

ATTACHMENT A

Turkey Point Unit No. 4

Section I	Valve Test Program
Section II	Pump Test Program

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ABSTRACT

The planned inservice inspection and testing programs were developed employing the R.G. 1.26, Revision 2, criteria for quality group classifications and standards (Quality Group A is the same as ASME Class 1, etc.).

Section I. Valve Test Program Outline

The valve test program shall be conducted in accordance with Subsection IWV of Section XI of the 1980 Edition of the ASME Boiler and Pressure Vessel Code thru Winter 1980 Addenda, except for specific relief requested in accordance with 10 CFR 50.55a (g) (5) (iii) which is identified in Subsections I.G. The period for this valve test program starts September 7, 1983, and ends September 7, 1993.

Section II. Pump Test Program Outline

The pump test program shall be conducted in accordance with Subsection IWP of Section XI of the 1980 Edition of the ASME Boiler and Pressure Vessel Code thru Winter 1980 Addenda, except for specific relief requested in accordance with 10 CFR 50.55a (g) (5) (iii) which is identified in Subsection II.A. The period for the pump test program starts September 7, 1983, and ends September 7, 1993.



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Section II.	Pump Test Program
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I.A. LIST OF DRAWINGS

FPL NO. F-503184, REV. 7
(5610-M-420-3, REV. 8)

CHEMICAL & VOLUME CONTROL
SYSTEM (3 SHEETS)

FPL NO. E-503185, REV. 7
(5610-M-470-5, REV. 8)

SAFETY INJECTION SYSTEM
(4 SHEETS)

FPL NO. F-503187, REV. 7
(5610-M-450-57, REV. 9)

AUXILIARY COOLANT SYSTEM
COMPONENT COOLING (3 SHEETS)

FPL NO. F-503188, REV. 7
(5610-M-450-57, REV. 9)

AUXILIARY COOLANT SYSTEM
COMPONENT COOLING (4 SHEETS)

FPL NO. F-503189, REV. 8
(5610-M-500-28, REV. 6)

WASTE DISPOSAL SYSTEM
(2 SHEETS)

FPL NO. F-503191, REV. 6
(5610-M-410-91, REV. 7)

REACTOR COOLANT SYSTEM
(2 SHEETS)

FPL NO. F-503193, REV. 7
(5610-M-480-1, REV. 9)

SAMPLING SYSTEM

FPL NO. F-503194, REV. 6
(5610-M-450-53, REV. 8)

AUXILIARY COOLANT SYSTEM
RESIDUAL HEAT REMOVAL

FPL NO. F-503195, REV. 4
(5610-M-450-54, REV. 6)

AUXILIARY COOLANT SYSTEM
SPENT FUEL PIT COOLING SYSTEM

FPL NO. F-502027, REV. 14
(5610-M-1, REV. 14)

STEAM SYSTEM

FPL NO. F-502028, REV. 15
(5610-M-2, REV. 18)

CONDENSATE AND FEEDWATER
SYSTEMS

FPL NO. F-502030, REV. 14
(5610-M-4, REV. 18)

LUBE OIL SERVICE AND
INSTRUMENT AIR

FPL NO. F-502031, REV. 14
(5610-M-5, REV. 17)

CIRCULATING WATER, SALT
WATER AND CHLORINATION SYSTEMS
(INTAKE COOLING WATER SYSTEM)

FPL NO. F-502033, REV. 8
(5610-M-7, REV. 10)

DIESEL OIL

FPL NO. F-502036, REV. 14
(5610-M-10, REV. 18)

FIRE, PRIMARY MAKE-UP, CONTAINMENT
COOLING WATER AND CHEMICAL
INJECTION SYSTEMS

FPL NO. F-502037, REV. 10
(5610-M-11, REV. 11)

CONTAINMENT VENTILATION
SYSTEM



I.A. LIST OF DRAWINGS (CONT'D)

FPL NO. F-502038, REV. 10
(5610-M-12, REV. 9)

CONTAINMENT AND RADWASTE
DRAINS AND VENTS

(5177-074-M-2, REV. 6)

BREATHING AIR SYSTEMS

NOTE: Architect Engineer drawing number shown in parentheses.



I.B. LIST OF VALVE CATEGORIES (IWV-2200)

- Category A - Valves for which seat leakage is limited to a specific maximum amount in the closed position for fulfillment of their function.
- Category B - Valves for which seat leakage in the closed position is inconsequential for fulfillment of their function.
- Category C - Valves which are self-actuating in response to some system characteristic, such as pressure (relief valves) or flow direction (check valves).
- Category D - Valves which are actuated by an energy source capable of only one operation, such as rupture disks or explosive-actuated valves.

NOTE: When more than one distinguishing category characteristic is applicable, all requirements of each of the individual categories are applicable, although duplication or repetition of common testing requirements is not necessary.



I.C. LIST OF DEFINITIONS

1. ACTIVE AND PASSIVE VALVES (IWV-2100)

- (a) ACTIVE VALVES - Valves which are required to change position to accomplish a specific function.
- (b) PASSIVE VALVES - Valves which are not required to change position to accomplish a specific function.

2. EXERCISING - Exercising is the demonstration, based on direct or indirect visual or other positive indication, that the valve exhibits the required change of disk position to fulfill its function.

- (a) FULL STROKE - is the valve stem or disc movement to the position required (to open or to close) to fulfill its function.
- (b) NORMALLY OPEN VALVE - verification of seating upon cessation or reversal of flow.
- (c) NORMALLY CLOSED VALVE - verification of opening upon cessation of pressure differential or initiation of flow or by mechanical force.



I.D. LIST OF VALVE TABLE SYMBOLS

VALVE NUMBER:

IDENTIFICATION

SIZE:

INCHES

TYPE:

GATE-GLOBE-CHECK-RELIEF-BALL

SAFE - SAFETY
DIAPH - DIAPHRAGM
BUTFY - BUTTERFLY
S/CHK - STOP CHECK
POWER - POWER ASSISTED CHECK
ASS'T
CHECK

ACTUATOR:

A/O - AIR OPERATOR
MO - ELECTRIC MOTOR OPERATOR
SO - SOLENOID OPERATOR
S/A, SA,
OR SELF - SELF ACTUATED

CODE CLASS:

1 - 2 - 3

CODE CATEGORY:

A - B - C (OR COMBINATION OF) SEE
TABLE A BELOW AND SUBSECTION I.B.

ACTIVE/PASSIVE:

A - P (SEE TABLE A BELOW)

NORMAL POSITION:

NO - NORMALLY OPEN
NC - NORMALLY CLOSED
LO - LOCKED OPEN
LC - LOCKED CLOSED

FAILURE MODE:

FO - FAIL OPEN
FC - FAIL CLOSED
FAI - FAIL AS IS

REMOTE POSITION
INDICATION:

YES - NO

TEST PERIOD:

1. REFUELING SHUTDOWN
2. COLD SHUTDOWN - SEE NOTE(S) 1
FOR DEFINITION UNDER SUB-SECTION
I.E. & F.
3. OPERATION - 3 MONTHS OR LESS
(CODE).

INSERVICE INSPECTION
(ISI) TESTS:

SEE APPROPRIATE LIST OF CATEGORY
LEGEND (SUB-SECTION I.E. & I.G.).

ASTERISK (*) - ONE (1) INSPECTION
INTERVAL (TEN YEARS).



I.D. LIST OF VALVE TABLE SYMBOLS (CONT'D)

RRB NO. NUMBER RELATES TO APPROPRIATE VALVE RELIEF REQUEST BASIS.

VALVE COORDINATE: LOCATION OF VALVE ON DRAWING.

REMARKS: RELATED TO SPECIAL ALTERNATE TESTING.

TABLE - A (TABLE IWV-3700-1) (2)

<u>VALVE CATEGORY</u>	<u>VALVE FUNCTION</u>	<u>CODE LEAK TEST</u>	<u>CODE EXERCISE TEST</u>	<u>SPECIAL TEST PROCEDURE</u>
A	ACTIVE	IWV-3420	IWV-3410	NONE
A	PASSIVE	IWV-3420	NONE	NONE
B	ACTIVE	NONE	IWV-3410	NONE
C - SAFETY AND RELIEF	ACTIVE	NONE	IWV-3510	NONE
C - CHECK (1)	ACTIVE	NONE	IWV-3520	NONE
D	ACTIVE	NONE	NONE	IWV-3600

NOTE: (1). Combination Category AC Check Valves shall be leak tested IWV-3420.

(2) No Tests required for Category B, C, and D passive valves.



I.E. LEGEND FOR TABLE I - TEST PARAMETERS TO CODE

CATEGORY A AND B VALVES

Exercising

- EF-1 Exercise valve (full stroke) for operability every 3 months (Code).
- EF-2 Exercise valve (full stroke) for operability during cold shutdown (Code).
- EF-3 Exercise valve (full stroke) for operability during refueling shutdown (Code).
- EF-4 Exercise valve (full stroke) for operability prior to return to service (Code).
- EF-5 Exercise valve with remote position indicator at least once every 2 years for verification that valve operation is accurately indicated. (Code)
- EF-7 Exercise valve (with Fail-Safe Actuators) to observe failure mode every 3 months (Code).
- EF-8 Exercise valve (with Fail-Safe Actuators) to observe failure mode during refueling shutdown (Code).
- EF-9 Exercise valve (with Fail-Safe Actuators) to observe failure mode during refueling shutdown (Code).

Measurement of Full Stroke Time

- EST-1 Exercise valve - power operated (full stroke) and measure time (Code) (5 seconds - Max.) (FPL)
- EST-2 Exercise valve - power operated (full stroke) and measure time (Code) (10 seconds - Max.) (FPL)
- EST-3 Exercise valve - power operated (full stroke) and measure time (Code) (60 seconds - Max.) (FPL)
- EST-4 Exercise valve - power operated (full stroke) and measure time (Code) (120 seconds - Max.) (FPL)
- EST-5 Exercise valve - power operated (full stroke) and measure time (Code) (180 seconds - Max.) (FPL)
- EST-6 Exercise valve - power operated (full stroke) and measure time (Code) (15 seconds - Max.) (FPL)



I.E. LEGEND FOR TABLE I - TEST PARAMETERS TO CODE

CATEGORY A AND B VALVES
(Continued)

Valve Leak Rate Tests

SLT-1 Seat leakage test valve during refueling but less than every 2 years (Code).

CATEGORY C VALVES

Check Valves

EF-1 Exercise valve (full stroke) for operability every 3 months (Code).

EF-2 Exercise valve (full stroke) for operability during cold shutdown (Code).

EF-3 Exercise valve (full stroke) for operability during refueling shutdown (Code).

Safety and Relief Valves

TF-1 Safety and relief valve tests (Set Point) to ASME Table IWV-3510-1 (Code).

NOTES:

1) Cold Shutdown - ISI test scheduled for Test Period 2 (Cold Shutdown) shall commence no later than 48 hours after reaching Cold Shutdown conditions, or no later than 64 hours if Cold Shutdown conditions are reached between Friday 1600 hours and Monday 0800 hours. In the case of frequent Cold Shutdowns, valve testing will not be performed more often than once every three (3) months for Category A, B, and C valves. Valves that are not tested during a specified Cold Shutdown, due to plant startup, will be identified to assure their testing in the event of untimely Cold Shutdowns within the three (3) month time period. In any event, plant startup shall not be delayed to complete valve testing.

2) Corrective Action - Where a valve fails to meet the requirements of the program and/or the reference Code, the condition(s) shall be reviewed by the Plant nuclear Safety Committee for disposition and determination of whether it involves an unreviewed safety question prior to commencing with plant startup or continuing with plant operation.



I.E. LIST OF CATEGORY (A-B-C): LEGEND FOR TABLE I (CONT'D)

- CATEGORY "C" VALVES-

NOTES:

- 3) Test Period - Column: Where test period is denoted as 1 or 2, tests may be performed during either period, dependent upon plant conditions.



I.F. LIST OF CATEGORY A: TABLE I.A. - SEAT LEAKAGE TEST
PARAMETERS TO FPL REQUIREMENTS.

SEAT LEAKAGE RATES BASED ON AIR TESTS

50 PSI = DELTA P AT Pa .

WHERE:

Pa = ACCIDENT PRESSURE



I.G. LIST OF CATEGORY (A-B-C): LEGEND FOR TABLE II -
TESTING TO CODE REQUIREMENTS DETERMINED TO BE
IMPRACTICAL AND RELIEF REQUESTED....PARAGRAPH
50.55a (G)5 (iii).

LEGEND - The LEGEND in Sub-Section I.E. applies
to this Table.

NOTES:

- 1) Notes 1, 2, and 3 of Sub-Section I.E. apply
to this Table.
- 2) Each valve in Table II is either
 - a) Partially tested to code or,
 - b) Relief requested and supported by Relief
Request Basis including Alternate Tests.



I.H. LIST OF VALVES EXEMPT FOR TESTING (IWV-1200)

TABLE III

(a) Valves used only for operator convenience including:

- o Vent Valves
- o Drain Valves
- o Instrument Valves
- o Test Valves
- o Valves used only for maintenance
- o Valves used for system control (such as pressure regulating valves)

(b) External control and protection systems responsible for sensing plant conditions and providing signals for valve operation.

(c) Non-nuclear Safety Valves

NOTE: Valves listed in table III are not included in this report. They are identified and included in plant records.

FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
4-333	3	GLOBE	MAN	2	A	P	NC	FAI	NO	1	SLT-1	C-17	

SYSTEM TITLE: CHEM. VOL. CONT. (CVCS)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-503184



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
4-945E	1	CHECK	S/A	2	AC	P	NC	---	NO	1	SLT-1	C-9	
MOV-4-843A	4	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-6 EF-5	C-13	
HQV-4-843B	4	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-6 EF-5	C-13	
MOV-4-867A	4	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-6 EF-5	C-13	
MOV-4-867B	4	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-6 EF-5	C-13	
MOV-4-867C	4	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-6 EF-5	C-13	
MOV-4-869	3	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-3 EF-5	D-12	
MOV-4-880A	6	GATE	MO	2	A	A	NC	FAI	YES	3 3 1	EF-1 EST-6 EF-5	B-7	
MOV-4-880B	6	GATE	MO	2	A	A	NC	FAI	YES	3 3 1	EF-1 EST-6 EF-5	B-7	
4-895V	3/4	GLOBE	MAN	2	A	P	LC	FAI	NO	1	SLT-1	D-12	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
4-702A	16	CHECK	S/A	3	C	A	NO	---	NO	3	EF-1	C-11	
4-702B	16	CHECK	S/A	3	C	A	NO	---	NO	3	EF-1	B-11	
4-702C	16	CHECK	S/A	3	C	A	NO	---	NO	3	EF-1	B-11	
MOV-4-749A	16	GATE	MO	3	B	A	NC	FAI	YES	3 3 1	EF-1 EST-5 EF-5	D-5	
MOV-4-749B	16	GATE	MO	3	B	A	NC	FAI	YES	3 3 1	EF-1 EST-5 EF-5	C-6	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
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NO TABLE I VALVES



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
PCV-4-1014	1	GLOBE	A/O	2	A	A	NC	FC	NO	3 3 1	EF-1 EF-7 SLT-1	D-6	
CV-4-4658A	3/4	DIAPH	A/O	2	B	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-2 EF-5	C-7	
CV-4-4658B	3/4	DIAPH	A/O	2	A	A	NO	FC	YES	3 3 3 1 1	EF-1 EF-7 EST-2 SLT-1 EF-5	C-6	
CV-4-4659A	3/4	DIAPH	A/O	2	B	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-2 EF-5	C-7	
CV-4-4659B	3/4	DIAPH	A/O	2	A	A	NO	FC	YES	3 3 3 1 1	EF-1 EF-7 EST-2 SLT-1 EF-5	C-6	
CV-4-4668A	3	DIAPH	A/O	2	B	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-2 EF-5	B-7	
CV-4-4668B	3	DIAPH	A/O	2	A	A	NO	FC	YES	3 3 3 1 1	EF-1 EF-7 EST-2 EF-5 SLT-1	B-6	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT..	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
RV-4-551A	4	SAFE	S/A	1	C	A	NC	---	NO	1	TF-1	A-7	
RV-4-551B	4	SAFE	S/A	1	C	A	NC	---	NO	1	TF-1	A-7	
RV-4-551C	4	SAFE	S/A	1	C	A	NC	---	NO	1	TF-1	A-6	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
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NO TABLE I VALVES

SYSTEM TITLE: SAMPLING (SS)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-503193

FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
4-741A	2	GLOBE	MAN	2	B	A	LO	FAI	NO	3	EP-1	D-11	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
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NO TABLE I VALVES

SYSTEM TITLE: AUX. COOLANT SPENT FUEL

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-503195

FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
RV-4-1400	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	C-10	
RV-4-1401	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	C-9	
RV-4-1402	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	C-10	
RV-4-1403	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	C-9	
RV-4-1405	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	B-10	
RV-4-1406	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	B-10	
RV-4-1407	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	B-11	
RV-4-1408	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	B-10	
RV-4-1410	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	C-11	
RV-4-1411	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	C-11	
RV-4-1412	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	C-11	
RV-4-1413	6	SAFE	S/A	2	C	A	NC	--	NO	1	TF-1	C-10	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
MOV-4-1403	3	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-3 EF-5	C-12	
MOV-4-1404	3	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-3 EF-5	D-12	
MOV-4-1405	3	GATE	MO	2	B	A	NC	FAI	YES	3 3 1	EF-1 EST-3 EF-5	D-12	
4-10-083	4	CHECK	S/A	3	C	A	NC	---	NO	3	EF-1	E-12	
4-10-085	4	CHECK	S/A	3	C	A	NC	--	NO	3	EF-1	E-12	
4-10-087	4	CHECK	S/A	3	C	A	NC	--	NO	3	EF-1	F-12	
4-10-119	3	S/CHK	S/A	3	C	A	NC	--	NO	3	EF-1	C-12	
4-10-219	3	S/CHK	S/A	3	C	A	NC	--	NO	3	EF-1	D-12	
4-10-319	3	S/CHK	S/A	3	C	A	NC	--	NO	3	EF-1	D-12	
4-10-120	4	S/CHK	S/A	3	C	A	NC	--	NO	3	EF-1	C-12	
4-10-220	4	S/CHK	S/A	3	C	A	NC	--	NO	3	EF-1	D-12	
4-10-320	4	S/CHK	S/A	3	C	A	NC	--	NO	3	EF-1	D-12	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
20-143	6	CHECK	S/A	3	C	A	NC	--	NO	3	EF-1	B-12	
20-243	6	CHECK	S/A	3	C	A	NC	--	NO	3	EF-1	C-12	
20-343	6	CHECK	S/A	3	C	A	NC	--	NO	3	EF-1	D-12	
4-20-140	4	CHECK	S/A	2	C	A	NC	--	NO	3	EF-1	B-10	
4-20-240	4	CHECK	S/A	2	C	A	NC	--	NO	3	EF-1	B-10	
4-20-340	4	CHECK	S/A	2	C	A	NC	--	NO	3	EF-1	C-11	
4-20-401	8	CHECK	S/A	3	C	A	NC	--	NO	3	EF-1	A-11	

SYSTEM TITLE: CONDENSATE AND FEEDWATER

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502028



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
CV-4-2803	2	GLOBE	A/O	2	A	P	LO	FO	NO	1	SLT-1	C-8	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
4-50-311	24	CHECK	S/A	3	C	A	NO	--	NO	3	EF-1	F-3	
4-50-321	24	CHECK	S/A	3	C	A	NO	--	NO	3	EF-1	F-4	
4-50-331	24	CHECK	S/A	3	C	A	NO	--	NO	3	EF-1	F-5	

SYSTEM TITLE: INTAKE COOL WATER

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502031.



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
70-006A	2	CHECK	S/A	3	C	A	NC	---	NO	3	EF-1	B-4	
70-006B	2	CHECK	S/A	3	C	A	NC	---	NO	3	EF-1	B-4	

SYSTEM TITLE: DIESEL OIL

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502033



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
CV-4-2810	6	GLOBE	A/O	2	B	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 EF-5	A-11	
CV-4-2812	6	GLOBE	A/O	2	B	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 EF-5	B-11	
CV-4-2814	6	GLOBE	A/O	2	B	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 EF-5	B-11	

SYSTEM TITLE: PRIMARY MAKEUP & CONT. COOL.

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502036



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
CV-4-2903	10	BUTFY	A/O	2	B	A	NO	FO	YES	3 3 3 1	EP-1 EP-7 EST-3 EP-5	A-8	
CV-4-2904	10	BUTFY	A/O	2	B	A	NO	FO	YES	3 3 3 1	EP-1 EP-7 EST-3 EP-5	B-8	
CV-4-2905	10	BUTFY	A/O	2	B	A	NO	FO	YES	3 3 3 1	EP-1 EP-7 EST-3 EP-5	B-8	
CV-4-2906	10	BUTFY	A/O	2	B	A	NC	FO	YES	3 3 3 1	EP-1 EP-7 EST-3 EP-5	A-11	
CV-4-2907	10	BUTFY	A/O	2	B	A	NC	FO	YES	3 3 3 1	EP-1 EP-7 EST-3 EP-5	B-11	
CV-4-2908	10	BUTFY	A/O	2	B	A	NC	FO	YES	3 3 3 1	EP-1 EP-7 EST-3 EP-5	C-11	

SYSTEM TITLE: PRIMARY MAKEUP & CONT. COOL.

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502036



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	VALVE COORD.	REMARKS
4-10-567	2	CHECK	S/A	2	AC	P	NC	--	NO	1	SLT-1	D-6	

SYSTEM TITLE: PRIMARY MAKEUP AND CONT. COOL

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502036

FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	VALVE COORD.	REMARKS
POV-4-2600	48	BUTFY	A/O	2	A	P	NC	FC	YES	1	SLT-1	H-2	
POV-4-2601	48	BUTFY	A/O	2	A	P	NC	FC	YES	1	SLT-1	B-3	
POV-4-2602	54	BUTFY	A/O	2	A	P	NC	FC	YES	1	SLT-1	D-2	
POV-4-2603	54	BUTFY	A/O	2	A	P	NC	FC	YES	1	SLT-1	D-3	

SYSTEM TITLE: CONTAINMENT VENTILATION

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502037



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
CV-4-2819	2	GLOBE	A/O	2	A	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 SLT-1	D-2	
CV-4-2826	2	GLOBE	A/O	2	A	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 SLT-1	D-1	



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
CV-4-2821	3	GLOBE	A/O	2	A	A	NO	FC	YES	3	EP-1	C-9	
										3	EP-7		
										3	EST-2		
										1	SLT-1		
										1	EP-5		
CV-4-2822	3	GLOBE	A/O	2	A	A	NO	FC	YES	3	EP-1	C-9	
										3	EP-7		
										3	EST-2		
										1	SLT-1		
										1	EP-5		

SYSTEM TITLE: CONT. & RAD. DRAINS & VENTS

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502038



FLORIDA POWER & LIGHT COMPANY
TABLE I - TESTS TO CODE.

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	VALVE COORD	REMARKS
4-BA-201	2 1/2	GATE	S/A	2	C	P	NC	---	NO	1	SLT-1	B-3	TABLE IWV-3700-1



I.F. LIST OF CATEGORY A: TABLE I.A. - SEAT LEAKAGE TEST
PARAMETERS TO FPL REQUIREMENTS.

SEAT LEAKAGE RATES BASED ON AIR TESTS

50 PSI = DELTA P AT Pa

WHERE:

Pa = ACCIDENT PRESSURE



TURKEY POINT UNIT NO. 4

FLORIDA POWER & LIGHT COMPANY
TABLE I.A. - VALVE LEAKAGE RATES

PENE. NO.	VALVE NUMBER	LEAKAGE CC/MIN	PENE. NO.	VALVE NUMBER	LEAKAGE CC/MIN	PENE. NO.	VALVE NUMBER	LEAKAGE CC/MIN
33	SV-4-2913	2,000	17	4-895V	500	32	PAHM-4-001B	3,000
5	SV-4-6385	1,000	19A	MOV-4-880A	5,000	33	SV-4-2911	2,000
6	4-519	2,500	19B	MOV-4-880B	5,000	34	4-40-204	2,000
7	CV-4-519A	2,000	20	CV-4-956C	1,000	34	4-40-205	2,000
8	CV-4-956A	1,000	23	CV-4-2821	2,000	35	POV-4-2600	15,000
9	CV-4-956B	1,000	23	CV-4-2822	2,000	35	POV-4-2601	15,000
10	CV-4-4658B	500	24A	4-298A	2,000	36	POV-4-2602	15,000
10	PCV-4-1014	2,000	24B	4-298B	2,000	36	POV-4-2603	15,000
14	CV-4-200A	3,000	24C	4-298C	2,000	42	CV-4-855	500
14	CV-4-200B	3,000	25	MOV-4-381	2,000	47	4-10-567	4,000
14	CV-4-200C	3,000	25	MOV-4-6386	2,000	52	CV-4-4668B	2,000
14	CV-4-204	1,000	29	CV-4-2803	2,000	53	HV-4-4	1,000
15	HCV-4-121	2,000	29	4-40-336	4,000	53	PAHM-4-002B	3,000
15	4-333	2,000	31	CV-4-4659B	500	54A	MOV-4-861A	7,000
15	4-312C	4,000	32	4-11-003	2,000	54B	MOV-4-861B	7,000
16	HV-4-2	1,000	32	SV-4-2912	2,000	55	CV-4-956D	1,000
16	PAHM-4-002A	3,000	32	PAHM-4-001A	3,000	63	CV-4-2819	2,000
30	4-BA-201	3,000	30	CV-4-6165	3,000	63	CV-4-2826	2,000
						42	4-945E	3,000

SYSTEM TITLE: VARIOUS

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: VARIOUS



I.G. LIST OF CATEGORY (A-B-C): LEGEND FOR TABLE II - TESTING TO CODE REQUIREMENTS DETERMINED TO BE IMPRACTICAL AND RELIEF REQUESTED....PARAGRAPH 50.55a (G)5 (iii).

LEGEND - The LEGEND in Sub-Section I.E. applies to this Table.

NOTES:

- 1) Notes 1, 2, and 3 of Sub-Section I.E. apply to this Table.
- 2) Each valve in Table II is either
 - a) Partially tested to code or,
 - b) Relief requested and supported by Relief Request Basis including Alternate Tests.



