaction.

FLANGE - CHECK

For the 18"ϕ duct in system 4316-6, the companion angles are made up of L1½"x1½"x⅛"

STRESS

The stress on an angle due to pressure is,

$$p_H = \frac{qr}{t_f}$$

$$P_H = \frac{.795\left(\frac{18}{2}\right)}{.125} = 58 \text{ psi} << 21{,}600 \text{ psi}$$

$\therefore$ OK

The stress on the flange from the tensile force on the bolts is estimated from,

$$W = \frac{F_{TBOLT,i} \times n}{\pi C} = \frac{1060 \times 16}{\pi\,(19.5)} = 276.8 \text{ lb/in}$$

The moment is,

$$M = Wh = 276.8(.3) = 83.1 \text{ in-lb/inch}$$

The stress per inch of flange is,

$$P_f = \frac{Mt_f}{2I} = \frac{83.1\,(.125)}{2\,\frac{(.125)^3(1)}{12}}$$

$$P_f = 31{,}910 < 32{,}400 \text{ psi} \quad \therefore \text{OK}$$

DEFLECTION

As mentioned previously, deflection of round sections is small.

WELD - CHECK

The flange angles are brazed to the sheet metal by 1" long brazes at intervals of 8" on center.

The weld reaction force 'Rw' was found to be,

$R_w = 382$ lb

The allowable is also the same as before,

$R_a = 1110$ lb

Thus

$R_w = 382$ lb < 1110 lb $\therefore$ OK

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FLANGE LEAKAGE - CHECK

Flange leakage will occur when the following criteria is satisfied,

$$F_{mk} - F_m - F_t - F_p = 0$$

where,

$$F_{mk} = \frac{(18,000)(16)\pi(3/8)^2}{4} = 31,808 \text{ lb}$$

$$F_m = \frac{4(73074)}{19.5} = 14,990 \text{ lb}$$

$$F_t = 1428 \text{ lb}$$

$$F_p = 27 \times 16 = 432 \text{ lb}$$

So,

$$31,680 - 14,990 - 1428 - 432 = 14958 > 0$$

$\therefore$ OK

4.4.2 16" x 16" ⊞

BOLT - CHECK

The forces producing tension on the bolts are,

$$F_{TBOLT} = F_{Tm} + F_{TT} + F_{TP}$$

For the 16"x16" square duct,

$F_{Tm} = .0165M$ (Pg 18)

where,

$$M = M_{DW} + M_{SEIS}$$

$$= 180 + 20 + 67{,}700 = 67{,}900 \text{ in-lb}$$

$$F_{Tm} = .0165\,(67{,}900) = 1120 \text{ lb}$$

$$F_{TT} = \frac{F_x}{n,\ \#\,\text{bolts}} = \frac{877}{16} = 55 \text{ lb}$$

$F_{TP} = 383$ lb (Pg 28)

$$F_{TBOLT} = 1120 + 55 + 383 = 1558 \text{ lb}$$

The tensile stress on the critical bolt is,

$$pt = \frac{F_{TBOLT}}{A_{bolt}} = \frac{1558}{.11} = 14{,}164 \text{ psi} < 20{,}000 \text{ psi}$$

∴ OK

The shear from torsional bending and shear forces is,

$$F_{vm} = \frac{M\, s_{max}}{\Sigma s^2}$$

$\Sigma s^2 = 1531 \text{ in}^2$, as was calculated earlier.

$$F_{vm} = \frac{(20+0)\left(\frac{\sqrt{2}\times 17.5}{2}\right)}{1531} \cong 1 \text{ lb}$$

$$F_{vv} = \frac{39+681}{16} = 45 \text{ lb}$$

$$F_{v_{BOLT}} = 45+1 = 46 \text{ lb}$$

The stress on a bolt,

$$p_v = \frac{46}{.11} = 420 \text{ psi} < 10{,}000 \text{ psi} \therefore \text{OK}$$

Checking interaction

$$\frac{14{,}164}{20{,}000} + \frac{420}{10{,}000} = .75 < 1 \therefore \text{OK}$$

FLANGE - CHECK

For the 16"x16" duct in system 43:6-7, the companion angles are made up of L1½"x1½"x1/4".

STRESS

The weight per inch acting on the flanges is,

$$w = (.795)(30) + \overset{\text{self wt}}{.39} = 24.3 \text{ lb/inch}$$

The moments and the forces are,

$$M_P = \frac{wL^2}{12} = \frac{24.3\,(16)^2}{12} = 519 \text{ in-lb}$$

$$F_A = \frac{wL}{2} = \frac{24.3\,(16)}{2} = 195 \text{ lb}$$

The stress is found from,

$$P_b = \frac{M_P}{Z} + \frac{F_A}{A}$$

where,

$Z = .262 \text{ in}^3$

$A = 1.375 \text{ inch}^2$

$$p_b = \frac{519}{.262} + \frac{195}{1.375} = 2130 < 21{,}600 \text{ psi}$$

$\therefore$ OK

The stress on the flange from the bolts is calculated from,

$$W = \frac{F_{TBOLT} \times n_t}{L_f}$$

$n_t = 5$, # bolts taking tension

$L_f = 19''$, full length of flange

$$W = \frac{1558 \times 5}{19} = 410 \text{ lb/inch}$$

The moment per inch of flange,

$$M = Wh = 410\,(.3) = 123 \text{ in-lb/inch}$$

The stress is,

$$p_f = \frac{Mt_f}{2I} = \frac{123\,(.25)}{2\;\frac{(.25)^3}{12}}$$

$$p_f = 11{,}810 \text{ psi} < 32{,}400 \text{ psi} \quad \therefore \text{OK}$$

DEFLECTION

As was calculated for other 16"x16", the allowable deflection criteria is met, as shown on Pg 59

$\Delta_{max} = .0026" < .16" = \Delta_{all}$ ∴ OK

WELD - CHECK

as calculated previously on Pg 47, the load acting on the weld is below the allowable,

$R_w = 191$ lb $< R_a = 1110$ lb ∴ OK

FLANGE LEAKAGE - CHECK

Flange leakage criteria is,

$$F_{mk} - F_m - F_t - F_p = 0$$

where,

$$F_{mk} = \frac{(18,000)(16)\pi(3/8)^2}{4} = 31,808 \text{ lb}$$

$$F_m = F_{TBOLT} \times n_t = 1558 \times 5 = 7790 \text{ lb}$$

$$F_p = 383 \times 16 = 6128$$

$$F_t = 877 \text{ lb}$$

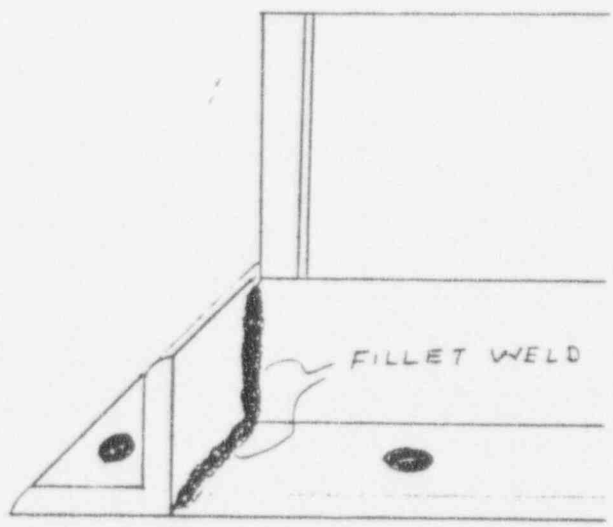
Thus,

$$31,808 - 7790 - 6128 - 877$$

$$= 17,013 > 0 \quad \therefore \text{No leakage}$$

4.5 ANGLE TO ANGLE WELD - CHECK

In the rectangular ducts, the angle irons, comprising the flange, are connected as shown below,

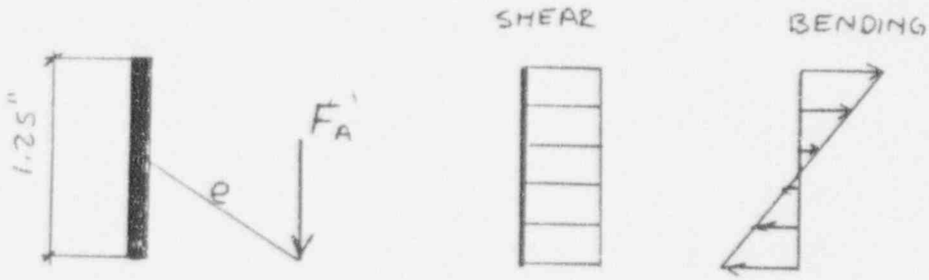


For $L 1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{4}$ angles, the length of weld on each leg is

$\ell = 1\frac{1}{2} - \frac{1}{4} = 1\frac{1}{4}"$

For conservatism, it can be assumed that minimum allowable weld thickness is $\frac{1}{8}"$, as recommended by AISC, section J2.2.b. for $\frac{1}{4}"$ thickness of material joined.

Assuming only one leg of the weld is effective in resisting bending, the loading on weld can be idealized as follows,



The 16"x16" duct will experience the largest force 'F', which comes from internal pressure bending the flange. As was determined earlier the moments and the forces at the edges of the flanges are,

M_p = 519 in-lb (Pg 57)

F_A = 195 lb

Because the flange consists of two angles, the load on each angle is,

$$M_p' = \frac{519}{2} = 260 \text{ in-lb}$$

$$F_A' = \frac{195}{2} = 98 \text{ lb}$$

The load on the weld is,

$$R_v = \frac{F_A'}{\ell} = \frac{98}{1.25} = 78.4 \text{ lb/inch}$$

$$R_b = \frac{M_p' c}{I}$$

where

$$c = \frac{1.25}{2} = .625''$$

$$I = \frac{1.25^3}{12} = .1627 \text{ in}^4\text{/inch}$$, moment of inertia per inch of thickness of weld

$$R_b = \frac{260(.625)}{.1627} = 998.4 \text{ lb}$$

The resultant force is,

$$R_n = \sqrt{(R_b)^2 + (R_v)^2} = \sqrt{78.4^2 + 998.4^2}$$

$$R_n = 1002 \text{ lb/inch}$$

The allowable weld strength is,

$$R_{nw} = .3 t_w \ell F_{EXX}$$

or

$$R_{nw} = .4 t_m \ell F_y$$

$t_w = \frac{1}{8}"$, $t_m = \frac{1}{4}"$, $\ell = 1.25"$

$F_{EXX} = 70{,}000$ psi , design strength for weld per [Ref 10 , SFA-5.1]

$F_y = 36{,}000$ psi , for base metal

Thus the allowable is the minimum of,

$R_{nw} = .3(\frac{1}{8})(.707)(1.25)(70^k) = 2320$ lb

$R_{nw} = .4(\frac{1}{4})(1.25)(36^k) = 4500$ lb

So

$R_n = 1002 < 2320$ lb $\therefore$ OK

4.6 FATIGUE CONSIDERATION

FOR SEISMIC ANALYSES, FATIGUE EVALUATION IS USUALLY NOT PERFORMED. HOWEVER, DUE TO AN ADJACENT FAN, FAN START UP WILL CAUSE CYCLIC LOADING ON THE DUCTS DUE TO INTERNAL PRESSURE.

BOLT FATIGUE

THE MAXIMUM EXPECTED BOLT FORCE CAUSED BY INTERNAL PRESSURE (FAN START-UP) IS:

$$F_T = 383 \text{ LB} \quad [\text{SEC } 3.3.1 \quad \text{PG } 28]$$

BOLT DIA = 3/8"

BOLT AREA, A_B = 0.078 IN² (MIN AREA) [AISC, 9TH ED, 4-147]

BOLT STRESS, $\sigma_B = F_T/A_B = 383 \text{ LBS}/0.078 \text{ IN}^2$

$$\sigma_B = 4910 \text{ psi}$$

THE MINIMUM EXPECTED BOLT FORCE WOULD BE DUE TO NO INTERNAL PRESSURE, AND THE LOADING WOULD BE ONLY FROM PRELOADING OF THE BOLT. SINCE THE ACTUAL PRELOAD IS NOT AVAILABLE, THE MAXIMUM CONSERVATIVE CYCLIC LOADING WOULD BE:

STRESS AMPLITUDE, $S_a = \left(\frac{\sigma_B - 0}{2}\right)$

$$S_a = \frac{4.911 - 0}{2} \text{ KSI}$$

$$S_a = 2455 \text{ psi} \quad \text{(BOLT)}$$

THE FANS ARE STARTED ABOUT TWICE MONTHLY [11]

THE LIFETIME EXPECTED START UP N_L:

N_L = 2 PER MONTH (12 MONTHS PER YEAR)(40 YEARS PER PLANT)

= 960 cycles

$N_L < 1 \times 10^3$ CYCLES

ASSUMING THE LOWEST GRADE OF MATERIAL FOR BOLTS, 2455 psi IS WELL BELOW THE ENDURANCE LIMIT FOR CARBON STEEL. FIGURE 4.6.1 [12, FIG I-9.1] DEMONSTRATES FATIGUE CURVES FOR CARBON, LOW ALLOY, AND HIGH TENSILE STEELS.

FROM THE FATIGUE CURVE IT IS OBVIOUS FATIGUE IS NOT A CONCERN.

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HA-05/94-0819
Revision 1

CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 12 DATE: MAY '94 PAGE: 78

SUBJECT: ANALYSIS BY: LLV CK: MAK SHT: 47 OF 49

1986 Edition APPENDIX 1 Fig. I-9.1

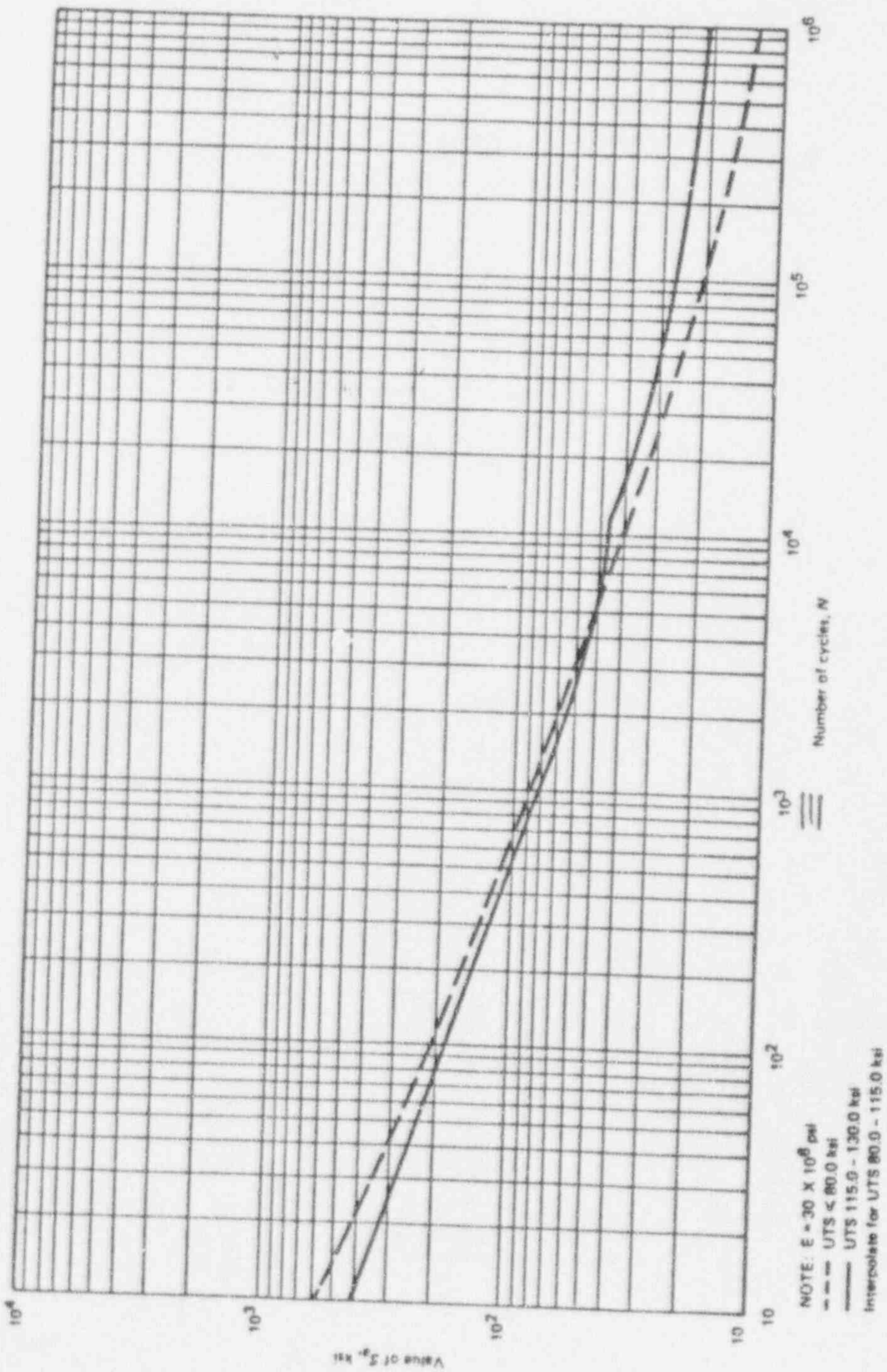

FIG. I-9.1 DESIGN FATIGUE CURVES FOR CARBON, LOW ALLOY, AND HIGH TENSILE STEELS FOR METAL TEMPERATURES NOT EXCEEDING 700°F

Table I-9.1 Contains Tabulated Values and a Formula for Accurate Interpolation of These Curves

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FIGURE 4.6.1

[12]

DUCTING FATIGUE

SIMILARLY, FOR DUCTING FATIGUE, (BOUNDING ANALYSIS)

$$S_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

FROM ANSI/ASME N509-80, THE MAXIMUM EXPECTED STRESS WOULD BE $0.6\,S_y$ AND THE MINIMUM STRESS WOULD BE ZERO.

LET $\sigma_{max} = 0.6 S_y = 0.6(36\,KSI) = 21.6\,KSI$

$\sigma_{min} = 0$

$$S_a = \frac{21.6 - 0}{2}$$

$$= \frac{21.6}{2}\ KSI$$

$$S_a = 10.8\ KSI \quad (\text{DUCTING})$$

AGAIN, FROM FIGURE 4.6.1, AT $N = 1 \times 10^3$ CYCLES, AND $S_a = 10.8$ KSI, DUCTING FATIGUE IS NOT A CONCERN.

NORMALLY, FATIGUE IS A CONCERN FOR $N_L > 10^4$. FOR $N_L < 10^4$, (LOW CYCLE FATIGUE - LCF) FATIGUE IS ONLY A CONCERN WHEN STRESSES EXCEED YIELD. FOR NORMAL OPERATION, INCLUDING THE FAN OPERATION, THE DUCTING SHOULD BE QUALIFIED BELOW YIELD, AND THEREFORE THE FEW FAN CYCLES WILL NOT AFFECT THE DESIGN OR LONGEVITY OF THE DUCTING.

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 12 DATE: MAY '94 PAGE: 80

SUBJECT: ANALYSIS BY: LLV CK: MJK SHT: 49 OF 49

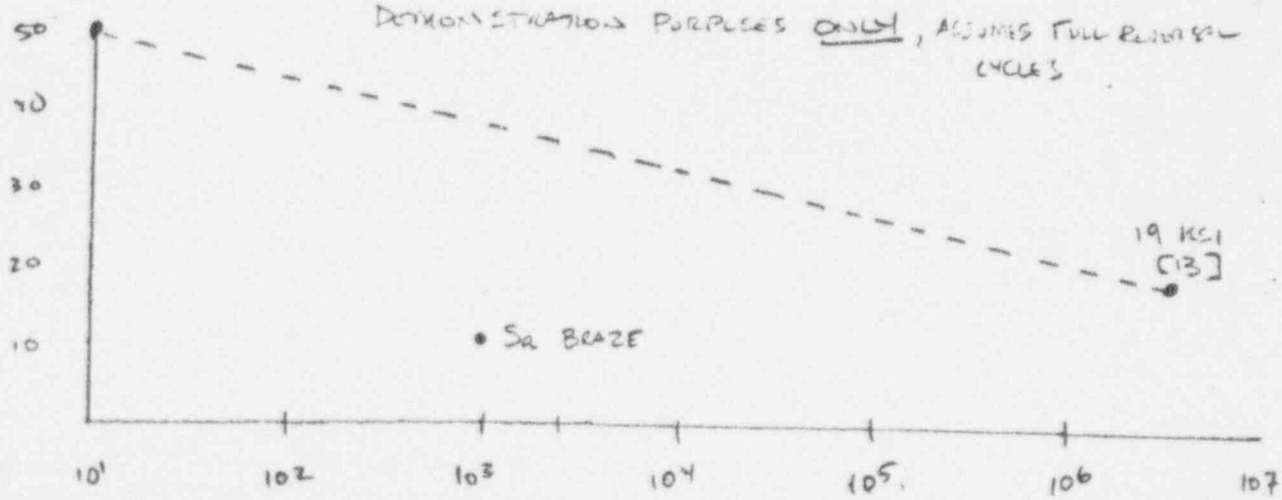
BRAZE FATIGUE

SIMILARLY FOR WELD FATIGUE, THE ALTERNATING STRESS CAN BE ASSUMED TO BE THE SAME AS THE DUCTING; S_a = 10.8 KSI. THE DUCTING SPEC CALLOUT FOR THE BRAZED JOINT IS ECuSi AWS A5.6 [1]. THE ASME BOILER AND PRESSURE VESSEL CODE, SECTION II, PART C - MATERIALS INCLUDES SFA 5.6 WHICH IS IDENTICAL TO AWS A5.6.

