MAINE YANKEE ATOMIC POWER COMPANY •
ENGINEERING OFFICE
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FRAMINGHAM, MASSACHUSETTS 01701
617-872-8100

May 1, 1981
FMY - 81-68
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United States Nuclear Regulatory Commission
Washington, D.C. 20555

Attention: Office of Nuclear Reactor Regulation
Division of Licensing
Robert A. Clark
Operating Reactors Branch No. 3

References:

(a) License No. DPR-36 (Docket 50-309)
(b) USNRC letter from Robert W. Reid, DOR to R. H. Groce, dated November 29, 1978
(c) YAEC letter to USNRC, dated December 29, 1978
(d) USNRC letter to MYAPC, dated October 22, 1980
(e) USNRC letter to MYAPC, dated September 17, 1979
(f) MYAPC letter to USNRC, WMY 79-126, dated November 8, 1979
(g) MYAPC letter to USNRC, WMY 79-144, dated December 10, 1979
(h) MAYPC letter to USNRC, WMY 79-146, dated December 12, 1979
(i) MYAPC letter to USNRC, WMY 80-78, dated May 16 1980

Enclosures:

(A) Valve Performance Calculations and Conclusions from Allis-Chalmers Company
(B) Valve Seismic Calculations and Conclusions from Allis-Chalmers Company
(C) Resultant Off-Site Doses from A LOCA During ON-Line Containment Purge at Maine Yankee

Subject: Unlimited Containment Purging During Normal Plant Operation

Dear Sir:

Reference (b) requested that the Maine Yankee Atomic Power Company commit to (1) cease all containment purge during operation and to propose an amendment relative to that commitment, or (2) provide justification for the continued practice of purging during operation whether limited, or unlimited.

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Reference (c) was provided in response to that request and reflected our position that unlimited purging could be justified for the Maine Yankee Plant.

Pending the completion of the long-term review to justify unlimited purging, Maine Yankee committed to limiting containment purge to 90 hours per year when not in a cold shutdown or refueling mode as stated in References (c) and (h). This section was found acceptable by the NRC in Reference (d).

The intent of this submittal is to provide the NRC with the necessary information to justify unlimited purge. Much of the required information for this justification has already been submitted to the Commission by Maine Yankee in References (c), (f), (g), (h) and (i).

The following information coupled with the information already submitted should complete the Maine Yankee justification for unlimited containment purge:

Operability of Containment Purge and Vent Valves

Reference (b) indicated that purging during normal operation may be permitted if the containment purge isolation valves are capable of closing against the dynamic forces of a design basis loss-of-coolant accident. Reference (e) from the NRC provided guidelines for valve operability when closing against the dynamic forces of a design basis loss-of-coolant accident. This evaluation was performed on the 42 inch containment purge valves at Maine Yankee by Allis-Chalmers, the valve manufacturer. The results of this evaluation are provided in Enclosure (A).

The original purchase requirements for the 42 inch containment purge valves required that the valves close in approximately 1.5 seconds from the full open position. It should be noted as specified in Reference (h), Maine Yankee will continue to restrict valve opening to 50° open (90° being full open). During a design basis accident, the containment pressure will reach the containment isolation set point of 5 psig in approximately 0.5 seconds. Therefore, assuming 0.1 seconds for instrument response, the containment purge valves should be closed in approximately 2.1 seconds at a respective containment pressure of approximately 15 psig.

In summary, the results of the evaluation performed by Allis-Chalmers show that the 42 inch containment purge valves are structually adequate to close against the dynamic loads of the design basis accident in 2.1 seconds. In addition, the results show that the valves can also close against a hypothetical pressure drop across each valve of up to 27 psi, and maintain a design safety factor of 2 on all valve parts and valve operator. At an even greater hypothetical pressure drop of 60 psi across each valve the combined shear stress on the valve shaft would increase to 14,928 psi. The shear yield for the shaft material is approximately 18,000 psi. Realizing that the 14,928 psi is a maximum stress that occurs only at one point on the outside diameter of the shaft, it is Allis-Chalmer's position that even at this hypothetical pressure, the valve would close and structural integrity will be maintained. Note, maximum containment pressure during a postulated design basis accident is apprximately 55 psig.

The guidelines provided for valve operability in Reference (c) also specified that siesmic loadings be addressed. The following was the original seismic requirements for these valves:

1. The equipment shall be designed within allowable working stresses for all normal loads, plus an earthquake load corresponding to a horizontally applied average acceleration of .09 g and a vertical pf .06 g, both acting simultaneously.

2. The equipment shall also be designed to withstand, without loss of structural integrity or function, all normal loads plus an earthquake load corresponding to a horizontally applied average acceleration of .12 g and a vertical of .08 g, both acting simultaneously.

Since these seismic requirements did not specify frequency, Maine Yankee requested Allis-Chalmers to re-evaluate the seismic integrity of the valves and operators at 2.9 g's based on the appropriate amplified response spectrum for Maine Yankee. The seismic evaluation and conclusions from Allis-Chalmers are provided in Enclosure (B).

In summary, the stresses calculated for the internal valve parts and operator mounting components were found acceptable.

Radiological Analysis

A radiological analysis was performed to determine the incremental dose associated with a loss-of-coolant accident occuring during a period of on-line purging. Several break sizes were analyzed, and in all cases the total calculated accident doses are within the 10 CFR100 guideline values. The details and results regarding this analysis are provided in Enclosure (C).

Because of the complexities that arise when considering the analysis of various break sizes, break locations and the associated impact of on-line purging, Maine Yankee proposes to install an additional containment purge isolation signal which would close the purge valves on a pre-set low pressurizer pressure signal. This signal, which will be set at approximately 2200 psig, would be anticipatory to a safety injection actuation signal and would reduce the effects of small and intermediate breaks.

The proposed 2200 psig trip for the purge valves will be designed in accordance with 1E criteria for safety grade instrument systems. In addition to the 2200 psig trip, a high pressurizer pressure trip for the purge valves will also be installed. The set point for this high pressure trip will be below the set point pressure for the pressurizer safety valves. Installation of these trips will be contingent upon the Commission approving Maine Yankee's request for unlimited purge. Upon approval, on-line purge will be limited to 90 hours until the installation of these additional purge valve trips have been completed. During either heat-up or cool down of the primary system, these two addition purge valve trips may be manually by-passed. The design of this manual by-pass will not affect the other purge valve trips previously identified in the References.

Additional conservatism is included in the doses resulting from a LOCA while on-line purging, since the releases calculated from the containment did not take into account the fact that the purge valves are limited to 50° open (90° bring full open) and the throttling action created during valve closure. The calculation was done assuming a continuous 42 inch diameter opening from the containment to atmosphere throughout the specified time period.

Miscellaneous

An error has been found in the response to question 2C on Page 3 of Reference (g). The response stated that the transition from piping to ductwork occurred outside the room containing the purge exhaust valves. The transition from pipe to circular ductwork occurs approximately one foot from the outer exhaust purge valve. The stuctural integrity and supporting of the short section of 12 gauge circular weld duct within the valve area has been reviewed and we have concluded that the information given in the response regarding this area is still correct even though it is classified as circular weld duct and not pipe.

In the area where the welded circular duct exists the purge valve enclose through the concrete roof slab, there is approximately a 2 inch clearance between the duct and the circular hole through the slab. To better seal off this area, the clearance will be reduced by the installation of an apron around the duct where it exits the slab.

It has also been found that a supply air duct to the auxiliary feedwater pump room is routed through the enclosure containing the containment purge exhaust filter and fan. Seismic failure of the purge duct in this area of enclosure could result in damage to this supply duct. This problem will be addressed and necessary changes made (if necessary) as part of our efforts related to IE Bulletin 79-01B.

Based on the enclosed information, Maine Yankee concludes that the practice of purging during operation is justified and that it should not be limited to 90 hours per year. Purging during operation will not be done on a routine basis and emphasis will continue to be placed on operating the containment in a passive mode as much as possible and on limiting all purging and venting times to as low as reasonably achievable.

Until further notice from the Commission, Maine Yankee will continue to limit purging during operation to 90 hours per year.

Very truly yours,

MAINE YANKEE ATOMIC POWER COMPANY

Robert H Groce

Robert H. Groce
Senior Engineer - Licensing

RHG/jlb

ENCLOSURE (A)

CUSTOMER Maine Yankee	DATE 2/18/80	SHEET 1 OF 10	
SUBJECT 42" Containment Supply & Exhaust Valves		PRELIM.	FINAL x
DRAWING NUMBER	ENGINEERING CALCULATION SHEET	CALCULATED BY Zeiders	
	LITHO IN U.S.A.–A-C		
	ALLIS-CHALMERS FORM 6715-1	P. SCHWARZ 3-5-80	

Valves: 42"-75WR Streamseal Butterfly Valves

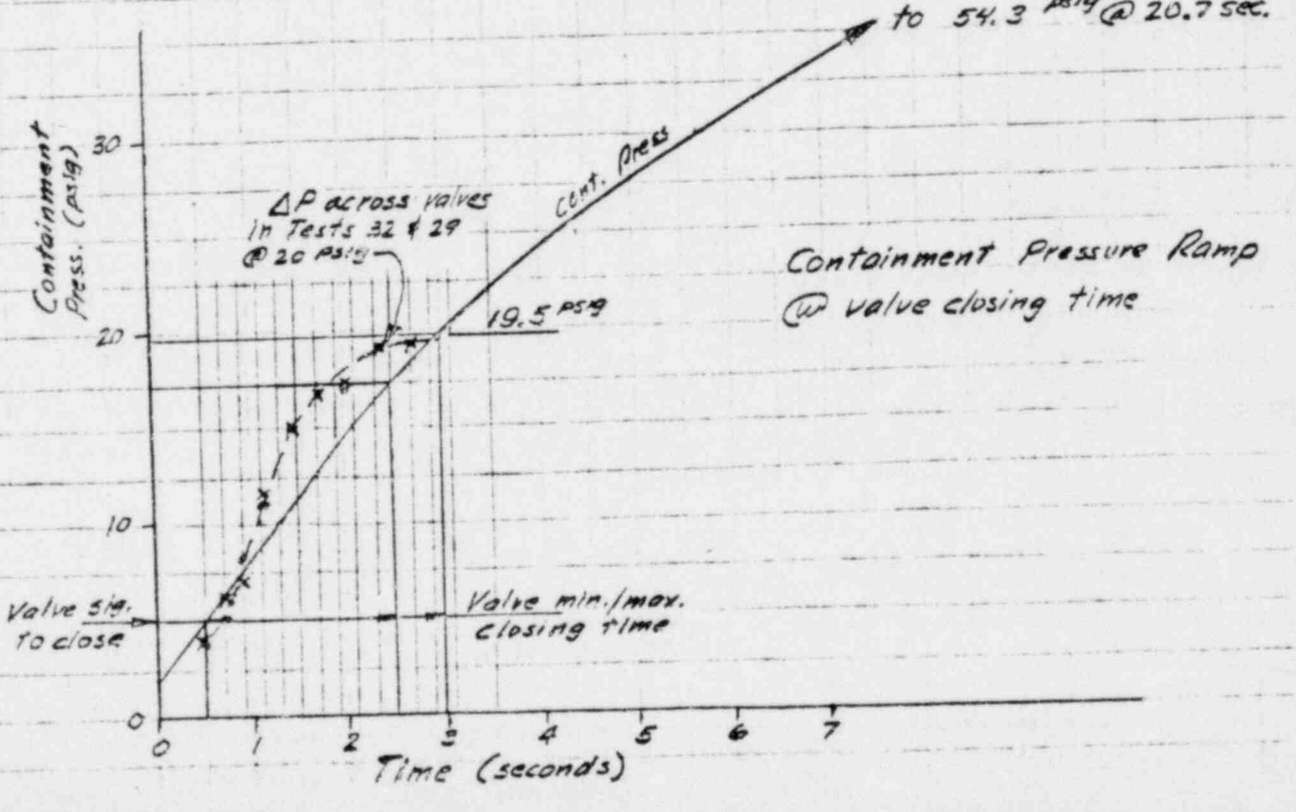
Operators: 8×30 Allis-Chalmers Standard Air Cyl.