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
CV-4-200A	2	GLOBE	A/O	1	A	A	NC	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	1	D-19	
CV-4-200B	2	GLOBE	A/O	1	A	A	NC	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	1	D-18	
CV-4-200C	2	GLOBE	A/O	1	A	A	NO	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	1	D-18	
CV-4-204	2	GLOBE	A/O	2	A	A	NO	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	2	D-17	
MOV-4-381	3	GATE	MO	2	A	A	NO	FAI	YES	2 2 1 1	EF-2 EST-3 SLT-1 EF-5	3	B-16	
MOV-4-6386	3	GATE	MO	2	A	A	NO	FAI	YES	2 2 1 1	EF-2 EST-3 SLT-1 EF-5	3	B-16	

FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
HCV-4-121	3	GLOBE	A/O	2	A	A	NO	FO	NO	2 2 1	EF-2 EF-8 SLT-1	5	C-17	
CV-4-310A	3	GLOBE	A/O	1	B	A	NO	FO	YES	2 2 2 1	EF-2 6 EF-8 EST-3 EF-5		C-19	
CV-4-310B	3	GLOBE	A/O	1	B	A	NC	FO	YES	2 2 2 1	EF-2 EF-8 EST-3 EF-5	6	C-19	
LCV-4-115C	4	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-3 EF-5	7	C-14	



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
LCV-4-115B	4	BUTFY	A/O	2	B	A	NC	FC	YES	2 2 2 1	EF-2 EF-8 EST-3 EF-5	8	A-14	
MOV-4-350	2	GATE	MO	2	B	A	NC	FAI	YES	2 2 1	EF-2 EST-3 EF-5	9	A-12	
4-312A	3	CHECK	S/A	1	C	A	NC	--	NO	2	EF-2	10	C-19	
4-312B	3	CHECK	S/A	1	C	A	NC	--	NO	2	EF-2	10	C-19	
4-351	2	CHECK	S/A	2	C	A	NC	--	NO	2	EF-2	11	A-12	
4-357	4	CHECK	S/A	2	C	A	NC	--	NO	2	EF-2	12	A-13	



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE.	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
4-298A	2	CHECK	S/A	1	AC	A	NO	--	NO	1 1	EF-3 SLT-1	13	A-20	
4-298B	2	CHECK	S/A	1	AC	A	NO	--	NO	1 1	EF-3 SLT-1	13	A-19	
4-298C	2	CHECK	S/A	1	AC	A	NO	--	NO	1 1	EF-3 SLT-1	13	A-18	
4-312C	3	CHECK	S/A	1	AC	A	NO	--	NO	1 1	EF-3 SLT-1	14	C-17	

SYSTEM TITLE: CHEM. VOL CONT. (CVCS)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-503184

POWER & LIGHT COMPANY
TABLE TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
MOV-4-860A	14	GATE	MO	2	B	A	NC	FAI	YES	2 2 1	EF-2 EST-4 EF-5	1	A-9	
MOV-4-860B	14	GATE	MO	2	B	A	NC	FAI	YES	2 2 1	EF-2 EST-4 EF-5	1	A-9	
MOV-4-861A	14	GATE	MO	2	A	A	NC	FAI	YES	2 2 1 1	EF-2 EST-4 SLT-1 EF-5	2	A-8	
MOV-4-861B	14	GATE	MO	2	A	A	NC	FAI	YES	2 2 2 1	EF-2 EST-4 SLT-1 EF-5	2	A-8	
MOV-4-863A	8	GATE	MO	2	B	A	LC	FAI	YES	2 2 1	EF-2 EST-3 EF-5	3	B-8	
MOV-4-863B	8	GATE	MO	2	B	A	LC	FAI	YES	2 2 1	EF-2 EST-3 EF-5	3	B-8	
MOV-4-872	8	GATE	MO	2	B	A	NC	FAI	YES	2 2 1	EF-2 EST-3 EF-5	4	A-7	
CV-4-855	1	GLOBE	A/O	2	A	A	NC	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	5	C-9	

SYSTEM TITLE: SAFETY INJECTION SYS. (SIS)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: E-503185



FL POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
MOV-4-866A	2	GLOBE	MO	1	B	A	LC	FAI	YES	2 2 1	EP-2 EST-3 EP-5	6	D-15	
MOV-4-866B	2	GLOBE	MO	1	B	A	LC	FAI	YES	2 2 1	EP-2 EST-3 EP-5	6	D-15	
4-876A	8	CHECK	S/A	1	C	A	NC	--	NO	2	EP-2	7	B-15	
4-876B	8	CHECK	S/A	1	C	A	NC	--	NO	2	EP-2	7	A-13	
4-876C	8	CHECK	S/A	1	C	A	NC	--	NO	2	EP-2	7	A-11	
4-876D	8	CHECK	S/A	1	C	A	BC	--	NO	2	EP-2	8	A-13	
4-876E	8	CHECK	S/A	1	C	A	NC	--	NO	2	EP-2	8	A-11	

SYSTEM TITLE: SAFETY INJECTION SYS (SIS)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: E-503185



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
879A	3	CHECK	S/A	2	C	A	NC	---	NO	3	EF-1	19	D-7	
879B	3	CHECK	S/A	2	C	A	NC	---	NO	3	EF-1	19	C-7	
879C	3	CHECK	S/A	2	C	A	NC	---	NO	3	EF-1	19	C-7	
879D	3	CHECK	S/A	2	C	A	NC	---	NO	3	EF-1	19	C-7	
4-875A	10	CHECK	S/A	1	C	A	NC	---	NO	2	EF-2	9	A-16	
4-875B	10	CHECK	S/A	1	C	A	NC	---	NO	2	EF-2	9	A-17	
4-875C	10	CHECK	S/A	1	C	A	NC	---	NO	2	EF-2	9	A-17	

SYSTEM TITLE: SAFETY INJECTION SYS (SIS)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: E-503185



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
MOV-878A	4	GATE	MO	2	B	A	NO	FAI	YES	1 1 1	EP-3 EST-3 EP-5	10	C-7	
MOV-878B	4	GATE	MO	2	B	A	NO	FAI	YES	1 1 1	EP-3 EST-3 EP-5	10	C-7	
MOV-4-864A	16	GATE	MO	2	B	A	LO	FAI	YES	1 1 1	EP-3 EST-4 EP-5	11	D-2	
MOV-4-864B	16	GATE	MO	2	B	A	LO	FAI	YES	1 1 1	EP-3 EST-4 EP-5	11	D-2	
MOV-4-862A	14	GATE	MO	2	B	A	LO	FAI	YES	1 1 1	EP-3 EST-5 EP-5	12	A-5	
MOV-4-862B	14	GATE	MO	2	B	A	LO	FAI	YES	1 1 1	EP-3 EST-5 EP-5	12	A-5	
SV-4-2905	2	GATE	SO	2	B	A	NC	FAI	NO	1	EP-3	13	C-10	
SV-4-2906	2	GATE	SO	2	B	A	NC	FAI	NO	1	EP-3	13	C-10	
SV-4-2907	2	GATE	SO	2	B	A	NC	FAI	NO	1	EP-3	13	C-9	

SYSTEM TITLE: SAFETY INJECTION SYS (SIS)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: E-503185

FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER.	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
SV-4-2908	2	GATE	SO	2	B	A	NC	FAI	NO	1	EP-3	13	C-9	
SV-4-2909	2	GATE	SO	2	B	A	NC	FAI	NO	1	EP-3	13	C-9	
SV-4-2910	2	GATE	SO	2	B	A	NC	FAI	NO	1	EP-3	13	C-9	
4-2918	2	CHECK	S/A	2	C	A	NC	--	NO	*	*	14	B-10	*SEE ALTER- NATE TESTING IN REQUEST FOR RELIEF BASIS.
4-2919	2	CHECK	S/A	2	C	A	NC	--	NO	*	*	14	B-9	
4-2920	2	CHECK	S/A	2	C	A	NC	--	NO	*	*	14	B-9	
4-2921	2	CHECK	S/A	2	C	A	NC	--	NO	*	*	14	B-10	
4-2922	2	CHECK	S/A	2	C	A	NC	--	NO	*	*	14	B-9	
4-2923	2	CHECK	S/A	2	C	A	NC	--	NO	8	8	14	B-9	
4-874A	2	CHECK	S/A	1	C	A	NC	--	NO	1	EP-3	15	D-17	
4-874B	2	CHECK	S/A	1	C	A	NC	--	NO	1	EP-3	15	D-17	

SYSTEM TITLE: SAFETY INJECTION SYS(SIS)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: E-503185



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
4-873A	2	CHECK	S/A	1	C	A	NC	--	NO	1	EP-3	16	C-15	
4-873B	2	CHECK	S/A	1	C	A	NC	--	NO	1	EP-3	16	C-15	
4-873C	2	CHECK	S/A	1	C	A	NC	--	NO	1	EP-3	16	C-14	
4-875D	10	CHECK	S/A	1	C	A	NC	--	NO	1	EP-3	17	B-15	
4-875E	10	CHECK	S/A	1	C	A	NC	--	NO	1	EP-3	17	B-13	
4-875F	10	CHECK	S/A	1	C	A	NC	--	NO	1	EP-3	17	B-11	
4-890A	6	CHECK	S/A	2	C	A	NO	--	NO	*	*	14	B-8	*SEE ALTER- NATE TESTING IN REQUEST FOR RELIEF BASIS.
4-890B	6	CHECK	S/A	2	C	A	NC	--	NO	*	*	14	B-8	

FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
MOV-4-865A	10	GATE	MO	2	B	P	LO	FAI	YES	---	---		B-15	TABLE IWV-3700-1
MOV-4-865B	10	GATE	MO	2	B	P	LO	FAI	YES	---	---		B-13	TABLE IWV-3700-1
MOV-4-865C	10	GATE	MO	2	B	P	LO	FAI	YES	---	---		B-11	TABLE IWV-3700-1



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
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NO TABLE II VALVES



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
CV-4-739	3	GLOBE	A/O	2	B	A	NC	FC	YES	2 2 2 1	EF-2 EF-8 EST-2 EF-5	1	B-13	
4-738	3	CHECK	S/A	2	C	A	NC	--	NO	2	EF-2	2	A-17	
MOV-4-716A	6	GATE	MO	3	B	A	NO	FAI	YES	2 2 1	EF-2 EST-6 EF-5	3	B-18	
MOV-4-716B	6	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-6 EF-5	4	B-17	
MOV-4-730	6	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-6 EF-5	5	C-13	
FCV-4-626	3	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-6 EF-5	6	B-13	



FLORIDA POWER AND LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
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NO TABLE II VALVES

SYSTEM TITLE: WASTE DISPOSAL LIQUID

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-503189



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
CV-4-519A	3	DIAPH	A/O	2	A	A	NO	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-4 SLT-1 EF-5	1	A-12	
4-519	3/4	CHECK	S/A	2	AC	A	NC	--	NO	2 1	EF-2 SLT-1	2	A-11	
SV-4-6385	3/8	GLOBE	SO	2	A	A	NC	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 SLT-1	3	A-12	



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
CV-4-956A	3/8	GLOBE	A/O	2	A	A	NC	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	1	H-11	
CV-4-956B	3/8	GLOBE	A/O	2	A	A	NC	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	1	H-11	
CV-4-956C	3/8	GLOBE	A/O	2	A	A	NC	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	1	G-11	
CV-4-956D	3/8	GLOBE	A/O	2	A	A	NC	FC	YES	2 2 2 1 1	EF-2 EF-8 EST-2 SLT-1 EF-5	1	F-11	



FLORIDA POWER LIGHT COMPANY
TABLE II - TO COMMAND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
MOV-4-744A	10	GATE	MO	2	B	A	NC	FAI	YES	2 2 1	EF-2 EST-3 EF-5	1	B-12	
MOV-4-744B	10	GATE	MO	2	B	A	NC	FAI	YES	2 2 1	EF-2 EST-3 EF-5	1	B-12	
4-753A	10	CHECK	S/A	2	C	A	NC	---	NO	2	EF-2	2	F-7	
4-753B	10	CHECK	S/A	2	C	A	NC	---	NO	2	EF-2	2	H-7	

SYSTEM TITLE: AUX. COOL. RES. HEAT REM.

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-503194



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
MOV-4-751	14	GATE	MO	1	B	P	LC	FAI	YES	---	---	3	C-5	TABLE IWV-3700-1
MOV-4-750	14	GATE	MO	1	B	P	NC	FAI	YES	---	---		B-5	TABLE IWV-3700-1
HCV-4-758	12	BUTFY	A/O	2	B	P	LO	FO	NO	---	---		G-11	TABLE IWV-3700-1

SYSTEM TITLE: AUX. COOL. RES. HEAT REM.

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-503194



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
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NO TABLE II VALVES

SYSTEM TITLE: AUX. COOLANT SPENT FUEL

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-503195

FLORIDA POWER AND LIGHT COMPANY
TABLE II - TO CORRECT AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
POV-4-2604	26	POWER ASST'D CHECK	A/CYL	2	C	A	NO	FC	YES	2 2 2 1	EF-2 EF-8 EST-1 EF-5	1	B-9	
POV-4-2605	26	POWER ASST'D CHECK	A/CYL	2	C	A	NO	FC	YES	2 2 2 1	EF-2 EF-8 EST-1 EF-5	1	B-10	
POV-4-2606	26	POWER ASST'D CHECK	A/CYL	2	C	A	NO	FC	YES	2 2 2 1	EF-2 EF-8 EST-1 EF-5	1	B-11	

SYSTEM TITLE: STEAM SYSTEM

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502027



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
MOV-4-1425	1	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-3 EF-5	1	E-11	
MOV-4-1426	1	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-3 EF-5	1	F-11	
MOV-4-1427	1	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-3 EF-5	1	F-11	
CV-4-6275A	2	GLOBE	A/O	2	B	A	NO	FC	YES	2 2 1	EF-8 EST-3 EF-5	2	C-10	
CV-4-6275B	2	GLOBE	A/O	2	B	A	NO	FC	YES	2 2 1	EF-8 EST-3 EF-5	2	C-10	
CV-4-6275C	2	GLOBE	A/O	2	B	A	NO	FC	YES	2 2 1	EF-8 EST-3 EF-5	2	C-11	
CV-4-2900	14	CHECK	A/O	2	C	A	NO	--	NO	2	EF-2	3	D-7	
CV-4-2901	14	CHECK	A/O	2	C	A	NO	--	NO	2	EF-2	3	D-8	
CV-4-2902	14	CHECK	A/O	2	C	A	NO	--	NO	2	EF-2	3	D-9	

SYSTEM TITLE: CONDENSATE & FEEDWATER

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502028



FLORIDA POWER AND LIGHT COMPANY
TABLE II - TO CORRECT AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
CV-4-2816	4	GLOBE	A/O	2	B	A	NC	FC	NO	3 3	EF-1 EF-7	4	B-10	
CV-4-2817	4	GLOBE	A/O	2	B	A	NC	FC	NO	3 3	EF-1 EF-7	4	B-11	
CV-4-2818	4	GLOBE	A/O	2	B	A	NC	FC	NO	3 3	EF-1 EF-7	4	C-11	
CV-4-2831	4	GLOBE	A/O	2	B	A	NC	FC	NO	3 3	EF-1 EF-7	4	B-10	
CV-4-2832	4	GLOBE	A/O	2	B	A	NC	FC	NO	3 3	EF-1 EF-7	4	B-11	
CV-4-2833	4	GLOBE	A/O	2	B	A	NC	FC	NO	3 3	EF-1 EF-7	4	C-11	

SYSTEM TITLE: CONDENSATE & FEEDWATER

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502028



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
4-40-204	2	GATE	MAN	2	A	A	LC	FAI	NO	2 1	EF-2 SLT-1	1	A-10	
4-40-205	2	CHECK	S/A	2	AC	A	NC	--	NO	2 1	EF-2 SLT-1	2	A-11	
4-40-203	2	GATE	MAN	2	B	A	LC	FAI	NO	2	EF-2	3	A-10	
4-40-336	2	CHECK	S/A	2	AC	A	NO	--	NO	1 1	EF-3 SLT-1	4	C-8	



FLORIDA POWER AND LIGHT COMPANY
TABLE 11 - TO CORRECT AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
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NO TABLE II VALVES

SYSTEM TITLE: INTAKE COOL WATER

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502031



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
CV-2046A	2	GLOBE	A/O	3	B	A	NC	--	NO	3	EF-1	2	C-3	
CV-2046B	2	GLOBE	A/O	3	B	A	NC	--	NO	3	EF-1	2	C-5	
SV-3522A	1 1/2	GLOBE	SO	3	B	A	NC	--	NO	3	EF-1	1	D-4	
SV-3522B	1 1/2	GLOBE	SO	3	B	A	NC	--	NO	3	EF-1	1	D-5	

SYSTEM TITLE: DIESEL OIL

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502033



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
MOV-4-1417	10	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-4 EF-5	1	B-8	
MOV-4-1418	10	GATE	MO	2	B	A	NO	FAI	YES	2 2 1	EF-2 EST-4 EF-5	1	C-10	

SYSTEM TITLE: PRIMARY MAKEUP AND CONT. COOL

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502036



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
HV-4-1	2	DIAPH	MAN	2	B	A	LC	FAI	NO	2	EF-2	1	F-10	
HV-4-2	2	DIAPH	MAN	2	A	A	LC	FAI	NO	2 1	EF-2 SLT-1	2	F-10	
HV-4-3	2	DIAPH	MAN	2	B	A	LC	FAI	NO	2	EF-2	1	F-10	
HV-4-4	2	DIAPH	MAN	2	A	A	LC	FAI	NO	2 1	EF-2 SLT-1	2	F-10	

SYSTEM TITLE: CONTAINMENT VENTILATION

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502037



FLORIDA POWER LIGHT COMPANY
TABLE II - TO CORRECT AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
4-11-003	2	CHECK	S/A	2	A/C	A	NO	--	NO	1 1	EF-3 SLT-1	3	D-7	
PAHM-4-001A	1	BALL	MAN	2	A	A	NC	FAI	NO	2 1	EF-2 SLT-1	2	D-8	
PAHM-4-001B	1	BALL	MAN	2	A	A	NC	FAI	NO	2	EF-2 SLT-1	2	D-8	
PAHM-4-002A	3/4	BALL	MAN	2	A	A	NC	FAI	NO	2 1	EF-2 SLT-1	4	F-10	
PAHM-4-002B	3/4	BALL	MAN	2	A	A	NC	FAI	NO	2 1	EF-2 SLT-1	4	F-10	

SYSTEM TITLE: CONTAINMENT VENTILATION SYSTEM

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: F-502037

FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
SV-4-2911	1	GLOBE	SO	2	A	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 SLT-1		C-8	
SV-4-2912	1	GLOBE	SO	2	A	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 SLT-1		D-8	
SV-4-2913	1	GLOBE	SO	2	A	A	NO	FC	YES	3 3 3 1	EF-1 EF-7 EST-3 SLT-1		C-8	



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CORRECT AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER.	ISI TESTS	RRB NO.	VALVE COORD.	REMARKS
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NO TABLE II VALVES

SYSTEM TITLE: CONT. & RADWASTE DRAINS & VENTS

PROGRM TITLE: VALVE TET PROGRAM

CHART TITLE: F-502038



FLORIDA POWER & LIGHT COMPANY
TABLE II - TO CODE AND RELIEF REQUEST

VALVE NUMBER	SIZE	TYPE	ACTU.	CODE CL.	CODE CAT.	ACT/ PAS	NORM. POS.	FAIL- URE MODE	REM. POS. IND.	TEST PER	ISI TESTS	RRB NO.	VALVE COORD	REMARKS
CV-4-6165	2 1/2	GATE	A/O	2	A	P	LC	FC	YES	1	SLT-1		B-4	TABLE IIV-3700-1

SYSTEM TITLE: BREATHING AIR SYSTEM (BA)

PROGRAM TITLE: VALVE TEST PROGRAM

CHART TITLE: 5177-074-M-2



ATTACHMENT A-1

TURKEY POINT UNIT NO. 4

RELIEF REQUEST BASIS



I.I. RELIEF REQUEST BASIS - ATTACHMENT A-1 CODE REQUIREMENTS
DETERMINED TO BE IMPRACTICAL....PARAGRAPH 50.55a (g) 5 (iii)
AND RELIEF REQUESTED.

Chemical and Volume Control System	Pg 1 to 8
Safety Injection System	Pg. 1 to 12
Auxiliary Coolant System Component Cooling	Pg. 1 to 3
Reactor Coolant System	Pg. 1 to 2
Sampling System	Pg. 1 to 1
Auxiliary Coolant System Residual Heat Removal	Pg. 1 to 2
Steam System	Pg. 1 to 1
Condensate and Feedwater Systems	Pg. 1 to 2
Lube Oil Service and Instrument Air	Pg. 1 to 3
Primary Make-up Containment Cooling Water and Chemical Injection Systems	Pg. 1 to 1
Containment Ventilation System	Pg. 1 to 3
Diesel Oil System	Pg. 1 to 1



RELIEF REQUEST BASIS

SYSTEM: -Chemical and Volume Control

1. Valve: CV-4-200A, CV-4-200B, CV-4-200C
Category: A
Class: 1

Function: Provides parallel letdown flow paths through the letdown orifices to control Chemical and Volume Control System letdown flow rate.

Test Requirement: IWV-3410

Basis for Relief: Testing these valves during plant operation would cause an unbalanced flow condition in the Chemical and Volume Control System. This could interrupt flow to the Reactor Coolant Pump Controlled Leakage Seal System, which could result in damage to the Reactor Coolant pumps; thereby placing the plant in an unsafe mode of operation.

Alternate Testing: These valves will be tested during cold shutdowns.

2. Valve: CV-4-204
Category: A
Class: 2

Function: Provides the letdown flowpath during plant operation.

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would cause an unbalanced flow condition in the Chemical and Volume Control System. This would interrupt flow to the Reactor Coolant Pump Controlled Leakage Seal System. This could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation. The loss of the letdown flow path will result in the loss of the regenerative heat exchanger function. This could subject the Reactor Coolant System piping to thermal shock due to cooler charging return flow from the Chemical and Volume Control System.

Alternate Testing: This valve will be tested during cold shutdowns.

RELIEF REQUEST BASIS

SYSTEM: Chemical and Volume Control

3. Valve: MOV-4-381 and MOV-4-6386
Category: A
Class: 2

Function: Provides the Reactor Coolant Pump seal injection return flow path to the Chemical and Volume Control System Volume Control Tank. In addition, provides the excess letdown flow path from the Reactor Coolant System to the Chemical and Volume Control System Control Tank.

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would interrupt flow to the Reactor Coolant Pump Controlled Leakage Seal System, which could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Chemical and Volume Control

5. Valve: HCV-4-121
Category: A
Class: 2

Function: Provides the charging flow path to the Reactor Coolant System. This valve is used to proportion flow between the seal injection supply to the Reactor Coolant pump Controlled Leakage Seal System and the charging flow to the Reactor Coolant System.

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would cause an unbalanced flow condition in the Chemical and Volume Control System. This could interrupt flow to the Reactor Coolant Pump Controlled Leakage Seal System, which could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.

6. Valve: CV-4-310A and CV-4-310B
Category: B
Class: 1

Function: Provides redundant charging flow paths to the Reactor Coolant System.

Test Requirement: IWV-3410

Basis for Relief: Testing these valves during plant operation would cause an unbalanced flow condition in the Chemical and Volume Control System. This could interrupt flow to the Reactor Coolant Pump Controlled Leakage Seal System, which could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: These valves will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Chemical and Volume Control

7. Valve: LCV-4-115C
Category: B
Class: 2

Function: Provides the flow path from the Volume Control Tank to the charging pump suction header.

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would cause an unbalanced flow condition in the Chemical and Volume Control System. This would interrupt flow to the Reactor Coolant Pump Controlled Leakage Seal System, which could result in damage to the Reactor pump; thereby, placing the plant in an unsafe mode of operation. Further, the failure of this valve in the closed position, by testing during plant operation, would isolate normal charging pump make-up.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Chemical and Volume Control

8. Valve: LCV-4-115B
Category: B
Class: 2

Function: Provides a flow path from the refueling water storage tank to the Reactor Coolant System.

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would result in the addition of Boron to the Reactor Coolant System. This could place the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.

9. Valve: MOV-4-350
Category: B
Class: 2

Function: Provides a flow path from the Boric Acid Storage Tanks to the Reactor Coolant System.

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would result in the addition of Boron to the Reactor Coolant System. This could place the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Chemical and Volume Control

10. Valve: 4-312A and 4-312B
Category: C
Class: 1

Function: Prevents reverse flow from the Reactor Coolant System to the Chemical and Volume Control System charging flow path.

Test Requirement: IWV-3520

Basis for Relief: Testing these valves during plant operation would cause an unbalanced flow condition in the Chemical and Volume Control System. This could interrupt flow to the Reactor Coolant Pump Controlled Leakage Seal System, which could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: These valves will be tested during cold shutdowns.

11. Valve: 4-351
Category: C
Class: 2

Function: Prevents reverse flow from the charging pump suction header to the Boron Addition System.

Test Requirement: IWV-3520

Basis for Relief: Testing this valve during plant operation would result in the addition of Boron to the Reactor Coolant System. This could place the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.

RELIEF REQUEST BASIS

SYSTEM: Chemical and Volume Control

12. Valve: 4-357
Category: C
Class: 2

Function: Prevents reverse flow from the charging pump suction header to the Refueling Water Storage Tank system.

Test Requirement: IWV-3520

Basis for Relief: Testing this valve during plant operation would result in the addition of Boron to the Reactor Coolant System. This could place the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.

13. Valve: 4-298A, 4-298B, and 4-298C
Category: AC
Class: 1

Function: Prevents reverse flow from the Reactor Coolant Pump Seal Injection Ssystem to the Chemical and Volume Control System.

Test Requirement: IWV-3520

Basis for Relief: Testing these valves during plant operation would interrupt flow to the Reactor Coolant Pump Controlled Leakage Seal System, which could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Further, the testing of these valves during cold shutdowns is impractical since it would require draining the Reactor Coolant Pump Seal Injection System to check the position of these valves. This would increase the possibility of causing damage to the Reactor Coolant pump seals due to the added frequency of venting the system prior to plant operation.

Alternate Testing: These valves will be tested during refueling shutdowns.

RELIEF REQUEST BASIS

SYSTEM: Chemical and Volume Control

14. Valve: 4-312C
Category: AC
Class: 1

Function: Prevents reverse flow from the Reactor Coolant System charging flow path to the Chemical and Volume Control System.

Test Requirement: IWV-3520

Basis for Relief: Testing this valve during plant operation would cause an unbalanced flow condition in the Chemical and Volume Control System. This could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Testing this valve during cold shutdown is impractical because it would require draining the charging system to check the position of the valve. This would cause a loss of the charging flow path that is routinely used to meet the Tech Spec requirements to have a Boron injection flow path to the Reactor Coolant System during cold shutdown.

Alternate Testing: This valve will be tested during refueling shutdowns.

RELIEF REQUEST BASIS

SYSTEM: Safety Injection

1. Valve: MOV-4-860A and MOV-4-860B
Category: B
Class: 2

Function: Provides the flow path from train "A" and train "B" containment recirculation sumps to the suction of the Residual Heat Removal pumps "A" and "B" respectively, during the recirculation phase following a LOCA.

Test Requirement: IWV-3410

Basis for Relief: In the event that maintenance is required upon the failure of either of these valves, by testing during plant operation, it would cause a loss of containment integrity.

Alternate Testing: These valves will be tested during cold shutdowns.

2. Valve: MOV-4-861A and MOV-4-861B
Category: A
Class: 2

Function: Provides the flow path from train "A" and train "B" containment recirculation sumps to the suction of the Residual Heat Removal pumps "A" and "B" respectively, during the recirculation phase following a LOCA.

Test Requirement: IWV-3410

Basis for Relief: The failure of either of these valves in the open position, by testing during plant operation, would cause a loss of containment integrity.

Alternate Testing: These valves will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Safety Injection

3. Valve: MOV-4-863A and MOV-4-863B
Category: B
Class: 2

Function: Provides the flow path to the alternate header to the Reactor Coolant System from the Low Pressure Safety Injection System. Also, provides the flow path to the High Pressure Safety Injection System during the recirculation mode.

Test Requirement: IWV-3410

Basis for Relief: The failure of either of these valves in the open position, by testing during plant operation, would result in diverting flow from the reactor core in the event of a safety injection signal.

Alternate Testing: These valves will be tested during cold shutdowns.

4. Valve: MOV-4-872
Category: B
Class: 2

Function: Provides the alternate flow path from the Low Pressure Safety Injection System to the Reactor Coolant System.

Test Requirement: IWV-3410

Basis for Relief: The failure of this valve in the open position, by testing during plant operation would result in diverting flow during the High Pressure Recirculation mode.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Safety Injection

5. Valve: CV-4-855
Category: A
Class: 2

Function: Provides the Nitrogen supply to maintain pressure in the Safety Injection Accumulators.

Test Requirement: IWV-3410

Basis for Relief: The failure of this valve in the closed position, by testing during plant operation, would cause a loss of Nitrogen make-up to the accumulators to maintain the pressure above Tech Spec limits.

Alternate Testing: This valve will be tested during cold shutdowns.

6. Valve: MOV-4-866A and MOV-4-866B
Category: B
Class: 1

Function: Provides High Pressure Safety Injection redundant flow paths to the Reactor Coolant System Hot Legs.

Test Requirement: IWV-3410

Basis for Relief: These valves are required by Tech Specs to be closed and locked-out at the breaker during plant operation. The testing of these valves during plant operation, coupled with the failure of Valve 4-874A or 4-874B, could subject the Safety Injection System to pressures in excess of its design pressure.

Alternate Testing: These valves will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Safety Injection

7. Valve: 4-876A, 4-876C and 4-876B
Category: C
Class: 1

Function: Prevents reverse flow from the Accumulator Safety Injection System and the High Pressure Safety Injection System to the Low Pressure Safety Injection System.

Test Requirement: IWV-3520

Basis for Relief: These valves cannot be tested during operation because the Low Pressure Safety Injection pumps do not develop sufficient discharge head to establish a flow path to the Reactor Coolant System.

Alternate Testing: These valves will be tested during cold shutdowns.

8. Valve: 4-876D and 4-876E
Category: C
Class: 1

Function: Prevents reverse flow from the Accumulator Safety Injection System and the High Pressure Safety Injection System to the Low Pressure Safety Injection System alternate flow path.

Test Requirement: IWV-3520

Basis for Relief: These valves cannot be tested during plant operation because the Low Pressure Safety Injection pumps do not develop sufficient discharge head to establish a flow path to the Reactor Coolant System.

Alternate Testing: These valves will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Safety Injection

9. Valve: 4-875A, 4-875B, 4-875C
Category: C
Class: 1

Function: Prevents reverse flow from the Reactor Coolant System to the Accumulator Safety Injection system, Low Pressure Safety Injection System, and High Pressure Safety Injection System.

Test Requirement: IWV-3520

Basis for Relief: These valves cannot be tested during plant operation because the High Pressure Safety Injection or Low Pressure Safety Injection pumps do not develop sufficient discharge head to establish a flow path to the Reactor Coolant System.

Alternate Testing: These valves will be tested during cold shutdowns.

10. Valve: MOV-878A and MOV-878B
Category: B
Class: 2

Function: Provides a flow path for any combination of two of the four High Pressure Safety Injection pumps to the Reactor Coolant System of either unit.

Test Requirement: IWV-3410

Basis for Relief: In the event that maintenance is required upon the failure of either of these valves, by testing during plant operation of both units, it would jeopardize the ability of the High Pressure Safety Injection pumps to support a LOCA; thereby, placing the units in an unsafe mode of operation.

In the event that maintenance is required upon the failure of either of these valves, by testing during cold shutdown of either unit, it would jeopardize the ability of the High Pressure Safety Injection pumps to support a LOCA on the operating unit; thereby, placing that unit in an unsafe mode of operation.

Alternate Testing: These valves will be tested during refueling shutdowns.



RELIEF REQUEST BASIS

SYSTEM: .Safety Injection

11. Valve: MOV-4-864A and MOV-4-864B
Category: B
Class: 2

Function: Provides the flow path to the High Pressure Safety Injection, Low Pressure Safety Injection, and Containment Spray pumps.

Test Requirement: IWV-3410

Basis for Relief: These valves are required by Tech Specs to be open and locked-out at the breaker during plant operation.

The failure of either of these valves in the closed position, by testing during plant operation, would cause a total loss of the Containment Spray and the Low Pressure Safety Injection Systems and, further, jeopardize the High Pressure Safety Injection System; thereby, placing the plant in an unsafe mode of operation.

The failure of either of these valves in the closed position, by testing during cold shutdown, would jeopardize the High Pressure Safety Injection System and cause a loss of an emergency flow path to the Boration Systems; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: These valves will be tested during refueling shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Safety Injection

12. Valve: MOV-4-862A and MOV-4-862B
Category: B
Class: 2

Function: Provides a flow path to the Low Pressure Safety Injection System.

Test Requirement: IWV-3410

Basis for Relief: These valves are required by Tech Specs to be open and locked-out at the breaker during plant operation. The failure of either of these valves in the closed position, by testing during plant operation, would cause a total loss of the Low Pressure Safety Injection System.

In the event that maintenance is required upon the failure of Valve 862A, by testing during cold shutdown, it would cause a loss of the RWST and jeopardize the ability of the High Pressure Safety Injection Pumps to support a LOCA on the operating unit. In the event that maintenance is required upon the failure of Valve 862B, by testing during cold shutdown, it would cause a loss of the ability to cool the core.

