Attachment
Director of NRR
December 2, 1981

CONTAINMENT
ISOLATION/PURGE VALVE ANALYSIS FOR
18" 2FII BUTTERFLY VALVE
WITH 744A-1SR BETTIS-OPERATOR
AND 746A-2SR BETTIS-OPERATOR

Project Site: Prairie Island Nuclear Power Plant

Customer: Northern States Power Company

Engineer: Fluor Power Services, Inc.

Original Specification: TS-M-635A

Original Purchase Order: HIA-195 Change #2 & HIA-1195 Change #2

Original Pratt Job No.: N-6903-1&2, N-6904-1&2

Valve Tag Nos.: CV-31310 thru 31317

CV-31621 CV-31627

CV-31622 CV-31628

General Arrangement Drawing: E-1683 Rev. 4

Prepared by: T. J. Wrona

Date: 10-1-81

Reviewed by: R. L. Wilson

Date: 10-1-81

Certified by: J V Ballun

Date: 10-2-81

JOHN V. BALLUN
39860
REGISTERED
PROFESSIONAL
ENGINEER
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CONTENTS

I. Introduction

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

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

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

II. Considerations

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

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

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

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

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

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

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

7&8. Drawings or written description of valve orientation with respect to piping immediately upstream, as well as direction of valve closure, are furnished by customer/engineer. In lieu of input, worst case conditions have been applied to the analysis; namely, 90^{o} elbow (upstream) oriented 90^{o} out-of-plane with respect to valve shaft, and leading edge of disc closing toward outer wall of elbow. Effects of downstream piping on system back pressure have been covered in paragraph A.2. (above).

B. This analysis consists of a static analysis of the valve components indicating if the stress levels under combined seismic and LOCA conditions are less than 90% of yield strength of the materials used.

A valve operator evaluation is presented based on the operators ability to resist the reaction of LOCA-induced fluid dynamic torques.

C. Sealing integrity can be evaluated as follows:

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

Valves at outside ambient temperatures below $0^{o}F$, if not properly adjusted, may have leakage due to thermal contraction of the elastomer, however, during a LOCA, the valve internal temperature would be expected to be higher than ambient which tends to increase sealing capability after valve closure. The presence of debris or damage to the seats would obviously impair sealing.

III. Method of Analysis

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

A. Torque calculation

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

Bearing friction torque is calculated from the following equation:

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

where

P = pressure differential, psia

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

U = bearing coefficient of friction

d = shaft diameter, in.

Fluid dynamic torque is calculated from the following equations:

For subsonic flow

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

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

For sonic flow

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

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

Where

T_D = fluid dynamic torque, in-lbs.

F_{RE} = Reynold number factor

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

P_1 = upstream static pressure at flow condition, psia

P_2 = downstream static pressure at flow condition, psia

D = disc diameter, in.

C_{T1} = subsonic torque coefficient

C_{T2} = sonic torque coefficient

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

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

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

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

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

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

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

D-29253-2 TORQUE TABLE 3 8 / 31 / 81

JOB:NORTH.ST:PRAIR.ISL P2-VARIABLE SIZE ADJUSTED(REYNLDS NO.FNCTN!)
SAT.STEAM/AIR MIXTURE WITH 1.4 LBS STEAM PER 1-LBS AIR
SPEC.GR.= .738255 MOL.WT.= 21.3872 KAPA(ISENT.EXP.)= 1.19775 R= 72.1972
GAS CONSTANT-CALC.
SONIC SPEED(MOVING MIXTR.)= 1353.63 FEET/SEC AT 264 DEG.

CRIT.CASE INLET VELOCITY IS 1.42357 TIMES HIGHER AS AIR CRIT.CASE INLET V1-OF 5 INCH MODEL

MAX. TORQUE IS AT THE CRITICAL PRESS.RATIO(.585-(5 IN)MODEL OR APPX .655842 (17.25 IN)WITH STMIX.)FIRST SONIC(@ 72 DEG.V.A.)
ABSOL.MAX.TORQUE(FIRST SONIC)AT 72-68 DG.VLV.ANG.= 18259 IN-LBS @ 72 DEG.
MAX.TORQUE INCLUDES SIZE EFFECT(REYNOLDS NO.ETC)APPX. X 1.16257 FOR 17.25 INCH BASIC LINE I.D.
ALL PRESSURES USED:STATIC(TAP)PRESS.-ABSOLUTE:P2 INCL.RECOVERY PRESS.
(TORQUE)CALC'S VALIDITY:P1/P2)1.07:

VALVE TYPE:	18"-2FII:2.25/4.75	CLASS 150
DISC SIZE:	17.37 INCHES	SYMMETRICAL DISC
SHAFT DIA.:	2.25 INCHES	
BEARING TYPE:	BRONZE	
SEATING FACTOR:	25	
INLET PRESS.VAR.MAX.:	50.2 PSIA	
OUTLET PRESSURE(P6):	26.85 PSIA (72 DEG. ACTUAL PRESS.ONLY(VAR.))	
MAX.ANG.FLOW RATE:	72023.2 CFM: 117867. SCFM: 6479.49 LB/MIN	
CRIT.SONIC FLOW-90DG:	6942.42 LB/MIN AT 30.0498 INLET PSIA	
VALVE INLET DENSITY:	8.99640E-02 LB/FT^3-MIN. .134812 LB/FT^3-MAX.	
FULL OPEN DELTA P:	9.79071 PSI	

SYSTEM CONDITIONS:
PIPE IN-PIPE-OUT -AND- AIR/STEAM MIXTURE SERVICE @ 264 DEG.F
MINIMUM 0.75 DIAM. PIPE DOWNSTREAM FROM CENT.LINE SHAFT.

P1 ABS. PRESSURE(ADJ.)FOLLOWS TIME/PRESS.TRANSIENT CURVE.
ABSOLUTE MAX.TORQUE IS DEPENDENT ON DELAY TIME AND 3.43 TO 2.15-TH POWER OF (P1/P2)IN WORST RANGE X LINEAR CONSTANT X DWNSTR.PRESS. P6-ABS.(75-60DEG.) IN SUBSONIC RANGE LIMITS-ONLY;SEE FORMULATIONS.-PER TESTS H.PRATT
THIS TQ. AT 72 DEG.SYMM. DISC (68=OFFSET SHAFT)CT=T/D^3/P2(ABS)

