ADVANCE COPY

ISOLATION/PURGE VALVE ANALYSIS FOR

6"-1200 BUTTERFLY VALVE

Project Site	Susquehanna Steam Electric Station
	Berwick, Pennsylvania
Customer	Pennsylvania Power & Light
Engineer	Bechtel Power Corporation
Specification No.	8856
Original Purchase Order	8856-P-31-AC
Original Pratt Job No.	D-0026-3
Valve Tag Nos.	HBB-BF-AO-5721

General Arrangement Drawings	C-2600	Rev. 5
Cross Section Drawing	C-2988	Rev. 2

Prepared by: Rao N. Koza

Date: 3/12/82

Reviewed by: T. J. Ulrona

Date: 3-12-82

Certified by: Boni M. Zarda

Date: 3/12/82

BONI M. ZAFCLA
30701
REGISTERED PROFESSIONAL ENGINEER OF ILLINOIS

[illegible]

CONTENTS

I. Introduction

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

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

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

II. Considerations

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

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

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

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

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

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

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

7&8. Drawings or written description of valve orientation with respect to piping immediately upstream, as well as direction of valve closure, are furnished by customer/engineer. In this report, 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 readiation 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, psi

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

U = bearing coefficient of friction

d = shaft diameter, in.

Fluid dynamic torque is calculated from the following equations:

For subsonic flow

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

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

For sonic flow

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

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

Where

T_D = fluid dynamic torque, in-lbs.

F_{RE} = Reynold number factor

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

P_1 = upstream static pressure at flow condition, psia

P_2 = downstream static pressure at flow condition, psia

D = disc diameter, in.

C_{T1} = subsonic torque coefficient

C_{T2} = sonic torque coefficient

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

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

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

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

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

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

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

D-34933(D0026-3)   TORQUE TABLE   1   3 / 10 / 82

JOB:SUSQUEHANA/BECHTEL
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.)= 1354.57   FEET/SEC AT 265   DEG.

ABSOL.MAX.TORQUE(FIRST SONIC)AT 72-68 DG.VLV.ANG.=   478 IN-LBS @ 68 DEG.
MAX.TORQUE INCLUDES SIZE EFFECT(REYNOLDS NO.ETC)APPX. X .996595   FOR 6   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: 6"-1200   CLASS 150
DISC SIZE: 5.2 INCHES   OFFSET ASYMMETRIC DISC
SHAFT DIA.:   1   INCHES
BEARING TYPE: BRONZE
SEATING FACTOR: 15
INLET PRESS.VAR.MAX.: 48.2   PSIA
OUTLET PRESSURE(P6): 17.2   PSIA (72 DEG. ACTUAL PRESS.ONLY(VAR.))
MAX.ANG.FLOW RATE: 6658.94   CFM; 7698.92   SCFM; 423.231   LB/MIN
CRIT.SONIC FLOW-90DG: 518.539   LB/MIN AT 19.651   INLET PSIA
VALVE INLET DENSITY: 6.35583E-02   LB/FT^3-MIN. .129262   LB/FT^3-MAX.
FULL OPEN DELTA P: 7.74903   PSI
SYSTEM CONDITIONS:
PIPE IN-PIPE-OUT -AND- AIR/STEAM MIXTURE SERVICE @ 265 DEG.F
MINIMUM 0.75 DIAM. PIPE DOWNSTREAM FROM CENT.LINE SHAFT.

P1 ABS. PRESSURE(ADJ.)FOLLOWS TIME/PRESS.TRANSIENT CURVE.

	--5 IN.MODEL EQUIV.VALUES------				ACTUAL SIZE VALUES-----					
ANGLE APPRX.	P1 PSIA	P2 PSIA	DELP PSI	PRESS. RATIO	FLOW (SCFM)	FLOW (LB/MIN)	TD ----	TB+TH INCHLBS----	TD-TB-TH	TIME(LOCA) SEC.
90	23.70	15.50	8.20	.654 CR	7698	423	140	12	128	0.20
85	27.98	15.80	12.18	.565	11888	653	169	14	154	0.44
80	31.32	15.99	15.32	.511	12896	708	209	18	190	0.68
75	34.23	16.07	18.16	.469 CR	13344	733	370	32	337	0.90
72	35.78	15.96	19.82	.446 CR	12888	708	489	42	446	1.02
70	36.74	15.94	20.80	.434 CR	12553	690	457	39	417	1.10
65	38.84	15.73	23.11	.405 CR	11543	634	439	38	401	1.27
60	40.51	15.50	25.00	.383 CR	10054	552	345	37	308	1.41
55	41.72	15.26	26.46	.366 CR	8558	470	302	45	256	1.52
50	42.45	15.10	27.36	.356	7045	387	225	52	172	1.58
45	42.70	14.97	27.73	.351	6983	383	187	59	128	1.60
40	42.80	14.89	27.91	.348	4807	264	130	64	65	1.62
35	43.11	14.81	28.30	.343	3674	201	82	69	12	1.68
30	43.60	14.76	28.84	.339	2779	152	49	73	-24	1.79
25	44.24	14.73	29.51	.333	1983	109	35	78	-43	1.93
20	45.01	14.71	30.30	.327	1241	68	28	83	-55	2.10
15	45.86	14.70	31.16	.321	705	38	12	88	-76	2.30
10	46.74	14.70	32.04	.315	338	18	7	92	-85	2.52
5	47.57	14.70	32.87	.309	108	5	5	96	-90	2.76
0	48.20	14.70	33.50	.305	0	0	628	88	539	3.00

SEATING + BEARING + HUB SEAL TORQUE (M/M)=   628 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) =   489 IN-LBS @ 70 DEG.

B. Valve Stress Analysis

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

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

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

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

where

S_{max} = maximum combined stress, psi

T_1 = direct tensile stress, psi

T_2 = tensile stress due to bending, psi

S_1 = direct shear stress, psi

S_2 = shear stress due to torsion, psi

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

C. OPERATOR EVALUATION

Model: Bettis 521C-SR 60

Rating: 7,000 in-lbs. at full open and closed positions only.

4620 in-lbs at 68°

4000 in-lbs at 45° (minimum rating)

Max. Valve Torque: 628 in-lbs.

The maximum torque generated during a LOCA induces reactive forces in the load carrying components of the actuator. Since the LOCA induced torque derived in this analysis is less than the maximum absorbtion rating of the operator, it is concluded that the Bettis models furnished are structurally suitable to withstand combined LOCA and seismic loads.

IV. Conclusion

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

ATTACHMENT 1A

PRATT PROPOSAL LETTER

TELEPHONE 312-844-1000 · TELEX 720-121

PRATT

HENRY PRATT COMPANY
creative engineering for fluid systems
401 SOUTH HIGHLAND AVENUE · AURORA, ILLINOIS 60507

April 16, 1981

Bechtel Power Corporation
P.O. Box 3965
San Francisco, CA 94119

Attention: Mr. E.B. Poser
Project Engineer

SUBJECT: Susquehanna Steam Electric Station
Containment Isolation/Purge Valve Analysis

Gentlemen:

With reference to your recent inquiry regarding suitability of the valves and actuators to withstand aerodynamic LOCA conditions, please note the following:

1. Torque calculations will be performed for aerodynamic torque generated as a result of LOCA. These calculations will be performed using the following data to be furnished by you.

   A. Containment Pressure - Time Curves

   B. Containment Temperature - Time Curves

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

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

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

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

PRATT

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

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

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

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

2. Based on the above results, a static load stress analysis will be provided for valve components affected by the dynamic torque loadings in combination with pressure and seismic loads.

The actuator supplier will be asked to verify the suitability of the actuator for the reaction or back drive force resulting from aerodynamic torque conditions.

3. The cost of performing the evaluation of the valve components will be $12,800 each size for 6", 18" and 24" valves.

4. The completion of this analysis is projected to be twenty-six (26) weeks after receipt of purchase order and data requested above based on availability of engineering schedule.

5. Our response to NRC's criteria for demonstrating operability of purge valves is included in the analysis.

This proposal is for investigative analysis only and is not intended to guarantee the adequancy of the equipment as furnished when subjected to LOCA loads currently being defined.

The proposal is valid for thirty (30) days. The terms of payment will be Net 30 Days.

We hope you will find the proposal responsive to your needs. If we can be of any additional assistance in this matter, please advise.

Very truly yours,

HENRY PRATT COMPANY

Glenn Beane

Glenn L. Beane
Manager-Application Engineering

GLB/tl

ATTACHMENT 1B

CUSTOMER/ENGINEER RESPONSE TO
REQUEST FOR INFORMATION

APPLICATION ENGRG.

JUN 19 1981

Bechtel Power Corporation
Engineers—Constructors

Fifty Beale Street
San Francisco, California
Mail Address: P.O. Box 3965, San Francisco, CA 94119

Henry Pratt Company
401 South Highland Avenue
Aurora, Illinois 60507

JUN 15 '81 141808

Attention: Mr. G. L. Beane

Subject: Susquehanna Steam Electric Station
Units 1 and 2 Job 8856
P.O. 8856-P-31-AC, Containment
Isolation/Purge Valve Analysis

Pratt SO# 34933

Gentlemen:

In order to perform the analysis Henry Pratt requested certain information. The following is our reply:

A. Containment pressure time curve is attached.

B. Containment temperature time curve is attached.

C. A back pressure of 19.7 psia should be used in this analysis. This back pressure is per the assumptions in your letter of April 16, 1981.

D. Minimum delay time is 0.1 seconds. Maximum delay time is 5 seconds.

E. Isometric drawings for both units are attached. We believe that Henry Pratt is in a better position to determine the direction of valve closure as viewed from the operator end. This information is not apparent on the drawings you submitted to Bechtel.

In addition, if Henry Pratt's 16 week analysis report shows the valves to be unqualified, Henry Pratt will state at what angle the valves must be blocked open in order to meet the NRC's interim position. Henry Pratt will also make recommendations on how to block the valves and to provide a detailed drawing of the stop.

We trust that the foregoing information is satisfactory and will enable you to complete the qualification of the subject valves. If you have any questions, please contact Al Daily at (415) 768-9235 or A. Tiongson at (415) 768-7770.

Very truly yours,

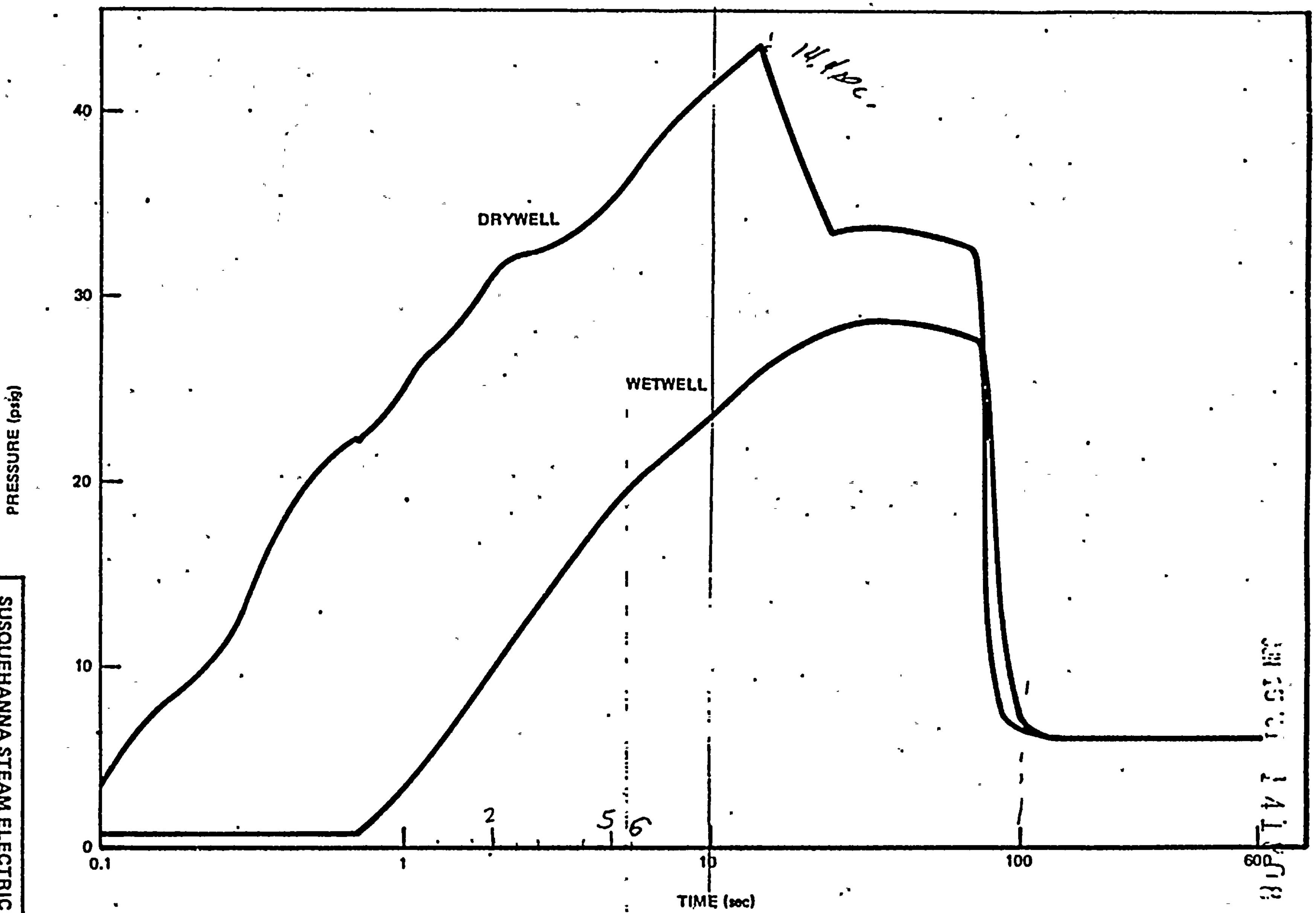
E. B. Poser
Project Engineer

Written Response Req'd: No
Design Document Changes: No
CHN/APT/cgs
WP30/3-1

cc: Mr. T. M. Crimmins, Jr. (PL) w/a

cc T. Wrona Pratt Engr.

