PRATT

HENRY PRATT COMPANY

creative engineering for fluid systems

STRESS REPORT FOR

48" RIA WITH BETTIS ACTUATOR

Project Site	Turkey Point Units 3 & 4
Customer	Florida Power & Light Co.
Engineer	Bechtel Power Corporation
Original Specification	5610-M-83
Original Purchase Order	5610-M-83
Original Pratt Job No.	7-3071-1 & 7-3071-2
Valve Tag Nos.	POV-3-2602 POV-3-2603
	POV-4-2602 POV-4-2603

General Arrangement Drawing E-586 Rev. 1

Prepared by:	Rao N. Kaza
Date:	9-15-81
Reviewed by:	T. J. Wrona
Date:	9-16-81
Certified by:	J V Ballun
Date:	9-18-81

JOHN V. BALLUN
39860
REGISTERED PROFESSIONAL ENGINEER OF
ILLINOIS

CONTENTS

I. Introduction

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

The analysis of the structural and operational adequacy of the valve assembly under such conditions is based principally upon containment pressure 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 vàlve 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 furnished by customer/ engineer or 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° elbow (upstream) oriented 90° out-of-plane with respect to valve shaft, and leading edge of disc closing toward outer wall of elbow. Effects of downstream piping on system back pressure have been covered in paragraph A.2. (above).

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 x 10^8 rads at a maximum incidence temperature of 350°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°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-27256-2 TORQUE TABLE 1 9 / 14 / 81

JOB:FLOR.PWR:TURKEY-PT 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.)= 1371.29 FEET/SEC AT 283 DEG.

CRIT.CASE INLET VELOCITY IS 1.48516 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 .692271 (47.375 IN)WITH STMIX.)FIRST SONIC(@ 72 DEG.V.A.)
ABSOL.MAX.TORQUE(FIRST SONIC)AT 72-68 DG.VLV.ANG.= 514276 IN-LBS @ 72 DEG.
MAX.TORQUE INCLUDES SIZE EFFECT(REYNOLDS NO.ETC)APPX. X 1.30269 FOR 47.375 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: 48"-R1A5:3/7.5 CLASS 75
DISC SIZE: 47.062 INCHES SYMMETRICAL DISC
SHAFT DIA.: 4 INCHES
BRG. COEF. OF FRCTN.: 5.00000E-03
SEATING FACTOR: 15
INLET PRESS.VAR.MAX.: 60.2 PSIA
OUTLET PRESSURE(P6): 33.93 PSIA (72 DEG. ACTUAL PRESS.ONLY(VAR.))
MAX.ANG.FLOW RATE: 550564. CFM: 1092862 SCFM: 60077.6 LB/MIN
CRIT.SONIC FLOW-90DG: 67628.4 LB/MIN AT 37.234 INLET PSIA
VALVE INLET DENSITY: .10912 LB/FT^3-MIN. .157531 LB/FT^3-MAX.
FULL OPEN DELTA P: 9.58431 PSI
SYSTEM CONDITIONS:
PIPE IN-PIPE-OUT -AND- AIR/STEAM MIXTURE SERVICE @ 283 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	41.70	29.38	12.32	.704	CR1092863	60077	0	1430	-1430	3.00
85	44.29	30.93	13.36	.698	1153003	63383	128955	1449	127506	3.29
80	46.31	31.38	14.93	.678	1158249	63672	189597	1496	188100	3.77
75	48.07	30.60	17.47	.637	1145101	62949	307545	1587	305957	4.13
72	49.01	28.67	20.34	.585	CR1054664	57977	489099	1727	487371	4.32
70	49.59	27.99	21.61	.564	CR1029718	56606	474955	1716	473238	4.45
65	50.86	25.10	25.77	.493	CR 928246	51028	442434	1691	440743	4.72
60	51.87	22.47	29.40	.433	CR 795188	43713	369698	1635	368063	4.95
55	52.61	19.97	32.64	.380	CR 668258	36735	361006	1647	359359	5.11
50	53.05	18.37	34.68	.346	CR 546198	30026	277366	1700	275666	5.22
45	53.20	17.21	35.99	.323	533747	29341	244680	1744	242936	5.25
40	53.33	16.45	36.88	.308	369400	20306	178488	1787	176700	5.28
35	53.72	15.67	38.04	.292	286493	15749	135682	1818	133863	5.39
30	54.34	15.22	39.12	.280	213264	11723	73408	1850	71557	5.55
25	55.16	14.97	40.19	.271	147567	8112	46702	1877	44825	5.76
20	56.15	14.82	41.32	.264	91239	5015	32506	1932	30574	6.05
15	57.23	14.72	42.50	.257	51981	2857	11899	2015	9883	6.38
10	58.34	14.71	43.63	.252	25970	1427	7606	2113	5492	6.72
5	59.40	14.70	44.69	.248	8656	475	4068	2209	1858	7.11
0	60.20	14.70	45.50	.244	0	0	35807	2141	33665	7.50

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37948 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 489099 IN-LBS @ 72 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 Evaluation

Model: Bettis 2744A-2SR-45

Rating: 125,000 in-lbs open and closed positions only
87,000 in-lbs intermediate positions

Max. valve torque: 489099 in-lbs

The Bettis cylinder operator furnished was specifically designed for the requirements of the original order which did not include LOCA conditions.

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

The operator model furnished has an approximate rating which exceeds the calculated valve torque for the following valve angles:

35 degrees open to 0 degrees (fully closed)

Listed in the attachments section of this report are the following documents used in evaluating the structural and operational adequacy of the actuators.

- Supplemental Torque Calculations (attachment #3)

IV. Conclusion

It is concluded that neither the valve structure (with present materials) nor the valve actuator are adequate to withstand the defined LOCA-induced loads based on the calculated torques developed in this analysis except for restricted valve opening as described below:

Specifically, the valve top shaft, disc pins, thrust bearing adjusting screw, trunnion bolts, operator bolts, and bonnet are shown to be overstressed except at valve disc angles of 50° or less. (See attachments #2 and #4.)

In addition, the calculated torques exceed the rating for the actuator except at valve disc angles of 35° or less.

V. Additional Information

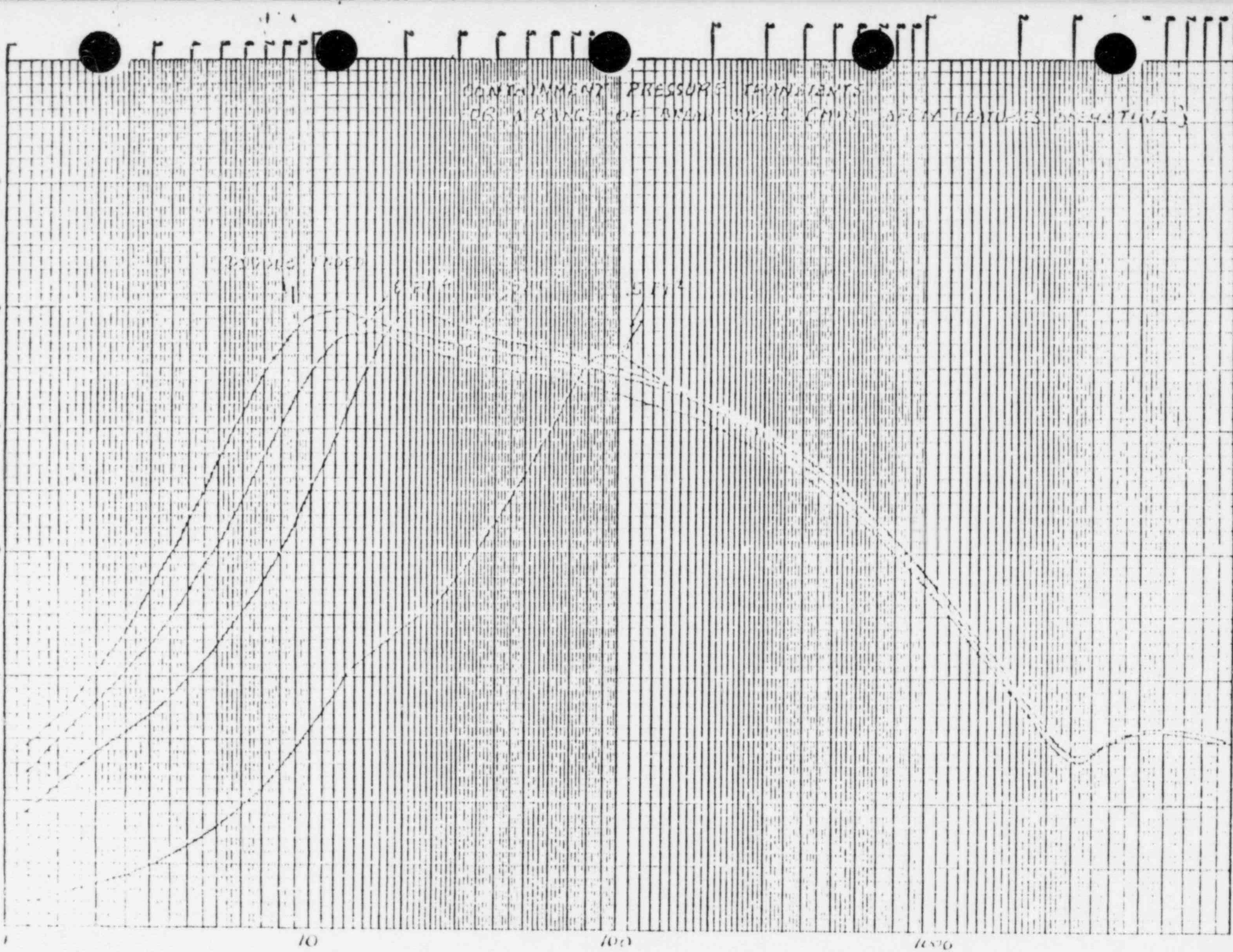
The following items are presented to describe how system factors affect torques developed in this analysis for your consideration and are informational only.

Further analysis by the customer/engineer is recommended prior to implementation.

1. An important factor governing the magnitude of the dynamic torque is delay time from the start of a LOCA incident to activation of the pressure sensing mechanism, which in turn initiates valve closure. Careful re-evaluation by the customer/engineer of the pressure sensing/timing sequence may render the present valve assembly functional through a significantly greater range of angles.
2. Installation of a convergent-divergent section downstream of the outside containment valve with a throat area sufficient to allow unrestricted ventilation during normal operation, but which will choke LOCA-induced flow while the valve is closing, through the critical range of 80°-60° open, could resultantly reduce the flow through the valve to subsonic levels.
3. An orifice plate installed similar to #2 above can also choke the system downstream and reduce flow through the valve to subsonic levels.
4. Mechanically restrict or block the valve disc to a maximum disc opening angle. (See attachment #3 for further illustration.)

ATTACHMENT 1A

PRESSURE vs. TIME GRAPHS


CONTAINMENT PRESSURE TRANSIENTS
60
50
40
30
20
10
0
CONTAINMENT PRESSURE, PSIG
1
10
100
1000
TIME, SECONDS
FIG 14.3.4-2

ATTACHMENT 1B

PRATT LETTER REGARDING
ADDITIONAL INFORMATION

PRATT

HENRY PRATT COMPANY

401 SOUTH HIGHLAND AVENUE · AURORA, ILLINOIS · 60507

February 16, 1981

Bechtel Power Corp.
Gaithersburg Power Division
15740 Shady Grove Road
Gaithersburg, MD 20760

Attention: Mr. Dick Baldwin

Subject: 48" and 54"
Purge Valve Analyses
Turkey Point Project
Florida Power & Light Co.
Pratt No.: D-27256

Dear Mr. Baldwin:

Confirming our recent telephone conversation, the Henry Pratt Company will furnish a revised purge valve analysis upon receipt of additional technical data.

Our general analysis of purge valves subjected to LOCA conditions indicates that this additional data has a significant impact on the maximum torque and resultant stresses in the valve assembly. It is, therefore, requested that the information provided be as accurate as possible.

We will require:

1. The combined resistance coefficient for all ventilation system components downstream or 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.

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2. Confirmation that the maximum delay time from LOCA to initiation of valve rotation is 4.2 seconds. Provide a minimum delay time as well.

This data will be used with previously submitted information to perform a new torque analysis, valve stress analysis, and operator evaluation as required.

Very truly yours,

HENRY PRATT COMPANY

T. J. Wrona

T. J. Wrona, Manager
Contract and Proposal Engineering

/kk

cc: G. L. Beane

ATTACHMENT 1C

CUSTOMER/ENGINEER RESPONSE
TO REQUEST FOR INFORMATION

Bechtel Power Corporation

Engineers—Constructors

15740 Shady Grove Road
Gaithersburg, Maryland 20760
301-258-3000

March 26, 1981

Mr. T. J. Wrona
Henry Pratt Company
401 South Highland Avenue
Aurora, Illinois 60507

Dear Mr. Wrona:

Turkey Point Units 3 & 4
Bechtel Job 5177-152
REA-TPN-31
Purge Valve Analysis
Bechtel Files: A-21, S-77.1
V-241

In response to the engineering data requirements listed in your letter dated February 16, 1981, we feel certain assumptions and considerations must accompany the numerical values and thus we answer as follows:

1) Regarding "The combined resistance coefficient of all ventilation components downstream of the valves (one for each valve size)...."

   We consider the conservative approach to be that condition which would pass the most Post Accident Flow. For that condition, all ductwork, except the seismically qualified and Q listed portions, would be removed in such a way as to not impede the accident flow path. The only qualified duct is the ten-foot penetration pipe between the two valves of any pair, which is the same diameter as the valve. Furthermore one of the two valves could be considered to fail in its blocked open position due to signal malfunction. Flow resistance coefficients vary considerably with valve angle. The entrance and exit coefficient for the penetration pipe also contributes to the total system resistance although the ten feet of pipe is essentially insignificant. The flow medium is a mixture of air ($k = 1.4$) and steam ($k = 1.3$) with the steam portion increasing as the accident progresses. The conservative approach would then be to use the lower friction of steam.

   Using 1979 Crane Technical Paper No. 410 - Flow of Fluids through Valves Fittings and Pipe, and 1977 ASHRAE Handbook of Foundamentals, we compile the following flow resistance coefficients:

PIPE	ITEM	COEF.	REF.
	Entry	0.78	Crane A-29
	Length	0.03	Crane 3-4 & A-22
	Exit	1.00	Crane A-29

Bechtel Power Corporation

VALVE	VALVE ANGLE	OPEN ANGLE	COEF.	REF.
	0	90	0.17	'79 ASHRAE 31.35
	10	80	0.52	
	20	70	1.6	
	30	60	3.9	
	40	50	10.8	
	50	40	33.	
	60	30	118.	
	70	20	751.	

NOTE: Take one valve at blocking angle selected during closure time and the other valve to vary from that angle to fully closed.