Alternate Testing: These valves will be tested during refueling shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Safety Injection

13. Valve: SV-4-2905 and SV-4-2906
SV-4-2907 and SV-4-2908
SV-4-2909 and SV-4-2910

Category: B
Class: 2

Function: Provides for redundant flow paths from the operating Containment Spray Headers to the associated Emergency Containment Filter.

Test Requirement: IWV-3410

Basis for Relief: These redundant self-contained, completely enclosed solenoid valves have no external valve position indicators. Therefore, stroke-time measurements and valve position verification is impractical.

Functional testing of these redundant valves by placing the containment spray system in operation would result in dousing the filters and other components located inside the containment building. Testing these valves by connecting an external water source to the containment spray header would also result in dousing the filters and other components located inside the containment building.

Alternate Testing: These redundant valves will be tested by connecting an external air supply with sufficient air pressure to verify the main disk moves to the open position.

These valves will be tested during refueling shutdowns.

RELIEF REQUEST BASIS

SYSTEM: Safety Injection

14. Valve: 4-2918 and 4-2921
4-2919 and 4-2922
4-2920 and 4-2923
4-890A and 4-890B

Category: C
Class: 2

Function: Prevents reverse flow from an operating Containment
Spray header to the other Containment Spray Header.

Test Requirement: IWV-3520

Basis for Relief: Functional testing of these redundant valves by placing the containment spray system in operation would result in dousing the components located inside the containment building. testing these redundant valves by connecting an external water source to the containment spray system would also result in dousing the components located inside the containment building.

Alternate Testing: Each of these redundant check valves will be disassembled, to inspect the valves' internals and to physically verify the valves' freedom of motion to the open and closed position, at least once each 120 month Inservice Inspection Interval. This inspection will be performed during refueling shutdowns. Any problems found during this inspection would be cause for inspecting the other valves. This provides for an inspection of one of these redundant check valves during refueling shutdowns over the 120 month Inservice Inspection Interval.

RELIEF REQUEST BASIS

SYSTEM: Safety Injection

15. Valve: 4-874A and 4-874B
Category: C
Class: 1.

Function: Prevents reverse flow from the Reactor Coolant System
Hot Legs to the High Pressure Safety Inspection System.

Test Requirement: IWV-3520

Basis for Relief: These valves cannot be tested during plant operation because the High Pressure Safety Inspection pumps do not develop sufficient discharge head to establish a flow path to the Reactor Coolant System.

Further, testing of these valves during cold shutdowns is impractical since it could subject the Reactor Coolant System to conditions exceeding Pressure-Temperature limits.

Alternate Testing: These valves will be tested during refueling shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Safety Injection

16. Valve: 4-873A and 4-873B, and 4-873C
Category: C
Class: 1

Function: Prevents reverse flow from the Accumulator Safety Injection System and the Low Pressure Safety Injection System to the High Pressure Safety Injection System.

Test Requirement: IWV-3520

Basis for Relief: These valves cannot be tested during plant operation because the High Pressure Safety Injection pumps do not develop sufficient discharge head to establish a flow path to the Reactor Coolant System.

Further, testing of these valves during cold shutdowns is impractical since it could subject the Reactor Coolant System to conditions exceeding Pressure-Temperature limits.

Alternate Testing: These valves will be tested during refueling shutdowns.

17. Valve: 4-875D, 4-875E, and 4-875F
Category: C
Class: 1

Function: Prevents reverse flow from the High Pressure Safety Injection System and the Low Pressure Safety Injection System to the Accumulator Safety Injection System.

Test Requirement: IWV-3520

Basis for Relief: These valves cannot be tested during plant operation because the accumulator pressure is insufficient to provide the differential pressure required to establish a flow path to the Reactor Coolant System.

Further, testing of these valves during cold shutdowns is impractical since it could subject the Reactor Coolant System to conditions exceeding Pressure-Temperature limits.

Alternate Testing: These valves will be tested during refueling shutdowns.

RELIEF REQUEST BASIS

SYSTEM: Safety Injection

19. Valve: 879A, 879B, 879C, and 879D
Category: C
Class: 2

Function: Prevents reverse flow from the High Pressure Safety System Supply Header to a non-operating High Pressure Safety Injection Pump.

Test Requirement: IWV-3520

Basis for Relief: These valves cannot be tested during plant operation because the High Pressure Safety Injection System Pumps do not develop sufficient discharge head to establish a flow path to the Reactor Coolant System.

Further, testing of these valves during cold shutdowns is impractical since it could subject the Reactor Coolant System to conditions exceeding pressure-temperature limits.

Alternate Testing: These check valves will be exercised during refueling shutdown.

Additional Testing: These check valves will be exercised quarterly during the performance of associated pump tests.



RELIEF REQUEST BASIS

SYSTEM: Auxiliary Coolant, Component Cooling Water

1. Valve: CV-4-739
Category: B
Class: 2

Function: Provides the component cooling water return flow path for the excess letdown heat exchanger.

Test Requirement: IWV-3410

Basis for Relief: The failure of this valve in the closed position, by testing during plant operation, would cause a loss of the Excess Letdown Heat Exchanger System function.

Alternate Testing: This valve will be tested during cold shutdowns.

Valve: 4-738
Category: C
Class: 2

Function: Prevents reverse flow from the excess letdown heat exchanger, shell side, to the component cooling water supply header.

Test Requirement: IWV-3520

Basis for Relief: The failure of this valve in the closed position, by testing during plant operation, would cause a loss of Excess Letdown Heat Exchanger System function.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Auxiliary Coolant, Component Cooling Water

3. Valve: MOV-4-716A
Category: B
Class: 3

Function: Provides the component cooling water supply flow path for the heat exchangers located in the Reactor Coolant pumps (motors and thermal barriers).

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would cause interruption of cooling water to the Reactor Coolant pumps' heat exchangers. This action could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.

4. Valve: MOV-4-716B
Category: B
Class: 2

Function: Provides the component cooling water supply flow path for the heat exchangers located in the Reactor Coolant pumps (motors and thermal barriers).

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would cause interruption of cooling water to the Reactor Coolant pumps' heat exchangers. This action could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Auxiliary Coolant, Component Cooling Water

5. Valve: MOV-4-730
Category: B
Class: 2

Function: Provides the component cooling water return flow path for the Reactor Coolant pumps' motor heat exchangers.

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would cause interruption of cooling water to the Reactor Coolant pumps' motor heat exchangers. This action could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.

6. Valve: FCV-4-626
Category: B
Class: 2

Function: Provides the component cooling return flow path for the Reactor Coolant Pumps' Controlled Leakage Seal System thermal barriers.

Test Requirement: IWV-3410

Basis for Relief: Testing this valve during plant operation would cause interruption of cooling water to the Reactor Coolant pumps' Controlled Leakage Seal System heat exchangers. This action could result in damage to the Reactor Coolant pumps; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Reactor Coolant

1. Valve: CV-4-519A
Category: A
Class: 2

Function: Provides a primary water flow path to either the pressurizer relief tank or the Reactor Coolant pumps' standpipes.

Test Requirement: IWV-3410

Basis for Relief: Failure of this valve in the open position, by testing during plant operation, would cause a loss of containment integrity.

Alternate Testing: This valve will be tested during cold shutdowns.

2. Valve: 4-519
Category: AC
Class: 2

Function: Prevents reverse flow from the Pressurizer Relief Tank to the Nitrogen system.

Test Requirement: IWV-3520

Basis for Relief: Failure of this valve in the open position, by testing during plant operation, would cause a loss of containment integrity.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Reactor Coolant

3. Valve: SV-4-6385
Category: A
Class: 2

Function: Provides flow path from pressurizer relief tank to gas analyzer.

Test Requirement: IWV-3320

Basis for Relief: These self contained, completely enclosed solenoid valves have no external valve position indication. Therefore direct observation of valve position indication is impractical.

Alternate Testing: These valves will be checked during local leak rate tests to verify that remote valve indications accurately reflect valve operation.



RELIEF REQUEST BASIS

SYSTEM: Sampling

1. Valve: CV-4-956A, CV-4-956B, CV-4-956C, and CV-4-956D
Category: A
Class: 2

Function: Provides the flow path from the pressurizer steam space, the pressurizer liquid space, the Reactor Coolant System, and the accumulators respectively, to the Sample System.

Test Requirement: IWV-3410

Basis for Relief: The failure of any one of these valves in the open position, by testing during plant operation, would cause a loss of containment integrity.

Alternate Testing: These valves will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Auxiliary Coolant, Residual Heat Removal

1. Valve: MOV-4-744A and MOV-4-744B
Category: B
Class: 2

Function: Provides a flow path from the Low Pressure Safety Injection System to the Reactor Coolant System.

Test Requirement: IWV-3410

Basis for Relief: The testing of these valves during plant operation, coupled with the failure of Valves 4-876A, 4-876B or 4-876C, could subject the Low Pressure Safety Injection System to pressures in excess of its design pressure.

Alternate Testing: These check valves will be tested during cold shutdown.

2. Valve: 4-753A and 4-753B
Category: C
Class: 2

Function: Prevents reverse flow from the Low Pressure Safety Injection (Residual Heat Removal) System supply header to a non-operating Low Pressure Safety Injection (Residual Heat Removal) pump.

Test Requirement: IWV-3520

Basis for Relief: These valves cannot be tested during plant operation because the Low Pressure Safety Injection (Residual Heat Removal) pumps do not develop sufficient discharge head to establish a flow path to the Reactor Coolant System.

Alternate Testing: These check valves will be exercised during cold shutdown.

Additional Testing: These check valves will be exercised quarterly during the performance of associated pump tests.



RELIEF REQUEST BASIS

SYSTEM: Auxiliary Coolant, Residual Heat Removal

3. Valve: MOV-4-751.
Category: B (previously categorized as A)
Class: 1

Function: Provides a flow path from the Reactor Coolant System to the Residual Heat Removal System for removal of decay heat from the reactor core.

Test Requirement: IWV-3420

Basis for Relief: Seat leak testing this valve when fuel is in the reactor pressure vessel would cause a loss of system function. This would result in violation of the current plant Technical Specifications.



RELIEF REQUEST BASIS

SYSTEM: Main Steam

1. Valve: POV-4-2604, POV-4-2605, and POV-4-2606
Category: C
Class: 2

Function: Provides the flow path from the associated steam generator to the main steam line header.

Test Requirement: IWV-3520

Basis for Relief: Testing any one of these valves during plant operation would isolate the associated steam generator from the main steam line header which would result in a reactor trip.

Alternate Testing: These valves will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Condensate and Feedwater

1. Valve: MOV-4-1425, MOV-4-1426, and MOV-4-1427
Category: B
Class: 2

Function: Provides the flow path from the associated steam generator secondary side to the Sampling System.

Test Requirement: IWV-3410

Basis for Relief: The failure of any one of these valves in the closed position, by testing during plant operation, would result in a loss of ability to sample the associated steam generator; thereby, affecting the ability to verify proper chemistry control and to detect radioactivity.

Alternate Testing: These valves will be tested during cold shutdowns.

2. Valve: CV-4-6275A, CV-4-6275B, and CV-4-6275C
Category: B
Class: 2

Function: Provides the flow path from the associated steam generator secondary side to the Blowdown system.

Test Requirement: IWV-3410

Basis for Relief: The failure of any one of these valves in the closed position, by testing during plant operation, would result in a loss of ability to blowdown the associated steam generator; thereby, affecting the ability to maintain proper chemistry control

Alternate Testing: These valves will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Condensate and Feedwater

3. Valve: CV-4-2900, CV-4-2901, and CV-4-2902
Category: C
Class: 2

Function: Prevents reverse flow from the associated steam generator to the Feedwater System.

Test Requirement: IWV-3520

Basis for Relief: Testing any one of these valves, during plant operation would cause an interruption of feedwater flow to the associated steam generator; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: These valves will be tested during cold shutdowns.

Valve: CV-4-2816, CV-4-2817, CV-4-2818
CV-4-2831, CV-4-2832, and CV-4-2833
Category: B
Class: 2

Function: Provides a flow path from the Auxiliary Feedwater Pump Discharge Header to the Steam Generator Nos. 4A, 4B, or 4C.

Test Requirement: IWV-3410

Basis for Relief: These valves are flow controlled modulating valves. therefore, valve stroke-time is not essential to fulfill their safety related function.

Alternate Testing: Exercising these valves quarterly will demonstrate that the moving parts of the valve function satisfactorily.



RELIEF REQUEST BASIS

SYSTEM: Service Air (Lube Oil, Service and Instrument Air)

1. Valve: 4-40-204
Category: A
Class: 2

Function: Provides the flow path from the Service Air System to the containment.

Test Requirement: IWV-3410

Basis for Relief: The failure of the valve in the open position, by testing during plant operation, would cause a loss of containment integrity.

Alternate Testing: This valve will be tested during cold shutdowns.

2. Valve: 4-40-205
Category: AC
Class: 2

Function: Prevents reverse flow from the containment atmosphere to the Service Air System located outside containment.

Test Requirement: IWV-3520

Basis for Relief: The failure of this valve in the open position, by testing during plant operation, would cause a loss of containment integrity.

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Service Air (Lube Oil, Service and Instrument Air)

3. Valve: 4-40-203
Category: B
Class: 2

Function: Provides the flow path from the Service Air System to the containment (valve in series with 4-40-204).

Test Requirement: IWV-3410

Basis for Relief: This valve is required to be tested only to provide the necessary flow path to valve 4-40-204 and will be tested during cold shutdowns (See 4-40-204 RRB No. 1).

Alternate Testing: This valve will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Instrument Air (Lube Oil, Service and Instruemtn Air)

4. Valve: 4-40-336
Category: AC
Class: 2

Function: Prevents reverse flow from the Instrument Air System, inside containment, to the Instrument Air system, located outside containment.

Test Requirement: IWV-3520

Basis for Relief: Testing this valve during plant operation would interrupt the instrument air supply to the components located inside containment that require instrument air for proper operation; thereby, placing the plant in an unsafe mode of operation.

Testing this valve during cold shutdown would interrupt the instrument air supply to the components located inside containment that require instrument air to maintain the plant in a safe shutdown condition; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: This valve will be tested during refueling shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Primary Make-up and Containment Cooling Water

1. Valve: MOV-4-1417 and MOV-4-1418
Category: B
Class: 2

Function: Provides the component cooling water supply (MOV-4-1417) and return (MOV-4-1418) flow paths for the normal containment coolers, the control rod drive mechanism coolers, and the primary shield cooling coils.

Test Requirement: IWV-3410

Basis for Relief: Testing these valve during plant operation would cause interruption of cooling water to the normal containment cooler, the control rod drive mechanism coolers, and the primary shield cooling coils. This action could result in damage to the control rod drive mechanisms and associated equipment; thereby, placing the plant in an unsafe mode of operation.

Alternate Testing: These valves will be tested during cold shutdowns.

RELIEF REQUEST BASIS

SYSTEM: Containment Ventilation

1. Valve: HV-4-1 and HV-4-3
Category: B
Class: 2

Function: Provides redundant flow paths for Post Loss of Coolant Accident purge.

Test Requirement: IWV-3410

Basis for Relief: In the event that maintenance is required upon the failure of either of these valves, by testing during plant operation, it would cause a loss of containment integrity.

Alternate Testing: These valves will be tested during cold shutdowns.

2. Valve: HV-4-2 and HV-4-4; PAHM-4-001A and PAHM-4-001B
Category: A
Class: 2

Function: Provides redundant flow paths for Post Loss of Coolant Accident purge.

Test Requirement: IWV-3410

Basis for Relief: The failure of either of these valves in the open position, by testing during plant operation, would cause a loss of containment integrity.

Alternate Testing: These valves will be tested during cold shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Containment Ventilation

3. Valve: 4-11-003
Category: AC
Class: 2

Function: Prevents reverse flow from the containment atmosphere to the Containment Gas and Particulate Radioactivity Detection System.

Test Requirement: IWV-3520

Basis for Relief: The Tech Specs require two independent systems to monitor reactor coolant leakage; one of which has to be sensitive to radioactivity. Testing this valve during plant operation would cause an interruption of the Reactor Coolant Leak Detection System which is sensitive to radioactivity.

Similarly, testing this valve during cold shutdown would result in an interruption of the Detection System which is sensitive to radioactivity.

Alternate Testing: This valve will be tested during refueling shutdowns.

4. Valve: PAHM-4-002A and PAHM-4-002B
Category: A
Class: 2

Function: Prevents reverse flow from the containment atmosphere to the Containment Gas and Particulate Radioactivity Detection System.

Test Requirement: IWV-3520

Basis for Relief: The Tech Specs require two independent systems to monitor reactor coolant leakage; one of which has to be sensitive to radioactivity. Testing this valve during plant operation would cause an interruption of the Reactor Coolant Leak Detection System which is sensitive to radioactivity.

Similarly, testing this valve during cold shutdown would result in an interruption of the Detection System which is sensitive to radioactivity.

Alternate Testing: This valve will be tested during refueling shutdowns.



RELIEF REQUEST BASIS

SYSTEM: Containment Ventilation

5. Valve: SV-4-2911, SV-4-2913, AND SV-4-2912
Category: A
Class: 2

Function: Provides flow path from the containment atmosphere to the containment gas and particulate radio-activity detection system and return to containment.

Test Requirement: IWV-3300

Basis for Relief: These self contained, completely enclosed solenoid valves have no external valve position indication. Therefore direct observation of valve position indication is impractical.

Alternate Testing: These valves will be checked during local leak rate tests to verify that remote valve indications accurately reflect valve operation.

RELIEF REQUEST BASIS

SYSTEM: Diesel Oil

1. Valve: SV-3522A and SV-3522B
Category: B
Class: 2

Function: Provides the flow path from the Emergency Diesel - generator Diesel Oil Day Tank to the Skid-mounted Diesel Oil Tank.

Test Requirement: IWV-3410

Basis for Relief: These self-contained, completely enclosed solenoid valves have no external valve position indicators. Therefore, stroke-time measurements and valve position verification is impractical.

Alternate Testing: An increase in level in the Skid-mounted Diesel Oil Tank while exercising these valves quarterly will demonstrate that the moving parts of the valve function satisfactorily.

2. Valve: CV-2046A and CV-2046B
Category: B
Class: 3

Function: Provides the flow path from the Emergency Diesel-generator Diesel Oil Transfer Pump Discharge Header to the Diesel Oil Day Tanks.

Test Requirement: IWV-3410

Basis for Relief: Valve stroke-time is not essential for the valve to fulfill its safety related function.

Alternate Testing: Exercising these valves quarterly will demonstrate that the moving parts of the valve function satisfactorily.



II. PUMP TEST PROGRAM

II.A. Table IV - Test Parameters

This subsection outlines the pumps which are provided an emergency power source.

II.B. Test Interval

In accordance with Relief Request to sub-section IWP-3400, the quarterly test interval as defined in this pump test program is only inclusive of operational modes Power Operation, Hot Standby, or Hot Shutdown (as appropriate) as defined in the Turkey Point Unit No. 4 Technical Specifications.

III.C. Attachment A-2

Relief Request Basis



FLORIDA POWER & LIGHT COMPANY
TABLE IV
PUMP TEST PROGRAM

PUMP	PUMP NO.	TEST PARAMETERS						
		Speed, H	Inlet Pressure P ₁	Differential Pressure DELTA P	Flow Rate Q	Vibration Amplitude, V	Bearing Temperature T _b ⁽¹⁾	RRB NO.
Auxiliary Feed	P2A	Yes	Yes	Yes	Yes ⁽³⁾	Yes	Yes	---
	P2B	Yes	Yes	Yes	Yes ⁽³⁾	Yes	Yes	---
	P2C	Yes	Yes	Yes	Yes ⁽³⁾	Yes	Yes	---
Intake Cooling Water	4-P9A	No	Yes	Yes	Yes ⁽³⁾	Yes	Yes	2
	4-P9B	No	Yes	Yes	Yes ⁽³⁾	Yes	Yes	2
	4-P9C	No	Yes	Yes	Yes ⁽³⁾	Yes	Yes	2
Residual Heat Removal	4-P210A	No	Yes	Yes	No ⁽²⁾	Yes	No	1&3
	4-P210B	No	Yes	Yes	No ⁽²⁾	Yes	No	1&3
Component Cooling	4-P211A	No	Yes	Yes	Yes ⁽³⁾	Yes	No	---
	4-P211B	No	Yes	Yes	Yes ⁽³⁾	Yes	No	---
	4-P211C	No	Yes	Yes	Yes ⁽³⁾	Yes	No	---
Containment Spray	4-P214A	No	Yes	Yes	No ⁽²⁾	Yes	Yes	1
	4-P214B	No	Yes	Yes	No ⁽²⁾	Yes	Yes	1



FLORIDA POWER & LIGHT COMPANY
TABLE IV
PUMP TEST PROGRAM

PUMP	PUMP NO.	TEST PARAMETERS						
		Speed, H	Inlet Pressure P_1	Differential Pressure DELTA P	Flow Rate Q	Vibration Amplitude, V	Bearing Temperature $T_b^{(1)}$	RRB NO.
High Head Safety Injection	P215A	No	Yes	Yes	No (2)	Yes	Yes	1
	P215B	No	Yes	Yes	No (2)	Yes	Yes	1
	P215C	No	Yes	Yes	No (2)	Yes	Yes	1
	P215D	No	Yes	Yes	No (2)	Yes	Yes	1

NOTE (1): BEARING TEMPERATURE, $T_{(b)}$, IS MEASURED ONLY ON THE ANNUAL TEST.

NOTE (2): FIXED HYD. RESISTANCE SYSTEM

NOTE (3): VARIABLE HYD. RESISTANCE SYSTEM.



ATTACHMENT A-2

TURKEY POINT UNIT NO. 4

RELIEF REQUEST BASIS



RELIEF REQUEST BASIS

1) PUMPS:

Residual Heat Removal Pump No. 4A	(4-P210A)
Residual Heat Removal Pump No. 4B	(4-P210B)
Containment Spray Pump No. 4A	(4-P214A)
Containment Spray Pump No. 4B	(4-P214B)
High Head Safety Injection Pump No. A	(P215A)
High Head Safety Injection Pump No. B	(P215B)
High Head Safety Injection Pump No. C	(P215C)
High Head Safety Injection Pump No. D	(P215D)

Test Requirement: IWP-4600 Flow Measurement

Basis for Relief: Section XI, ASME Boiler & Pressure Vessel Code, requires measurement of flow rate using a rate or quantity meter installed in the pump test circuit. These pumps use a fixed hydraulic resistance system with an orifice installed in the pump recirculation line (IWP-1400).

Alternate Testing: Measure differential pressure (IWP-4240) across the pump during the quarterly pump test. measured differential pressure across the pump shall then be compared to the established reference value. This provides for an indirect measure of flow rate and verifies the operational readiness of the pump (IWP-1500).



RELIEF REQUEST BASIS

2) PUMPS:

Intake Cooling Water Pump No. 4A	(4-P9A)
Intake Cooling Water Pump No. 4B	(4-P9B)
Intake Cooling Water Pump No. 4C	(4-P9C)

Test Requirement: IWP-4310 (Bearing Temperature Measurement)

Basis for Relief: The pump bearings for these vertical centrifugal pumps are located inside the pump housing which is submerged. The bearings are water lubricated with no provisions for temperature measurement.

Alternate Testing: Measurement of vibration amplitude and the other required parameters during each pump test will provide for the detection of changes in the mechanical characteristics of the pump.

3) PUMPS:

Residual Heat Removal Pump No. 4A	(4-P210A)
Residual Heat Removal Pump No. 4B	(4-P210B)

Test Requirement: IWP-4310 (Bearing Temperature Measurement)

Basis for Relief: The pump bearings for these vertical centrifugal pumps are located inside the pump housing which is submerged. The bearings are water lubricated with no provisions for temperature measurement.

Alternate Testing: Measurement of vibration amplitude and the other required parameters during each pump test will provide for the detection of changes in the mechanical characteristics of the pump.

PRATT

HENRY PRATT COMPANY

creative engineering for fluid systems

401 SOUTH HIGHLAND AVENUE • AURORA, ILLINOIS 60607

~ CONTAINMENT ISOLATION / PURGE VALVE ANALYSIS

Florida Power & Light

Turkey Point Units 3 & 4

- 48" RIA Valve with Bettis Actuator

Customer P.O.: 5177-099

Pratt Order No.: D-027256

[illegible]

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

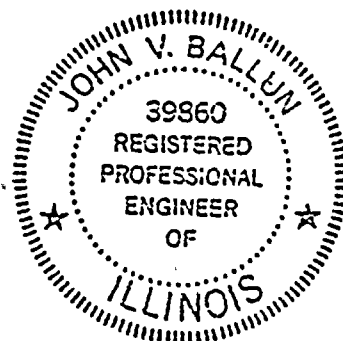
Valve Tag Nos. POV-3-2602 POV-3-2603
POV-4-2602 POV-4-2603

General Arrangement Drawing E-586 Rev. 1

Prepared by: Rao N. Kaza
Date: 9-15-81

Reviewed by: T. J. Whona
Date: 9-16-81

Certified by: J. V. Ballun
Date: 9-18-81



REGULATORY DOCKET FILE COPY

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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 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 1×10^8 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, in²

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

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

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

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 (≈ 1.2 for air/steam mix)

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

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

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

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

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

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

D-27256-2

TORQUE TABLE 1

9 / 14 / 81

JOB: FLOR. PWR; TURKEY-PT P2-VARIABLE SIZE ADJUSTED (REYNOLDS 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.)

ABSD. 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 FRICTN.: 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/FT³-MIN. .157531 LB/FT³-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. (75-80 DEG.)

IN SUBSONIC RANGE LIMITS-ONLY; SEE FORMULATIONS. -PER TESTS H. PRATT

THIS TO. AT 72 DEG. SYMM. DISC (68=OFFSET SHAFT) CT=T/D³/P2 (ABS)

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

ANGLE	P1	P2	DELP	PRESS.	FLOW	FLOW	TD	TB+TH	TIME (LOCAL)
APPRX.PSIA	PSIA	PSI	RATIO		(SCFM)	(LB/MIN)	-----INCHLBS-----	TD-TB-TH	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 3.39
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	440742 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	359358 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	244630	1744	242936 5.35
40	53.33	16.45	36.88	.308	369400	20306	178488	1787	176700 5.38
35	53.72	15.67	38.04	.292	286493	15749	135682	1818	133963 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 5.78
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	3656	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, PSIG

FIG. 14.3.4-2

CONTAINMENT PRESSURE EQUIVALENT
FOR A RANGE OF AMPLITUDES (CM) OF CYCLES (SEC)

60

50

40

30

20

10

0

2X DUC 1000

5 F12

5 F12

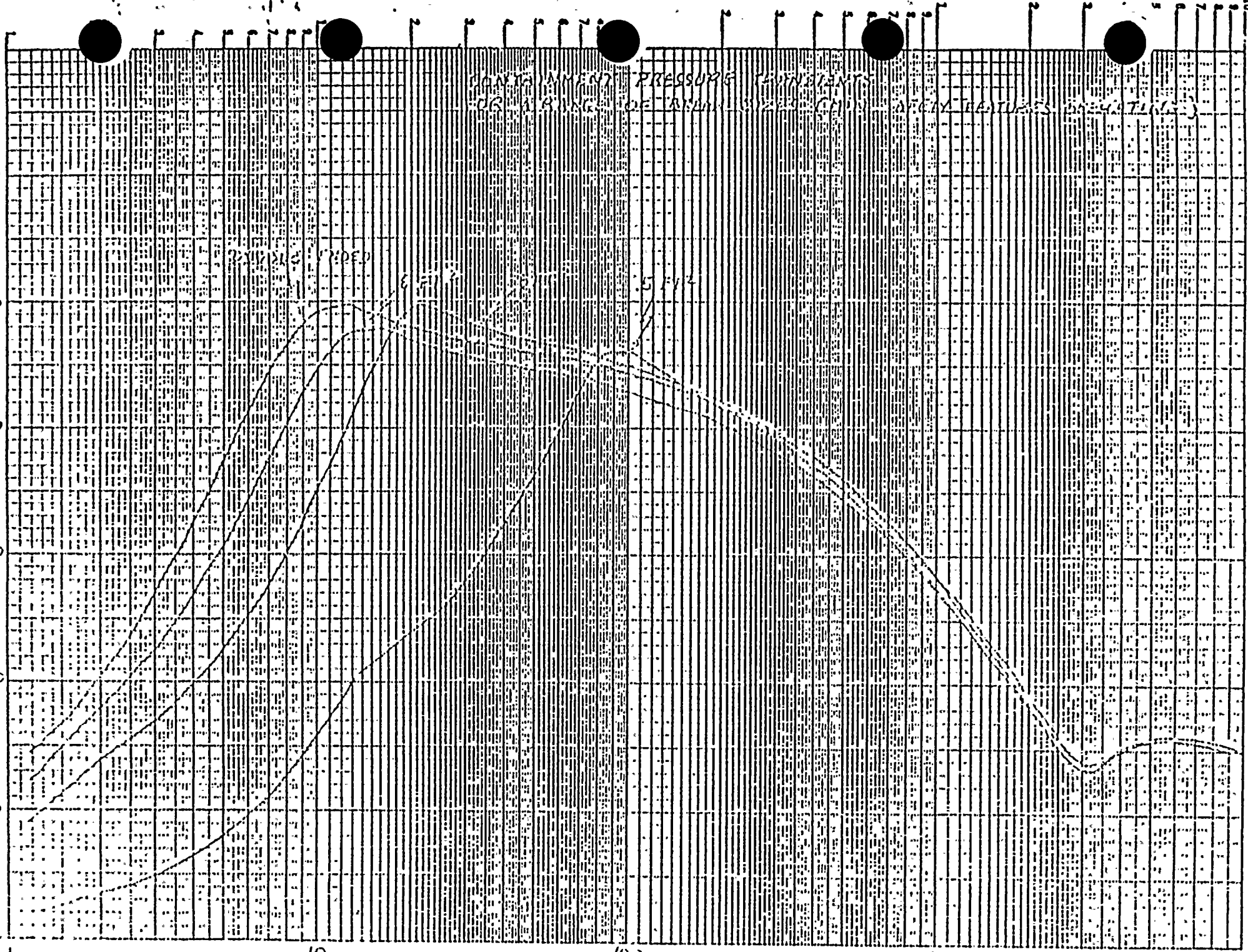
5 F12

10

100

1000

TIME, SECONDS





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 of the valve (one for each valve size), or