@ 70 ^{psig} min. supply pressure

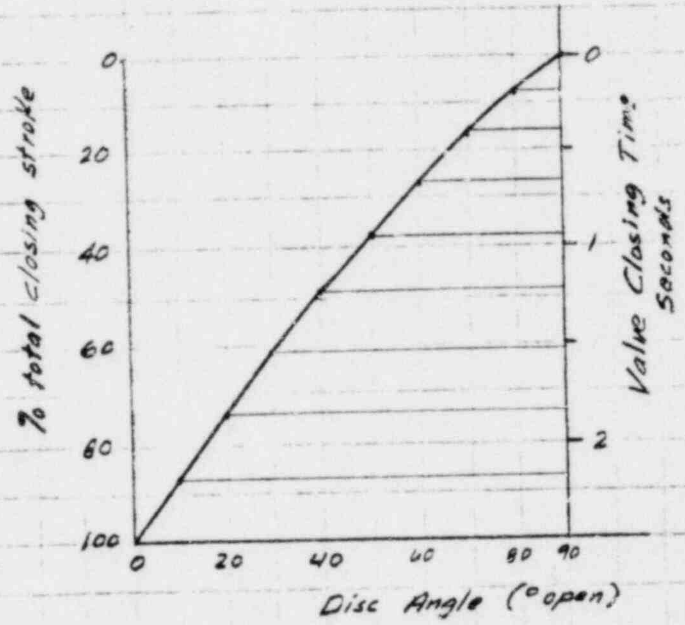
Valve Orientation: Per information from Yankee Atomic on 12/10/79, both supply and exhaust valves have the flat side of the disc toward containment. The supply valves are oriented as shown in Allis-Chalmers Test Report Fig. 10. The exhaust valves are oriented as shown in Fig. 9.

The thickness to diameter ratio for these valves is $t/d = 6.0/39.6 = .152$. Data from Test # 32 will be used for the supply valves and Test # 29 for the exhaust valves. t/d for tests # 32 & # 29 is .17.

Valve shaft diameter = 2.875" with a torque rating of 3400 $^{1\#}$ per AWWA C504-74.



CUSTOMER Maine Yankee	DATE 2/19/80	SHEET 2 OF 10
SUBJECT	PRELIM.	FINAL
DRAWING NUMBER	LITHO IN U.S.A.-A-C	CALCULATED BY Zeiders
	ENGINEERING CALCULATION SHEET	
	ALLIS-CHALMERS FORM 6715-1	P. SCHWARZ 3-5-80



Graph assumes constant velocity of piston rod.

Assume as worst case that valve takes 2.5 sec. to close:

Valve Closing Time	Disc Angle	Containment Pressure	* ΔP valve Test 32	* ΔP valve Test 29
0	90	5 psig	4	4
.2	80	6	5	6
.4	70	7.5	8	7
.65	60	9	11	11.5
.95	50	11	15	15
1.2	40	12.5	17	17
1.55	30	14.5	17	17.5
1.85	20	16.5	19	19
2.2	10	18.5	19	19
2.5	0	19.5		

* 20 psig upstream pressure test

CUSTOMER Maine Yankee	DATE 2/18/80		SHEET 3 OF 10
SUBJECT 42" Containment Isolation Valves		PRELIM.	FINAL
DRAWING NUMBER		CALCULATED BY Zeiders	
		P. SCHWARZ 3-5-80	

LITHO IN U.S.A.—A-C

ENGINEERING CALCULATION SHEET

ALLIS-CHALMERS FORM 6715-1

Test #32 @ 15 ^{PSIG}

Disc Angle	ΔP Valve	C_T	T_d	T_b	T_o	Operator Torq. @ 70 PSIG
90	2	-2.4	176	35	211	3541
80	3	0	0	53	53	4062
70	5	4.7	845	89	756	4550
60	6	3.9	841	106	735	5002
50	9	2.4	776	159	617	5410
40	11	.8	316	195	121	5771
30	12	0	0	212	212	6080
20	13	-1.6	748	230	978	6321
10	13	-2.0	934	230	1164	6475

Test #32 @ 20 ^{PSIG}

90	4	-11.8	1696	71	1767	3541
80	5	-5	899	89	988	4062
70	8	3.9	1121	142	979	4550
60	11	3.4	1344	195	1149	5002
50	15	1.1	593	266	327	5410
40	17	-.4	244	301	545	5771
30	17	-.6	367	301	668	6080
20	19	-.7	478	336	814	6321
10	19	-1.3	878	336	1214	6475

Test #32 @ 30 ^{PSIG}

90	3.5	-26.7	3359	62	3421	3541
80	6	-6.4	1380	106	1486	4062
70	11	1.6	593	195	398	4550
60	17	1.5	916	301	615	5002
50	24	.27	233	425	192	5410
40	26	-.5	467	460	927	5771
30	28	-.34	342	496	838	6080
20	28	-.69	694	496	1190	6321
10	28.5	-1.13	1157	504	1661	6475

CUSTOMER Maine Yankee

DATE 2/18/80

SHEET 4 OF 10

SUBJECT 42" Containment Isolation Valves

PRELIM. FINAL

DRAWING NUMBER

LITHO IN U.S.A.-A-C

ENGINEERING CALCULATION SHEET

ALLIS-CHALMERS FORM 6715-1

CALCULATED BY Zeiders

P. SCHWARZ 3-5-8

Test #32 @ 40 ^{psig}

Disc Angle	ΔP Valve	C_T	T_d	T_b	T_o	Operator Torq @ 70 ^{psig}
90	6	-21	4528	106	4634	3541
80	10	5.5	1977	177	1800	4062
70	19	.51	348	336	12	4550
60	28	.57	573	496	77	5002
50	33	-.195	231	584	816	5410
40	36	-.9	1164	637	1801	5771
30	38	-.59	806	673	1479	6080
20	38	-.85	1161	673	1834	6321
10	38.5	-1.0	1384	681	2065	6475

Test #32 @ 50 ^{psig}

90	10	-14.1	5067	177	5244	3541
80	11	-7.1	2807	195	3002	4062
70	22	0	0	389	389	4550
60	28	.6	604	496	108	5002
50	36	-.4	518	637	1155	5410
40	42	-.8	1208	743	1951	5771
30	44	-.7	1107	779	1886	6080
20	44	-.9	1423	779	2202	6321
10	46	-.8	1323	814	2137	6475

Test #32 @ 60 ^{psig}

90	15	-3.[illegible]	4528	266	4794	3541
80	18	-3.2	2070	319	2389	4062
70	23	0	0	407	407	4550
60	32	.8	920	566	354	5002
50	43	-.3	464	761	1225	5410
40	49	-.8	1409	867	2276	5771
30	52	-.6	1121	920	2041	6080
20	53	-.9	1714	938	2652	6321
10	55	-.9	1779	974	2753	6475

CUSTOMER Maine Yankee

DATE 2/18/80

SHEET 5 OF 10

SUBJECT 42" Containment Isolation Valves

PRELIM.

FINAL

DRAWING NUMBER

LITHO IN U.S.A.—A-C

ENGINEERING CALCULATION SHEET

ALLIS-CHALMERS

FORM 6715-1

CALCULATED BY Zaiders

P. SCHWARZ 3-5-80

Test # 29 @ 15 PSIG

Disc Angle	ΔP Valve	C_T	T_d	T_b	T_o	Operator Torq. @ 70 PSIG
90	2.5	.6	54	44	10	3541
80	2.5	5.1	458	44	414	4062
70	5	6.2	1114	88	1026	4550
60	7.5	3.3	890	133	757	5002
50	9.5	1.0	341	168	173	5410
40	11.5	-.6	248	204	452	5771
30	12	-1.1	474	212	686	6080
20	12	-1.3	561	212	773	6321
10	12.5	-1.8	809	221	1030	6475

Test # 29 @ 20 PSIG

90	4	-10	1438	71	1509	3541
80	6	-4	863	106	969	4062
70	7	9.2	2315	124	2191	4550
60	11.5	4.2	1736	204	1534	5002
50	15	1.1	593	266	327	5410
40	17	-2.4	1466	301	1767	5771
30	17.5	-2.7	1698	310	2008	6080
20	19	-3.4	2322	336	2658	6321
10	19	-4.2	2868	336	3204	6475

Test # 29 @ 30 PSIG

90	5	-9.44	1696	88	1784	3541
80	11	-2.3	909	195	1104	4062
70	13	0	0	230	230	4550
60	17.5	0	0	310	310	5002
50	22.5	-.558	451	398	849	5410
40	26	-1.2	1121	460	1581	5771
30	28	-1.12	1127	496	1623	6080
20	29	-1.08	1125	513	1638	6321
10	30	-1.57	1692	531	2223	6475

CUSTOMER Maine Yankee

DATE 2/18/80

SUBJECT 42" Containment Isolation Valves

SHEET 6 OF 10

DRAWING NUMBER

PRELIM.

FINAL

LITHO IN U.S.A.—A-C

ENGINEERING CALCULATION SHEET

ALLIS-CHALMERS

FORM 6715-1

CALCULATED BY Zeiders

P. SCHWARZ 3-5-80

Test #29 @ 40 psig

Disc Angle	ΔP Valve	C_T	Td	Tb	To	Operator Torq. @ 70 psig
90	10	-6.29	2260	177	2437	3541
80	10	-3.74	1344	177	1521	4062
70	15	-1.99	1073	266	1339	4550
60	22.5	-.66	534	398	932	5002
50	30	-1.05	1132	531	1663	5410
40	35	-1.36	1711	620	2331	5771
30	38	-.985	1345	673	2018	6080
20	39	-1.15	1612	690	2302	6321
10	39.5	-1.21	1718	699	2417	6475

Test #29 @ 50 psig

90	10	-7.1	2552	177	2729	3541
80	10	-4.11	1477	177	1654	4062
70	17.5	-1.28	805	310	1115	4550
60	27.5	-.65	642	487	1129	5002
50	35	-1.17	1472	620	2092	5410
40	42.5	-1.48	2261	752	3013	5771
30	44	-1.36	2151	779	2930	6080
20	46.5	-1.35	2256	823	3079	6321
10	47	-1.27	2145	832	2977	6475

Test #29 @ 60 psig

90	14.5	-4.34	2262	257	2519	3541
80	17	-2.42	1478	301	1779	4062
70	24.5	-1.28	1127	434	1561	4550
60	34	-.93	1136	602	1738	5002
50	42	-1.25	1887	743	2630	5410
40	50	-1.57	2821	885	3706	5771
30	54.5	-1.44	2821	965	3786	6080
20	55.5	-1.42	2832	982	3814	6321
10	57	-1.18	2417	1009	3426	6475

CUSTOMER Maine Yankee	**DATE** 2/20/80	**SHEET** 7 **OF** 10	
SUBJECT	**PRELIM.**		**FINAL**
DRAWING NUMBER	**ENGINEERING CALCULATION SHEET**	**CALCULATED BY** Zeiders P.R. SCHWARZ 3-5-80	

LITHO IN U.S.A.-A-C

ALLIS CHALMERS

FORM 6715-1

Valve seating torque at the closed position:

$T_{os} = T_s + T_b$ T_s = rubber seating torque

T_b = shaft bearing torque

$T_s = C_s D^2$ where C_s = coef. of seating = 180

D = valve dia. in feet

$$T_s = 180 \left(\frac{40}{12}\right)^2 = 2000^{1\#}$$

$T_b = 4.71 \times D^2 \times d \times f \times \Delta P$ where d = shaft dia. (inches)

f = coef. of friction

$$T_b = 4.71 \times \left(\frac{40}{12}\right)^2 \times 2.875 \times .12 \times \Delta P$$

$$= 18\,\Delta P$$

Shutoff Pressure psig	T_b	T_s	$T_b + T_s$
15	270	2000	$2270^{1\#}$
20	360	↓	2360
30	540		2540
40	720		2720
50	900		2900
60	1080		3080

Operator capability at the closed position @ 70^{psig} supply press. is $6491^{1\#}$.