For example assuming a 30 degree blocking angle and the failure of the outboard valve to close, the inboard valve would have the following downstream flow resistance coefficients:

at 30 degrees 0.03 + 118 + 1.00 = 119.03 say 119
at 20 degrees 0.03 + 118 + 1.00 = 119.03 say 119

total system resistance coefficient, restricting the flow is:

at 30 degrees 0.78 + 118 + 0.03 + 118 + 1.00 = 237.81 say 238
at 20 degrees 0.78 + 751 + 0.03 + 118 + 1.00 = 870.81 say 871

the downstream resistance coefficient of the outboard valve is 1.00.

If both the valves were to operate and the recommended blocking angle were to be 50 degrees, the downstream resistance coefficient of the inboard valve would be tables as follows:

at 50 degrees 0.03 + 10.8 + 1.00 = 11.83 say 12
at 40 degrees 0.03 + 33 + 1.00 = 34.03 say 34
at 30 degrees 0.03 + 118 + 1.00 = 119.03 say 119
at 20 degrees 0.03 + 751 + 1.00 = 752.03 say 752

total system resistance coefficient would then be:

at 50 degrees 24
at 40 degrees 67
at 30 degrees 238
at 20 degrees 1,504

Assuming the generic k coefficients of butterfly valves are applicable to the specific Pratt valves supplied to Turkey Point, we have developed a family of curves to indicate minimum valve back pressure with maximum Post LOCA flow. (See Enclosure 1)

2) Regarding the "Confirmation that maximum delay time from LOCA to initiation of valve rotation is 4.2 seconds. Provide a minimum delay time as well."

Minimum delay times would certainly be more realistic. However, we must conservatively consider only one time for each accident containment pressure curve. Three of the four curves can be grouped together and the worst case envelope considered. The delay time for the envelope curve which is the double ended break is 2.7 seconds. The delay time for the 0.5 ft^2 pressure curve is approximately 5.3 seconds. Delay time is found by adding 1.5 seconds to the point on the graph when 6 psi is reached. Time of full closure will vary with blocking angle, however, we would expect an approximate linear relationship in regards to the maximum 90 degree closure interval of 5 seconds. The figure of 4.2 seconds mentioned in your letter is not a starting time for the Turkey Point curves and thus cannot be confirmed.

If there are any further questions, please contact us.

Yours very truly,

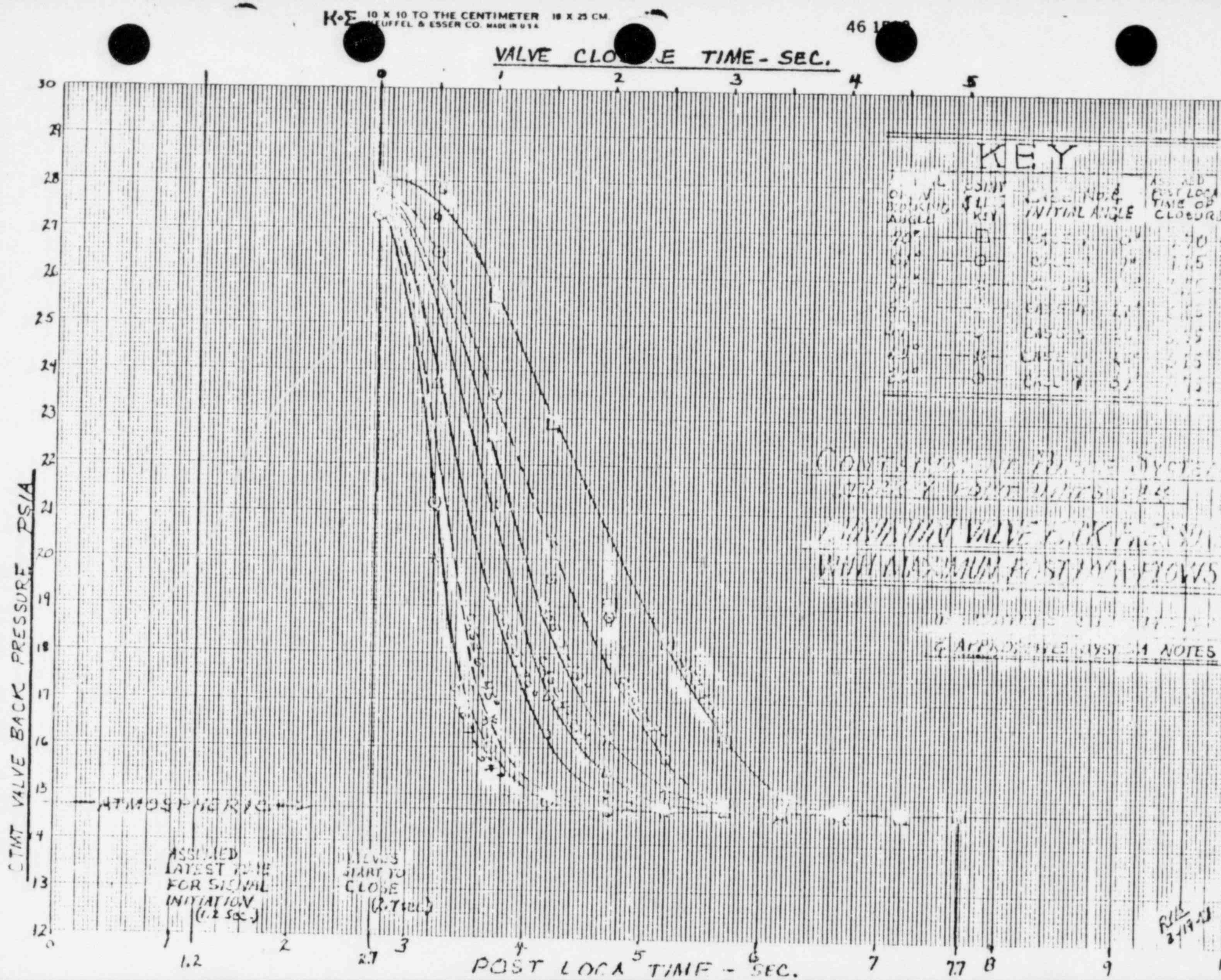
A. W. Wilk

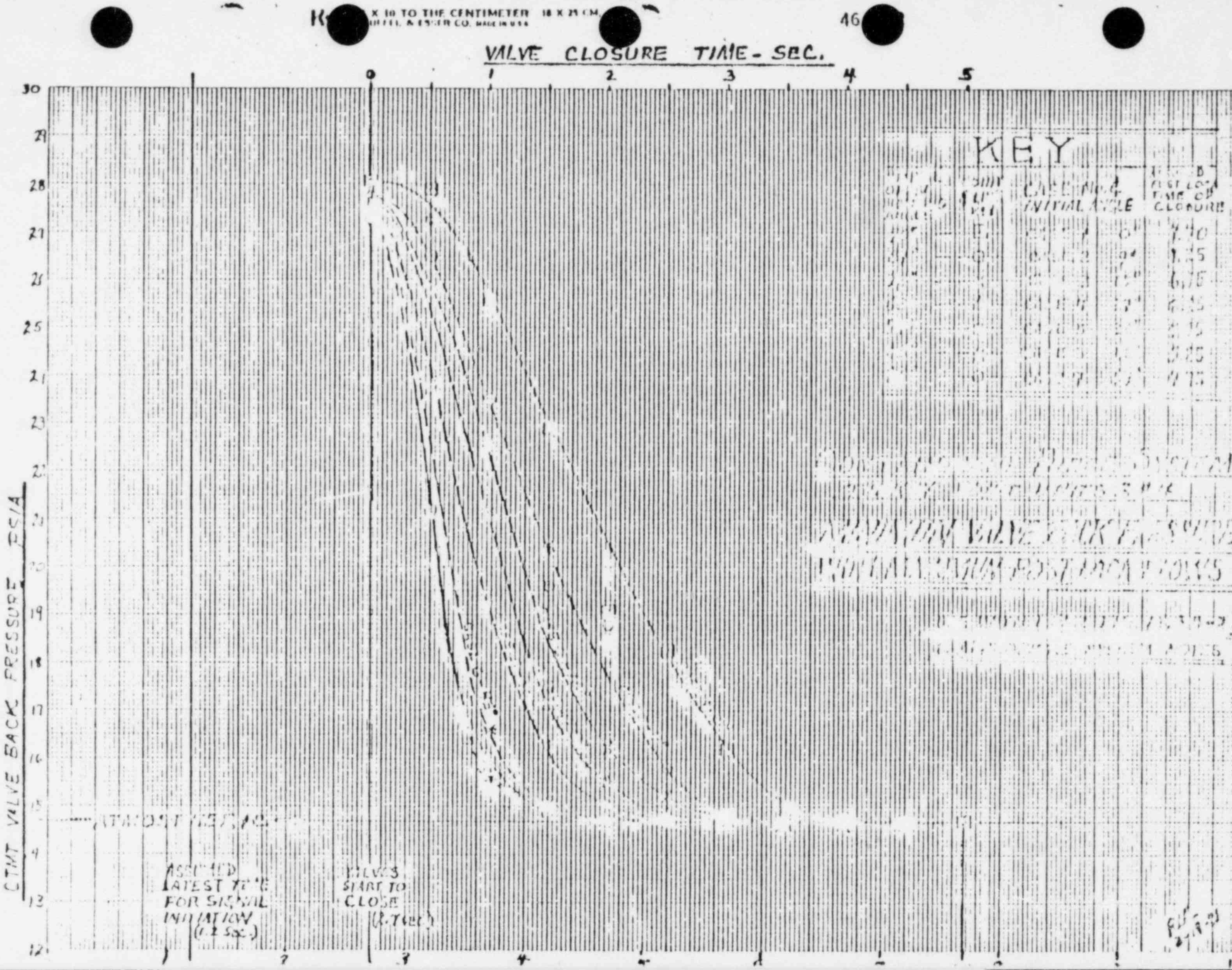
A. W. Wilk
Project Engineer

AWW/RVB:mfa

Enclosure: Curves

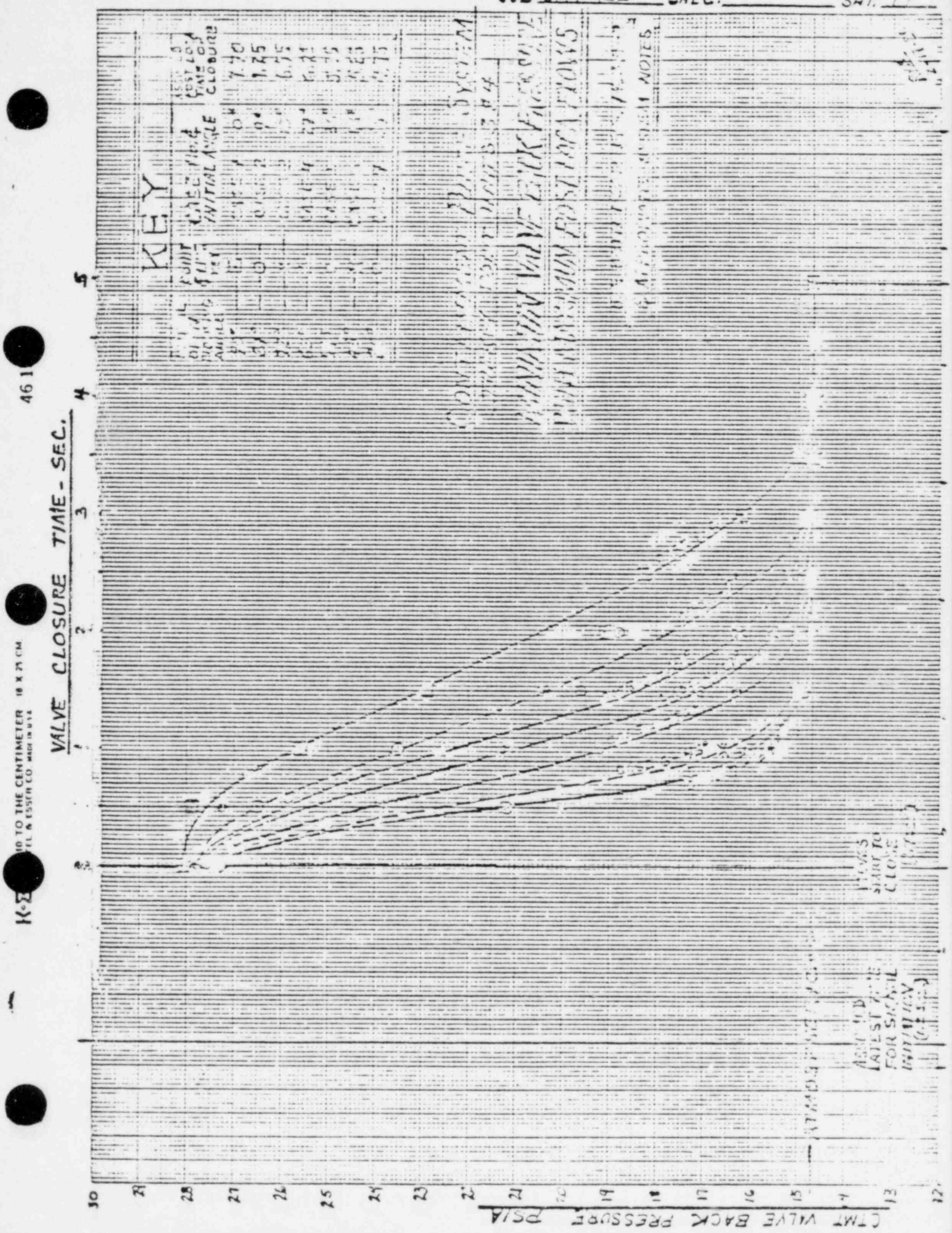
cc: W. H. Rogers, Jr., w/o
H. D. Mantz, w/o
S. G. Brain, w/3
G. R. Gram, w/1
F. A. Panzani, w/1
R. J. Acosta/R. Li, w/1
M. Crisler, w/1
D. W. Haase, w/1
D. T. Hughes, w/1
D. E. Douthit, w/1