--5 IN.MODEL EQUIV.VALUES------ACTUAL SIZE VALUES-----

ANGLE APPRX.	P1 PSIA	P2 PSIA	DELP PSI	PRESS. RATIO	FLOW (SCFM)	FLOW (LB/MIN)	TD ----INCHLBS	TB+TH ----	TD-TB-TH	TIME(LOCA) SEC.
90	33.5	23.71	9.79	0.708	117867	6479	0	1058	-1058	2.25
85	36.1	24.97	11.16	0.691	126251	6940	4937	1095	3842	2.47
80	38.2	25.49	12.70	0.667	128359	7056	7285	1228	6056	2.68
75	40.0	25.16	14.82	0.629	127989	7035	11933	1502	10430	2.89
72	40.9	23.97	16.97	0.585	121436	6675	18302	1877	16425	2.98
70	41.5	23.56	17.97	0.567	115910	6371	17814	1848	15965	3.05
65	42.8	21.71	21.11	0.507	104842	5763	13274	1581	11692	3.21
60	43.9	19.98	23.87	0.456	90047	4950	13462	1592	11869	3.33
55	44.6	18.30	26.29	0.410	75855	4170	11990	1779	10211	3.42
50	45.0	17.22	27.83	0.382	62025	3409	8669	1952	6717	3.48
45	45.2	16.42	28.78	0.363	61256	3367	7641	2099	5541	3.50
40	45.3	15.90	29.39	0.351	42190	2319	5408	2238	3170	3.52
35	45.6	15.37	30.20	0.337	32155	1767	3879	2374	1504	3.58
30	46.0	15.06	30.96	0.327	23931	1315	1985	2506	-520	3.67
25	46.6	14.88	31.72	0.319	16764	921	1181	2676	-1494	3.79
20	47.3	14.78	32.52	0.313	10769	592	798	2913	-2115	3.95
15	48.1	14.72	33.36	0.306	6115	336	351	3183	-2831	4.12
10	48.9	14.71	34.16	0.301	2923	160	296	3288	-2992	4.32
5	49.6	14.70	34.92	0.296	941	51	192	3373	-3181	4.53
0	50.2	14.70	35.50	0.293	0	0	10605	3165	7439	4.75

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 15771 IN-LBS @ 0 DEG.
MAX.DYN. + BEARING + HUB SEAL TORQUE (M/M)= 19502 IN-LBS @ 70 DEG.

B. Valve Stress Analysis

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

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

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

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

where

S_{max} = maximum combined stress, psi

T_1 = direct tensile stress, psi

T_2 = tensile stress due to bending, psi

S_1 = direct shear stress, psi

S_2 = shear stress due to torsion, psi

The calculated maximum valve torque resulting from LOCA conditions is used in the seismic stress analysis, attachment #2, along with "G" loads per design specification. The calculated stress values are compared to code allowables if possible, or LOCA allowables of 90% of the yield strength of the material used.

C. Operator Analysis

Model: Bettis 744A-1SR

Rating: 152,400 in-lbs at full open and closed positions only

101,600 in-lbs at intermediate positions

Model: Bettis 746A-2SR

Rating: 143,200 in-lbs at full open and closed positions only

94,500 in-lbs at intermediate positions

Max. valve torque: 18,302 in-lbs

The maximum torque generated during a LOCA induces reactive forces in the load carrying components of the actuator.

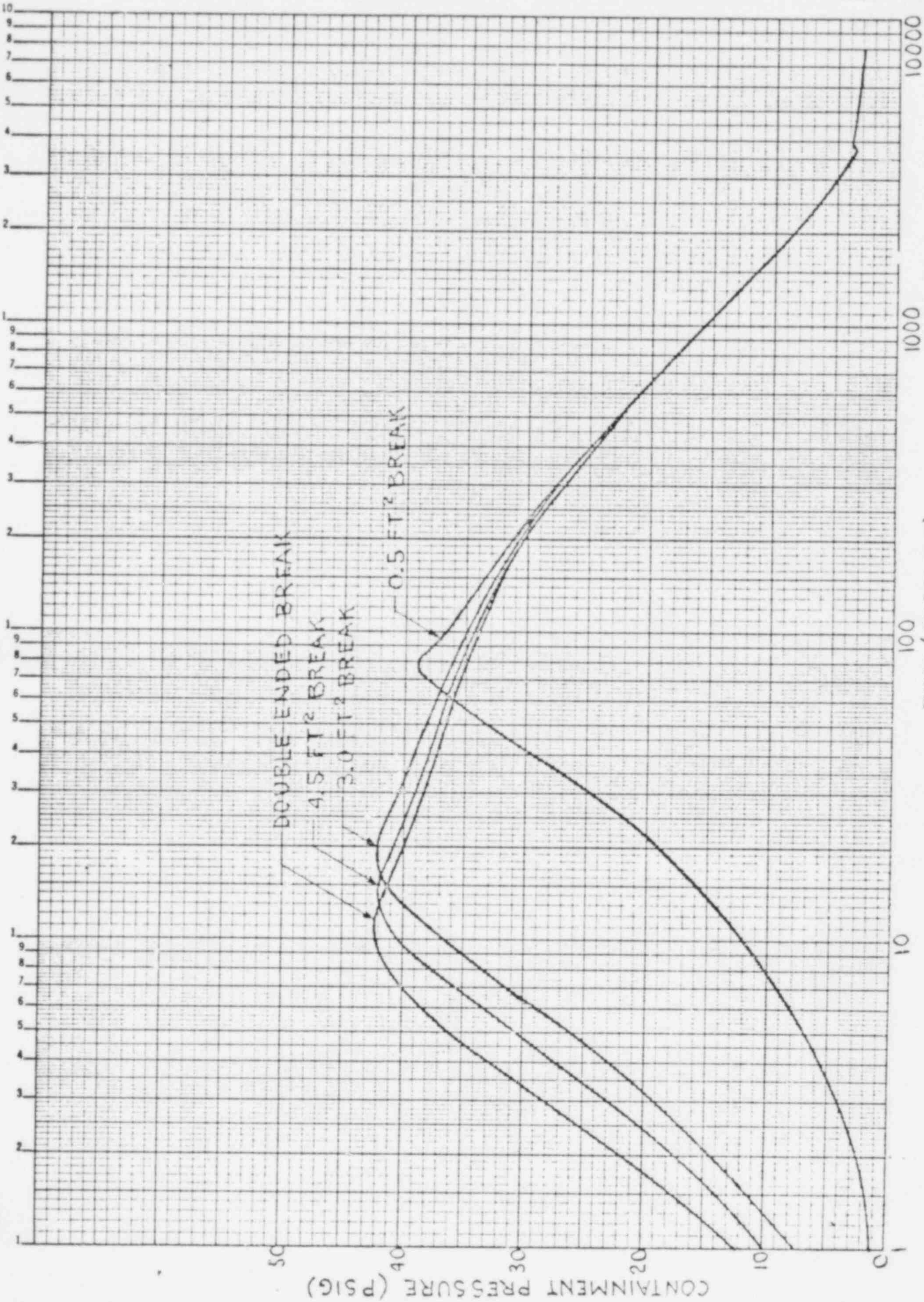
Since the LOCA induced torque derived in this analysis is less than the maximum absorbtion rating of the operator, it is concluded that the Bettis models furnished are structurally suitable to withstand combined LOCA and seismic loads.