CUSTOMER Maine Yankee

SUBJECT

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FINAL

LITHO IN U.S.A.

ENGINEERING CALCULATION SHEET

ALLIS-CHALMERS

FORM 6715-1

CALCULATED BY Zeiders

P.R. Schultz 3-5-8

Conclusions :

1. Using Tests #32 & 29, the pressure drop across the valve closely follows or exceeds containment ramp pressure. With the valves closing in 2.5 seconds (3 sec. total time from 0), the cylinder operators should close the valves with adequate margin from the full open position.

2. Supply valve (Ref. Test #32)
   Valve should close from wide open with containment pressures up to 30 psig provided that connecting piping contributes to pressure drop exposing the valve to ΔP shown in the test results.
   For containment pressures greater than 30 psig and up to 60 psig, the valve opening should be limited to 80°. With this opening limitation and ΔP across the valve as shown in the test results, the valve should close from 80° open.

3. Exhaust valve (Ref. Test #29)
   Valve should close from wide open with containment pressures up to 50 psig provided that connecting piping contributes to pressure drop exposing the valve to ΔP shown in the test results.
   For containment pressures greater than 50 psig, the valve shaft may be exposed to torques beyond the 3400 '# rating.

4. Only single valve closure is considered in this report. Simultaneous closure of valves in series would tend to reduce torque.

CUSTOMER Maine Yankee	DATE 2/22/80	SHEET 9 OF 10
SUBJECT	PRELIM.	FINAL
DRAWING NUMBER	LITHO IN U.S.A.—A-C **ENGINEERING CALCULATION SHEET** ALLIS-CHALMERS FORM 6715-1	CALCULATED BY Zeiders P. SCHWARZ 3-5-80

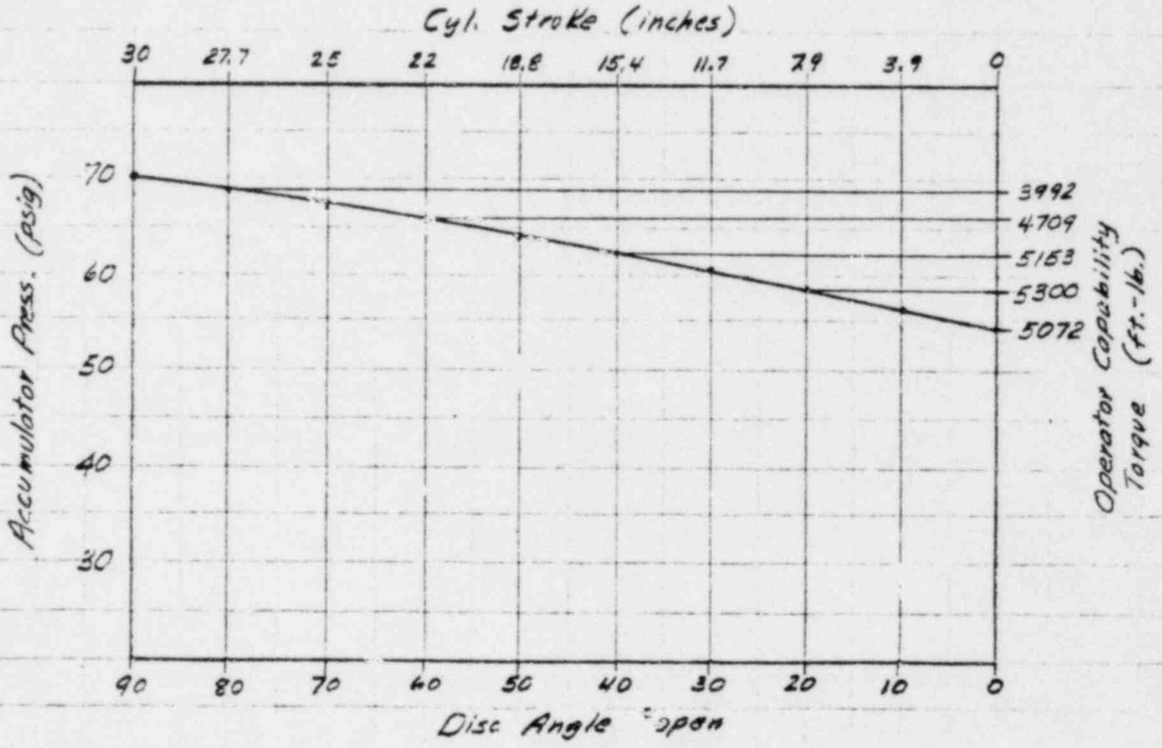
5 Accumulator size:

Volume = 30 gal. × 231 $\frac{in^3}{gal}$ = 6930 $in.^3$ = 4 ft.

Pressure = 70 psig

Cylinder size (full stroke disregarding rod)

$\frac{\pi}{4} \times 8^2 \times 30 = 1508\ in^3 = .873\ ft.^3$



Above graph shows valve closure using accumulator tank. Accumulator pressure decay does not change conclusions 2 & 3.

CUSTOMER Maine Yankee

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SHEET 10 OF 10

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LITHO IN U.S.A.—A-C

ENGINEERING CALCULATION SHEET

ALLIS-CHALMERS

FORM 6715-1

CALCULATED BY Zeiders

P. SCHWARZ 3-5-80

6. Valve shaft stresses:

During the valve closing stroke, the shaft is subjected to direct shear due to the pressure drop and to torsional shear due to the required closing torque.

$$S_{SD} = \frac{\frac{\pi}{4} \times D^2 \times \Delta P}{2 A_s} = \frac{\pi (39.6)^2 \Delta P}{8 (6.49)} = 94.89\ \Delta P$$

$$S_{ST} = \frac{T \times 12}{Z_p} = \frac{12}{4.67} \times T = 2.57\ T$$

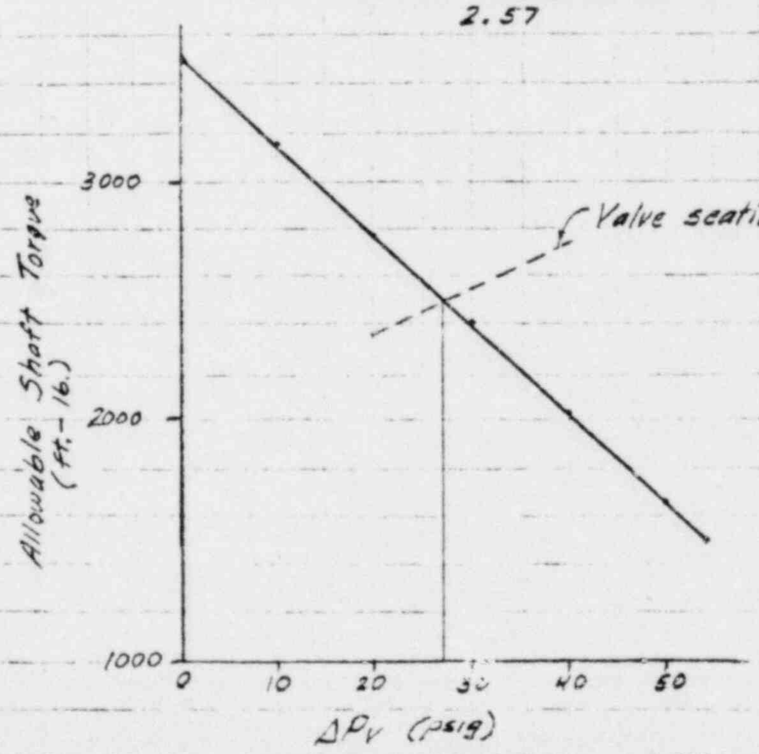
$$S_{SD} + S_{ST} = 94.89\ \Delta P + 2.57\ T$$

Assume $S_{SD} + S_{ST} = 9000^{psi}$ max. for type 304 stainless steel

If $\Delta P_{max.} = 54.3^{psig}$ then:

$$9000 = 94.89 (54.3) + 2.57\ T$$

$$T = \frac{9000 - 94.89 (54.3)}{2.57} = 1497^{'\#}$$



Valve should be closed before pressure reaches 27^{psig} to avoid combined shear stress $> 9000^{psi}$ on shaft.

Also note Test # 29 @ 20^{psig} shows combined stress higher than allowable at 10°. This is an instaneous peak and should not be a structural concern.

CUSTOMER Maine Yankee

DATE 3/3/80

SHEET 1 OF 1

SUBJECT 42" Containment Supply & Exhaust Valves

PRELIM. FINAL

DRAWING NUMBER

LITHO IN U.S.A.-A-C

ENGINEERING CALCULATION SHEET

ALLIS-CHALMERS FORM 6715-1

CALCULATED BY R. Zeiders

P. R. SCHWARZ 3-5-80

Addendum #1

Conclusions (continued)

7. Per our discussions with Yankee Atomics, we have been advised that the subject valves will be blocked 50° open. With this condition, closing time should be 1.5 sec. max. and pressures seen by the valve will decrease.

Without considering the above margin, the maximum anticipated combined shear stress on the shaft would be:

(from Test #29 @ 60 psig)

$$S_{SD} + S_{ST} = 94.89(54.3) + 3814(2.57)$$
$$= 5126 + 9802$$
$$= 14,928^{PSI}$$

The shear yield for the shaft material is ≈ 18,000 ^{PSI}. Realizing that the 14,928 ^{PSI} is a max. stress that occurs only at one point on the O.D. of the shaft, we feel that even in this worst case the valve would close and maintain structural integrity.

ENCLOSURE (3)

ALLIS-CHALMERS
BOX M-93 • YORK, PENNSYLVANIA 17405/717-848-1126

YORK PLANT
VALVE DIVISION

January 28, 1981

Yankee Atomic Electric Co.
1671 Worcester Road
Framingham, Mass 01701

ATTENTION: Bob Benson

Gentlemen,

In accordance with Yankee Atomic P.O. # 28191, Allis-Chalmers has submitted a static Seismic Analysis VER 0218 which verifies the ability of the equipment to withstand 2.9 g's applied simultaneously in three directions. Internal valve items, and operator mounting components have been analyzed to verify acceptable stress levels during a seismic event and the first mode natural frequency of the extended structure has been analyzed and found to be in excess of 33 hz, therefore the equipment is rigid.