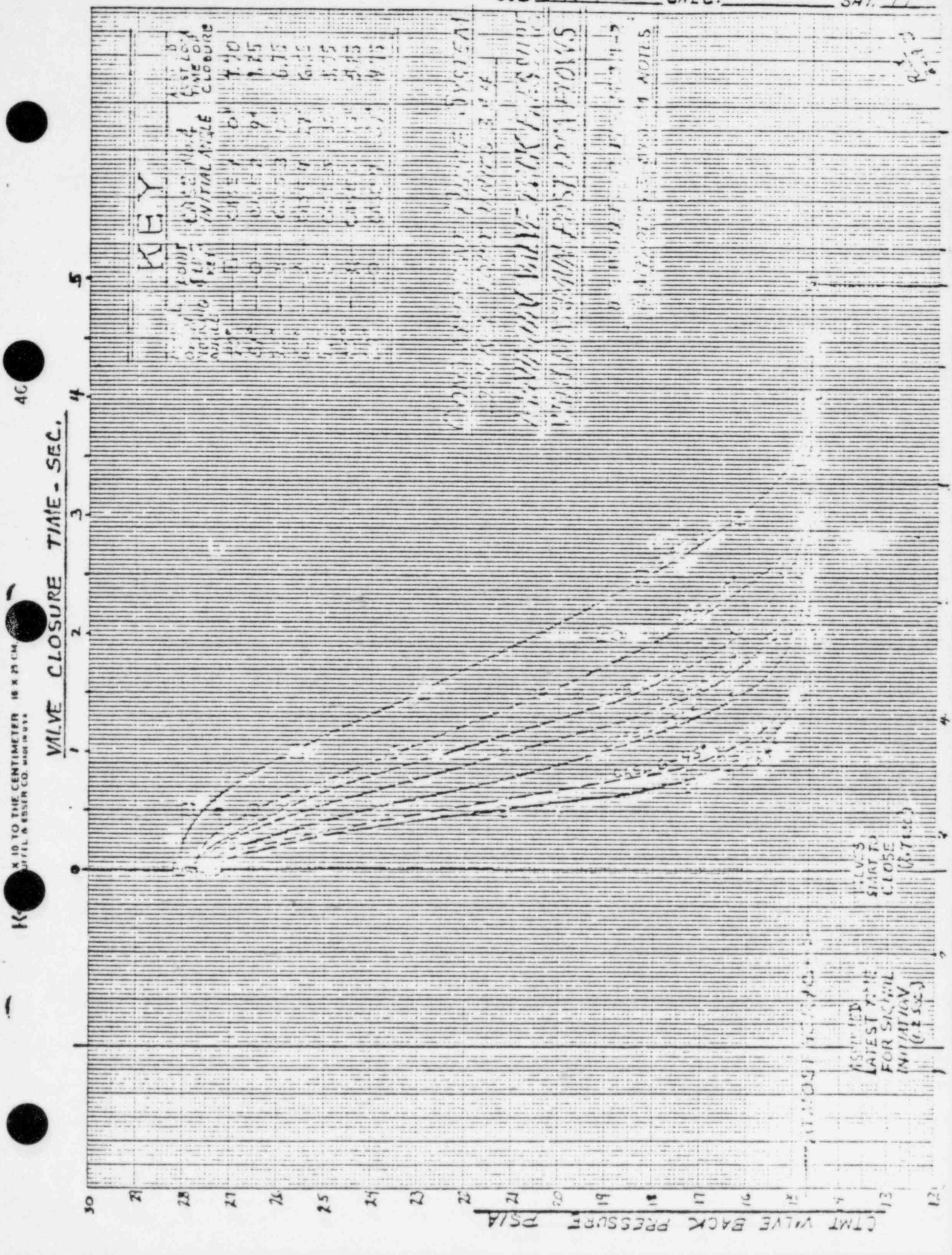

K•E 10 X 10 TO THE CENTIMETER 18 X 25 CM.
KEUFFEL & ESSER CO. MADE IN U.S.A.
46 1
VALVE CLOS E TIME - SEC.
KEY
KEY
INITIAL ANGLE
CASE
CONT
WITH MAXIMUM POST LOCA FLOWS
APPROXIMATE SYSTEM NOTES
CTMT VALVE BACK PRESSURE PSIA
ATMOSPHERIC
ASSUMED LATEST TIME FOR SIGNAL INITIATION (1.2 Sec.)
VALVES START TO CLOSE (2.7 sec.)
POST LOCA TIME - SEC.
0 1 2 3 4 5
12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30
0 1 1.2 2 2.7 3 4 5 6 7 7.7 8 9 10


X 10 TO THE CENTIMETER 18 X 25 CM.
46
VALVE CLOSURE TIME - SEC.
0
1
2
3
4
5
30
28
27
25
23
19
18
17
13
12
CTMT VALVE BACK PRESSURE PSIA
KEY
INITIAL
CLOSURE
ASSUMED LATEST TIME FOR SIGNAL INITIATION (1.2 Sec.)
VALVES START TO CLOSE

ENCLOSURE 1 V-241

JOB 5177-152 CALC. SHT. 17




VALVE CLOSURE TIME - SEC.
CTMT VALVE BACK PRESSURE PSIA
KEY
K&E 10 X 10 TO THE CENTIMETER 18 X 25 CM. KEUFFEL & ESSER CO. MADE IN U.S.A.
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ATTACHMENT 2

Nuclear Purge Valve

Stress Analysis

SEISMIC ANALYSIS

FOR 48 INCH

NUCLEAR PURGE VALVE

TABLE OF CONTENTS

LIST OF FIGURES

NOMENCLATURE

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

ANALYSIS NOMENCLATURE

A_f	Effective fluid pressure area based on fully corroded interior contour for calculating crotch primary membrane stress (NB-3545.1(a)), in^2
A_m	Metal area based on fully corroded interior contour effective in resisting fluid force on A_f (NB-3545.1 (a)), in^2
A_1	
A_2	Tensile area of thrust bearing adjusting screw.
A_3	Tensile area of bottom cover bolt, in^2
A_4	Shear area of bottom cover bolt, in^2
A_5	Tensile area of trunnion bolt, in^2
A_6	Shear area of trunnion bolt, in^2
A_7	Tensile area of operator bolt, in^2
A_8	Shear area of operator bolt, in^2
A_9	Tensile area of hub retainer bolts, in^2
A_{10}	Shear area of hub bolts, in^2
A_{11}	Tensile area of hub bolts, in^2
A_{12}	Shear area of thrust bearing retainer bolts, in^2
A_{13}	Tensile area of thrust bearing retainer bolts, in^2
B_1	Unsupported shaft length, in.
B_2	Bearing bore diameter, in.
B_3	Bonnet bolt tensile area, in.
B_4	Bonnet bolt shear area, in^2
B_5	Bonnet body cross-sectional area, in^2
B_6	Top bonnet weld size, in.
B_7	Bottom bonnet weld size, in.
B_8	Distance to outer fiber of bonnet from shaft on y axis, in.

B_9	Distance to outer fiber of bonnet from shaft on x axis, in.
C	A factor depending upon the method of attachment of head, shell diminsions, and other items as listed in NC-3225.2, dimensionless (Fig. NC-3225.1 thru Fig. NC-3225.3)
C_b	Stress index for body bending secondary stress resulting from moment in connected pipe (NB-3545.2 (b))
C_p	Stress index for body primary plus secondary stress, inside surface, resulting from internal pressure (NB-3545.2(a))
C_2	Stress index for thermal secondary membrane stress resulting from structural discontinuity.
C_3	Stress index for maximum secondary membrane plus bending stress resulting from structural discontinuity
C_6	Product of Young's modulus and coefficient of linear thermal expansion, at 500°F, psi/°F (NB-3550)
C_7	Distance to outer fiber of disc for bending along the shaft, in.
C_8	Distance to outer fiber of disc for bending about the shaft, in.
d	Inside diameter of body neck at crotch region (NB-3545.1(a)), in.
D_1	Valve nominal diameter, in.
D_2	Shaft diameter, in.
D_3	Hub retainer bore diameter, in.
D_4	Thrust collar outside diameter, in.
D_5	Thrust bearing bolt diameter, in.
D_6	Cover cap bolt diameter, in.
D_7	Trunnion bolt diameter, in.
D_8	Operator bolt diameter, in.
D_9	Bonnet bolt diameter, in.

D_{10}	Diameter of thrust bearing adjusting stud, in.
D_{11}	Outer diameter of trunnion, in.
E	Modulus of elasticity, psi
F_b	Bending modulus of standard connected pipe, as given by Figs. NB-3545.2-4 and NB-3545.2-5, $in.^3$
F_d	1/2 x cross-sectional area of standard connected pipe, as given by Figs. NB-3545.2-2 and NB-3545.2-3, $in.^2$
F_N	Natural frequency of respective assembly, hertz
F_x	W_3g_x--Seismic force along x axis due to seismic acceleration acting on operator extended mass, pounds.
F_y	W_3g_y--Seismic force along y axis due to seismic acceleration acting on operator extended mass, pounds.
F_z	W_3g_z--Seismic force along z axis due to seismic acceleration acting on operator extended mass, pounds.
g	Gravitational acceleration constant, $inch\text{-}per\text{-}second^2$
G_b	Valve body section bending modulus at crotch region (NB-3545.2(b)), in^3
G_d	Valve body section area at crotch region (NB-3545.2 (b)), in^2
G_t	Valve body section torsional modulus at crotch region (NB-3545.2(b)), in^3
g_x	Seismic acceleration constant along x axis
g_y	Seismic acceleration constant along y axis
g_z	Seismic acceleration constant along z axis
h_g	Gasket moment arm, equal to the radial distance from the center line of the bolts to the line of the gasket reaction (NC-3225), in.
H_1	Disc hub key height, in.
H_2	Top trunnion bolt square, in.
H_3	Bottom trunnion bolt square, in.

H_4 Bonnet bolt square, in.

H_5 Operator bolt square, in.

H_6 Bonnet bolt circle, in.

H_7 Operator bolt circle, in.

H_8 Bonnet height, in.

H_9 Actual body wall thickness, in.

I_1 Bonnet body moment of inertia about x axis, in^4

I_2 Bonnet body moment of inertia about y axis, in^4

I_3 Disc area moment of inertia for bending about the shaft, in^4

I_4 Disc area moment of inertia for bending along the shaft, in^4

I_5 Moment of inertia of valve body, in^4

I_6 Moment of inertia of shaft, in^4

I_7 Disc area moment of inertia for bending of unsupported flat plate, in^4

I_8 Moment of inertia of top trunnion plate.

J_1 Distance to neutral bending axis for top trunnion bolt pattern along x axis, in.

J_2 Distance to neutral bending axis for top trunnion bolt pattern along y axis, in.

J_3 Distance to neutral bending axis for bonnet bolt pattern along x axis, in.

J_4 Distance to neutral bending axis for bonnet bolt pattern along y axis, in.

J_5 Distance to neutral bending axis for operator bolt pattern along x axis, in.

J_6 Distance to neutral bending axis for operator bolt pattern along y axis, in.

K Spring Constant

K_1 Distance of bonnet leg from shaft centerline, in.

K_2	Thickness of disc above shaft, in.
K_3	Length along z axis to c.g. of bonnet plus adapter plate assembly, in.
K_4	Top trunnion width, in.
K_5	Top trunnion depth, in.
K_6	Height of top trunnion, in.
L_1	Valve body face-to-face dimension, in.
L_2	Thickness of operator housing under trunnion bolt, in.
L_3	Length of engagement of cover cap bolts in bottom trunnion, in.
L_4	Length of engagement of trunnion bolts in top trunnion, in.
L_5	Bearing length, in.
L_6	Length of shaft after retainer groove, in.
L_7	Length of engagement of bonnet bolts in adapter plate, in.
L_8	Length of engagement of bonnet bolts in bonnet, in.
L_9	Length of engagement of stub shaft in disc, in.
L_{10}	Disc hub key length, in.
L_{11}	Top trunnion weld height, in.
m	Reciprocal of Poisson's ratio
M	Mass of component
M_x	$W_3(g_y Z_o + g_z Y_o)$, operator extended mass seismic bending moment about the x axis, acting at the base of the operator, in-lbs.
M_y	$W_3(g_x Z_o + g_z X_o)$, operator extended mass seismic bending moment about the y axis, acting at the base of the operator, in-lbs.
M_z	$(W_3(g_x Y_o + g_y X_o)$, operator extended mass seismic bending moment about the z axis, in-lbs.
$\overline{Mx}$	Mx+FyT5, operator extended mass seismic bending moment about the x axis, acting at the bottom of the adapter plate, in-lbs.

$\overline{My}$	My+FxT5, operator extended mass seismic bending moment about the y axis, acting at the bottom of the adapter, in-lbs.
$\overline{\overline{Mx}}$	$Mx+Fy(T_5+H_8)+gyW_4K_3$, operator extended mass seismic bending moment about the x axis, acting at the base of the bonnet, in-lbs.
$\overline{\overline{My}}$	$My+Fx(T_5+H_8)+g_xW_4K_3$, operator extended mass seismic bending moment about the y axis, acting at the base of the bonnet, in-lbs.
M_8	Bending moment at joint of flat plate to disc hub, in-lbs.
N_a	Permissible number of complete start-up/shut-down cycles at hr/100°F/hr/hr fluid temperature change rate (NB-3545.3)
NA	Not applicable to the analysis of the system.
N_1	Number of top disc pins
N_2	Number of operator bolts
N_3	Number of trunnion bolts
Pd	Design pressure, psi
Pr	Primary pressure rating, pound
Ps	Standard calculation pressure psi
Pe	Largest value among Peb, Ped, Pet, psi
Peb	Secondary stress in crotch region of valve body caused by bending of connection standard pipe, calculated according to NB-3545.2(b), psi
Ped	Secondary stress in crotch region of valve body caused by direct or axial load imposed by connected standard piping, calculated according to NB-3545.2(b), psi
Pet	Secondary stress in crotch region of valve body caused by twisting of connected standard pipe, calculated according to NB-3545.2(b), psi
P_m	General primary membrane stress intensity at crotch region, calculated according to NB-3545.1(a), psi
P_m'	Primary membrane stress intensity in body wall, psi

Q_p	Sum of primary plus secondary stresses at crotch resulting from internal pressure, (NB-3545.2(a)), psi
Q_T	Thermal stress in crotch region resulting from 100°F/hr fluid temperature change rate, psi
Q_{T1}	Maximum thermal stress component caused by through wall temperature gradient associated with 100°F/hr fluid temperature change rate (NB-3545.2(c)), psi
Q_{T2}	Maximum thermal secondary membrane stress resulting from 100°F/hr fluid temperature change rate, psi
Q_{T3}	Maximum thermal secondary membrane plus bending stress resulting from structural discontinuity and 100°F/hr fluid temperature change rate, psi
Q_1	Distance to bolts in bolt pattern on hub block, in.
Q_2	Distance to bolts in bolt pattern on hub block, in.
Q_3	Distance to bolts in bolt pattern on hub block, in.
Q_4	Distance to bolts in bolt pattern on hub block, in.
Q_5	Distance to bolts in bolt pattern on hub block, in.
Q_6	Distance to bolts in bolt pattern on hub block, in.
Q_7	Distance from shaft centerline to disc plate, in.
r	Mean radius of body wall at crotch region (Fig. NB-3545.2(c)-1), in.
r_i	Inside radius of body at crotch region for calculating Qp (NB-3545.2(a)), in.
r_2	Fillet radius of external surface at crotch (NB-3545.2(a)), in.
R_4	Disc radius, in.
R_5	Shaft radius, in.
R_m	Mean radius of body wall, in.
R_6	Radius to gasket in cover cap, in.
R_7	Distance from shaft centerline to retaining bolt of thrust bearing.
S	Assumed maximum stress in connected pipe for calculating P_e (NB-3545.2(b)), 30,000 psi

S_m	Design stress intensity, (NB-3533), psi
S_n	Sum of primary plus secondary stress intensities at crotch region resulting from 100°F/hr temperature change rate (NB-3545.3), psi
S_{p1}	Fatigue stress intensity at inside surface in crotch region resulting from 100°F/hr fluid temperature change rate (NB-3545.3), psi
S_{p2}	Fatigue stress intensity at outside surface in crotch region resulting from 100°F/hr fluid temperature change rate (NB-3545.3), psi

S(1) through S(83) are listed after the alphabetical section.

t_e	Minimum body wall thickness adjacent to crotch for calculating thermal stresses (Fig. NB-4545.2(c))-1), in.
t_m	Minimum body wall thickness as determined by C.C. 1678, in.
T_e	Maximum effective metal thickness in crotch region for calculating thermal stresses, (Fig. NB-3545.2 (c)-1), in.
$\Delta T2$	Maximum magnitude of the difference in average wall temperatures for walls of thicknesses t_e, T_e, resulting from 100°F/hr fluid temperature change rate, °F.
T_1	Thickness of cover cap behind bolt head, in.
T_2	Thickness of adjusting screw head, in.
T_3	Thrust collar retaining plate thickness, in.
T_4	Cover cap thickness, in.
T_5	Adapter plate thickness, in.
T_6	Thickness of bottom bonnet plate, in.
T_7	Thickness of top bonnet plate, in.
T_8	Maximum required operating torque for valve, in-lbs.
T_9	Shaft retainer thickness on hub, in.
T_{10}	Bottom cover plate thickness, in.
T_{11}	Top trunnion wall thickness, in.
T_{12}	Thickness of top trunnion plate, in.