IV. Conclusion

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

ATTACHMENT 1A

PRESSURE vs. TIME GRAPHS



CONTAINMENT PRESSURE TRANSIENT

FIGURE 14.[illegible].3

ATTACHMENT 1B

PRATT LETTER REGARDING
ADDITIONAL INFORMATION

PRATT

HENRY PRATT COMPANY

creative engineering for fluid systems

401 SOUTH HIGHLAND AVENUE · AURORA, ILLINOIS 60507

December 3, 1980

Northern States Power Co.
R.R. #2
Welch, MN 55089

Attention: Mr. A.D. Smith
Project Engineer

Subject: Prairie Island Nuclear Generating Plant
36" and 18" Purge Valve Analysis
MQ-05174

Dear Mr. Smith:

Recent findings in the general analysis of purge valves subjected to LOCA conditions have necessitated a request for additional technical data from the customer/engineer.

Delay time, system back pressure and valve orientation have a significant impact upon maximum torque and resultant stresses in the valve assembly. To properly complete the purge valve analysis referenced above, the following information is required:

1. The combined resistance coefficient for all ventilation system components downstream of the valve (one for each valve size) <u>or</u>

   A graph of back pressure vs. LOCA time at a distance 10-12 diameters downstream of the valve. Consider also the capacity of the piping, filter and duct work to resist increases in back pressure.

2. Maximum and minimum delay times from LOCA to <u>initiation</u> of valve rotation.

3. Drawings or written description of valve orientation with respect to elbow immediately upstream of valve (within 6 diameters), as well as direction of valve closure (clockwise or counterclockwise) as viewed from operator end.

In the absence of the above information, the following assumptions will apply to the purge valve analysis:

1. Back pressure of 19.7 psia throughout valve closing cycle. Higher back pressure increases maximum dynamic torque and valve stresses.

2. Delay time from LOCA to initiation of valve rotation shall be chosen to permit initial sonic flow condition and critical valve disc angle to coincide, resulting in maximum possible dynamic torque.

3. 90° elbow immediately upstream, oriented 90° out-of-plane with respect to valve shaft, with leading edge of disc closing away from outside radius of elbow. Such orientation and closure will increase torque values by 20% or more.

Your prompt response within 30 days would be appreciated.

Very truly yours,

HENRY PRATT COMPANY

T. J. Wrona

T.J. Wrona, Manager
Contract and Proposal Engineering

/sw

CC: R.D. Nelson

ATTACHMENT 1C

CUSTOMER/ENGINEER RESPONSE
TO REQUEST FOR INFORMATION

NSP

NORTHERN STATES POWER COMPANY

PRAIRIE ISLAND NUCLEAR GENERATING PLANT

RR #2

Welch, MN 55089

December 26, 1980

Henry Pratt Company
401 South Highway Ave.
Aurora, ILL 60507

Attention: Mr. T.J. Wrona

Subject: Your letter of December 3, 1980 requesting additional technical data for Purge Valve Analysis MQ-05174.

Dear Mr. Wrona:

1. Neither the combined resistance coefficient nor a graph of back pressure vs LOCA time at a distance 10-12 diameters downstream of the valve is available. Also, we do not consider a back pressure of 19.7 PSIA a conservative assumption.

2. Since delay times from LOCA to initiation of valve rotation is an unknown we accept your sonic flow condition assumption.

3. Table 1 attached is a summary of valve orientation.

   If we can be of further assistance, please call.

Very truly yours,

F.P. Tierney, Jr.
Plant Manager

By Gary Miller
Gary Miller

FPT/GM/jmc
Attachments
cc: subj. file - ventilation

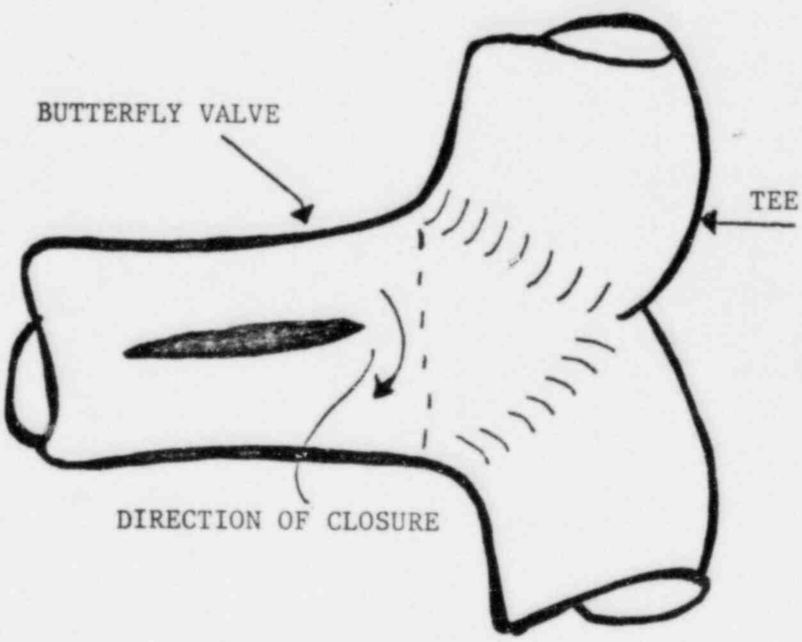
TABLE 1

Butterfly Valve Tag No.	VLV Closure Direction	Upstream Elbow Orientation
CV-31310	Clockwise	Note 2
CV-31311	Clockwise	Note 1
CV-31312	"	Note 2
CV-31313	"	Note 1
CV-31314	"	Note 2
CV-31315	"	Note 1
CV-31316	"	Note 2
CV-31317	"	Note 1
CV-31569	"	Note 2
CV-31570	"	Note 3
CV-31574	"	Note 2
CV-31575	"	Note 3
CV-31633	"	Note 2
CV-31634	"	Note 1
CV-31635	"	Note 2
CV-31636	"	Note 1

Notes

1. Inlet of butterfly valve opens directly into containment with no ductwork including elbows upstream of valve.
2. There are two Pratt Butterfly valves in series separated by a 21-30 inch straight length of pipe. This is the second valve.
3. There is a tee directly upstream of this valve. See Sketch 1 for valve Orientation with respect to the tee.

FIGURE 1



FLUOR POWER SERVICES, INC.