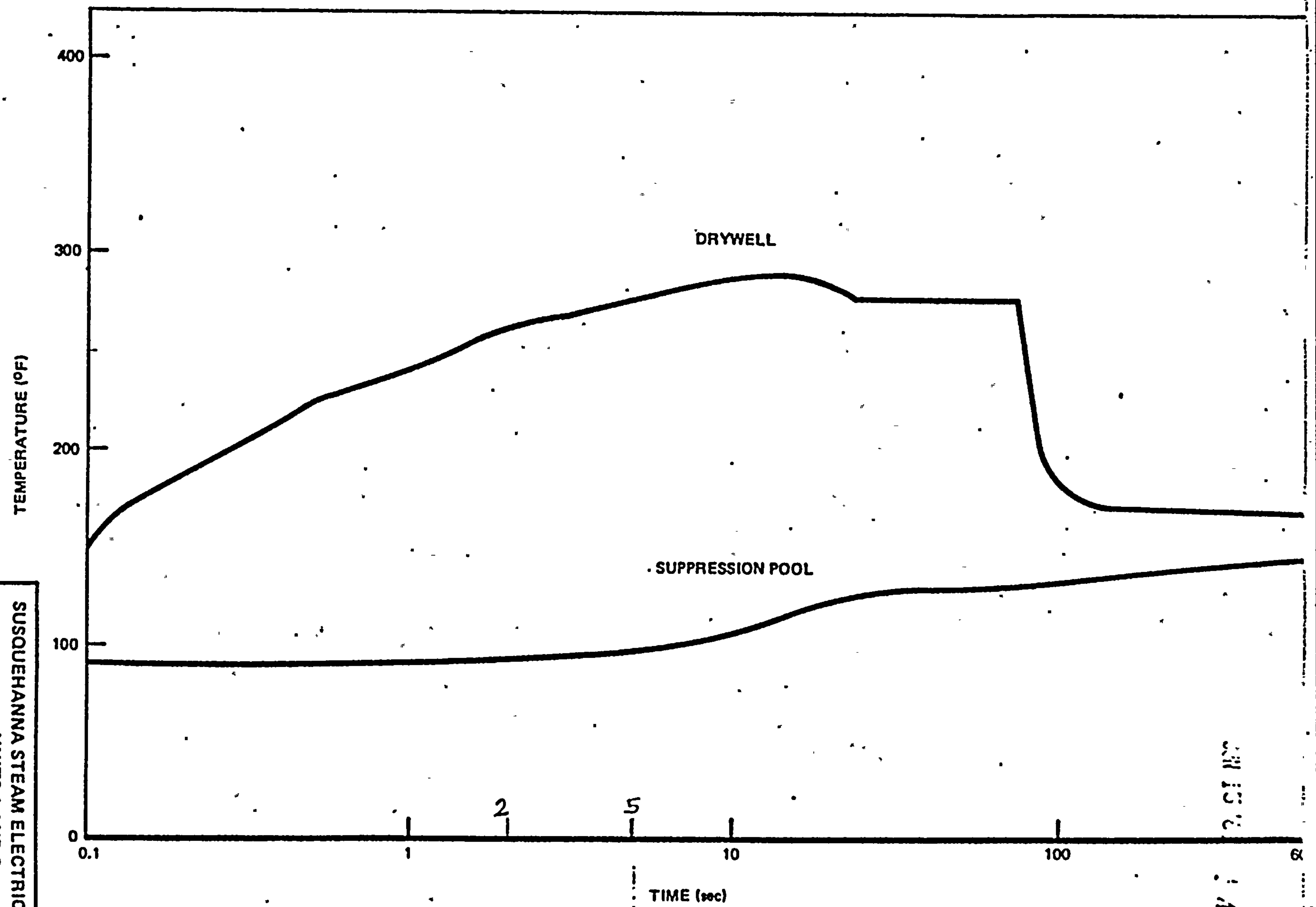
5026



SUSQUEHANNA STEAM ELECTRIC STATION
UNITS 1 AND 2
FINAL SAFETY ANALYSIS REPORT

PRESSURE RESPONSE FOR
RECIRCULATION LINE BREAK

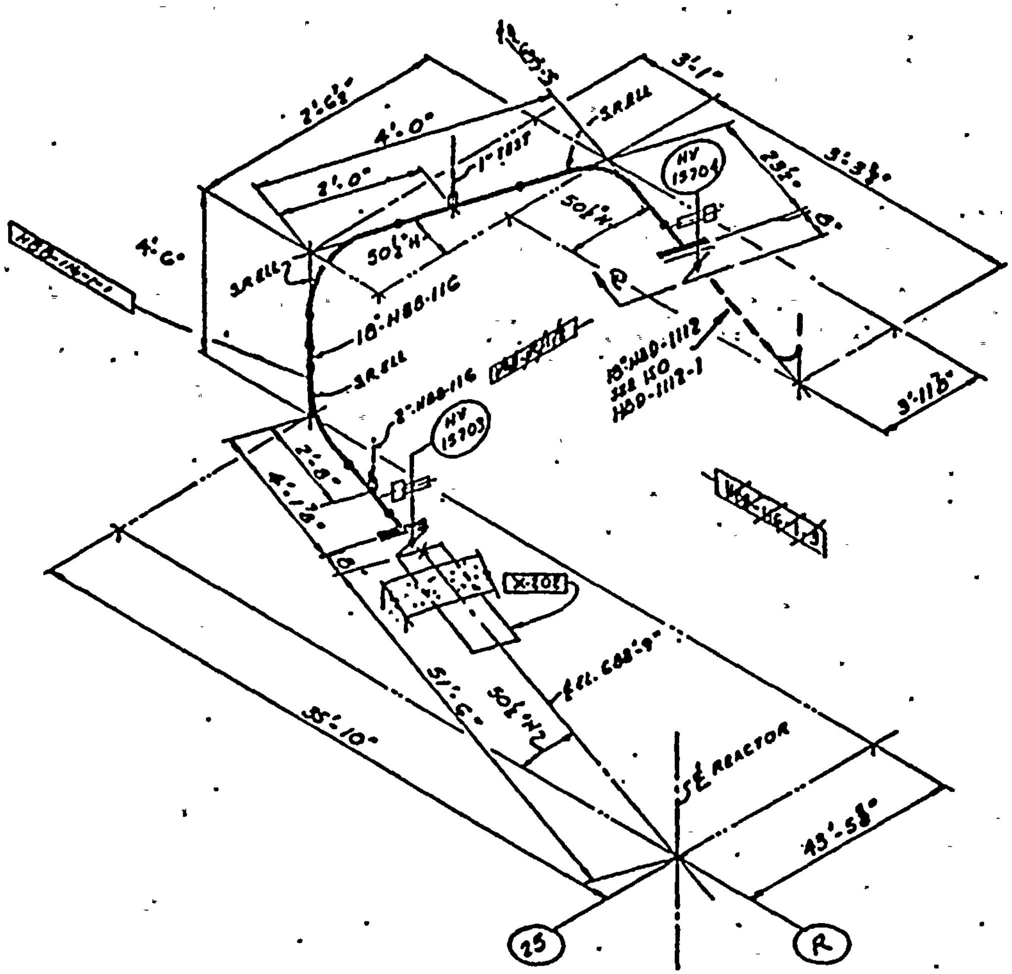
FIGURE 6.2-2



SUSQUEHANNA STEAM ELECTRIC STATION
UNITS 1 AND 2
FINAL SAFETY ANALYSIS REPORT

TEMPERATURE RESPONSE FOR RECIRCULATION LINE BREAK

FIGURE 6.2-3



NO PIPE SUPPORTS REQ'D

COMPONENTS ON
THIS DWG. ARE Q-LISTED
SPEC. 8856-G-9 APPLIES

REFERENCE DRAWINGS

M-157 SH.1 REV.4 P.&I.D.
M-27-3 " 17 PIPING PLAN-AREA 27

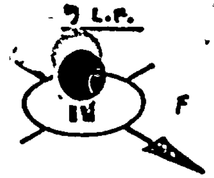
ISSUED FOR FABRICATION

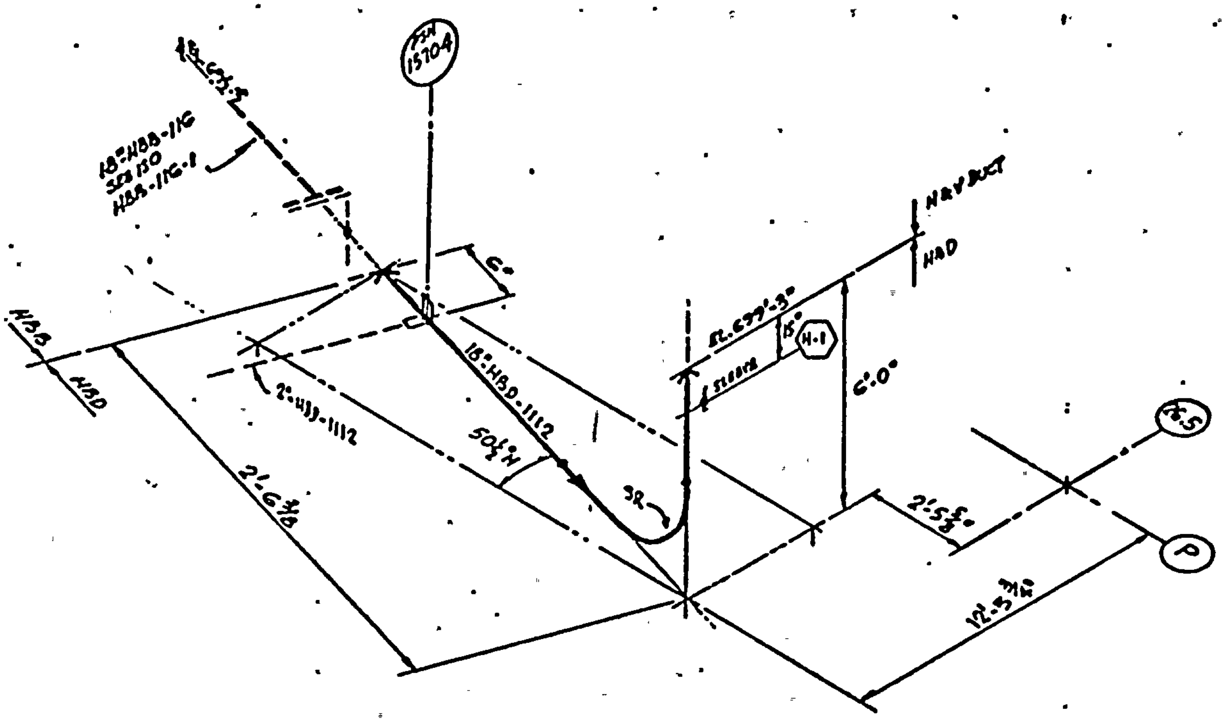
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNITS 1 AND 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 1

8856 | HBB-116-1 | 4





REFERENCE DRAWINGS

M- 157 SH1 REV. 8 P.&I.D.
M- 27-3 " 18" PIPING PLAN - AREA 27

ORIGINALLY ISSUED AS PART OF
150 HBB-116-1

2		ADDED PIPE SUPPORT	RJ					
1		ISSUED FOR CONSTRUCTION	RJ					
0		ISSUED FOR FABRICATION	RJ					

DRAWN RJ

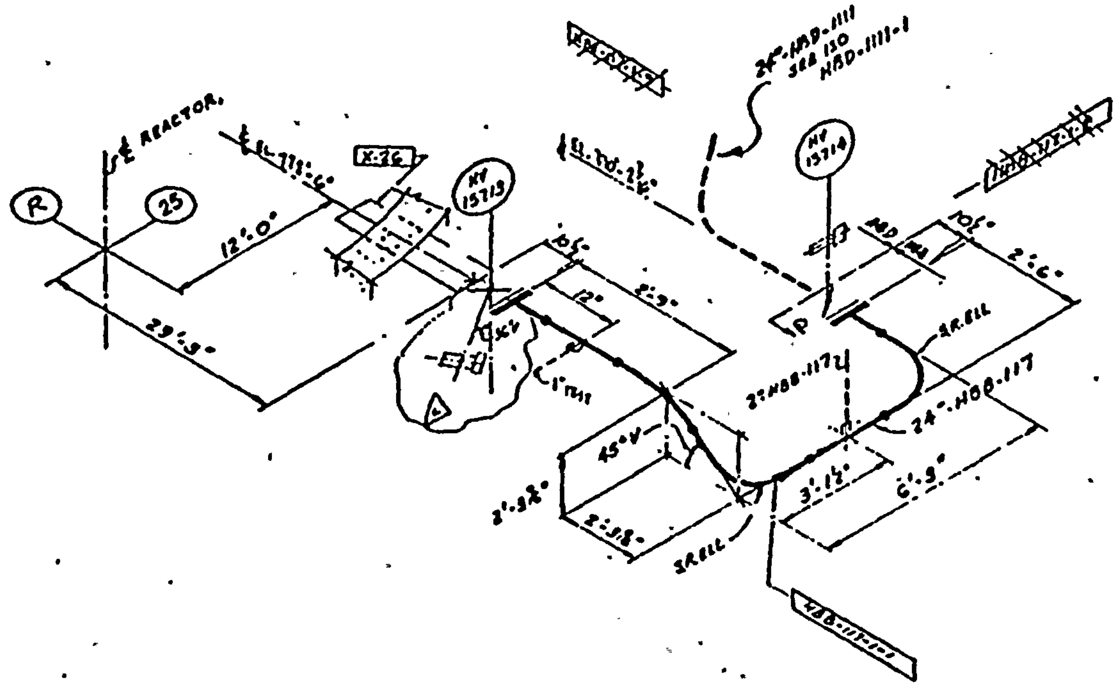
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATM. CONTROL - UNIT 1