U_1	Area of bottom bonnet weld, in^2
U_2	Area of top bottom weld, in^2
U_3	Thrust bearing coefficient of friction
U_4	Bearing friction torque due to pressure loading (shaft journal bearing)
U_5	Bearing friction torque due to pressure loading plus seismic loading (shaft journal bearings)
U_6	Thrust bearing friction torque
V_1	Distances to bolts in bolt pattern on adapter plate, in.
V_2	Distances to bolts in bolt pattern on adapter plate, in.
V_3	Distances to bolts in bolt pattern on adapter plate, in.
V_4	Distances to bolts in bolt pattern on adapter plate, in.
V_5	Distance to bolts in bolt pattern on bonnet, in.
V_6	Distance to bolts in bolt pattern on bonnet, in.
V_7	Distance to bolts in bolt pattern on bonnet, in.
V_8	Distance to bolts in bolt pattern on bonnet, in.
W	Total bolt load, pounds
W_1	Valve weight, pounds
W_2	Banjo weight, pounds
W_3	Operator weight, pounds
W_4	Bonnet and adapter plate assembly weight, pounds.
W_6	Weld size of disc structural welds, inches
W_7	Weight of disc, pounds
W_8	Length of weld around perimeter of bonnet, in.
X_o	Eccentricity of center of gravity of operator extended mass along x axis, inches.
Y_o	Eccentricity of center of gravity of operator extended mass along y axis, inches.

z_o	Eccentricity of center of gravity of operator extended mass along z axis, inches
z_1	Bending section modulus of bonnet welds in x direction, in^3
z_2	Bending section modulus of bonnet welds in y direction, in^3
z_3	Torsional section modulus of bottom bonnet welds, in^3
z_4	Torsional section modulus of top bonnet welds, in^3
Δy	Maximum static deflection of component, inches
z_7	Distance to edge of disc hub, inches
z_8	Thrust bearing stud diameter, in.

ANALYSIS NOMENCLATURE

S (1) = Combined bending stress in disc, psi

S (2) = Bending stress in disc due to bending along the shaft, psi

S (3) = Bending stress in disc due to bending about the shaft, psi

S (4) = Combined stress in shaft, psi

S (5) = Combined bending stress in shaft, psi

S (6) = Combined shear stress in shaft, psi

S (7) = Bending stress in shaft due to seismic and pressure load along x-axis, psi

S (8) = Bending stress in shaft due to seismic load along y-axis, psi

S (9) = Torsional shear stress in top shaft due to operating torque, psi

S(10) = Direct shear stress in shaft due to seismic and pressure loads, psi

S(11) = Shear tear out of retainer in shaft groove, psi

S(12) = Shear tear out of shaft groove, psi

S(13) = Bearing stress on retainer and groove, psi

S(14) = Tensile stress in retainer bolts, psi

S(15) = Bearing stress on hub keyway, psi

S(16) = Shear stress on key, psi

S(17) = Combined stress on hub block bolts, psi

S(18) = Combined tensile stress on hub block bolts, psi

S(19) = Shear stress in hub block bolts, psi

S(20) = Shear tear out of shaft through hub block, psi

S(21) = Compressive load on shaft bearings, lbs.

S(22) = Bearing stress on thrust collar, psi

S(23) = Shear stress in adjusting screw head, psi

S(24) = Combined stress in adjusting screw, psi

S(25) = Direct tensile stress on adjusting screw, psi

S(26) = Torsional shear stress on adjusting screw, psi

S(27) = Shear stress in adjusting screw threads, psi

S(28) = Combined stress in retainer bolts, psi

S(29) = Tensile stress in retainer bolts, psi

S(30) = Shear stress in retainer bolts, psi

S(31) = Shear tear out of thrust bearing bolts, psi

S(32) = Shear stress in cover plate, psi

S(33) = Shear tear out of bolts through tapped holes in trunnion, psi

S(34) = Shear tear out of cover cap bolt head through bottom cover cap, psi

S(35) = Combined stress in cover cap bolts, psi

S(36) = Shear stress in cover cap bolts due to torsional loads, psi

S(37) = Direct tensile stress in cover cap bolts, psi

S(38) = Combined stress in cover cap, psi

S(39) = Radial stress in cover cap, psi

S(40) = Tangential stress in cover cap, psi

S(41) = Shear stress in cover cap, psi

S(42) = Shear tear out of trunnion bolt through tapped hole in trunnion, psi

S(43) = Bearing stress of trunnion bolt on tapped hole in trunnion, psi

S(44) = Bearing stress of trunnion bolt on through hole in bonnet plate, psi

S(45) = Shear tear out of trunnion bolt head through bonnet plate, psi

S(46) = Combined stress in trunnion bolt, psi

S(47) = Direct tensile stress in trunnion bolt, psi

S(48) = Tensile stress in trunnion bolt due to bending moment, psi

S(49) = Direct shear stress in trunnion bolt, psi

S(50) = Shear stress in trunnion bolt due to torsional load, psi

S(51) = Shear tear out of operator bolt head through bonnet, psi

S(52) = Bearing stress of operator bolt on through hole in bonnet, psi

S(53) = Combined stress in operator bolts, psi

S(54) = Direct tensile stress in operator bolts, psi

S(55) = Tensile stress in operator bolt due to bending moment, psi

S(56) = Direct shear stress in operator bolts, psi

S(57) = Shear stress in operator bolt due to bending moment, psi

S(58) = Combined stress in bonnet body, psi

S(59) = Direct tensile stress in bonnet body, psi

S(60) = Tensile stress in bonnet body due to bending moment, psi

S(61) = Direct shear stress in bonnet body, psi

S(62) = Shear stress in bonnet body due to torsional load, psi

S(63) = Combined shear stress in bottom bonnet weld, psi

S(64) = Total tensile stress in bottom bonnet weld, psi

S(65) = Total shear stress in bottom bonnet weld, psi

S(66) = Direct tensile stress in bottom bonnet weld, psi

S(67) = Tensile stress in bottom bonnet weld due to bending moment, psi

S(68) = Direct shear stress in bottom bonnet weld, psi

S(69) = Shear stress in bottom bonnet weld due to torsional load, psi

S(70) = Combined shear stress in top bonnet weld, psi

S(71) = Total tensile stress in top bonnet weld, psi

S(72) = Total shear stress in top bonnet weld, psi

S(73) = Direct tensile stress in top bonnet weld, psi

S(74) = Tensile stress in top bonnet weld due to bending moment, psi

S(75) = Direct shear stress in top bonnet weld, psi

S(76) = Shear stress in top bonnet weld due to torsional load, psi

S(77) = Combined stress in the trunnion body, psi

S(78) = Direct tensile stress, psi

S(79) = Bending tensile stress, psi

S(80) = Direct shear stress, psi

S(81) = Torsional shear stress, psi

SUMMARY TABLE INTRODUCTION

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

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

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

Component Name

Code Reference (when applicable)

Stress Level Name and Symbol

Analysis Reference Page

Material Specification

Actual Stress Level

Allowable Stress Level

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

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

Summary Table Introduction

Component Name

Natural Frequency Symbol

Analysis Reference Page

Component Material

Natural Frequency

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

Pages 21 - 29 , stress level summary, frequency analysis summary and valve dimensional data sheets have been assembled at the beginning of the report submittal. They are located directly behind the design review record for the corresponding production order.

TABLE 1 STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABL STRESS LEV PSI
BODY	NB-3545.1	Primary Membrane Stress in Crotch Region	P_m	36	ASTM A-36	1611	Sm 12600
		Primary Membrane	P_m'	36	ASTM A-36	1033	Sm 12600
	NB-3542.2	Primary Plus Secondary Stress Due to Internal Pressure	Q_p	38	ASTM A-36	3579	Sm 12600
	NB-3545.2	Pipe Reaction Stress Axial Stress Bending Torsion	 P_{ed} P_{eb} P_{et}	38	ASTM A-36	 2542 4773 4773	1.5Sm 18900
	NB-3545.2	Thermal Secondary Stress	Q_t	38	ASTM A-36	968	Sm 12600
	NB-3545.2	Primary Plus Secondary Stress	S_n	38	ASTM A-36	6958	3Sm 37800
	NB-3545.3	Normal Duty Fatigue Stress $N_a \geq 2000$	S_p	39	ASTM A-36	7101	1.5Sm 18900
Disc	NB-3546.2	Combined Bending Stress in Disc	S(1)	40	ASTM A-36	7682	1.5Sm 18900
Shaft	NB-3545.3	Combined Stress in Shaft	S(4)	41	ASTM A-276 Type 316 Condition A	54126	1.5Sm 30000

TABLE 1 STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWAB STRESS LE PSI
DISC PINS		Shear Stress in Top Pin	S(20A)	43A	ASTM A-276 Type 316	32730	(.9)(.6)S 16200
		Bearing Stress on Top Pins in Shaft	S(20B)	43A	ASTM A-276 Type 316	12847	.9Sy 27000
Shaft Bearings		Compressive Stress on Shaft Roller Bearings	S(21)	45	(SKF-#23222C)	76688	125,000
Thrust Bearing		Bearing Stress on Thrust Collar	S(22)	46	ASTM B-164 Condition A	491	Sm 13600
		Shear Stress in Adjusting Screw Threads	S(24)	46	ASTM B-164 Condition A	37211	(.9)(.6)S 16200
		Combined Stress in Retainer Bolts	S(28)	48	ASTM B-164 Condition A	18816	.9Sy 27000
		Shear Tear out of Thrust Bearing Retainer Bolts	S(31)	48	ASTM B-164 Condition A	736	(.9)(.6)S 16200

TABLE 1

STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABL STRESS LEV PSI
Bottom Cover		Shear Tear Out of Bolts in Bottom Trunnion	S(33)	49	ASTM A-36	1824	.6Sm 7560
		Shear Tear Out of Cover Bolt Head Through Bottom Cover	S(34)	49	ASTM A-36	1360	.6Sm 7560
		Combined Stress in Cover Bolts	S(35)	49	ASTM A-193 GR.B7	7958	Sm 25000
		Combined Stress in Cover	S(38)	49	ASTM A-36	1088	Sm 12600
Operator Mounting		Shear Tear Out of Trunnion Bolts Thru Tapped Hole in Trunnion	S(42)	51	ASTM A-36	812	.6Sm 7560
		Bearing Stress on Tapped Holes in Trunnion	S(43)	51	ASTM A-36	12330	Sm 12600
		Bearing Stress of Trunnion Bolt on Through Hole in Bonnet	S(44)	51	ASTM A-36	19974	Sm 12600
		Shear Tear Out of Trunnion Bolt Heads Through Bonnet	S(45)	53	ASTM A-36	702	Sm 12600
		Combined Stress in Trunnion Bolt	S(46)	53	SAE GR.2	53873	.9Sy 25200
		Shear Tear Out of Operator Bolt Head Thru Bonnet	S(51)	53	ASTM A-36	721	.6Sm 7560

TABLE 1

STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVEL PSI
Operator Mounting (Cont'd)		Bearing Stress on Through Holes in Bonnet	S(52)	53	ASTM A-36	22047	Sm 12600
		Combined Stress in Operator Bolts	S(53)	53	SAE GR.2	35396	.9Sy 25200
		Combined Stress in Bonnet Body	S(58)	55	ASTM A-36	85075	Sm 12600
		Combined Shear Stresses in Bottom Welds	S(63)	57		6267	.6Sm 72000
		Combined Shear Stress in Top Bonnet Weld	S(70)	57		6234	.6Sm 7200
		Combined Stress in Trunnion Body	S(77)	58	ASTM A-36	759	Sm 12600

TABLE 2 NATURAL FREQUENCIES OF VALVE COMPONENTS

COMPONENT NAME	NATURAL FREQUENCY SYMBOL	REF. PAGE	MATERIAL	NATURAL FREQUENCY (HERTZ)
BODY	F_{N_1}	63	ASTM A-36	5460
BANJO	F_{N_2}	64	ASTM A-276 Type 316 Condition A	777
COVER CAP	F_{N_3}	64	ASTM A-36	1016
BONNET	F_{N_4}	65	ASTM A-36	515

DIMENSIONAL DATA

Job Number: D-27256 Valve Size: 48" R1A5

Operator Mounting: BONNET Operator: BETTS 2744A-20R-45

A_f	185.06	B_8	5.126	F_d	28
A_m	10.03	B_9	5.126	F_x	2100
A_1	N/A	C	.3	F_y	2100
A_2	.373	C_b	1	F_z	2800
A_3	.334	C_p	3	G_b	3928.3
A_4	.302	C_2	.42	G_d	330.46
A_5	.606	C_3	.45	G_t	7856.6
A_6	.551	C_6	249	g_x	3
A_7	.334	C_7	4.625	g_y	3
A_8	.302	C_8	4.625	g_z	4
A_9	.142	D_1	48"	H_1	.875
A_{10}	.551	D_2	4	H_2	12.25
A_{11}	.606	D_3	1.25	H_3	7.4246
A_{12}	.068	D_4	3.75	H_4	12.25
A_{13}	.078	D_5	.375	H_5	N/A
B_1	4.125	D_6	.375	H_6	N/A
B_2	4	D_7	.75	H_7	12.375
B_3	.606	D_8	.75	H_8	8
B_4	.551	D_9	1	H_9	2.125
B_5	20.12	D_{10}	.75	I_1	296.34
B_6	N/A	D_{11}	11	I_2	186.76
B_7	N/A	F_b	625	I_3	749.04

I_4	749	L_{11}	.375	R_4	23.531
I_5	101400	m	3.5	R_5	2
I_6	12.5663	M_x	8007	R_6	4.5
I_7	339.02	M_y	8007	R_7	1.50
I_8	1.229	M_z	0	R_m	24.75
J_1	1.25	$\overline{M_x}$	26187	S	30000
J_2	1.25	$\overline{M_y}$	26187	t_e	1.75
J_3	1.25	$\overline{M_x}$	26187	t_m	.59
J_4	1.25	$\overline{M_y}$	26187	ΔT_2	4
J_5	N/A	N_a	2000	T_1	1
J_6	N/A	N_1	2	T_2	0.25
K_1	4.75	N_2	8	T_3	.438
K_2	2	N_3	4	T_4	1
K_3	3.4117	P_d	75	T_5	N/A
K_4	14.75	P_r	75	T_6	1
K_5	14.75	P_s	85	T_7	1
K_6	6.63	Q_{T1}	550	T_8	489099
L_1	20	Q_1	1.5	T_9	.438
L_2	1	Q_2	6.5	T_{10}	1
L_3	1.37	Q_3	8.5	T_{11}	1.75
L_4	1.62	Q_4	2.0	T_{12}	1
L_5	N/A	Q_5	2.688	U_1	19
L_6	.50	Q_6	9.926	U_2	19
L_7	N/A	Q_7	3.125	U_3	.25
L_8	.62	r	4.625	U_4	N/A
L_9	10.5	r_i	23.6875	U_5	N/A
L_{10}	3.50	r_2	.63	U_6	2438

V_1 N/A

V_2 N/A

V_3 N/A

V_4 N/A

V_5 1.283

V_6 4.632

V_7 9.368

V_8 12.716

W_1 6000

W_2 1300

W_3 700

W_4 115

W_6 .75

W_7 1045

W_8 36.5

X_0 0.0

Y_0 0.0

Z_0 3.813

Z_1 182

Z_2 72

Z_3 80.5

Z_4 80.5

Z_7 N/A

Z_8 .75

ANALYSIS INTRODUCTION

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

The air purge valve is designed in accordance with Code Case 1678 of the code.