200 WEST MONROE STREET
CHICAGO ILLINOIS 60606
TELEPHONE (312) 368-3500

April 29, 1981

Mr. T. J. Wrona, Manager ✓
Contract and Proposal Engineering
Henry Pratt Company
401 S. Highland Avenue
Aurora, Illinois 60507

FX - 110

File: 000 268.1
1.2

Ref: None

Subject: Prairie Island Nuclear Generating Plant
217450-268
36" and 18" Purge Valve Analysis

Dear Mr. Wrona:

The enclosed table contains the calculated post LOCA pressures at several points in the containment purge supply and exhaust ducts.

Please call me if you have any questions.

Very truly yours,

D. F. Wheeler
Principal Mechanical Engineer

DFW/dm
encls.

cc: Mr. A. D. Smith, Northern States Power Company

SUPPLY SYSTEMS

LOCATION	UNIT 1 PRESSURE, PSIG 3 SECONDS	UNIT 1 PRESSURE, PSIG PEAK	UNIT 2 PRESSURE, PSIG 3 SECONDS	UNIT 2 PRESSURE, PSIG PEAK
Containment	27.8	42.6	27.8	42.6
Damper	25.4	38.6	25.2	38.3
10-12 Diameters *	22.3	33.4	21.5	32.5
Supply Fan	17.5	25.8	19.8	29.4

EXHAUST SYSTEMS

LOCATION	Unit 1 PRESSURE, PSIG 3 SECONDS	Unit 1 PRESSURE, PSIG PEAK	Unit 2 PRESSURE, PSIG 3 SECONDS	Unit 2 PRESSURE, PSIG PEAK
Containment	27.8	42.6	27.8	42.6
Damper	21.2	31.5	21.2	31.5
10-12 Diameters*	18.0	26.1	18.6	27.2
Stack Inlet	2.1	3.0	2.2	2.8

* 10-12 Diameters downstream of isolation valves.

ATTACHMENT 2

Nuclear Purge Valve

Stress Analysis

FOREWORD

This report presents applicable design criteria and analysis method for 2FII and MKII line of butterfly valves. The basic difficulty in the development of the applicable criteria is the compact shape of the valve body. This body-shape is not easily adaptable to the "Body-Shape Rule of ASME Section III, Boiler and Pressure Vessel Code".

The analysis methods and allowable stress intensities used are in accordance with the requirement of Nuclear Class I Valves. For the Valve Components not covered by the Code, analysis method presented are in accordance with the best engineering practice and judgement.

Wherever applicable, valve components are also analyzed for 5g level of Seismic loadings in three orthogonal directions.

APPLICABLE CRITERIA & ANALYSIS METHODS

Scope:

The procedures provided in this document are consistant with the requirement of article NB-3500, ASME Section III, Boiler and Pressure Vessel Code.

Abstract:

For the purpose of analysis the valve is divided in the following components:

1. Pressure - Temperature Rating
2. Analysis of Valve Body and Trunnion.
3. Analysis of Disc
4. Analysis of Stem
5. Analysis of Pin
6. Analysis of Bearings
7. Analysis of Mounting Bolts for the operator.

Stresses in these components are generated for most severe possible working condition combined with the Seismic loadings.

For each of the above areas, body of the report identifies the applicable criteria (article) of ASME Section III and the method of analysis. Actual calculations, results and comments for a particular valve size are included at the end.

PRESSURE - TEMPERATURE RATING

Criteria:

The service temperature and pressure of the valve will be in accordance with the article NB-3531,

Method:

Tables NB-3531-1 to NB-3531-7 provide design temperatures and corresponding service pressure for 150, 300, 400, 600, 900, 1500 and 2500 psi class valves. The tables may be interpolated for design temperatures other than those listed.

ANALYSIS OF VALVE BODY

Minimum Wall Thickness:

Criteria:

The minimum wall thickness of the valve body including the neck shall be in accordance with the article NB-3541.

Method:

Table NB-3542.1 identifies the minimum wall thickness (t_m), for inside diameter (d_m) of the valve body for primary pressure rating (Pr).

Body Shape Rules:

Criteria:

The valve body in question has practically no structural discontinuities. The body is compact and axi-symmetric. Hence the body shape rules of NB-3544 are not applicable in their entirety.

The external fillet radius at the junction of trunnion and body shall be in accordance with NB-3544.1.

Method:

The external fillet radius (r_2) at the junction of trunnion and body shall be governed by :

$$r_2 \geq 0.5\,(t_m)$$

where t_m = minimum required wall thickness.

Body Primary and Secondary Stresses:

Criteria:

Limits of the primary and secondary stresses will be in accordance with the requirement of NB-3545. The primary membrane stresses will be due to internal pressure. The secondary stresses will be calculated for the following load conditions:

(a) Discontinuity stresses in the crotch area

(b) Secondary stresses due to pipe-reactions

(c) Thermal stresses.

Method:

The primary-membrane stress (Pm), due to internal design pressure (P) will be:

$$P_m = \frac{(P)\,(d_m)}{2\,T_m} + \frac{P}{2}$$

Primary plus secondary stresses at the crotch region (Qp), due to internal pressure will be

$$Q_P = C_P \left(\frac{r_i}{T_m} + 0.5 \right) P_s$$

where $r_i = \frac{dm}{2}$

$Cp = 3$

Ps = Standard calculation pressure from fig. NB-3545.1-1 or 2

Secondary stress due to pipe-reactions are calculated as follows:

$$P_{ed} = \frac{(F_d)(S)}{G_d}$$

where Ped = stress due to direct or axial load effect

S = 30,000 psi

Gd = Metal area in the valve body at the junction of body and trunnion

Fd = 1/2 the cross-section area of standard connected pipe.

or

the value from figs NB-3542.2-2 or NB-3545.2-3

$$Peb = \frac{Cb.\ Fb.\ S}{Gb}$$

Where Peb = Stress due to bending of connected pipe

Fb = Bending Modulus of connected pipe

or

the Value from figs NB-3545.2-4 and NB-3545.2-5

Gb = Section Modulus of the body at the junction of trunnion and body.

Cb = stress index from fig NB-3545.2-6

$$Pct = \frac{2Fb.\ S}{Gt}$$

Where Pct = stress due to torsional load of the connecting pipe

Gt = Torsional modulus of the body at cross-section of the trunnion

For valve application at temperature less than 500°F, secondary thermal stresses are negligible.

For number of operating cycles less than 2000, no fatigue analysis is required.

For the normal duty operation the combined stress intensity (Sn) in the valve body shall be given by

$$Sn = Qp + Pe$$

Where Pe is the greater of Ped, Peb, or Pet.

Analysis of Trunnion

Criteria:

Trunnion shall be considered as a critical part of the valve body since it supports the operator and is subjected to the normal operating load and seismic load from the operator.

The allowable stress intensity in the trunnion shall be in accordance with NB-3546.3.

Method:

The stresses in the trunnion shall be computed at Section A-A due to the most severe mounting condition as shown in the figure 1.