JOB NO. 8856 — DRAWING NO. HBD-1112-1 — REV 2

01 | 150 REAC DD CTMT ATMOS CON | M



REFERENCE DRAWINGS

M-157 SH.1 REV. 4 P.&I.D.
M-25-5 " " PIPING PLAN-AREA 25

NO PIPE SUPPORTS REQ'D

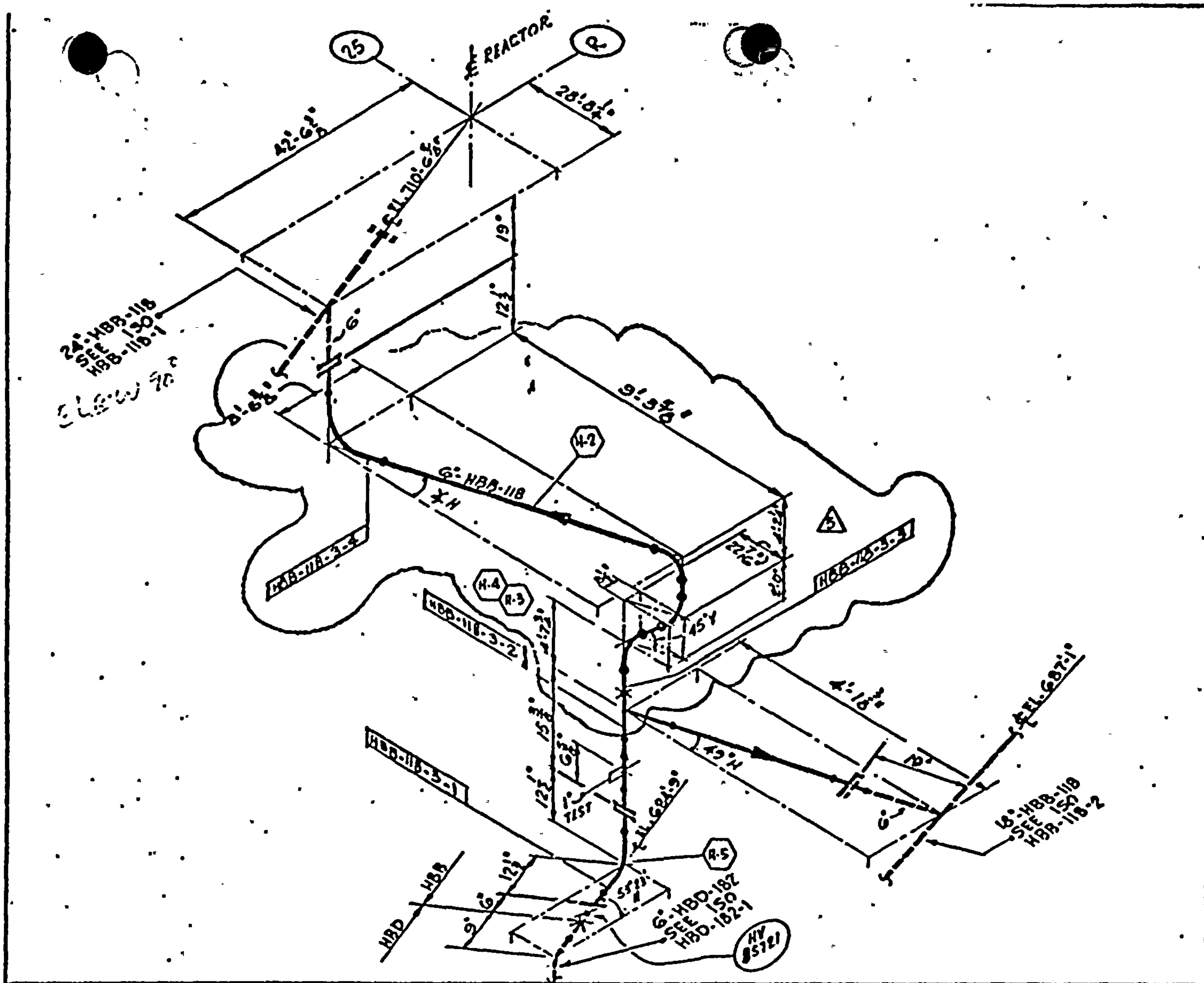
COMPONENTS ON THIS DWG. ARE Q-LISTED SPEC. 8856-G-9 APPLIES

Q

PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2
BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 1

8856 | HBB-117-1 | 6



N

REV. 5 NOTES

REVISED SPOOL No. 3. DIM. 3'-8 7/8" WAS 2'-9 7/8", 3'-3 7/8" WAS 7'-5 3/16" AND ADDED OFF-SET.

REFERENCE DRAWINGS

M-157 SH-1 REV-4 P.&I.D.
M-29-9 REV. 25 PIPING PLAN AREA-29

COMPONENTS ON THIS DWG. ARE Q-LISTED SPEC. 8856-G-9 APPLIES

1150 REACT BD CTMT ATM CONT

No.	Date	Revisions	By	Chk				
5	[illegible]	SEE REV. 5 NOTE AS PER FCR-2451	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
4	[illegible]	ADDED PIPE SUPPORTS	R.J.	[illegible]	[illegible]	[illegible]	[illegible]	
3	[illegible]	[illegible]	R.J.	[illegible]	[illegible]	[illegible]	[illegible]	
2	[illegible]	[illegible]	CDS	[illegible]	[illegible]	[illegible]	[illegible]	
1	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
0	[illegible]	ISSUED FOR FABRICATION	DKB	[illegible]	[illegible]	R.J.	[illegible]	

Scale ~ Designed ~ Drawn DKB

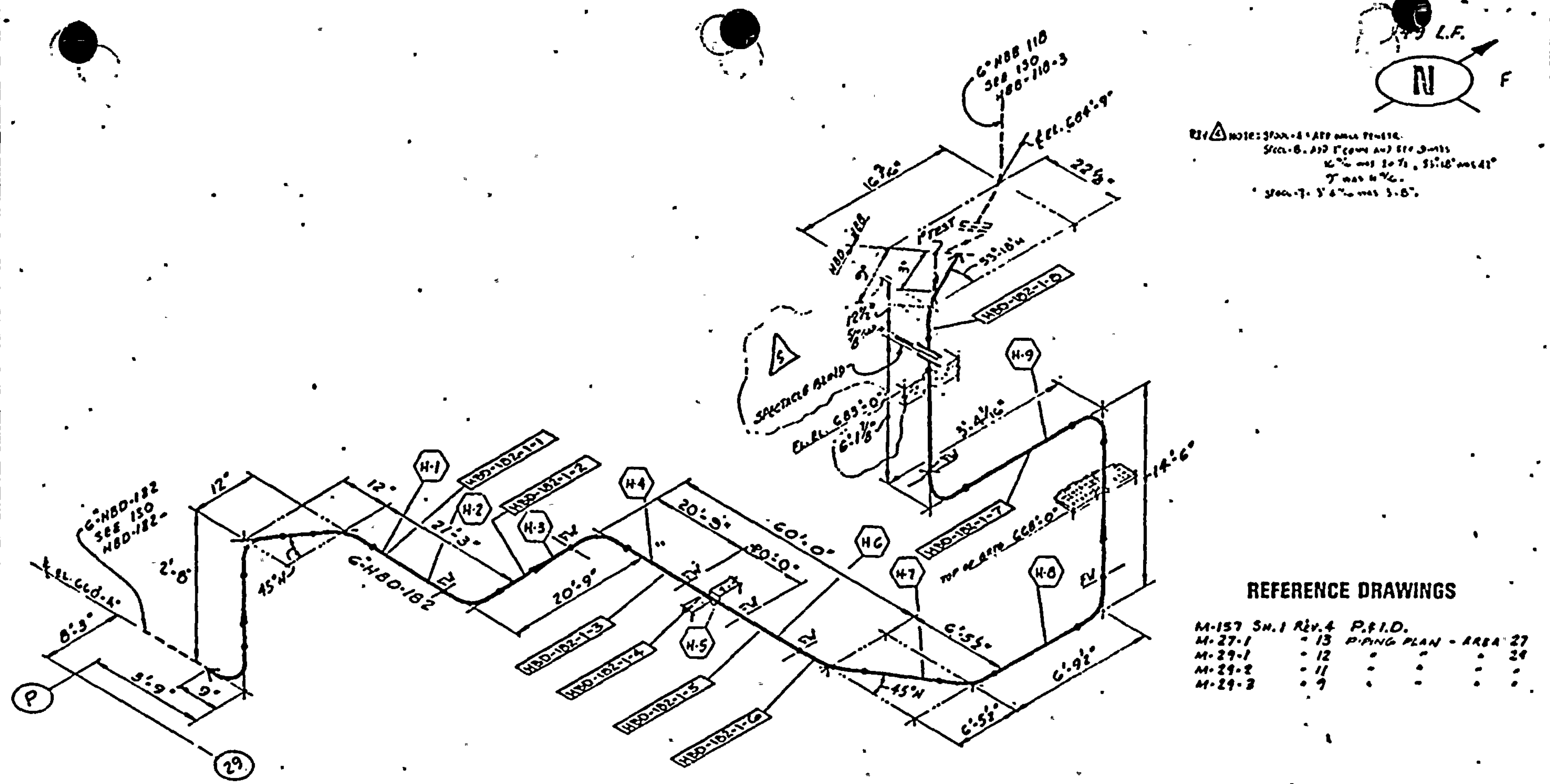
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

Q

ISOMETRIC – REACTOR BLDG. CONTAINMENT ATMOS. CONTROL-UNIT-1

Job No.	57-5 Drawing No.	Rev.
8856	HBB-118-3	5



REFERENCE DRAWINGS

M-157 SH. 1	REV. 4	P.&I.D.	
M-27-1	" 13	PIPING PLAN - AREA	27
M-29-1	" 12	" " "	29
M-29-2	" 11	" " "	"
M-29-3	" 9	" " "	"

ADDED PIPE SUPPORTS

ISSUED FOR FABRICATION

PENNSYLVANIA POWER & LIGHT COMPANY

ALLENTOWN, PENNSYLVANIA

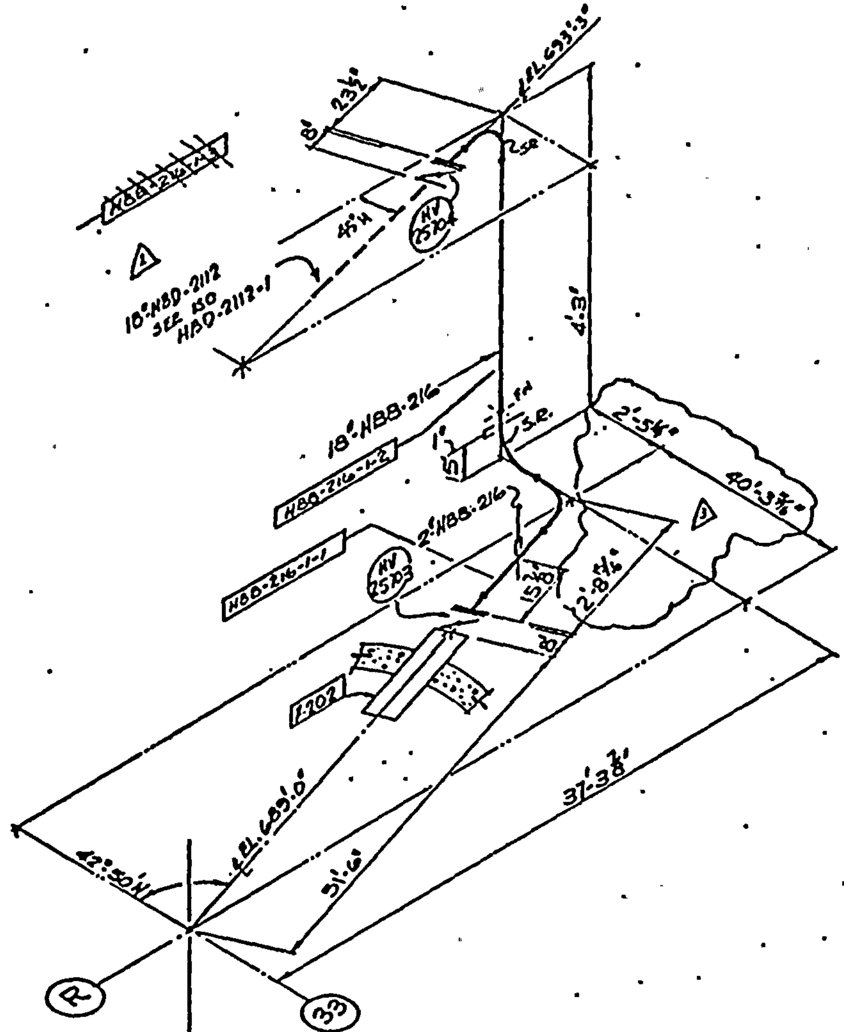
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.

CONTAINMENT ATMOS. CONTROL - UNIT 1

JOB NO.	DRAWING NO.	REV
8856	HBD-182-1	5



REFERENCE DRAWINGS

M-2157 SH.1 REV 0 P&ID
M-30-3 " " PIPING PLAN-AREA 30

COMPONENTS ON THIS DWG. ARE Q-LISTED SPEC. 8856-G-9 APPLIES

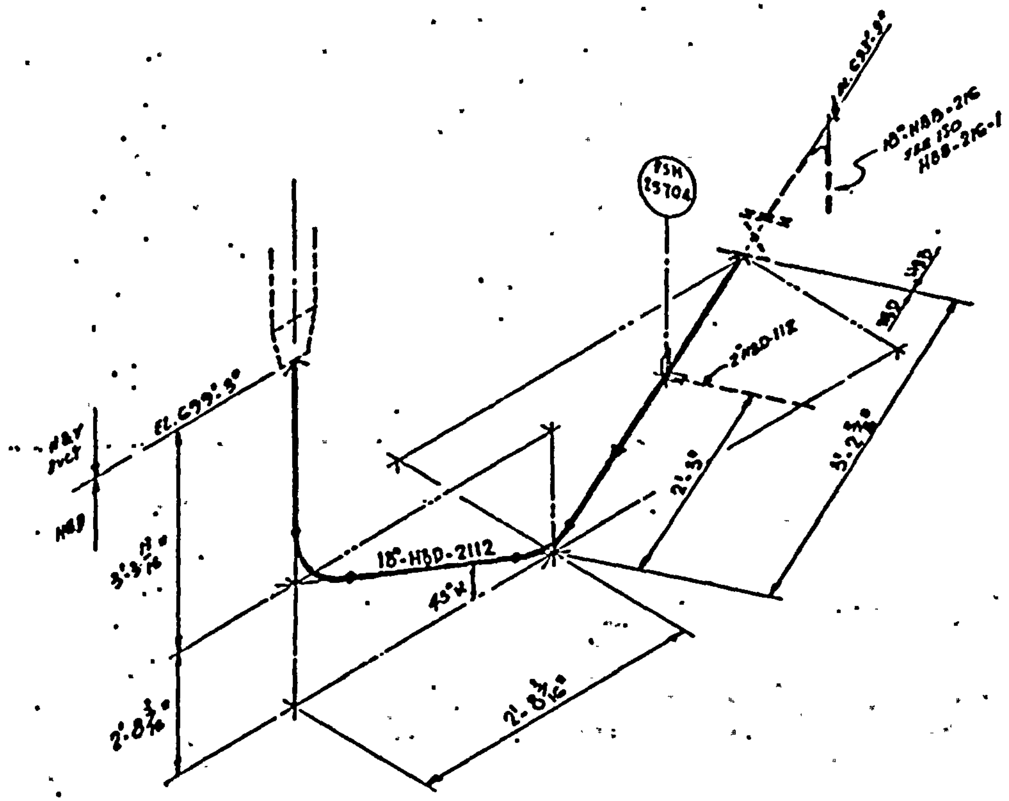
3		REVISED PER FCR P-...	DM	
2			RJ	
1			RJ	
0		ISSUED FOR FABRICATION	DRW	

PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION UNIT 1, UNIT 2
BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 2