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

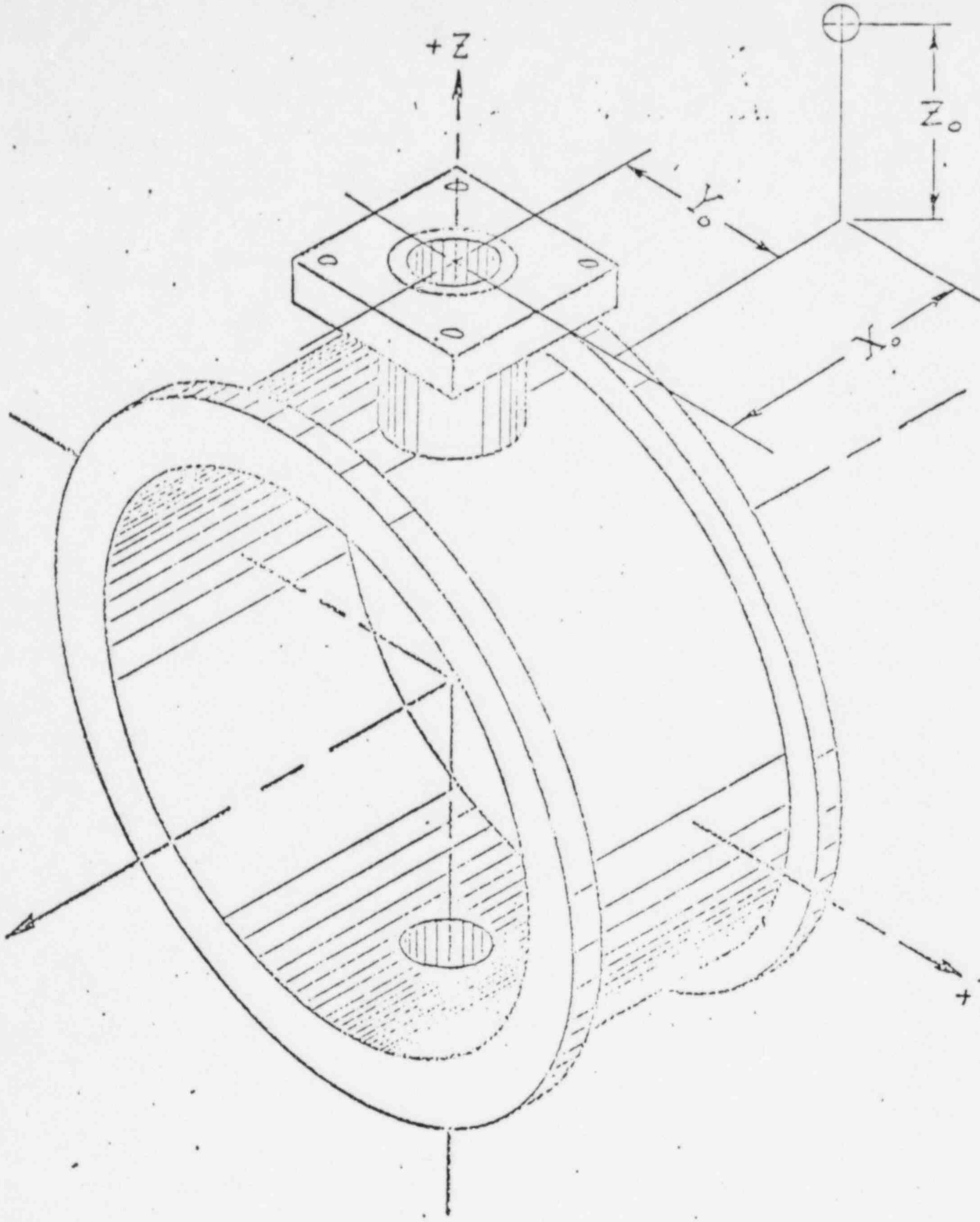
Seismic loads are made an integral part of the analysis by the inclusion of the acceleration constants g_x, g_y, g_z. The symbols g_x, g_y, g_z represent accelerations in the x, y and z directions respectively. These directions are defined with respect to the valve body centered coordinate system as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

Analysis Introduction

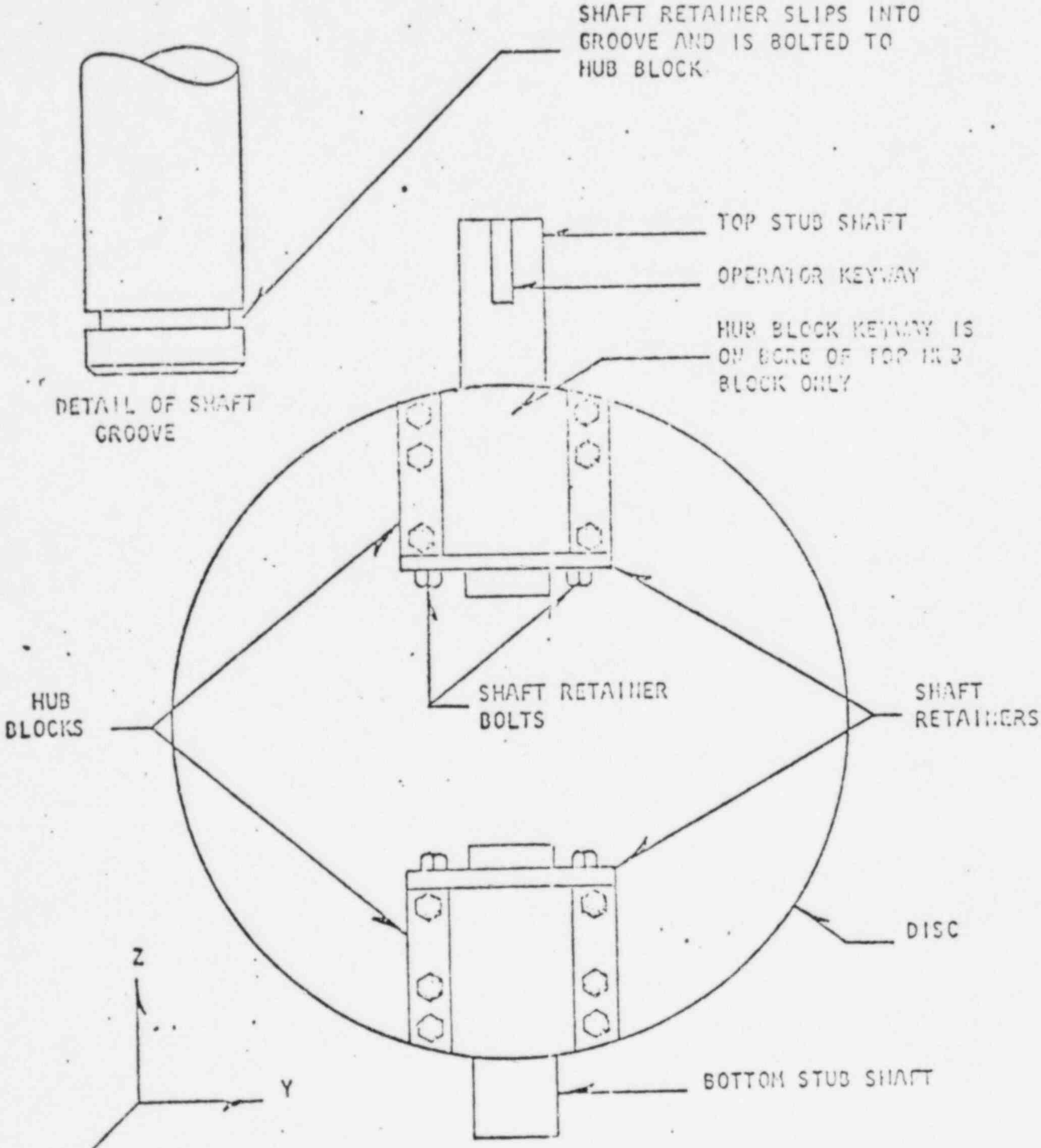
Valve orientation with respect to gravity is taken into account by adding the appropriate quantity to the seismic loads. The justification for doing this is that a gravitational load is completly equivalent to a 1g seismic load.

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

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


+Z
Z_o
Y_o
X_o
+X
+Y

OF BANJO ASSEMBLY



FLANGE ANALYSIS

The flange analysis is in accordance with appendix II, para. VA-56 of section VIII, division I of the ASME codes for pressure vessels and AWWA C-207.

BODY ANALYSIS

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

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

1. Minimum Body Wall Thickness

   Paragraph NB-3542 gives minimum body wall thickness requirements for standard pressure rated valves. The actual minimum wall thickness in the purge valve occurs behind the seat retaining screws.

2. Body Shape Rules

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

3. Primary Membrane Stress Due to Internal Pressure

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

The code defines the stresses in the crotch area using the pressure area method. The equation presented is found in paragraph NB-3545.1.

$$Pm = (Af/Am + .5)\ Ps$$

Applying the code rules to the crotch region results in a membrane stress considerable less than if applied to the region not containing the trunnion. The trunnion increases the metal area (Am) which decreases the Af/Am ratio and reduces the result. For a section not containing the trunnion, the fluid area to metal area ratio (Af/Am) reduces to the body inside radius to the shell thickness (Rm/Hg) since the depths are the same. The resulting membrane stress equation is then:

$$Pm = (Rm/Hg + .5)\ Ps$$

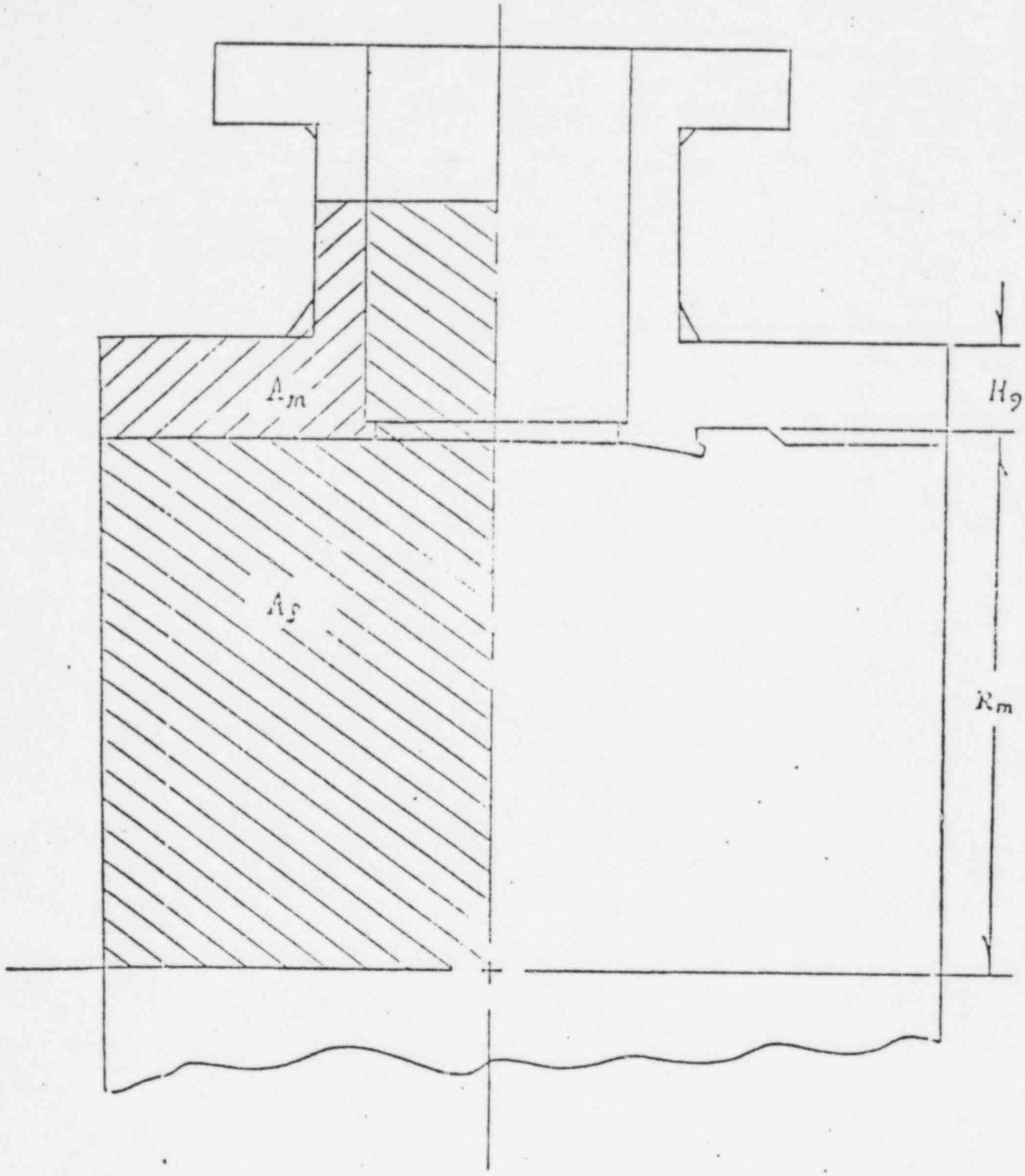
This equation results in the highest stressed area and complies with the intent of the code.