The load characteristics in the trunnion shall be as follows:

1. Torsional load due to the operating torque
2. Bending load due to the weight of the operator
3. Bending load due to the combined effect of horizontal and vertical seismic load acting at the C.G. of the operator.

The Shear Stress (Ss) in the trunnion due to the operating load will be:

$$Ss = \frac{TC}{J}$$

Where T = operating torque

C = Distance of the outer most fibre from the N.A.

J = Polar moment of inertia of the trunnion.

The bending stress (S_1) at the section A-A due to dead wt of the operator will be:

$$S_1 \quad \frac{J \cdot C}{Ixx}$$

Where M_1 = B.M. at the Section A-A

Ixx = Moment of Inertia of the trunnion around the x - axis

Bending stress (S_2) at the section A-A due to vertical seismic load will be:

$$S_2 = \frac{(M_2) \cdot C}{Ixx}$$

Where = M_2 = B.M. at section A-A due to Vert. Seismic Load.

Bending stress (S_3) at the Section A-A due to horizontal seismic load will be:

$$S_3 = \frac{(M_3)\ (C_z)}{Izz}$$

Shere M_3 = B.M. at section due to horizontal seismic load.

C_z = Dist. of the outer most fibre from Z axis

Izz = Moment of inertia of trunnion around Z axis.

Resultant Bending Stress (S_B) in the trunnion at Section A-A will be:

$$S_B = \left[(S_1 + S_2)^2 + (S_3)^2 \right]^{1/2}$$

Maximum principle stress (σ) in the trunnion at Section A-A will be

$$\sigma = \left(\frac{S_B}{2}\right) + \left[\left(\frac{S_B}{2}\right)^2 + (S_s)^2\right]^{1/2}$$

ANALYSIS OF DISC

CRITERIA:

The disc shall be considered as a pressure retaining component.

The allowable stress intensity in the disc shall be in accordance with NB-3546.3.

METHOD:

The disc shall be considered as a beam simply supported along the stem axis and fixed (cantelever) about the stem axis.

The stresses in the disc will be calculated for pressure load and inertial seismic loads.

The max. bending stress (σ_{al}) along the shaft axis due to pressure and seismic load will be:

$$\sigma_{al} = \left[M_{alp} + M_{als} \right] \cdot \frac{C_{al}}{I_{al}}$$

Where Malp = B.M. along the stem due to pressure

Mals = B.M. along the stem due to seismic acceleration

Ial = Moment of inertia of the disc along the shaft axis.

Cal = Dist of the outer most fibre of the disc from the shaft axis.

The max. bending stress (σ_{ab}) about the stem axis due to pressure and seismic load will be:

$$\sigma_{ab} = \left[M_{abp} + M_{abs} \right] \cdot \frac{C_{ab}}{I_{ab}}$$

Where subscript ab indicate corresponding values about the stem axis.

The max. primary bending stress in the disc shall be:

$$\sigma_D = \left[(\sigma_{al})^2 + (\sigma_{ab})^2 \right]^{1/2}$$

ANALYSIS OF STEM

CRITERIA:

Stem shall be considered as a critical component since it's failure can lead to gross violation of pressure retaining parts. The allowable stresses in the stem shall be in accordance with NB-3546.3.

METHOD:

The stresses in the stem shall be computed for pressure, seismic and torsional loadings.

The bending stress (σ_y) in the stem along y axis due to pressure load and seismic acceleration of the disc will be:

$$\sigma_y = \frac{32}{\pi d^3} \left[\left(\frac{\pi}{4}\right)\left(D_D\right)^2\left(\frac{P}{2}\right) + \left(\frac{W_D}{2}\right)\left(g_y\right) \right] \cdot a$$

Where a = Stem overhang

Bending stress (σ_Z) in the stem along z axis due to seismic acceleration of the disc will be:

$$\sigma_Z = \frac{32}{\pi d^3} \left[\left(\frac{W_D}{2}\right) \left(g_Z\right) (a) \right]$$

The max. bending stress (σ_{BS}) in the stem will be:

$$\sigma_{BS} = \left[\left(\sigma_y\right)^2 + \left(\sigma_Z\right)^2 \right]^{1/2}$$

∴ Torsional stress (τ_s) in the stem due to operating load will be:

$$\tau_s = \frac{16\,T}{\pi d^3}$$

The max. stress intensity (σ_s) in the stem will be:

$$\sigma_s = \left(\frac{\sigma_{Bs}}{2}\right) + \left[\left(\frac{\sigma_{Bs}}{2}\right)^2 + (\tau_s)^2\right]^{1/2}$$

ANALYSIS OF PIN

CRITERIA:

Allowable stresses in the pin will be in accordance with NB-3546.2.

METHOD:

The pin will be in double shear due to the operating torque and inertial loading due to seismic acceleration of the disc.

The max. shear stress (τ_p) in the pin will be:

$$\therefore \tau_p = \frac{4}{n \pi d_p^2} \left[\left(\frac{T}{d} \right)^2 + \left(\frac{W_D \cdot g_x}{2} \right)^2 \right]^{1/2}$$

Where dp = dia. of pin

n = no. of pins

ANALYSIS OF BEARINGS

CRITERIA:

The bearing analysis and the allowable stresses in the bearing shall be in accordance with the standard engineering practice.

METHOD:

Each bearing shall be considered to share one-half of the total load. The stresses in the bearing will be due to the pressure loading and due to the seismic acceleration of the disc.

The max bearing pressure will be:

$$\sigma_{bp} = \left[\frac{\pi}{4} \cdot D_D^2 \cdot P + W_D \cdot g_x \right] \cdot \frac{1}{2(L_B)d}$$

Where L_B = Length of the bearing.

ANALYSIS OF BOLTS

CRITERIA:

Bolts are critical components from the stand-point of seismic analysis.

METHOD:

Considering the combined effect of seismic and direct loading, the max. load (P_2) carried by a bolt will be:

$$P_2 = \frac{(W)(L_y)(d_{12})}{\left[(d_{11})^2 + (d_{12})^2\right] N} + \frac{(W)(\gamma_x)(L_y)(d_{12})}{\left[(d_{11})^2 + (d_{12})^2\right] N}$$

Where N = no. of bolts in row 2

Other Nomenclatures are shown in fig. 1.

Max. tensile stress (σ_t) in the bolt will be:

$$\sigma_t = \frac{P_2}{A_t}$$

Where A_t = Stress area of the bolt.