SFA 5.6 GIVES AWS MECHANICAL PROPERTY REQUIREMENTS FOR ECuSi. THE MIN TENSILE STRENGTH IS 50 KSI. AN ADDITIONAL REFERENCE [13 PG 22-36] GIVES A FATIGUE STRENGTH FOR ECuSi (SILICON BRONZE) AT ROOM TEMPERATURE

S_a = 19 KSI FOR 300 x 10^6 CYCLES

FOR N_L < 1000, FATIGUE OF THE BRAZE JOINT IS NOT A CONCERN. FIG 4.6.2



4.6.2 DEMONSTATIVE FATIGUE CURVE FOR SILICON BRONZE

CONCLUSIONS

In response to a U.S.N.R.C. Docket 50-341, the structural integrity, air tightness and fatigue strength of transverse joints of duct systems 2848-3, 4316-1, 4316-6 and 4316-7 was investigated.

It was found that all components of the transverse joints meet the allowable strength requirements as set forth by applicable codes. It was also determined that a conservative flange leakage criteria was satisfied by all systems. Meaning, air tightness of the transverse joint will not be violated. Based on the expected number of loading cycles on the ducting systems, fatigue life of the transverse joint is not expected to be compromised.

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 12 DATE: MAY '94 PAGE: 82

SUBJECT: 6.0 REFERENCES BY: A.R. CK: MAL SHT: 1 OF 3

REFERENCES

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2. Hopper & Associates, "Structural Evaluation of Ducting Systems 2848-3, 4316-1, 4316-6, 4316-7", Client: DECo Fermi-2, HA-05/89-686, May, 1989.

3. Materials Engineering, "1992 Materials Selector", A Penton Publication, December 1991.

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6. AISI, "Cold Formed Steel Design Manual", American Iron and Steel Institute, 1989.

7. AWS, "Structural Welding Code - Steel", D1.1, American Welding Society, 1983.

8. AISC, "Steel Construction Manual", 9th Edition, American Institute of Steel Construction, 1989.

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10. ASME, "1992 Boiler & Pressure Vessel Code - Materials, Part C.", American Society of Mechanical Engineers, 1992.

11. Telecon, Philip Hasrouni with Dave Jax, DECo Fermi 2, May 2, 1994.

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14. Timoshenko, "Theory of Plates and Shells", 2nd Edition, 1959.

15. Desai, S.C. et. al. "Structural Testing of Seismic Category I HVAC Duct Specimens", Civil Engineering and Nuclear Power, Vol. 1, ASCE, 1980.

Attachment 9 to
NRC-94-0050

HOPPER AND ASSOCIATES
ENGINEERS

DECO - USNRC DOCKET NO. 50-341

FERMI 2 CCHVAC DUCTING SYSTEMS

CONCERN ITEM NO. 13

EVALUATION CALCULATIONS

Prepared for: Detroit Edison Company
Nuclear Engineering Fermi 2
6400 North Dixie Highway
Newport, MI 48166

Prepared by: Hopper and Associates
300 Vista Del Mar
Redondo Beach, CA 90277

May, 1994

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PREPARED BY: [signature] VENDEMIA 5-11-94

VERIFIED BY: M. Amir Khan 12MAY94

ALL OTHER SECTIONS;

PREPARED BY: Alex Reizman 5-11-94

VERIFIED BY: M. Amir Khan 12MAY94

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 13 DATE: MAY '94 PAGE: 1

SUBJECT: 1.0 INTRODUCTION BY: LLV CK: MAK SHT: 1 OF 3

1.1 PROBLEM STATEMENT

ON MARCH 3, 1994, THE U.S.N.R.C SUBMITTED DOCKET NO 50-341 TO DETROIT EDISON COMPANY. THIS DOCKET CONSISTS OF A REQUEST FOR ADDITIONAL INFORMATION ON FERMI 2 CCHVAC SYSTEM DESIGN AND OPERATION. IN PARTICULAR, THERE ARE SEVERAL CONCERNS REGARDING THE STRUCTURAL INTEGRITY CALCULATIONS, HA-05/89-686 AND HA-09/89-696, PERFORMED BY HOPPER AND ASSOCIATES.

THIS PACKAGE IS PREPARED IN ORDER TO RESPOND TO ONE OF THE COMMENTS POSED BY THE NRC REGARDING HA-05/89-686. NAMELY,

"Corner Lap" or "Pittsburgh Lock" type corners and middle longitudinal "Acme Lock" type seams structural integrity was not evaluated. Therefore, the air tightness of the ducts cannot be assured.

1.2 INVESTIGATION APPROACH

THE "PITTSBURGH LOCK" CORNER SEAM AND THE "ACME LOCK" TYPE SEAM OF SYSTEMS ANALYZED IN HA-05/89-686 WILL BE REVIEWED. THE STRUCTURAL INTEGRITY OF THE SEAMS WILL BE EVALUATED FROM LOADS FROM COMPUTER RUNS PERFORMED BY HOPPER AND ASSOCIATES. ADDITIONALLY, AIR TIGHTNESS AND FATIGUE CAPACITY WILL BE EVALUATED

1.3 RESULT SUMMARY

Based upon calculations performed in this package it was determined that longitudinal "Pittsburgh Lock" and "ACME Lock" seams in systems 2848-3, 4316-1, 4316-6 and 4316-7 are structurally adequate to withstand operational and seismic loading, without exceeding the applicable Code allowable stresses, and without compromising air tightness.

SYSTEM DESCRIPTION

FOUR DUCTING SYSTEMS WERE EVALUATED IN HA-05/89-686;

2848-3

THIS SYSTEM IS LOCATED IN THE AUXILIARY BUILDING AT AN ELEVATION OF 690'-3". THE SYSTEM CONSISTS OF 18" DIAMETER ROUND DUCTS.

4316-1

THIS SYSTEM IS LOCATED IN THE AUXILIARY BUILDING AT ELEVATION 690'-6". THE SYSTEM CONSISTS OF 14" x 14" SQUARE DUCTS.

4316-6

THIS SYSTEM IS LOCATED IN THE AUXILIARY BUILDING AT ELEVATION 690'-8". THE SYSTEM CONSISTS OF 16"x16" SQUARE DUCTS AND 17.5" DIAMETER ROUND DUCTS.

4316-7

THIS SYSTEM IS LOCATED IN THE AUXILIARY BUILDING AT ELEVATION 690'-3". THE SYSTEM CONSISTS OF 16"x16" SQUARE DUCTS AND 18" DIAMETER ROUND DUCTS.

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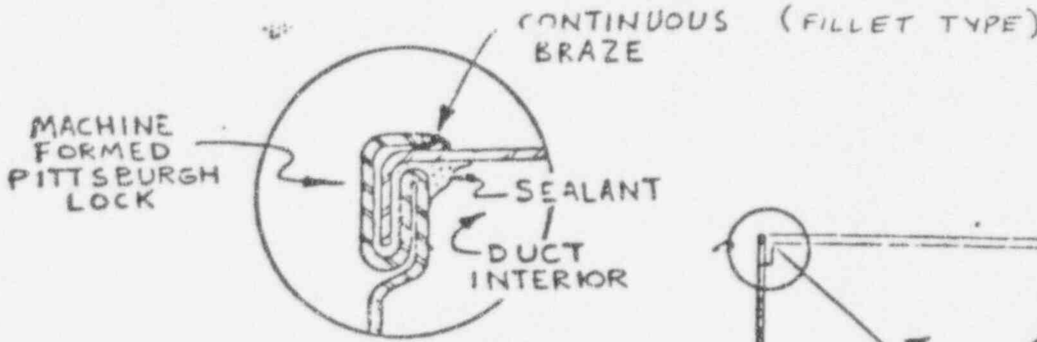
CALCULATION SHEET

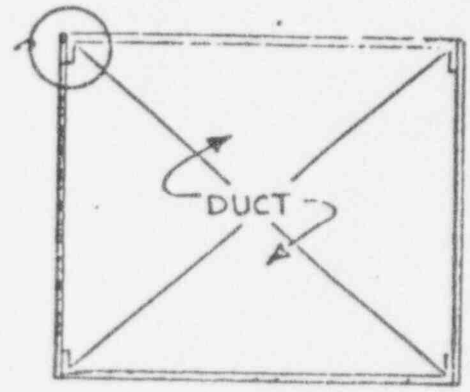
TITLE: RESPONSE TO CONCERN 13 DATE: MAY '94 PAGE: 5

SUBJECT: 2.0 SYSTEM DESCRIPTION BY: LLV CK: MAK SHT: 2 OF 3

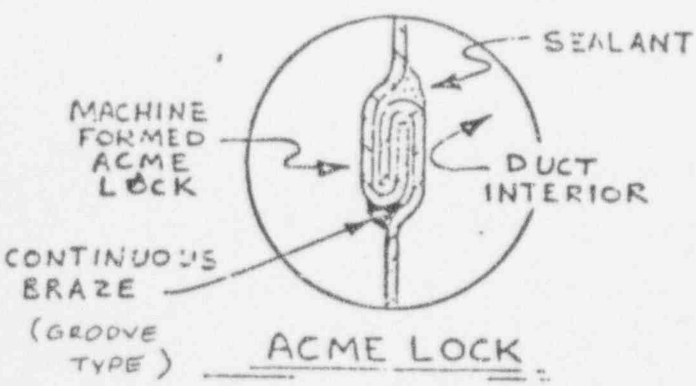
2.1 LONGITUDINAL SEAMS [REF 1, pg DB-4]



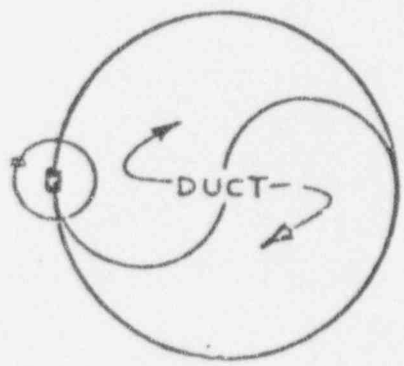
PITTSBURGH LOCK



TYPICAL RECTANGULAR DUCT



ACME LOCK



TYPICAL ROUND DUCT

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 13 DATE: MAY '94 PAGE: 6

SUBJECT: 2.0 SYSTEM DESCRIPTION BY: LLV CK: MAK SHT: 3 OF 3

2.2 MATERIALS [REF 1, PP. DB-2, 2A]

SHEET METAL	A-527 (G-90)
SEALANT	G.E. SILICONE # 1200
BRAZE	SILICON BRONZE ECuSi AWS-A5.6

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 13 DATE: MAY '94 PAGE: 7

SUBJECT: 3.0 ANALYSIS APPROACH BY: A.R CK: MAK SHT: 1 OF 18

3.1 ASSUMPTIONS

1. The AISI and AWS Codes are applicable to ductwork design. The applicability of this assumption is duscussed in [Ref 10, Pg 11-3.1]
2. Material A527 has the same mechanical strength properties as carbon steel A36.
3. Structural integrity of duct sheet material is properly evaluated in HA-05/89-686 [Ref 2].
4. Braze thickness is equal to the thickness of the thinner material being connected times 0.707. This is done to account for effective throat.
5. The brazed joint is qualified to Code.

3.2 LOADS

THE LOADS DUE TO DEADWEIGHT ARE OBTAINED FROM COMPUTER RUNS PERFORMED IN HA-05/89-686 [REF 2]. THE SEISMIC INDUCED FORCES ARE OBTAINED FROM A SITE SPECIFIC EARTHQUAKE (SSE) WITH 2% DAMPING. THE COMPUTER RESULTS ARE BASED ON THE GALVANIZED SHEET METAL THICKNESS.

TO ESTIMATE THE FORCES AND MOMENTS WHICH RESULT BASED ON A BARE METAL THICKNESS, ALL FORCES REPORTED IN HA-05/89-686 ARE MULTIPLIED BY A CONSERVATIVE FACTOR OF 1.1 (10% INCREASE). THE INCREASE ACCOUNTS FOR THE FREQUENCY SHIFT DUE TO THE ADDED FLEXIBILITY.

THE DESIGN PRESSURE IN ALL DUCT SYSTEMS IS 22 INCH H_2O OR 0.795 psi.

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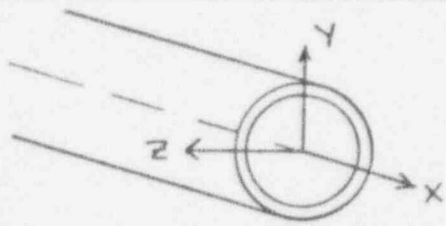
CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 13 DATE: APR 94 PAGE: 9

SUBJECT: ANALYSIS APPROACH BY: A.R. CK: MAK SHT: 3 OF 18

DUCT SYSTEM Z848-3, 18"φ		CRITICAL LOCATION: ELEMENT 30 NODE 30
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	Ø	730
F_y	-121	20
F_z	Ø	840
M_x	-4100	1380
M_y	Ø	52730
M_z	-874	410



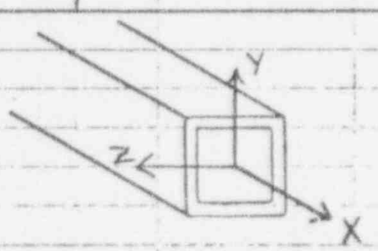
FORCES : LBS

MOMENTS : IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.1 LOADS FOR SYSTEM Z848-3

DUCT SYSTEM 4316-1, 14"□		CRITICAL LOCATION: ELEMENT 21 NODE 22
	SELF WEIGHT	SSE @ 2% DAMP. *
F_x	Ø	1050
F_y	-39	37
F_z	Ø	1856
M_x	-144	40
M_y	Ø	63260
M_z	-317	170



FORCES; LBS

MOMENTS; IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.2 LOADS FOR SYSTEM 4316-1

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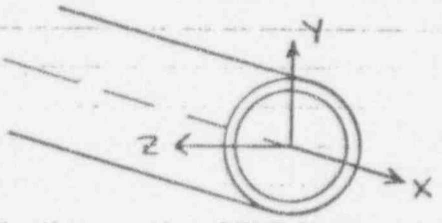
CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 13 DATE: APR 94 PAGE: 11

SUBJECT: ANALYSIS APPROACH BY: A.R. CK: MAK SHT: 5 OF 18

DUCT SYSTEM 4316-6, 17.5"Ø		CRITICAL LOCATION: ELEMENT 50 NODE 47
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	Ø	495
F_y	-303	Ø
F_z	Ø	302
M_x	Ø	Ø
M_y	Ø	9750
M_z	605	Ø



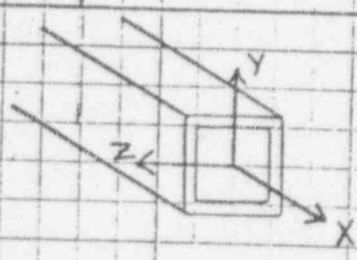
FORCES : LBS

MOMENTS : IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.3 LOADS FOR SYSTEM 4316-6

DUCT SYSTEM 4316-6, 16"□		CRITICAL LOCATION: ELEMENT 27 NODE 13
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	Ø	625
F_y	-62	39
F_z	Ø	302
M_x	-309	210
M_y	15	10870
M_z	58	140



FORCES: LBS
MOMENTS: IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.3 CONT'D

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CALCULATION SHEET

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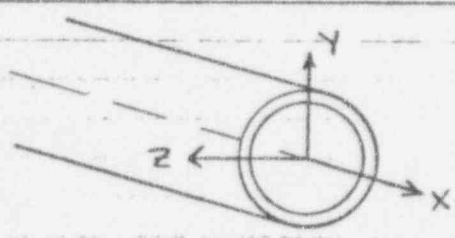
DUCT SYSTEM 4316-7 18" ∅		CRITICAL LOCATION: ELEMENT 53 NODE 730
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	0	0
F_y	491	0
F_z	0	3035
M_x	692	0
M_y	3	5649
M_z	205	0



FORCES : LBS

MOMENTS: IN-LBS

* Values include 1.1 frequency adjustment factor

TABLE 3.2.4 LOADS FOR SYSTEM 4316-7

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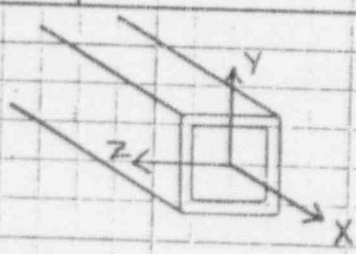
DUCT SYSTEM 4316-7 16" □		CRITICAL LOCATION: ELEMENT 42 NODE 7.5
	SELF WEIGHT	SSE @ 2% DAMP.*
F_x	0	829
F_y	105	[illegible]
F_z	0	2371
M_x	80	0
M_y	10	47923
M_z	384	0



FORCES; LBS

MOMENTS: IN-LBS

* Values include 1.1 frequency adjustment factor

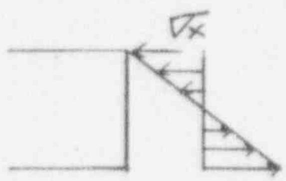
TABLE 3.2.4 CONT'D

3.3 ANALYTICAL IDEALIZATION

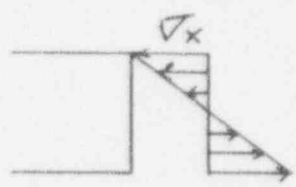
The loads acting on a duct cross-section and the corresponding stress distribution is as shown below.

3.3.1 EARTHQUAKE + DEADWEIGHT

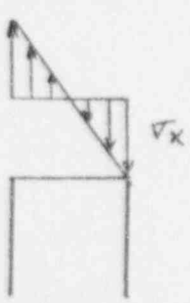
- BENDING 1



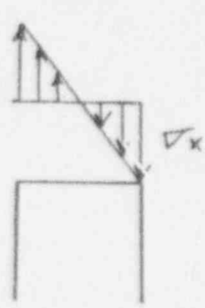
$$\sigma_x = \frac{M_z C}{I}$$



- BENDING 2



$$\sigma_x = \frac{M_y C}{I}$$



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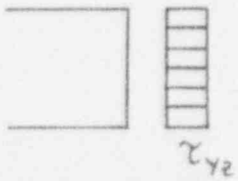
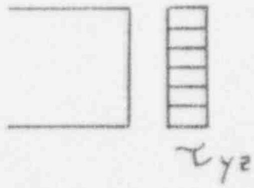
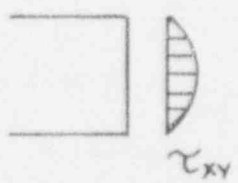
- TORSION



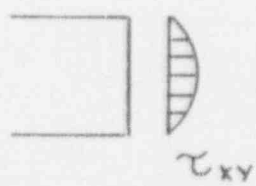
$$\tau_{yz} = \frac{Tc}{J}$$



- SHEAR 1



$$\tau_{xy} = \frac{V_y Q}{It}$$



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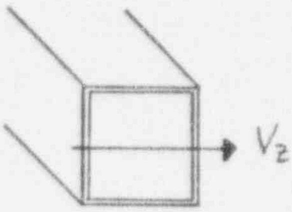
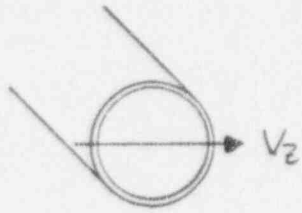
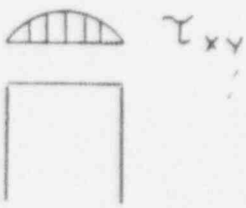
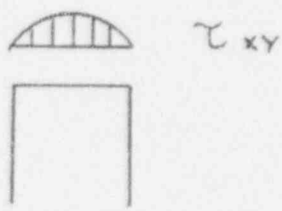
- SHEAR Z



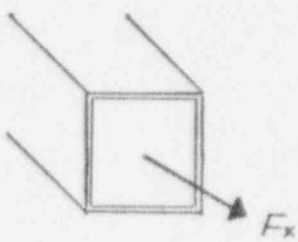


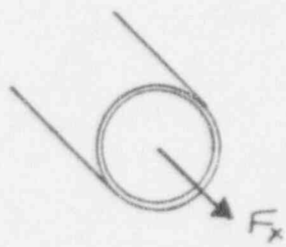


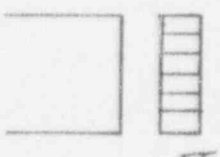
$$\tau_{xy} = \frac{V_z Q}{I t}$$



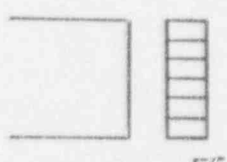
- AXIAL







$$\sigma_x = \frac{F_x}{A}$$



I = Moment of Inertia for section

J = Polar Moment of Inertia

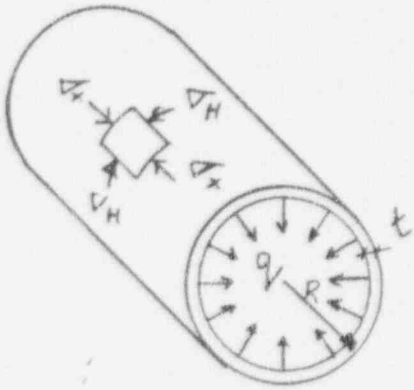
Q = Moment of Area

A = Cross-Sectional area

t = Thickness of cut section

3.3.2 PRESSURE

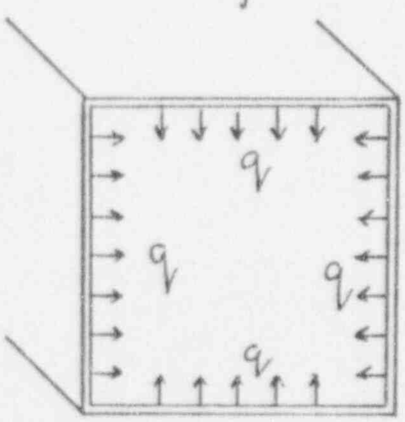
ROUND



$$\sigma_x = \frac{qR}{2t} \qquad \sigma_H = \frac{qR}{t}$$

SQUARE

Square ducts are idealized as an assembelage of plates loaded by a uniformely distributed pressure.