4. Secondary Stresses

A. Body Primary plus secondary stress due to internal pressure.

BODY CROSS-SECTION

Figure 3



Paragraph NB-3545.2 (a) of Section III of the code defines the formulas used in calculating this stress.

$$Q_p = C_p \quad \frac{ri}{t_e} + .5 \quad P_s$$

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

$$P_{ed} = \frac{F_d S}{G_d} = \text{Direct or axial load effect}$$

$$P_{eb} = \frac{C_b F_b S}{G_b} = \text{Bending load effect}$$

$$P_{et} = \frac{2F_b S}{G_t} = \text{Torsional load effect}$$

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

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

where

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

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

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

5. Valve fatigue requirements - Para. NB-3545.3 of Section III of the code defines requirements for normal duty valve fatigue. The allowable stress level is found from Figure I-9.0. Since the number of cycles is unknown, a maximum value of 2000 is assumed. The

allowable stress can then be found from Figure I-9.1 for carbon steel. This then gives an allowable stress of 65000 psi.

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

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

where:

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

DISC ANALYSIS

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

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

Primary Bending Stress

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

Combined bending stress in disc:

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

where

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

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

SHAFT ANALYSIS

The shaft is analyzed in accordance with para. NB-3546.3 of section III of the code. The shaft loading is a combination of seismic, pressure, and operating loads. Maximum torsional loading is either a combination of seating and bearing torque <u>or</u> bearing and dynamic torque. Columnar stress is not considered in the shaft loading due to its negligible effect on the stress levels. Figure 2 shows the banjo assembly with the stub shafts.

Shaft stresses due to pressure, seismic, and operating loads:

$$S(4) = \frac{S(5)}{2} + \frac{(S(5)^2 + 4\ S(6)^2)^{1/2}}{2}$$

Where:

$S(5) = (S(7)^2 + S(8)^2)^{1/2}$ = combined bending stress, PSI

$S(7) = \dfrac{(\pi R_4^2 P_s + W_2 g_x)\ .25\ B_1 R_5}{\pi\ .25\ R_5^4}$ = Bending tensile stress due to pressure & seismic loads along x - axis, PSI

$S(8) = \dfrac{.25\ W_2 g_y B_1 R_5}{.25\ \pi\ R_5 4}$ = Bending tensile stress due to seismic loads along y - axis, PSI

$S(6) = (S(9)^2 + S(10)^2)^{1/2}$ = combined shear stress, PSI

$S(9) = \dfrac{T_8 R_5}{.5\ \pi\ R_5^4}$ = Torsional shear stress, PSI

$S(10) = 1.333 \left[\dfrac{.5\pi R_4^2 P_s + .5 W_2 (g_x^2 + g_y^2)^{1/2}}{\pi R_5^2} \right]$ = Direct Shear Stress PSI

DISC PIN ANALYSIS

The valve assembly or cross-section drawing shows the two stub shafts and the disc pins. The top disc pins are subjected to torsional load as they transmit the operating torque.

Combined Shear Stress in Top Disc Pin:

$$S(20A) = \frac{T_8 - .5\ U_5}{2N_1\ R_5\ .785\ D_{12}{}^2}$$

Bearing Stress on Top Pins in Shaft:

$$S(20B) = \frac{T_8 - .5\ U_5}{(R_5 + .5\ K_2)\ 2K_2 D_{12} N_1}$$

Where

D_{12} = Disc Pin Diameter, in.

P_o = Actual Shut Off Pressure, psi.

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

$U_5 = U_4 + W_2 g \times U_3 R_5$

SHAFT BEARING ANALYSIS

The roller bearings in the trunnion are subjected to both seismic and pressure loads.

$$S(21) = \frac{\pi P_s R_4^2 + W_2 (g_x^2 + g_y^2)^{\frac{1}{2}}}{2} = \text{Compressive load on shaft bearing, lbs.}$$

THRUST BEARING ANALYSIS

As shown in figure 5, the thrust bearing assembly is located in the bottom trunnion. It provides restraint for the banjo in the z direction, assuring that the disc edge remains correctly positioned to maintain optimum sealing. Formulas used to analize the assembly are given below.

1. Bearing stress on thrust collar due to seismic load.

$$S(22) = \frac{W_2 \, g_z}{.785\ (D_4^{\,2} - D_{10}^{\,2})}$$

2. Shear stress in adjusting screw head due to seismic load.

$$S(23) = \frac{W_2 \, g_z}{\pi \, D_{10} \, T_2}$$

3. Combined stress in adjusting screw.

$$S(24) = \frac{S(25)}{2} + \frac{(S(25)^2 + 4\ S(26)^2)^{\frac{1}{2}}}{2}$$

Where:

$$S(25) = \frac{W_2 \, g_z}{A_2}$$ = Direct tensile stress due to seismic load.

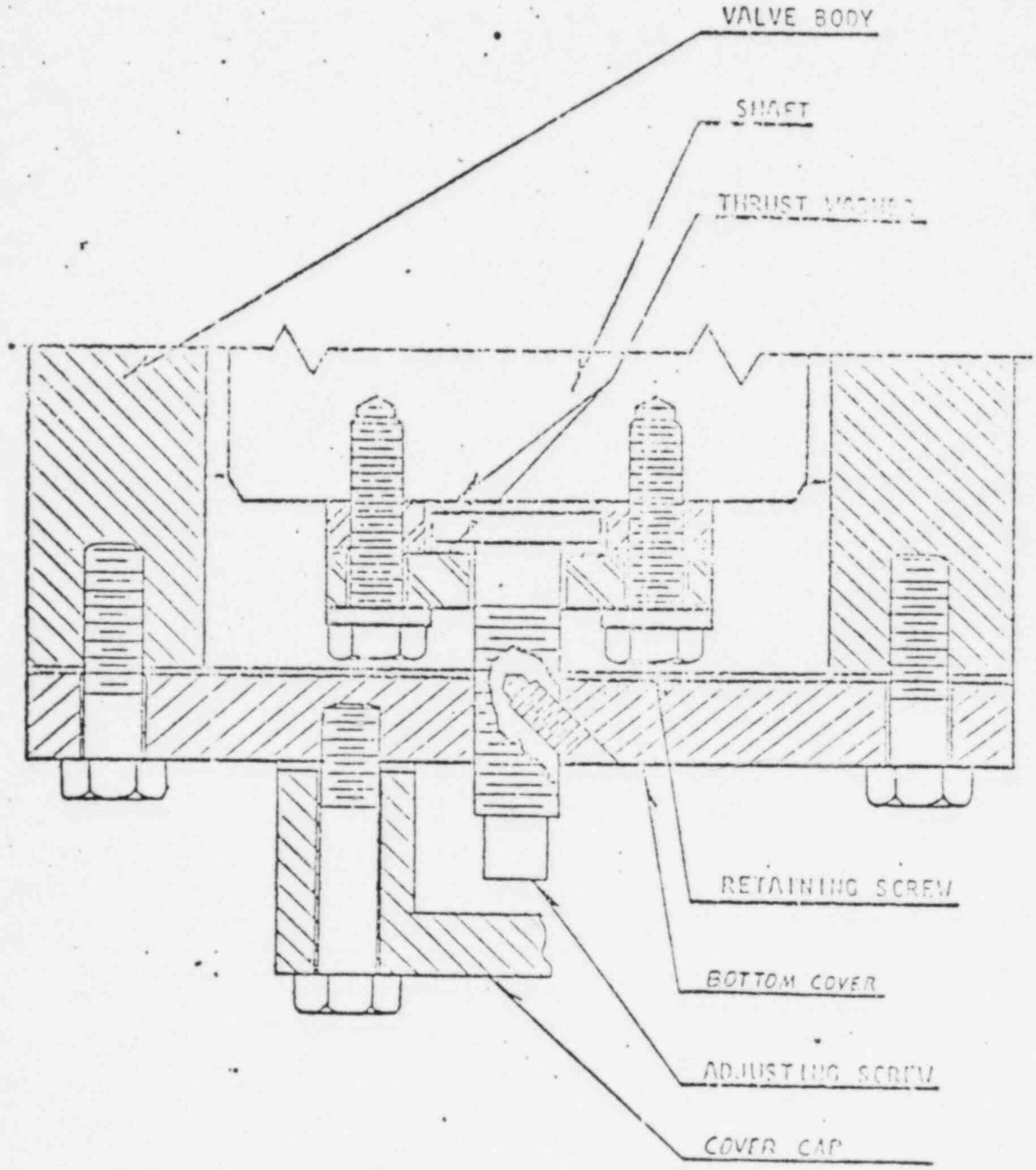
$$S(26) = \frac{16\ U_6}{\pi \, D_{10}^{\,3}}$$ = Torsional shear stress due to thrust bearing seismic friction torque.

$$U_6 = W_2 \, g_z \, U_3 \, (.5\ D_{10} + .5\ (D_4 - D_{10}))$$

4. Shear stress in adjusting screw threads due to seismic loads.

$$S(27) = \frac{W_2 \, g_z}{.9\ \pi \, D_{10} \, T_{10}}$$

ESSENTIAL FEATURES OF
THRUST BEARING ASSEMBLY



5. Combined stress in retainer bolts due to seismic loads.

$$S(28) = \frac{S(29)}{2} + \frac{\left(S(29)^2 + 4\ S(30)^2\right)^{\frac{1}{2}}}{2}$$

Where:

$$S(29) = \frac{W_2\ g_z}{6\ A_{13}} = \text{Tensile stress due to seismic load.}$$

$$S(30) = \frac{U_6}{6\ R_7\ A_{12}} = \text{Shear stress due to seismic load.}$$

6. Shear tear out of thrust bearing retainer bolts.

$$S(31) = \frac{W_2\ g_z}{6\ \pi\ D_5}$$

BOTTOM COVER ANALYSIS

Figure 5 shows the bottom trunnion assembly, including the bottom cover and bottom cover bolts.

1. Bottom Cover Bolt Stresses

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

   Shear tear out of bolts through tapped holes in trunnion:

   $$S(33) = \frac{W_2\, g_z + \pi\, P_s\, R_6^{\,2}}{4\, L_3\, (2.83)\, D_6}$$

   Shear tear out of bolt heads through bottom cover, PSI.

   $$S(34) = \frac{W_2\, g_z + \pi\, P_s\, R_6^{\,2}}{4\, T_1\, (5.2)\, D_6}$$

   Combined stress in bolts, PSI

   $$S(35) = \frac{S(37)}{2} + \frac{(S(37)^2 + 4\, S(36)^2)^{\frac{1}{2}}}{2}$$

   Where:

   $$S(36) = \frac{U_6}{.707\, H_3\, 4\, A_4}$$ = Shear stress in bolts due to torsional load.

   $$S(37) = \frac{W_2\, g_z + \pi\, P_s\, R_6^{\,2}}{4\, A_3}$$ = Tensile stress in bolts due to seismic and pressure loads, PSI.

2. Bottom Cover Stresses

   The combined stress in the bottom cover is calculated using the following formulas:

   $$S(38) = \frac{S(39) + S(40)}{2} + \frac{(\,(S(39) + S(40))^2 + 4\, S(41)^2)^{\frac{1}{2}}}{2}$$

Where:

$$S(39) = \frac{3(.785)(D_4 + .25)^2 P_s + W_2 g_z}{4 \pi T_4^2} = \text{Radial stress}$$

$$S(40) = \frac{3\ (.785)(D_4 + .25)^2 P_s + W_2 g_z}{4 \pi \dot{T}_4^2 m} = \text{Tangential stress}$$

$$S(41) = \frac{.785\ (D_4 + .25)^2 P_s + W_2 g_z}{\pi (D_4 + .25) T_4} = \text{Shear stress}$$

OPERATOR MOUNTING ANALYSIS

The operator mounting consists of the top trunnion, the bonnet, the operator housing and the bolt connections as shown in Fig.

1. Bolt Stresses and Localized Stress Due to Bolt Loads.

The following assumptions are used in the development of the equations:

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

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

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

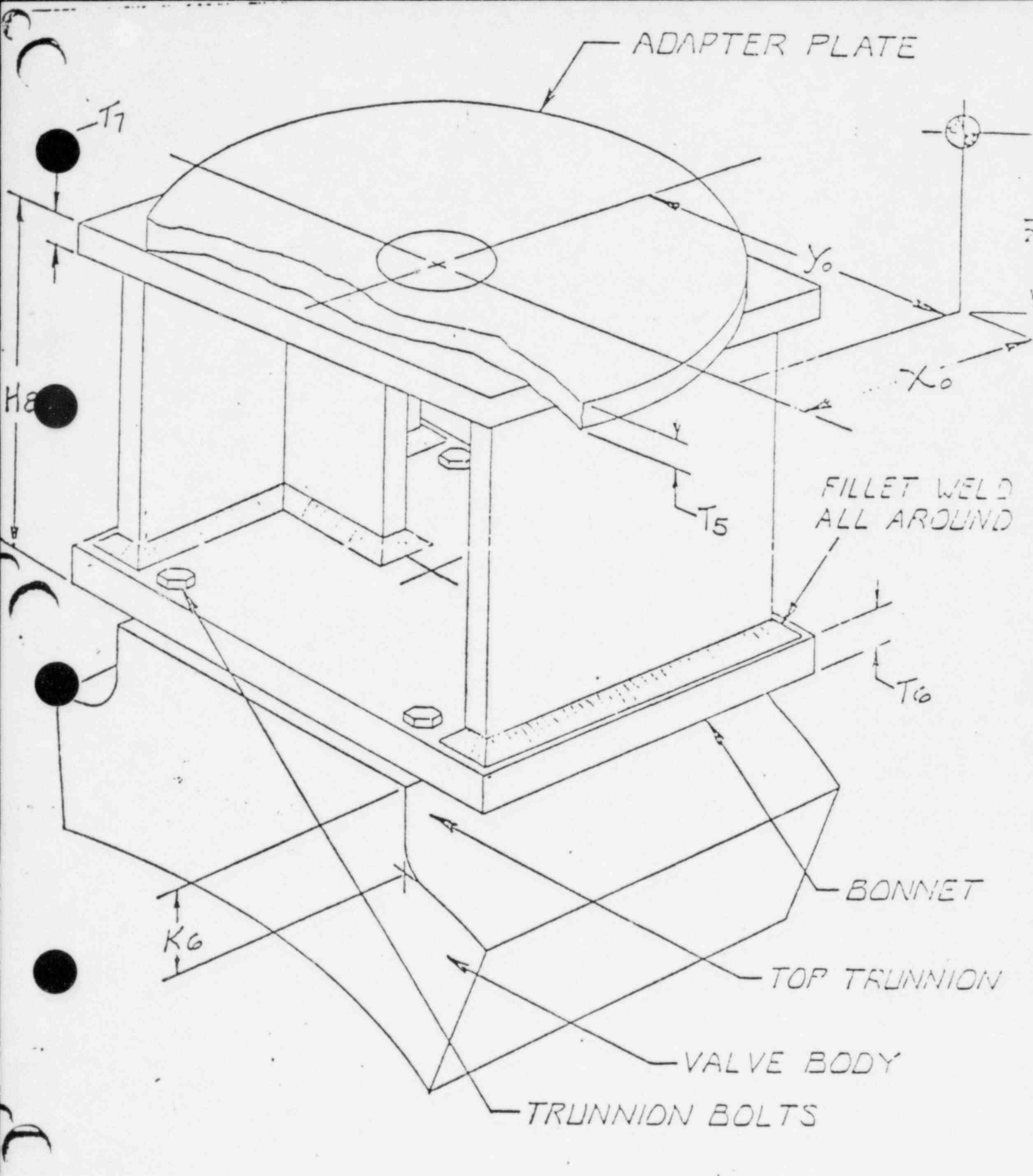
$$S(42) = \frac{\dfrac{F_z+W_4\sqrt{g_x^{\,2}+g_y^{\,2}+g_z^{\,2}}}{4} + \dfrac{\overline{\overline{M_x}}(J_2+H_2)}{2J_2^{\,2}+2(J_2+H_2)^2} + \dfrac{\overline{\overline{M_y}}(J_1+H_2)}{2J_1^{\,2}+2(J_1+H_2)^2}}{.9\pi L_4 D_7}$$

b. Bearing stress on tapped holes in trunnion.