Max. bearing stress (σ_b) on the threads will be:

$$\sigma_b = \frac{4 \cdot P_2}{\pi h (d_1^2 - d_2^2) n}$$

Where n = No. of threads per inch

h = Length of thread engagement

d_1 = dia. of bolt

d_2 = minor dia. of bolt

Max. shearing stress (τ_b) in the bolt will be:

$$\tau_b = \frac{P_s}{2(A_s)}$$

Where P_s = Shear load

A_s = Area of bolt in shear

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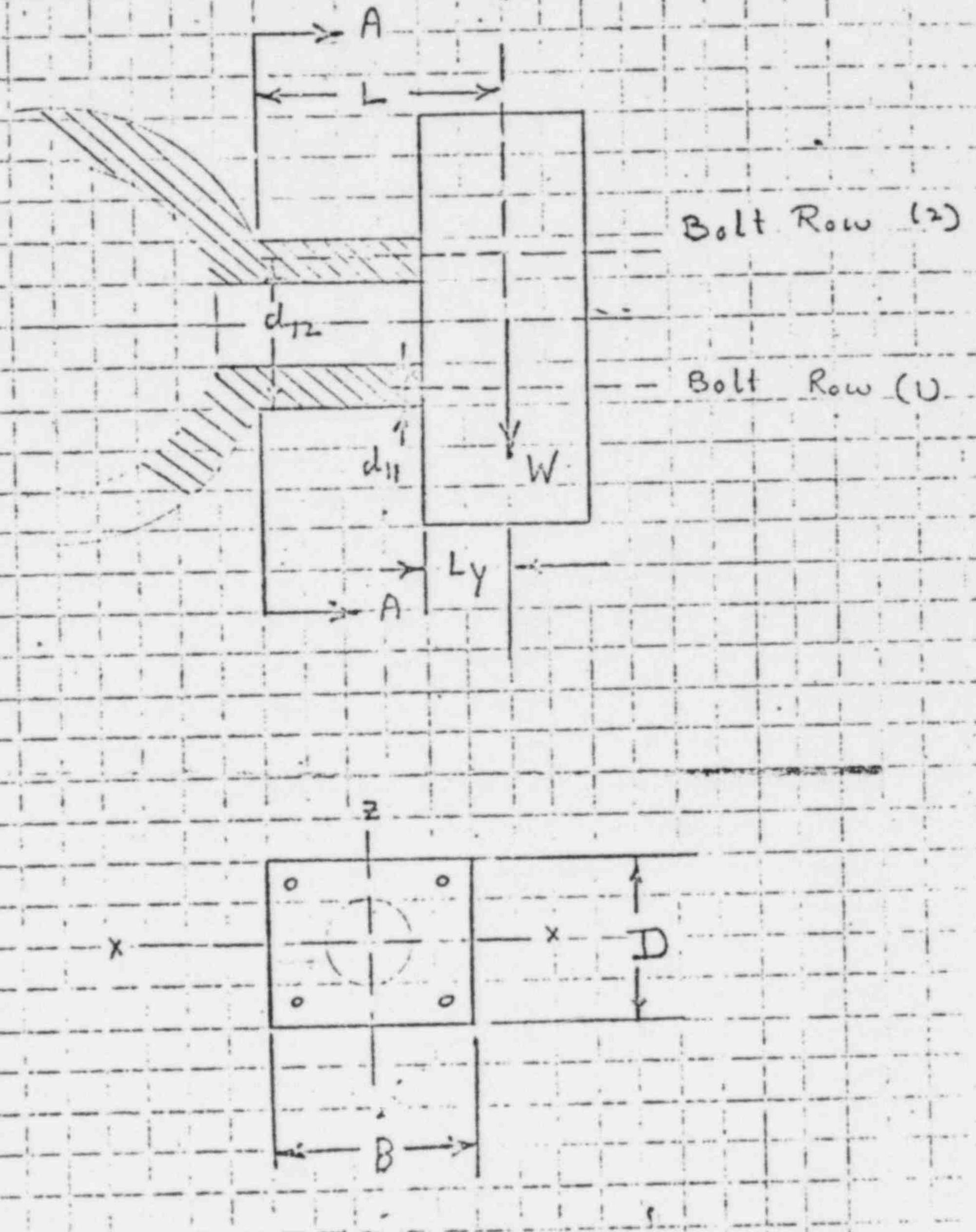

Cross-Section at A-A

Fig. 1

ANALYTICAL CALCULATIONS

FOR

VALVE SIZE: 18 inch

TYPE: 2FII

Prepared For: Pioneer Serv. & Engr. Co/Northern States Power

Customer Order No.: HIA-195 and 1195

Henry Pratt Order No.: N-6903-1 and 2

N-6904-1 and 2

SUMMARY OF RESULTS

BODY ANALYSIS

STRESS CLASSIFICATION	SECTION III SYMBOL	ANALYSIS REF. PAGE	STRESS LEVEL	ALLOWABLE STRESS
Primary Membrane stress intensity	Pm	5	470.0	18,300
Pr. + secondary stress due to internal press	Qp	6	2117.04	27,450
Secondary stresses due to Pipe reaction	Ped	7	718.0	27,450
	Peb		1702.43	27,450
	Pet		1702.43	27,450
Combined stress intensity	Sn	8	3819.47	54,900

SUMMARY OF RESULTS

VALVE COMPONENTS

COMPONENT	LOAD CONDITION	ANALYSIS REF. PAGE	STRESS LEVEL	ALLOWABLE STRESSES
Body-Trunnion	Operating load + Seismic load	11	1215	21,960
Disc	Op. Load + Seismic load	13	10,199.53	24,000
Stem	Op. Load + Seismic load	15	16932	24,000
Pin	Op. Load + Seismic load	16	10967	16200
Bearings	Op. load + Seismic load	17	1857.15	2000
Bolts	Op. load + Seismic load	19	15,312.89	67,500

PRESSURE-TEMPERATURE RATINGS

CALCULATIONS:

Valve Size	18	inches
End Connections (Flanged/Weld)	Flanged	
Pressure Class	150	psi
Service Temp (from customer's spec.)	100	°F
Allowable Service Press. (from tables NB-3531-1 to 7)	≈ 275	psi
Max. line press (from customers spec.)	≈ 46	psi

Comments:

VALVE BODY

Calculations:

Minimum Wall Thickness:

Inside dia. of valve body (dm) = 18.125 inches

Primary Pressure Rating (Pr) = 150 psi

Minimum Wall thickness (tm) = .48 inches
(from table NB-3542-1)

Actual Wall thickness (Tm) = 3.4375 inches

Body Shape Rule:

Actual external fillet radius (R_2) = N/A inches

Required Fillet radius $(r_2) \geq .3\,(T_m)$

N/A

Comments:

Primary Membrane Stress:

Design Pressure (P) = 150 psi

Primary Membrane Stress (pm) = $\frac{P.\ dm}{2Tm} + \frac{P}{2}$

$$= \frac{(150)(18.125)}{2(3.4375)} + \frac{150}{2}$$

$$= 470 \text{ psi}$$

Allowable Primary Membrane stress (Sm) for the valve body material = ASTM - A 516, GR 55

(from Appendix I, Tables I-1.1,2,3) is = 18,300 psi

Comment:

Primary + Secondary stress due to Internal Pressure:

$r_i = \frac{dm}{2} =$ 9.0625 inches

Ps (from fig. NB-3545.1-1 or 2) = 225 psi

$$Qp = 3\left(\frac{r_i}{T_m} + .5\right)P_s$$

$$= 3\left[\frac{9.0625}{3.4375} + .5\right]225 =$$

$= \quad \approx 2117.04$ psi

Secondary Stresses

Outside dia. of Valve body (d_o) = 25 inches

$$Gd = \frac{\pi}{4}\left[d_o^2 - d_m^2\right]$$

$$= \frac{\pi}{4}\left[(25)^2 - (18.125)^2\right]$$

$= \quad 296.48$ Sq. In.