In addition, Allis-Chalmers previously supplied performance calculations for the subject Containment Isolation Valve. These calculations evaluate the functional capability of the valve during a LOCA event. Based upon the criteria for Containment Pressure Ramp supplied to Allis-Chalmers by Yankee Atomic, our analysis shows that the subject valves will function and further that valve structural integrity will be maintained. Refer to our report and performance calculations sent to you in Feb. and March 1980.

Therefore the requirement for seismic qualification and functional qualification during LOCA have been seperately satisfied.

Sincerely,

G. Andzulis

GA/pmm
3/pmm/5259

ALLIS-CHALMERS CORPORATION

ENCLOSURE (B)

ALLIS-CHALMERS CORPORATION
VALVE DIVISION, YORK, PA

VER 0218
Page 1 of 28

Seismic Analysis for a 42" Containment Purge Valve for Maine Yankee Atomic Power

Customer Purchase Order # 28191
Allis-Chalmers Order # 43381

A-C Manufacturing No.: 76447
Tag # P.O. 28191

Prepared By: H. Carl Barnes

Checked By: Barbara H. Gaither

Approved By: 1/18/80

Pg 1A of 28
12/00

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Form Eng. .016-1

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ALLIS-CHALMERS CORP. YORK PLANT, YORK, PA.

BY H.C. BARNUM DATE 11-13-80 SUBJECT MAINE YANKEE SHEET NO 2 OF 28
CHKD. BY BHY DATE 11-17-80 42"-175 W.R. BUTTERFLY VALVE w/ JOB NO
8"x30" AIR CYLINDER

The calculations for the valve and the mounting of the operators are based on the general practice of the Valve Industry and AWWA Specification C504 for butterfly valves, or the requirements of ASME Section III and ANSI B16.34 where applicable. The valve disc, shafts, body, mounting bracket, pins, bolts and bracket welding have all been reviewed.

The design stresses are within the maximum allowable stresses given in ASME Code Section III. The valve, the mounted operator and bolting meet code requirements.

Calculations shown are for valves with 75 psi design pressure at a temperature of 285 °F.

Seismic factors:

G. 2.9 horizontal _____ one direction ✓ two orthogonal directions

G. 2.9 vertical

WORST CASE → The valve is oriented in a VERT. pipeline with the axis of the valve shaft HORIZ. and the operator ________ the valve.

Disc stresses and deflections are based upon weight, siesmic factors and pressure. Shaft bearing and end cover stress calculations used disc loads, shaft weights, pressure and/or operating torques.

Body features exceeded requirements. Weights and center of gravities for the mounting bracket and operator are used in the calculation for the bolting and for the critical section in the mounting bracket.

Reference is made to:

1. "Formulas for Stress and Strain" by R. J. Roark, Table 26 Case 8b. 4TH EDITION

2. "Design of Welded Structures" by Omer W. Blodgett, Sections 2.10 and 7.0

3. "Vibration Theory and Application" by William T. Thomas, Page 7.

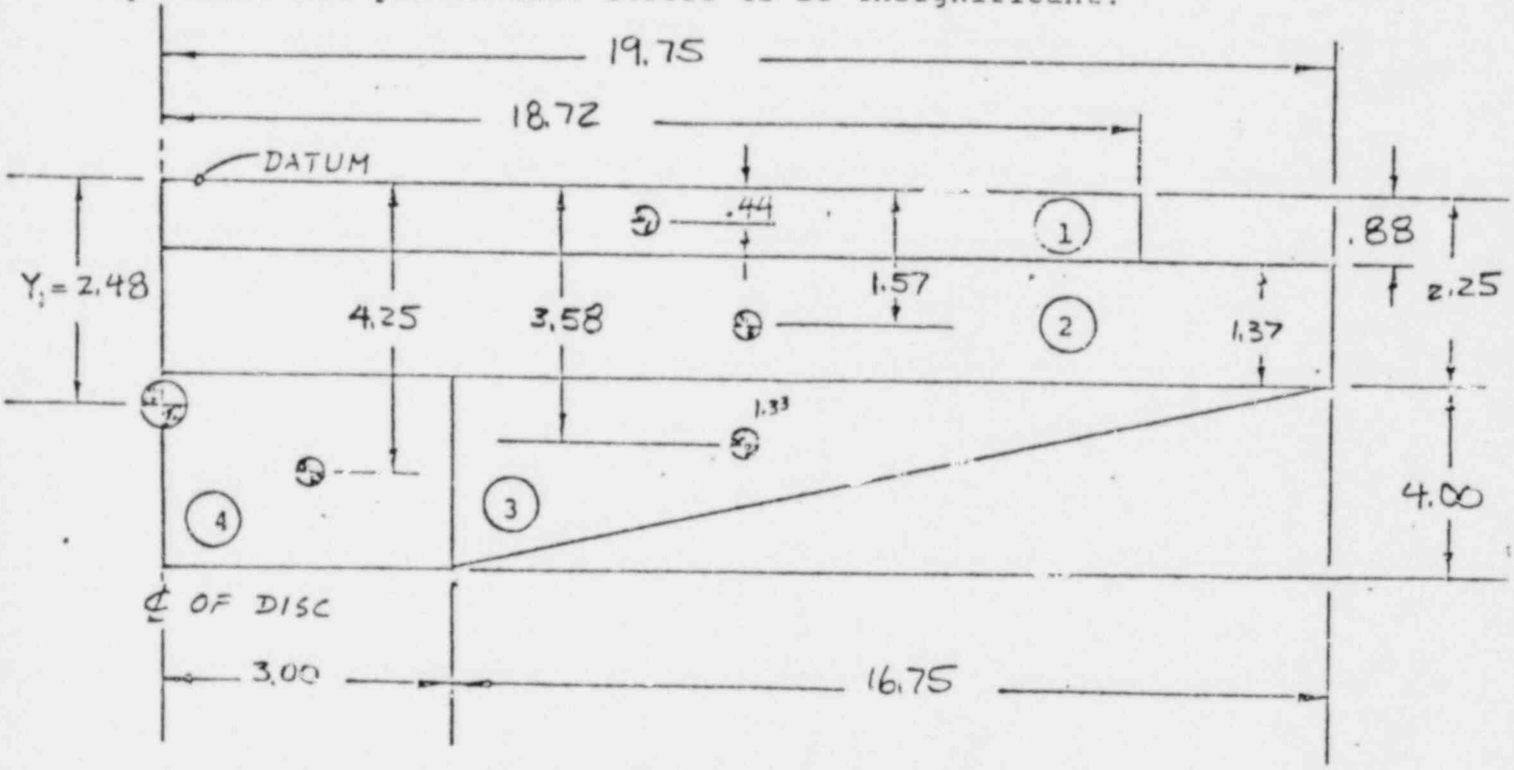
3/djc/4406

By H.C.B. Date 11-13-80
Chkd by ~~RHL~~ Date 11-17-80

Subject Seismic Calculation
42"-75 WR. BFV

Sheet No 3 of 28
Job No

1. DISC STRESS & DEFLECTION:- Disc is calculated as a simple beam with maximum bending stress being on a plane thru the center perpendicular to the shaft centerline. Actual stress strain tests indicate the weakest section being in the center of the disc. Triangle loading is used to simulate a round disc, since the load is increasing towards the center and results in a higher bending load than using a uniform load. Cantilever bending calculation (90° to loading shown) has not been shown, since past experience has proven this stress to be insignificant.



Section No.	A	Y	AY	I_o	d	Ad^2	I_D
1	16.47	.44	7.25	1.06	2.04	68.54	69.60
2	27.06	1.57	42.48	4.23	0.91	22.41	26.64
3	33.50	3.58	119.93	29.78	1.10	40.54	70.32
4	12.00	4.25	51.00	16.00	1.77	37.59	53.59
Total	89.03		220.66				220.15

$$2I_D = 440.3 \text{ IN}^4 = I_T$$

A = Area of section (sq. in)
Y = Distance from datum to centroid of section (in)
Y_1= Distance from datum to neutral axis of disc (in)
I_o= Moment of inertia of "A" section about its centroid

$$I_o \underset{\substack{1+2 \\ \text{AND } 4}}{\text{SECT}} = \frac{bh^3}{12} \qquad I_o \underset{3}{\text{SECT}} = \frac{bh^3}{36}$$

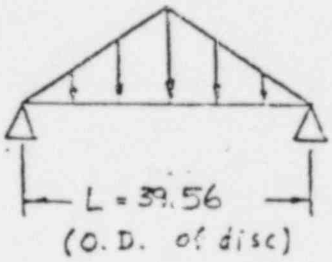
d = Distance from neutral axis of disc to centroid of section
I_D= Moment of inertia of section about neutral axis of disc

$$I_D = I_o + Ad^2$$

ALLIS-CHALMERS MFG. CO. YORK PLANT, YORK, PA.

By H.C.B. Date 11-13-80 Subject SEISMIC CALCULATIONS Sheet No. 4 of 28
Chkd. by Date 11-17-80 42"-75 WR BFV Job No.

1.2 PRESSURE LOAD ON DISC (W_1)



πr^2

W_1 = Pressure x disc area
W_1 = 75 PSI x 1229.14 IN²
W_1 = 92,186 LBS.

1.3 BENDING MOMENT (M_1)

$$① \; M_1 = \frac{W_1 L}{6} = \frac{(92,186)(39.56)}{6} = 607,813 \text{ IN·LBS.}$$

1.4 BENDING STRESS (S_1)

$$S_1 = \frac{M_1 x C}{I_T} = \frac{(607,813) \times (3.77)}{440.3} = 5204 \text{ PSI}$$

M_1 = bending moment
C = distance from neutral axis to outermost fiber 3.77
I_T = moment of inertia of disc = $2I_D$ 440.3

1.5 DEFLECTION FROM PRESSURE (Y_p)

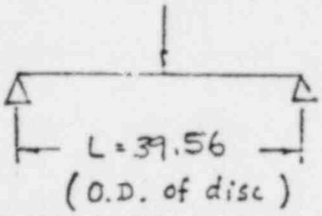
$$Y_p = \frac{W_1 L^3}{60 \; EI_T} = \frac{(92,186)(39.56)^3}{(60)(30 \times 10^6)(440.3)} = .0072 \text{ IN}$$

W_1 = pressure load
L = O.D. of disc
E = modulus of eleasiticy (in tension) 30×10^6 PSI
I_T = moment of inertia of disc

1 Ref from R. J. Roark, Table III, Case #17

By H.C.B. Date 11-13-80 Subject SEISMIC CALCULATIONS
Chkd. by [illegible] Date 11-17-80 42" 75 WR B.F.V.
Sheet No. 5 of 28
Job No.