8856 | 52-4 | HBB-216-1 | 3

G2 ISO REAC BLD CTMT ATMOS CON



REFERENCE DRAWINGS

M-2157 SH-1 REV.0 P.&I.D.
M-30-3 " PIPING PLAN-AREA 30

ORIGINALLY ISSUED AS PART OF
ISO HBB-216-1

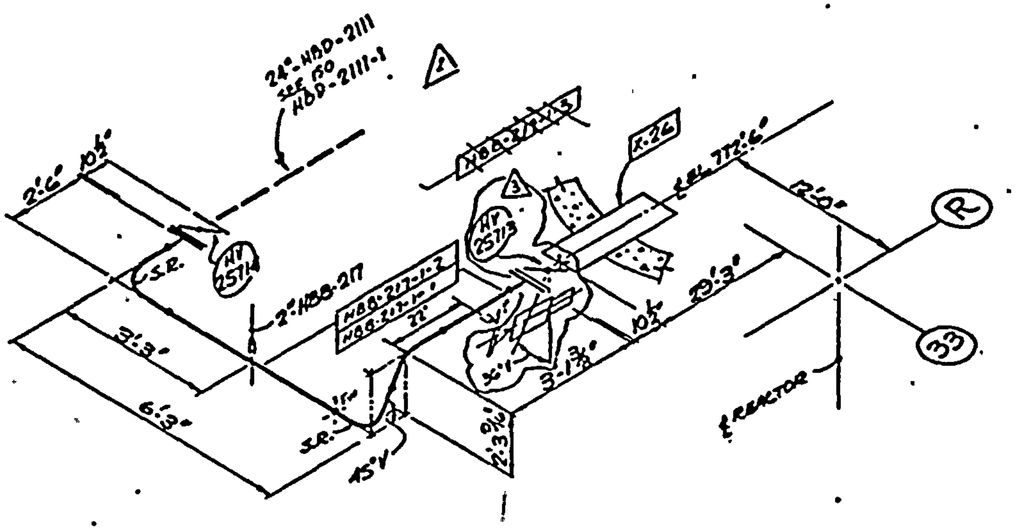
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2
BECHTEL - SAN FRANCISCO

ISOMETRIC- REACTOR BLDG.
CONTAINMENT ATM. CONTROL - UNIT 2

8856 | HBD-2112-1 | 1

157

REV. 3 NOTE
. ROTATE VALVE HV-25713



REFERENCE DRAWINGS

M-2157 SK.1 REV. 3 P&ID
M-32-5 " 20 PIPING PLAN-AREA 32

COMPONENTS ON THIS DWG. ARE Q-LISTED SPEC. 8856-G-9 APPLIES

Q

3		INCORPORATED PCR-R-3063 SEE REV 3 NOTE	53	FS				
2		[illegible]	RJ					
1		[illegible]	RJ					
0		ISSUED FOR FABRICATION	PRW					

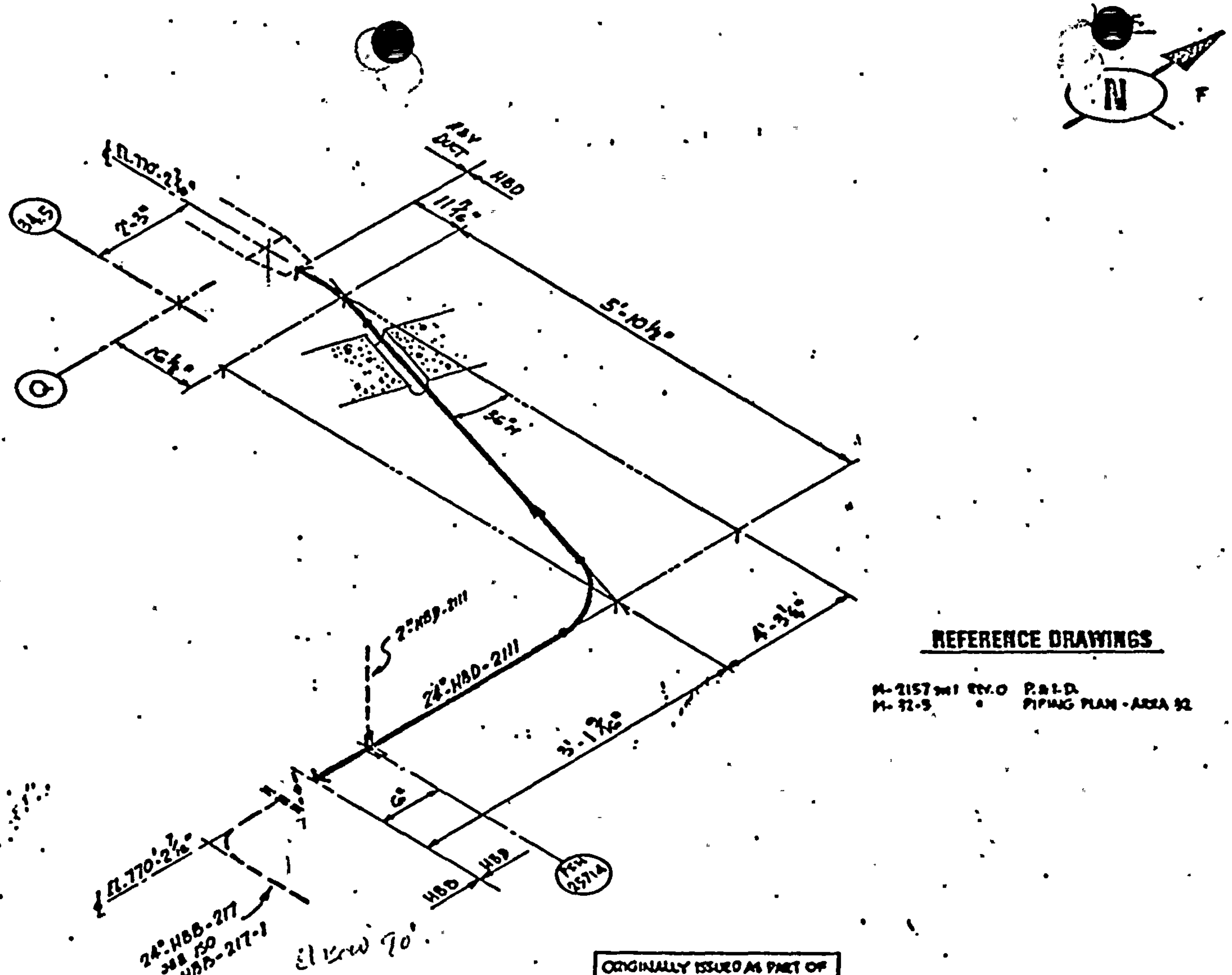
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 2

Job No.	Drawing No.	Rev
8856	57-4 HBB-217-1	3

G2 | ISO REAC 3D CTMT ATMOS CON | N



ORIGINALLY ISSUED AS PART OF
ISO HBB-217-1

REFERENCE DRAWINGS

M-2157 SH 1 REV. 0 P. & I. D.
M-32-5 " PIPING PLAN - AREA 32

△								
△	4/..	ISSUED FOR CONSTRUCTION	LJ					
△	3/..	ISSUED FOR FABRICATION	2.J					

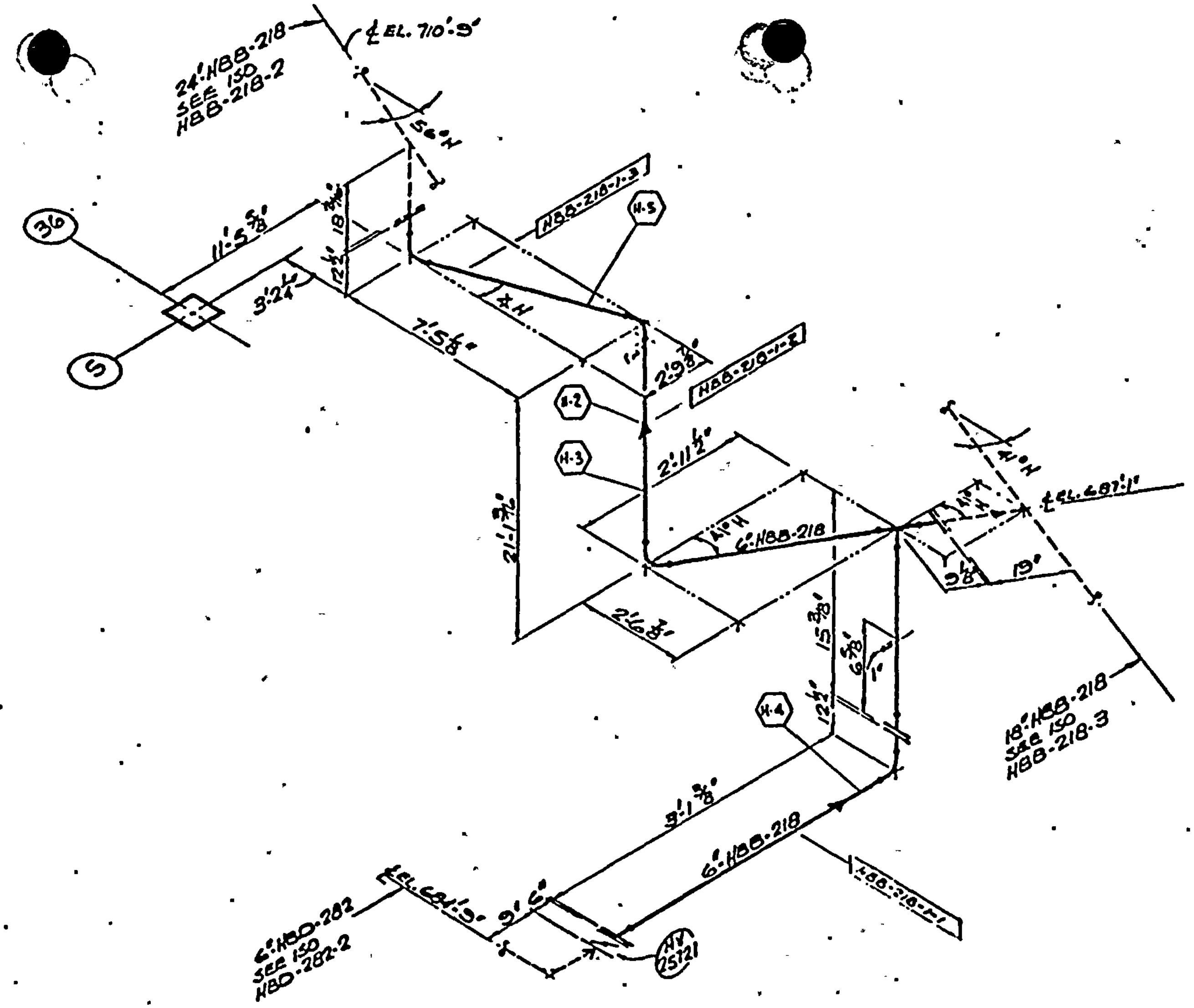
DRAWN RJ

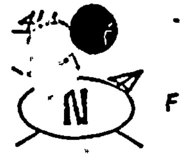
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG
CONTAINMENT ATM. CONTROL - UNIT 2