$$S(43) = \frac{\dfrac{(M_z+T_8)}{4(.707H_2)} + \dfrac{(F_x^{\,2}+F_y^{\,2})^{1/2}}{4} + \dfrac{W_4(g_x^{\,2}+g_y^{\,2})^{1/2}}{4}}{D_7L_4}$$

c. Bearing stress on through hole in bonnet.

$$S(44) = \frac{\dfrac{M_z+T_8}{4(.707H_2)} + \dfrac{(F_x^{\,2}+F_y^{\,2})^{1/2}}{4} + \dfrac{W_4(g_x^{\,2}+g_y^{\,2})^{1/2}}{4}}{D_7T_6}$$



TOP TRUNNION MOUNTING

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

$$S(45) = \frac{\dfrac{F_z+W_4g_z}{4} + \dfrac{\overline{\overline{M_x}}(J_2+H_2)}{2J_2^{\,2}+2(J_2+H_2)^2} + \dfrac{\overline{\overline{M_y}}(J_1+H_2)}{2J_1^{\,2}+2(J_1+H_2)^2}}{5.2\ D_7T_6}$$

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

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

Where,

$$S(47) = \frac{F_z + W_4g_z}{4A_5} \quad = \text{ Direct tensile stress, psi}$$

$$S(48) = \frac{\overline{\overline{M_x}}(J_2+H_2)}{2J_2^{\,2}+2(J_2+H_2)^2} + \frac{\overline{\overline{M_y}}(J_1+H_2)}{2J_1^{\,2}+2(J_1+H_2)^2} \quad = \text{ Tensile stress due to extended mass bending moment, psi}$$

$$S(49) = \frac{(F_x^{\,2}+F_y^{\,2})^{\frac{1}{2}} + W_4(g_x^{\,2}+g_y^{\,2})^{\frac{1}{2}}}{4A_6} \quad = \text{ Direct shear stress, psi}$$

$$S(50) = \frac{M_z+T_8}{(.707H_2)4A_6} \quad = \text{ Shear stress due to torsional load, psi}$$

f. Shear tearout of operator bolt head through bonnet.

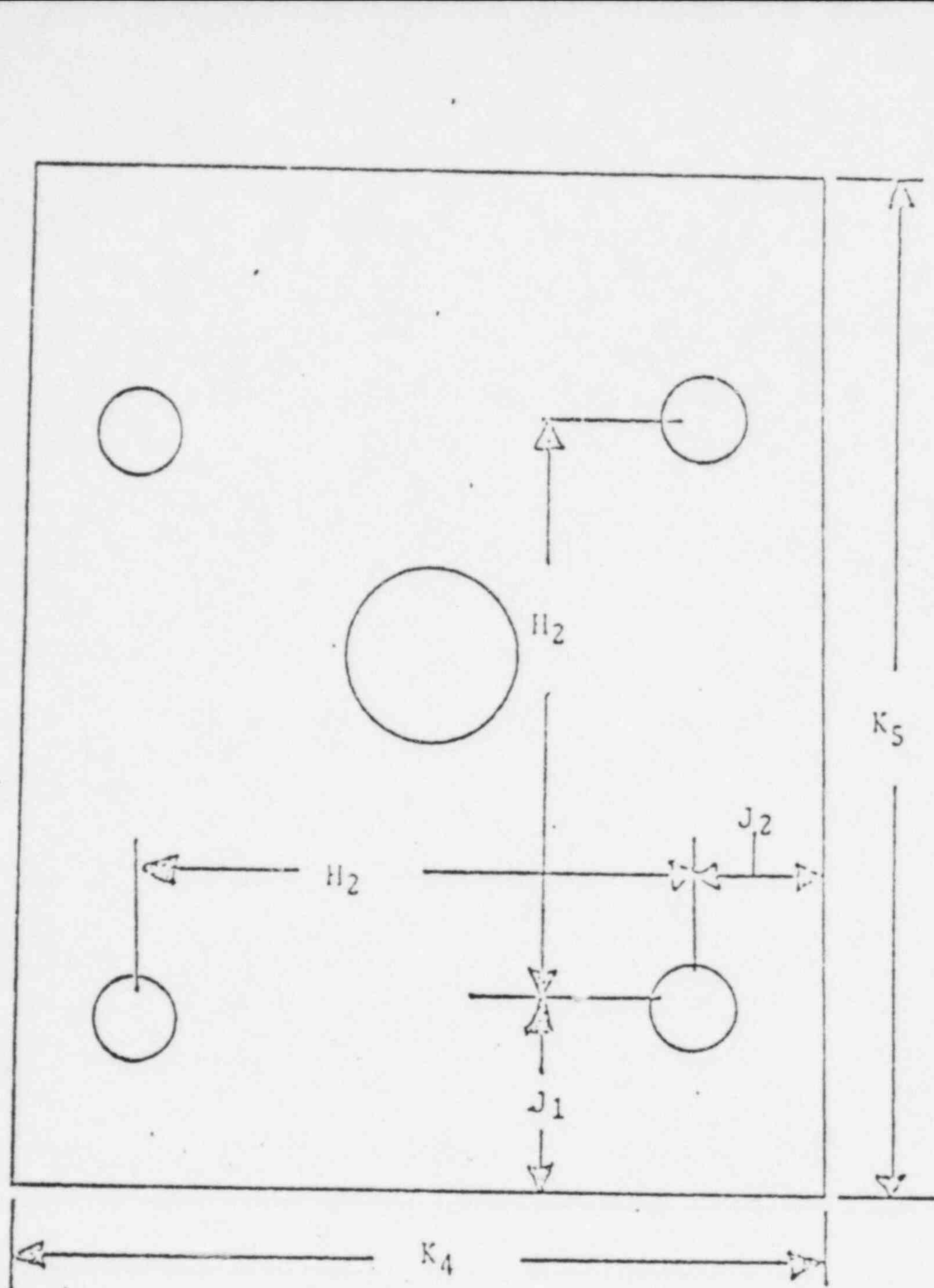
$$S(51) = \frac{\dfrac{(M_x+M_y)V_4}{2(V_1^{\,2}+V_2^{\,2}+V_3^{\,2}+V_4^{\,2})} + \dfrac{F_z}{4}}{5.2\ D_8T_7}$$

g. Bearing stress on through holes in bonnet.

$$S(52) = \frac{M_z+T_8}{.5H_78T_7D_8}$$

h. Combined stress in operator bolts (Fig. 10)

$$S(53) = \frac{S(54)+S(55)}{2} + \frac{((S(54)+S(55))^2 + 4(S(56)+S(57))^2)^{\frac{1}{2}}}{2}$$



TOP TRUNNION BOLTING

Figure 6

Where,

$$S(54) = \frac{F_z}{4A_7} = \text{ Direct tensile stress, psi.}$$

$$S(55) = \frac{(M_x+M_y)\ V_4}{2(V_1{}^2+V_2{}^2+V_3{}^2+V_4{}^2)A_7} = \text{ Tensile stress due to bending moment, psi.}$$

$$S(56) = \frac{(F_x{}^2+F_y{}^2)^{1/2}}{4A_8} = \text{ Direct shear stress, psi}$$

$$S(57) = \frac{M_z+T_8}{.5H_7 8A_8} = \text{ Shear stress due to torsion, psi}$$

BONNET STRESSES

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

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

$$S(58) = \frac{S(59)+S(60)}{2} + \frac{((S(59)+S(60))^2 + 4(S(61)+S(62))^2)^{1/2}}{2}$$

= Combined stress in bonnet legs.

Where,

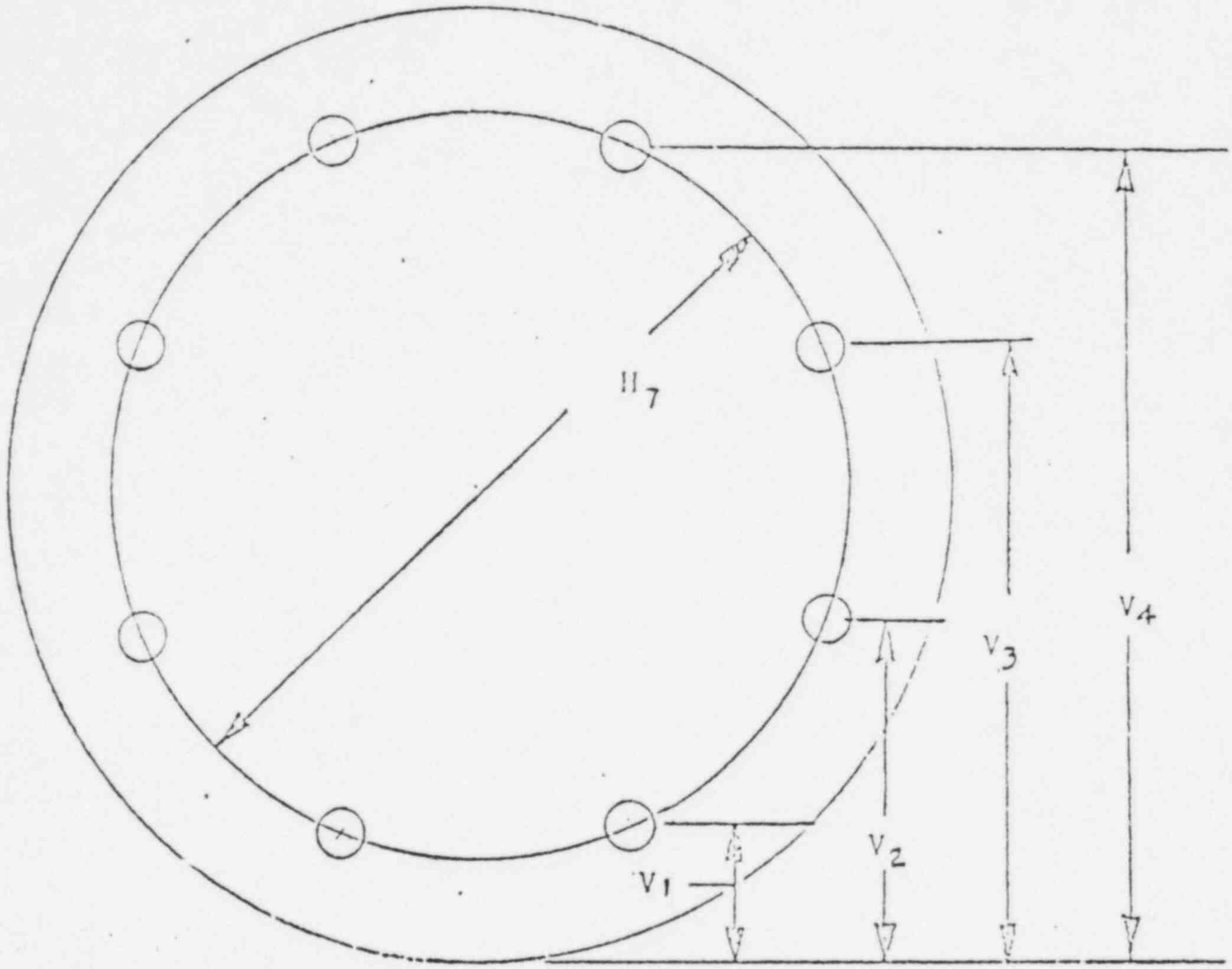
$$S(59) = \frac{F_z+W_4g_z}{B_5} = \text{ Direct tensile stress, psi}$$

$$S(60) = \frac{\overline{\overline{M}}_x B_8}{I_1} + \frac{\overline{\overline{M}}_y B_9}{I_2} = \text{ Tensile stress due to bending moment, psi}$$

$$S(61) = \frac{(F_x{}^2+F_y{}^2)^{1/2} + W_4(g_x{}^2+g_y{}^2)^{1/2}}{B_5} = \text{ Direct shear stress, psi}$$

$$S(62) = \frac{T_8C_0}{K_0} = \text{ Shear stress in bonnet body due to torsional load, psi}$$

Y



OPERATOR BOLT PATTERN

Figure 8

Where,

T_8 = Operating torque, in-lbs.

C_0 = Torsional constant for non-circular cross-section

K_0 = Function of cross-section, in^4.

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

BOTTOM BONNET WELDS

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

Where,

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

$$S(66) = \frac{F_z + W_4 g_z}{U_1}$$ = Direct tensile stress, psi

$$S(67) = \frac{\overline{\overline{M}}_x}{Z_1} + \frac{\overline{\overline{M}}_y}{Z_2}$$ = Bending tensile stress, psi

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

$$S(68) = \frac{(F_x{}^2 + F_y{}^2)^{1/2} + W_4(g_x{}^2 + g_y{}^2)^{1/2}}{U_1}$$ = Direct shear stress, psi

$$S(69) = \frac{M_z + T_8}{Z_3}$$ = Torsional shear stress, psi

TOP BONNET WELDS

$$S(70) = \frac{((S(71))^2 + 4(S(72))^2)^{1/2}}{2}$$ = Combined shear stress in top bonnet weld, psi

Where,

$S(71) = S(73) + S(74)$

$$S(73) = \frac{F_z}{U_2} = \text{Direct tensile stress, psi}$$

$$S(74) = \frac{\overline{M_x}}{Z_1} + \frac{\overline{M_y}}{Z_2} = \text{Bending tensile stress, psi}$$

$$S(72) = S(75) + S(76) = \text{Total shear stress, psi}$$

$$S(75) = (F_x^{\,2} + F_y^{\,2})^{1/2} = \text{Direct shear stress, psi}$$

$$S(76) = \frac{M_z + T_8}{Z_4} = \text{Torsional shear stress, psi}$$

TRUNNION BODY STRESS

The trunnion body stresses are calculated using the following assumptions.