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. 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

RECEIVED

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 SAE Handbook of Fundamentals, we compile the following flow resistance coefficients:

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



<u>VALVE</u>	<u>VALVE</u> <u>ANGLE</u>	<u>OPEN</u> <u>ANGLE</u>	<u>COEF.</u>	<u>REF.</u>
	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)



Mr. T. J. Wrona
Page 2
V-241

- 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 ft² 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
Project Engineer

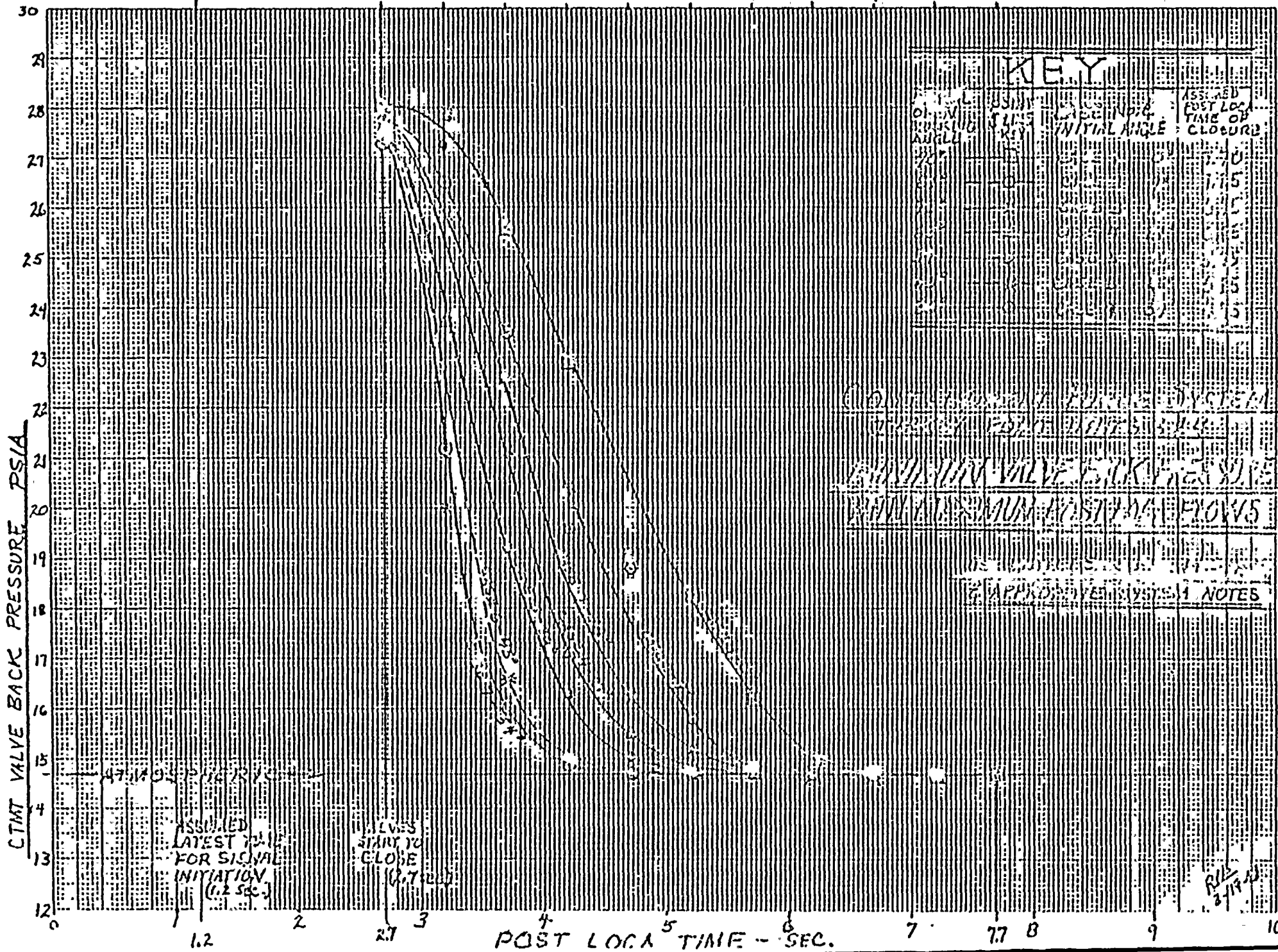
AWW/RVB:mfa

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

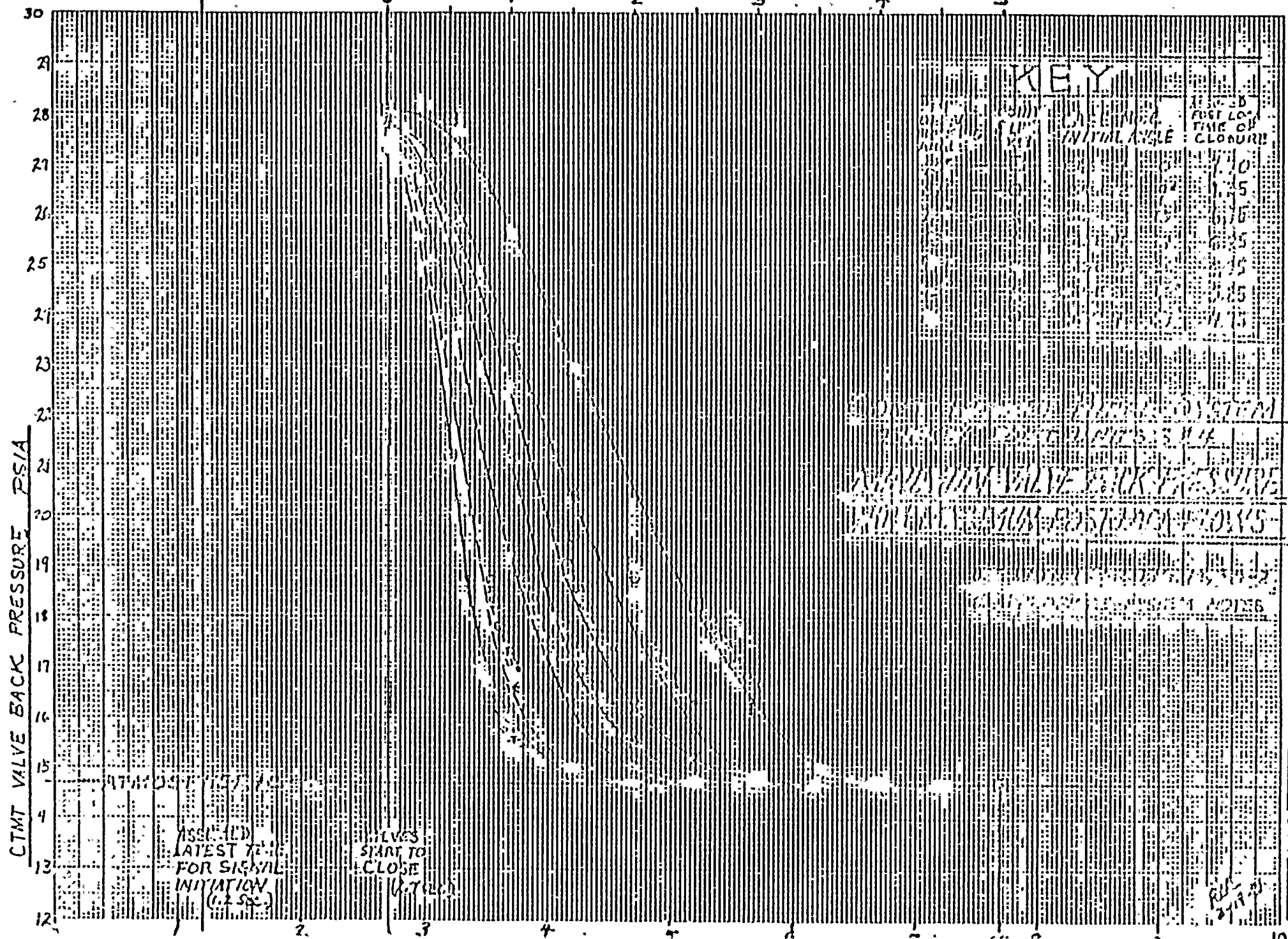


VALVE CLOSURE TIME - SEC.





1	2	3	4
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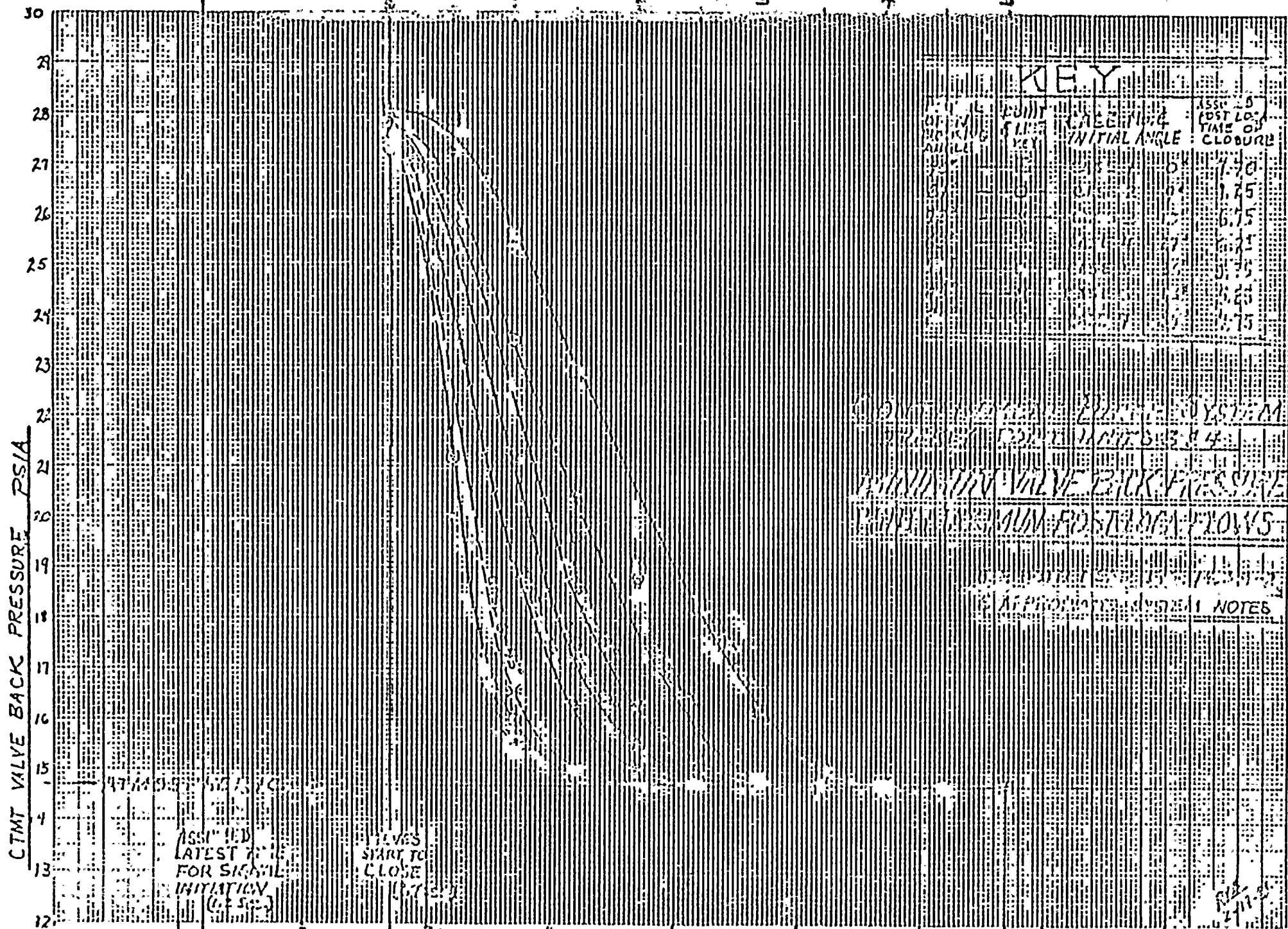


108 5177-152

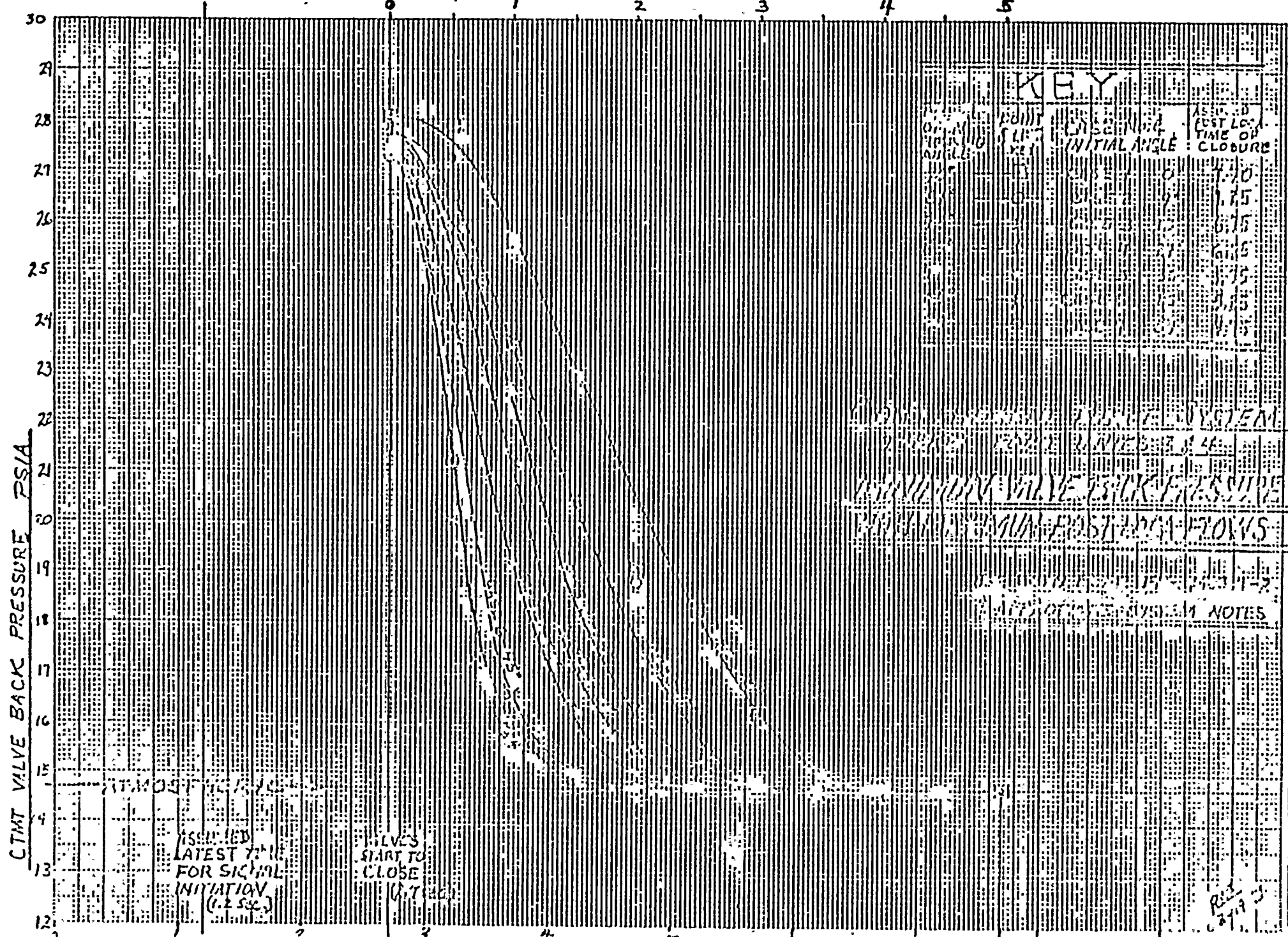
Chlor.

SH7-17

VALVE CLOSURE TIME - SEC.



VALVE CLOSURE TIME - SEC.



KEY			
POINT OF VALVE CLOSURE	INITIAL ANGLE	ASSIGNED TIME OF CLOSURE	
1	0	4.70	
2	0	1.75	
3	0	6.75	
4	0	6.75	
5	0	6.75	
6	0	6.75	
7	0	6.75	
8	0	6.75	
9	0	6.75	
10	0	6.75	

1.0 IN. DIAMETER VALVE SYSTEM
2.0 IN. DIAMETER VALVE SYSTEM
3.0 IN. DIAMETER VALVE SYSTEM
4.0 IN. DIAMETER VALVE SYSTEM
5.0 IN. DIAMETER VALVE SYSTEM
6.0 IN. DIAMETER VALVE SYSTEM
7.0 IN. DIAMETER VALVE SYSTEM
8.0 IN. DIAMETER VALVE SYSTEM
9.0 IN. DIAMETER VALVE SYSTEM
10.0 IN. DIAMETER VALVE SYSTEM

NOTES

JOB 5177-152

CALC.

SHT. 17



ATTACHMENT 2

Nuclear

Purge Valve

Stress

Analysis

SEISMIC ANALYSIS
FOR 48 INCH
NUCLEAR PURGE VALVE



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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 grotch 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.



ANALYSIS NOMENCLATURE

B ₉	Distance to outer fiber of bonnet from shaft on x axis, in.
C	A factor depending upon the method of attachment of head, shell dimensions, and other items as listed in NC-3225.2, dimensionless (Fig. NC-3225.1 thru Fig. NC-3225.3)
C _b	Stress index for body bending secondary stress resulting from moment in connected pipe (NB-3545.2 (b))
C _p	Stress index for body primary plus secondary stress, inside surface, resulting from internal pressure (NB-3545.2(a))
C ₂	Stress index for thermal secondary membrane stress resulting from structural discontinuity.
C ₃	Stress index for maximum secondary membrane plus bending stress resulting from structural discontinuity
C ₆	Product of Young's modulus and coefficient of linear thermal expansion, at 500°F, psi/°F (NB-3550)
C ₇	Distance to outer fiber of disc for bending along the shaft, in.
C ₈	Distance to outer fiber of disc for bending about the shaft, in.
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.

ANALYSIS NOMENCLATURE

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	$1/2 \times$ 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	$W_3 g_x$ --Seismic force along x axis due to seismic acceleration acting on operator extended mass, pounds.
F_y	$W_3 g_y$ --Seismic force along y axis due to seismic acceleration acting on operator extended mass, pounds.
F_z	$W_3 g_z$ --Seismic force along z axis due to seismic acceleration acting on operator extended mass, pounds.
g	Gravitational acceleration constant, inch-per-second ²
G_b	Valve body section bending modulus at crotch region (NB-3545.2(b)), in ³
G_d	Valve body section area at crotch region (NB-3545.2(b)), in ²
G_t	Valve body section torsional modulus at crotch region (NB-3545.2(b)), in ³
g_x	Seismic acceleration constant along x axis
g_y	Seismic acceleration constant along y axis
g_z	Seismic acceleration constant along z axis
h_g	Gasket moment arm, equal to the radial distance from the center line of the bolts to the line of the gasket reaction (NC-3225), in.
H_1	Disc hub key height, in.
H_2	Top trunnion bolt square, in.
H_3	Bottom trunnion bolt square, in.

ANALYSIS NOMENCLATURE

H ₄	Bonnet bolt square, in.
H ₅	Operator bolt square, in.
H ₆	Bonnet bolt circle, in.
H ₇	Operator bolt circle, in.
H ₈	Bonnet height, in.
H ₉	Actual body wall thickness, in.
I ₁	Bonnet body moment of inertia about x axis, in ⁴
I ₂	Bonnet body moment of inertia about y axis, in ⁴
I ₃	Disc area ₄ moment of inertia for bending about the shaft, in ⁴
I ₄	Disc area ₄ 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 ₄ 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.

ANALYSIS NOMENCLATURE

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	$W_3(g_{zO} + g_{yO})$, operator extended mass seismic bending moment about the x axis, acting at the base of the operator, in-lbs.
M_y	$W_3(g_{xO} + g_{zO})$, operator extended mass seismic bending moment about the y axis, acting at the base of the operator, in-lbs.
M_z	$(W_3(g_{yO} + g_{xO}))$, operator extended mass seismic bending moment about the z axis, in-lbs.
$\overline{M_x}$	$M_x + F_y T_5$, operator extended mass seismic bending moment about the x axis, acting at the bottom of the adapter plate, in-lbs.

ANALYSIS NOMENCLATURE

$\overline{M_y}$	$M_y + F_x T_5$, operator extended mass seismic bending moment about the y axis, acting at the bottom of the adapter, in-lbs.
$\overline{M_x}$	$M_x + F_y (T_5 + H_8) + g y W_4 K_3$, operator extended mass seismic bending moment about the x axis, acting at the base of the bonnet, in-lbs.
$\overline{M_y}$	$M_y + F_x (T_5 + H_8) + g x W_4 K_3$, 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 $hr/100^\circ F/hr/hr$ fluid temperature change rate (NB-3545.3)
NA	Not applicable to the analysis of the system.
N_1	Number of top disc pins
N_2	Number of operator bolts
N_3	Number of trunnion bolts
P_d	Design pressure, psi
P_r	Primary pressure rating, pound
P_s	Standard calculation pressure psi
P_e	Largest value among P_{eb} , P_{ed} , P_{et} , psi
P_{eb}	Secondary stress in crotch region of valve body caused by bending of connection standard pipe, calculated according to NB-3545.2(b), psi
P_{ed}	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
P_{et}	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



ANALYSIS NOMENCLATURE

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 Q_p (NB-3545.2(a)), in.
r_2	Fillet radius of external surface at crotch (NB-3545.2(a)), in.
R_4	Disc radius, in.
R_5	Shaft radius, in.
R_m	Mean radius of body wall, in.
R_6	Radius to 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



ANALYSIS NOMENCLATURE

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.
ΔT_2	Maximum magnitude of the difference in average wall temperatures for walls of thicknesses t_e , T_e , resulting from 100° F/hr fluid temperature change rate, °F.
T_1	Thickness of cover cap behind bolt head, in.
T_2	Thickness of 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.



ANALYSIS NOMENCLATURE

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.

ANALYSIS NOMENCLATURE

z_0	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



ANALYSIS NOMENCLATURE

- 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 S_m for tensile stresses and $.6 S_m$ for shear stresses. The allowable levels are the same whether the calculated stress is a combined stress or results from a single load condition. S_m 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	ALLOWABLE STRESS LEVEL, PSI
BODY	NB-3545.1	Primary Membrane Stress in Crotch Region	P_m	36	ASTM A-36	1611	S_m 12600
		Primary Membrane	P'_m	36	ASTM A-36	1033	S_m 12600
	NB-3542.2	Primary Plus Secondary Stress Due to Internal Pressure	Q_p	38	ASTM A-36	3579	S_m 12600
	NB-3545.2	Pipe Reaction Stress Axial Stress Bending Torsion	P_{ed} P_{eb} P_{et}	38	ASTM A-36	2542 4773 4773	$1.5S_m$ 18900
	NB-3545.2	Thermal Secondary Stress	Q_t	38	ASTM A-36	968	S_m 12600
	NB-3545.2	Primary Plus Secondary Stress	S_n	38	ASTM A-36	6958	$3S_m$ 37800
	NB-3545.3	Normal Duty Fatigue Stress $N_a \geq 2000$	S_p	39	ASTM A-36	7101	$1.5S_m$ 18900
Disc	NB-3546.2	Combined Bending Stress in Disc	$S(1)$	40	ASTM A-36	7682	$1.5S_m$ 18900
Shaft	NB-3545.3	Combined Stress in Shaft	$S(4)$	41	ASTM A-276 Type 316 Condition A	54126	$1.5S_m$ 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) 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) 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) 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" RIA5

Operator Mounting: BARNET

Operator: BETTIS 2744-A-2SR-45

A_F 185.06

B₈ 5.126

F_d 2.5

A_m 10.03

B₉ 5.126

F_x 2100

A₁ N/A

C .3

F_y 2100

A₂ .373

C_b 1

F_z 2800

A₃ .334

C_p 3

G_b 3928.3

A₄ .302

C₂ .42

G_d 330.46

A₅ .606

C₃ .45

G_t 7856.6

A₆ .551

C₆ 2.49

g_x 3

A₇ .334

C₇ 4.625

g_y 3

A₈ .302

C₈ 4.625

g_z 4

A₉ .142

D₁ 48"

H₁ .875

A₁₀ .551

D₂ 4

H₂ 12.25

A₁₁ .606

D₃ 1.25

H₃ 7.4246

A₁₂ .068

D₄ 3.75

H₄ 12.25

A₁₃ .076

D₅ .375

H₅ N/A

B₁ 4.125

D₆ .375

H₆ N/A

B₂ 4

D₇ .75

H₇ 12.375

B₃ .606

D₈ .75

H₈ 8

B₄ .551

D₉ 1

H₉ 2.125

B₅ 20.12

D₁₀ .75

I₁ 296.34

B₆ N/A

D₁₁ 11

I₂ 186.76

B₇ N/A

F_b 625

I₃ 749.04

Dimensional Data (Cont.)

I ₄ <u>749</u>	L ₁₁ <u>.375</u>	R ₄ <u>23.531</u>
I ₅ <u>101400</u>	m <u>3.5</u>	R ₅ <u>2</u>
I ₆ <u>12.5663</u>	M _x <u>8007</u>	R ₆ <u>4.5</u>
I ₇ <u>339.02</u>	M _y <u>8007</u>	R ₇ <u>1.50</u>
I ₈ <u>1.229</u>	M _z <u>0</u>	R _m <u>24.75</u>
J ₁ <u>1.25</u>	$\overline{M_x}$ <u>26187</u>	S <u>30000</u>
J ₂ <u>1.25</u>	$\overline{M_y}$ <u>26187</u>	t _e <u>1.75</u>
J ₃ <u>1.25</u>	$\overline{M_x}$ <u>26187</u>	t _m <u>.59</u>
J ₄ <u>1.25</u>	$\overline{M_y}$ <u>26187</u>	ΔT ₂ <u>Δ</u>
J ₅ <u>N/A</u>	N _a <u>2000</u>	T ₁ <u>1</u>
J ₆ <u>N/A</u>	N ₁ <u>2</u>	T ₂ <u>0.25</u>
K ₁ <u>4.75</u>	N ₂ <u>8</u>	T ₃ <u>.438</u>
K ₂ <u>2</u>	N ₃ <u>4</u>	T ₄ <u>1</u>
K ₃ <u>3.4117</u>	P _d <u>75</u>	T ₅ <u>N/A</u>
K ₄ <u>14.75</u>	P _r <u>75</u>	T ₆ <u>1</u>
K ₅ <u>14.75</u>	P _s <u>85</u>	T ₇ <u>1</u>
K ₆ <u>6.63</u>	Q _{T1} <u>550</u>	T ₈ <u>Δ89099</u>
L ₁ <u>20</u>	Q ₁ <u>1.5</u>	T ₉ <u>.438</u>
L ₂ <u>1</u>	Q ₂ <u>6.5</u>	T ₁₀ <u>1</u>
L ₃ <u>1.37</u>	Q ₃ <u>8.5</u>	T ₁₁ <u>1.75</u>
L ₄ <u>1.62</u>	Q ₄ <u>2.0</u>	T ₁₂ <u>1</u>
L ₅ <u>N/A</u>	Q ₅ <u>2.688</u>	U ₁ <u>19</u>
L ₆ <u>.50</u>	Q ₆ <u>9.926</u>	U ₂ <u>19</u>
L ₇ <u>N/A</u>	Q ₇ <u>3.125</u>	U ₃ <u>.25</u>
L ₈ <u>.62</u>	r <u>4.625</u>	U ₄ <u>N/A</u>
L ₉ <u>10.5</u>	z _i <u>23.6875</u>	U ₅ <u>N/A</u>
L ₁₀ <u>3.50</u>	r ₂ <u>.63</u>	U ₆ <u>24.38</u>



Dimensional Data (Cont.)

V_1 NIA
 V_2 NIA
 V_3 NIA
 V_4 NIA
 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 NIA
 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°F. Seismic accelerations are simultaneously applied in each of three mutually perpendicular directions.