JOB No.	DRAWING No.	REV.
8860	HBD-2111-1	1





REFERENCE DRAWINGS

M-2157 SH.1 REV. P&ID
M-34-3 " " PIPING PLAN-AREA 34

COMPONENTS ON THIS DWG. ARE Q-LISTED SPEC. 8856-G-9 APPLIES

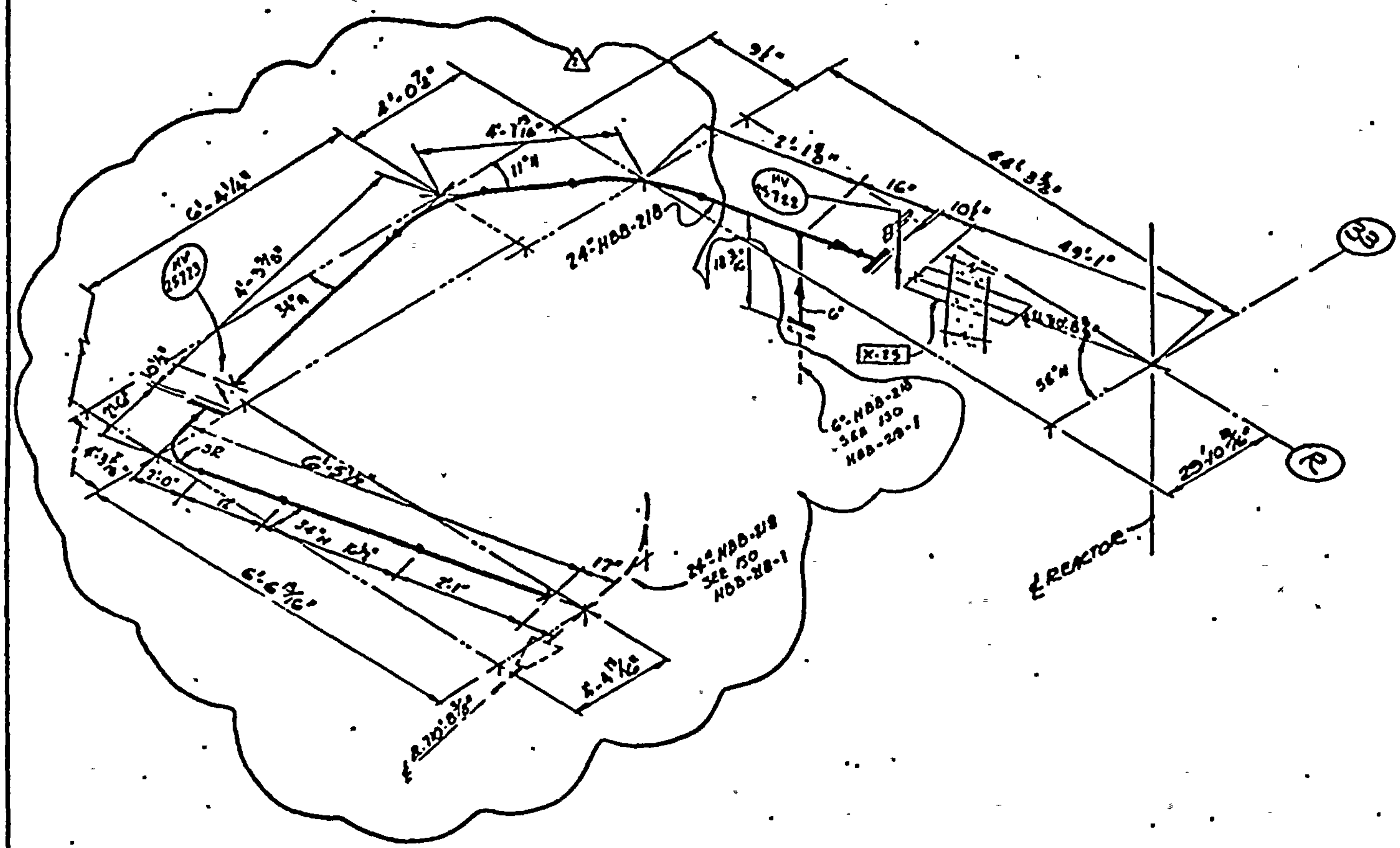
3		ADDED PIPE SUPPORTS		
2				
1		ISSUED FOR FABRICATION		

PENNSYLVANIA POWER & LIGHT COMPANY

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 2

8856 | 57-2 | HBB-218-1 | 2



REFERENCE DRAWINGS

M-2157 SH.1 REV. 1 P&ID
M-34-3 REV. 17 PIPING PLAN-AREA 34

COMPONENTS ON THIS DWG. ARE Q-LISTED SPEC. 8856-G-9 APPLIES

Q

ISSUED FOR FABRICATION

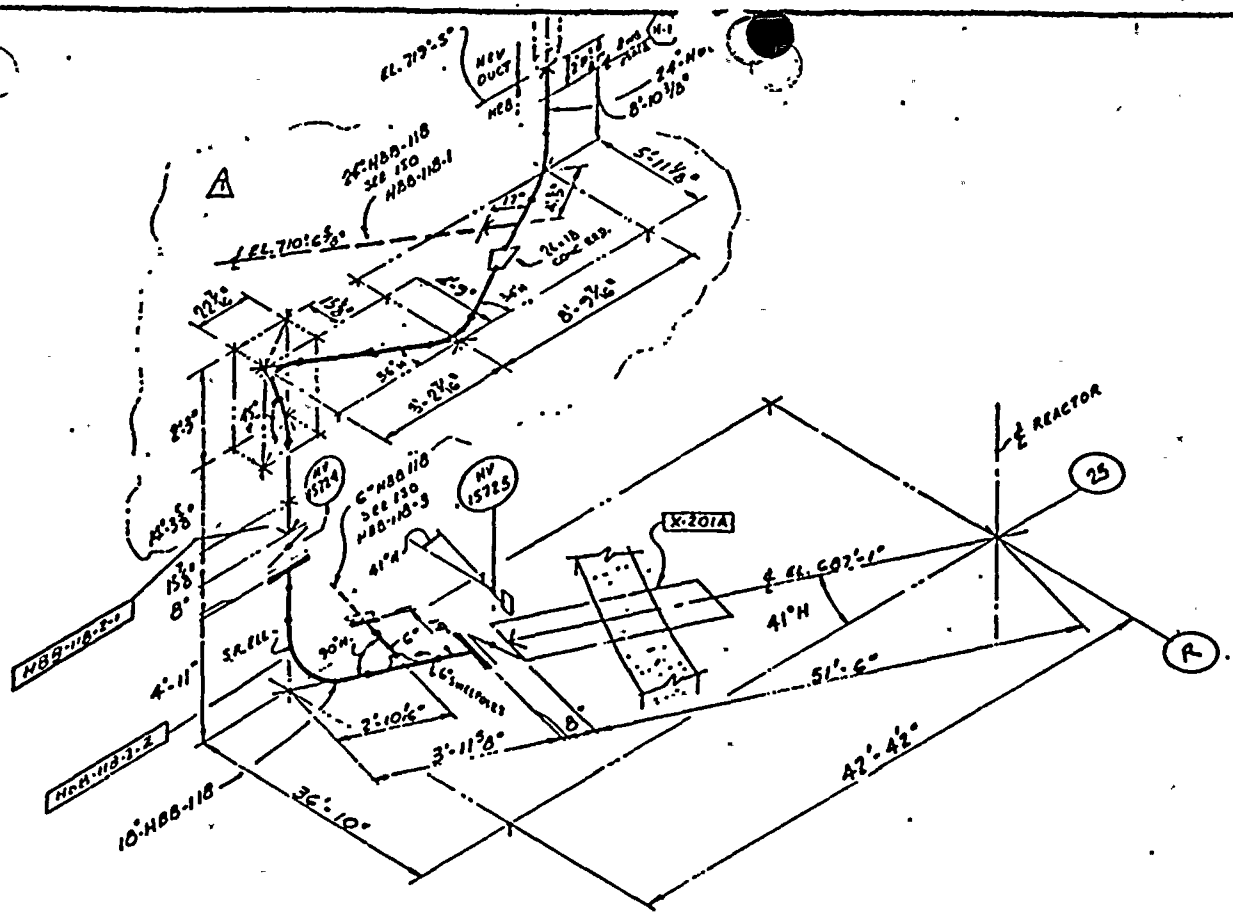
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 2

JOB No.	DRAWING No.	REV.
8856	HBB-218-2	2

ISO REAC BD CTMT ATMOS CON



52

N F

REV. △ NOTE: SPOOL-1: 24" BRANCH ELEV 710'-6 3/8 was 710'-9. REV. DIMS TO SUIT.

REV. △ NOTE: REROUTED CONN. TO HBV DUCT. BLDG. CONN. PER 150-1 FCR P-2142

REFERENCE DRAWINGS

M-157 SH. 1 REV. 4 P. & I. D.
M-29-5 " 11 PIPING PLAN - AREA 29

COMPONENTS ON
THIS DWG. ARE Q-LISTED
SPEC. 8856-G-9 APPLIES

NO	DATE	REVISION	BY	CHK				APPD
△	[illegible]	SEE REV △ NOTE	RJ					
△	[illegible]	SEE REV △ NOTE [illegible]	RJ					
△	[illegible]	SPOOL-2 [illegible]	RJ					
△	[illegible]	ADDED FOR SUPPORT H-1	RJ					
△	[illegible]	[illegible]	RJ					
△	[illegible]	[illegible]						
△	[illegible]	[illegible]	RJ					
△	[illegible]	ISSUED FOR FABRICATION						

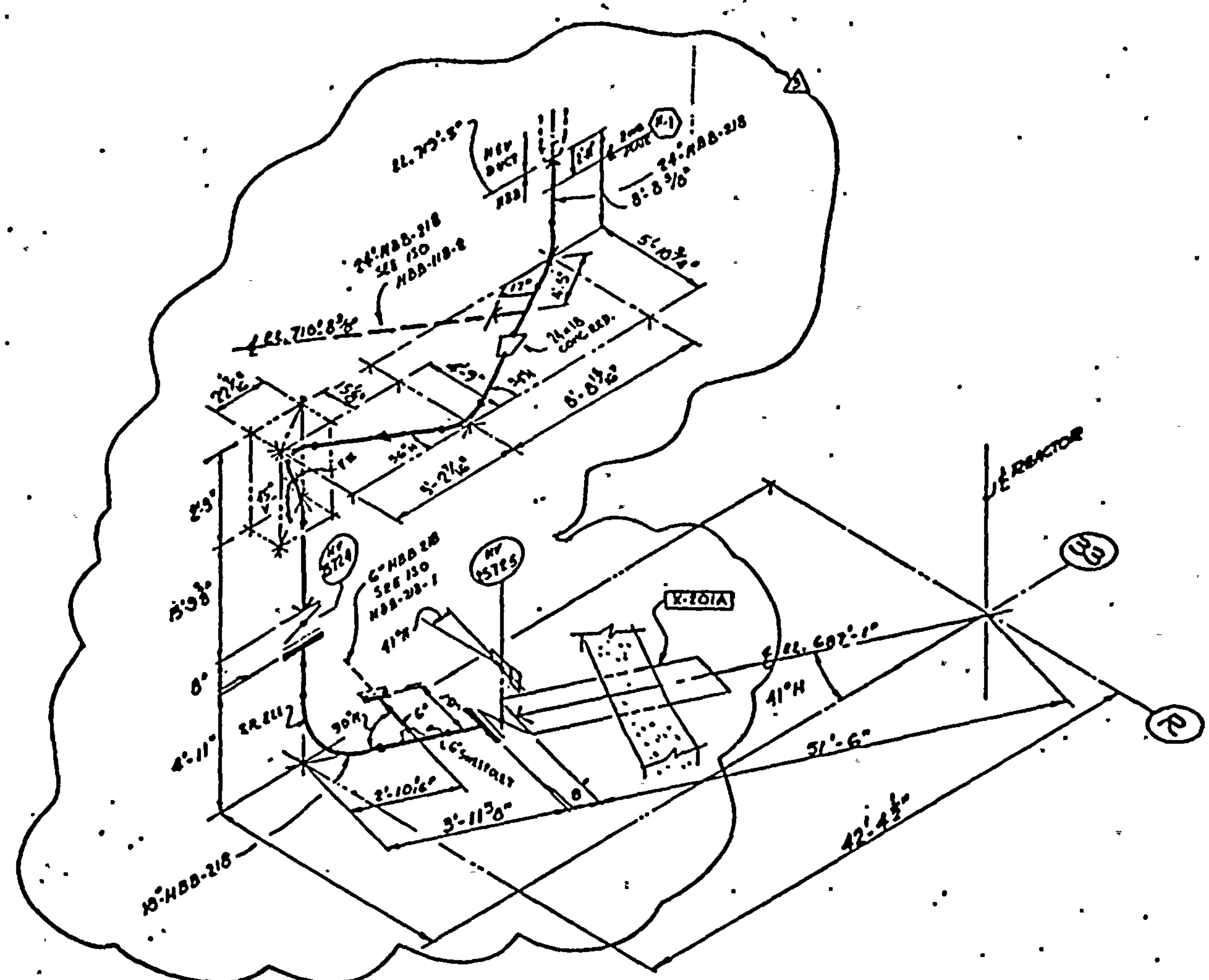
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 1

JOB NO.	DRAWING NO.	REV
8858	HBB-118-2	7

GIII 135 REACT BLDG CTMT ATM CONT



REFERENCE DRAWINGS

M-2157 SH.1 REV. 17 P&ID
M-34-3 PIPING PLAN-AREA 34

COMPONENTS ON THIS DWG. ARE Q-LISTED SPEC. 8856-G-9 APPLIES

No.	Date	Revision	By	Chk'd				
3		INCORP. FCR P-3648				ES		
2		ADDED PIPE SUPPORT	RJ	RER				
1			RJ					
0	7-4-77	ISSUED FOR FABRICATION	RRW					

Scale — | Drawn DRW

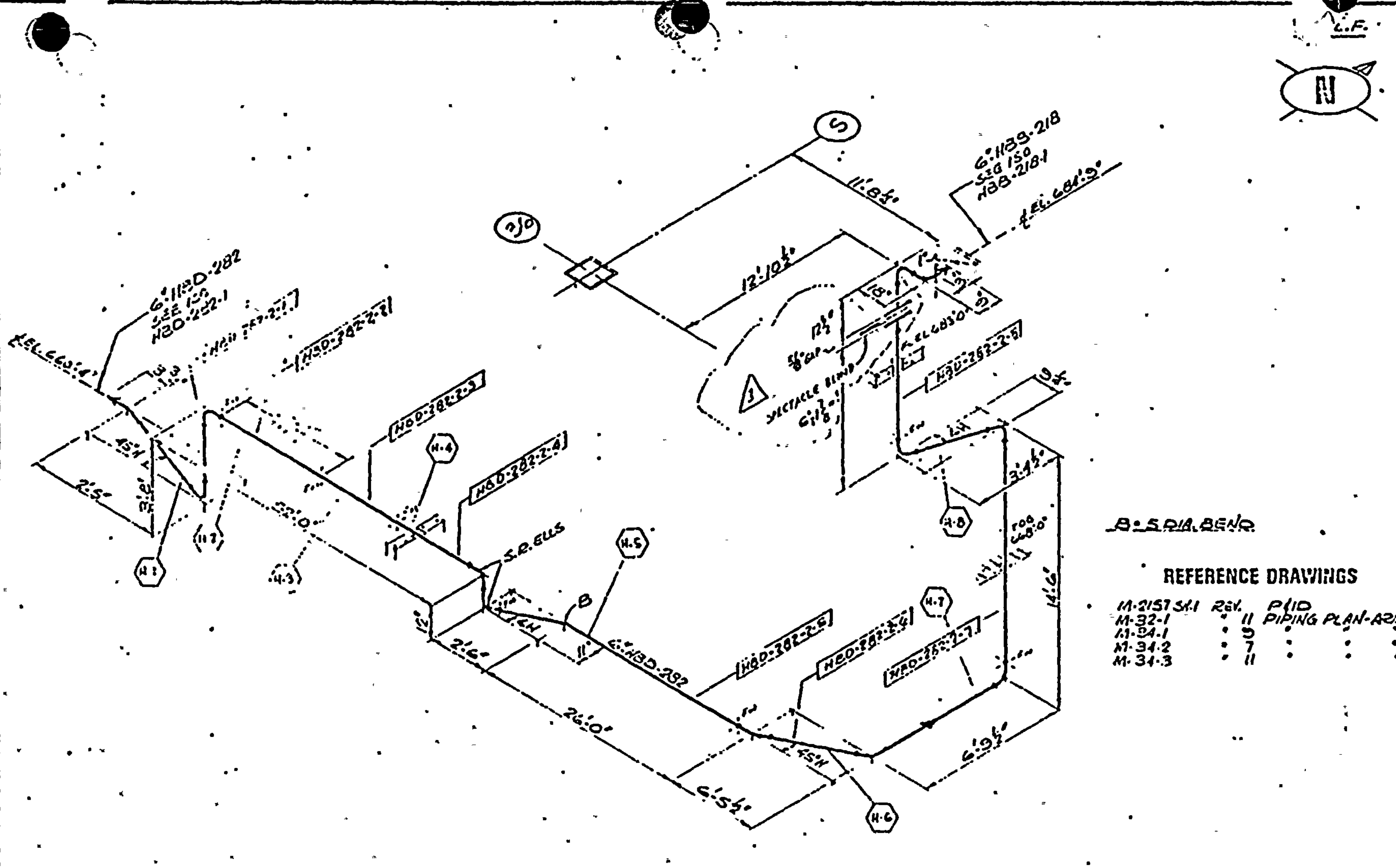
PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC-REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL- UNIT 2

Job No. 8858 | Drawing No. 57-3 HBB-218-3 | Rev. 3

G2 ISO REAC BD CTMT ATMOS CON M



B = 5 DIA. BEND

REFERENCE DRAWINGS

M-2151 SH.1	REV.	P&ID	
M-32-1	" 11	PIPING PLAN-AREA 32	
M-34-1	" 9	" " "	34
M-34-2	" 7	" " "	"
M-34-3	" 11	" " "	"