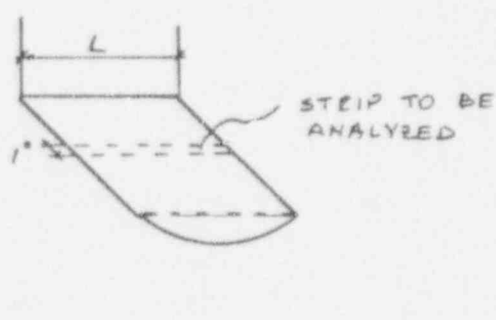


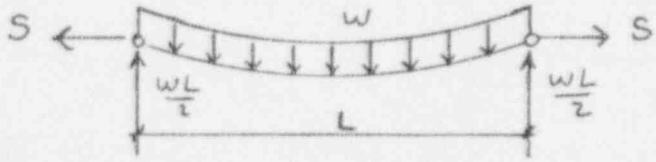
The length to width ratio of these plates is

$$\frac{14}{30} = .47$$

$$\frac{16}{30} = .53$$

From [Ref 3, Pg 422], for length to width ratio of < 2/3, the solution is very close to those obtained for an infinitely long plate. Therefore, the duct panels can be analyzed as long plates bent to a cylindrical surface.





The stresses in the panel consist of two components; bending and membrane. The influence of each depends on the boundary condition. The total stress in the panel, i.e. bending plus membrane, is maximized when fixed boundary is assumed. However, a simply supported boundary will produce the largest membrane stresses alone. Because stresses in the braze depend only on membrane stresses, a simply supported boundary will produce the most conservative result.

From [Ref 4, Pg 480], the membrane stress is determined as shown,

$$\frac{qb^4}{Et^4}$$

where,

q = .795 psi, maximum pressure

b = 16" or 14", duct width

E = 29×10^6 psi

t = .0478", panel thickness (Gage 18)

Thus,

$$\frac{(.795)(16)^4}{(29\times10^6)(.0478)^4} = 344 \qquad (16")$$

$$\frac{(.795)(14)^4}{(29\times10^6)(.0478)^4} = 202 \qquad (14")$$

From the table,

$$\sigma_m = 17.5\frac{Et^2}{b^2} \qquad (16")$$

$$\sigma_m = 10.9\frac{Et^2}{b^2} \qquad (14")$$

$$\sigma_m = \frac{17.5\,(29\times10^6)(.0478)^2}{16^2} = 4530 \text{ psi} \qquad (16")$$

$$\sigma_m = \frac{10.9\,(29\times10^6)(.0478)^2}{14^2} = 3685 \text{ psi} \qquad (14")$$

3.4 EVALUATION CONDITIONS AND CRITERIA

3.4.1 APPLICABLE CODES AND STANDARDS

AISI - Cold Formed Steel Design Manual [Ref 5]

AWS - Structural Welding Code - Steel, D1.1, 1983 [Ref 6]

- Sheet Metal Welding Code D9.1, 1990. [Ref 7]

ANSI/ASME N509-80 [Ref 8]

3.4.2 BRAZE STRENGTH CRITERIA

In providing the requirements for tubular structures, the structural systems most similar to ducts, the AWS code specifies the following.

TYPE OF WELD	STRESS	ALLOWABLE STRESS
FILLET AND GROOVE WELD	Tension or compression parallel to axis of weld	Same as for base metal
	Shear stress on effective throat	$.3 F_{EXX}$ (weld) $.4 F_y$ (base metal)

For thin sheet the AISI Code is more applicable. As discussed in AISI [Ref 5, Pg II-33] Commentary, for connected material less than 0.15 inch in thickness, tearing of the sheet metal is the controlling failure mode. The specified allowable stress for fillet weld is,

LONGITUDINAL - FILLET : $\frac{.75 F_u}{2.5} = .3 F_u$

- FLARE GROOVE : " $= .3 F_u$

TRANSVERSE - FILLET : $\frac{F_u}{2.5} = .4 F_u$

- FLARE GROOVE : $\frac{.833 F_u}{2.5} = .33 F_u$

The conclusions drawn in AISI commentary are based on welded rather than brazed connections. However, a typical code like the ASME, B&PV Code, section IX [Ref 9, Pg 183], specifies that for a brazed connection to be considered acceptable, the base metal should brake, before the braze. This acceptance criteria is identical to that for a weld. Thus the AISI conclusions appear applicable for a brazed connection.

The allowable stresses on the base metal are governed by ANSI N509-1980, Parag. 5.10.3.3, which specifies a maximum stress of 0.9 of yield stress for combined loads which include the the Safe Shutdown Earthquake, and 0.6 of yield for normal plant operations.

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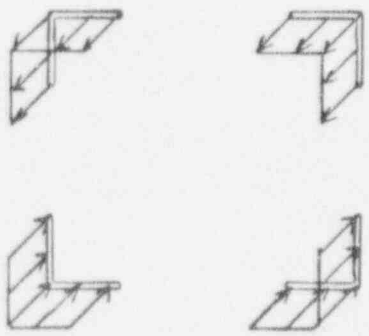
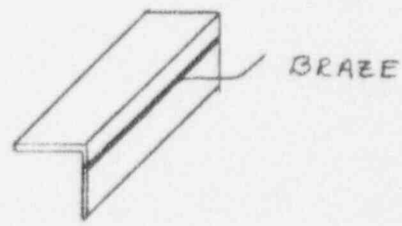
4.1 DEADWEIGHT AND SEISMIC LOADING

The bending moments arising from deadweight and seismic loading, introduce longitudinal stresses on the ducts. For rectangular ducts, these stresses are assumed acting on the corners of the duct only,





The corners are nearly under a pure compressive and tensile stress from bending. Similarly the same is true for the round duct. Therefore the braze is under direct tension or compression from the bending moments. Of course, for axial loading the braze is again under pure tension or compression.

For tension or compression parallel to axis of the weld, as is the case in longitudinal seams, the AWS Code states,

"Groove and fillet welds parallel to the longitudinal axis of tension or compression members, except in connection areas, are not considered as transferring stress and hence may take the same stress as that in the base metal, regardless of the electrode (filler metal) classification."
[Ref 6, Pg 161, Note 3]

Because the base metal was checked, for the above described loading of two bending moments and axial force, in HA-05/89-686 [Ref 2], the determined stresses will not be recalculated in this package.

The remaining forces that do effect the braze are; torsional moment, the shearing forces in the two directions and internal pressure.

TORSIONAL MOMENT

The shear stress due to torsion is determined from,

- ROUND

$$\tau = \frac{M_t R}{J} \qquad M_t = M_{x\,SEIS} + M_{x\,DW}$$

$R = 9"$ or $8.75"$, radius of round duct

$J = 2I = 2(\pi R^3 t)$

$J_{9"} = 2(\pi(9)^3(.0478)) = 219 \text{ in}^4 \quad (9")$

$J_{8.75} = 2(\pi(8.75)^3(.0478)) = 201 \text{ in}^4 \quad (8.75")$

Therefore

$\tau = \frac{9}{219} M_t = .0411\, M_t \quad (18" \phi)$

$\tau = \frac{8.75 M_t}{201} = .0436\, M_t \quad (17.5" \phi)$

The maximum shearing stress due to torsion is,

DUCT SIZE	SYSTEM	M_t	τ_{max}
18" ϕ	2848-3	5480 in-lb	226 psi
17.5" ϕ	4316-6	0	0 psi

- SQUARE

$$\tau = \frac{M_t\, c}{J}$$

c = 8" or 7", half the duct width

$$J = \frac{4A^2 t}{S}, \quad S = \text{perimeter of duct}$$

$$J_{8"} = \frac{4\,(16 \times 16)^2\,(.0478)}{4 \times 16} = 195.8 \text{ in}^4$$

$$J_{7"} = \frac{4\,(14 \times 14)^2\,(.0478)}{4 \times 14} = 131.1 \text{ in}^4$$

Therefore,

$$\tau = \frac{8\,M_t}{195.8} = .0409\,M_t \qquad (16" \times 16")$$

$$\tau = \frac{7\,M_t}{131.1} = .0534\,M_t \qquad (14" \times 14")$$

The maximum shearing stress on a square duct due to torsion is,

DUCT SIZE	SYSTEM	Mt	τ_{max}
16"x16"	4316-6	519 in-lb	22 psi
14"x14"	4316-1	184 in-lb	10 psi

SHEARING FORCES

The shearing stress due to shearing forces is determined from,

$$\tau = \frac{VQ}{It}$$

where,

$$V = (V_{SEIS} + V_{DW})_Y + (V_{SEIS} + V_{DW})_Z$$

- ROUND

$t = .0478"$, duct thickness

$I = \pi R^3 t = 109.5 \text{ in}^4$ (18" Ø)

$= 100.6 \text{ in}^4$ (17.5" Ø)

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$$Q = A\bar{y}$$

$$A = \pi R t$$

$$\bar{y} = \frac{4}{3\pi} \frac{R^3 - R_i^3}{R^2 - R_i^2}$$

DUCT	R	R_i	t	A (in²)	$\bar{y}$ (in)	Q (in³)
18" ϕ	9"	8.952"	.0478"	1.351	5.714	7.72
17.5 ϕ	8.75"	8.702"	.0478"	1.314	5.556	7.3

The maximum shear stress on the round duct due to shear forces,

DUCT SIZE	SYSTEM	V	τ_{max}
18" ϕ	4316-7	3084 lb	2276 psi
17.5 ϕ	4316-6	605 lb	460 psi

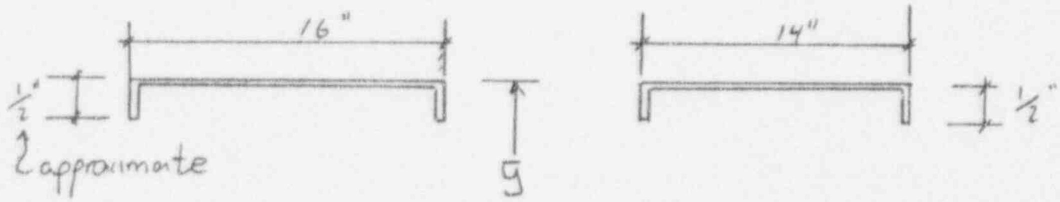
- SQUARE

$t = .0478"$

$I = 129.4 \text{ in}^4$ $(16" \times 16")$

$= 86.6 \text{ in}^4$ $(14" \times 14")$

$Q = A\bar{y}$



$A = (16 \times .0478) + 2(.5)(.0478)$ $A = (14 \times .0478) + 2(.5)(.0478)$

$= .8126 \text{ in}^2$ $= .717 \text{ in}^2$

$\bar{y} = 8"$ (conservative) $\bar{y} = 7"$

$Q_{16} = 6.5 \text{ in}^3$ $Q_{14} = 5.02 \text{ in}^3$

The maximum shear stress is

DUCT SIZE	SYSTEM	V	τ_{max}
16" x 16"	4316-7	2476 lb	1304 psi
14" x 14"	4316-1	1932 lb	1172 psi

4.2 PRESSURE

Negative pressure loading introduces compressive stresses in a round duct and tensile stresses on the panels of the square duct.

ROUND

For the round duct the stresses in the hoop direction are determined from,

$$\sigma_H = \frac{qR}{t}$$

where,

$q = .795$ psi

$R = 9"$ or $8.75"$

$t = .0478"$

So,

$$\sigma_H = \frac{.795\,(9)}{.0478} = 150 \text{ psi} \qquad (18"\ \phi)$$

$$\sigma_H = \frac{.795\,(8.75)}{.0478} = 146 \text{ psi} \qquad (17.5"\ \phi)$$

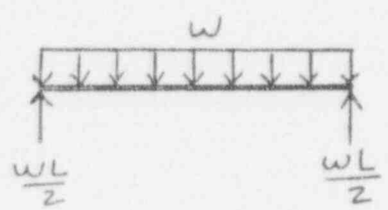
SQUARE

The tensile membrane stresses in the square ducts were determined from Pg 21, to be

σ_m = 4530 psi (16" x 16")

σ_m = 3685 psi (14" x 14")

In addition, the pressure introduces shear forces on the panels,



$V_p = \frac{wL}{2}$

$w = .795$ lb/in

L = 16" or 14"

$V_p = \frac{.795\,(16)}{2} = 6.4$ lb (16" x 16")

$V_p = \frac{.795\,(14)}{2} = 5.6$ lb (14" x 14")

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The shear stress is determined from

$$\tau_p = \frac{3 V_p}{2 A}$$

where,

$A = 1 \times .0478 = .0478 \text{ in}^2$, area of strip

$$\tau_p = \frac{3\,(6.4)}{2\,(.0478)} = 201 \text{ psi} \qquad (16" \times 16")$$

$$\tau_p = \frac{3\,(5.6)}{2\,(.0478)} = 176 \text{ psi} \qquad (14" \times 14")$$

4.3 BRAZE - CHECK

As discussed previously the stresses on the braze are due to torsional moment and shear forces introduced by Seismic and Deadweight loading, and also due to internal negative pressure. Combining all these stresses we get the total stress on the material at the braze. To get the stress on the braze, the stress on the material is multiplied by $\sqrt{2}$ to account for effective throat on the braze. The combined stresses are shown below,

DUCT SIZE	SYSTEM*	MATERIAL τ (psi)	MATERIAL σ (psi)	BRAZE τ (psi)	BRAZE σ (psi)
18" φ	4316-7	2502	150	3539	212
17.5" φ	4316-6	460	146	650	207
16"x16"	4316-7	1527	4530	2160	6407
14"x14"	4316-1	1358	3685	1921	5212

*contributing stresses from different systems are added although as discussed in Section 3.4.2 the braze is not the weak part of the connection, it will nontheless be checked for stress. The allowable stresses are as follows,

$F_W = .3 F_{Exx}$, shear on braze (AWS)

$F_M = .4 F_y$, shear on metal (AWS)

$F_m = .3 F_u$, Longitudinal to weld (AISI)

$F_m = .33 F_u$, Transverse to weld (AISI)

$F_m = .6 F_y$, operational loads (ANSI N509)

$F_m = .9 F_y$, combined loads (ANSI N509)

where,

$F_y = 36,000$ psi , carbon steel

$F_u = 58,000$ psi , carbon steel

$F_{Exx} = 50,000$ psi , Class E CuSi [Ref 11, SFA-5.6]

The minimum allowable stresses for any direction or type of loading are,

$F_W = .3(50^k) = 15,000$ psi

$F_M = .4(36^k) = 14,400$ psi

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The tensile and shearing stresses are conservatively added and then compared to the allowables. That is,

$$(\tau + \sigma)_{MAT} < F_m = 14,400 \text{ psi}$$

$$(\tau + \sigma)_{BRAZE} < F_B = 15,000 \text{ psi}$$

So,

18" ϕ

2502 + 150 = 2652 < 14,400 psi O.K

3539 + 212 = 3751 < 15,000 psi O.K

17.5" ϕ

460 + 146 = 606 < 14,400 psi O.K.

650 + 207 = 857 < 15,000 psi O.K

16"x16"

1527 + 4530 = 6057 < 14,400 psi O.K.

2160 + 6407 = 8567 < 15,000 psi O.K.

14"x14"

1358 + 3685 = 5043 < 14,400 psi OK.

1921 + 5212 = 7133 < 15,000 psi OK

1

IMAGE EVALUATION TEST TARGET (MT-3)

150mm

6"

1

IMAGE EVALUATION TEST TARGET (MT-3)

150mm

6"

4.4 AIR TIGHTNESS

As was shown in previous Section 4.2 the stresses on the brazed joint in the "Pittsburgh Lock" and "ACME Lock" longitudinal seams are well below the Code allowables. In addition, to insure air tightness, the longitudinal seams are sealed with G.E. Silicone #1200.

From the above discussion it can be concluded that the air tightness of the longitudinal seams will be properly maintained.

4.5 FATIGUE CONSIDERATIONS

FATIGUE EVALUATIONS ARE NOT USUALLY CONSIDERED FOR SEISMIC ANALYSES. THE ONLY OTHER FATIGUE CONSIDERATION IS THE STARTUP/SHUTDOWN CYCLES OF ADJACENT FANS WHICH CAUSE INTERNAL PRESSURE. THESE CYCLES ARE PART OF THE OPERATIONAL LOADS.

THE FANS ARE STARTED ABOUT TWICE MONTHLY [12]

∴ THERE ARE TWO STARTUP/SHUTDOWN CYCLES PER MONTH

∴ $N_L \equiv$ THE LIFETIME EXPECTED STARTUP

N_L = 2 CYCLES PER MONTH x 12 MONTHS x 40 YEARS

= 960 CYCLES

$N_L < 1 \times 10^3$ CYCLES

NORMALLY, FATIGUE IS CONSIDERED WHEN $N_L > 10^4$. FOR $N_L < 10^4$ (LOW CYCLE FATIGUE ≡ LCF) FATIGUE IS A CONCERN WHEN STRESSES EXCEED YIELD. FOR NORMAL OPERATION, INCLUDING THE FANS OPERATION, THE STRESSES ON THE DUCTING ARE BELOW YIELD, AND THEREFORE THE FAN CYCLES WILL NOT AFFECT THE DESIGN OR LONGEVITY OF THE DUCTING. THIS WILL BE DEMONSTRATED IN THE NEXT FEW PAGES.

FOR THE DUCTING MATERIAL, THE DESIGN STRESS PER ANSI/ASME N509-80 IS 0.6 Sy. ALLOWING THE MAXIMUM STRESS TO APPROACH 0.6Sy:

$$S_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

LET $\sigma_{max} \equiv 0.6\,S_y = 21.6$ KSI

$\sigma_{min} = 0$ KSI

$$\therefore\ S_a = \frac{21.6 - 0}{2} = 10.8 \text{ KSI}$$

FROM FIG 4.5.1 : [13, APPENDIX I, FIG I-9.1]

NO REDUCTION IN THE MATERIAL STRENGTH IS EXPECTED.

AS FOR THE BRAZE, THE STRUCTURAL WELDING CODE ANSI/AWS D1.1-83 DOES NOT CONSIDER $N < 10^4$ FOR FATIGUE. FIG 4.5.2 DEMONSTRATES FATIGUE OF WELDED PIPE, THE DUCTING WOULD BE SIMILAR. ADDITIONALLY, THE STRESS IN THE WELDMENT CAN BE ASSUMED TO BE THE SAME AS THE DUCT. $\therefore\ S_a = 10.8$ KSI

THE BRAZE MATERIAL IS ECuSi, AWS A5.6 [1].

ASME BOILER AND PRESSURE VESSEL CODE, SECTION II, PART C - MATERIALS INCLUDES SFA 5.6 WHICH IS IDENTICAL TO AWS A5.6.