1.5 CONCENTRATED LOAD ON DISC DUE TO SEISMIC SHOCK (W_2)



W_2 = Wt. of disc x seismic factor
$W_2 = 1060 \quad \times 2.9$
$W_2 = 3074$

1.7 BENDING MOMENT (M_2)

$$② \quad M_2 = \frac{W_2 L}{4} = \frac{(3074)(39.56)}{4} = 30,402$$

1.8 BENDING STRESS (S_2)

$$S_2 = \frac{M_2 C}{I_T} = \frac{(30,402)(3.77)}{440.3} = 260 \text{ PSI}$$

M_2 = bending moment
C = distance from neutral axis to outermost fiber 3.77
I_T = moment of inertia of disc = $2I_D$ 443.84

1.9 DEFLECTION FROM SEISMIC LOAD (Y_S)

$$Y_S = \frac{W_2 L^3}{48EI_T} = \frac{(3074)(39.56)^3}{48(30\times10^6)(440.3)} = .00030 \text{ IN}$$

TOTAL DISC STRESS (S_3)

$S_3 = S_1 + S_2 = 5204 + 260$

$S_3 = 5464$ PSI

TOTAL DISC DEFLECTION (Y_T)

$Y_T = Y_P + Y_S = .0072 + .0003$

$Y_T = .0075$

FACTOR OF SAFETY (F_S)

$$F_S = \frac{\text{Max. Allowable Stress}}{\text{Disc Stress}} = \frac{16,000}{5464} = 2.93$$

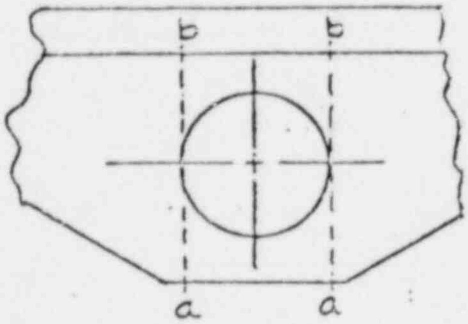
ATE STEEL
STM-A516 GR60 S_{YIELD} = 32,000 PSI, $S_{S\,YIELD}$ = 16,000 PSI

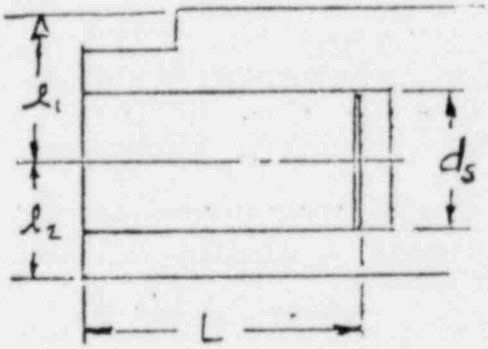
② Ref. from R. J. Roark. Table III, Case #11

BY H.C.B DATE 11-13-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 6 OF 28

CHKD. BY RHG DATE 11-17-80 42" - 75 WR BFV JOB NO.

1.10 Shear stress in disc due to shaft loading ($\sigma all =$)





d_s = 2.88
L = 5.50
l_1 = 3.25
l_2 = 3.00

Assume the shaft will shear out of the disc along plane "a-b". The plane of least resistance will be in the direction of the least distance l_1 or l_2.

For Pressure load W_1

$$\sigma sp = \frac{W_1}{4lL} = \frac{92,186}{4(3.00)(5.50)} = 1397 \text{ PSI}$$

For Seismic load W_2

$$\sigma sg = \frac{W_2}{4lL} = \frac{3074}{4(3.00)(5.50)} = 46.6 \text{ PSI}$$

Combined shear stress $\sigma s = \sigma sp + \sigma sg = 1397 + 46.6 = 1443.6$

1.11 Bearing Stress in shaft bore
(σall = 32,000 PSI)

Pressure loading W_1

$$\sigma bp = \frac{W_1}{2d_sL} = \frac{92,186}{2(2.88)(5.5)} = 2909.9 \text{ PSI}$$

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BY H.C.B. DATE 11-13-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 7 OF 28

CHKD. BY RHY DATE 11-17-80 42"-75 WR. BFV JOB NO.

1.11 Continued

Seismic Loading W_2

$$\sigma bg = \frac{W_2}{2d_sL} = \frac{3074}{2(2.88)(5.50)} = 97 \text{ PSI}$$

8" x 30 CYL.
70 PSI AIR
22.88" LEVER ARM

1.12 Bearing Stress on Disc Due to Disc Pins
(σall = 32,000 PSI)

T = Torque
T = 80,488 IN·LBS

$$\sigma b = \frac{T}{2rndpl}$$

where r = avg. distance from shaft ℄ to the ℄ of the pin bearing area.
n = number of pins loaded
dp = avg. pin diameter
l = total length of pin bearing on the disc.

$$\sigma b = \frac{80,488}{2(1.75)(2)(.803)(6.62)} : 2163 \text{ PSI}$$

σb =

2.0 PIN: Shear Stress
(σsall = 50,000 PSI)

TAPER PINS
.TR. SST. TYPE 416

T = Torque

$$\sigma s = \frac{2T}{\pi dp^2 n} = \frac{2(80,488)}{\pi(.803)^2 2} = 39,733 \text{ PSI}$$

3.0 SHAFT: Shear Stress

Pressure Load = W_1 from paragraph 1.11

Seismic Load = W_2 from paragraph 1.11
W_2+Wt. for Horizontal Shaft add weight of Disc.

Total Shear Stress

$$\sigma_s = \frac{W_1+W_2}{2\ A_s} + \frac{16T}{2\pi d^3} = \frac{95,260}{2(6.49)} + \frac{16(80,488)}{2(\pi)(2.88)^3}$$

σs = 15,919 PSI

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BY H.C.B. DATE 11-13-80 SUBJECT SEISMIC CALCULATION SHEET NO. 8 OF 28

CHKD. BY RHY DATE 11-17-80 42"-75 W.P. B.F.V. JOB NO.

4.0 Bearing Stress ON BODY SHAFT BORE

Using "W_1" & "W_2" from 3.0

$\sigma b = \frac{W_1 + W_2}{2A_b}$ Ab = Bearing Area = $\overset{2.88}{D}$ x $\overset{4.75}{\text{length}}$

$\sigma b = \frac{92{,}186 + 3074}{2(2.88 \times 4.75)} = 3482$ PSI

5.0 Tensile Stress in End cover bolts

$S_T = \frac{P}{AN} = \frac{4980}{(.640)\ 4} = 1945$ PSI

P = pressure x area + seismic factor x $\left(\overset{1060}{\text{wt. of disc}} + \overset{78}{\text{WT. OF SHAFTS}}\right)$ IF VERTICAL

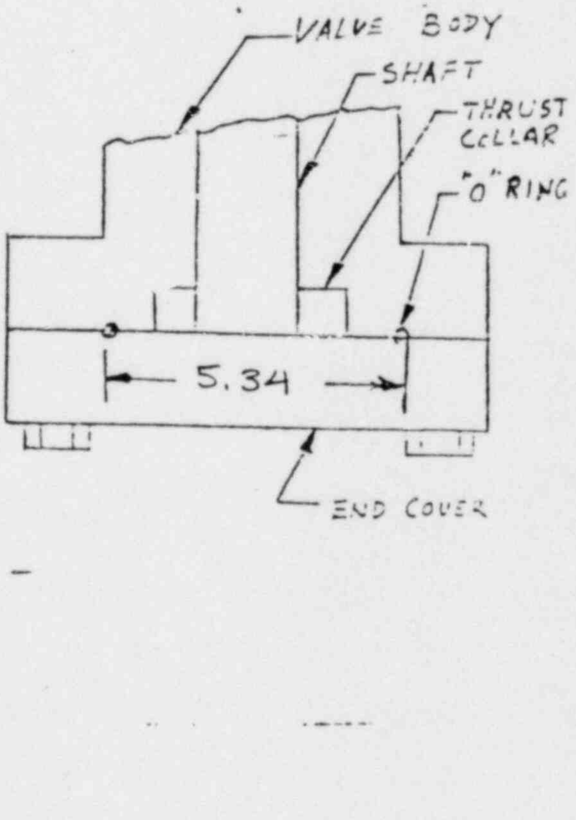
$P = 75 \times 22.4\,\text{in}^2 + 2.9$ =

$A = .7854\ (D - \frac{.9743}{n})^2$

$A = .7854\ (1 - \frac{.9743}{10})^2 = .640$

N = NUMBER OF BOLTS = 4

$n = 10$



6.0 End Cover Stresses

(Reference: Formulas for Stress & Strain
R. J. Roark Table 26 Case 8b)

σ_b = BENDING STRESS

$$\sigma_b = \frac{3W_p}{2\pi t^2}\left[\left\langle(1+V)\ \ln\ \frac{2b}{r_c'}\right\rangle + \beta_1\right]$$

$$= \frac{3(4980)}{2\pi(1.5)^2}\left[\left\langle(1.3)\ \ln\ \frac{2(9.5)}{2.67}\right\rangle + (-.238)\right] =$$

$$= 2444 \text{ PSI}$$

Where t = thickness of end cover = 1.50

b = shortest side of rectangular end cover = 9.50

β_1 = dimension factor = -.238

$W_p = \pi r_o^2 P$ for pressure loads + seismic factor x (wt of vane & shafts) = $\pi(2.67)^2 75 + 2.9(1138)$ = 4980 LBS
1060 +78 = 1138

V = Poisson ratio = .3

r_o' = the equivalent radius of contact for a load concentrated on a very small area and is given by $r_o' = \sqrt{1.6 r_o^2 + t^2} - 0.675t$ if $r_o < 0.5t$ and $r_o' = r_o$ if $r_o > 0.5t$

= 2.67 ($r_o > .5t$)

σ = BENDING STRESS

a = LONGEST SIDE OF RECTANGULAR END COVER, = 9.50 IN

BY H.C.B. DATE 11-13-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 10 OF 28
CHKD. BY DATE 11-17-80 42"-75 WR BFV JOB NO.

7.0 Body

7.1 Body Walls

Actual minimum wall thickness for the body flow passage is .75 in. Wall thickness for the flow passage, shaft penetration, bolt holes, etc. are all in accordance with ANSI B16.34-1977 Steel Valves.

2.1 Pipe loads

The moment of inertia, section modulus, and torsional resistance of the valve body is many times greater than the connecting pipe.

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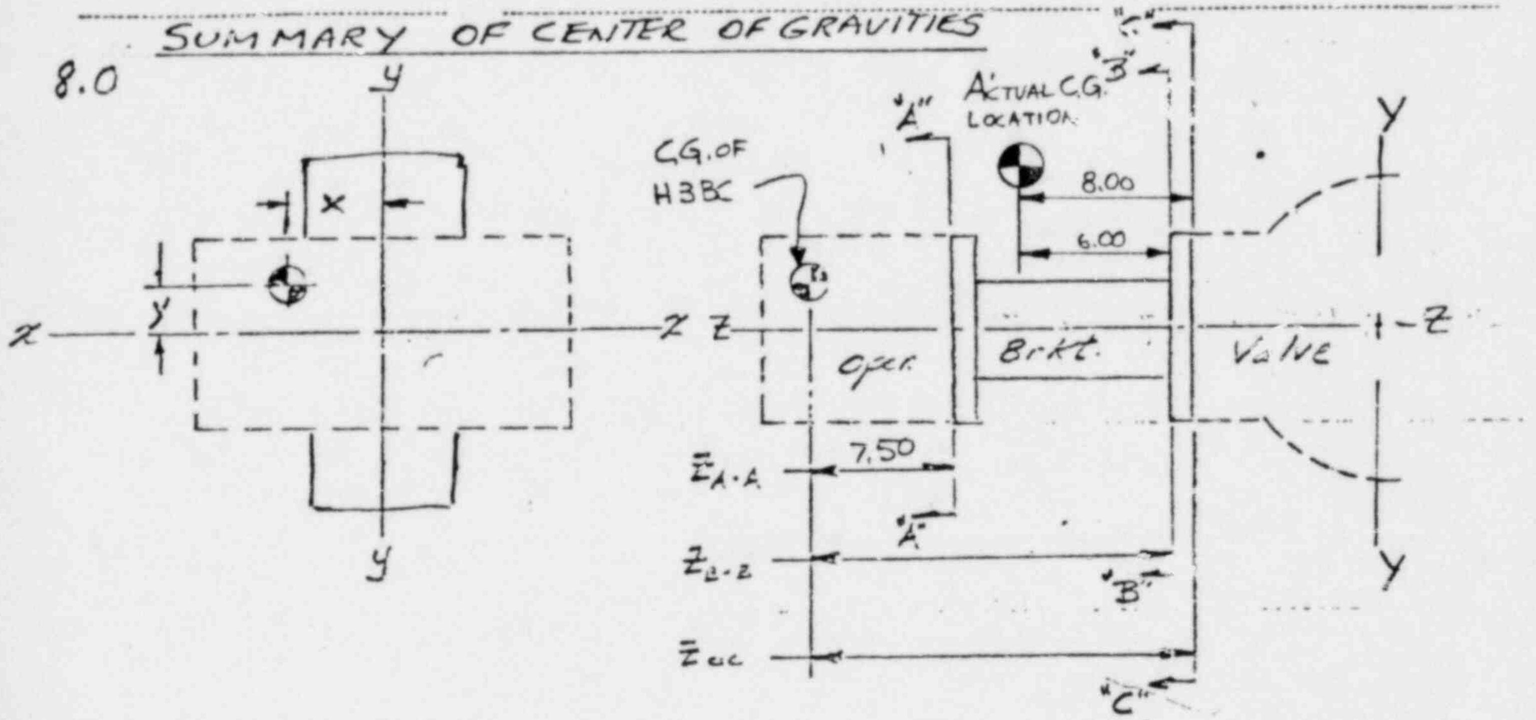
BY H.C.B DATE 11-14-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 11 OF 28

CHKD. BY AHY DATE 11-17-80 42"-75 WR. B.F.V. JOB NO.