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

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

$$S(77) = \frac{S(78)+S(79)}{2} + \frac{((S(78)+S(79))^2 + 4(S(80)+S(81))^2)^{1/2}}{2}$$

Where,

$$S(78) = \frac{F_z + W_4 g_z}{K_4 K_5 - .785 B_2^{\,2}} = \text{Direct tensile stress, psi}$$

$$S(79) = \frac{(M_x+F_y K_6)\,.5K_4}{.0833K_5K_4^{\,3} - \frac{\pi B_2^{\,4}}{64}} + \frac{(M_y+F_x K_6)\ .5K_5}{.0833K_4K_5^{\,3} - \frac{\pi B_2^{\,4}}{64}} = \text{Bending tensile stress, psi}$$

$$S(80) = \frac{(F_x^{\,2}+F_y^{\,2})^{\frac{1}{2}} + W_4(g_x^{\,2}+g_y^{\,2})^{\frac{1}{2}}}{K_4K_5-.785B_2^{\,2}} = \text{Direct shear stress, psi}$$

$$S(81) = \frac{(M_z+T_8)\ \ .5(K_4^{\,2}+K_5^{\,2})^{\frac{1}{2}}}{.0833(K_4K_5^{\,3} + K_5K_4^{\,3})-\frac{\pi B_2^{\,4}}{32}} = \text{Torsional shear stress, psi}$$

The top trunnion assembly consists of the top trunnion plate, the top trunnion, the welds and the body material immediately adjacent to the trunnion. Fig. 10 illustrates the elements of the assembly.

1. Combined shear stress in the top trunnion plate welds is a maximum due to seismic and torsional loads.

$$S(77) = (S(78)^2 + S(79)^2)^{1/2}$$

Where,

$$S(78) = \frac{4(\overline{\overline{M_x}}^2+\overline{\overline{M_y}}^2)^{1/2}}{.707(.5)\pi D_{11}^2} + \frac{F_z}{\pi D_{11}L_{11}}$$ = Shear stress due to operator eccentricity

$$S(79) = \frac{4(M_z+T_8)(3D_{11}+2T_{11})}{3(1.41)\pi L_{11}(D_{11}+2T_{11})^3}$$ = Torsional shear due to operator eccentricity and operator torque

2. Combined stress in base of trunnion body due to combined bending, torsion and seismic loads.

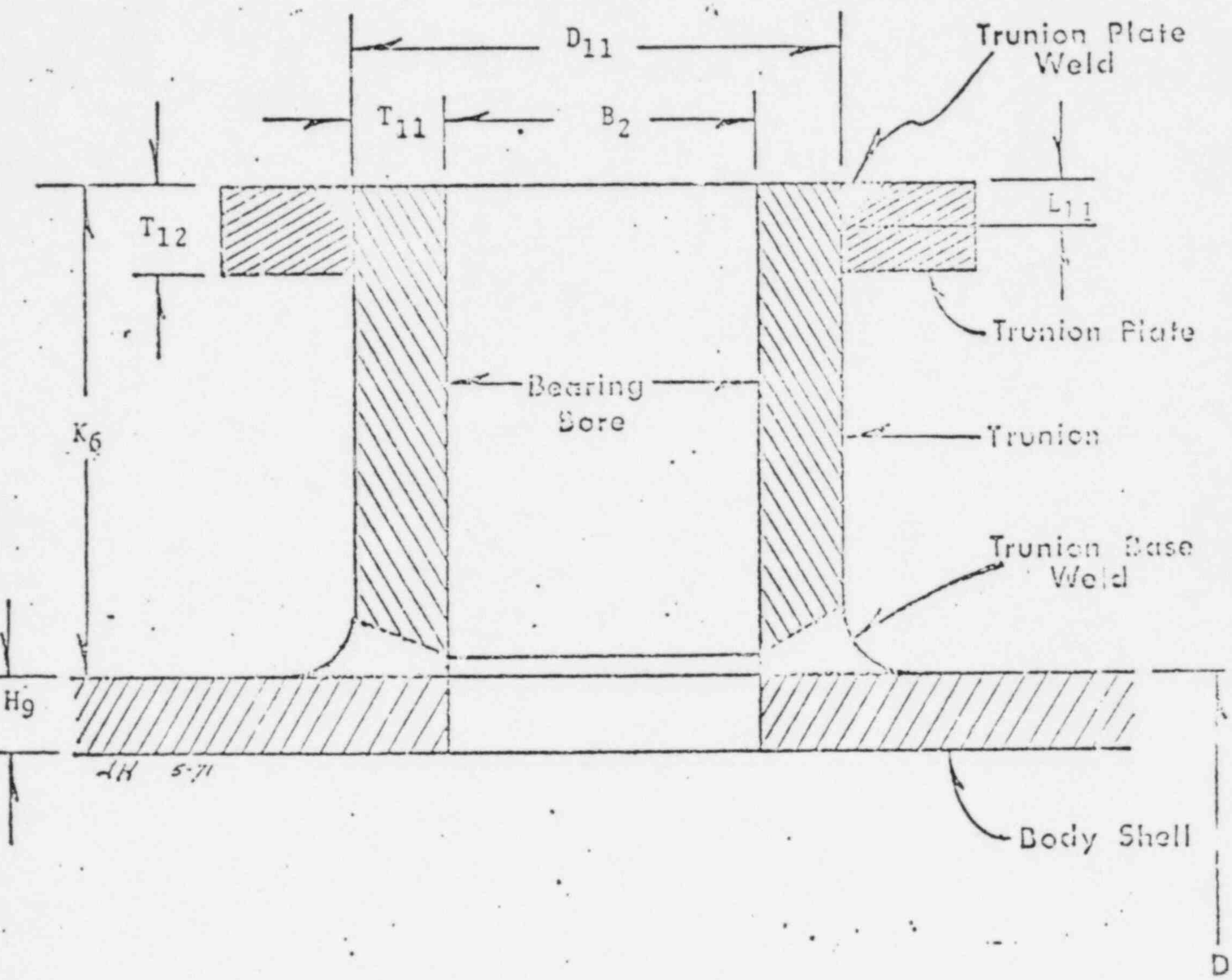
$$S(80) = \frac{(S81)+S(82)}{2} + \frac{(S(81)+S(82))^2 + 4(S(83)+S(84))^2)^{1/2}}{2}$$

Where,

$$S(81) = \frac{F_z+W_4g_z}{.25\pi(D_{11}^2-B_2^2)}$$ = Direct tensile stress, psi.

$$S(82) = \frac{32((\overline{\overline{M_x}} + F_yK_6)^2 + (\overline{\overline{M_y}} + F_xK_6)^2)^{1/2} D_{11}}{\pi(D_{11}^4-B_2^4)}$$

= Bending tensile stress, psi



D_{11}
T_{11}
B_2
T_{12}
L_{11}
K_6
H_9
D
Trunion Plate Weld
Trunion Plate
Bearing Bore
Trunion
Trunion Base Weld
Body Shell
5-71

$$S(83) = \frac{(F_x^2+F_y^2)^{\frac{1}{2}} + W_4(g_x^2+g_y^2)^{\frac{1}{2}}}{.25\pi(D_{11}^2-B_2^2)}$$ = Direct shear stress, psi

$$S(84) = \frac{16(M_z+T_8)\ D_{11}}{\pi(D_{11}^4-B_2^4)}$$ = Torsional shear stress, psi

3. Combined shear stress in top trunnion to shell weld is a maximum due to seismic and torsional loads.

$$S(85) = (S(86)^2 + S(87)^2)^{\frac{1}{2}}$$

Where,

$$S(86) = \frac{4((\overline{\overline{M_x}} + F_yK_6)^2 + (\overline{\overline{M_y}} + F_xK_6)^2)^{\frac{1}{2}}}{.707(.5)\pi D_{11}^2} + \frac{F_z}{\pi D_{11}L_{11}}$$

= Shear stress due to operator eccentricity

$$S(87) = \frac{4(M_z+T_8)(3D_{11}+2T_{11})}{3(1.41)\pi L_{11}(D_{11}+2T_{11})^3}$$ = Torsional shear due to operator eccentricity and operating torque

FREQUENCY ANALYSIS

A. Introduction

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

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

or

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

or

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

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

B. Valve Body Assembly

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

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

Where

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

C. Banjo Assembly

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

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

Where

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

D. Cover Cap Assembly

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

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

Where

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

E. Bonnet Assembly

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

Frequency Analysis

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

Natural frequency of bonnet:

$$F_{N4} = \left(\frac{9.8}{\Delta y_4}\right)^{\frac{1}{2}}$$

Where

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

ATTACHMENT 3

SUPPLEMENTAL TORQUE CALCULATIONS

ATTACHMENT 3

The following pages illustrate the combined effects of disc blockage and delay time on dynamic torque. In each case, the delay time is fixed at that which produced the worst case torque for the full open, unblocked condition. The initial disc angle is reduced by blocking to illustrate the resultants of several different initial angles of opening.

D-27256-2 TORQUE TABLE 2 9 / 14 / 81

JOB:FLOR.PWR:TURKEY-PT 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.)= 1371.29 FEET/SEC AT 283 DEG.

CRIT.CASE INLET VELOCITY IS 1.57264 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 .692271 (47.375 IN)WITH STMIX.)FIRST SONIC(@ 72 DEG.V.A.)
ABSOL.MAX.TORQUE(FIRST SONIC)AT 72-68 DG.VLV.ANG.= 179419 IN-LBS @ 35 DEG.
MAX.TORQUE INCLUDES SIZE EFFECT(REYNOLDS NO.ETC)APPX. X 1.23418 FOR 47.375 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:	48"-R1A5:3/7.5	CLASS 75
DISC SIZE:	47.062 INCHES	SYMMETRICAL DISC
SHAFT DIA.:	4 INCHES	
BRG. COEF. OF FRCTN.:	5.00000E-03	
SEATING FACTOR:	15	
INLET PRESS.VAR.MAX.:	51.191 PSIA	
OUTLET PRESSURE(P6):	33.93 PSIA (72 DEG. ACTUAL PRESS.ONLY(VAR.))	
MAX.ANG.FLOW RATE:	108209. CFM: 214793. SCFM: 11807.7 LB/MIN	
CRIT.SONIC FLOW-90DG:	57551. LB/MIN AT 31.6857 INLET PSIA	
VALVE INLET DENSITY:	.10912 LB/FT^3-MIN. .133956 LB/FT^3-MAX.	
FULL OPEN DELTA P:	13.4092 PSI	

SYSTEM CONDITIONS:
PIPE IN-PIPE-OUT -AND- AIR/STEAM MIXTURE SERVICE @ 283 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 TO. 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.
35	41.70	22.36	19.34	.536	214792	11807	51642	1586	50055	3.09
30	44.03	15.04	28.99	.342	172255	9469	47548	1716	45831	3.28
25	45.55	14.89	30.66	.327	122772	6749	30708	1759	28949	3.68
20	46.34	14.78	31.56	.319	75817	4167	21406	1808	19597	3.85
15	46.59	14.72	31.88	.316	42217	2320	8815	1864	6950	3.90
10	47.69	14.71	32.98	.308	21243	1167	6323	1947	4376	4.07
5	49.50	14.70	34.80	.297	6992	384	3823	2019	1804	4.37
0	51.19	14.70	36.49	.287	0	0	35650	1984	33665	4.75

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37635 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 51642 IN-LBS @ 35 DEG.

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 40 DEG.

MAX.ANG.FLOW RATE: 157316 CFM: 312270. SCFM: 17166.4 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37674 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 91788 IN-LBS @ 35 DEG.

AT 3 SEC.DELAY TIME TO 5 CLOSED VLV.(LOCA)TIME(41.7 TO 52.3009 PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.32458
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.45002

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 45 DEG.

MAX.ANG.FLOW RATE: 219543. CFM: 435791. SCFM: 23956.6 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37705 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 129383 IN-LBS @ 40 DEG.

AT 3 SEC.DELAY TIME TO 5.25 CLOSED VLV.(LOCA)TIME(41.7 TO 53.2 PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.29674
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.41954

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 50 DEG.

MAX.ANG.FLOW RATE: 213950. CFM: 424688. SCFM: 23346.3 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37736 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 176303 IN-LBS @ 45 DEG.

AT 3 SEC.DELAY TIME TO 5.5 CLOSED VLV.(LOCA)TIME(41.7 TO 54.0939 PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.26153
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.381

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 55 DEG.

MAX.ANG.FLOW RATE: 264052 CFM: 524140. SCFM: 28813.4 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37766 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 200052 IN-LBS @ 50 DEG.

AT 3 SEC.DELAY TIME TO 5.75 CLOSED VLV.(LOCA)TIME(41.7 TO 54.9701 PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.26097
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.38038

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 60 DEG.

MAX.ANG.FLOW RATE: 318087. CFM; 631399 SCFM; 34709.7 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37796 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 275702 IN-LBS @ 55 DEG.

AT 3 SEC.DELAY TIME TO 6 CLOSED VLV.(LOCA)TIME(41.7 TO 55.8265 PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.24212
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.35975

****

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 65 DEG.

MAX.ANG.FLOW RATE: 379236. CFM; 752778. SCFM; 41382.3 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37825 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 357203 IN-LBS @ 65 DEG.

AT 3 SEC.DELAY TIME TO 6.25 CLOSED VLV.(LOCA)TIME(41.7 TO 56.6602 PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.21566
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.33078

****

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 70 DEG.

MAX.ANG.FLOW RATE: 430453. CFM; 854443. SCFM; 46971.1 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37853 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 390847 IN-LBS @ 65 DEG.

AT 3 SEC.DELAY TIME TO 6.5 CLOSED VLV.(LOCA)TIME(41.7 TO 57.4675 PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.21019
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.3248

****

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 75 DEG.

MAX.ANG.FLOW RATE: 494335. CFM: 981249. SCFM: 53942. LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37880 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 430363 IN-LBS @ 70 DEG.

AT 3 SEC.DELAY TIME TO 6.75 CLOSED VLV.(LOCA)TIME(41.7 TO 58.2428 P
SIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.19917
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.31273

****

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 80 DEG.

MAX.ANG.FLOW RATE: 518135. CFM: 1028492 SCFM: 56539. LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37906 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 449227 IN-LBS @ 70 DEG.

AT 3 SEC.DELAY TIME TO 7 CLOSED VLV.(LOCA)TIME(41.7 TO 58.9771 PSIA
UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.19521
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.3084

****

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 85 DEG.

MAX.ANG.FLOW RATE: 547259. CFM: 1086302 SCFM: 59717.1 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 37929 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 463889 IN-LBS @ 70 DEG.

AT 3 SEC.DELAY TIME TO 7.25 CLOSED VLV.(LOCA)TIME(41.7 TO 59.6528 P
SIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.19221
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.30511

ATTACHMENT 4

GENERAL ARRANGEMENT

AND CROSS-SECTION DRAWINGS