1986 Edition APPENDIX I Fig. I-9.1

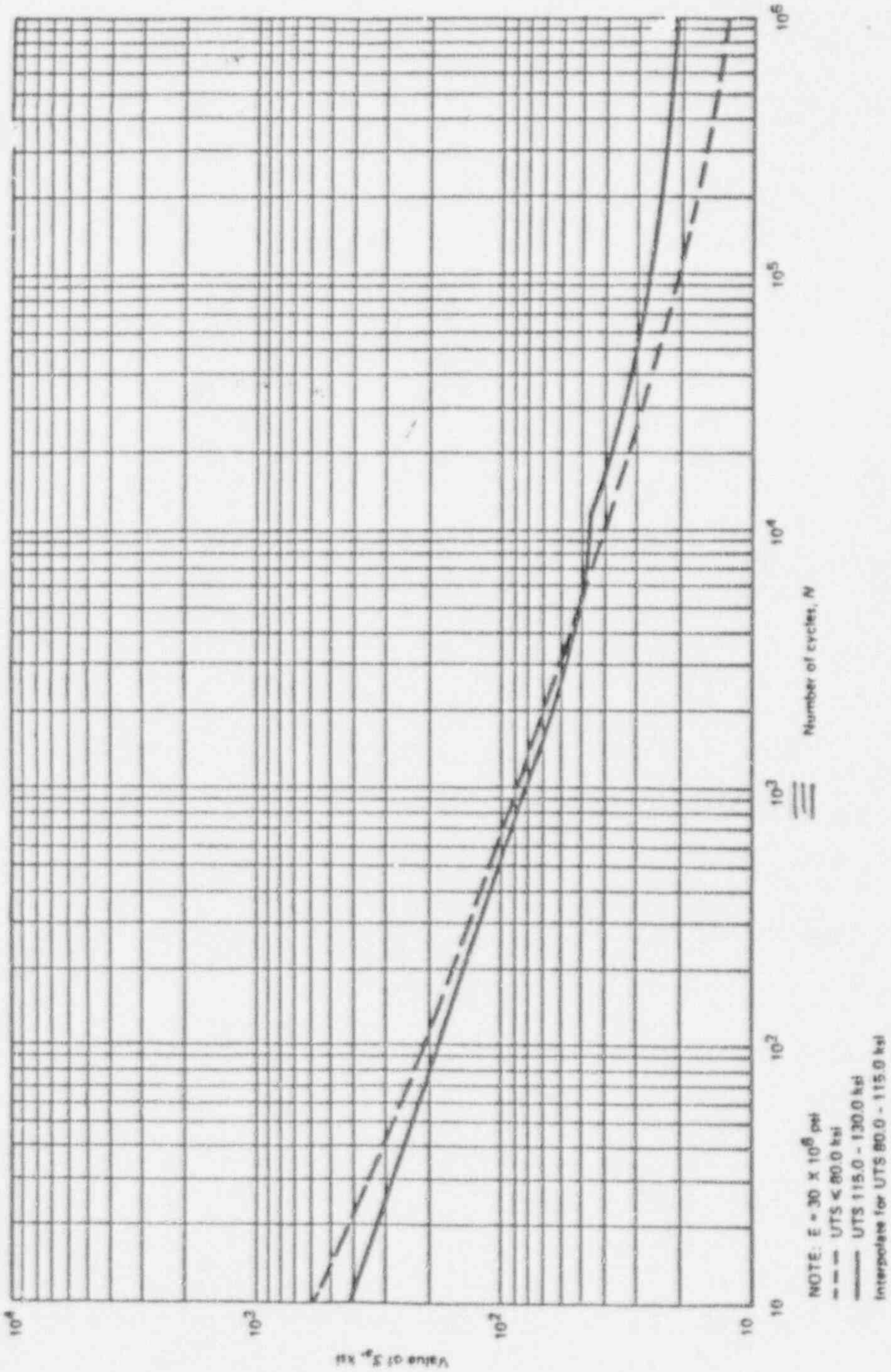
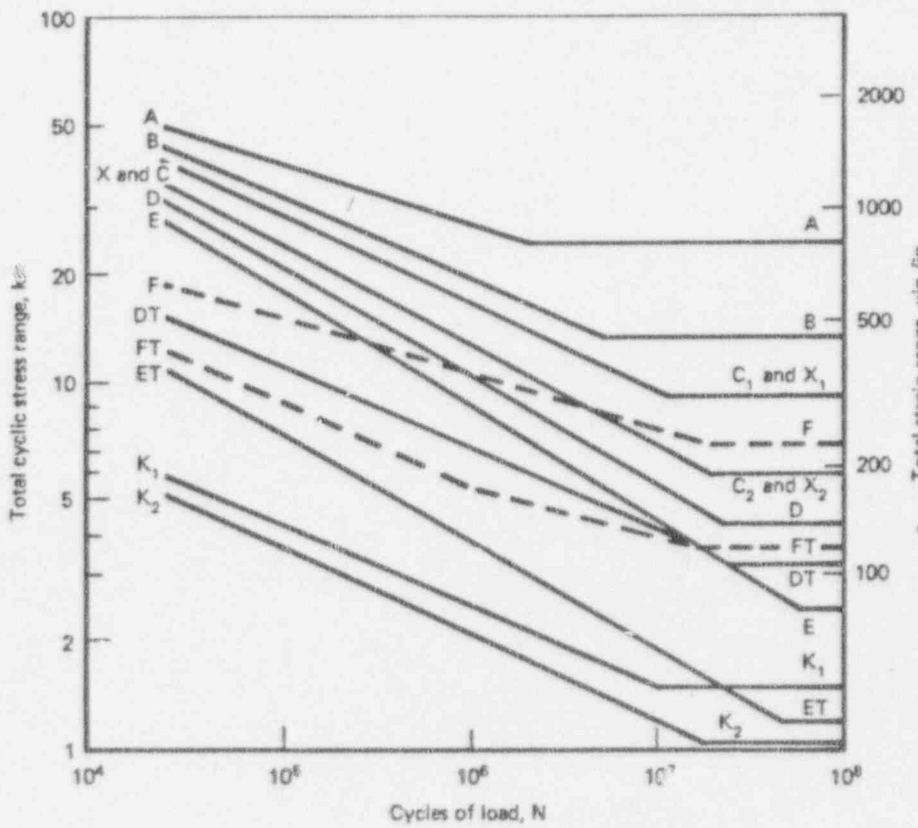


FIG. I-9.1 DESIGN FATIGUE CURVES FOR CARBON, LOW ALLOY, AND HIGH TENSILE STEELS FOR METAL TEMPERATURES NOT EXCEEDING 700°F

Table I-9.1 Contains Tabulated Values and a Formula for Accurate Interpolation of These Curves

FIG. 4.5.1 FATIGUE CURVE [13, APPENDIX I, FIG I-9.1]

STRESS CATEGORY B = PIPE WITH LONGITUDINAL SEAM [6, TAB 10.7.3]
(SIMILAR TO DUCTS, FOR DEMONSTRATION ONLY)



Allowable fatigue stress and strain ranges for stress categories redundant structures in atmospheric service

FIG. 4.5.2 [6, FIG 10.7.4]

SFA 5.6 GIVES AWS MECHANICAL PROPERTY REQUIREMENTS FOR ECuSi. THE MINIMUM TENSILE STRENGTH IS 50 KSI. AN ADDITIONAL REFERENCE [14, PG 22.36] GIVES A FATIGUE STRENGTH FOR ECuSi (SILICON BRONZE) AT ROOM TEMPERATURE:

$$S_a = 19 \text{ KSI FOR } 300 \times 10^6 \text{ CYCLES}$$

FOR $N_L < 1000$, NO REDUCTION IN THE BRAZE MATERIAL STRENGTH IS EXPECTED.

HOPPER AND ASSOCIATES
ENGINEERS

CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 13 DATE: MAY '94 PAGE: 44

SUBJECT: 5.0 CONCLUSIONS BY: A.R. CK: MAK SHT: 1 OF 1

CONCLUSIONS

In response to a U.S.N.R.C. Docket 50-341, the structural integrity, air tightness and fatigue strength of the longitudinal "Pittsburgh Lock" and "ACME Lock" seams of duct systems 2848-3, 4316-1, 4316-6 and 4316-7 was investigated.

It was found that the brazed joints of the longitudinal seams meet the allowable stress requirements of the applicable codes. Air tightness is expected to be maintained because of the presence of Silicone sealant. also, based on the expected number of loading cycles on the ducting systems, fatigue life of the longitudinal seams is not expected to be compromised

REFERENCES

1. Robert Irsay Company, "Duct Construction Brochure Enrico Fermi Atomic P.P. #2 3071-104-Type 1", (Edison File 89-686)

2. Hopper & Associates, "Structural Evaluation of Ducting Systems 2848-3, 4316-1, 4316-6, 4316-7", Client: DECo Fermi-2, HA-05/89-686, May, 1989.

3. Timoshenko & Krieger, "Theory of Plates and Shells", 2nd Edition, 1959.

4. Young, W.C., "Roark's Formulas for Stress and Strain", 6th Edition McGraw-Hill, 1989.

5. AISI, "Cold Formed Steel Design Manual", American Iron and Steel Institute, 1989.

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 13 DATE: MAY '94 PAGE: 46

SUBJECT: REFERENCES BY: A.R. CK: MAK SHT: 2 OF 3

6. AWS, "Structural Welding Code - Steel", D1.1, American Welding Society, 1983.

7. AWS, "Sheet Metal Welding Code" D9.1, American Welding Society, 1990.

8. ANSI/ASME N509-80, "Nuclear Power Plant Air Cleaning Units and Components", American Society of Mechanical Engineers, 1980.

9. ASME, Boiler & Pressure Vessel Code, Section IX, "Welding and Brazing Qualifications", American Society of Mechanical Engineers, 1992.

10. Desai, S.C. et. al. "Structural Testing of Seismic Category I HVAC Duct Specimens", Civil Engineering and Nuclear Power, Vol. 1, ASCE, 1980.

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 13 DATE: MAY 94 PAGE: 47

SUBJECT: REFERENCES BY: A.P. CK: MMK SHT: 3 OF 3

11. ASME, 1992 Boiler & Pressure Vessel Code - Materials, Part C, American Society of Mechanical Engineers, 1992.
12. Telecon, Philip Hasrouni with Dave Jax, DECo Fermi 2, May 2, 1994.
13. ASME, 1986 Boiler & Pressure Vessel Code, Section III, Appendix I
14 Smithels Metals Reference Book, ed. by E.A. Brandes, 6th Edition, Butterworth Co Ltd, London, 1983.

Atta:hment 10 to
NRC-94-0050

HOPPER AND ASSOCIATES
ENGINEERS

DECO - USNRC DOCKET NO. 50-341

FERMI 2 CCHVAC DUCTING SYSTEMS

CONCERN ITEM NO. 14

EVALUATION CALCULATIONS

Prepared for: Detroit Edison Company
Nuclear Engineering Fermi 2
6400 North Dixie Highway
Newport, MI 48166

Prepared by: Hopper and Associates
300 Vista Del Mar
Redondo Beach, CA 90277

April, 1994

HOPPER AND ASSOCIATES
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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 14 DATE: MAR '94 PAGE: i

SUBJECT: TABLE OF CONTENTS BY: A.R. CK: MAK SHT: i OF i

TABLE OF CONTENTS

PREPARED BY: Alex Reizman 4-5-94

CHECKED BY: M. Amir Khan 4-19-94

1.1 PROBLEM STATEMENT

On March 3, 1994, the U.S.N.R.C submitted Docket No 50-341 to Detroit Edison Company. This Docket consists of a request for additional information on Fermi 2 CCHVAC system design and operation. In particular, there are several concerns regarding the structural integrity calculations, HA-05/89-686 and HA-09/89-696, performed by Hopper & Associates.

This package is prepared in order to respond to one of the comments posed by the NRC regarding HA-09/89-696. Namely,

"The existing internal loads due to seismic loading were amplified by the product of the ratio of the SSE ZPA to the OBE ZPA and the ratio of the peak response acceleration to the ZPA, presumably of the SSE. The damping used for this calculation was not specified. However, there are two horizontal and one vertical component for each earthquake, and these are also dependent on the evaluation within the building. It is unclear how these directional components were actually applied in these calculations."

1.2 INVESTIGATION APPROACH

The procedure used in obtaining the load factors, implemented throughout the bounding analyses of HA-09/89-696, will be documented. The approach taken in analyzing each system will then be summarized in the form of tables.

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 14 DATE: MAR 94 PAGE: 3

SUBJECT: 2.0 ANALYSIS APPROACH BY: A.R. CK: MAK SHT: 1 OF 25

2.1 EXISTING ANALYSIS

An analysis of the ducting systems was performed by Fluor - Pioneer [Ref 1]. The results of these analyses are documented in form of design reports. With an exception of Duct System 2850-2, the structural evaluation work in HA-09/89-696 is based upon data contained in these design reports.

2.2 ASSUMPTIONS

The analyses performed by Fluor - Pioneer, and documented in the design reports, were based upon the following assumptions;

1. All hangers and anchors supporting the Ductwork are rigid.
2. Most of the ducting systems were considered rigid (natural frequency >33Hz) Therefore, a static analysis was performed.

3. In some cases, the natural frequencies of the ducting systems were found to be < 33 Hz. For these systems, a dynamic analysis was performed applying seismic response spectra as load input.
4. The seismic analyses for all ducting systems were based upon the Operating Basis Earthquake (OBE).
5. No internal pressure loads were considered in the analyses.

2.3 LOAD FACTORS

The amplification factors used in the HA-09/89-696 analyses are defined as follows,

$$S = \frac{\text{SSE ZPA}}{\text{OBE ZPA}}$$

$$D = \frac{\text{SSE PEAK ACCELERATION}}{\text{SSE ZPA}}$$

The values used correspond to 2% damping Response Spectra. The Spectra are included as Figures 2.3.1 through 2.3.12. The SSE is from [Ref 2] and OBE is from [Ref 3].

Thus, for Slab 3 (EL. 641.5')

$$S_{N-S} = \frac{.38}{.16} = 2.38$$

$$S_{E-W} = \frac{.43}{.155} = 2.77$$

$$S_{VERT} = \frac{.52}{.18} = 2.89$$

and,

$$D_{N-S} = \frac{3.4}{.38} = 8.9$$

$$D_{E-W} = \frac{3.1}{.43} = 7.2$$

$$D_{VERT} = \frac{4.1}{.52} = 7.9$$

LOAD FACTOR = $S \times D$

$$L.F._{N-S} = 2.38 \times 8.9 = 21.1$$

$$L.F._{E-W} = 2.77 \times 7.2 = 19.9$$

$$L.F._{VERT} = 2.89 \times 7.9 = 22.8$$

For simplicity, a Load Factor of 22 was used in all three directions.

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 14 DATE: MAR '94 PAGE: 6

SUBJECT: ANALYSIS APPROACH BY: A.R. CK: MAK SHT: 4 OF 25

FOR <u>Slab 5 (EL. 684.5')</u>

$$S_{N-S} = \frac{.49}{.2^*} = 2.45$$

$$S_{E-W} = \frac{.54}{.2^*} = 2.7$$

$$S_{VERT} = \frac{.52}{.2^*} = 2.6$$

* .2g is used in all directions to maintain consistancy with Fluor-Pioneer calculations

and,

$$D_{N-S} = \frac{4.1}{.49} = 8.4$$

$$D_{E-W} = \frac{4.0}{.54} = 7.4$$

$$D_{VERT} = \frac{4.1}{.52} = 7.9$$

The Load Factor is therefore

$$L.F._{N-S} = 2.45 \times 8.4 = 20.6$$

$$L.F._{E-W} = 2.7 \times 7.4 = 20.0$$

$$L.F._{VERT} = 2.6 \times 7.9 = 20.5$$

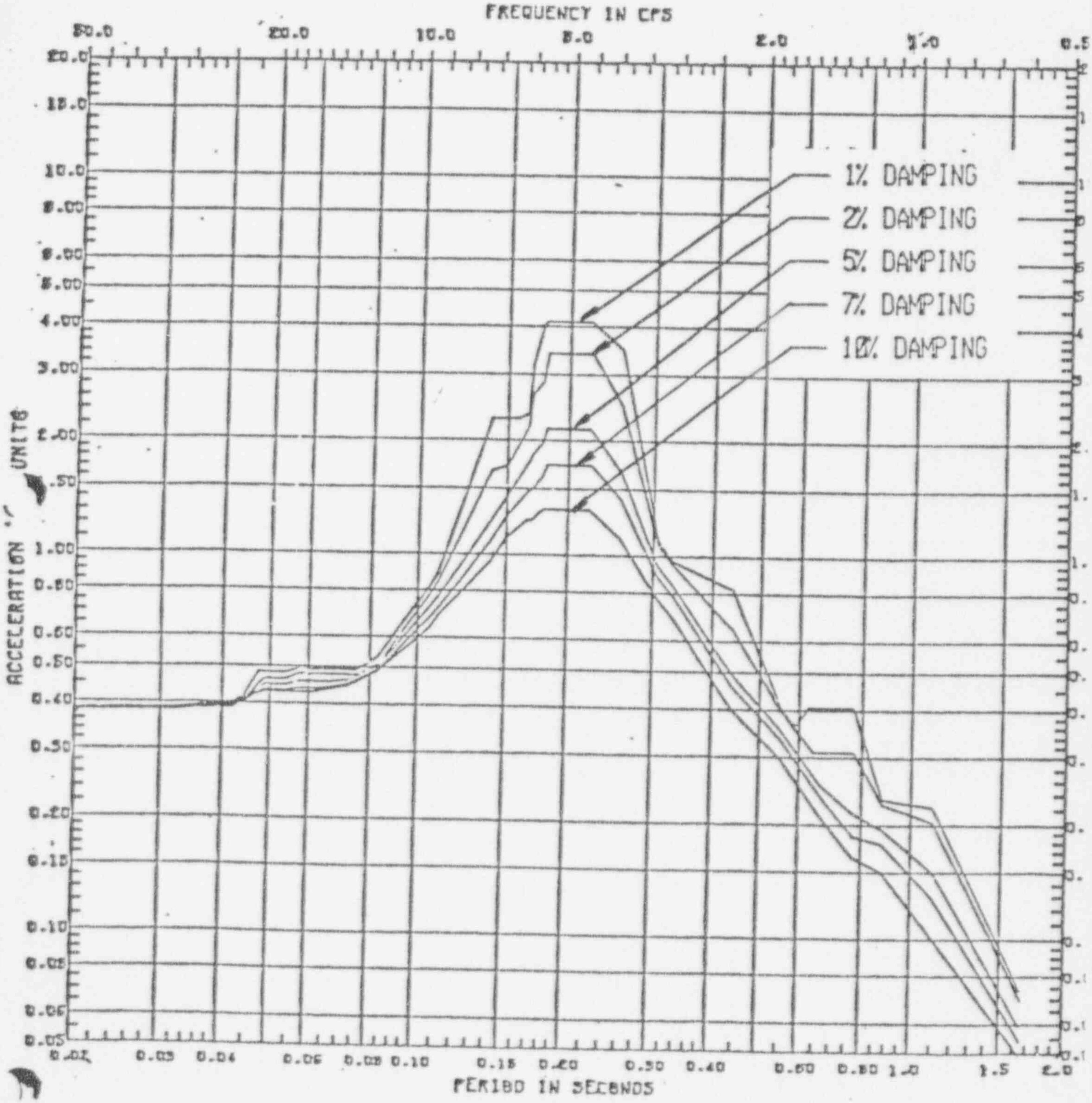
again for simplicity, a Load Factor of <u>20</u> was used in all three directions.

NOTE: Vertical spectra are for Reactor Building only. Vertical spectra for Auxiliary Bldg are significantly lower. Thus, vertical direction is conservative.