SUMMARY OF CENTER OF GRAVITIES

8.0



	Weight	X	Y	$\bar{Z}_{A-A}$	$\bar{Z}_{B-B}$	$\bar{Z}_{C-C}$
Operator	360	-.78	-2.84	7.50		
Operator + Bracket to "B-B"	788	-9.44	17.19		6.00	
Operator + Complete Brkt	995	-9.44	24.38			8.00

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BY H.C.B. DATE 11-14-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 12 OF 28

CHKD. BY [illegible] DATE 11-17-80 42"- 75 WR BFV. JOB NO.

8.0 Extended Portions CONTINUED
(Reference drawing for weight and
combined C.G. of operator and mounting bracket)

8.1 Torsional Loads

8.1.1 Operating Torque

To = 80,488 IN LBS

$G_H = \sqrt{2.9^2 + 2.9^2} = 4.10$

8.1.2 Induced seismic torque

$G_V = 2.9$

8.1.2.1 Horizontal valve shaft

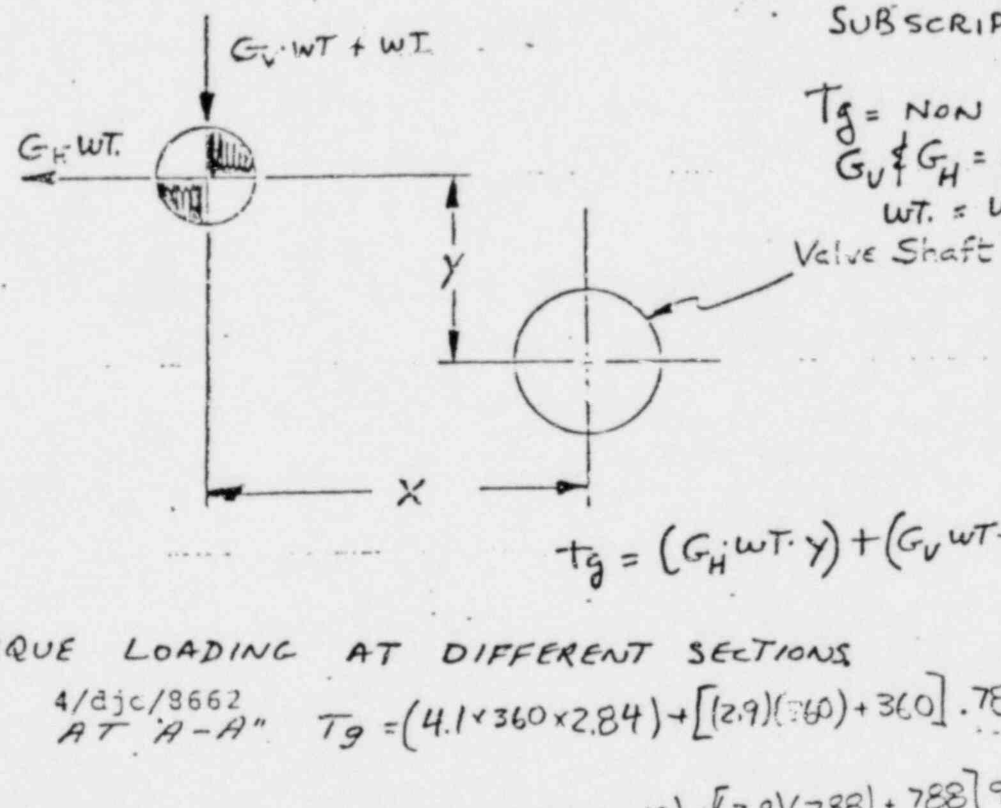
G = SEISMIC FACTORS

SUBSCRIPT V = VERTICAL
H = HORIZONTAL

Tg = NON OPERATING TORQUE

G_V & G_H = SEISMIC FACTORS

WT. = WEIGHT



$$T_g = (G_H WT \cdot Y) + (G_V WT + WT)X$$

TORQUE LOADING AT DIFFERENT SECTIONS

4/djc/8662

AT "A-A" $T_g = (4.1 \times 360 \times 2.84) + [(2.9)(360) + 360] .78 = 5287$ IN LBS

AT "B-B" $T_g = (4.1 \times 788 \times 17.19) + [(2.9)(788) + 788] 9.44 = 84,548$ IN LBS

AT "C-C" $T_g = (4.1 \times 995 \times 24.38) + [(2.9)(995) + 995] 9.44 = 136,090$ IN LBS

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BY H.C.B DATE 11-14-80 SUBJECT SEISMIC CALCULATIONS
CHKD. BY DATE 11-17-80 4" 75 WR BFV
SHEET NO. 13 OF 28
JOB NO.

8.1.2.2 Vertical Valve Shaft

$$T_g = GH \cdot WT\sqrt{X^2 + Y^2}$$

TORQUE LOADING AT DIFFERENT SECTIONS.

AT "A-A" $T_g = G_H \cdot WT\sqrt{X^2+Y^2} =$

AT "B-B" $T_g = G_H \cdot WT\sqrt{X^2+Y^2} =$

AT "C-C" $T_g = G_H \cdot WT\sqrt{X^2+Y^2} =$

NA

8.1.3 Total Torsional Load

T_o = OPERATING TORQUE.
To = 80,488 IN·LBS

T = To + Tg =

AT "A-A" T = 80,488 + 5287 = 85,776 IN·LBS

AT "B-B" T = 80,488 + 84,548 = 165,036 IN·LBS

AT "C-C" T = 80,488 + 136,090 = 216,578 IN·LBS.

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BY H.C.B. DATE 11-14-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 14 OF 28
CHKD. BY AHH DATE 11-17-80 42" - 75 WR B.F.V. JOB NO.

8.2 Toppling Moments and Bending Moment in Bracket

8.2.1 Horizontal Valve Shaft

At "A" - "A"

Mv (x-x) = [(wTGv) + (wt of operator)] (Z"A-A") =

[(360 2.9) + (360)] × 7.50 = 10,530 IN·LBS

Mh (y-y) = (Ghwt)(X) = (4.1)(360)(.78) = 1151 IN·LBS

Mh (x-x) = (GhwT)(Y) = (4.1)(360)(2.84) = 4192 IN·LBS

At "B-B"

Mv (x-x) = [(wTGv) + wt of oper. & brkt)] Z "B-B"

[(788)(2.9) + (788)] 6.00 = 18,439 IN·LBS.

Mh (y-y) = Gh (x)(wT) =

(4.1)(9.44) 788 = 30,499 IN·LBS.

Mh (x-x) = Gh (y)(wT) = (4.1)(17.19)(788) = 55,537 IN·LBS

At "C-C"

Mv (x-x) = [(wTGv) + (wt. of oper. & brkt)] Zc-c

[(995)(2.9) + (995)] 8.00 = 31,044 IN·LBS

Mh (y-y) = Gh (x)(wt) =

(4.1)(9.44)(995) = 38,510 IN·LBS

Mh (x-x) = Gh (y)(wT) =

(4.1)(24.38)(995) = 99,458 IN·LBS

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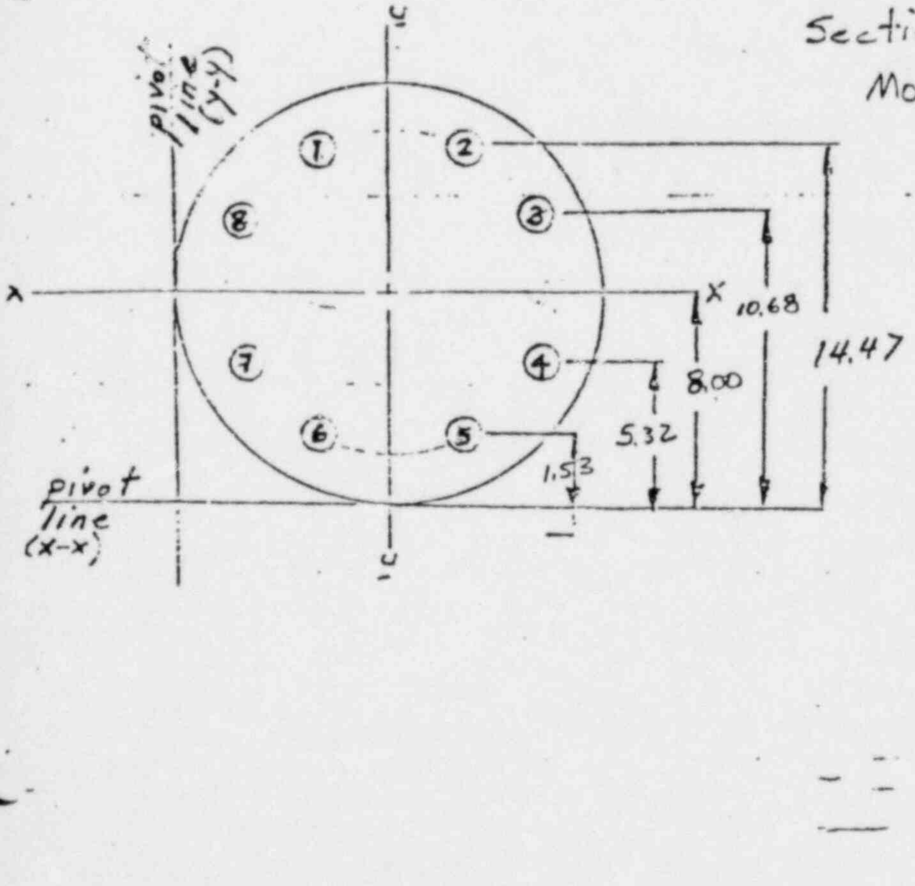
BY H.C.B DATE 11-14-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 15 OF 28

CHKD. BY AHG DATE 11-17-80 42"-75 WR B.F.V. JOB NO.

TYPICAL MOUNTING HOLE PATTERN



Section A-A

MOUNTING FOR H3BC

Eight hole mounting.