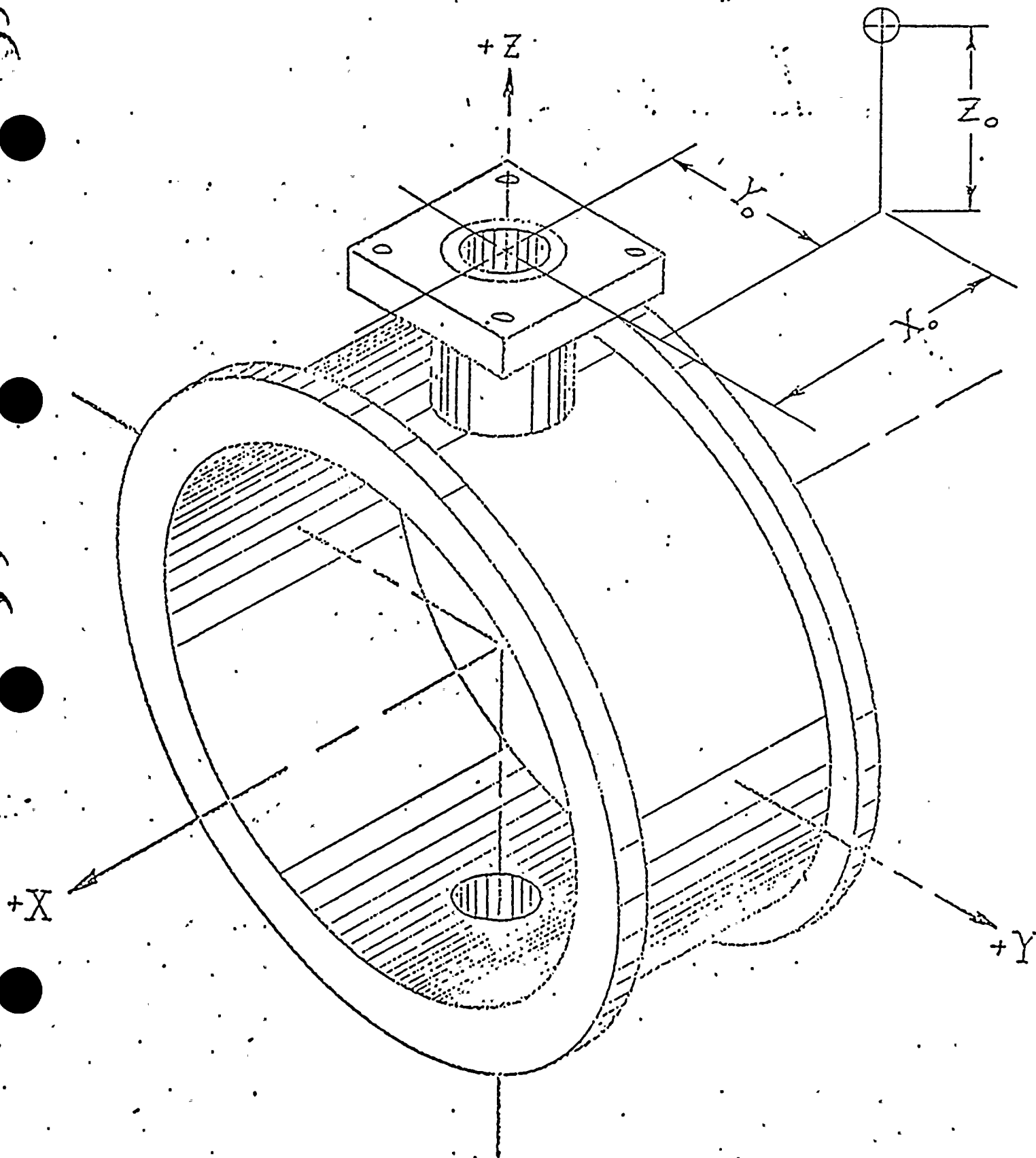
Seismic loads are made an integral part of the analysis by the inclusion of the acceleration constants g_x , g_y , g_z . The symbols g_x , g_y , g_z represent accelerations in the x, y and z directions respectively. These directions are defined with respect to the valve body centered coordinate system 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.

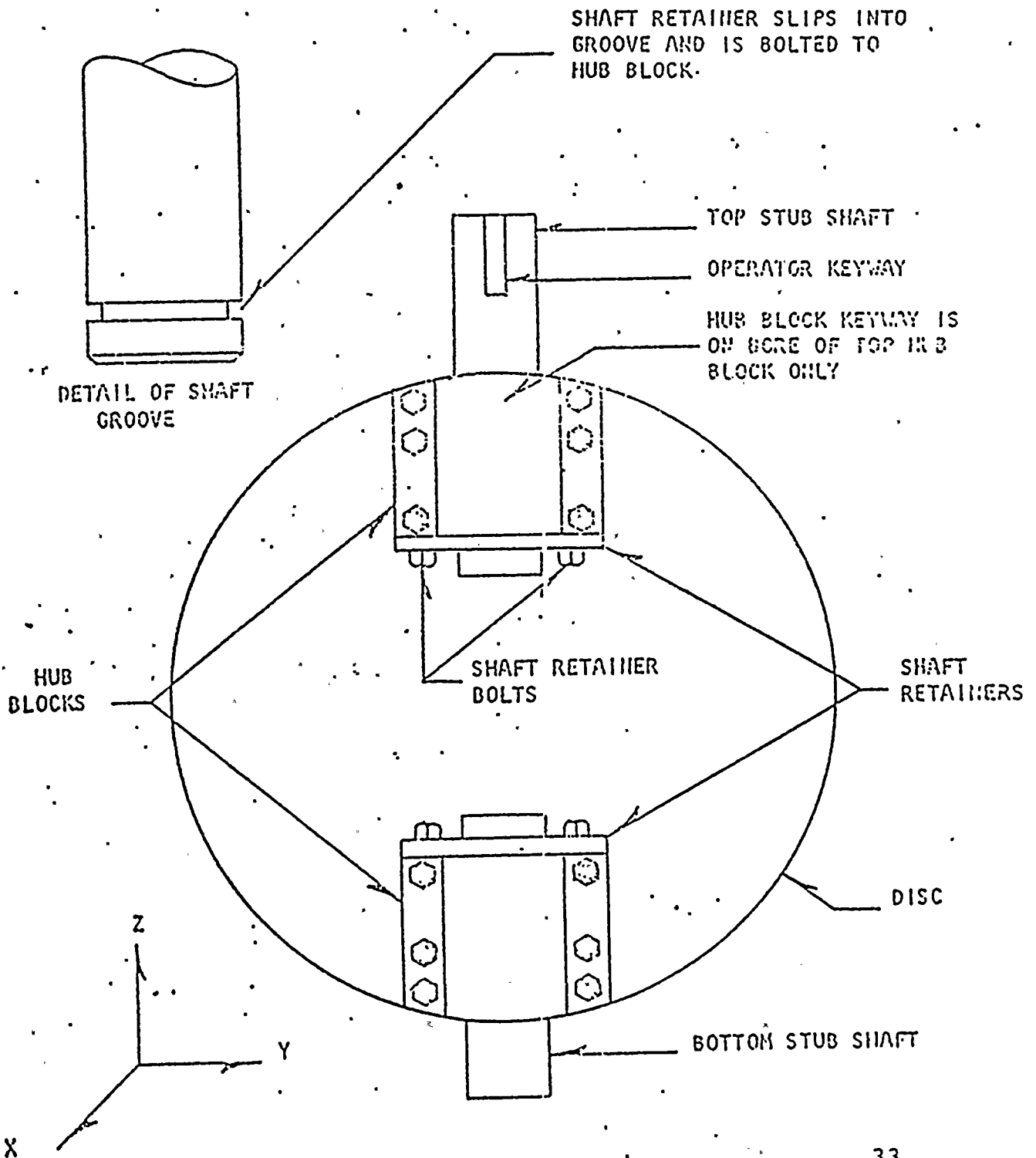
Analysis Introduction

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 completely equivalent to a 1g seismic load.

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

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





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.

1



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.

$$P_m = (A_f/A_m + .5) P_s$$

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 (A_m) which decreases the A_f/A_m ratio and reduces the result. For a section not containing the trunnion, the fluid area to metal area ratio (A_f/A_m) reduces to the body inside radius to the shell thickness (R_m/H_g) since the depths are the same. The resulting membrane stress equation is then:

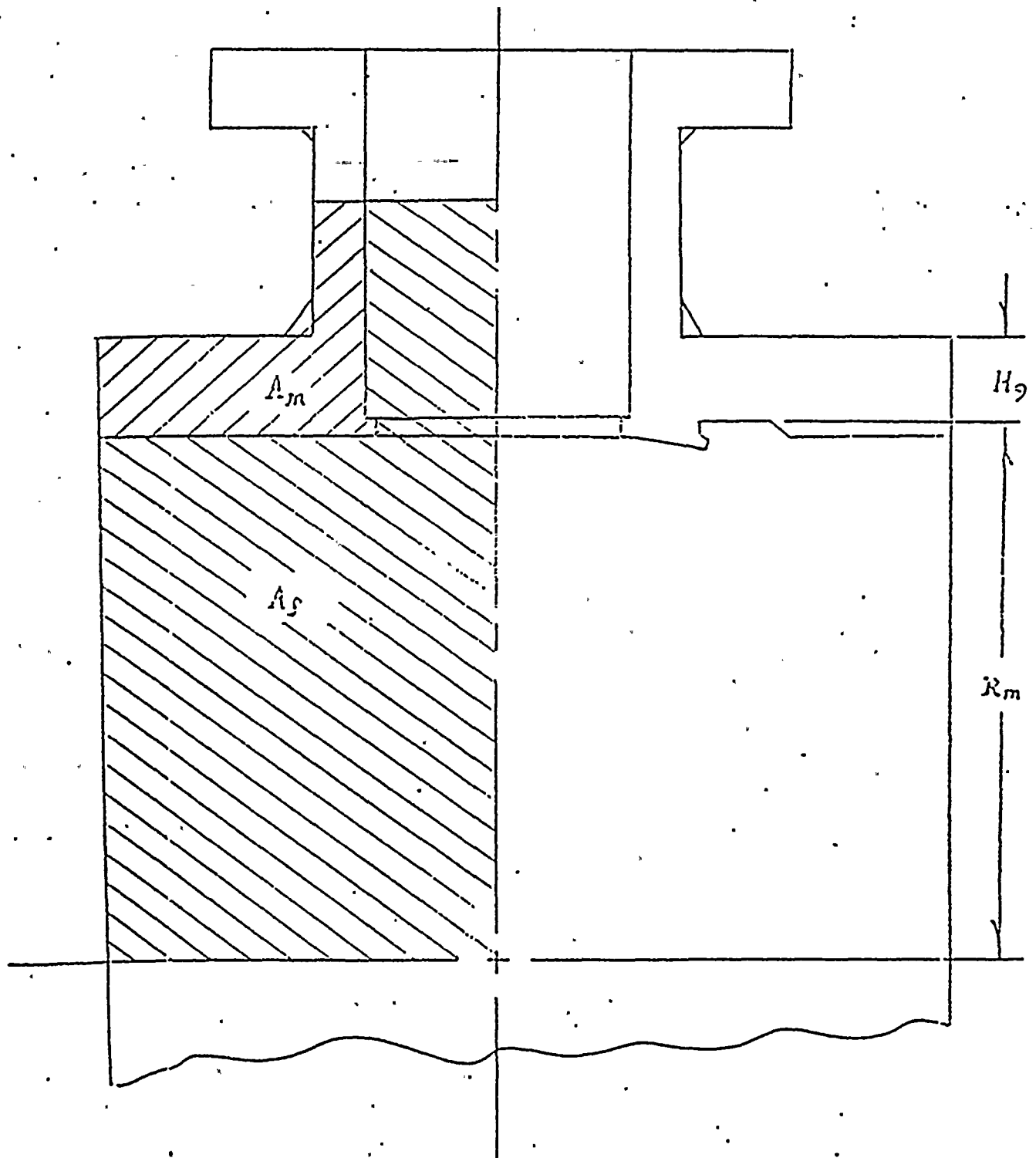
$$P_m = (R_m/H_g + .5) P_s$$

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.

Figure 3



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

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

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 stress are mentioned in this section. For a flat plate such as the valve disc, membrane stress is not defined until 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:

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

where

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

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



SHAFT ANALYSIS

The shaft is analyzed in accordance with para. NR-3546.3 of section III of the code. The shaft loading is a combination of seismic, pressure, and operating loads. Maximum torsional loading is either a combination of seating and bearing torque or bearing and dynamic torque. Columnar stress 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:

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

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

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

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

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

$$S(10) = 1.333 \left[\frac{.5 \pi R_4^2 P_s + .5 W_2 (g_x^2 + g_y^2)^{1/2}}{\pi R_5^4} \right] = \text{Direct Shear Stress, PSI}$$



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 analyze 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.

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

Where:

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

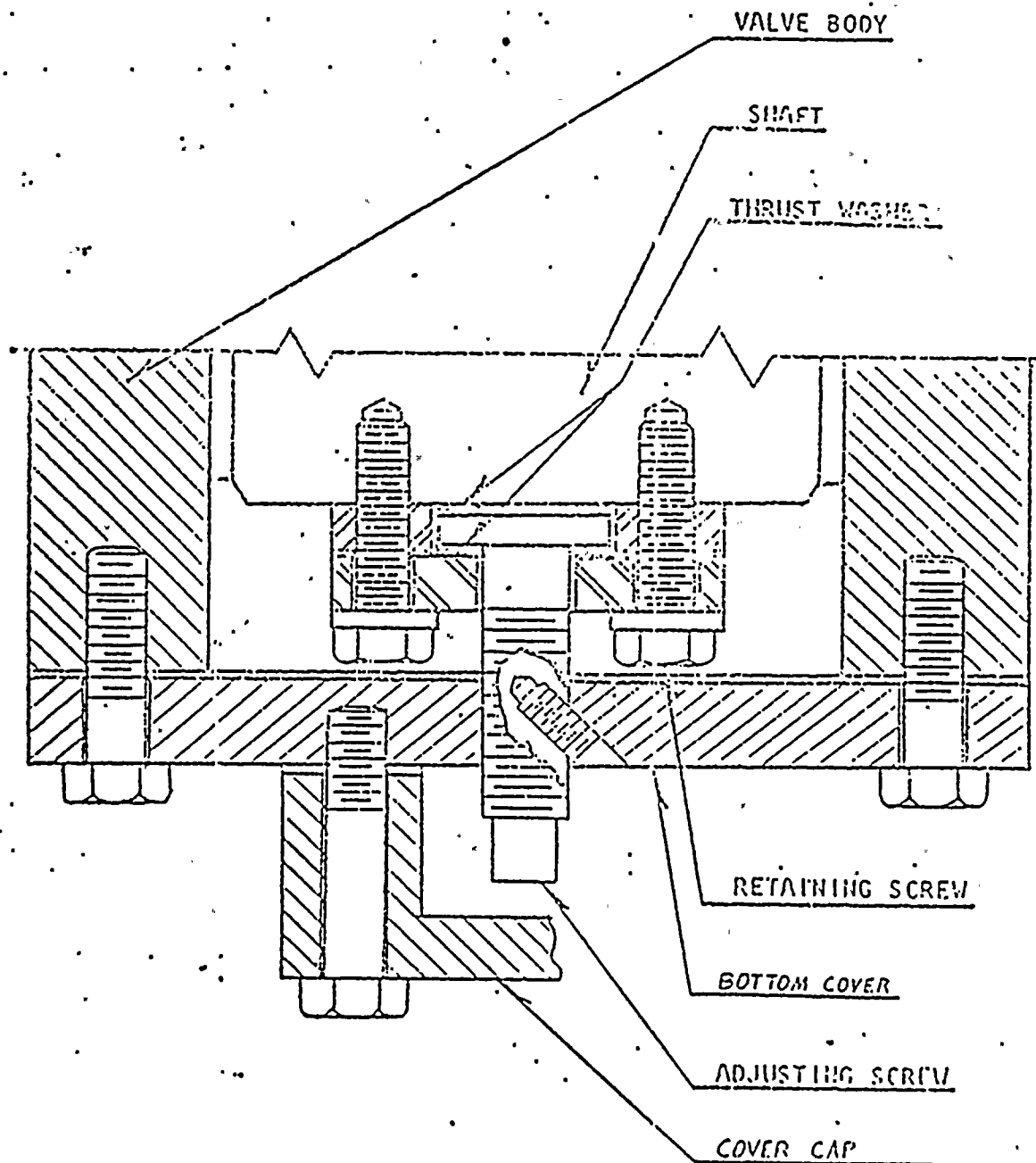
$$S(26) = \frac{16 U_6}{\pi D_{10}^3} = \text{Torsional shear stress due to thrust bearing seismic friction torque.}$$

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

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
A THRUST BEARING ASSEMBLY





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

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

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:

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

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

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:

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

$$S(40) = \frac{3(.785)(D_4 + .25)^2 P_s + W_2 g_z}{4 \pi T_4^2 m} = \text{Tangential stress}$$

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



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.

$$S(42) = \frac{F_z + W_4 \sqrt{g_x^2 + g_y^2 + g_z^2}}{4} + \frac{\overline{M}_x (J_2 + H_2)}{2J_2^2 + 2(J_2 + H_2)^2} + \frac{\overline{M}_y (J_1 + H_2)}{2J_1^2 + 2(J_1 + H_2)^2} \\ \cdot 9\pi L_4 D_7$$

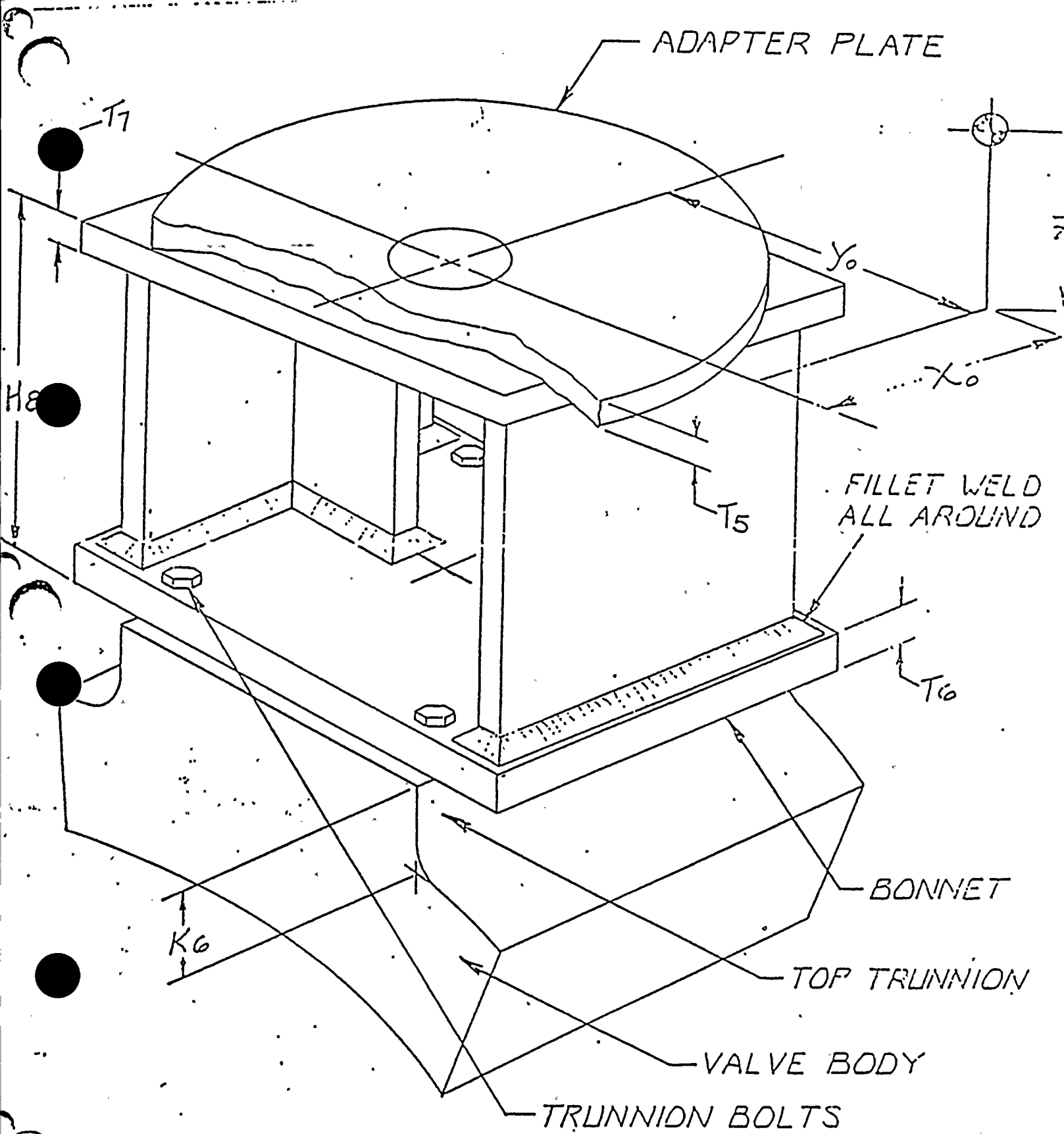
- b. Bearing stress on tapped holes in trunnion.

$$S(43) = \frac{(M_z + T_8)}{4(.707H_2)} + \frac{(F_x^2 + F_y^2)^{\frac{1}{2}}}{4} + \frac{W_4 (g_x^2 + g_y^2)^{\frac{1}{2}}}{4} \\ D_7 L_4$$

- c. Bearing stress on through hole in bonnet.

$$S(44) = \frac{M_z + T_8}{4(.707H_2)} + \frac{(F_x^2 + F_y^2)^{\frac{1}{2}}}{4} + \frac{W_4 (g_x^2 + g_y^2)^{\frac{1}{2}}}{4} \\ D_7 T_6$$







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

$$S(45) = \frac{\frac{F_z + W_4 g_z}{4} + \frac{\overline{M}_x (J_2 + H_2)}{2J_2^2 + 2(J_2 + H_2)^2} + \frac{\overline{M}_y (J_1 + H_2)}{2J_1^2 + 2(J_1 + H_2)^2}}{5.2 D_7 T_6}$$

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,

$$S(47) = \frac{F_z + W_4 g_z}{4A_5} = \text{Direct tensile stress, psi}$$

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

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

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

f. Shear tearout of operator bolt head through bonnet.

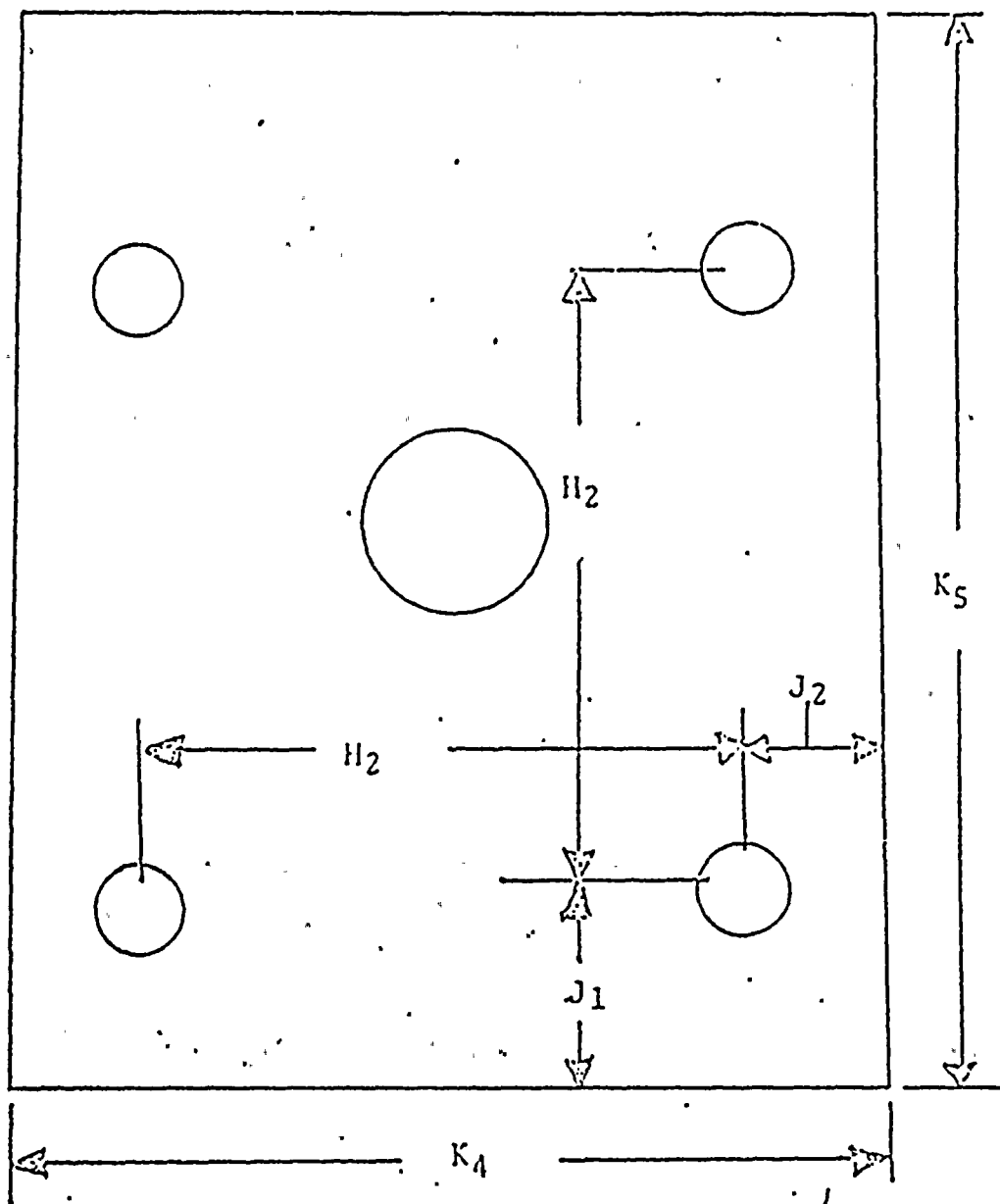
$$S(51) = \frac{\frac{(\overline{M}_x + \overline{M}_y) V_4}{2(V_1^2 + V_2^2 + V_3^2 + V_4^2)} + \frac{F_z}{4}}{5.2 D_8 T_7}$$

g. Bearing stress on through holes in bonnet.

$$S(52) = \frac{M_z + T_8}{.5 H_7 8 T_7 D_8}$$

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,

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

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

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

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

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:

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

= Combined stress in bonnet legs.

Where,

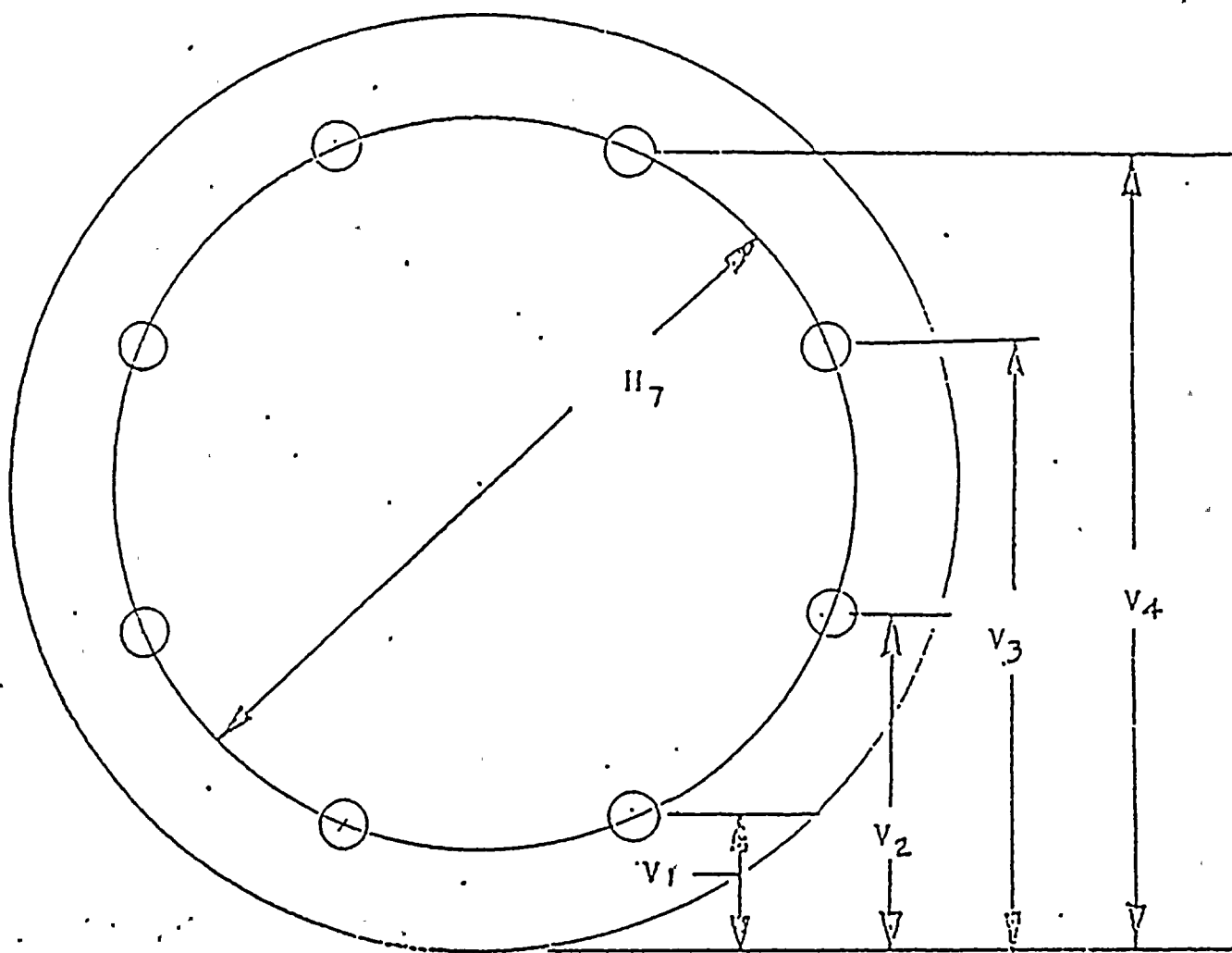
$$S(59) = \frac{F_z + W_4 g_z}{B_5} = \text{Direct tensile stress, psi}$$

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

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

$$S(62) = \frac{T_8 C_0}{K_0} = \text{Shear stress in bonnet body due to torsional load, psi}$$





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, in⁴.

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

BOTTOM BONNET WELDS

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

Where,

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

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

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

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

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

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

TOP BONNET WELDS

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

Where,

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

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

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

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

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

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

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:

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

Where,

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

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



$$S(80) = \frac{(F_x^2 + F_y^2)^{1/2} + W_4(g_x^2 + g_y^2)^{1/2}}{K_4 K_5 - .785 B_2^2} = \text{Direct shear stress, psi}$$

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



TOP TRUNNION ASSEMBLY

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.