ENRICO FERMI UNIT 2

A. KROLIKOWSKI DA
SYSTEM ENGR: YA/AEK 11/11/91
Y. ANAND DAT

SITE SPECIFIC EARTHQUAKE 5% SITE DAMPING



REACTOR AUXILLARY BUILDING: RESPONSE SPECTRUM

FIGURE NO 2.31

SPECTRUM	NODE	ELEV	DIRECT	LOCATION	
SES-B-33	3	641-6	NS	REA-AUX BLDG	SLAB 3

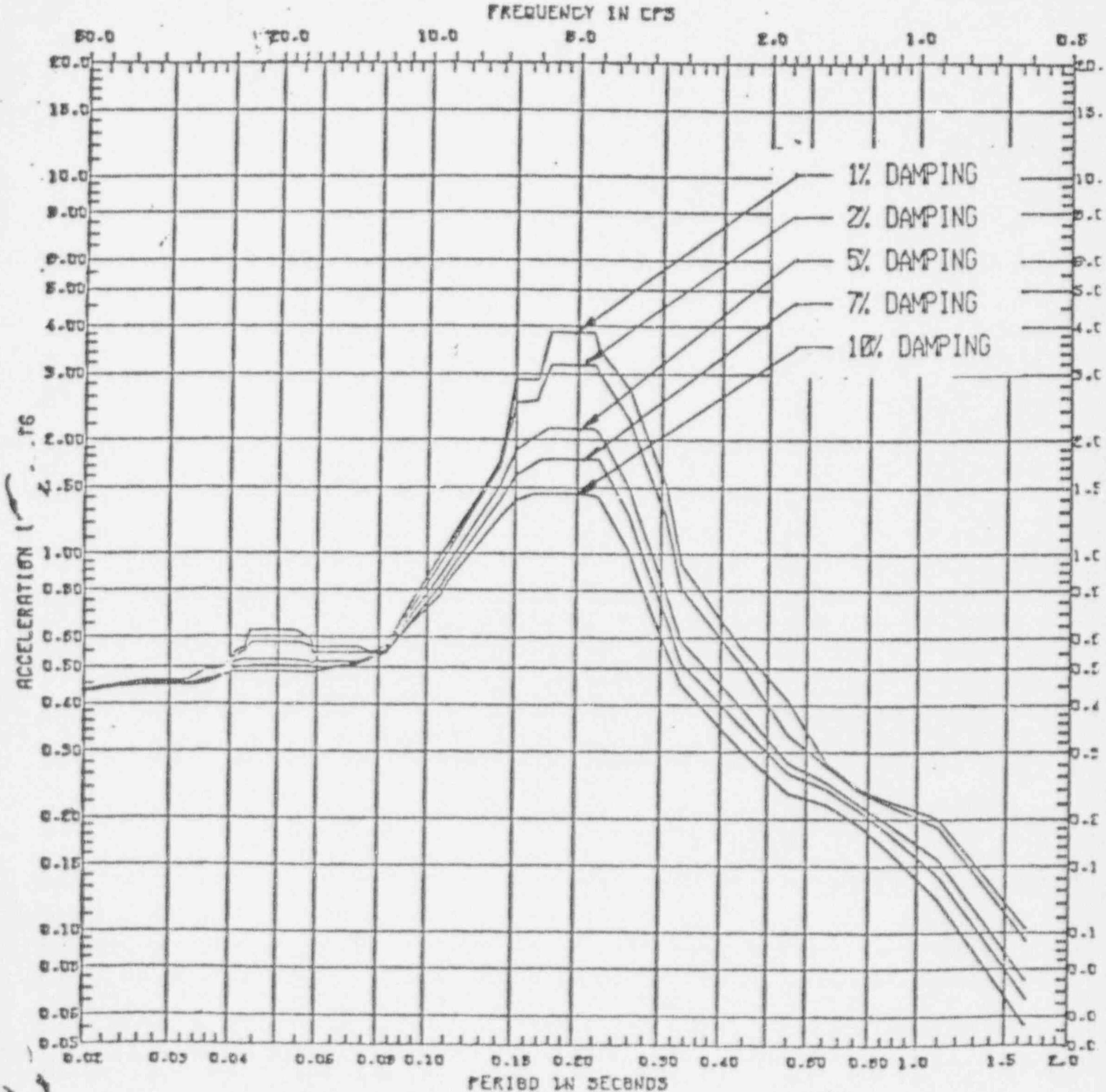
ENRICO FERMI UNIT 2

PREPARED BY: A. Krolikowski 10/20/81
A. KROLIKOWSKI DATE

SYSTEM ENGR: YA/AEK 11/11/81
Y. ANAND DATE

SITE SPECIFIC EARTHQUAKE 5% SITE DAMPING

8
6/2



REACTOR AUXILLARY BUILDING: RESPONSE SPECTRUM

FIGURE NO. 2.3.2

SPECTRUM	NODE	ELEV	DIRECT	LOCATION	
		641-6	EW	REA-AUX BLDG	SLAB 3

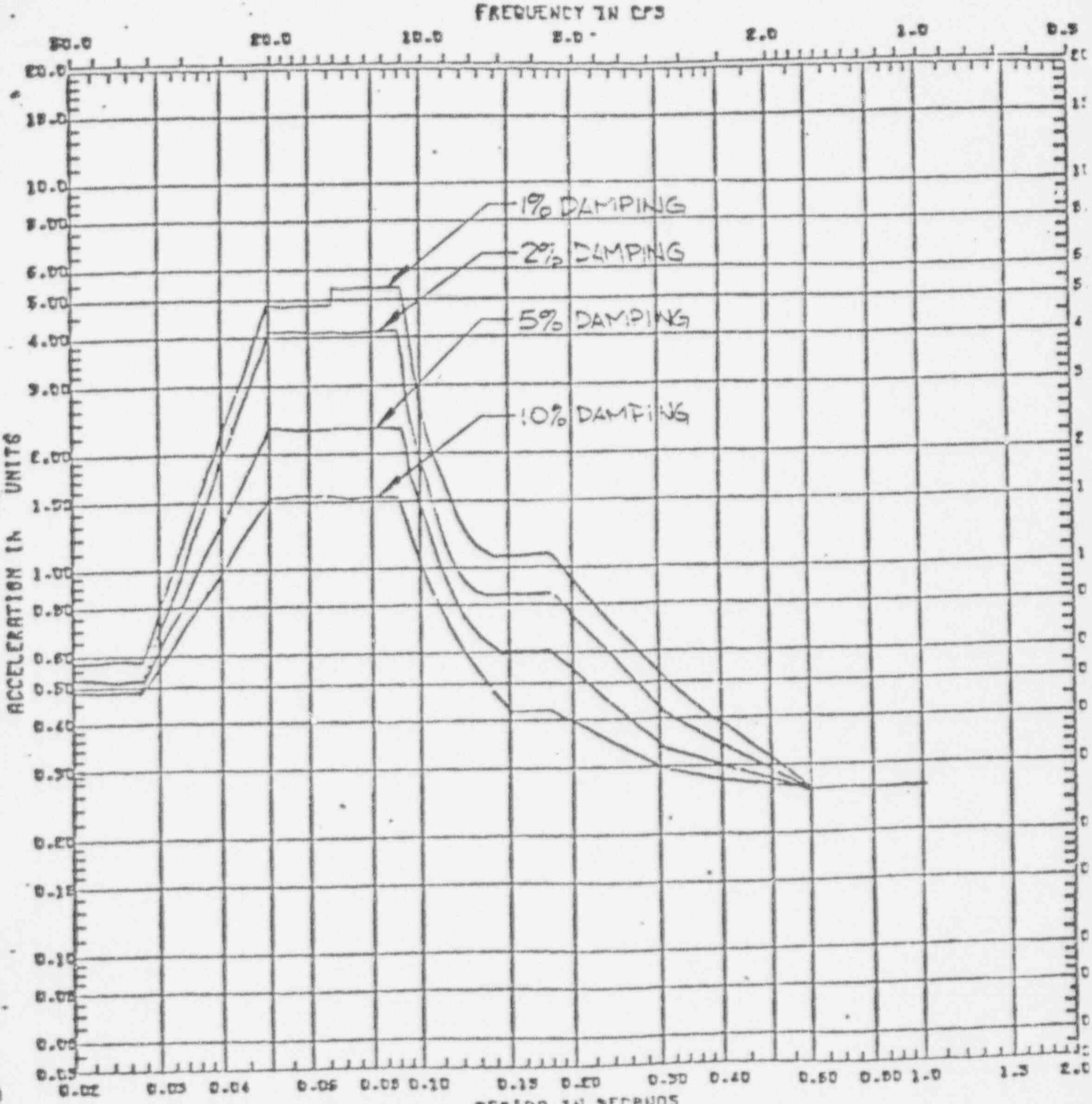
ENRICO FERMI
UNIT 2

PREPARED BY: A. KROLIKOWSKI DATE

SYSTEM ENGR: ANAND 11/11/8_ DATE

SITE SPECIFIC EARTHQUAKE 5% SITE DAMPING

9
7/25



REACTOR-AUXILLIARY BUILDING: RESPONSE SPECTRUM

FIGURE NO.

SPECTRUM	NODE	ELEV	DIRECT	LOCATION
SE5-C-17		641-6	VERT	REACTOR BUILDING SLAB

PREPARED BY: [illegible]
A. KROLIKOWSKI

SYSTEM ENGR: [illegible]
Y. ANAND

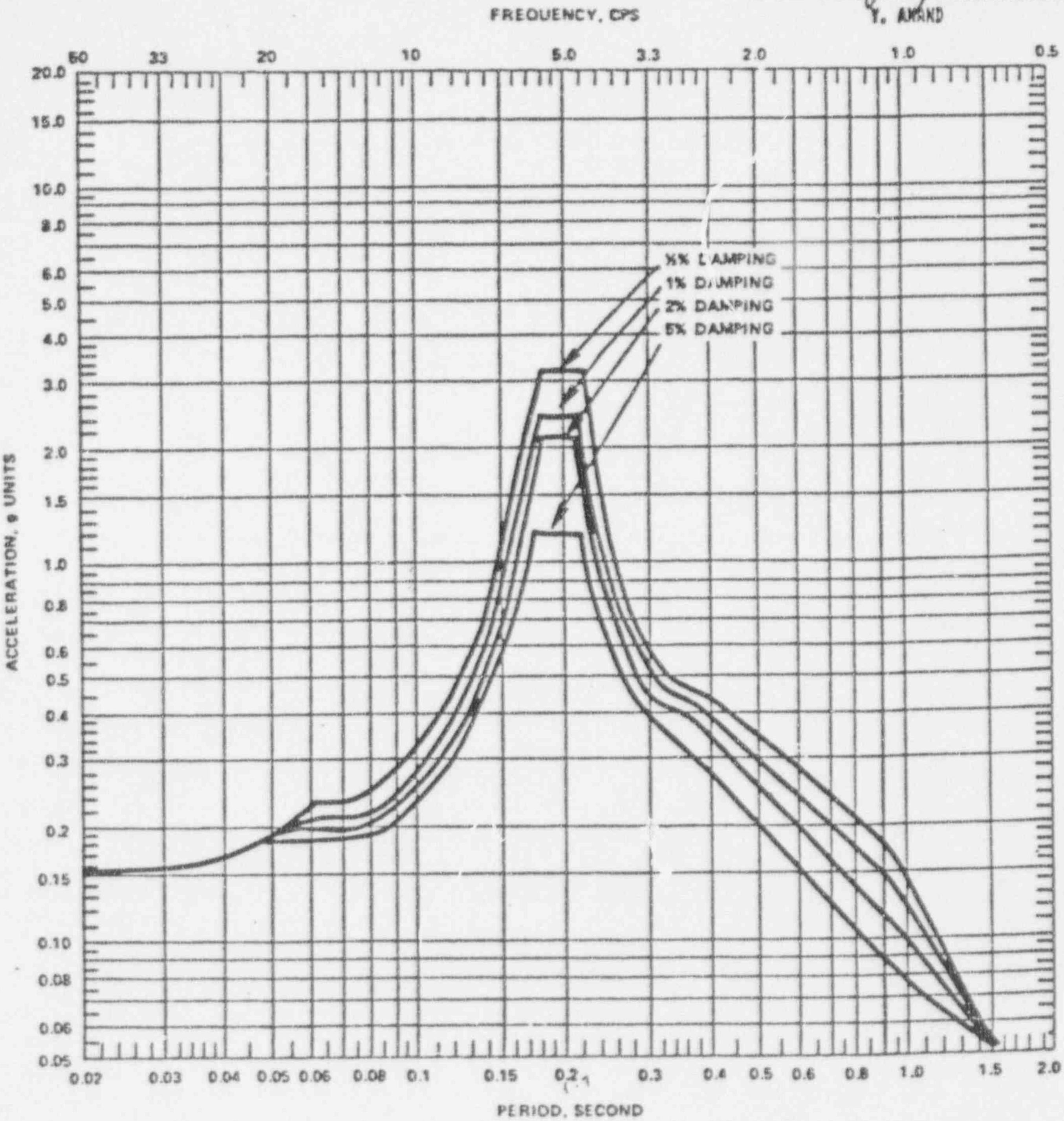


FIGURE 2.3.4

ENRICO FERMI ATOMIC POWER PLANT UNIT 2
HORIZONTAL FLOOR RESPONSE SPECTRA OPERATING BASIS EARTHQUAKE ELEVATION – 641'–6" (SLAB NO. 3) REACTOR/AUXILIARY BUILDING NORTH – SOUTH COMPONENT

SARGENT & LUNDY REPORT NO. SL–2682

PREPARED BY: A. Krolikowski 11/5/..
A. KROLIKOWSKI DAT

SYSTEM ENGR: H.A./OSE · 10/11/8..
T. ANAND DAT

11
9/2

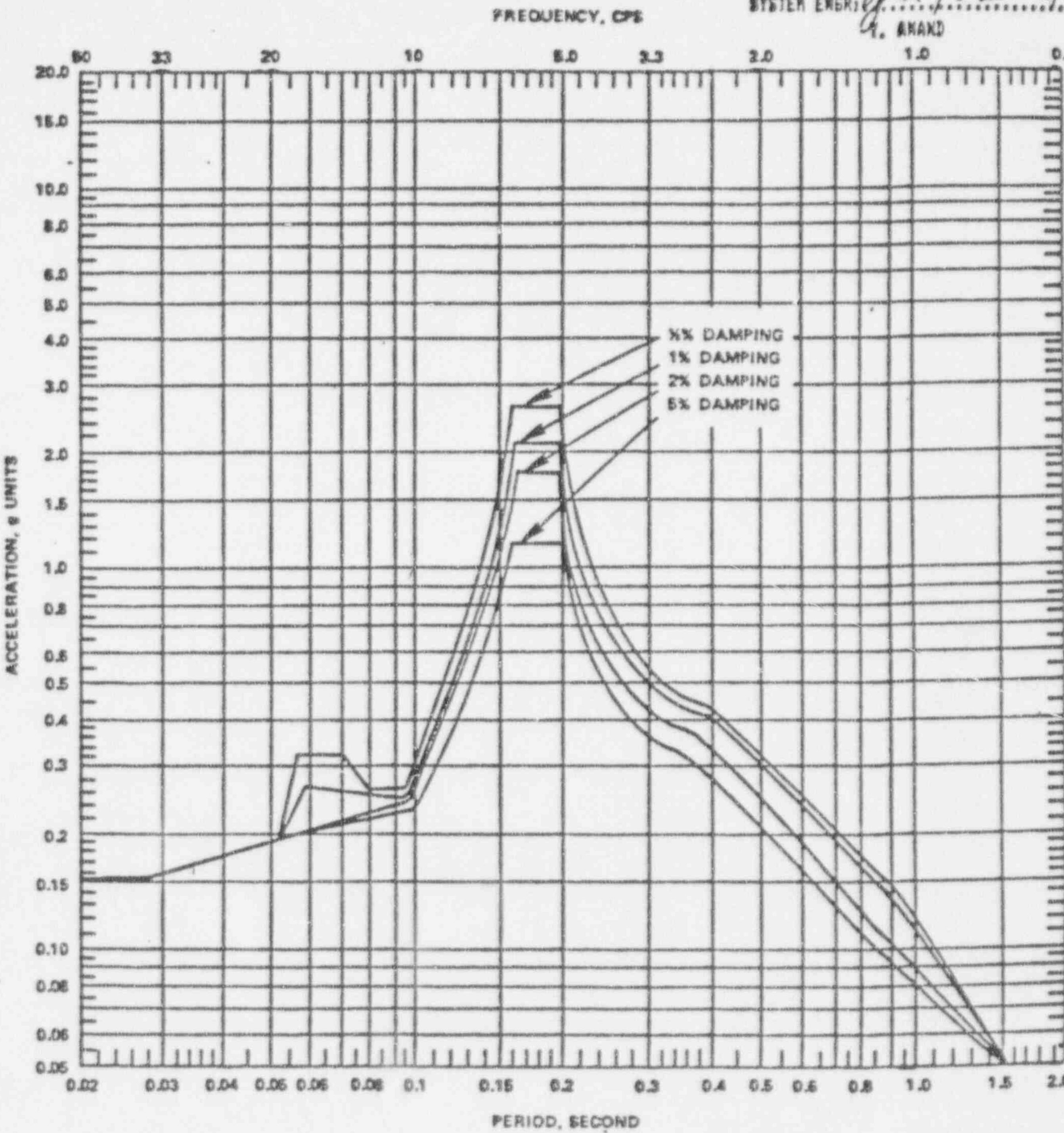


FIGURE 2.3.5

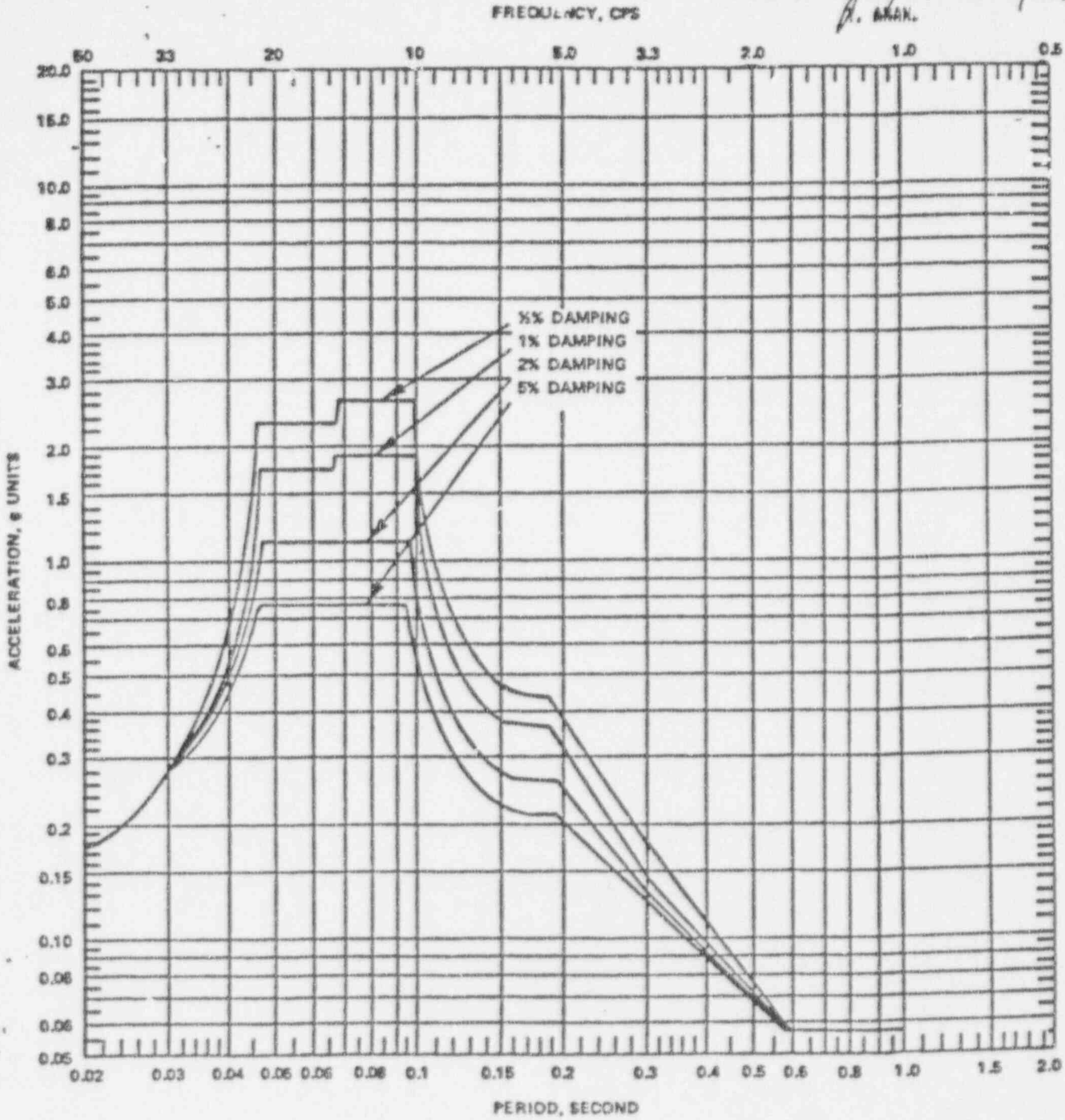
ENRICO FERMI ATOMIC POWER PLANT
UNIT 2

HORIZONTAL FLOOR RESPONSE SPECTRA
OPERATING BASIS EARTHQUAKE ELEVATION –
641'–6" (SLAB NO. 3) REACTOR/AUXILIARY
BUILDING EAST – WEST COMPONENT

SARGENT & LUNDY REPORT NO. SL–2682

PREPARED BY: ..U.C.K... 11/21/81
A. KROLIKOWSKI DATE

SYSTEM ENGR: ..YA/AEK.. 4/6/81
A. ANAN. DATE



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10/2

FIGURE 2.3.6

ENRICO FERMI ATOMIC POWER PLANT
UNIT 2

VERTICAL RESPONSE SPECTRA OPERATING BASIS EARTHQUAKE REACTOR BUILDING SLAB EL. 641'-6", 659'-6" & 684'-6"