PENNSYLVANIA POWER & LIGHT COMPANY

ALLENTOWN, PENNSYLVANIA

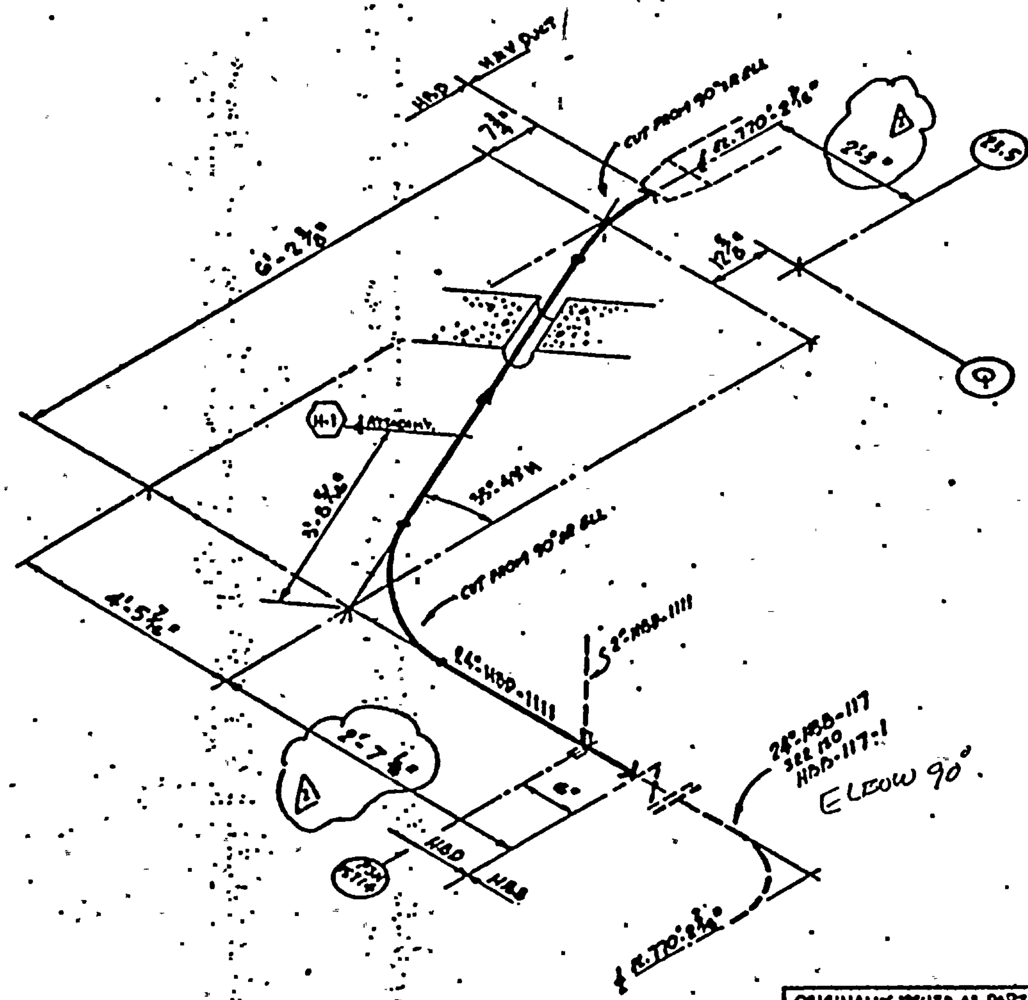
SUSQUEHANNA STEAM ELECTRIC STATION UNIT 1 UNIT 2

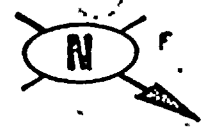
BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 2

	8856	HBD-282-2	3

57





REFERENCE DRAWINGS

M-157 SH.1 REV. 8 P. & I.D.
M-25-5 " PIPING PLAN - AREA 25

ORIGINALLY ISSUED AS PART OF
ISO HBD-117-1

2			DJ					
1		ISSUED FOR CONSTRUCTION	DJ					
0		ISSUED FOR FABRICATION	DJ					

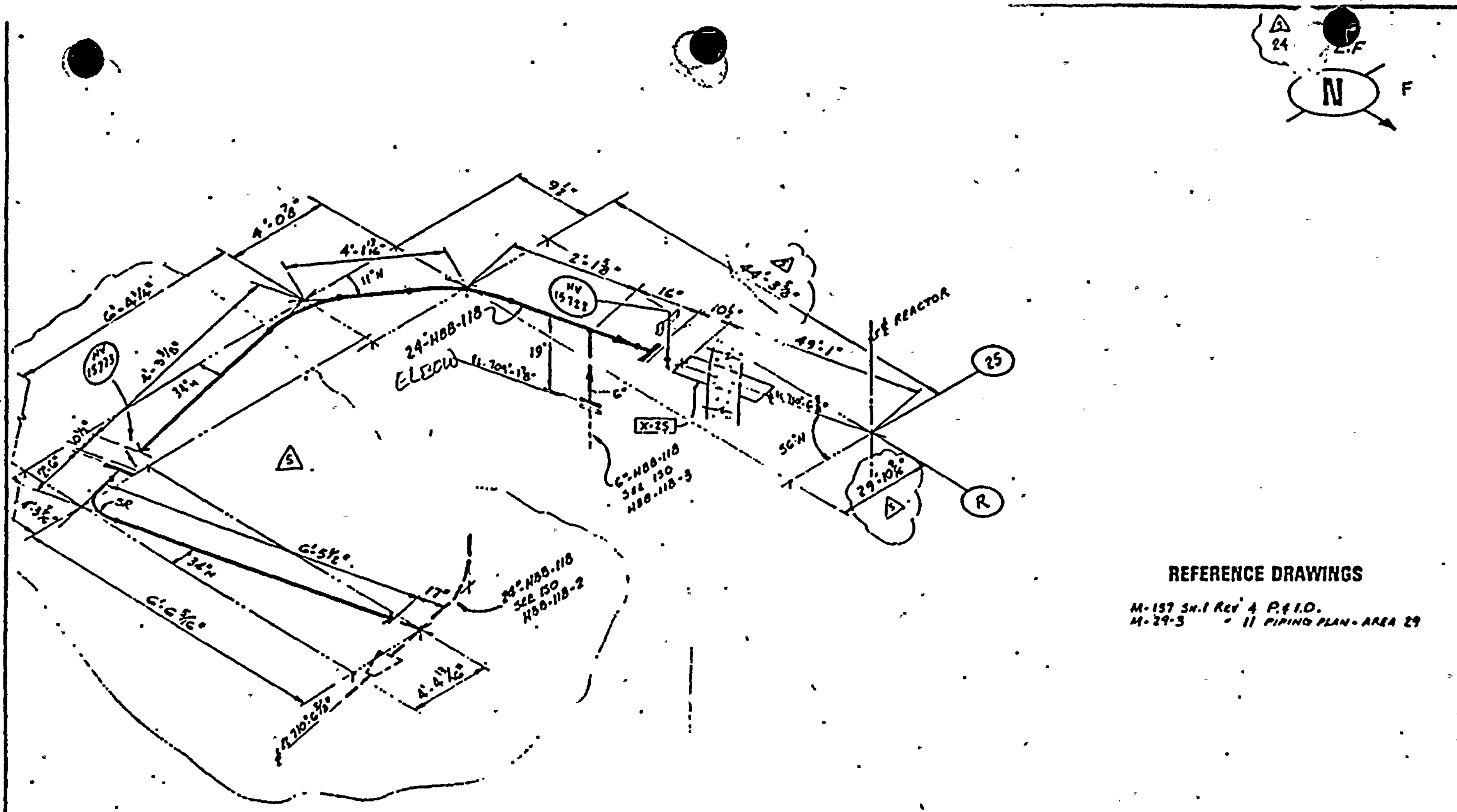
DRAWN DJ

PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATM. CONTROL - UNIT 1

JOB No.	51-5 DRAWING No.	REV.
8856	HBD-1111-1	2



REFERENCE DRAWINGS

M-157 SH.1 REV 4 P.&I.D.
M-29-3 " 11 PIPING PLAN - AREA 29

COMPONENTS ON THIS DWG. ARE Q-LISTED SPEC. 8856-G-9 APPLIES

NO	DATE	REVISIONS	BY	CHK				APPD
5	[illegible]	[illegible] ECR R-7147	RJ					
4	[illegible]	[illegible] PER P-1960	RJ					
3	[illegible]	DIM WAS 2'-6 5/16" WAS [illegible]	CG					
2	[illegible]	REVISED ISO REF.						
1	[illegible]	[illegible]	RJ	J.		RJ		
0	[illegible]	ISSUED FOR FABRICATION		RJ		RJ		

SCALE — | DESIGNED | DRAWN

PENNSYLVANIA POWER & LIGHT COMPANY
ALLENTOWN, PENNSYLVANIA
SUSQUEHANNA STEAM ELECTRIC STATION - UNIT 1, UNIT 2

BECHTEL - SAN FRANCISCO

ISOMETRIC - REACTOR BLDG.
CONTAINMENT ATMOS. CONTROL - UNIT 1

JOB No.	57-4 DRAWING No.	REV
8856	HBB-118-1	5

0111 150 REACT BD CTMT ATM CONT

ATTACHMENT 2
Nuclear Purge Valve

Stress Analysis

SEISMIC ANALYSIS
FOR 6 INCH
NUCLEAR PURGE VALVE

TABLE OF CONTENTS

LIST OF FIGURES

NOMENCLATURE

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

ANALYSIS NOMENCLATURE

A_f	Effective fluid pressure area based on fully corroded interior contour for calculating crotch primary membrane stress (NB-3545.1(a)), in^2
A_m	Metal area based on fully corroded interior contour effective in resisting fluid force on A_f (NB-3545.1(a)), in^2
A_3	Tensile area of cover cap bolt, in^2
A_4	Shear area of cover cap 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
B_1	Unsupported shaft length, in.
B_2	Bearing bore diameter, in.
B_3	Bonnet bolt tensile area, in^2
B_4	Bonnet bolt shear area, in^2
B_5	Bonnet body cross-sectional area, in^2
B_6	Top bonnet weld size, in.
B_7	Bottom bonnet weld size, in.
B_8	Distance to outer fiber of bonnet from shaft on y axis, in.
B_9	Distance to outer fiber of bonnet from shaft on x axis, in.
C	A factor depending upon the method of attachment of head, shell 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))

ANALYSIS NOMENCLATURE

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.
C_9	Distance to outer fiber of flat plate of disc for bending of unsupported flat plate, in.
d	Inside diameter of body neck at crotch region (NB-3545.1(a)), in.
d_m	Inside diameter used as basis for determining body minimum wall thickness, (NB-3541), in.
D_1	Valve nominal diameter, in.
D_2	Shaft diameter, in.
D_3	Disc pin diameter, in.
D_4	Thrust collar outside diameter, in.
D_5	Spring pin 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.
E	Modulus of elasticity, psi
F_b	Bending modulus of standard connecting pipe, as given by Figures NB-3545.2-4 and NB-3545.2-5, in^3
F_d	1/2 x cross-sectional area of standard connected pipe, as given by Figures NB-3545.2-2 and NB-3545.2-3, $in.^2$
F_N	Natural frequency of respective assembly, hertz

ANALYSIS NOMENCLATURE

F_x	W_3g_x--Seismic force along x axis due to seismic acceleration acting on operator extended mass, pounds
F_y	W_3g_y--Seismic force along y axis due to seismic acceleration acting on operator extended mass, pounds
F_z	W_3g_z--Seismic force along z axis due to seismic acceleration acting on operator extended mass, pounds
g	Gravitational acceleration constant, inch-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 centerline of the bolts to the line of the gasket reaction (NC-3225), in.
H_2	Top trunnion bolt square, in.
H_3	Bottom trunnion bolt square, in.
H_4	Bonnet bolt square, in.
H_5	Operator bolt square, in.
H_6	Bonnet bolt circle, in.
H_7	Operator bolt circle, in.
H_8	Bonnet height, in.
H_9	Actual body wall thickness, in.
I_1	Bonnet body moment of inertia about x axis, in^4
I_2	Bonnet body moment of inertia about y axis, in^4
I_3	Disc area moment of inertia for bending about the shaft in^4

ANALYSIS NOMENCLATURE

I_4	Disc area moment of inertia for bending along the shaft, in^4
I_5	Moment of inertia of valve body, in^4
I_6	Moment of inertia of shaft, in^4
I_7	Disc area moment of inertia for bending of unsupported flat plate, in^4
J_1	Distance to neutral bending axis for top trunnion bolt pattern along x axis, in.
J_2	Distance to neutral bending axis for top trunnion bolt pattern along y axis, in.
J_3	Distance to neutral bending axis for bonnet bolt pattern along x axis, in.
J_4	Distance to neutral bending axis for bonnet bolt pattern along y axis, in.
J_5	Distance to neutral bending axis for operator bolt pattern along x axis, in.
J_6	Distance to neutral bending axis for operator bolt pattern along y axis, in.
K	Spring constant
K_1	Distance of bonnet leg from shaft centerline, in.
K_2	Thickness of disc above shaft, in.
K_3	Length along z axis to cg 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 structural disc hub welds, 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.
m	Reciprocal of Poisson's ratio
M	Mass of component
Mx	$W_3(g_yZ_o+g_zY_o)$, operator extended mass seismic bending moment about the x axis, acting at the base of the operator, in-lbs.
My	$W_3(g_xZ_o+g_zX_o)$, operator extended mass seismic bending moment about the y axis, acting at the base of the operator, in-lbs.
Mz	$W_3(g_xY_o+g_yX_o)$, operator extended mass seismic bending moment about the z axis, in-lbs.
$\overline{Mx}$	$Mx+F_yT_5$, operator extended mass seismic bending moment about the x axis, acting at the bottom of the adapter plate, in-lbs.
$\overline{My}$	$My+F_xT_5$, operator extended mass seismic bending moment about the y axis, acting at the bottom of the adapter plate, in-lbs.
$\overline{\overline{Mx}}$	$Mx+F_y(T_5+H_8)+g_yW_4K_3$, operator extended mass seismic bending moment about the x axis, acting at the base of the bonnet, in-lbs.
$\overline{\overline{My}}$	$My+F_x(T_5+H_8)+g_xW_4K_3$, operator extended mass seismic bending moment about the y axis, acting at the base of the bonnet, in-lbs.
M_8	Bending moment at joint of flat plate to disc hub, in-lbs.
Na	Permissible number of complete start-up/shut-down cycles at hr/100°F/hr/hr fluid temperature change rate (NB-3545.3)
NA	Not applicable to the analysis of the system
N_1	Number of top disc pins

ANALYSIS NOMENCLATURE

N_2	Number of operator bolts
N_3	Number of trunnion bolts
Pd	Design pressure, psi
Pr	Primary pressure rating, pounds
Ps	Standard calculation pressure from Figure NB-3545.1-1, psi
Pe	Largest value among Peb, Ped, Pet, psi
Peb	Secondary stress in crotch region of valve body caused by bending of connected standard pipe, calculated according to NB-3545.2(b), psi
Ped	Secondary stress in crotch region of valve body caused by direct or axial load imposed by connected standard piping, calculated according to NB-3545.2(b), psi
Pet	Secondary stress in crotch region of valve body caused by twisting of connected standard pipe, calculated according to NB-3545.2(b), psi
P_m	General primary membrane stress intensity at crotch region, calculated according to NB-3545.1(a), psi
$P_{m'}$	Primary membrane stress intensity in body wall, psi
Q_p	Sum of primary plus secondary stresses at crotch resulting from internal pressure, (NB-3545.2(a)), psi
Q_T	Thermal stress in crotch region resulting from 100°F/hr fluid temperature change rate, psi
Q_{T1}	Maximum thermal stress component caused by through wall temperature gradient associated with 100°F/hr fluid temperature change rate (NB-3545.2(c)), psi
Q_{T2}	Maximum thermal secondary membrane stress resulting from 100°F/hr fluid temperature change rate, psi
Q_{T3}	Maximum thermal secondary membrane plus bending stress resulting from structural discontinuity and 100°F/hr fluid temperature change rate, psi
r	Mean radius of body wall at crotch region (NB-3545.2 (c)-1), in.
r_i	Inside radius of body at crotch region for calculating Q_p (NB-3545.2(a)), in.