Fd (from figs NB-3545.2-2 or 3) = 7.1

Fb (From figs.NB-3545.2-4 or 5) = 63

$$\frac{t_e}{r} = \frac{4 \cdot T_m}{D_o + d_m}$$

$$= \frac{4(3.4375)}{(25 + 18.125)}$$

$= \quad .31884$

Cb (For the value of $\frac{t_e}{r}$, from fig NB-3545.2-6) = 1.0

$$Gt = \frac{\pi}{16}\left[\frac{d_o^4 - d_m^4}{d_o}\right]$$

$$= \frac{\pi}{16}\left[\frac{(25)^4 - (18.125)^4}{25}\right]$$

= 2220.35

$$Ped = \frac{Fd.\ (30000)}{Gd.}$$

$$= \frac{7.1\ (30000)}{296.48}$$

= 718.43

$$Peb = \frac{Cb.\ Fb.\ (30000)}{Gb}$$

$$= \frac{(1)\ (63)(30000)}{1110.18}$$

= 1702.43 psi

$$Pet = \frac{2.Fb.\ (30000)}{Gt}$$

$$= \frac{(2)\ (63)\ (30,000)}{2220.35}$$

= 1702.43 psi

$$G_b = \frac{1}{2}\ G_t = 1110.18$$

Max. Secondary Stress (Pe) due to Pipe reaction will be greater of Ped, Peb or Pet

Pe = 1702.43 psi

Max. stress intensity (Sn) in valve body:

Sn = Qp + Pe

= 2117.04 + 1702.43

= 3819.47 psi

Allowable stress intensity (3Sm) = 54,900 psi

Comments:

BODY TRUNNION

CALCULATIONS:

Max. Operating Torque (T) =	18302 in. lbs.
Wt. of the operator (W) =	776 lbs.
Dist. of C.G. of the operator from Section A-A (L) =	6.5 inches
Width of the trunnion (B) =	8 inches
Depth of the trunnion (D) =	5.25 inches
Dia. of the Stem (d) =	2.25 inches
Dist. of the outer most fibre from x-axis (C) =	2.625 inches
Dist. of the outer-most fibre from z-axis (C_z) =	4.00 inches
Seismic accelerations g_x, g_y, g_z =	5

$$I_{xx} = 1/12\ BD^3 - \frac{\pi}{64} d^4$$

$$= \frac{1}{12}(8)(5.25)^3 - \frac{\pi}{64}(2.25)^4$$

$$= \underline{95.21} \text{ in.}^4$$

$$I_{zz} = 1/12\ DB^3 - \frac{\pi}{64} d^4$$

$$= \frac{1}{12}(5.25)(8)^3 - \frac{\pi}{64}(2.25)^4$$

$$= \underline{222.75} \text{ in}^4$$

$$J = I_{xx} + I_{zz}$$

$$= 95.21 + 222.75$$

$$= \underline{317.96} \text{ in.}^4$$

$$M_1 = WL$$

$$= (776)(6.5)$$

$$= \underline{5044.00} \text{ in. lbs.}$$

$$M_2 = (W)(g_z)(L)$$

$$= (776)(6)(6.5)$$

$$= \underline{30,264.0} \text{ in.lbs.}$$

$$M_3 = (W)(g_x)(L)$$

$$= (776)(5)(6.5)$$

$$= \underline{25,220.0} \text{ in.lbs.}$$

$$S_1 = \frac{(M_1)\ C}{I_{xx}}$$

$$= \frac{(5044.0)(2.625)}{95.21}$$

$$= \underline{139.066} \text{ psi}$$

$S_2 = \frac{(M_2)\ C}{I_{xx}}$

$= \frac{(30264.0)(2.625)}{95.21}$

$=$ 834.398 psi

$S_3 = \frac{(M_3)(C_2)}{I_{zz}}$

$= \frac{(16867.5)(4.0)}{222.75}$

$=$ 695.33 psi

$S_s = \frac{(T)C}{J}$

$= \frac{(18302)(2.625)}{317.96}$

$=$ 151 psi

$S_B = \left[(S_1 + S_2)^2 + (S_3)^2\right]^{1/2}$

$= \left[(139.06 + 834.40)^2 + (695.33)^2\right]^{1/2}$

$=$ 1196.29 psi

$\sigma = \frac{S_B}{2} + \left[\left(\frac{S_B}{2}\right)^2 + (S_s)^2\right]^{1/2}$

$= \frac{1196.29}{2} + \left[\left(\frac{1196.29}{2}\right)^2 + (151)^2\right]^{1/2}$

$=$ 1215 psi

Allowable stress intensity for class I and II seismic operation

(1.2 Sm) = 21,960 psi

Allowable Stress Intensity for Class III Seismic operation

(1.5 Sm) = 27,450 psi

Comments:

DISC

CALCULATIONS:

Dia. of the Disc (D_D) = 17.37 inches

Wt. of the Disc (W_D) = 145 lbs.