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

Where,

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

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

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

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

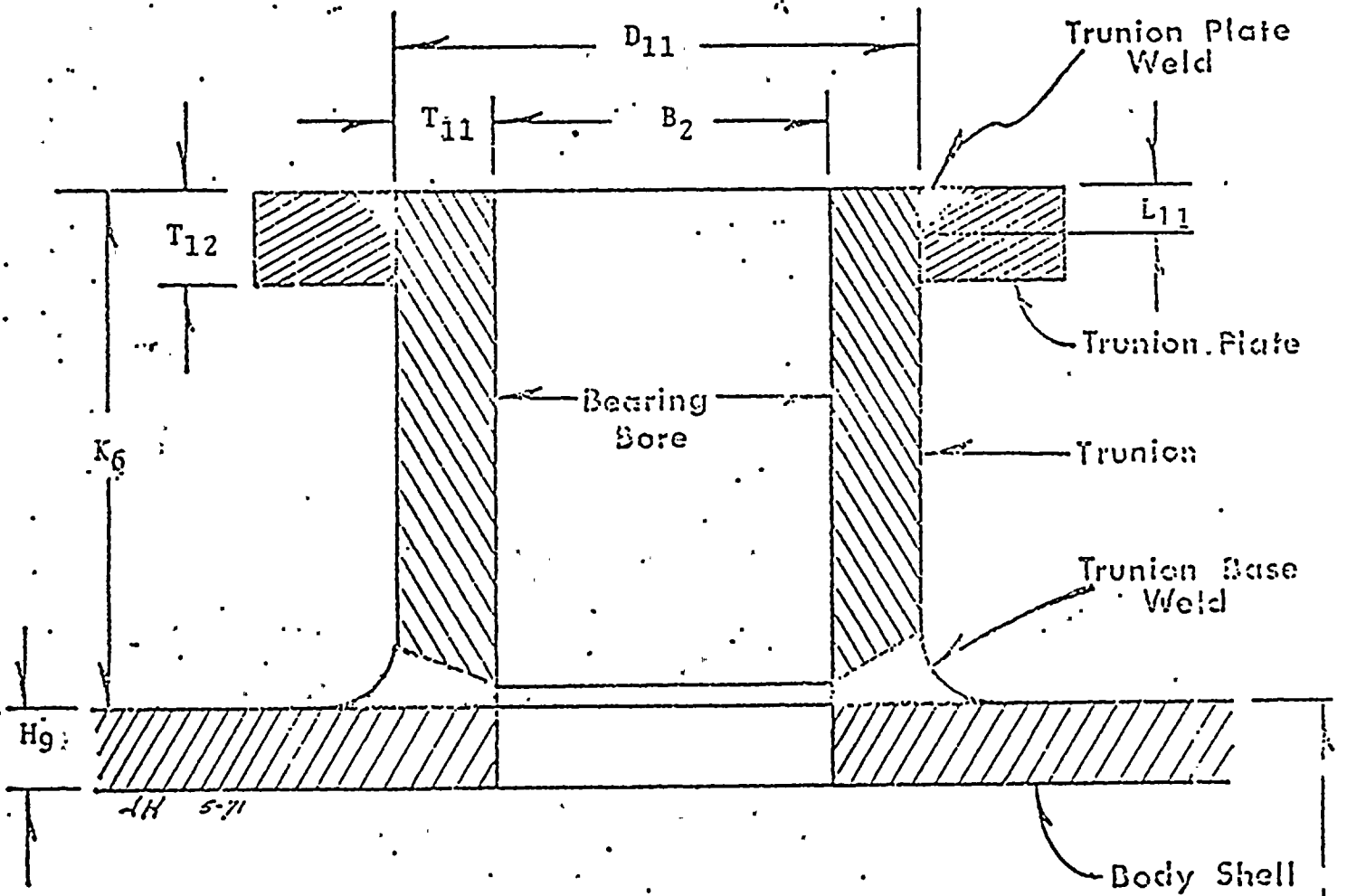
Where,

$$S(81) = \frac{F_z + W_4 g_z}{.25\pi(D_{11}^2 - B_2^2)} = \text{Direct tensile stress, psi.}$$

$$S(82) = \frac{32((\overline{M}_x + F_y K_6)^2 + (\overline{M}_y + F_x K_6)^2)^{\frac{1}{2}} D_{11}}{\pi(D_{11}^4 - B_2^4)}$$

= Bending tensile stress, psi







Top Trunnion Assembly Cont'd

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

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

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

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

Where,

$$S(86) = \frac{4((\overline{M}_x + F_y K_6)^2 + (\overline{M}_y + F_x K_6)^2)^{1/2}}{.707(.5)\pi D_{11}^2} + \frac{F_z}{\pi D_{11} L_{11}}$$

= Shear stress due to operator eccentricity

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



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:

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

Frequency Analysis

Where

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

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

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

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

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

E. Bonnet Assembly

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

Frequency Analysis

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

Natural frequency of bonnet:

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

Where

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

ATTACHMENT 3
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 (REYNOLDS 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 FRICTN.: 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/FT³-MIN. .133956 LB/FT³-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 DOWNSTR. PRESS. P6-ABS. (75-60 DEG.)

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	P1	P2	DELP	PRESS.	FLOW	FLOW	TD	TB+TH	TIME (LOCA)
APPRX. PSIA	PSIA	PSI	RATIO	(SCFM)	(LB/MIN)	INCH LBS	TD-TB-TH	SEC.	
35	41.70	22.36	19.34	.536	214793	11807	51642	1586	50055 3.00
30	44.03	15.04	28.99	.342	172255	9469	47548	1716	45831 3.38
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.

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

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

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

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 P
SIA UPSTR.PRESS.)

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



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 P
SIA 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





HENRY PRATT COMPANY

creative engineering for fluid systems

401 SOUTH HIGHLAND AVENUE • AURORA, ILLINOIS 60307

~ CONTAINMENT ISOLATION / PURGE VALVE ANALYSIS

Florida Power & Light
Turkey Point Units 3 & 4

~ 54" RIA Valve w/Pratt 2 Plate Cylinder Operator

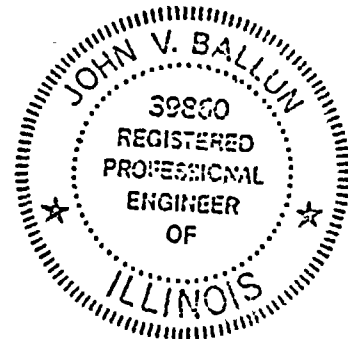
Customer P.O.: 5177-099

Pratt Order No.: D-027256



STRESS REPORT FOR

54" RIA WITH PRATT TWO PLATE CYLINDER OPERATOR

Project Site Turkey Point Units 3 & 4Customer Florida Power & Light Co.Engineer Bechtel Power CorporationOriginal Specification 5610-M-83Original Purchase Order 5610-M-83Original Pratt Job No. 7-3071-3 & 7-3071-4Valve Tag Nos. POV-3-2602 POV-3-2603POV-4-2602 POV-4-2603General Arrangement Drawing E-585 Rev. 3Prepared by: Rao N. KazaDate: 9/14/81Reviewed by: J. J. WronaDate: 9-15-81Certified by: J. V. BallunDate: 9-16-81



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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 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 1×10^8 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, in²

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

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

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

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 (≈ 1.2 for air/steam mix)

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

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

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

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

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



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

D-27256-1

TORQUE TABLE 1

9 / 11 / 81

JOB: FLOP. PWR: TURKEY PT P2-VARIABLE SIZE ADJUSTED (REYNOLDS 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 1.49311 TIMES HIGHER AS AIR CRIT. CASE INLET V1-OF
5 INCH MODELMAX. TORQUE IS AT THE CRITICAL PRESS. RATIO (.585- (5 IN) MODEL OR APPX .696352
(53.25 IN) WITH STMIX.) FIRST SONIC @ 72 DEG. V.A.)

ABSOL. MAX. TORQUE (FIRST SONIC) AT 72-68 DG. VLV. ANG. = 751527 IN-LBS @ 72 DEG.

MAX. TORQUE INCLUDES SIZE EFFECT (REYNOLDS NO. ETC) APPX. X 1.32454 FOR 53.25
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: 54"-R1A; 3/7.5 CLASS 75
 DISC SIZE: 53.062 INCHES SYMMETRICAL DISC
 SHAFT DIA.: 4.375 INCHES
 BRG. COEF. OF FRICTN.: 5.00000E-03
 SEATING FACTOR: 15
 INLET PRESS. VAR. MAX.: 60.2 PSIA
 OUTLET PRESSURE (P6): 34.13 PSIA (72 DEG. ACTUAL PRESS. ONLY (VAR.))
 MAX. ANG. FLOW RATE: 695628 CFM; 1380813 SCFM; 75907.1 LB/MIN
 CRIT. SONIC FLOW-90DG: 85975.5 LB/MIN AT 37.2315 INLET PSIA
 VALVE INLET DENSITY: .10912 LB/FT³-MIN. .157531 LB/FT³-MAX.
 FULL OPEN DELTA P: 9.48389 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 DNSTR. PRESS. P6-ABS. (75-60 DEG.)

IN SUBSONIC RANGE LIMITS-ONLY; SEE FORMULATIONS. -PER TESTS H. PRATT

THIS TQ. AT 72 DEG. SYMM. DISC (68=OFFSET SHAFT) $CT = T/D^{3/2} / A2 (ABS)$

--5 IN.MODEL EQUIV.VALUES-----ACTUAL SIZE VALUES-----										
ANGLE	P1	P2	DELP	PRESS.	FLOW	FLOW	TD	TB+TH	TIME (LOCA)	
APPRX.PSIA	PSIA	PSI	RATIO		(SCFM)	(LB/MIN)	----	INCHLBS----	TD-TB-TH	SEC.
90	41.70	29.38	12.32	.704	CR1380813	75907	0	1610	-1610	3.00
85	44.29	30.92	13.36	.698	1456702	80078	187449	1640	185809	3.39
80	46.31	31.38	14.93	.678	1463337	80443	275664	1706	273958	3.77
75	48.07	30.60	17.47	.637	1446726	79530	447458	1835	445623	4.13
72	49.01	28.67	20.35	.585	CR1330633	73148	712696	2034	710662	4.32
70	49.59	27.98	21.61	.564	CR1300953	71516	691837	2018	689818	4.45
65	50.86	25.10	25.77	.493	CR1172749	64469	645369	1983	643386	4.72
60	51.87	22.47	29.40	.433	CR1004642	55227	540019	1904	538115	4.95
55	52.61	19.97	32.64	.380	CR 844278	46412	527993	1909	526083	5.11
50	53.05	18.37	34.68	.346	CR 690066	37934	405970	1981	403988	5.22
45	53.20	17.21	35.99	.323	674335	37070	358280	2042	356238	5.25
40	53.33	16.45	36.88	.308	466700	25655	261477	2101	259375	5.28
35	53.72	15.67	38.04	.292	361955	19897	198745	2145	196600	5.39
30	54.34	15.22	39.12	.280	269438	14811	107525	2189	105336	5.55
25	55.16	14.97	40.19	.271	186718	10264	68689	2225	66464	5.78
20	56.15	14.82	41.32	.264	115503	6349	48297	2297	45999	5.05
15	57.23	14.72	42.50	.257	65662	3609	17838	2408	15430	6.38
10	58.34	14.71	43.63	.252	32810	1803	11262	2542	8720	6.73
5	59.40	14.70	44.69	.248	10936	601	5831	2695	3135	7.11
0	60.20	14.70	45.50	.244	0	0	45133	2600	42533	7.50

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47734 IN-LBS @ 0 DEG.
 MAX. DYN. - BEARING - HUB SEAL TORQUE (M/M) = 712696 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: 2 Plate Cyl. Operator 14 x 18 w/spring
.. to close.

Rating: 75000 in-lbs

Max. valve torque: 712696 in-lbs

The two plate cylinder operator furnished was specifically designed for the requirements of the original order which did not include specific 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:

30 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 40° or less. (See attachments #2 and #4.)

In addition, the calculated torques exceed the rating for the actuator except at valve disc angles of 30° 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.

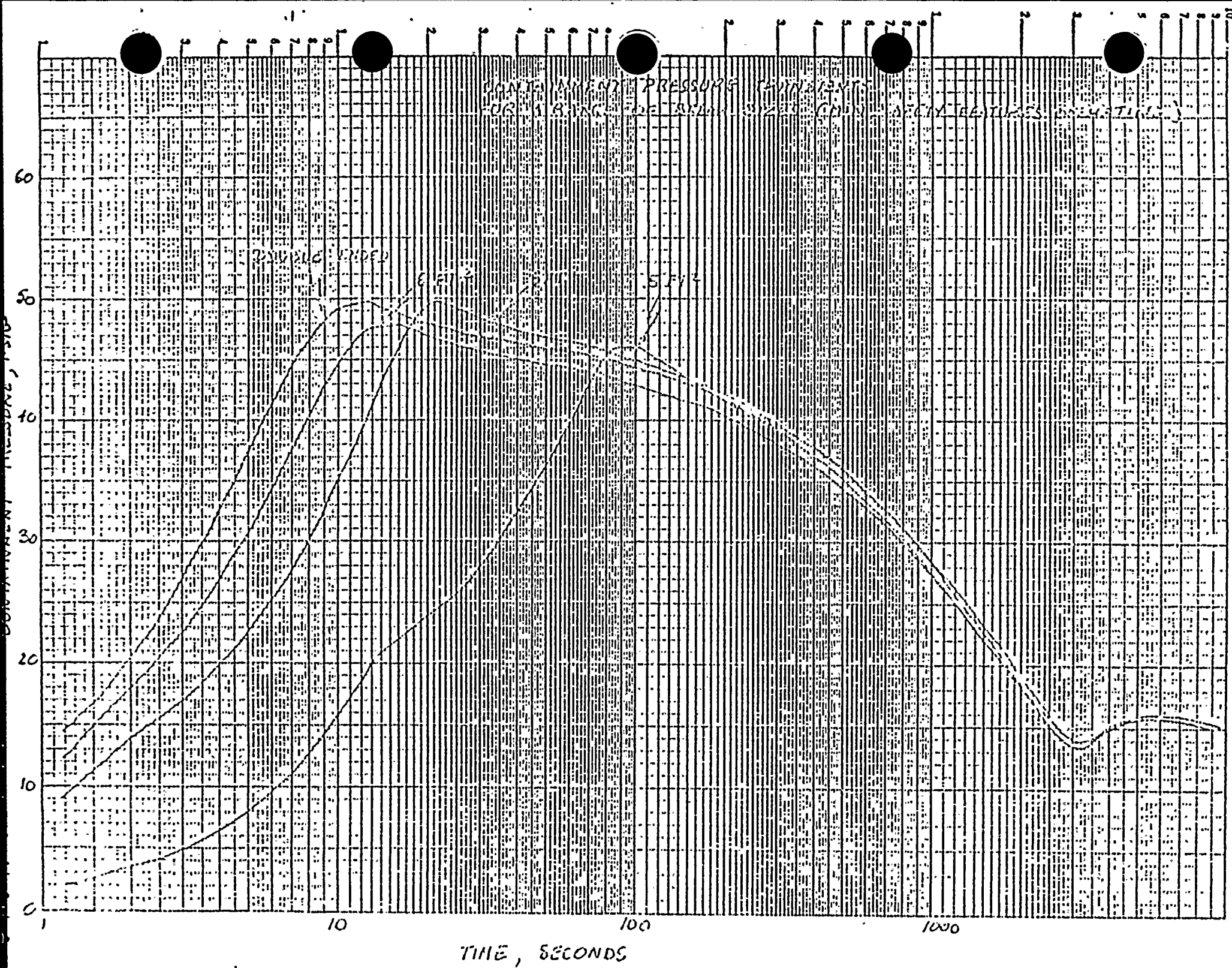
Further analysis by the customer/engineer is recommended prior to implementation.

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







ATTACHMENT 1B

PRATT LETTER REGARDING
ADDITIONAL INFORMATION

PRATT

HENRY PRATT COMPANY

301 SOUTH HIGHLAND AVENUE • AURORA, ILLINOIS 60507

February 16, 1981

Becitel 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 of the valve (one for each valve size), or

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.



PRATT

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 Fundamentals, we compile the following flow resistance coefficients:

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

<u>VALVE</u>	<u>VALVE</u> <u>ANGLE</u>	<u>OPEN</u> <u>ANGLE</u>	<u>COEF.</u>	<u>REF.</u>
	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)

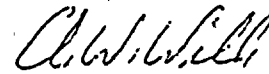


- 2) Regarding the "Confirmation that maximum delay time from LOCA to initiation of valve rotation is 4.2 seconds. Provide a minimum delay time as well."

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 ft² 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
Project Engineer

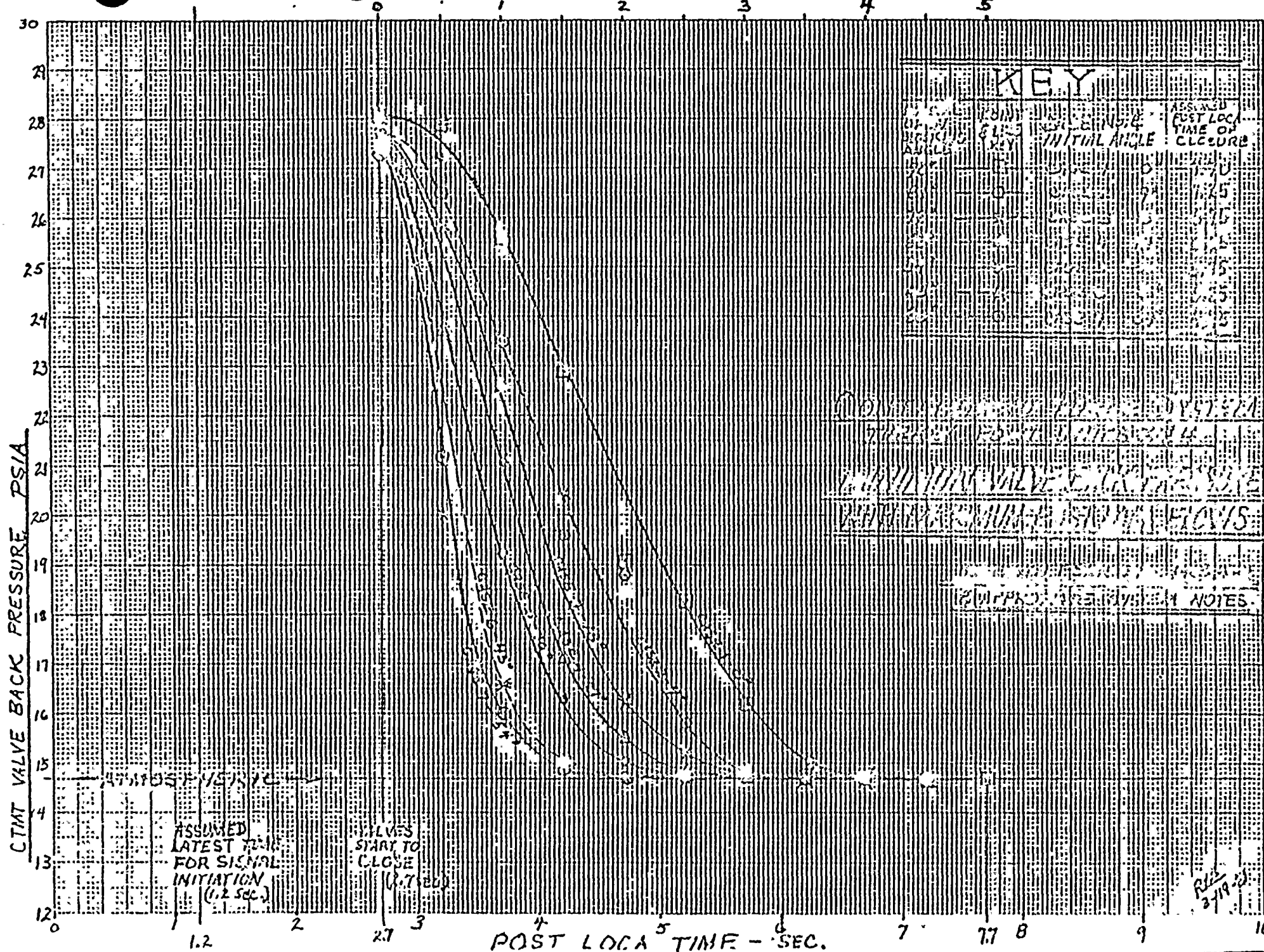
AWW/RVB:mfa

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

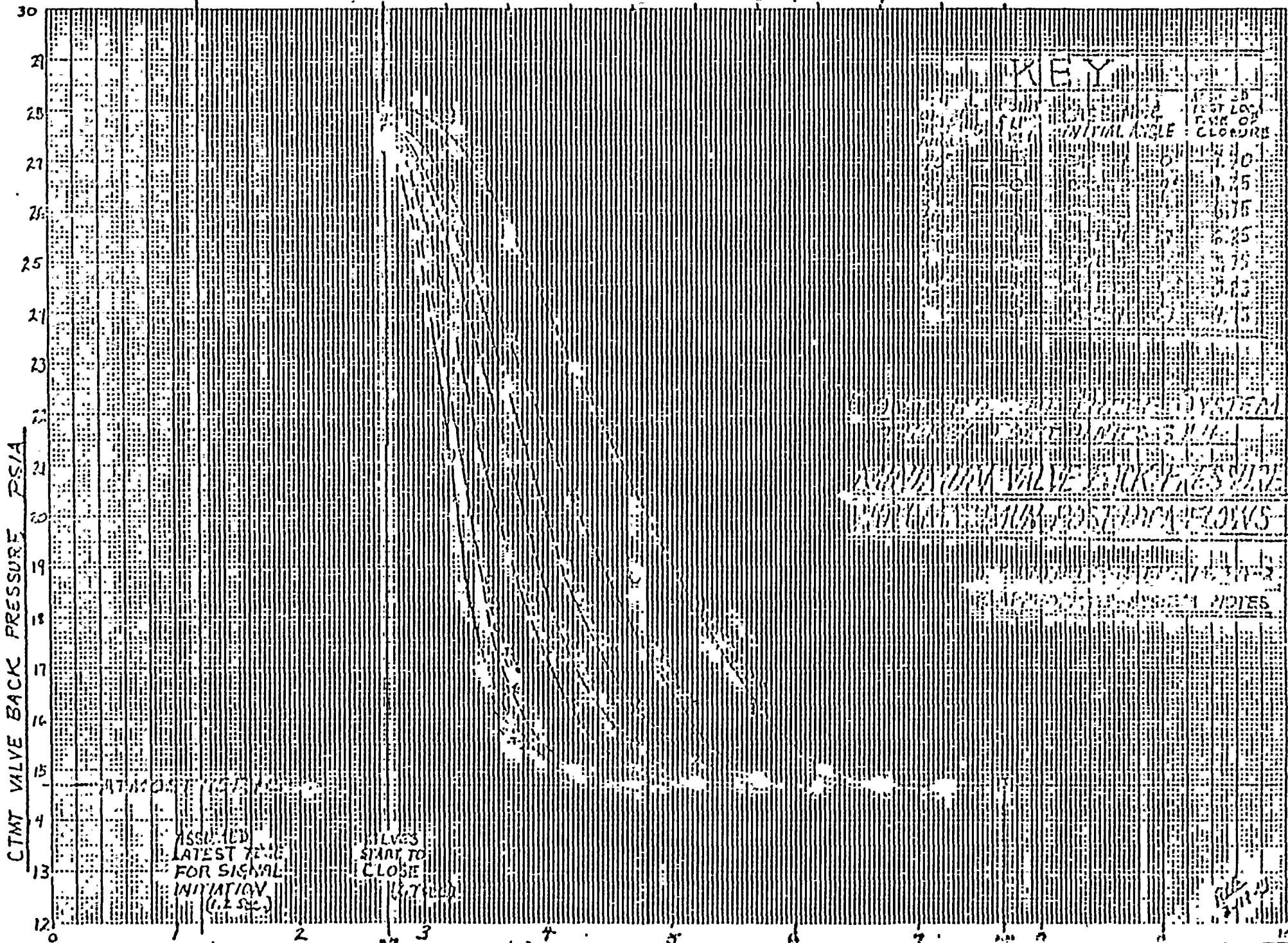


VALVE CLOSE TIME - SEC.





VALVE CLOSURE TIME - SEC.



KEY

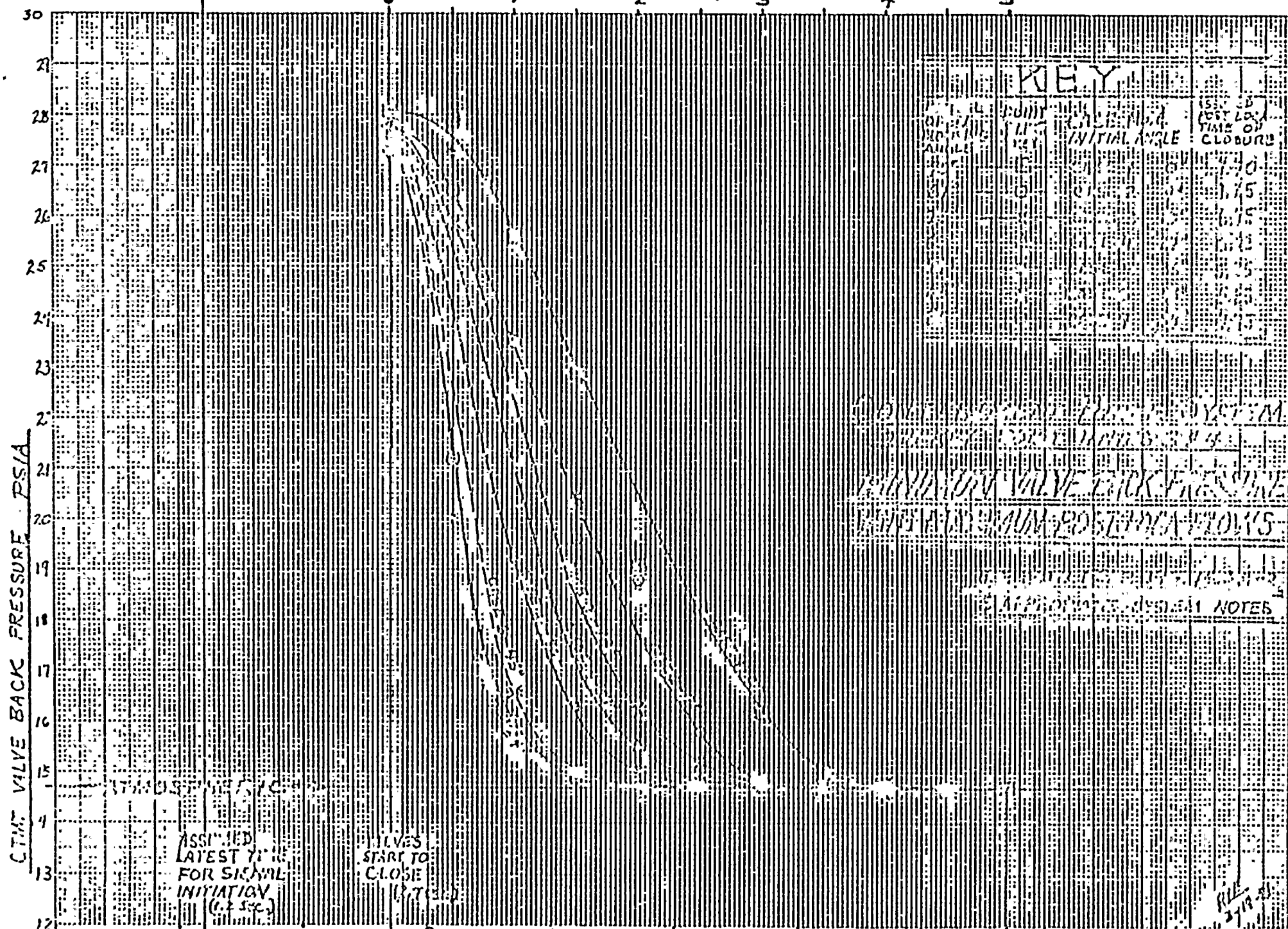
INITIAL ANGLE	TIME OF CLOSURE
0	1.40
1	1.75
2	2.15
3	2.45
4	2.75
5	3.15
6	3.45
7	3.75
8	4.15

1. SYSTEM
 2. VALVE
 3. VALVE
 4. VALVE
 5. VALVE
 6. VALVE
 7. VALVE
 8. VALVE
 9. VALVE
 10. VALVE

NOTES

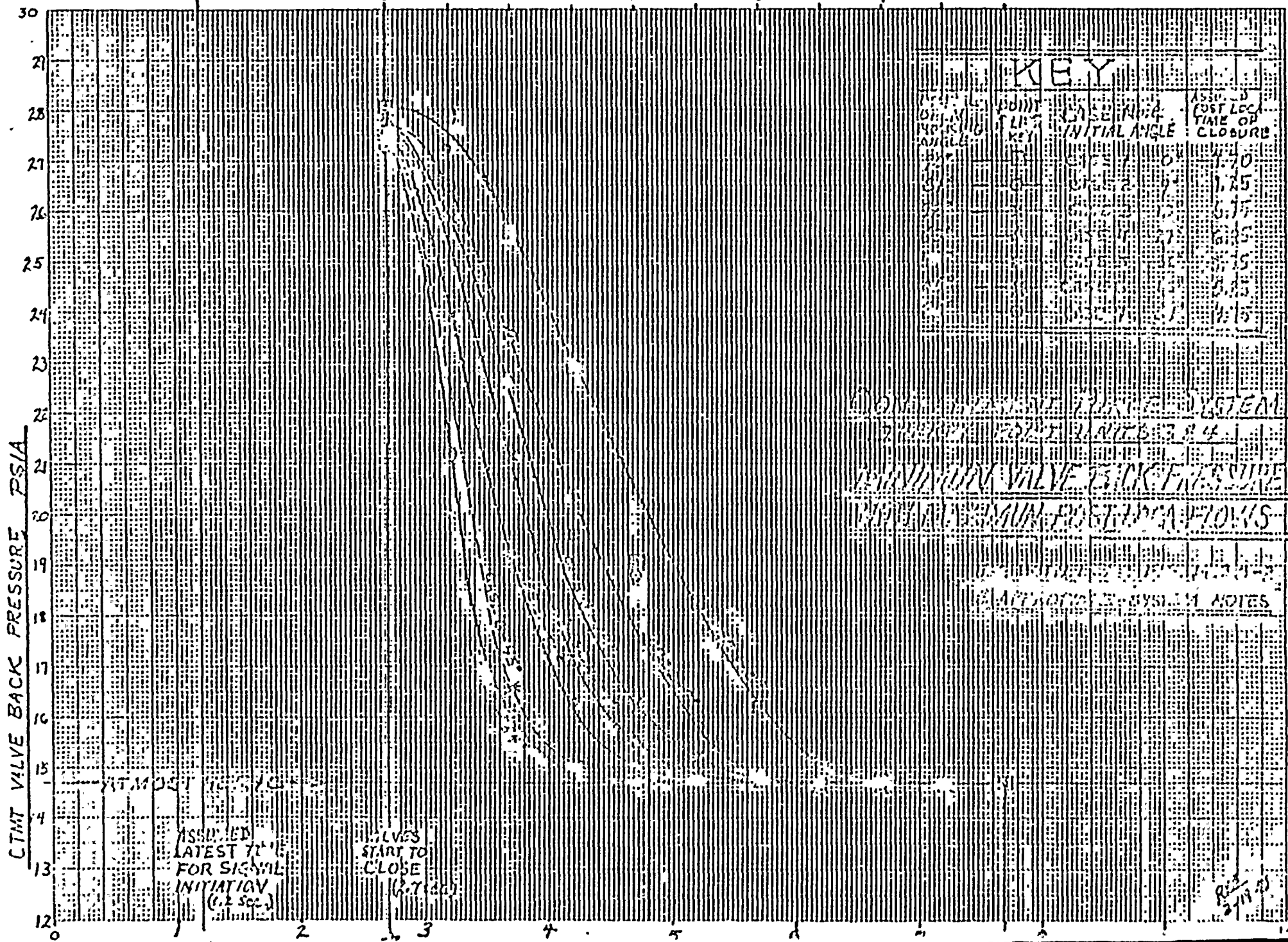


VALVE CLOSURE TIME - SEC.