Two bolts per row

r = 7.00 in

Bolt size = $\frac{3}{4}$ - 10NC x 2.75LG

Bolt Mat'l = ASTM-A307 GR.A

YIELD STRESS = 60,000 PSI

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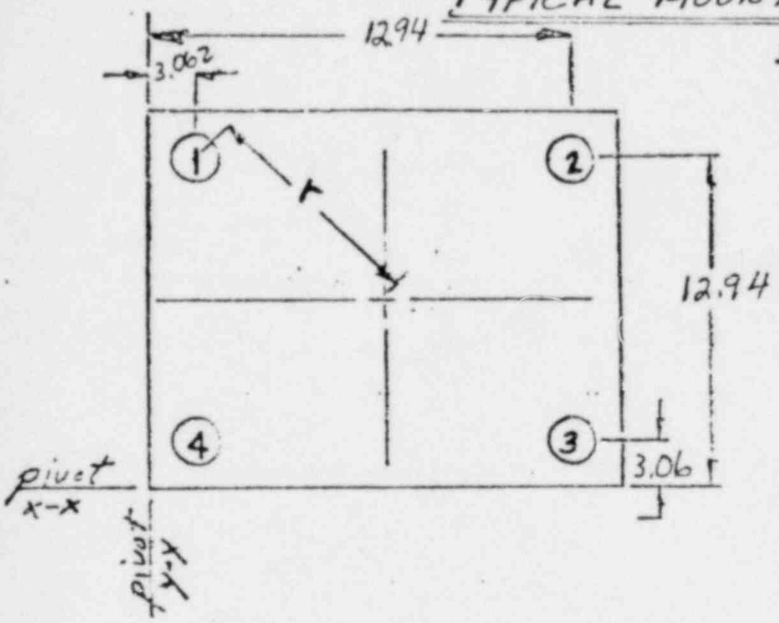
BY H.C.B. DATE 11-14-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 16 OF 28
CHKD. BY CHG DATE 11-17-80 42" - 75 W.R. BFV. JOB NO.

TYPICAL MOUNTING HOLE PATTERN

section C-C



Four hole rectangular mounting
Two bolts per Row

r = 7.00 in

Bolt size = 1.125-8N × 3.50 LG.

Bolt Mat'l = ASTM-A307 GR A

YIELD STRESS = 60,000 PSI

BY H.C.B DATE 11-14-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 17 OF 28
CHKD. BY AHY DATE 11-18-80 42"-75 W.R B.F.V. JOB NO.

$G_H = 4.1$
$G_V = 2.9$

8.3.1 BOLTS: Shear stress for section A-A

T = total torque in lb (8.1.3) = 85,776 IN·LBS
r = radius to bolts from centroid in.
N = number of bolts
D = bolt diameter in.
n = threads per inch
A = area of bolt in^2 = $.7854\left(D - \frac{1.3}{n}\right)^2$

$$A = .7854\left(.75 - \frac{1.3}{10}\right)^2 = .302\ IN^2$$

For Horizontal Shaft (8.1.2.1)
Seismic Load $L_S = \sqrt{(G_V\ wt.+wt.)^2+(G_H\ wt.)^2}$

$$L_S = \sqrt{(2.9 \times 360 + 360)^2 + (4.1 \times 360)^2} = 2037\ LBS$$

$T = T_o = 85,776$ IN·LBS

Shear stress per bolt = direct + torsional

$$\sigma_S = \frac{L_S}{NA} + \frac{T}{NrA} =$$

$$\frac{2037}{8(.302)} + \frac{85,776}{8(7.00)(.302)} = 5915\ PSI$$

8.3.1 Continued

For Vertical Shaft

Seismic Load $L_S = G_H$ wt. (8.1.2.2)

N.A.

Shear stress per bolt = direct + torsional

$$\sigma_S = \frac{L_S}{NA} + \frac{T}{NrA} =$$

BY H.C.B DATE 11-14-80 SUBJECT SEISMIC CALCULATIONS SHEET NO. 19 OF 28
CHKD. BY CHG DATE 11-18-80 42"-75 W.R. B.F.V. JOB NO.

8.3.2. BOLTS: TENSILE STRESS FOR SECTION

Tensile Load on Critical Bolt Section A-A

For each toppling moment the critical bolt is that bolt most distant from the point or line about which the adapter would pivot if free to move.

From the previous calculations moments about X-X and y-y are treated separately. Where there are two moments about either x-x or y-y they are combined in the analysis.

1. Load due to moments about x-x (The critical bolt is marked No. 1&2 on the sketch of the section)

$$L_{T1} = \frac{M_{(x-x)} d_k}{N_1 d_1^2 + N_2 d_2^2 + \ldots N_n d_n^2}$$

where $M_{(x-x)}$ = Sum of all moments about x-x = 14,722

d_k = Distance from pivot point to most distant bolt. = 14.47

$$L_{T1} = \frac{14,722\,(14.47)}{2(1.53^2 + 5.32^2 + 10.68^2 + 14.47^2)} = 300.8 \text{ LBS}$$

N_n = number of bolts per row = 2

d_n = Distance from pivot point to each row of bolts

2. Load due to moments about y-y on bolt No. 3 & 4

$$L_{T2} = \frac{M_{(y-y)} d_k}{N_1 d_1^2 + N_2 d_2^2 + N_n d_n^2}$$

$M(y-y) = 1151$

$d_k = 14.47$

$$L_{T2} = \frac{1151\,(14.47)}{2(1.53^2 + 5.32^2 + 10.68^2 + 14.47^2)} = 23.5 \text{ LBS}$$

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8.3.2. BOLTS: TENSILE STRESS FOR SECTION A-A

FOR VERTICAL SHAFT ONLY

3. Now take the critical bolt due to moments about x-x and find the load on that bolt due to y-y moments.

$$L_{T3} = \frac{M_{(y-y)}\, d}{N_1 d_1^2 + N_2 d_2^2 + \ldots N_n d_n^2}$$

where d = Distance from adapter pivot to critical bolt found in .1 = ____ in.

The total load on bolt No. ___ is

$L_{T4} = L_{T1} + L_{T3} =$ ____ = ________ lbs.

4. The load on bolt No. ___ due to x-x moments is

$$L_{T5} = \frac{M_{(x-x)}\, d}{N_1 d_1^2 + N_2 d_2^2 + \ldots N_n d_n^2} = \_\_\_\_\_\_ = \_\_\_\_\_\_ \text{ lbs}$$

The total load on bolt No. ___ is

$L_{T6} = L_{T2} + L_{T5} =$ ____ = ________ lbs.

5. Therefore the critical bolt is No. ___.

6. Tensile Stress on Critical Bolt

$$\sigma_T = \frac{L_T}{A} = \frac{300.8 + 23.5}{.334} = 971 \text{ psi}$$

$$\text{where } A = .7854 \left(d - \frac{.9743}{n}\right)^2$$

$$A = .7854 \left(.75 - \frac{.9743}{10}\right)^2$$

$$A = .334 \text{ in}^2$$

8.4. CRITICAL BOLT

$$\text{MAX. SHEAR} - \sigma_{S_M} = \sqrt{\sigma_S^2 + \frac{\sigma_T^2}{4}} = \sqrt{5915^2 + \frac{(971)^2}{4}} = 5935 \text{ PSI}$$

$$\text{MAX. TENSILE} = \sigma_{T_M} = \frac{\sigma_T}{2} + \sqrt{\sigma_S^2 + \frac{\sigma_T^2}{2}} = \frac{971}{2} + \sqrt{5915^2 + \frac{(971)^2}{2}}$$

$$= 6440 \text{ PSI}$$

5/pmm/700037

8.3.1 BOLTS: Shear stress for section C-C

BOLTS
1,125-8N

$G_H = 4.1$
$G_V = 2.9$

T = total torque in lb (8.1.3) = 216,578
r = radius to bolts from centroid in. = 7,0 IN
N = number of bolts 4
D = bolt diameter in.
n = threads per inch
A = area of bolt in^2 = $.7854 \left(D - \frac{1.3}{n}\right)^2$

$$A = 7854\left(1.125 - \frac{1.3}{8}\right)^2$$

$$A = .728 \text{ IN}^2$$

For Horizontal Shaft (8.1.2.1)
Seismic Load $L_S = \sqrt{(G_V \text{ wt.} + \text{wt.})^2 + (G_H \text{ wt.})^2}$

$$L_S = \sqrt{(2.9 \times 995 + 995)^2 + (4.1 \times 995)^2} = 5630 \text{ LBS}$$

$T = T_o = 216,578$ IN·LBS (SECTION C-C)

Shear stress per bolt = direct + torsional

$$\sigma_S = \frac{L_S}{NA} + \frac{T}{NrA} = \frac{5630}{4 \times .728} + \frac{216,578}{(4)(7)(.728)}$$

$$\sigma_S = 12,558 \text{ PSI}$$

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8.3.2. BOLTS: TENSILE STRESS FOR SECTION CC

Tensile Load on Critical Bolt Section

For each toppling moment the critical bolt is that bolt most distant from the point or line about which the adapter would pivot if free to move.

From the previous calculations moments about X-X and y-y are treated separately. Where there are two moments about either x-x or y-y they are combined in the analysis.

1. Load due to moments about x-x (The critical bolt is marked No. 1+2 on the sketch of the section)

$$L_{T1} = \frac{M_{(x-x)} d_k}{N_1 d_1^2 + N_2 d_2^2 + \ldots N_n d_n^2}$$

where $M_{(x-x)}$ = Sum of all moments about x-x = 130,502 IN-LBS

d_k = Distance from pivot point to most distant bolt. = 12.94 IN.

$$L_{T1} = \frac{(130{,}502)(12.94)}{2(3.06^2 + 12.94^2)} = 4776 \text{ LBS}$$

N_n = number of bolts per row

d_n = Distance from pivot point to each row of bolts

2. Load due to moments about y-y on bolt No. 2+3

$$L_{T2} = \frac{M_{(y-y)} d_k}{N_1 d_1^2 + N_2 d_2^2 + N_n d_n^2}$$

M(y-y) = 38,510 IN-LBS.

d_k = 12.94 IN

$$L_{T2} = \frac{38{,}510 \times 12.94}{2(3.06^2 + 12.94^2)} = 1409 \text{ LBS.}$$

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8.3.2. BOLTS: TENSILE STRESS FOR SECTION

FOR VERTICAL SHAFT ONLY

3. Now take the critical bolt due to moments about x-x and find the load on that bolt due to y-y moments.

$$L_{T3} = \frac{M_{(y-y)}\ d}{N_1 d_1^2 + N_2 d_2^2 + \ldots N_n d_n^2}$$

where d = Distance from adapter pivot to critical bolt found in .1 = in.

The total load on bolt No. ___ is

$L_{T4} = L_{T1} + L_{T3} =$ = ______________ lbs.

4. The load on bolt No. ___ due to x-x moments is

$$L_{T5} = \frac{M_{(x-x)}\ d}{N_1 d_1^2 + N_2 d_2^2 + \ldots N_n d_n^2} = \underline{\qquad\qquad} = \underline{\qquad\qquad} \text{ lbs}$$

The total load on bolt No. ___ is

$L_{T6} = L_{T2} + L_{T5} =$ = ______________ lbs.