Cal = 1.343 inches

Cab = 1.343 inches

Ial = 12.62 $in.^4$

Iab = 35.10 $in.^4$

C.G. of the half disc (X_C) = 7.3 in.

$$Malp = .113\ P(D_D)^3$$

$$= .113(.150)(17.37)^3$$

$$= \qquad 88831.94 \text{ in. lbs.}$$

$$Mals = \left(\frac{W_D}{2}\right)(g_z)\frac{(D_D)}{2}$$

$$= \left(\frac{145}{2}\right)(6)(17.37)$$

$$= \qquad 3777.98 \text{ in. lbs.}$$

$$Mabs = \left(\frac{W_D}{2}\right)(g_z)(X_c)$$

$$= \left(\frac{145}{2}\right)(6)(7.3)$$

$$= \qquad 3175.5 \text{ in. lbs.}$$

$$Mabp = ..0833\ P\ (D_D)^3$$

$$= .0833(150)(11.37)^3$$

$$= \qquad 65,484.08 \text{ in. lbs.}$$

$$\sigma_{al} = \left[\text{Malp} + \text{Mals} \right] \cdot \frac{\text{Cal}}{\text{Ial}}$$

$$= \left[88831.94 + 3777.98 \right] \frac{1.343}{12.62}$$

$$= \underline{9855.4} \text{ psi}$$

$$\sigma_{ab} = \left[\text{Mabp} + \text{Mabs} \right] \cdot \frac{\text{Cab}}{\text{Iab}}$$

$$= \left[65484.08 + 3175.5 \right] \frac{1.343}{35.1}$$

$$= \underline{2627.06} \text{ psi}$$

$$\sigma_D = \left[(\sigma_{al})^2 + (\sigma_{ab})^2 \right]^{1/2}$$

$$= \left[(9855.4)^2 + (2627.06)^2 \right]^{1/2}$$

$$= \underline{10,199.53} \text{ psi}$$

Allowable primary membrane stress (Sm) for the disc material: ASTM-A351, CF8CF8

(from Appendix I, Tables I-1.1,2,3) is 20,000 psi

The max. allowable primary bending stress intensity

(1.5 Sm) = 30,000 psi

Allowable stress intensity for Class I and II Seismic operation (1.2 Sm) = 24,000 psi

Allowable stress intensity for Class III seismic operation

(1.5 Sm) = 30,000 psi

Comments:

STEM

CALCULATIONS:

Dia. of the stem (d) = 2.25 inches

Stem overhang (a) = 0.8 inches

$$\sigma_y = \frac{32}{\pi d^3}\left[\left(\frac{\pi}{4}\right)(D_D)^2\left(\frac{P}{2}\right) + \left(\frac{W_D}{2}\right)(g_y)\right]a$$

$$= \frac{32}{\pi(2.25)^3}\left[\frac{\pi}{4}(17.37)^2\left(\frac{150}{2}\right) + \left(\frac{145}{2}\right)(5)\right](0.8)$$

$= 12,973.63$ psi

$$\sigma_z = \frac{32}{\pi d^3}\left[\left(\frac{W_D}{2}\right)(g_z)(a)\right]$$

$$= \frac{32}{\pi(2.25)^3}\left[\left(\frac{145}{2}\right)(6)(0.8)\right]$$

$= 311.1816$ psi

$$\sigma_{BS} = \left[(\sigma_y)^2 + (\sigma_z)^2\right]^{1/2}$$

$$= \left[(12973.63)^2 + (311.1816)^2\right]^{1/2}$$

$= 12,977.36$ psi

$$\tau_s = \frac{16T}{\pi d^3}$$

$$= \frac{16(18302)}{\pi(2.25)^3}$$

$= 8183$ psi

$$\sigma_s = \left(\frac{\sigma_{bs}}{2}\right) + \left[\left(\frac{\sigma_{bs}}{2}\right)^2 + (\tau_s)^2\right]^{1/2}$$

$$= \left(\frac{12,977.36}{2}\right) + \left[\left(\frac{12977.36}{2}\right)^2 + (8183)^2\right]^{1/2}$$

$$= \underline{16932} \text{ psi}$$

The max. allowable stress intensity (Sm) for the stem material ASTM-A479, Type 316

from Appendix I, Table I-1.1,2,3) = 20,000 psi

Allowable stress intensity for Class I and II Seismic operation (1.2 Sm) = 24000 psi

Allowable Stress intensity for Class III Seismic Operation (1.5 Sm) = 30000 psi

Comments:

PIN

CALCULATIONS:

Average dia. of the pin (d_p) = 0.6875 inches

$$\tau_p = \frac{4}{\pi d_p^2}\left[\left(\frac{T}{d}\right)^2 + \left(\frac{W_D \cdot g_x}{2}\right)^2\right]^{1/2}$$

$$= \frac{4}{2\pi(0.6875)^2}\left[\left(\frac{18302}{2.25}\right)^2 + \left(\frac{(145)(5)}{2}\right)^2\right]^{1/2}$$

= 10967 p.s.i

The allowable shear stress intensity for the pin material = ASTM - A477, Type 316

is $(.6)(.9S_y)$ = 16200 psi

Comment:

BEARING

CALCULATIONS:

Length of the bearing (L_B) = 4.34375 inches

$$\sigma_{bp} = \left[\frac{\pi}{4}\cdot(D_D)^2 T + (W_D)(\gamma_x)\right]\cdot\frac{1}{2(L_B)d}$$

$$= \left[\frac{\pi}{4}(17.37)^2(150) + (145)(5)\right]\cdot\frac{1}{2(4.34375)(2.25)}$$

$$= 1857.15 \text{ psi}$$

The allowable bearing pressure for bearing material Nylon

is 2000 psi

Comment:

BOLTS

CALCULATIONS:

Type of bolt	1/2 - 13 NC	
Bolt Material	SAE - GR2	
No. of threads per inch (n) =	13	
Length of thread engagement (h) =	.8125	inches
Dia. of bolt (d_1) =	0.5	inches
Minor dia. of bolt (d_2) =	0.4	inches
Tensile Area (At) =	0.1419	Sq. In.
Shear Area (As) =	0.1257	Sq. In.
From the fig. 1 Ly =	4.625	inches
d_{11} =	0.625	
d_{12} =	4.875	

$$P_2 = \frac{(W)(Ly)(d_{12})}{\left[(d_{11})^2 + (d_{12})^2\right] N} + \frac{(W)(gx)(Ly)(d_{12})}{\left[(d_{11})^2 + (d_{12})^2\right] N}$$

$$= \frac{(776)(4.625)(4.875)}{[(0.625)^2 + (4.875)^2]2} + \frac{(776)(5)(4.625)(4.875)}{[(0.625)^2 + (4.875)^2]2}$$

$$= \underline{2172.90} \text{ lbs.}$$

Max. Tensile Stress (σ_t) = $\frac{P_2}{A_t} = \frac{2172.9}{.1419}$

$$= \underline{15,312.89} \text{ psi}$$

Max. Bearing stress on threads (σ_b) = $\frac{4.\ P_2}{\pi h (d_1^2 - d_2^2) n}$

$$= \frac{(4)(2172.9)}{\pi(0.8125)[(.5)^2 - (.4)^2](13)}$$

$$= \underline{2910.31} \text{ psi}$$

Max. Shear load (P_S) = W + W(gx)

$$= 776 + 776(5)$$

$$= \underline{46,56.00} \text{ lbs.}$$

Max. Shear Stress (τ_b) = $\frac{Ps}{As}$

$$= \frac{4656}{.1257}$$

$$= \underline{37,040.57} \text{ psi}$$

Allowable Stress Intensity $\underline{67,500}$ psi

Comment:

ATTACHMENT 3

General Arrangement Drawings

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