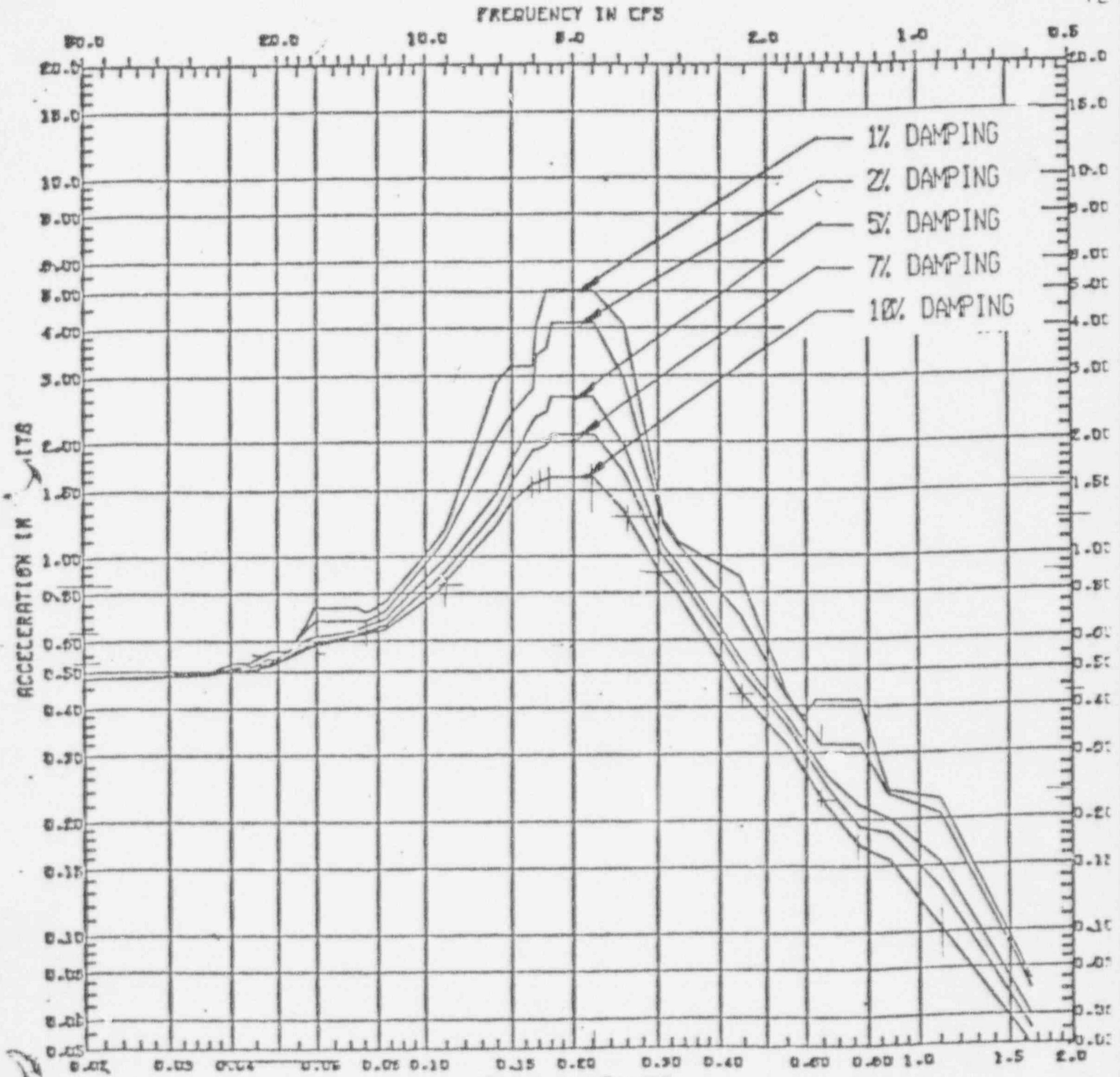
SARGENT & LUNDY REPORT NO. SL-2682

ENRICO FERMI
UNIT 2

PREPARED BY: [signature] A. KROLIKOWSKI DATE

SYSTEM ENGR: [signature] 4/1/81 A. ANAND DATE

SITE SPECIFIC EARTHQUAKE 5% SITE DAMPING



REACTOR AUXILLARY BUILDING: RESPONSE SPECTRUM

FIGURE NO. 2.3.7

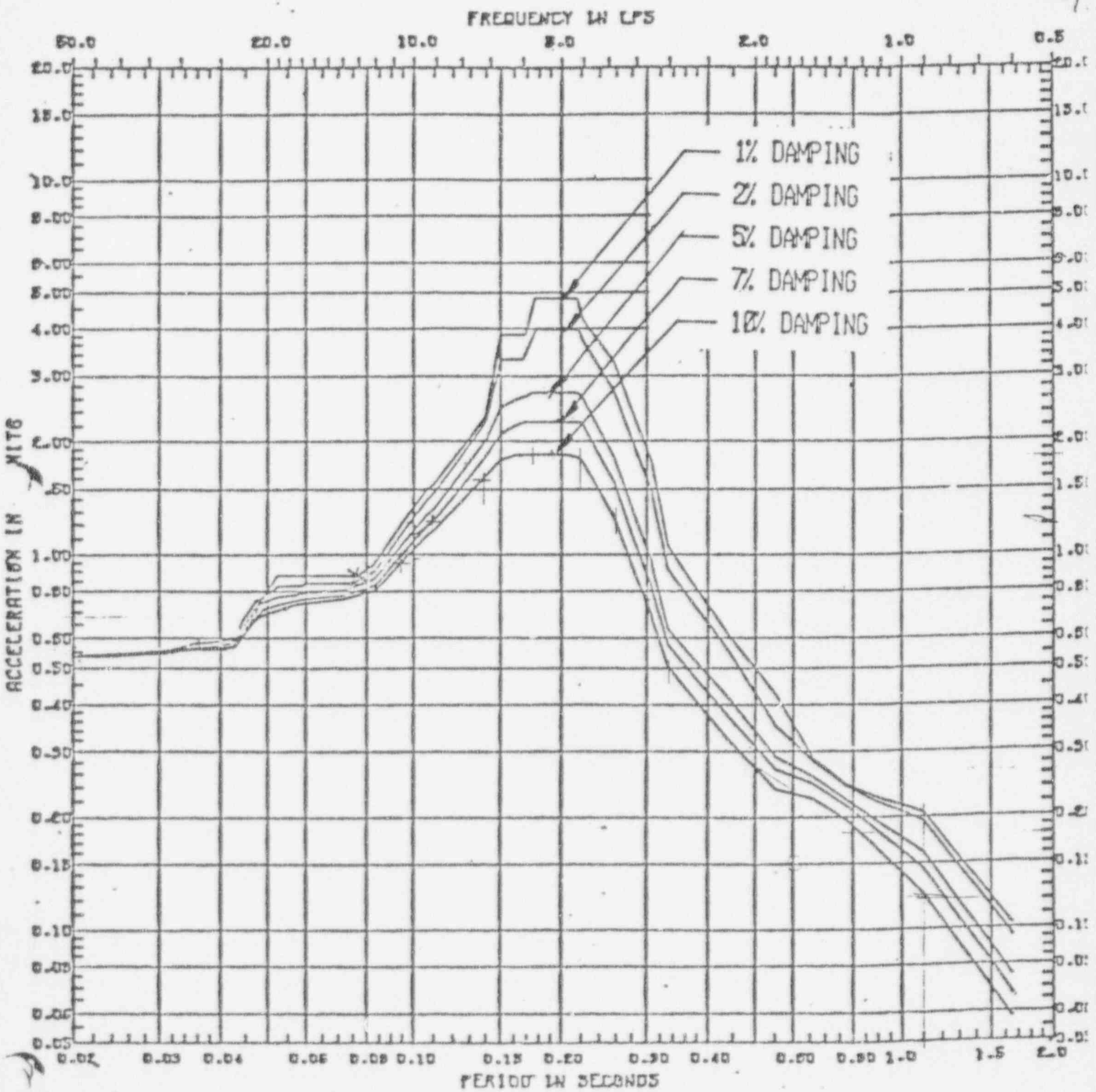
SPECTRUM	NODE	ELEV	DIRECT	LOCATION	
SES-B-37	5	684-6	NS	REA-AUX BLDG	SLAB 5

ENRICO FERMI
UNIT 2

PREPARED BY: A. Krolikowski
A. KROLIKOWSKI DATE

SYSTEM ENGR: Y. Anand 4/4/91
Y. ANAND DATE

SITE SPECIFIC EARTHQUAKE 5% SITE DAMPING



REACTOR AUXILLARY BUILDING: RESPONSE SPECTRUM

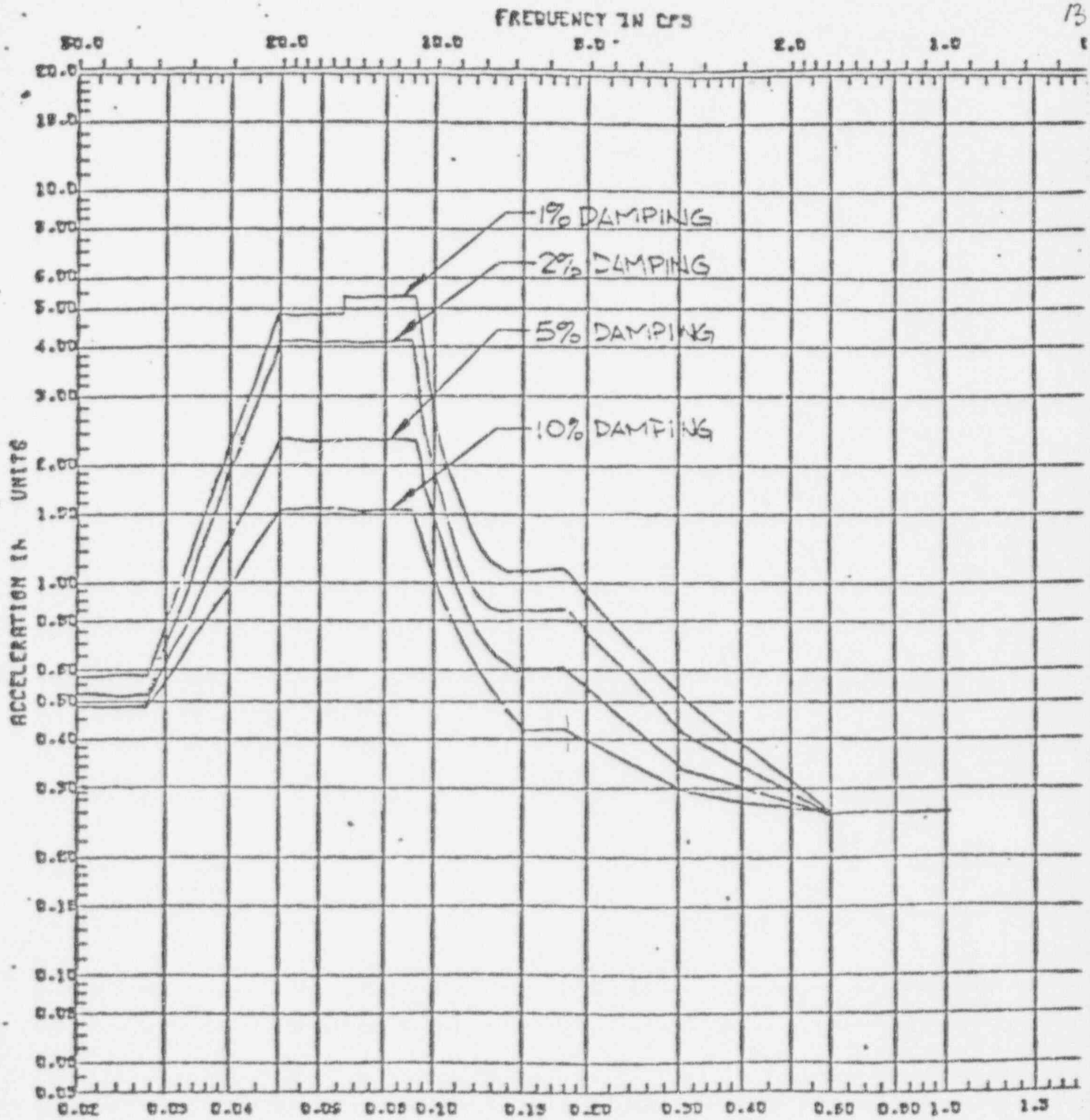
FIGURE NO 2.3.8

SPECTRUM	NODE	ELEV	DIRECT	LOCATION
SES-B-38	5	684-6	EW	REA-AUX BLDG SLAB 5

ENRICO FERMI
UNIT 2

PREPARED BY: A. KROLIKOWSKI DATE
SYSTEM ENGR: ANAND 11/11/8 DATE

SITE SPECIFIC EARTHQUAKE 5% SITE DAMPING



REACTOR-AUXILLIARY BUILDING: RESPONSE SPECTRUM

FIGURE N

SPECTRUM	NODE	ELEV	DIRECT	LOCATION
SE5-C-17		641-6	VERT	REACTOR BUILDING SLAB

2.3.9

PREPARED BY: 11/5/81
A. KROLIKOWSKI DATE

SYSTEM ENGR: Y.A./AEK 11/4/81
Y. ANAND DATE

16
14/25

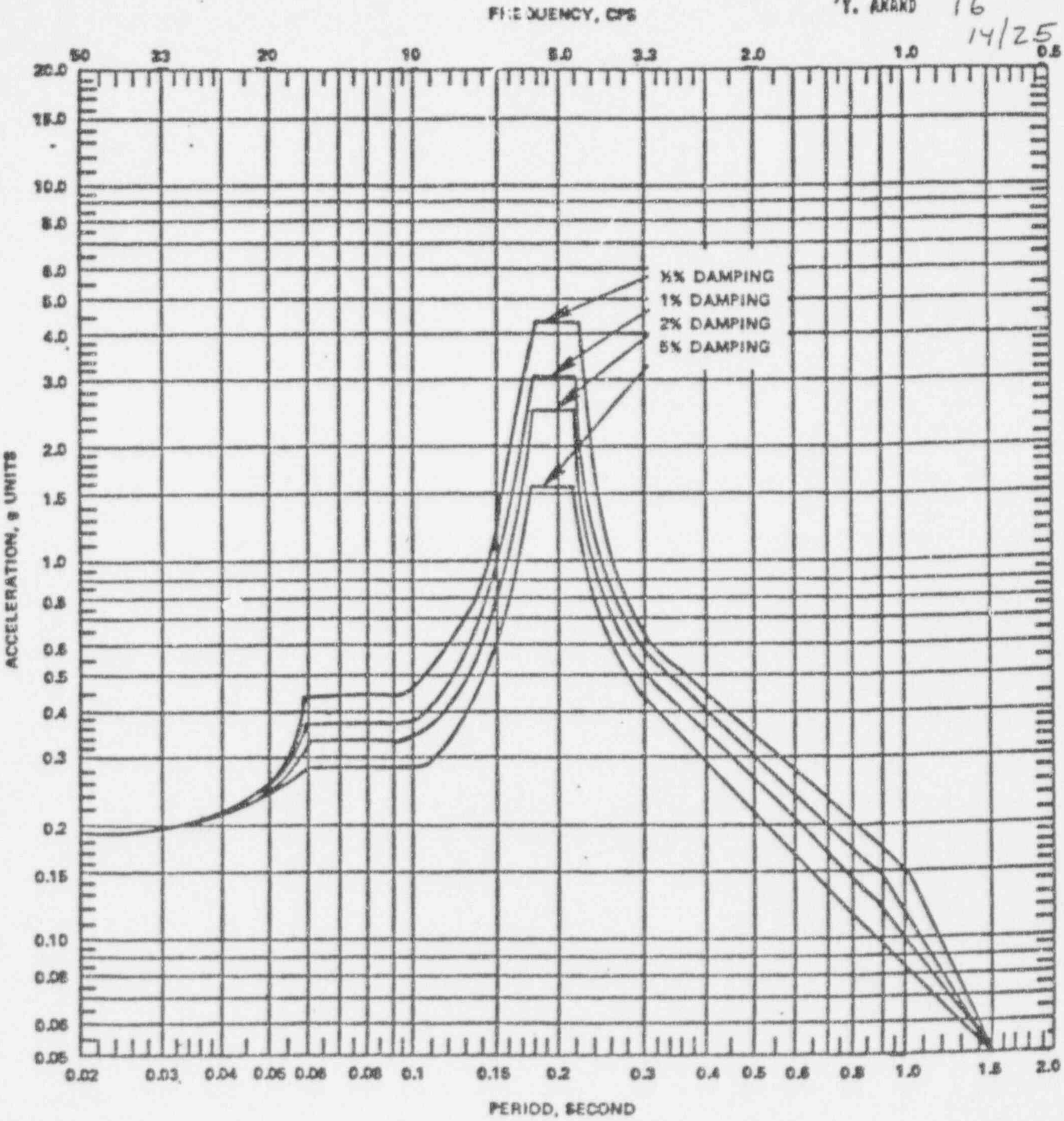


FIGURE 2.3.10

ENRICO FERMI ATOMIC POWER PLANT UNIT 2
HORIZONTAL FLOOR RESPONSE SPECTRA OPERATING BASIS EARTHQUAKE ELEVATION – 684'–6" (SLAB NO. 5) REACTOR/ AUXILIARY BUILDING NORTH – SOUTH COMPONENT

SARGENT & LUNDY REPORT NO. SL–2662

PREPARED BY: A. Krolikowski
A. KROLIKOWSKI DA

SYSTEM ENGR: Y. P. Anand 4/4/8
Y. ANAND DA

17
15/25

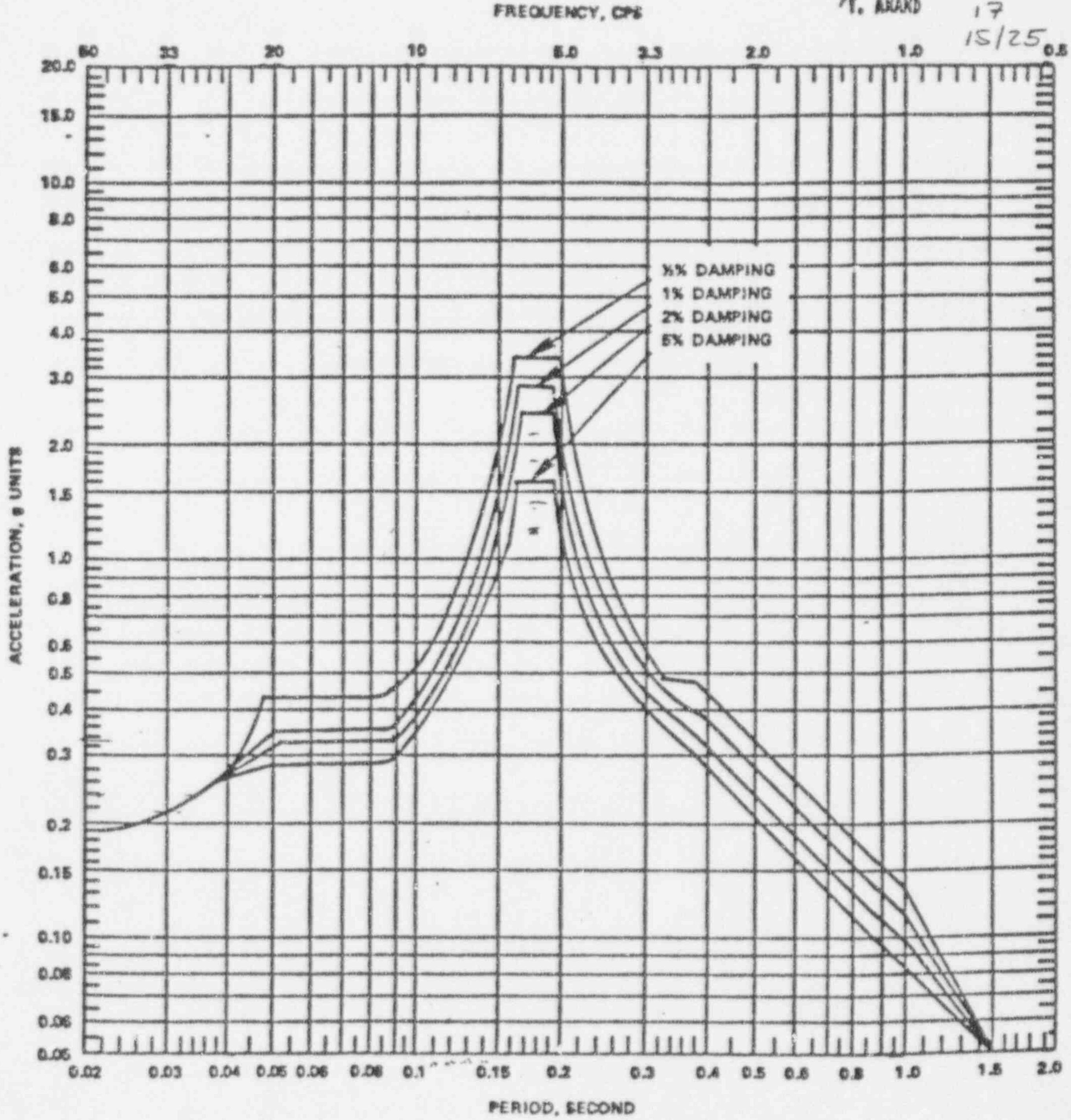


FIGURE 2.3.11

ENRICO FERMI ATOMIC POWER PLANT UNIT 2
HORIZONTAL FLOOR RESPONSE SPECTRA OPERATING BASIS EARTHQUAKE ELEVATION – 684'–6" (SLAB NO. 5) REACTOR/ AUXILIARY BUILDING EAST – WEST COMPONENT