5. Therefore the critical bolt is No. ___.

6. Tensile Stress on Critical Bolt

$$\sigma_T = \frac{L_T}{A} = \frac{4776 + 1409}{.790} = \underline{7829} \text{ psi}$$

$$\text{where } A = .7854\ (d - \frac{.9743}{n})^2$$

$$A = .7854\ (1.125 - \frac{.9743}{8})^2$$

$$A = \underline{.790} \text{ in}^2$$

8.4. CRITICAL BOLT

$$\text{MAX. SHEAR} - \sigma_{S_M} = \sqrt{\sigma_S^2 + \frac{\sigma_T^2}{4}} = \sqrt{12{,}558^2 + \frac{7829^2}{4}} = 13{,}154 \text{ PSI}$$

$$\text{MAX. TENSILE} = \sigma_{T_M} = \frac{\sigma_T}{2} + \sqrt{\sigma_S^2 + \frac{\sigma_T^2}{2}} = \frac{7829}{2} + \sqrt{12{,}558^2 + \frac{7829^2}{2}}$$

$$= 17{,}639 \text{ PSI}$$

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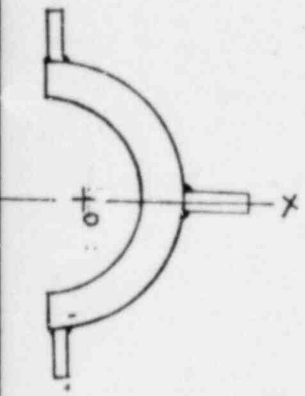
8.6 Bracket Section B-B

Minimum I is determined for the critical section



A = SECTION AREA
I_T = MOMENT OF INERTIA.

$$I_x = \cancel{\frac{bh^3}{12}} = I_{\text{HALF CIR}} + I_x \text{ of WELDED BARS.} = 532.8 \text{ in}^4$$

TRANSFER OF AXIS

$$I_T = 2I = 1066 \text{ in}^4$$

$$A = \cancel{2bh} = \text{TOTAL BRACKET AREA} = 13.96 \text{ in}^2$$

$$I_x = .1098(r_2^4 - r_1^4) - \frac{.283(r_2^2 r_1^2)(r_2 - r_1)}{r_2 + r_1}$$

HALF CIRCLE

8.6.1 Point farthest from the neutral axis

For horizontal shaft:

$$\sigma_{Ta} = \frac{WT \times Gv \times Z_{bb} \times C}{I_T} = \frac{788 \times 2.9 \times 6 \times 7}{1066} = 90.0$$

a) Tensile (flexural)

C = farthest distance from neutral axis = 7.0 IN
Wt. = weight of operator + bracket to "B-B" = 788

b) Tensile (direct) $\sigma_{Tb} = \frac{WT \times G_H}{A}$

A = section area

$$\sigma_{Tb} = \frac{788 \times 4.1}{13.96} = 231 \text{ PSI}$$

c) Stress (ToT) $\sigma_T = \sigma_{TA} + \sigma_{TB}$

$$\sigma_T = 90.0 + 231 = 321 \text{ PSI}$$

8.6.J. CONTINUED

For Vertical Shaft:

a) Tensile (flexural) $\sigma_{Ta} = \frac{W_T \times G_H \times Z_{bb} \times C}{I_T}$

N.A.

b) Tensile (direct) $\sigma_{tb} = \frac{W_T \times G_V \pm W_T}{A}$

c) Stress (Tot) $\sigma_T = \sigma_{TA} + \sigma_{TB}$

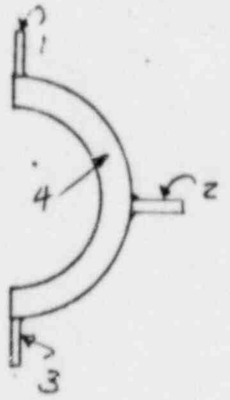
B.6.2 Point of highest torsional stress in bracket section

a) Shear Stress (Torsonal)

(Reference: Blodgett, Design of Welded Structures Section 2.10)

Torsional Resistance factor "R" for the section is the sum of the individual factors for each element of the section $R_1+R_2+R_3+\ldots R_n$.

For the section sketched the value of "R" is calculated below.



Element	d	b	b/d	β	R
1	2.67	.75		.320	4.57
2	.75	2.65		.320	.36
3	2.65	.75		.320	4.47
4					26.5

Total $R_T = 36$ $2R_T = 72$

$R = \frac{.0982(d_2^4 - d_1^4)}{2}$ SECTION 4

$R = \beta d^3 b$ SECTIONS 1, 2 + 3

The shear stress

$\sigma_{Sa} = \frac{Tt}{2R}$ for open sections where T = torque in inch lbs. @ B-B 165,036 LB

$= \frac{165036(.75)}{72} = 1719$

t = Thickness of the thickest plate = .75

$\sigma_{Sa} = \frac{165036(.75)}{72} = 1719$ psi

b) Shear stress (direct)

$G_H = 4.1$
$G_V = 2.9$

SHAFT DIA = 2.875
∴ A = 6.49 IN²

For horizontal shaft: $\sigma_{Sb} = \frac{WT. \times Gv}{A}$

$= \frac{995(2.9)}{6.49} = 445$ PSI

For vertical shaft: $\sigma_{Sb} = \frac{WT. \times GH}{A}$ N.A.

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8,6.2 CONTINUED

c) Stress (Tot' $\sigma_s = \sigma_{sa} + \sigma_{sb}$

$1719 + 445 = 2164$ PSI

d) Combine shear and tensile stresses

$$\sigma_{T\ max} = \frac{\sigma_t}{2} + \sqrt{\sigma_s^2 + \frac{\sigma_t^2}{4}}$$

$$\sigma_T = \frac{321}{2} + \sqrt{2164^2 + \frac{321^2}{4}} = 2330 \text{ PSI}$$

PL. STL
ASTM-A516 GR.60

$\sigma_{SHEAR\ YIELD} = 15,000$ PSI

$$\text{Factor of safety} = \frac{\sigma_t \text{ allowable}}{\sigma_{T\ max.}}$$

$$= \frac{15,000}{2330} = 6.44$$

Note: Plate weldments are designed to provide equivalent area, section modulus, and strength in the weld and base metal. The allowable stress has been limited to a level appropriate for a weldment.

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Natural Frequency at "B-B"

$$f = \frac{1}{2\pi}\sqrt{\frac{3gEI}{W(B_{B-B})^3}}$$ cycles per second

g = Acceleration of gravity = 32.2 $\frac{ft}{sec^2}$

E = Modulus of elasticity 30×10^6 PSI

I_T = Moment of inertia (Lesser of $I_{(x-x)}$ or $I_{(y-y)}$) $I_x = 1066$

W = Wt. of oper. & adapter beyond "B-B" = 788 LBS.

B_{B-B} = Distance from "B-B" to c.g. 6 IN.

$$f = \frac{1}{2\pi}\sqrt{\frac{3\times32.2(30\times10^6)(1066)}{788\,(6^3)}} = 678$$ cps

5/pmm/6380

RESULTANT OFF-SITE DOSES FROM A LOCA DURING ON-LINE CONTAINMENT PURGE AT MAINE YANKEE

BACKGROUND INFORMATION

Radiation detectors in the containment and in the primary vent stack sample gaseous and particulate activitity continuously. These detectors are designed to secure on-line purge on high gaseous activity in either the containment or the primary vent stack.

All on-line purge flow (23,000 CFM) is routed through a filter bank consisting of prefilters, HEPA filters, and a deep bed charcoal filter. The charcoal filter consists of a one-inch guard bed of activated charcoal followed by a four-inch bed of activated impregnated charcoal.

These features would serve to greatly reduce the off-site doses in the event of a LOCA occurring during a period of on-line purging; however, no credit has been taken in the analyses presented for either the on-line purge filter or automatic purge isolation on high radiation.

Several radiological analyses were performed covering the full spectrum of break sizes. The results are summarized below for each break size analyzed.

DESIGN BASIS LOCA WITH ON-LINE PURGE

The design basis accident with on-line purge was analyzed using conservative values for purge valve closure time and primary coolant concentrations. With the 42-inch diameter purge and supply valves open and with the purge fans running, the pressure rise following the DBA would reach the containment isolation setpoint of 5 psig in approximately 0.5 seconds. The purge valves are designed to close in approximately 1.5 seconds after receiving the containment isolation signal. A value of 2.1 seconds was used for the time period from the onset of the DBA to the time at which the purge valves are closed. The quantity of steam and air mixture which will be expelled from the containment under DBA conditions before the purge valves are secured was calculated to be 1600 LBM of air and 1040 LBM of water.

The source term considered in the analysis is the primary coolant activity at 1% failed fuel with a pre-existing Iodine spike of 60 uCi/gram dose equivalent iodine 131. The value of 60 uCi/gram dose equivalent iodine is the Technical Specification limit for periods not to exceed 48 hours. (The long-term operating limit for dose equivalent I-131 is 1 uCi/gram.)

Using these assumptions, the additional contribution to the thyroid and whole body gamma two-hour exclusion area boundary doses are 25 Rem and 0.01 Rem respectively. The DBA LOCA doses (without on-line purge considerations) are 178 Rem thyroid and 4.7 Rem whole body gamma. The total two-hour exclusion area boundary doses for the DBA occurring during a period of on-line purging would therefore be 203 Rem thryroid and 4.71 Rem whole body gamma.

SMALL BREAK LOCA WITH ON-LINE PURGE

An analysis was performed on the largest line-break which would not actuate an alarm for minimum letdown (60 gpm). It is assumed that after twenty minutes, the operator will manually secure the on-line purge valves, thus terminating the release. The primary coolant concentrations were set at their 1% failed fuel values, with a pre-existing Iodine spike of 60 uCi/gram dose equivalent I-131. The additional thyroid and whole body gamma doses for the small break, due to the on-line purge operation, are 3.53 Rem and 0.0015 Rem respectively.

INTEMEDIATE BREAK LOCAL WITH ON-LINE PURGE

The intermediate break analysis assumed the largest break which would not result in containment isolation on high pressure. In determining this break size, the in-containment pressure response was analyzed with the purge lines open and the purge fans running. Under these conditions, the largest break size which would not give a containment isolation on high containment pressure (5 psig) was calculated to the 0.1 ft^2. A break of this magnitude would depressurize the primary coolant system to the safety injection setpoint in approximately 25 seconds.

Because of the complexities that arise when considering the analysis of various break sizes, break locations and the associated impact of on-line purging, Maine Yankee proposes to install an additional containment purge isolation signal which would close the purge valves on a pre-set low pressurizer pressure signal. This signal, which will be set at approximately 2200 psig, would be anticipatory to a safety injection actuation signal and would reduce the effects of small and intermediate breaks.

Analysis of the 0.1 ft^2 break size shows that the primary coolant pressure would drop below a 2200 psig purge line isolation setpoint in less than 1 sec. A value of 2.6 seconds was used for the time interval from the 0.1 ft^2 break until the on-line purge valves are closed by the proposed low pressure signal. During the 2.6 second interval between the line break and the purge valve closure, approximately 879 gallons of primary coolant would be released into the containment atmosphere. This quantity represents about 1% of the active primary coolant. During the 2.6 second time interval, approximately 80 lbm of steam is expelled from the containment via the open purge valves. Using conservative assumptions for primary coolant concnetrations (1% failed fuel and a pre-existing Iodine spike of 60 uCi/ml), the additional LOCA dose from a 0.1 ft^2 break during on-line purge operations has been calculated to be 1.57 E-01 Rem thyroid and 7.18 E-05 Rem whole body gamma.

The radiological analysis of various LOCA conditions occurring during periods of on-line purging has shown that with the incorporation of the low pressure trip signal described, the limiting conditions for offsite doses is the large break DBA LOCA. Further, it has been shown that the offsite doses resulting from a large break LOCA with on-line purge would be well within the limits set forth in 10CFR100.