VALVE CLOSURE TIME - SEC.



KEY			
POINT	LINE	INITIAL ANGLE	ASSUMED POST LOC. TIME OF CLOSURE
1	1	0	4.40
2	1	0	4.70
3	1	0	5.15
4	1	0	5.75
5	1	0	6.15
6	1	0	6.75
7	1	0	7.15
8	1	0	7.75
9	1	0	8.15
10	1	0	8.75
11	1	0	9.15
12	1	0	9.75
13	1	0	10.15
14	1	0	10.75
15	1	0	11.15

1.05 INCHES DIA. PORT SYSTEM
 2.00 INCHES DIA. PORT SYSTEM
 1.00 INCH DIA. VALVE BACK PRESSURE
 1.00 INCH DIA. PORT SYSTEM
 1.00 INCH DIA. PORT SYSTEM
 1.00 INCH DIA. PORT SYSTEM

108.5177-152
 CALC.
 SHY. 17



ATTACHMENT 2

Nuclear

Purge Valve

Stress

Analysis



SEISMIC ANALYSIS
FOR 54 INCH
NUCLEAR PURGE VALVE



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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 grotch 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.

ANALYSIS NOMENCLATURE

B ₉	Distance to outer fiber of bonnet from shaft on x axis, in.
C	A factor depending upon the method of attachment of head, shell dimensions, and other items as listed in NC-3225.2, dimensionless (Fig. NC-3225.1 thru Fig. NC-3225.3)
C _b	Stress index for body bending secondary stress resulting from moment in connected pipe (NB-3545.2 (b))
C _p	Stress index for body primary plus secondary stress, inside surface, resulting from internal pressure (NB-3545.2(a))
C ₂	Stress index for thermal secondary membrane stress resulting from structural discontinuity.
C ₃	Stress index for maximum secondary membrane plus bending stress resulting from structural discontinuity
C ₆	Product of Young's modulus and coefficient of linear thermal expansion, at 500°F, psi/°F (NB-3550)
C ₇	Distance to outer fiber of disc for bending along the shaft, in.
C ₈	Distance to outer fiber of disc for bending about the shaft, in.
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.



ANALYSIS NOMENCLATURE

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	$1/2 \times$ 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	$W_3 g_x$ --Seismic force along x axis due to seismic acceleration acting on operator extended mass, pounds.
F_y	$W_3 g_y$ --Seismic force along y axis due to seismic acceleration acting on operator extended mass, pounds.
F_z	$W_3 g_z$ --Seismic force along z axis due to seismic acceleration acting on operator extended mass, pounds.
g	Gravitational acceleration constant, inch-per-second ²
G_b	Valve body section bending modulus at crotch region (NB-3545.2(b)), in ³
G_d	Valve body section area at crotch region (NB-3545.2(b)), in ²
G_t	Valve body section torsional modulus at crotch region (NB-3545.2(b)), in ³
g_x	Seismic acceleration constant along x axis
g_y	Seismic acceleration constant along y axis
g_z	Seismic acceleration constant along z axis
h_g	Gasket moment arm, equal to the radial distance from the center line of the bolts to the line of the gasket reaction (NC-3225), in.
H_1	Disc hub key height, in.
H_2	Top trunnion bolt square, in.
H_3	Bottom trunnion bolt square, in.



ANALYSIS NOMENCLATURE

H_4	Bonnet bolt square, in.
H_5	Operator bolt square, in.
H_6	Bonnet bolt circle, in.
H_7	Operator bolt circle, in.
H_8	Bonnet height, in.
H_9	Actual body wall thickness, in.
I_1	Bonnet body moment of inertia about x axis, in ⁴
I_2	Bonnet body moment of inertia about y axis, in ⁴
I_3	Disc area ₄ moment of inertia for bending about the shaft, in ⁴
I_4	Disc area ₄ 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 ₄ 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.

ANALYSIS NOMENCLATURE

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	$W_3(g_z Z_O + g_y Y_O)$, operator extended mass seismic bending moment about the x axis, acting at the base of the operator, in-lbs.
M_y	$W_3(g_z Z_O + g_x X_O)$, operator extended mass seismic bending moment about the y axis, acting at the base of the operator, in-lbs.
M_z	$(W_3(g_y Y_O + g_x X_O))$, operator extended mass seismic bending moment about the z axis, in-lbs.
$\overline{M_x}$	$M_x + F_y T_5$, operator extended mass seismic bending moment about the x axis, acting at the bottom of the adapter plate, in-lbs.

ANALYSIS NOMENCLATURE

\overline{M}_y	$M_y + F_x T_5$, operator extended mass seismic bending moment about the y axis, acting at the bottom of the adapter, in-lbs.
\overline{M}_x	$M_x + F_y (T_5 + H_8) + g_y W_4 K_3$, operator extended mass seismic bending moment about the x axis, acting at the base of the bonnet, in-lbs.
\overline{M}_y	$M_y + F_x (T_5 + H_8) + g_x W_4 K_3$, 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 100°F/hr/hr fluid temperature change rate (NB-3545.3)
NA	Not applicable to the analysis of the system.
N_1	Number of top disc pins
N_2	Number of operator bolts
N_3	Number of trunnion bolts
P_d	Design pressure, psi
P_r	Primary pressure rating, pound
P_s	Standard calculation pressure psi
P_e	Largest value among P_{eb} , P_{ed} , P_{et} , psi
P_{eb}	Secondary stress in crotch region of valve body caused by bending of connection standard pipe, calculated according to NB-3545.2(b), psi
P_{ed}	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
P_{et}	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



ANALYSIS NOMENCLATURE

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 Q_p (NB-3545.2(a)), in.
r_2	Fillet radius of external surface at crotch (NB-3545.2(a)), in.
R_4	Disc radius, in.
R_5	Shaft radius, in.
R_m	Mean radius of body wall, in.
R_6	Radius to 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



ANALYSIS NOMENCLATURE

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.
ΔT_2	Maximum magnitude of the difference in average wall temperatures for walls of thicknesses t_e , T_e , resulting from 100°F/hr fluid temperature change rate, °F.
T_1	Thickness of cover cap behind bolt head, in.
T_2	Thickness of 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.



ANALYSIS NOMENCLATURE

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.

ANALYSIS NOMENCLATURE

z_0	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



ANALYSIS NOMENCLATURE

- 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

ANALYSIS NOMENCLATURE

- 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 S_m for tensile stresses and $.6 S_m$ for shear stresses. The allowable levels are the same whether the calculated stress is a combined stress or results from a single load condition. S_m 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	ALLOWABLE STRESS LEVEL PSI
BODY	NB-3545.1	Primary Membrane Stress in Crotch Region	P_m	36	ASTM A-36	1603	S_m 12600
		Primary Membrane	P'_m	36	ASTM A-36	939	S_m 12600
	NB-3542.2	Primary Plus Second- ary Stress Due to Internal Pressure	Q_p	38	ASTM A-36	2690	S_m 12600
	NB-3545.2	Pipe Reaction Stress Axial Stress Bending Torsion	P_{ed} P_{eb} P_{et}	38	ASTM A-36	2118 4548 4548	1.5 S_m 18900
	NB-3545.2	Thermal Secondary Stress	Q_t	38	ASTM A-36	1068	S_m 12600
	NB-3545.2	Primary Plus Second- ary Stress	S_n	38	ASTM A-36	5803	3 S_m 37800
	NB-3545.3	Normal Duty Fatigue Stress $N_a \geq 2000$	S_p	39	ASTM A-36	6959	1.5 S_m 18900
Disc	NB-3546.2	Combined Bending Stress in Disc	$S(1)$	40	ASTM A-36	8185	1.5 S_m 18900
Shaft	NB-3546.3	Combined Stress in Shaft	$S(4)$	41	ASTM A-276 Type 316 Condition A	73413	1.5 S_m 30000

TABLE 1

STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVEL PSI
DISC PINS		Shear Stress in Top Pin	S (20A)	43A	ASTM A-276 Type 316	43273	(.9) (.6) Sy 16200
		Bearing Stress on Top Pins in Shaft	S (20B)	43A	ASTM A-276 Type 316	15524	.9Sy 27000
Shaft Bearings		Compressive Stress on Shaft Roller Bearings	S (21)	45	(SKF-#23222C)	97589	125,000 1
Thrust Bearing		Bearing Stress on Thrust Collar	S (22)	46	ASTM B-164 Condition A	641	Sm 13600
		Shear Stress in Adjusting Screw Threads	S (24)	46	ASTM B-164 Condition A	65040	(.9) (.6) Sy 16200
		Combined Stress in Retainer Bolts	S (28)	48	ASTM B-164 Condition A	24447	.9Sy 27000
		Shear Tear out of Thrust Bearing Retainer Bolts	S (31)	48	ASTM B-164 Condition A	1647	(.9) (.6) Sy 16200

TABLE 1

STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVE PSI
Bottom Cover		Shear Tear Out of Bolts in Bottom Trunnion	S(33)	49	ASTM A-36	2970	.6Sm 7560
		Shear Tear Out of Cover Bolt Head Through Bottom Cover	S(34)	49	ASTM A-36	1907	.6Sm 7560
		Combined Stress in Cover Bolts	S(35)	49	ASTM A-193 GR.B7	11164	Sm 25000
		Combined Stress in Cover	S(38)	49	ASTM A-36	2582	Sm 12600
Operator Mounting		Shear Tear Out of Trunnion Bolts Thru Tapped Hole in Trunnion	S(42)	51	ASTM A-36	5673	.6Sm 7560
		Bearing Stress on Tapped Holes in Trunnion	S(43)	51	ASTM A-36	21352	.9Sy 63000
		Bearing Stress of Trunnion Bolt on Through Hole in Bonnet	S(44)	51	ASTM A-36	40142	.9Sy 63000
		Shear Tear Out of Trunnion Bolt Heads Through Bonnet	S(45)	53	ASTM A-36	5776	Sm 12600
		Combined Stress in Trunnion Bolt	S(46)	53	SAE GR.2	67913	.9Sy 25200
		Shear Tear Out of Operator Bolt Head Thru Bonnet	S(51)	53	ASTM A-36	3222	(.6)(.9)Sy 37800

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	25921	.9Sy 63000
		Combined Stress in Operator Bolts	S(53)	53	SAE GR.2	92647	.9Sy 25200
		Combined Stress in Bonnet Body	S(58)	55	ASTM A-36	124303	.9Sy 63000
		Combined Shear Stresses in Bottom Welds	S(63)	57		12656	.6Sm 7200
		Combined Shear Stress in Top Bonnet Weld	S(70)	57		20567	.6Sm 7200
		Combined Stress in Trunnion Body	S(77)	58	ASTM A-36	2340	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	6178
BANJO	F_{N_2}	64	ASTM A-276 Type 316 Condition A	782
COVER CAP	F_{N_3}	64	ASTM A-36	888
BONNET	F_{N_4}	65	ASTM A-36	242



DIMENSIONAL DATA

Job Number: D-27256-1¹/₂

Valve Size: 54" - R1A5

Operator Mounting: DIRECT

Operator: THE PLATE 18X14 CYLINDER.

A _F <u>227.84</u>	B ₈ <u>5.126</u>	F _d <u>33</u>
A _m <u>12.412</u>	B ₉ <u>5.126</u>	F _x <u>6.000</u>
A ₁ <u>.373</u>	C <u>0.3</u>	F _y <u>6000</u>
A ₂ <u>.373</u>	C _b <u>1.0</u>	F _z <u>8000</u>
A ₃ <u>.334</u>	C _p <u>3.0</u>	G _b <u>62.66.26</u>
A ₄ <u>.302</u>	C ₂ <u>.50</u>	G _d <u>467.49</u>
A ₅ <u>.606</u>	C ₃ <u>.67</u>	G _t <u>12532</u>
A ₆ <u>.551</u>	C ₆ <u>2.49</u>	g _x <u>3</u>
A ₇ <u>.334</u>	C ₇ <u>5.01</u>	g _y <u>3</u>
A ₈ <u>.302</u>	C ₈ <u>5.01</u>	g _z <u>4</u>
A ₉ <u>.142</u>	D ₁ <u>54</u>	H ₁ <u>1</u>
A ₁₀ <u>.551</u>	D ₂ <u>4.375</u>	H ₂ <u>12.25</u>
A ₁₁ <u>.606</u>	D ₃ <u>1.43</u>	H ₃ <u>6.7175</u>
A ₁₂ <u>.0678</u>	D ₄ <u>3.75</u>	H ₄ <u>N/A</u>
A ₁₃ <u>.0775</u>	D ₅ <u>.75</u>	H ₅ <u>N/A</u>
B ₁ <u>3.9845</u>	D ₆ <u>.375</u>	H ₆ <u>N/A</u>
B ₂ <u>4.375</u>	D ₇ <u>.75</u>	H ₇ <u>12.375</u>
B ₃ <u>.606</u>	D ₈ <u>.75</u>	H ₈ <u>8</u>
B ₄ <u>.551</u>	D ₉ <u>N/A</u>	H ₉ <u>2.656</u>
B ₅ <u>20.12</u>	D ₁₀ <u>.75</u>	I ₁ <u>296.34</u>
B ₆ <u>N/A</u>	D ₁₁ <u>11</u>	I ₂ <u>186.76</u>
B ₇ <u>N/A</u>	F _b <u>950</u>	I ₃ <u>1091.4</u>

Dimensional Data (Cont.)

I ₄ <u>1091.4</u>	L ₁₁ <u>.375</u>	R ₄ <u>26.531</u>
I ₅ <u>183.895</u>	m <u>3.4</u>	R ₅ <u>2.1875</u>
I ₆ <u>17.984</u>	M _x <u>371500</u>	R ₆ <u>5.5</u>
I ₇ <u>284.5</u>	M _y <u>81500</u>	R ₇ <u>1.5</u>
I ₈ <u>1.230</u>	M _z <u>256500</u>	R _m <u>28.016</u>
J ₁ <u>1.25</u>	\overline{M}_x <u>421840</u>	S <u>30000</u>
J ₂ <u>1.25</u>	\overline{M}_y <u>131840</u>	t _e <u>2.656</u>
J ₃ <u>1.25</u>	\overline{M}_x <u>421840</u>	t _m <u>.69</u>
J ₄ <u>1.25</u>	\overline{M}_y <u>131840</u>	ΔT ₂ <u>4.0</u>
J ₅ <u>N/A</u>	N _a <u>2000</u>	T ₁ <u>1.0</u>
J ₆ <u>N/A</u>	N ₁ <u>2</u>	T ₂ <u>.25</u>
K ₁ <u>N/A</u>	N ₂ <u>8</u>	T ₃ <u>.437</u>
K ₂ <u>2.188</u>	N ₃ <u>4</u>	T ₄ <u>1.0</u>
K ₃ <u>3.4117</u>	P _d <u>75</u>	T ₅ <u>N/A</u>
K ₄ <u>14.75</u>	P _r <u>75</u>	T ₆ <u>1.0</u>
K ₅ <u>14.75</u>	P _s <u>85</u>	T ₇ <u>1.0</u>
K ₆ <u>6.63</u>	Q _{T1} <u>570</u>	T ₈ <u>712696 IN/LBS</u>
L ₁ <u>2.0</u>	Q ₁ <u>1.50</u>	T ₉ <u>.438</u>
L ₂ <u>1</u>	Q ₂ <u>8.50</u>	T ₁₀ <u>1.0</u>
L ₃ <u>1.18</u>	Q ₃ <u>11.50</u>	T ₁₁ <u>1.75</u>
L ₄ <u>1.88</u>	Q ₄ <u>1.9375</u>	T ₁₂ <u>1.0</u>
L ₅ <u>4.5</u>	Q ₅ <u>2.84</u>	U ₁ <u>.19</u>
L ₆ <u>1.0</u>	Q ₆ <u>11.00</u>	U ₂ <u>19</u>
L ₇ <u>N/A</u>	Q ₇ <u>3.375</u>	U ₃ <u>0.25</u>
L ₈ <u>N/A</u>	r <u>4.718</u>	U ₄ <u>N/A</u>
L ₉ <u>13.44</u>	r _i <u>26.688</u>	U ₅ <u>N/A</u>
L ₁₀ <u>9.5</u>	r ₂ <u>.63</u>	U ₆ <u>3188</u>



Dimensional Data (Cont.)

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 9.362
 V_8 12.716
 W_1 8500
 W_2 1700
 W_3 2000
 W_4 195
 W_6 .75
 W_7 1640
 W_8 36.5
 X_0 3.25
 Y_0 39.5
 Z_0 9.25
 Z_1 182
 Z_2 72
 Z_3 80.5
 Z_4 80.5
 Z_7 25.763
 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°F. Seismic accelerations are simultaneous 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 completely equivalent to a lg 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.



VALVE BODY CENTERED
COORDINATE SYSTEM

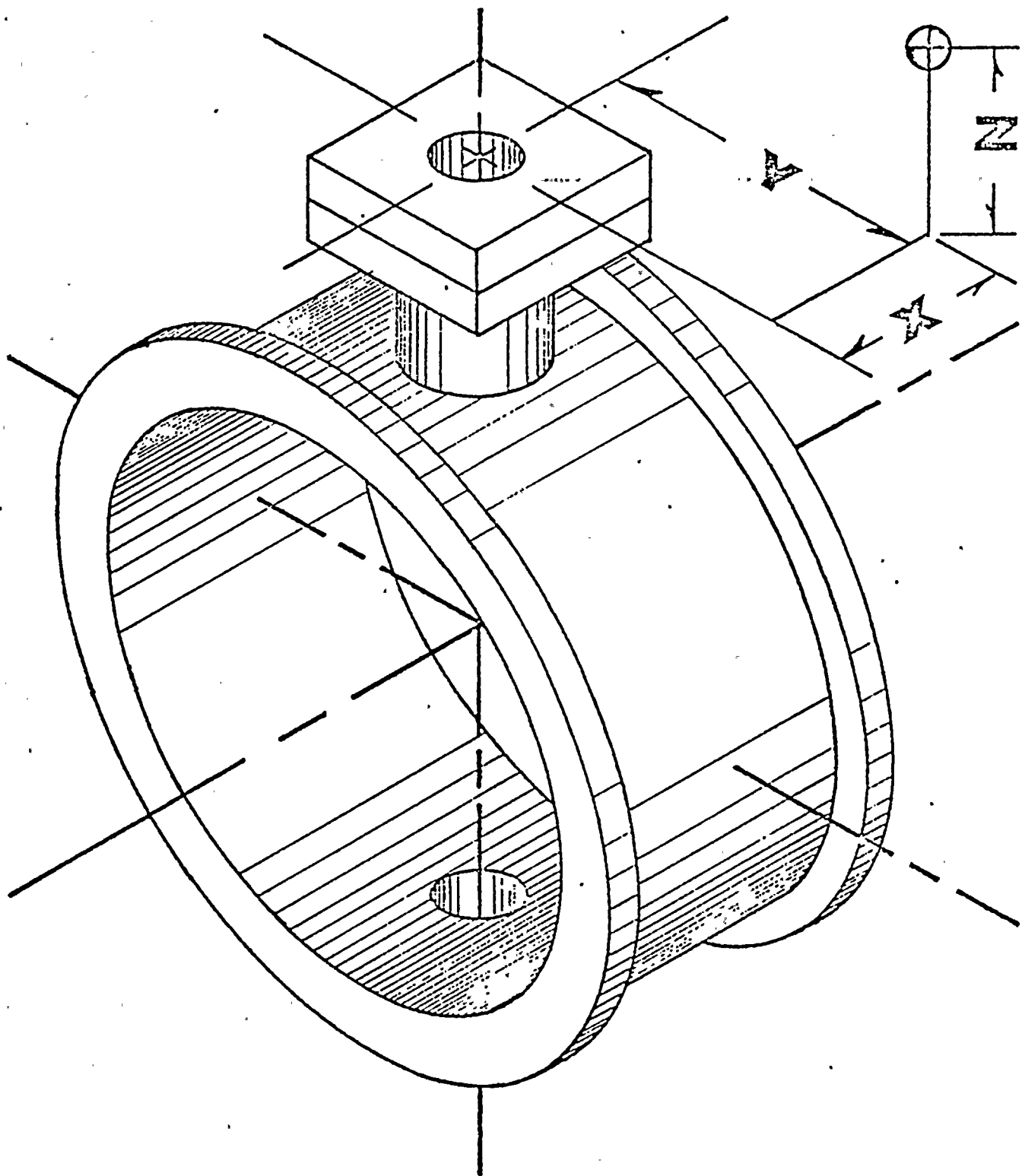
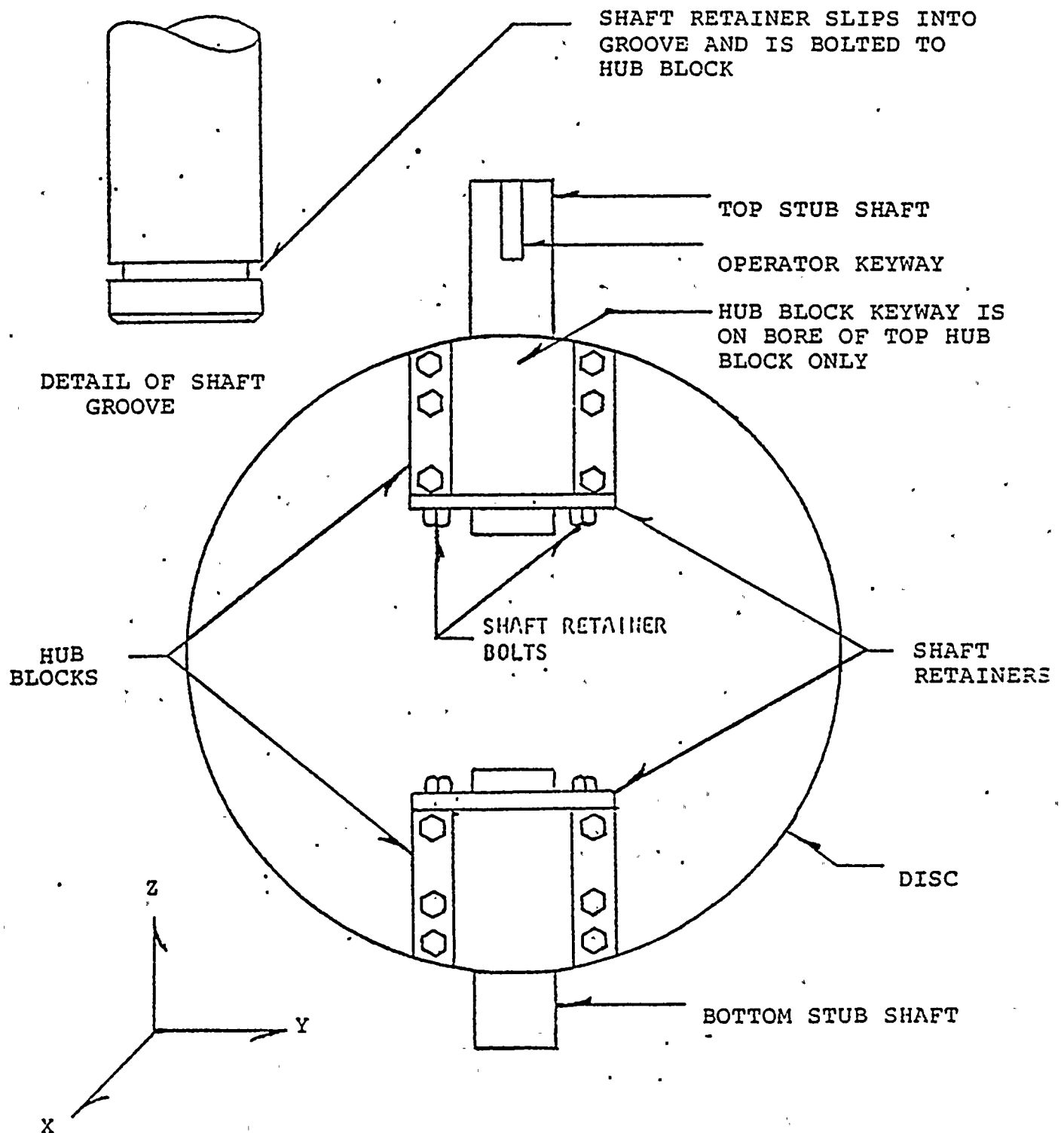


FIGURE 2 ESSENTIAL FEATURES
OF BANJO ASSEMBLY



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.

$$P_m = (A_f/A_m + .5) \cdot P_s$$

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 (A_m) which decreases the A_f/A_m ratio and reduces the result. For a section not containing the trunnion, the fluid area to metal area ratio (A_f/A_m) reduces to the body inside radius to the shell thickness (R_m/H_g) since the depths are the same. The resulting membrane stress equation is then:

$$P_m = (R_m/H_g + .5) P_s$$

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

4. Secondary Stresses

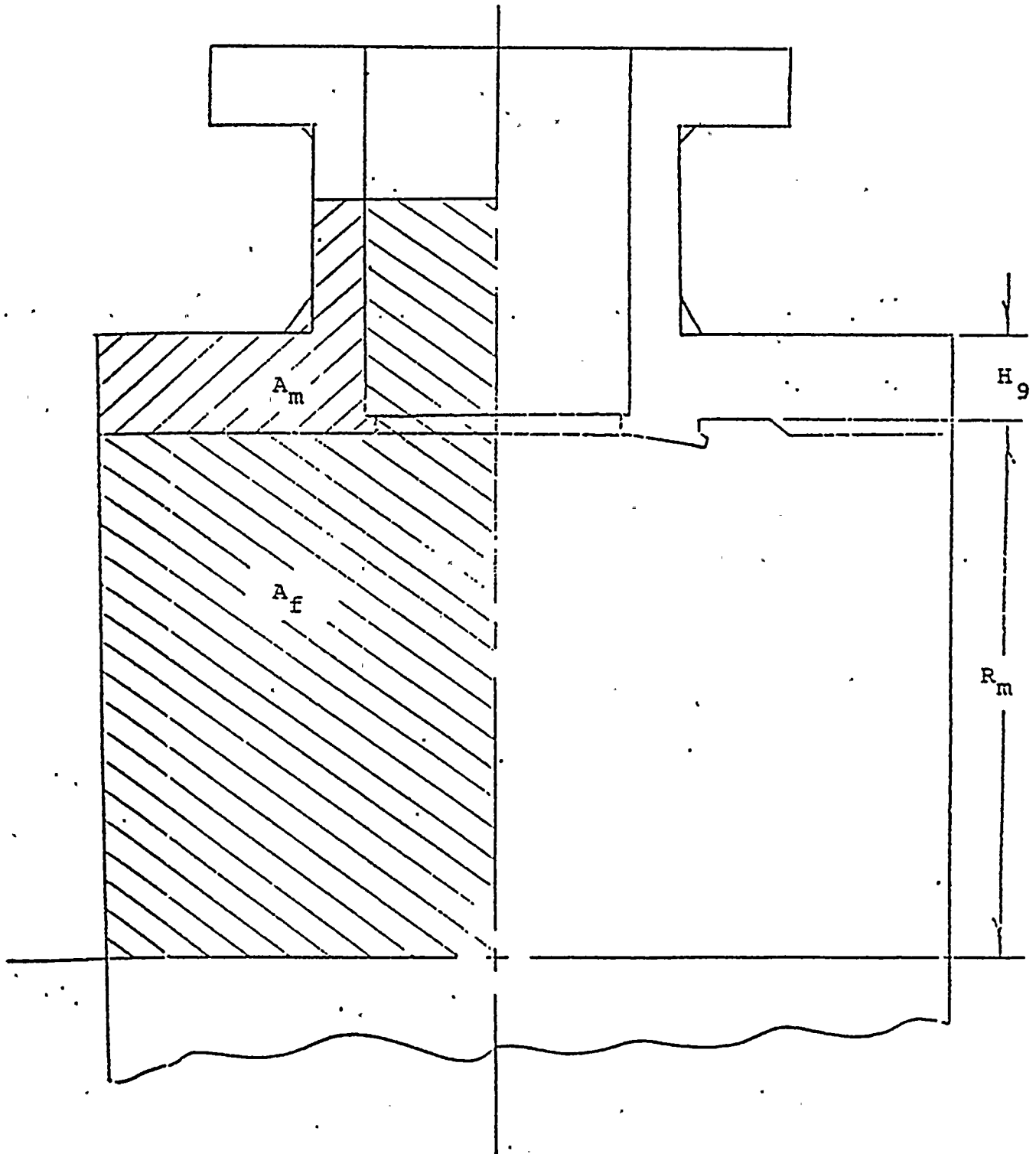
A. Body Primary plus secondary stress due to internal pressure.