SARGENT & LUNDY REPORT NO. SL-2682

PREPARED BY: OEK 11/5/81
A. KROLIKOWSKI DATE
SYSTEM ENGR: [illegible] 4/4/81
A. ANAND DATE
18
16/125

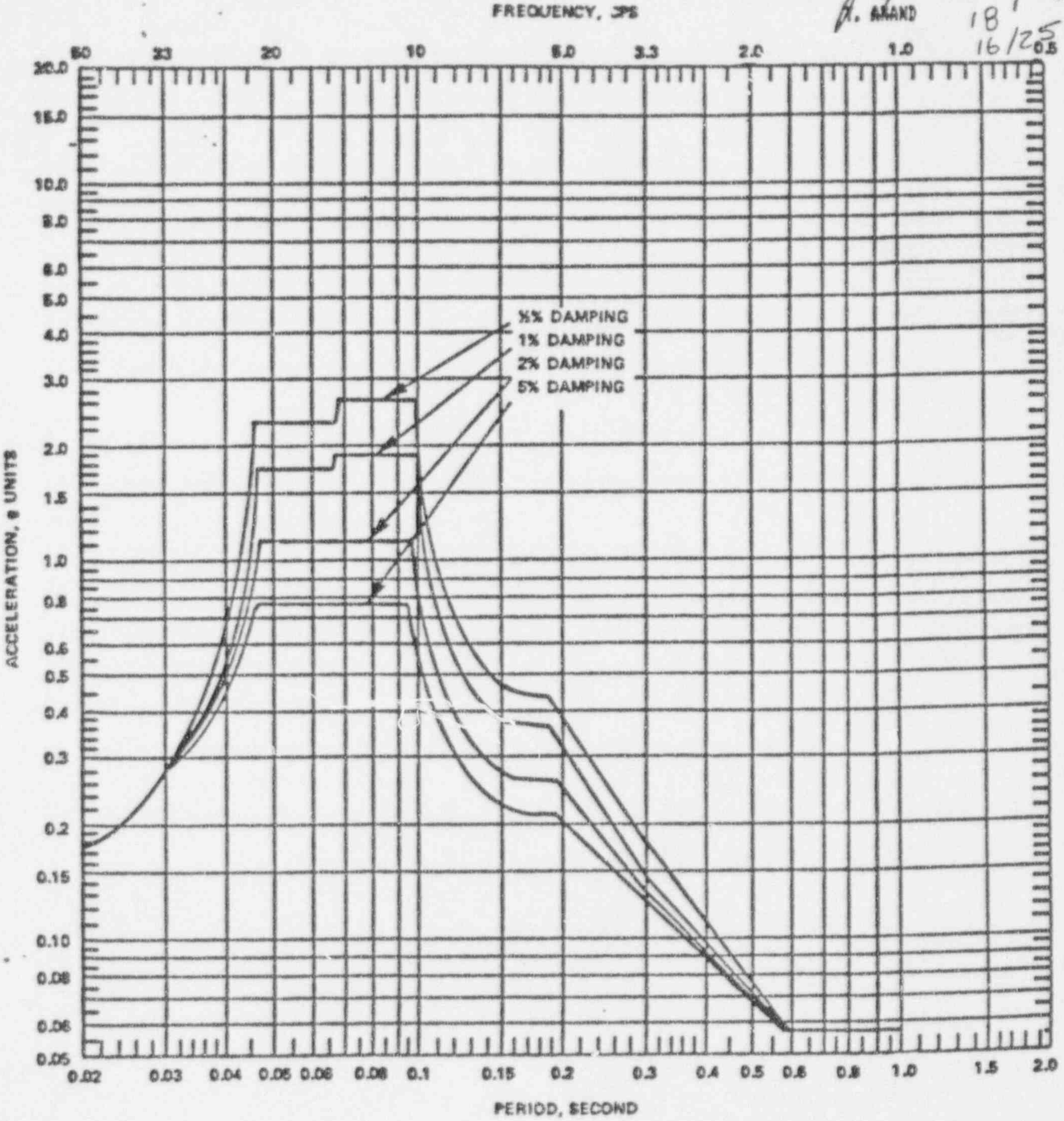


FIGURE 2.3.12

ENRICO FERMI ATOMIC POWER PLANT UNIT 2
VERTICAL RESPONSE SPECTRA OPERATING BASIS EARTHQUAKE REACTOR BUILDING SLAB EL. 641'-6", 659'-6" & 684'-6"

SARGENT & LUNDY REPORT NO. SL-2882

In some cases, where the system was deemed rigid, it was necessary to remove some conservatism. In these situations only the S factor was used as the multiplier.

$$S = \frac{SSE\ ZPA}{OBE\ ZPA}$$

System 2850-2 was analyzed independently using detailed computer analysis.

The procedure taken in analyzing each system is summarized in Tables 2.1 and 2.2.

The analysis procedure are as follows,

1. Stresses based on Fluor-Pioneer output multiplied by factor of S×D.
2. Stresses based on Fluor-Pioneer output multiplied by factor S. This is valid only for rigid systems.
3. Stresses determined from computer run performed by Hopper & Associates.

HOPPER AND ASSOCIATES
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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 14 DATE: MAR 94 PAGE: 20

SUBJECT: ANALYSIS APPROACH BY: A.R. CK: MML SHT: 18 OF 25

SYSTEM NO.	RECTANGULAR DUCT SIZE (INCHxINCH)	ANALYSIS PROCEDURE
2848-1	14X14	1
2848-1-1A	16X20	1
	24X20	1
	32X20	1
	20X16	1
	16X43	1
	16X66	1
2848-1-1B	12X7	1
	18X7	1
	24X7	1
	36X7	1
	48X7	1
	7X24	1
	8X57	2
	8X66	1
	8X36	1
2848-1-1C	16X9	1
	24X9	1
	32X9	1
	8X41	1
	8X66	1
2848-1-2A	32X40	1
	48X40	1
	64X40	1
	32X32	1
	24X49	1
	24X66	1
	24X32	1
2848-1-2B	21X40	1
	21X51	1
	21X80	1

TABLE 2.1 SUMMARY TABLE, RECTANGULAR DUCT

HOPPER AND ASSOCIATES
ENGINEERS

CALCULATION SHEET

TITLE: Response to Concern 14 | DATE: Mar 94 | PAGE: 21

SUBJECT: Analysis Approach | BY: A.R. | CK: MAK | SHT: 19 OF 25

2848-1-2B (CONT'D)	21X53 16X66	1 1
2848-1-2C	29X40 29X55 29X80 29X40 24X40 24X66 24X55	1 1 1 1 1 1 1
2848-1-2D	16X66 16X44 16X20 20X10	1 1 1 1
2848-1-2E	20X10 16X10 16X66	1 1 1
2848-2A	51.5X51.5 78X34	1 1
2848-2B	74X36 51X38	1 1
2848-5	10X10	1
2849-1	20X16 18X12 16X16 12X10 12X8 10X8 10X6 16X12	1 1 1 1 1 1 1 1
2849-2	19X19	1
2849-3	60X26 38X34	1 1

TABLE 2.1 (CONT'D)

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 14 DATE: MAR 94 PAGE: 22

SUBJECT: ANALYSIS APPROACH BY: A.R. CK: MMC SHT: 20 OF 25

2849-3 (CONT'D)	78X34	1
	32X30	1
	50X20	1
	22X12	1
2849-4	12X6	1
2849-5	74X30	1
	124X48	1
2849-6	10X10	1
2849-9	74X30	1
	124X76	1
	72X30	1
	52X44	1
2850-1	24X18	1
	24X14	1
	24X8	1
	34X20	1
	28X14	1
	20X18	1
	20X14	1
	16X14	1
	14X10	1
	14X8	1
	12X12	1
	10X8	1
2850-2	40X32	3
	40X26	3
	34X26	3
	34X20	3
	26X10	3
	16X10	3
	14X10	3
	14X8	3
	10X8	3
	10X6	3
	26X16	3

TABLE 2.1 <u>(CONT'D)</u>

2850-2 (CONT'D)	16X14 14X14 8X8	3 3 3
2850-3	60X26 40X40	1 1
2850-4	20X16 16X12 16X9 12X12 12X10 10X8 8X8	1 1 1 1 1 1 1
2850-5	16X9 14X6 12X6 10X6	2 2 1 1
2850-6	14X6 12X7 12X6 10X6	2 2 2 1
2850-7	36X30 40X20	1 1
2850-8	12X6	1
2850-9	40X20 40X21 10X6	1 1 1
2850-10	10X6	1
2853-1	40X20 32X20 26X20 20X20 20X12 24X18	1 1 1 1 1 1

TABLE 2.1 (CONT'D)

2853-2	36X30 32X30	1 1
2853-3	40X40 38X38 38X32 38X28 38X20 32X20 22X20 20X12	1 1 1 1 1 1 1 1
2854-1	36X30 36X10 34X22 32X20 26X20 24X16 20X10 20X6	1 1 1 1 1 1 1 1
2854-2	36X10 34X22 34X20 30X24 30X12 26X20 20X12 20X9	1 1 2 1 2 1 1 2
2854-3	38X28 38X30 36X30 36X18 48X14 36X12 28X16	1 1 1 1 1 1 1
2854-4	36X12 24X20 24X16 24X8	1 2 2 1

TABLE 2.1 (CONT'D)

4126-1	48X48	1
4316-2	19X19	1
4316-3	10X10	1

ANALYSIS PROCEDURE

(1) Stresses based on Fluor-Pioneer output multiplied by factor of SxD.

(2) Stresses based on Fluor-Pioneer output multiplied by factor S. This is valid only for rigid systems.

(3) Stresses determined from computer run performed by Hopper & Associates.

$$S = \frac{\text{SSE ZPA}}{\text{OBE ZPA}}$$

$$D = \frac{\text{PEAK SPECTRAL ACCELERATION}}{\text{ZPA ACCELERATION}}$$

TABLE 2.1 (CONT'D)

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SYSTEM NO.	CIRCULAR DUCT DIAMETER (INCHES)	ANALYSIS PROCEDURE
2268-2A	15.28 24	 1
2268-2B	12 16	1 1
2268-2C	15.28 24	1 1
2268-2D	12 16	1 1
2268-3A	28 8	1 1
2268-3B	12 16 8	1 1 1
2268-3C	11.28 8	1 1
2268-3D	12 16 8	1 1 1
2268-4A	11.282 8.625	1 1
2268-4B	11.282 8.625	1 1
2848-4A	13.625	1
2848-4B	13.625	1
2849-4	10	1

TABLE 2.2 SUMMARY TABLE, CIRCULAR DUCT

2849-5	46.75	1
2849-6	10	1
2849-7	10 9.625	1 1
2849-8	10 9.625	1 1
2849-9	46.75	1
2849-10	46.75 48	1 1
4126-1	46.75 48.25	1 1

ANALYSIS PROCEDURE

(1) Stresses based on Fluor-Pioneer output multiplied by factor of SxD.

$$S = \frac{\text{SSE ZPA}}{\text{OBE ZPA}}$$

$$D = \frac{\text{PEAK SPECTRAL ACCELERATION}}{\text{ZPA ACCELERATION}}$$

TABLE 2.2 (CONT'D)

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TITLE: NRC DOCKET No 50-341 DATE: MAR 94 PAGE: 28

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3.0 REFERENCES

1. Fluor-Pioneer, Inc. Design Reports on Microfilm Rolls "AA" Through "AH", Project 10-4221, Client: Robert Irsay. (Microfilm Rolls are on file with DECo.)
2. DECo EF2-55,599; November 23, 1981 Floor Response Spectra and Envelopes Reactor-Auxiliary Building and RHR Complex.
3. Sargent & Lundy Report SL-2682, Sept., 1982; Seismic Analysis of the Reactor Auxiliary Building Complex Enrico Fermi Atomic Power Plant, Unit 2.

Attachment 11 to
NRC-94-0050

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PREPARED BY: Alex Reizman 4-7-94

CHECKED BY: Edwin Oberg 4-26-94

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 15 DATE: APR 94 PAGE: 1

SUBJECT: 1.0 INTRODUCTION BY: A.R. CK: SO SHT: 1 OF 2

1.1 PROBLEM STATEMENT

On March 3, 1994, the U.S.N.R.C. submitted Docket No. 50-341 to Detroit Edison Company. This Docket consists of a request for additional information on Fermi 2 CCHVAC system design and operation. In particular, there are several concerns regarding the structural integrity calculations, HA-05/89-686 and HA-09/89-696, performed by Hopper & Associates.

This package is prepared in order to respond to one of the comments posed by the NRC regarding HA-09/89-696. Namely,

> Details of the calculations to determine the maximum permissible internal pressure from the stresses under combined loads were not reported. Maximum permissible internal pressures are shown for two load combinations: DW and DW + SSE. An examination of the stated values indicates that for the rectangular ductwork the permissible internal pressure under combined DW + SSE loading is higher than under DW alone; in fact, in some sections it is considerable higher (in one case about 20 times larger). The basis for these results has not been presented. Since the details of these calculations are not shown, it is not possible to assess the quality or validity of these calculations, and the maximum allowable pressure results must therefore be considered as questionable.

1.2 INVESTIGATION APPROACH

From the 55 ducting systems reviewed in HA-09/89-696, two duct sizes will be chosen for a detailed evaluation. The basis for selecting the two duct sizes is to capture the representative procedure used in analyzing all systems. For this reason, one of the duct sizes will be shown not to buckle under any loading combination, and the other buckles under the DW+Seismic loading.

An analysis approach for determining the maximum allowable internal pressure will be shown. A detailed calculation of two representative duct sizes will be performed to demonstrate the procedure.

SYSTEM DESCRIPTION

Detailed calculations will be performed for two duct systems, namely

1. 2849_1 (DUCT SIZE 20"x16")
2. 2848_1_1B (DUCT SIZE 7"x24")

These systems are representative of all the rectangular ducts reviewed in HA-09/89-696.

SYSTEM	LOCATION	GAGE	THICKNESS	
			GALVANIZED	BASE METAL
① 20"x16"	SLAB 5	18	.0516"	.0478"
② 7"x24"	SLAB 5	18	.0516"	.0478"

STIFFNER

Angle 1½" x 1½" x ⅛" @ 30" O.C.

3.1 LOAD CONDITIONS

1. DEAD LOAD
2. DEAD LOAD + SEISMIC

3.2 EXISTING ANALYSIS

An analysis of the ducting systems was performed by Fluor-Pioneer [Ref 1]. The results of these analyses are documented in form of design reports. All the structural evaluation work in HA-09/89-696 is based upon data contained in these design reports. The assumptions made by Fluor-Pioneer are summarized in HA-09/89-696.

3.3 LOAD FACTORS

As discussed in "Response to Concern 14" [Ref 7] the following load factors are used to magnify the Fluor-Pioneer Loads,

SLAB 3: L.F. = 22

SLAB 5: L.F. = 20

3.4 ANALYSIS CRITERIA

The duct stresses are compared to the requirements of ANSI/ASME N509-1980, ¶ 5.10.3.3. [Ref 5]

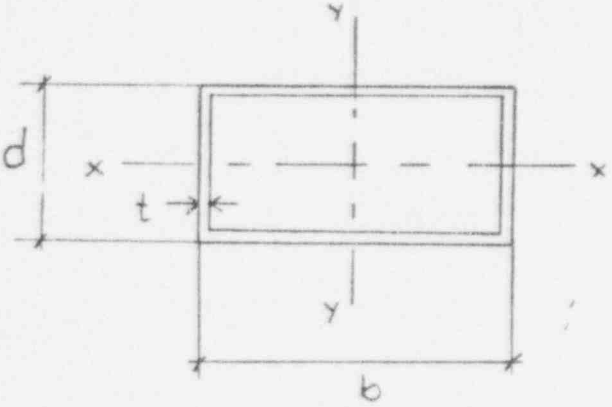
i) For Normal Plant Operation Loads
allowable stress = 0.6 of yield stress

ii) For Combined Loads including Site Specific Earthquake (SSE) loads
allowable stress = 0.9 of yield stress

iii) For Deflection
Panel - 1/8" per foot of maximum panel span.
Stiffner - 1/8" per foot of span

3.5 DUCT CAPABILITY EVALUATION

3.5.1 DUCT SECTION PROPERTIES

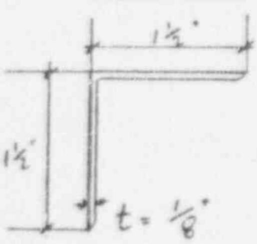


$$I_{xx} = \frac{t d^2}{6}(3b+d)$$

$$I_{yy} = \frac{t b^2}{6}(b+3d)$$

SYSTEM	d	b	t GALV.	t BARE	I_{xx} (in⁴) GALV.	I_{xx} (in⁴) BARE	I_{yy} (in⁴) GALV.	I_{yy} (in⁴) BARE
2849-1	16"	20"	.0516"	.0478"	167.3	155	233.9	216.7
2848-1-1B	7"	24"	.0516"	.0478"	33.3	30.8	222.9	206.5

ANGLE



$$\bar{y} = \frac{(1.5)(\tfrac{1}{8})(1\tfrac{1}{2}-\tfrac{1}{16}) + (1.5)(\tfrac{1}{8})(\tfrac{1.5}{2})}{2(1.5 \times \tfrac{1}{8})}$$

$$\bar{y} = 1.094"$$

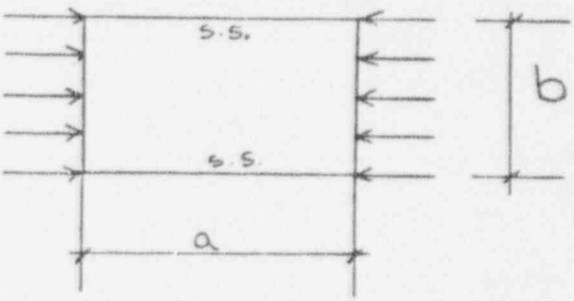
$$I_{xx} = \frac{(1.5)^3(.125)}{12} + (1.5 \times \tfrac{1}{8})(1.094 - \tfrac{1.5}{2})^2 + (1.5 \times \tfrac{1}{8})(1.5 - \tfrac{1}{16} - 1.094)^2$$

$$I_{xx} = .079 \text{ in}^4$$

3.5.2 BUCKLING STRESS [Ref 2, Pg 350]

Plate elastic compressive buckling stress.

$$\sigma_{cr} = \frac{K \pi^2 E}{12(1-\nu^2)} \left(\frac{h}{b}\right)^2$$



where,

$E = 29 \times 10^6$ psi

$\nu = .29$

$h = .0516''$ (galvanized)

$.0478''$ (bare)

$K = \left(\frac{a}{b} + \frac{b}{a}\right)^2$, $K = 4$ ($b/a < 1$)

a = distance between stiffners

b (in)	a (in)	b/a	K
20	30	<1	4
24	30	<1	4

For 2849_1

σ_{cr} = 693 psi (galv.)

σ_{cr} = 595 psi (bare)

For 2848_1_1B

σ_{cr} = 482 psi (galv)

σ_{cr} = 413 psi (bare)

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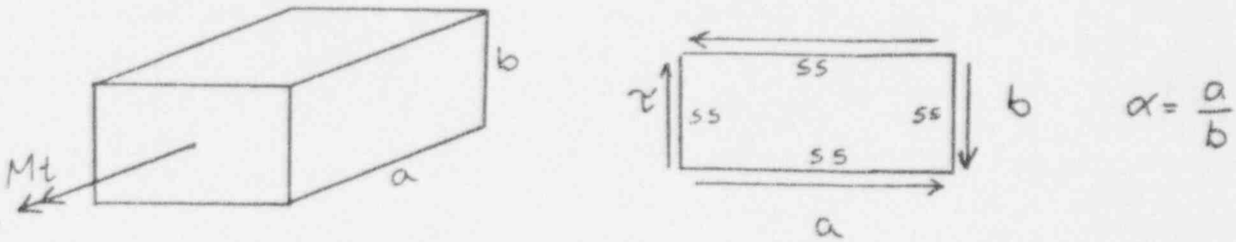
For torsion buckling, shear stress,



$$\tau_{cr} = K \frac{E}{1-\nu^2} \left(\frac{t}{b}\right)^2$$ [Ref 4, Table 35 Case 4]

where,

$E = 29 \times 10^6$ psi

$\nu = .29$

$K = 4.4$, the lower bound for an infinite plate

$t = .0516"$ (galv.)

$.0478"$ (bare)

For,

b	τ_{cr} (galv)	τ_{cr} (bare)
20	927 psi	796 psi
24	644 psi	553 psi

3.6 PRESSURE CAPABILITY PROCEDURE

Rectangular duct panel dimensions are based upon the cross-section side length and the stiffner spacing. A simple analysis approach, shown below, was taken to determine the allowable operating pressures for various stress conditions.