ANALYSIS NOMENCLATURE

r_2	Fillet radius of ecternal 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 O-ring in cover cap, in.
S	Assumed maximum stress in connected pipe for calculating P_e (NB-3545.2(b)), 30,000 psi
S_m	Design stress intensity, (NB-3533), psi
S_n	Sum of primary plus secondary stress intensities at crotch region resulting from 100°F/hr temperature change rate (NB-3545.2), 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(71) are listed after the alphabetical section.

t_e	Minimum body wall thickness adjacent to crotch for calculating thermal stresses (NB-3545.2(c)-1), in.
t_m	Minimum body wall thickness as determined by NB-3541, in.
T_e	Maximum effective metal thickness in crotch region for calculating thermal stresses, (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 shaft behind spring pin, in.
T_3	Thrust collar thickness, in.
T_4	Cover cap thickness, in.
T_5	Adapter plate thickness, in.

ANALYSIS NOMENCLATURE

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.
U_1	Area of bottom bonnet weld, in^2
U_2	Area of top bonnet weld, in^2
U_3	Shaft bearing coefficient of friction
U_4	Bearing friction torque due to pressure loading (shaft journal bearings)
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, in.

Z_1	Bending section modulus of bonnet welds along x-axis, $in.^3$
Z_2	Bending section modulus of bonnet welds along y-axis, $in.^3$
Z_3	Torsional section modulus of bottom bonnet welds, $in.^3$
Z_4	Torsional section modulus of top bonnet welds, $in.^3$
Z_7	Distance to edge of disc hub, inches
Δy	Maximum static deflection of component, inches
U_3	Shaft bearing coefficient of friction
U_4	Bearing friction torque due to pressure loading (shaft journal bearings)
U_5	Bearing friction torque due to pressure loading plus seismic loading (shaft journal bearings)
U_6	Thrust bearing friction torque

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)	Shear tear out of shaft through disc, psi
S(5)	Combined stress in shaft, psi
S(6)	Combined bending stress in shaft, psi
S(7)	Combined shear stress in shaft, psi
S(8)	Bending stress in shaft due to seismic and pressure loads along x axis, psi
S(9)	Bending stress in shaft due to seismic load along y axis, psi
S(10)	Torsional shear stress in shaft due to operating loads, psi
S(11)	Direct shear stress in shaft due to pressure and seismic loads, psi
S(12)	Torsional shear stress at reduced pin cross-section, psi
S(13)	Combined shear stress in pin, psi
S(14)	Direct shear stress in pin due to seismic load, psi
S(15)	Shear stress in pin due to torsional load, psi
S(16)	Bearing stress on pin, psi
S(17)	Compressive stress on shaft bearing due to seismic and pressure loads, psi
S(18)	Shear tear out of cover cap bolt through tapped hole in bottom trunnion.
S(19)	Shear tear out of cover cap bolt through cover cap, psi

ANALYSIS NOMENCLATURE

S(20) Combined stress in cover cap bolts, psi

S(21) Shear stress in cover cap bolts due to torsional loading, psi

S(22) Direct tensile stress in cover cap bolts due to seismic and pressure loads, psi

S(23) Combined stress in cover cap, psi

S(24) Radial stress in cover cap, psi

S(25) Tangential stress in cover cap, psi

S(26) Shear stress in cover cap, psi

S(27) Bearing stress on thrust collar, psi

S(28) Shear load on thrust collar spring pin, pounds

S(29) Bearing stress of spring pin on thrust collar, psi

S(30) Shear tear out of spring pin through thrust collar, psi

S(31) Shear tear out of spring pin through bottom of the shaft, psi

ANALYSIS NOMENCLATURE

S(32)	Shear tear out of trunnion bolt through tapped hole in trunnion, psi
S(33)	Bearing stress of trunnion bolt on tapped hole in trunnion, psi
S(34)	Bearing stress of trunnion bolt on through hole in bonnet plate, psi
S(35)	Shear tear out of trunnion bolt head through bonnet plate, psi
S(36)	Combined stress in trunnion bolt, psi
S(37)	Direct tensile stress in trunnion bolt, psi
S(38)	Tensile stress in trunnion bolt due to bending moment, psi
S(39)	Direct shear stress in trunnion bolt, psi
S(40)	Shear stress in trunnion bolt due to torsional load, psi
S(41)	Shear tear out of operator bolt head through hole in bonnet, psi
S(42)	Bearing stress of operator bolt on through hole in bonnet, psi
S(43)	Combined stress in operator bolts, psi
S(44)	Direct tensile stress in operator bolts, psi
S(45)	Tensile stress in operator bolts due to bending moment, psi
S(46)	Direct shear stress in operator bolts, psi

ANALYSIS NOMENCLATURE

S(47)	Shear stress in operator bolts due to torsional loads, psi
S(48)	Combined stress in bonnet body, psi
S(49)	Direct tensile stress in bonnet body, psi
S(50)	Tensile stress in bonnet body due to bending moment, psi
S(51)	Direct shear stress in bonnet body, psi
S(52)	Shear stress in bonnet body due to torsional load, psi
S(53)	Combined shear stress in bottom bonnet weld, psi
S(54)	Total tensile stress in bottom bonnet weld, psi
S(55)	Direct tensile stress in bottom bonnet weld, psi
S(56)	Tensile stress in bottom bonnet weld due to bending moment, psi
S(57)	Total shear stress in bottom bonnet weld, psi
S(58)	Direct shear stress in bottom bonnet weld, psi
S(59)	Shear stress in bottom bonnet weld due to torsional load, psi
S(60)	Combined shear stress in top bonnet weld, psi
S(61)	Total tensile stress in top bonnet weld, psi
S(62)	Direct tensile stress in top bonnet weld, psi
S(63)	Tensile stress in top bonnet weld due to bending moment, psi
S(64)	Total shear stress in top bonnet weld, psi
S(65)	Direct shear stress in top bonnet weld, psi
S(66)	Shear stress in top bonnet weld due to torsional load, psi

ANALYSIS NOMENCLATURE

S(67)	Combined stress in trunnion body, psi
S(68)	Direct tensile stress in trunnion body, psi
S(69)	Tensile stress in trunnion body due to bending moment, psi
S(70)	Direct shear stress in trunnion body, psi
S(71)	Shear stress in trunnion body due to torsional load, psi

SUMMARY TABLE INTRODUCTION

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

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

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

Component Name

Code Reference (when applicable)

Stress Level Name and Symbol

Analysis Reference Page

Material Specification

Actual Stress Level

Allowable Stress Level

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

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

Summary Table Introduction

Component Name

Natural Frequency Symbol

Analysis Reference Page

Component Material

Natural Frequency

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

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	P_m	35	ASME SA-516 GR.55	564	Sm 13700
		Primary membrane stress in body	P'_m	36	ASME SA-516 GR.55	941	Sm 13700
	NB-3545.2	Primary plus secondary stress due to internal pressure	Q_p	36	ASME SA-516 GR.55	2800	Sm 13700
	NB-3545.2	Pipe Reaction stresses Axial Load Bending Load Torsional Load	 Ped Peb Pet	 36 36 36	ASME SA-516 GR.55	 2655 4100 4100	1.5 Sm 20550
	NB-3545.2	Thermal Secondary stress	Q_t	38	ASME SA-516 Gr.55	1105	Sm 13700
	NB-3545.2	Primary plus secondary stress	S_n	38	ASME SA-516 Gr.55	7109	3Sm 41100
	NB-3545.3	Normal duty fatigue stress Na $\geq$ 2000	S_p	38	ASME SA-516 Gr.55	5469	Sm 13700
Disc	NB-3546.2	Combined bending stress in disc	S(1)	39	ASME SA-516 Gr.55	1591	1.5 Sm 20550
	NB-3546.2	Shear tear out of shaft thru disc	S(4)	41	ASME SA-516 Gr.55	916	.6Sm 8220

TABLE 1 STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVEL PSI
Shaft	NB-3546.3	Combined stress in shaft	S(5)	42	ASME SA-564 Type 630 Cond. H-1150	12745	Sm 33700
	NB-3546.3	Torsional shear stress at reduced pin cross-section	S(12)	43	ASME SA-564 Type 630 Cond. H-1150	5702	.6Sm 20220
Disc Pin	NB-3546.3	Combined shear stress in top disc pin	S(13)	44	ASME SA-320 Gr. B8M	3928	.6Sm 8160
	NB-3546.3	Bearing stress on top pin	S(16)	44	ASME SA-320 Gr. B8M	1787	Sm 13600
Shaft Bearing		Comprehensive stress on shaft bearing	S(17)	45	ASTM B-438 Gr. 1 Type II	2964	Sm 4000
Cover Cap	NB-3546.1	Shear tear out of cover cap bolt through tapped holes in bottom trunnion	S(18)	46	ASME SA-516 Gr.55	938	.6Sm 8220
	NB-3546.1	Shear tear out of cover cap bolt head thru cover cap	S(19)	46	ASME SA-516 Gr.70	201	.6Sm 10500
	NB-3546.1	Combined stress in cover cap bolts	S(20)	46	ASME SA-193 Gr.B7	5973	Sm 25000
		Combined stress in cover cap	S(23)	46	ASME SA-516 Gr.70	2698	Sm 17500

TABLE 1

STRESS LEVELS FOR VALVE COMPONENTS

COMPONENT	CODE REF. PARAGRAPH	SYMBOL & NAME		REF. PAGE	MATERIAL	STRESS LEVEL, PSI	ALLOWABLE STRESS LEVEL PSI
Thrust Bearing		Bearing stress on thrust collar	S(27)	49	SAE-660	174	Sm 8800
		Shear load on thrust collar spring pin	S(28)	49	AISI 420	291	Pm 1540#
		Bearing stress of spring pin on thrust collar	S(29)	49	SAE-660	1147	Sm 8800
		Shear tear out of spring pin thru bottom of shaft.	S(31)	49	ASME SA-564 Type 630 Cond. H-1150	832	.6Sm 20220

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		Shear tear out of trunnion bolt thru tapped hole in trunnion	S(32)	50	ASME SA-516 Gr.55	3333	.6Sm 8220
		Bearing stress of trunnion bolt on tapped hole	S(33)	50	ASME SA-516 Gr.55	3604	Sm 13700
		Bearing stress of trunnion bolt on thru hole in bonnet	S(34)	50	ASME SA-36	4505	Sm 12600
		Shear tear out of trunnion bolt head thru bonnet	S(35)	52	ASME SA-36	2244	.6Sm 7560
		Combined stress in trunnion bolt	S(36)	52	ASTM A-193 Gr.B7	23917	Sm 25000
		Shear tear out of operator bolt head thru hole in bonnet	S(41)	52	ASME SA-36	999	.6Sm 7560
		Bearing stress of operator bolt on thru hole in bonnet	S(42)	52	ASME SA-36	3537	Sm 12600
		Combined stress in operator bolt	S(43)	54	ASTM A-193 Gr.B7	12972	Sm 25000
		Combined stress in bonnet body	S(48)	54	ASME SA-36	2705	Sm 12600

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		Combined shear stress in bottom bonnet welds	S(53)	56		953	.6Sm 7200
		Combined shear stress in top bonnet welds	S(60)	57		537	.6Sm 7200
		Combined stress in trunnion body	S(67)	58	ASME SA-516 Gr.55	2786	Sm 13700

Table 2 NATURAL FREQUENCIES OF VALVE COMPONENTS

Component Name	Natural Frequency Symbol	Ref. Page	Material	Natural Frequency (Hertz)
Body	F_{N1}	59	ASME SA-516 Gr. 55	13825
Banjo	F_{N2}	60	ASME SA-564 Type 630 Cond. H-1150	8765
Cover Cap	F_{N3}	60	ASME SA-516 Gr. 70	4212
Bonnet	F_{N4}	61	ASME SA-36	383

DIMENSIONAL DATA

Job Number: D-34933(D-0026-3) Valve Size: 6"-1200

Operator Mounting: TEE BONNET Operator: 521C-SR60

A_f	9.74	C_3	.50	g	32.2
A_m	6.59	C_6	249	G_b	48.29
A_3	.078	C_7	1.40	G_d	30.51
A_4	.068	C_8	1.50	G_T	96.58
A_5	.142	C_9	.50	g_x	5
A_6	.126	d	6.065	g_y	5
A_7	.142	d_m	6.065	g_z	5
A_8	.126	D_1	6	H_2	2.5
B_1	.63	D_2	1.0	H_3	2.5
B_2	1.25	D_3	.435	H_4	N/A
B_3	N/A	D_4	2.6	H_5	3.188
B_4	N/A	D_5	.25	H_6	N/A
B_5	6.5	D_6	.375	H_7	N/A
B_6	.25	D_7	.50	H_8	7
B_7	.25	D_8	.50	H_9	1.317
B_8	3.5	D_9	N/A	I_1	60.79
B_9	2.5	E	30E6	I_2	10.45
C	.3	F_b	6.6	I_3	4.94
C_b	1	F_d	2.7	I_4	5.17
C_p	3	F_x	560	I_5	209.92
C_o	1.02	F_y	560	I_6	.049
C_2	.42	F_z	560	I_7	N/A

J_1	.75	M_z	5561	ΔT_2	1
J_2	.50	$\overline{M_x}$	10532	T_1	1.188
J_3	N/A	$\overline{M_y}$	7066	T_2	.172
J_4	N/A	$\overline{\overline{M_x}}$	10532	T_3	.75
J_5	.50	$\overline{\overline{M_y}}$	7066	T_4	.374
J_6	.50	M_8	N/A	T_5	N/A
K_o	.86	N_a	2000	T_6	.50
K_1	N/A	N_1	N/A	T_7	.50
K_2	.50	N_2	4	T_8	628
K_3	3.5	N_3	4	U_1	7.5
K_4	3.5	P_d	275	U_2	7.5
K_5	4.5	P_r	150	U_3	.25
K_6	1.0	P_s	285	V_1	N/A
L_1	5	Q_{T1}	1000	V_2	N/A
L_2	N/A	r	1.203	V_3	N/A
L_3	.468	r_i	3.03	V_4	N/A
L_4	.625	r_2	1.0	V_5	N/A
L_5	1.0	R_4	2.6	V_6	N/A
L_6	N/A	R_5	.5	V_7	N/A
L_7	N/A	R_m	3.69	V_8	N/A
L_8	.50	R_6	1.4	W_1	124
L_9	5.2	S	30,000	W_2	12.5
m	3.5	t_e	1.092	W_3	112
M_x	6087	t_m	.28	W_4	30
M_y	2621	T_e	1.317	W_6	N/A

W_7	9
W_8	N/A
X_o	1.87
Y_o	8.06
Z_o	2.81
Z_1	21.46
Z_2	6.14
Z_3	23.76
Z_4	23.76
Z_7	N/A

Standard Stress Report

for

NRS Butterfly Valve

with

Bonnet Mounted

Cylinder Operator

ANALYSIS INTRODUCTION

Described in the following pages is the analysis used in verifying the structural adequacy of the main elements of the NRS 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.