PRESSURE AREA ANALYSIS

BODY CROSS-SECTION

FIGURE 3



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

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

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



Body Analysis (Cont.)

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 stress are mentioned in this section. For a flat plate such as the butterfly valve disc, membrane stress is not defined until the deflection of the disc reaches one-half the disc thickness. Since total deflection of the disc is much less than one-half the thickness, membrane stresses are not applicable to the analysis.

Figure 5 shows the disc for the 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:

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

where

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

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



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 or 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:

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

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

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

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

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

$$S(10) = 1.333 \left[\frac{.5 \pi R_4^2 P_s + .5 W_2 (g_x^2 + g_y^2)^{1/2}}{\pi R_5^2} \right] = \text{Direct shear stress, PSI}$$



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 analyze 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.

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

Where:

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

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

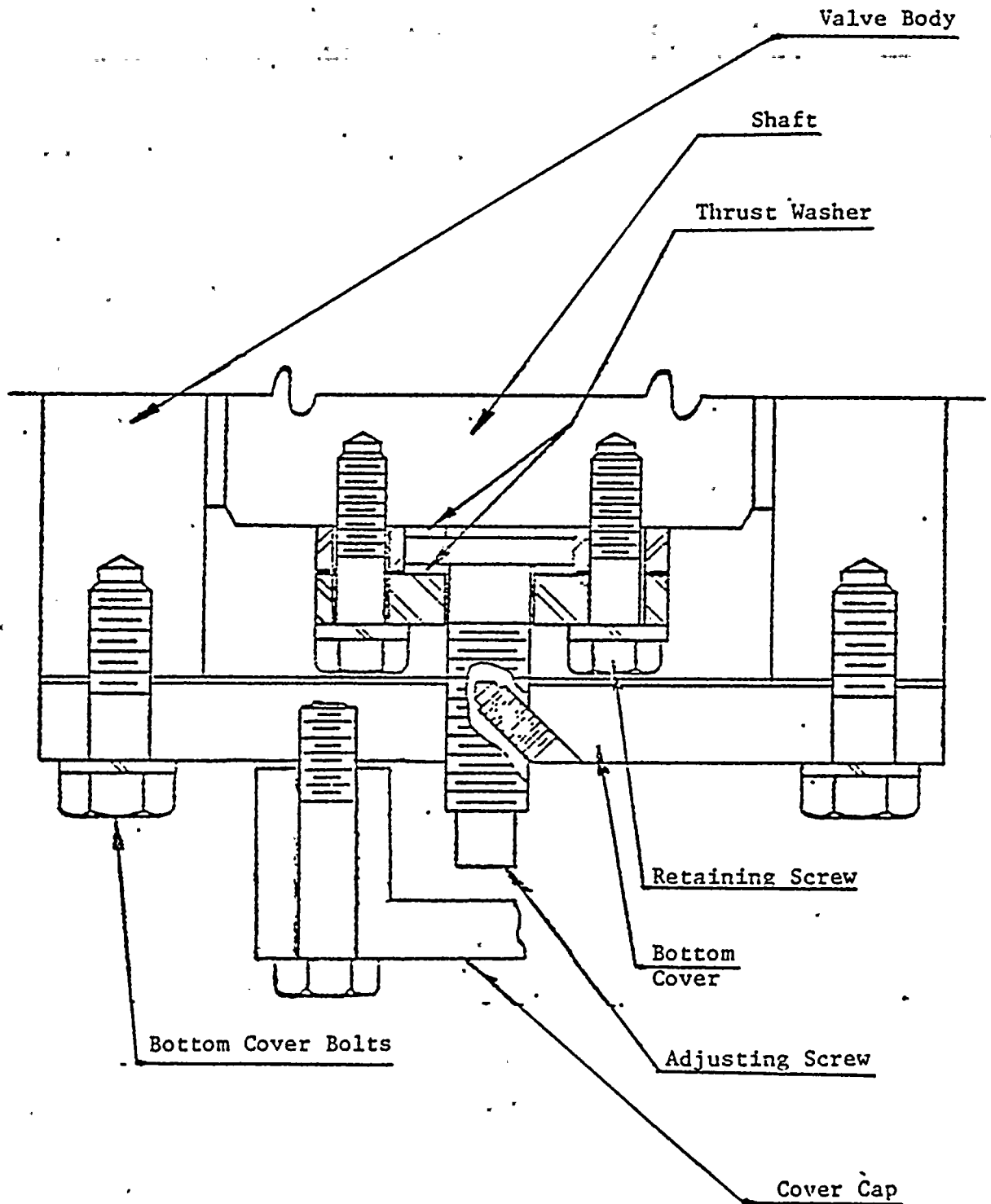
$$U_6 = W_2 g_z U_3 (.5 D_{10} + .5 (D_4 - D_{10}))$$

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

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



FIGURE 5
ESSENTIAL FEATURES
OF THRUST BEARING ASSEMBLY





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

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

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

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

Where:

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

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

2. Bottom Cover Stresses

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

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



Where:

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

$$S(40) = \frac{3(.785)(D_4 + .25)^2 P_s + W_2 g_z}{4 \pi T_4^2 m} = \text{Tangential stress}$$

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



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.

$$S(42) = \frac{F_z + W_4 \sqrt{g_x^2 + g_y^2 + g_z^2}}{4} + \frac{\overline{M}_x (J_2 + H_2)}{2J_2^2 + 2(J_2 + H_2)^2} + \frac{\overline{M}_y (J_1 + H_2)}{2J_1^2 + 2(J_1 + H_2)^2} \\ \cdot 9\pi L_4 D_7$$

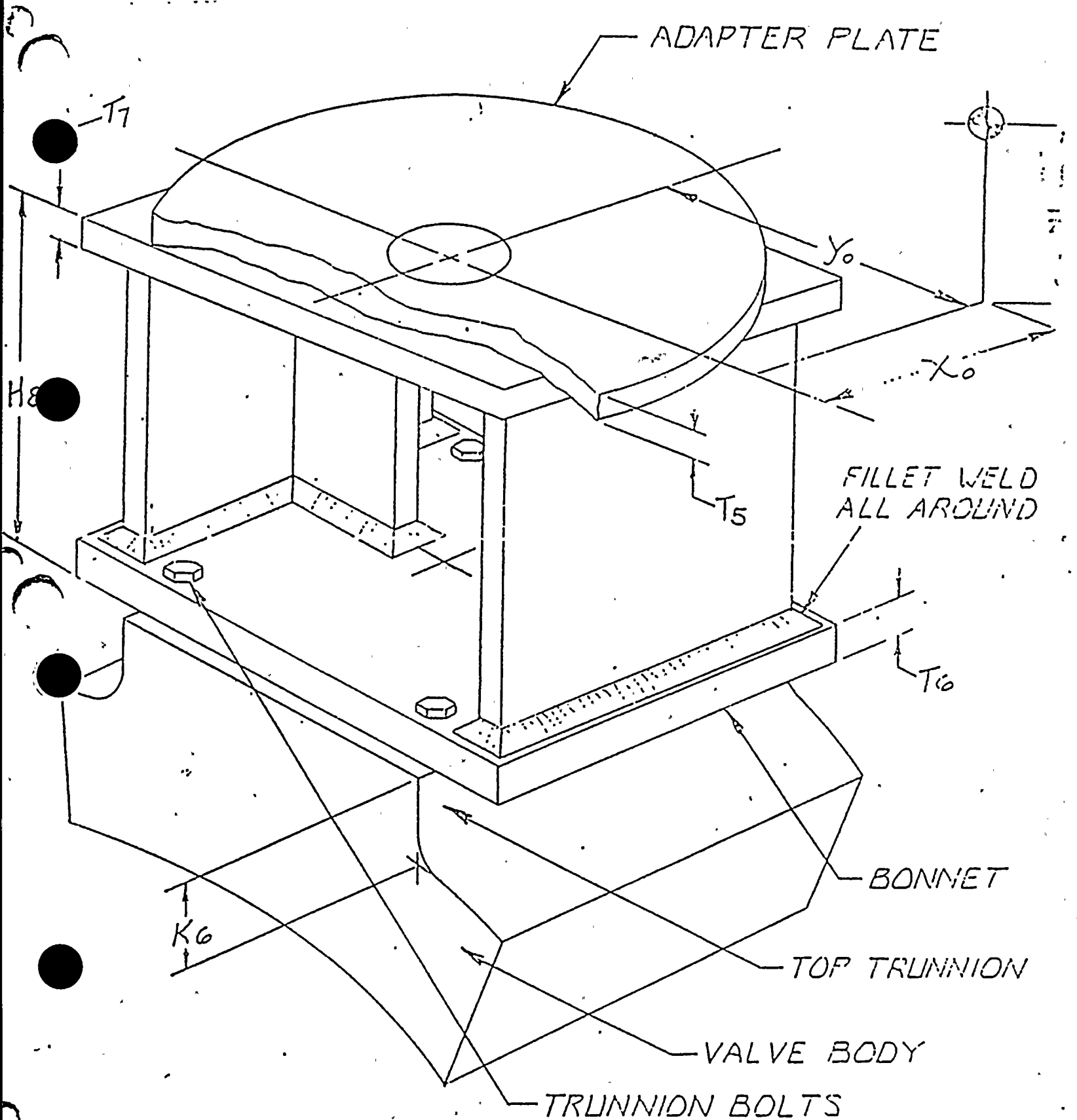
- b. Bearing stress on tapped holes in trunnion.

$$S(43) = \frac{\frac{M_z + T_8}{4(.707H_2)} + \frac{(F_x^2 + F_y^2)^{\frac{1}{2}}}{4} + \frac{W_4 (g_x^2 + g_y^2)^{\frac{1}{2}}}{4}}{D_7 L_4}$$

- c. Bearing stress on through hole in bonnet.

$$S(44) = \frac{\frac{M_z + T_8}{4(.707H_2)} + \frac{(F_x^2 + F_y^2)^{\frac{1}{2}}}{4} + \frac{W_4 (g_x^2 + g_y^2)^{\frac{1}{2}}}{4}}{D_7 T_6}$$





TOP TRUNNION MOUNTING

FIGURE 6



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

$$S(45) = \frac{\frac{F_z + W_4 g_z}{4} + \frac{\overline{M}_x (J_2 + H_2)}{2J_2^2 + 2(J_2 + H_2)^2} + \frac{\overline{M}_y (J_1 + H_2)}{2J_1^2 + 2(J_1 + H_2)^2}}{5.2 D_7 T_6}$$

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,

$$S(47) = \frac{F_z + W_4 g_z}{4A_5} = \text{Direct tensile stress, psi}$$

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

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

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

f. Shear tearout of operator bolt head through bonnet.

$$S(51) = \frac{\frac{(\overline{M}_x + \overline{M}_y) V_4}{2(V_1^2 + V_2^2 + V_3^2 + V_4^2)} + \frac{F_z}{4}}{5.2 D_8 T_7}$$

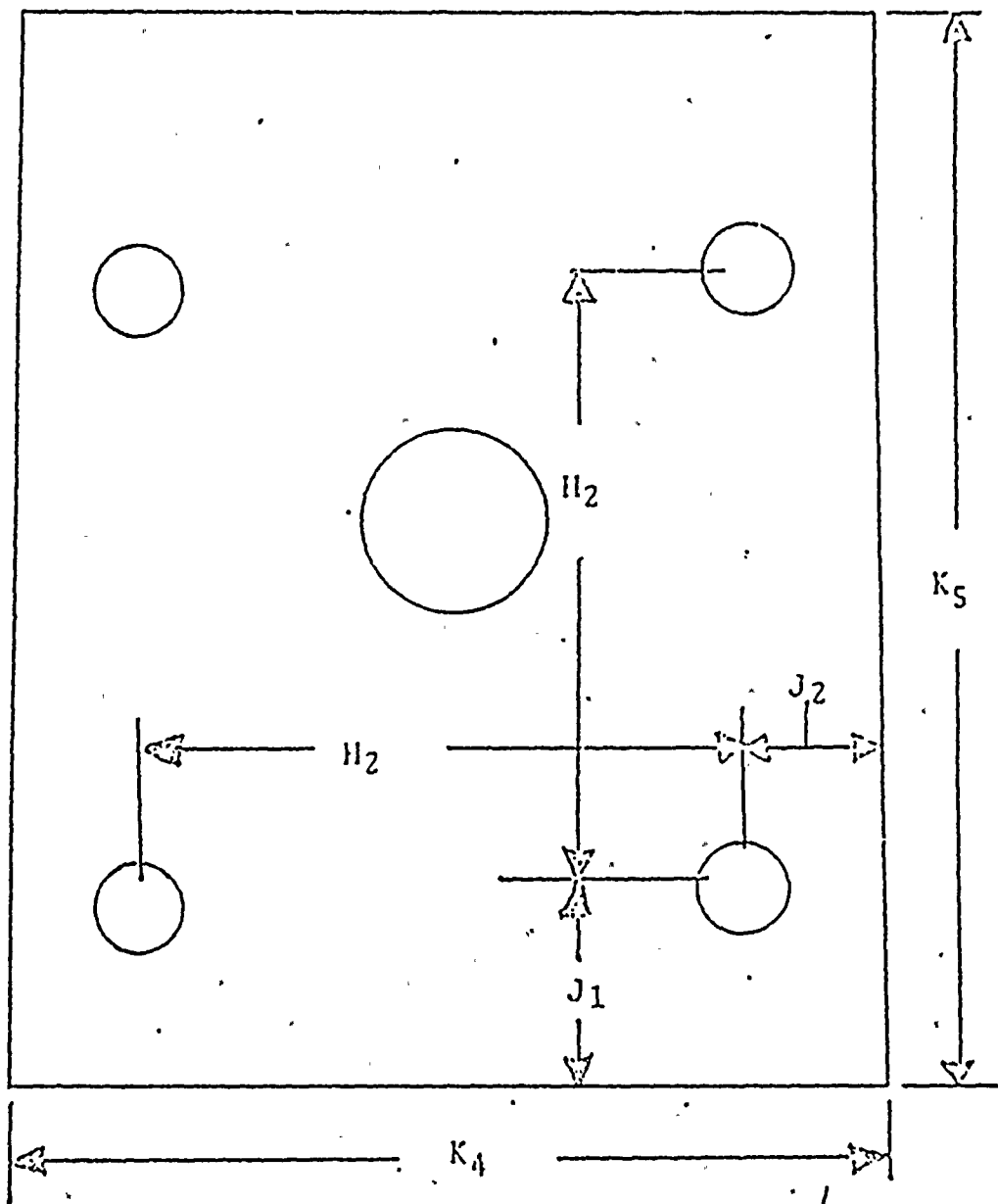
g. Bearing stress on through holes in bonnet.

$$S(52) = \frac{M_z + T_8}{.5H_7 8T_7 D_8}$$

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 7



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,

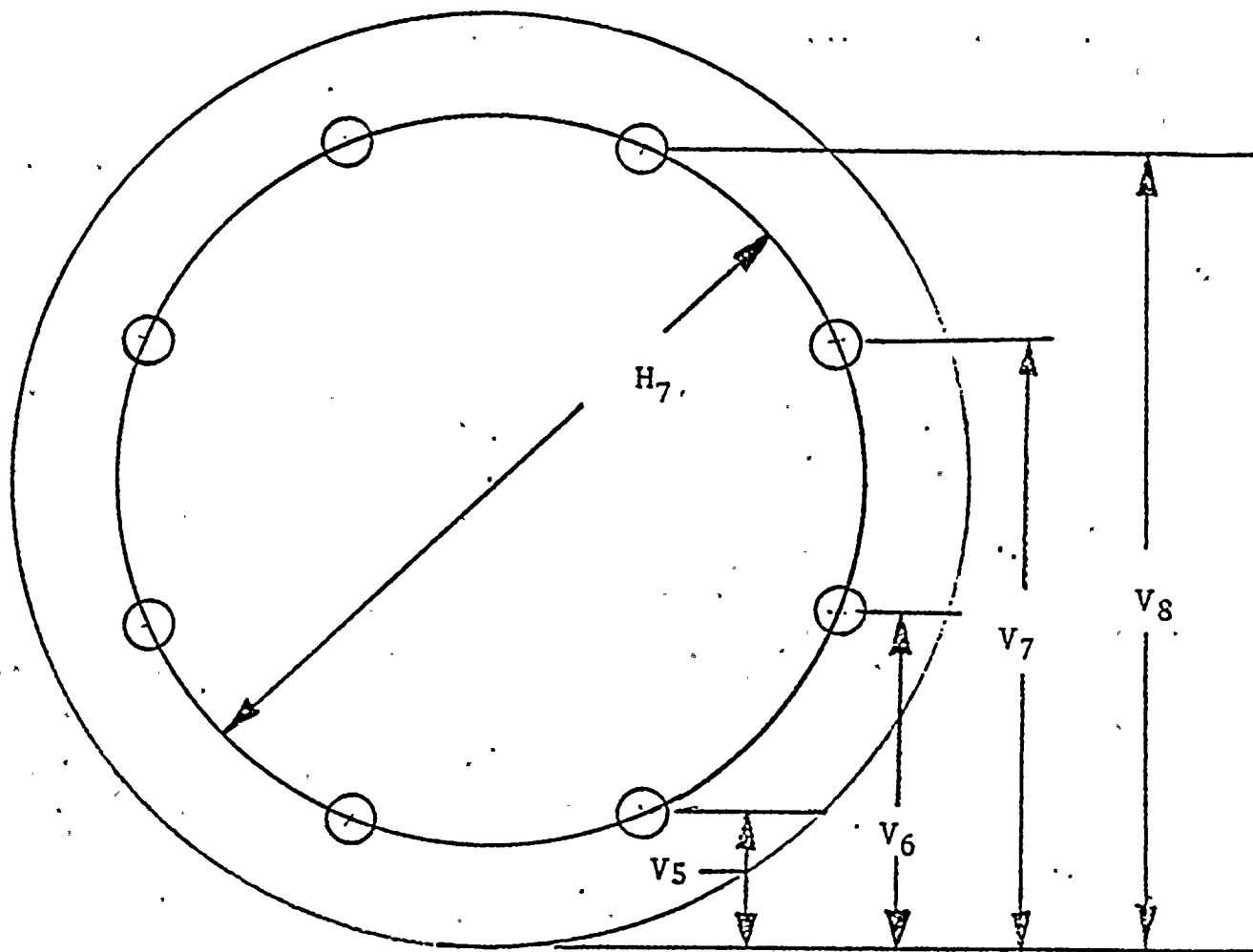
$$S(59) = \frac{F_z + W_4 g_z}{B_5} = \text{Direct tensile stress, psi}$$

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

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

$$S(62) = \frac{T_8 C_0}{K_0} = \text{Shear stress in bonnet body due to torsional load, psi}$$





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, in⁴.

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

BOTTOM BONNET WELDS

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

Where,

$$S(64) = S(66) + S(67) = \text{Total tensile stress, psi}$$

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

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

$$S(65) = S(68) + S(69) = \text{Total shear stress, psi}$$

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

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

TOP BONNET WELDS

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

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,

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

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



$$S(80) = \frac{(F_x^2 + F_y^2)^{1/2} + W_4(g_x^2 + g_y^2)^{1/2}}{K_4 K_5 - .785 B_2^2} = \text{Direct shear stress, psi}$$

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



TOP TRUNNION ASSEMBLY

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.

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

Where,

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

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

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

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

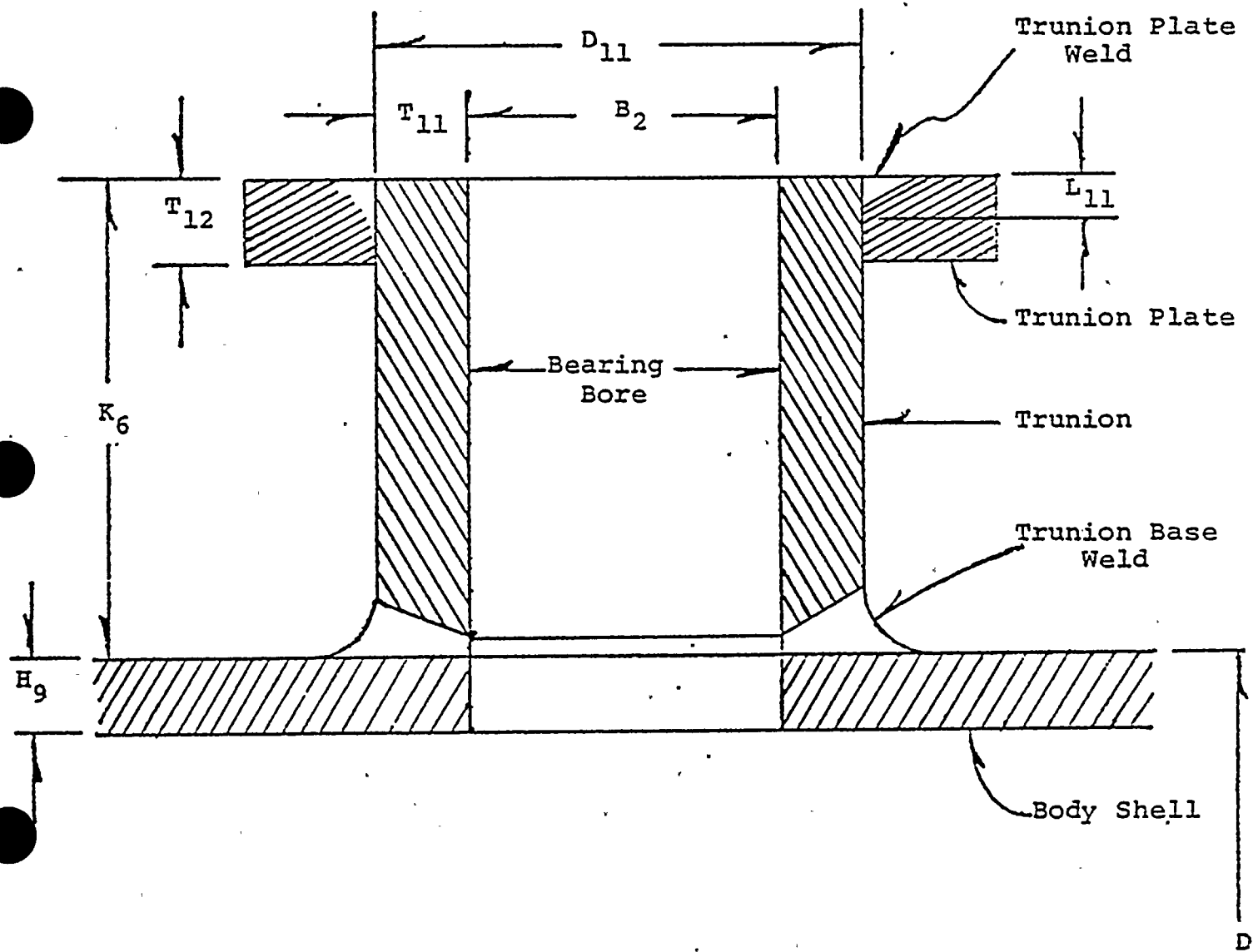
Where,

$$S(81) = \frac{F_z + W_4 g_z}{.25\pi (D_{11}^2 - B_2^2)} = \text{Direct tensile stress, psi.}$$

$$S(82) = \frac{32((\overline{M}_x + F_y K_6)^2 + (\overline{M}_y + F_x K_6)^2)^{\frac{1}{2}} D_{11}}{\pi (D_{11}^4 - B_2^4)}$$

= Bending tensile stress, psi

FIGURE 9
TOP TRUNION ASSEMBLY





Top Trunnion Assembly Cont'd

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

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

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

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

Where,

$$S(86) = \frac{4((\bar{M}_x + F_y K_6)^2 + (\bar{M}_y + F_x K_6)^2)^{1/2}}{.707(.5)\pi D_{11}^2} + \frac{F_z}{\pi D_{11} L_{11}}$$

= Shear stress due to operator eccentricity

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



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:

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

Frequency Analysis

Where

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

C. Banjo Assembly

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

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

Where

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

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:

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

Where

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

E. Bonnet Assembly

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



Frequency Analysis

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

Natural frequency of bonnet:

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

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-27256-1

TORQUE TABLE 1

9 / 11 / 81

JOB: FLOR. PWR; TURKEY-PT P2-VARIABLE SIZE ADJUSTED (REYNOLDS 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 1.5676 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 .696352
(53.25 IN) WITH STMIX.) FIRST SONIC @ 72 DEG. V.A.)

ABSOL. MAX. TORQUE (FIRST SONIC) AT 72-68 DG. VLV. ANG. = 70174 IN-LBS @ 0 DEG.

MAX. TORQUE INCLUDES SIZE EFFECT (REYNOLDS NO. ETC) APPX. X 1.22629 FOR 53.25 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: 54"-R1A; 3/7.5

CLASS 75

DISC SIZE: 53.062 INCHES

SYMMETRICAL DISC

SHAFT DIA.: 4.375 INCHES

BRG. COEF. OF FRCTN.: 5.00000E-03

SEATING FACTOR: 15

INLET PRESS. VAR. MAX.: 49.9846 PSIA

OUTLET PRESSURE (P6): 34.13 PSIA (72 DEG. ACTUAL PRESS. ONLY (VAR.))

MAX. ANG. FLOW RATE: 110150. CFM; 218647. SCFM; 12019.6 LB/MIN

CRIT. SONIC FLOW-90DG: 59572. LB/MIN AT 25.7975 INLET PSIA

VALVE INLET DENSITY: .10912 LB/FT³-MIN. .130799 LB/FT³-MAX.

FULL OPEN DELTA P: 13.4968 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. (75-60 DEG.)

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	P1	P2	DEL P	PRESS.	FLOW	FLOW	TD	TB+TH	TIME (LOCA)
APPRX. PSIA	PSIA	PSI	RATIO	(SCFM)	(LB/MIN)	----INCH LBS----	TD-TB-TH	SEC.	
30	41.70	22.35	19.35	.536	218646	12019	40667	1847	38820 3.00
25	43.99	14.87	29.12	.338	152176	8365	41416	2046	39369 3.38
20	45.36	14.78	30.58	.326	95700	5260	29366	2132	27233 3.65
15	45.84	14.72	31.13	.321	52410	2881	12904	2185	10718 3.75
10	46.52	14.71	31.81	.316	26181	1439	9150	2288	6862 3.85
5	48.23	14.70	33.52	.305	8562	470	5434	2396	3037 4.12
0	49.98	14.70	35.28	.294	0	0	44886	2353	42533 4.50

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47240 IN-LBS @ 0 DEG.

MAX. DYN. - BEARING - HUB SEAL TORQUE (M/M) = 41416 IN-LBS @ 25 DEG.



SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 35 DEG.

MAX.ANG.FLOW RATE: 136622. CFM: 271193. SCFM: 14908.2 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47298 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 75262 IN-LBS @ 35 DEG.

AT 3 SEC.DELAY TIME TO 4.75 CLOSED VLV. (LOCA) TIME(41.7 TO 51.191 PSIA
UPSTR.PRESS.)

REYNOLDS NO.FACTOR(MULTIPL.)= 1.34857
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9= 1.48508

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 40 DEG.

MAX.ANG.FLOW RATE: 198618. CFM: 394255. SCFM: 21673.3 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47352 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 134174 IN-LBS @ 35 DEG.

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

REYNOLDS NO.FACTOR(MULTIPL.)= 1.33881
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9= 1.47434

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 45 DEG.

MAX.ANG.FLOW RATE: 277186. CFM: 550210. SCFM: 30246.6 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47395 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 139173 IN-LBS @ 40 DEG.

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

REYNOLDS NO.FACTOR(MULTIPL.)= 1.31068
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9= 1.44335

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 50 DEG.

MAX.ANG.FLOW RATE: 270127. CFM: 536199. SCFM: 29476.3 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47439 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 257751 IN-LBS @ 45 DEG.

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

REYNOLDS NO.FACTOR(MULTIPL.)= 1.27509
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9= 1.40416



SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 55 DEG.

MAX.ANG.FLOW RATE: 333144. CFM; 661287. SCFM; 36352.7 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47481 IN-LBS @ 0 DEG.

MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 292362 IN-LBS @ 50 DEG.

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

REYNLDS NO.FACTOR(MULTIPL.)= 1.27452

TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9= 1.40353

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 60 DEG.

MAX.ANG.FLOW RATE: 401607. CFM; 797185. SCFM; 43823.4 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47522 IN-LBS @ 0 DEG.

MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 402742 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.)= 1.25547

TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9= 1.38255

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 65 DEG.

MAX.ANG.FLOW RATE: 478813. CFM; 950436. SCFM; 52248.1 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47563 IN-LBS @ 0 DEG.

MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 520710 IN-LBS @ 65 DEG.

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

REYNLDS NO.FACTOR(MULTIPL.)= 1.22865

TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9= 1.35302

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 70 DEG.

MAX.ANG.FLOW RATE: 543479. CFM; 1078798 SCFM; 59304.5 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47602 IN-LBS @ 0 DEG.

MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 569863 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.)= 1.2232

TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9= 1.34702



SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 75 DEG.

MAX.ANG.FLOW RATE: 624134. CFM; 1238898 SCFM; 68105.6 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47639 IN-LBS @ 0 DEG.

MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 626775 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.) = 1.21205

TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9 = 1.33475

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 80 DEG.

MAX.ANG.FLOW RATE: 654186. CFM; 1298550 SCFM; 71384.9 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47675 IN-LBS @ 0 DEG.

MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 654312 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.) = 1.20805

TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9 = 1.33034

SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 85 DEG.

MAX.ANG.FLOW RATE: 691429. CFM; 1372477 SCFM; 75448.9 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M) = 47707 IN-LBS @ 0 DEG.

MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 675698 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.) = 1.20502

TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)+(P6/P2)+J9 = 1.327



ATTACHMENT 4
GENERAL ARRANGEMENT
AND CROSS-SECTION DRAWINGS