From Timoshenko's, Theory of Plates and Shells. [Ref 3, Pg 13]

Membrane Stress (max.) -

$$\sigma_m = \sigma_1 = \frac{E u^2}{3(1-\nu^2)}\left(\frac{h}{\ell}\right)^2 \qquad \text{[EQ 1]}$$

Bending Stress (max) -

$$\sigma_B = \sigma_2 = \frac{q}{2}\left(\frac{\ell}{h}\right)^2 \Psi_1(u) \qquad \text{[EQ 2]}$$

$$\Psi_1(u) = \frac{3(u - \tanh(u))}{u^2 \tanh(u)} \qquad \text{[EQ 3]}$$

u is found from

$$\frac{E^2 h^8}{(1-\nu^2)^2 q^2 \ell^8} = -\frac{81}{16u^7 \tanh(u)} - \frac{27}{16u^6 \sinh^2(u)} + \frac{27}{4u^8} + \frac{9}{8u^6}$$

[EQ 4]

where,

$E = 29 \times 10^6$ psi

$\nu = .29$

$h = .0516"$ (galvanized)

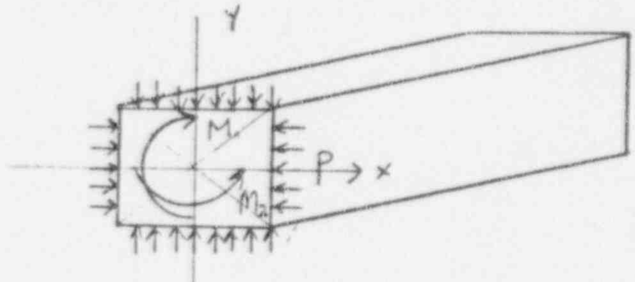
$= .0478"$ (bare)

q = pressure in psi

The pressure capabilities are calculated for Deadweight and for Deadweight + Seismic load conditions. If the Deadweight + Seismic load pressure capability is less than the Deadweight load capability, only the membrane stresses are considered. Thus 'u' is determined directly from [Eq 1], and then inserted into [Eq 4] to solve for allowable pressure 'q'. again, this pressure capability is based on membrane stress alone, since bending stress vanishes with the formation of a hinge by yielding at the boundaries.

3.7 MECHANICS OVERVIEW

The loading acting on the duct is as shown



Note:

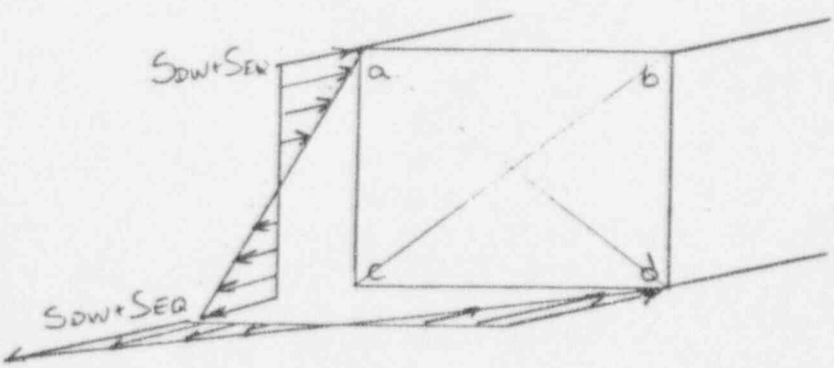
$M_1 = M_x$

$M_2 = M_y$

M_1, M_2 = Loading due to Deadweight and Seismic.

P = Loading due to Pressure

STRESS – Seismic + Deadweight



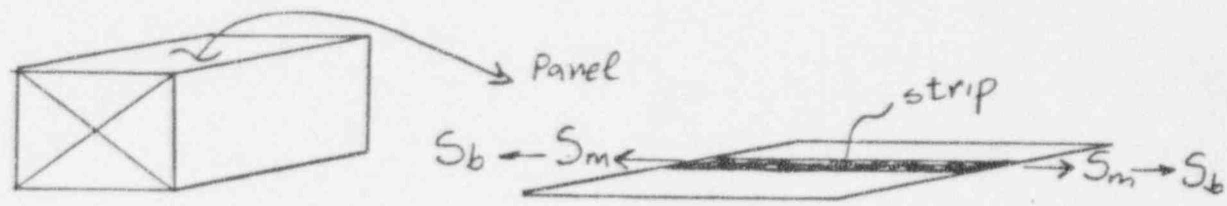
S_{EQ} = Earthquake Stress

S_{DW} = Deadweight Stress

Corner "b" is under worst case compression

Corner "c" is under worst case tension

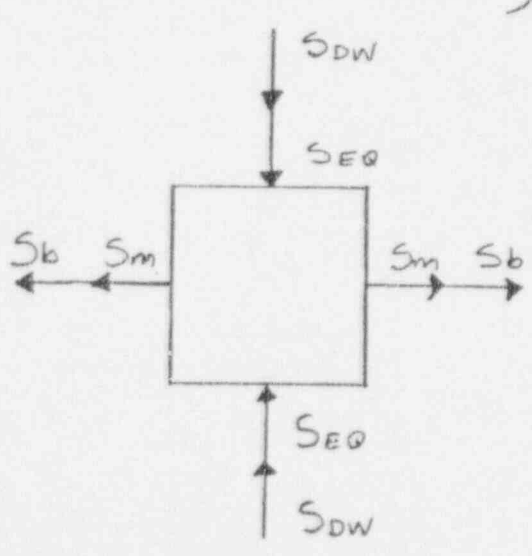
STRESS - Pressure



S_m = Membrane Stress due to pressure

S_b = Bending Stress due to pressure

In our situation the worst case location is at corner "b", where the state of stress is as shown,



From Tresca yield criterion [Ref 6, Pg 317], the maximum stress is,

$$S_{DW} + S_{EQ} + S_b + S_m \leq .9\, S_{yield}$$

3.8 DEFLECTION LIMIT [Ref 3, Pg 15]

The maximum deflection on a panel is calculated from,

$$w = \frac{q l^4}{384 D} f_1(u) \qquad \text{[Eq 5]}$$

$$D = \frac{E h^3}{12(1-\nu^2)} \qquad \text{[Eq 6]}$$

$$f_1(u) = \frac{24}{u^4}\left[\frac{u^2}{2} + \frac{u}{\sinh u} - \frac{u}{\tanh u}\right] \qquad \text{[Eq 7]}$$

Stiffner deflection is calculated from,

$$w = \frac{5 p l^4}{384 E I} \qquad p = \#/\text{in} \qquad \text{[Eq 8]}$$

3.8.1 AREA REDUCTION

To obtain an upper bound area reduction, assume panel deflection occures around the entire perimeter.

$$A.R. = \frac{A_0 - A_t}{A_0} \times 100\%$$

A_0 = Initial cross-sectional area

A_t = Area taken by deflection

3.9 APPLIED LOADS [Ref 1]

2849_1

From Fluor-Pioneer Output

Deadweight,

TORSION = 0 ft#

M_1 (OUT-PLANE) = 2 ft#

M_2 (IN-PLANE) = 210 ft#

Seismic, including Load Factor multiplier,

TORSION = 1 (20) = 20 ft#

M_1 (OUT-PLANE) = 3 (20) = 60 ft#

M_2 (IN-PLANE) = 4 (20) = 80 ft#

2848_1_1B

Deadweight

TORSION = 8.05 ft#

M_1 (OUT-PLANE) = 4.56 ft#

M_2 (IN-PLANE) = 257.3 ft#

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Seismic,

TORSION = 15 (20) = 300 ft#

M_1 (OUT-PLANE) = 43 (20) = 860 ft#

M_1 (IN-PLANE) = 47 (20) = 940 ft#

Note: Fluor-Pioneer did not publish axial forces. But based on the type of analysis performed, it is expected that axial forces will not be significant.

4.1 ANALYSIS OF 2849_1

DW + Pressure

$M_1 = 2 \text{ ft\#} \quad M_2 = 210 \text{ ft\#}$

$$\sigma_{DW} = \frac{M_2 \left(\frac{12 \text{ in}}{ft}\right)\left(\frac{d}{2}\right)}{I_{xx}} + \frac{M_1 \left(\frac{12 \text{ in}}{ft}\right)\left(\frac{b}{2}\right)}{I_{yy}}$$

$$= \frac{(210)(12)(16)}{(2)(167.3)} + \frac{(2)(12)(20)}{(2)(233.5)} \quad \text{(galv.)}$$

$$= 122 \text{ psi} < \sigma_{cr} = 693 \text{ psi} \quad \text{(galv.)}$$

∴ No buckling

$$= \frac{(210)(12)(16)}{(2)(155)} + \frac{(2)(12)(20)}{(2)(216.7)} \quad \text{(bare)}$$

$$= 132 \text{ psi} < 595 \text{ psi} \quad \text{(bare)}$$

again, No buckling

To calculate the allowable pressure,

$$\sigma_{DW} + (\sigma_B + \sigma_m) = .6\,\sigma_y$$

$$122 + (\sigma_B + \sigma_m) = .6\,(36{,}000) \quad \text{(galv.)}$$

$$\sigma_B + \sigma_m = 21{,}478 \text{ psi} \quad \text{(galv.)}$$

$$\sigma_B + \sigma_m = 21{,}468 \text{ psi} \quad \text{(bare)}$$

The difference between galvanized and bare metal is small.

Using the equation shown in Section 3.6 of this package, the pressure 'q' is determined. Adding [EQ 1] and [EQ 2] we get,

$$\sigma_m + \sigma_B = \frac{E u^2}{3(1-\nu^2)}\left(\frac{h}{\ell}\right)^2 + \frac{q}{2}\left(\frac{\ell}{h}\right)^2\left[\frac{3(u-\tanh(u))}{u^2\tanh(u)}\right]$$

Solving for q_1,

$$q_1 = \left[(\sigma_m+\sigma_B) - \frac{E u^2}{3(1-\nu^2)}\left(\frac{h}{\ell}\right)^2\right] * 2 * \left(\frac{h}{\ell}\right)^2 * \left[\frac{3(u-\tanh(u))}{u^2\tanh(u)}\right]^{-1}$$

Similarly solving [EQ 4] for q_2,

$$q_2 = \left[\frac{E^2 h^8}{(1-\nu^2)^2 \ell^8} * \left(-\frac{81}{16u^2\tanh(u)} - \frac{27}{16u^4\sinh^2(u)} + \frac{27}{4u^8} + \frac{9}{8u^6}\right)^{-1}\right]^{\frac{1}{2}}$$

A solution is obtained by iterating 'u' until $q_1 = q_2$.

Thus, for $\sigma_B + \sigma_m = 21{,}478$ psi , $\ell = 20''$
at $u = 7.0$ (galv.) $u = 7.61$ (bare)
$q_1 = q_2 = .655$ psi (") $q_1 = q_2 = .60$ psi (")
or, allowable pressure is,

$$\boxed{\begin{array}{ll} q = .655\left(\frac{407}{14.7}\right) = 18.1'' H_2O & \text{(galv.)} \\ q = .60 \text{ psi or } 16.6'' H_2O & \text{(bare)} \end{array}}$$

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DW + SEISMIC + PRESSURE

$M_1 = 80\ ft\#$ $M_2 = 60\ ft\#$

$$\sigma_s = \frac{(80)(12)(16)}{2(167.3)} + \frac{(60)(12)(20)}{2(233.9)} \quad (galv.)$$

$= 77\ psi < 693\ psi$

$\therefore$ No buckling

$$= \frac{(80)(12)(16)}{2(155)} + \frac{60(12)(20)}{2(216.7)} \quad (bare)$$

$= 83\ psi < 595\ psi$

Now,

$$\sigma_{DW} + \sigma_s + (\sigma_B + \sigma_m) = .9\,\sigma_y$$

$$122 + 77 + (\sigma_B + \sigma_m) = .9(36000) \quad (galv.)$$

$$132 + 83 + (\sigma_B + \sigma_m) = .9(36000) \quad (bare)$$

So,

$\sigma_B + \sigma_m = 32{,}200\ psi$ (galv.)

$\sigma_B + \sigma_m = 32{,}185\ psi$ (bare)

Solving for q as before,

at $u = 8.74$ (galv) $u = 9.5$ (bare)

$q = 1.2\ psi$	or	$32.6''\ H_2O$	(galv.)
$q = 1.1\ psi$	or	$30''\ H_2O$	(bare)

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4.2 ANALYSIS OF 2848_1_1B

DW + PRESSURE

$M_1 = 4.56$ ft # $\quad M_2 = 257.3$ ft #

$$\sigma_{DW} = \frac{(257.3)(12)(.7)}{2(33.3)} + \frac{(4.56)(12)(24)}{2(222.9)} \quad \text{(galv.)}$$

$$= 327 < 482 \text{ psi}$$

$\therefore$ No buckling

$$= \frac{(257.3)(12)(7)}{2(30.8)} + \frac{(4.56)(12)(24)}{2(206.5)} \quad \text{(bare)}$$

$$= 354 < 413 \text{ psi}$$

$$\sigma_{DW} + (\sigma_B + \sigma_m) = .6\,\sigma_y$$

$(\sigma_B + \sigma_m)_{galv} = 21{,}273$ psi $\quad (\sigma_B + \sigma_m)_{bare} = 21246$ psi

at $u = 8.5$ (galv.) $\quad u = 9.2$ (bare)

$q = .53$ psi	or $14.6''$ H_2O	(galv.)
$q = .48$ psi	or $13.4''$ H_2O	(bare)

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SUBJECT: ANALYSIS BY: A.R. CK: GO SHT: 5 OF 11

DW + SEISMIC + PRESSURE

$M_1 = 860 \text{ ft\#} \quad M_2 = 940 \text{ ft\#}$

$$\sigma_S = \frac{(.940)(12)(7)}{2(33.3)} + \frac{(.860)(12)(24)}{2(222.9)}$$

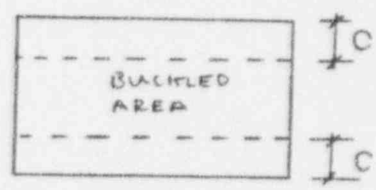
$= 1741 > 482$ psi

$\therefore$ buckling occures.

The effective strip is determined from

$$c = \frac{\pi h}{\sqrt{12(1-\nu^2)}} \sqrt{\frac{E}{\sigma_y}}$$

$= 1.39$ inch



The redistributed stresses are then,

$\sigma_{max}(2c + 2h) + \sigma_{cr}(b - (2c+2h)) = (\sigma_S + \sigma_{DW})b$

$\sigma_{max}(2.88) + 482(24 - 2.88) = (1741+327)\,24$

$\sigma_{max} = 13{,}700$ psi

Now

$\sigma_{max} + (\sigma_B + \sigma_m) = .9\sigma_y$

$13{,}700 + (\sigma_B + \sigma_m) = .9(36{,}000)$

$(\sigma_B + \sigma_m)_{galv} = 18{,}700$ psi

at $u = 7.91$ (galv.) $u = 8.6$ (bare)

$$\boxed{\begin{array}{llll} q = .44 \text{ psi} & \text{or} & 12.1'' \; H_2O & \text{(galv.)} \\ q = .40 \text{ psi} & \text{or} & 11.2'' \; H_2O & \text{(bare)} \end{array}}$$

As mentioned previously, if the DW + Seismic load pressure capability is less than DW load capability, only the membrane stresses are considered. Thus, u is determined from

$$\sigma_m = \frac{E u^2}{3(1-\nu^2)} \left(\frac{h}{\ell}\right)^2 = 18,700 \text{ psi}$$

where,

$\ell = 24"$

$$u = \sqrt{\frac{3(1-.29^2)(18,700)}{29\times10^6}\left(\frac{24}{.0516}\right)^2}$$

$u = 19.6$ (galv) $\quad$ $u = 21.1$ (bare)

Inserting 'u' into [Eq 4] from Section 3.6 and solving for q,

$q = 5.4$ psi	or	150" H_2O	(galv.)
$q = 5.0$ psi	or	137" H_2O	(bare)

4.3 DEFLECTION

4.3.1 PANEL - due to buckling

In this analysis the buckled system is 2848_1_1B Duct size 7"x24".
The largest panel size is 24"x30".
From [Ref. 2, Pg 413]

$$w = f \cos \frac{\pi x}{2b} \cos \frac{\pi y}{2a}$$

where,

$$f = \left[\frac{6.42 a^2 e - 4.058 h^2}{5.688} \right]^{1/2}$$

$$e = \frac{(1-\nu^2)\sigma_{cr}}{E}$$

a = ½ width, 24/2 = 12"

h = .516" (galv.)

.0478" (bare)

ν = .29

E = 29×10^6 psi

This result is for a simply supported square plate loaded above its elastic buckling limit. To obtain a bounding deflection value for rectangular plates, multiply the deflection by a factor of 2, to obtain a conservative result. Selecting,

$x = y = 0$, i.e. at center of plate

$\omega' = 2f$

For $7'' \times 24''$

$\sigma_{cr} = 482$ psi (galv.)

413 psi (bare)

$$f = \left[\frac{6.42(12)^2\left(\frac{(1-.29^2)\,482}{29 \times 10^6}\right) - 4.058(.0516)^2}{5.688} \right]^{1/2}$$

$f = .03''$ (galv.)

$f = .03''$ (bare)

$\omega' = .03 \times 2 = .06''$

4.3.2 PANEL - Pressure (7"x24")

According to ANSI N509-1980 [Ref 5] the maximum panel deflection is 1/8" per foot of maximum panel span. In our case the maximum span is the distance between stiffners, ie 30". Therefore,

$$w_{max} = w_p - w' = \tfrac{1}{8}\left(\tfrac{30}{12}\right) - .06 = .2525 \text{ inch}$$

Solving [EQ 5], from section 3.8 in terms of q,

$$q = \frac{384\, w_{max}\, D}{\ell^4}\left[f_1(u)\right]^{-1}$$

also,

$$q = \left[\frac{E^2 h^8}{(1-\nu^2)^2 \ell^8}\cdot\left(-\frac{81}{16u^7 \tanh(u)} - \frac{27}{16u^6 \sinh^2(u)} + \frac{27}{4u^8} + \frac{9}{8u^6}\right)^{-1}\right]^{1/2}$$

Iterating u will yield the allowable pressure,

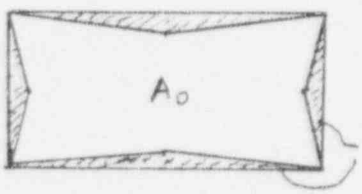
at $u = 13.6$ $q_{max} = .784$ psi or 21.7" H_2O (galv)

$u = 14.65$ $q_{max} = .714$ psi or 19.8" H_2O (bare)

4.4 AREA REDUCTION

Assuming the panels will deflect the ANSI N509-1980 allowable distance, the reduction in airflow area is determined,

$$w_{max} = \frac{1}{8}\left(\frac{30}{12}\right) = .3125''$$



$$A.R = \frac{A_o - A_t}{A_o} \times 100\%$$

For $20'' \times 16''$

$$A.R. = \frac{20 \times 16 - 2[(.3125)(12) + (.3125)(8)]}{20 \times 16} \times 100\%$$

= 96.1% left over or 3.9% reduction

For $7'' \times 24''$

$$A.R. = \frac{24 \times 7 - 2[(.3125)(3.5) + (.3125)(12)]}{24 \times 7} \times 100\%$$

= 94.2% left over or 5.8% reduction

4.5 STIFFNER

In this analysis the longest stiffner length is 24". According to ANSI N509-1980 the allowable stiffner deflection is

$$w_{max} = \frac{1}{8}\left(\frac{24}{12}\right) = .25''$$

Solving [Eq 8] for p,

$$p = \frac{384 E I w_{max}}{5 \ell^4}$$

where,

$I = .079 \text{ in}^4$

$\ell = 24''$

$$p = \frac{384\,(29 \times 10^6)\,(.079)\,(.25'')}{5\,(24)^4}$$

$p = 132$ #/in

Now

$q_{max} = \frac{p}{a}$ where $a = 30''$, tributary length

$$q_{max} = \frac{132}{30} = 4.4 \text{ psi}$$

or $121''$ H_2O

CONCLUSIONS

In response to a comment posed by the NRC in Docket NO. 50-341, a more detailed procedure for calculating allowable pressure, as published in HA-09/89-696, is presented. This is achieved by selecting two representative ducting systems and performing detailed calculations to determine the allowable pressures.

A summary of the results is shown in Table 5.1. However, the results are not as important as the procedure in obtaining them. The purpose of this package is to expand on the results published in HA-09/89-696. This is achieved by providing a sample of clear and complete calculations.

In Summary

System No.	Duct Size	Analysis Type	Allowable Pressure		
			DW (inch-water)	DW + Seis. (inch-water)	DW + Seis. (membrane stresses only) (inch-water)
2848_1_1B	7"x24"	galvanized bare metal	14.6 13.4	12.1 11.2	150 137
2849_1	20"x16"	galvanized bare metal	18.1 16.6	32.6 30	- -

TABLE 5.1 SUMMARY OF RESULTS

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CALCULATION SHEET

TITLE: RESPONSE TO CONCERN 15 DATE: APR. '94 PAGE: 29

SUBJECT: 6.0 REFERENCES BY: A.R. CK: GO SHT: 1 OF 2

REFERENCES

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3. Timoshenko. "Theory of Plates and Shells", 2nd Edition, 1959.

4. Young, "Roarks Formulas for Stress and Strain", 6th Edition, McGraw-Hill, 1989. [Table 35, Case 4]

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TITLE: RESPONSE TO CONCERN 15 DATE: APR 94 PAGE: 30

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7. Hopper & Associates, " DECo - USNRC Docket No. 50-341 Fermi 2 CCHVAC Ducting Systems, Concern Item No. 14", April 1994.