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 co-ordinate 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.

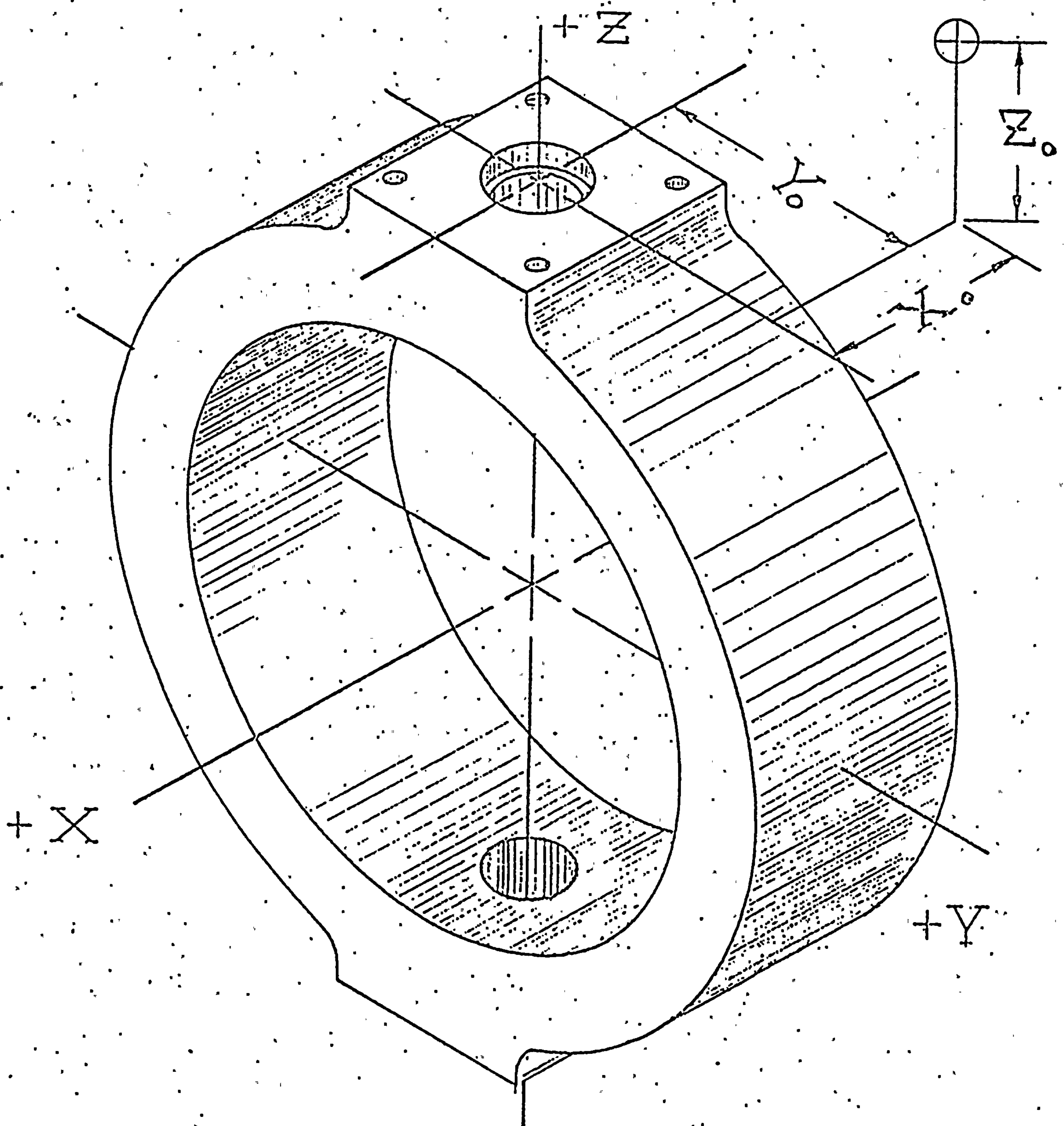


Figure 1 VALVE BODY SPATIAL ORIENTATION

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.

Disc Analysis

Shear Tear Out of Shaft

The disc is designed so the minimum thickness of material surrounding the shaft extension in the disc is above the shaft on the arch side. The loading is due to both seismic and pressure loads.

$$S(4) = \frac{\pi P_s R_4^2 + W_2 \sqrt{g_x^2+g_y^2+g_z^2}}{2L_9(K_2+\frac{D_2}{2}(1-\text{SIN } 45^o))} = \text{Shear tear out shaft through disc, psi.}$$

SHAFT ANALYSIS

The shaft is analyzed in accordance with Paragraph 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 through shaft.

Shaft stresses due to pressure, seismic and operating loads:

$$S(5) = \frac{S(6)}{2} + \frac{(S(6)^2+4\ S(7)^2)^{1/2}}{2}$$

where

$$S(6) = (S(8)^2+S(9)^2)^{1/2}$$ = Combined bending stress, psi

$$S(8) = \frac{(\pi R_4{}^2 P_S+W_2 g_x)\,.25\ B_1 R_5}{\pi\ .25\ R_5{}^4}$$ = Bending tensile stress due to pressure and seismic loads along x axis, psi.

$$S(9) = \frac{.25W_2 g_y\ B_1 R_5}{.25\ \pi\ R_5{}^4}$$ = Bending tensile stress due to seismic loads along y axis, psi.

$$S(7) = (S(10)^2+S(11)^2)^{1/2}$$ = Combined shear stress, psi

$$S(10) = \frac{T_8 R_5}{.5\pi R_5{}^4}$$ = Torsional shear stress, psi

$$S(11) = 1.333\left[\frac{.5\pi R_4{}^2 P_S+.5W_2(g_x{}^2+g_y{}^2)^{1/2}}{\pi R_5{}^2}\right]$$ = Direct shear stress, psi

Also worthy of attention is the torsional shear stress at the reduced cross-section where the pin passes through the shaft.

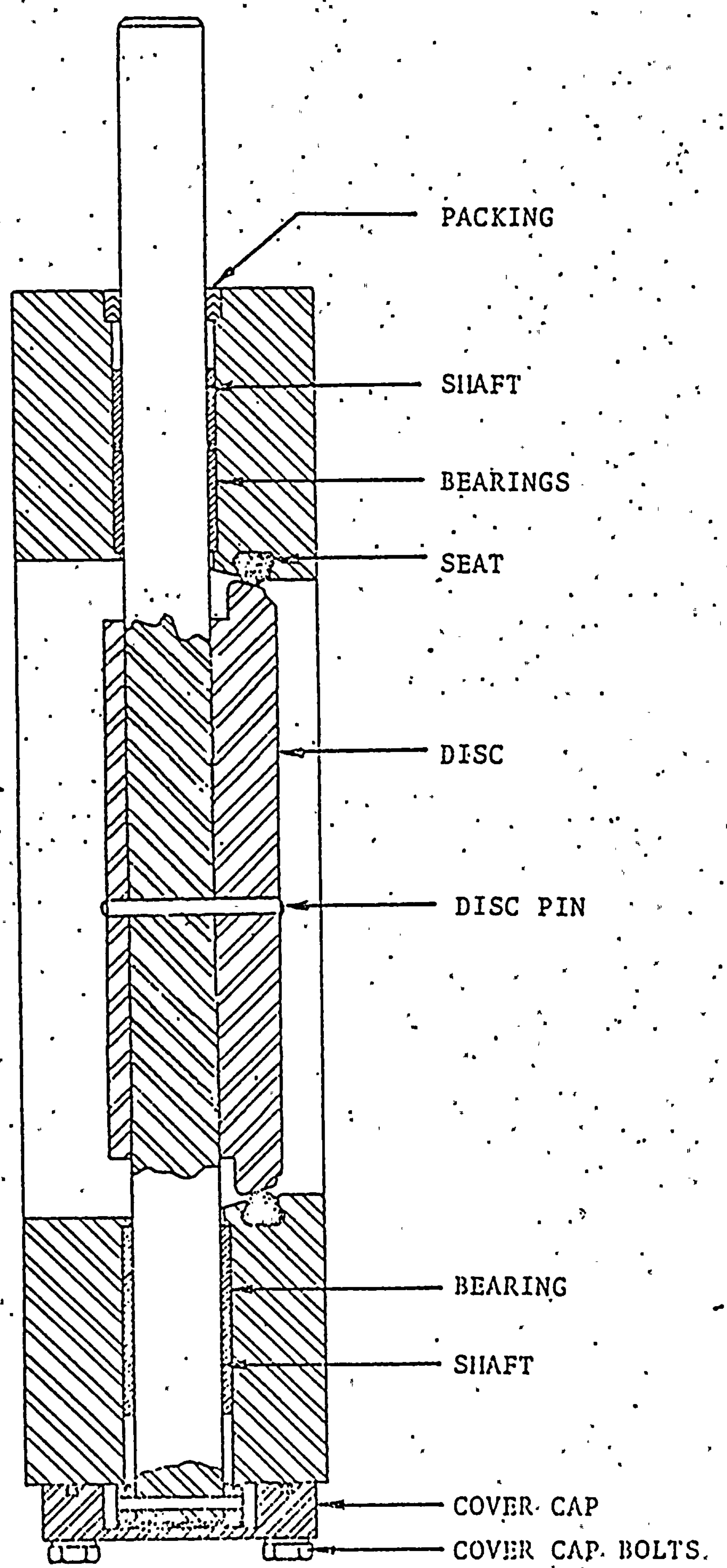


FIGURE 2 VALVE CROSS-SECTION

END CONNECTION ANALYSIS

The NRS butterfly valve is a uniflange design. Rather than having flanges that are external to and distinct from the body, the body shell is fabricated so that the end connections are machined directly into the body shell. The outside and inside diameter of the body shell conform to the requirements of the American National Standards Institute (ANSI) standard B16.5. The end connections, either flanged or weld end, also conform to this standard.

BODY ANALYSIS

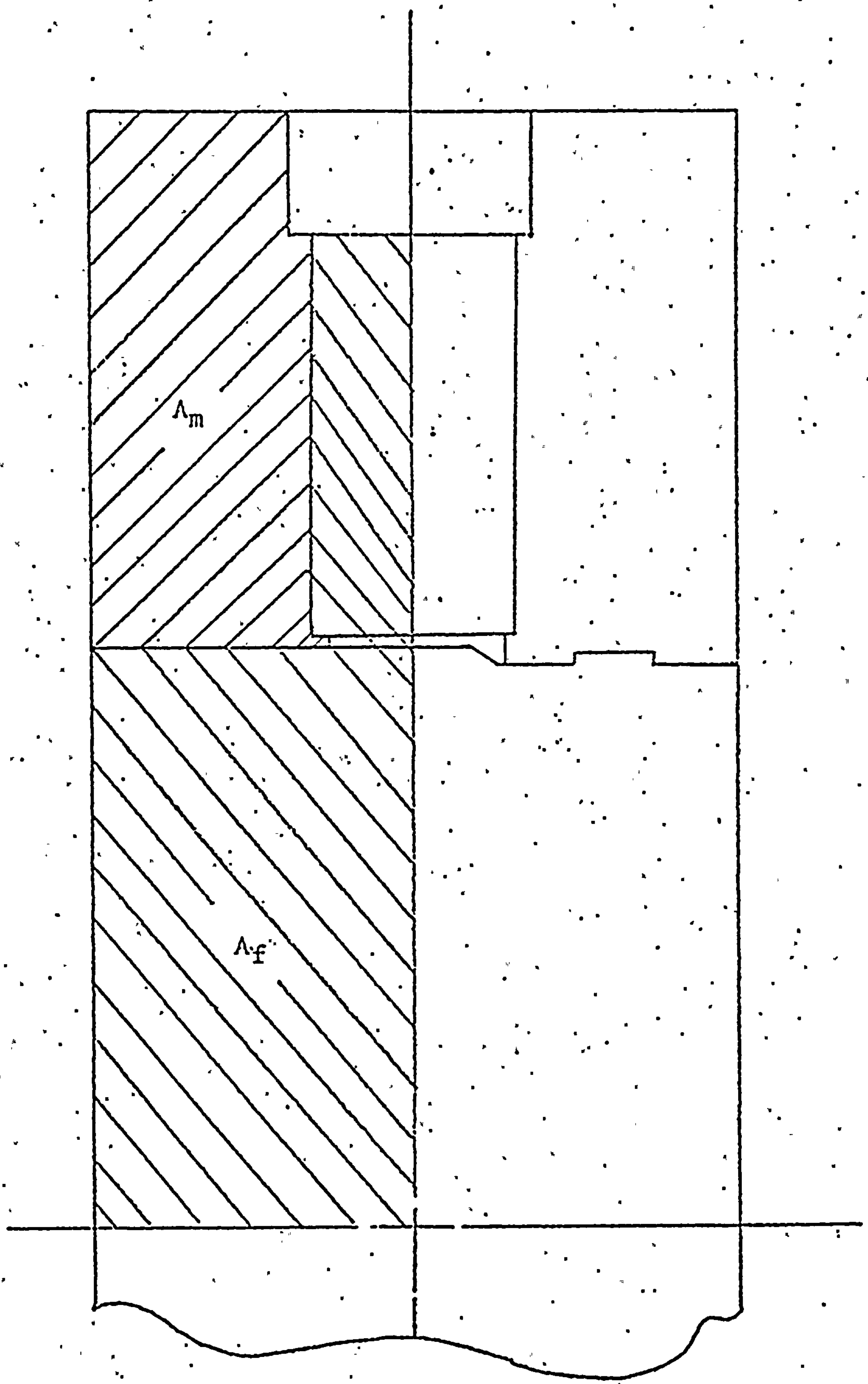
The body analysis consists of calculations as detailed in Paragraph NB-3540 of Section III of the Code. Paragraph NB-3540 is not highly oriented to butterfly valves as related to various design and shape rules. Therefore, certain of the design equations cannot be directly applied for butterfly valves. Where interpretation unique to the calculation is necessary, it is explained in the subsection 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. <u>Minimum Body Wall Thickness</u>

Paragraph NB-3542 gives minimum body wall thickness requirements for standard pressure rated valves.

The actual minimum wall thickness in the NRS valve occurs between the flange bolt holes and body bore.



PRESSURE-AREA ANALYSIS

BODY CROSS-SECTION

Figure 3

Body Analysis

2. Body Shape Rules

The NRS 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 this area using the pressure area method. As seen in Figure 3, certain code-defined dimensions are not applicable to this style of butterfly valve. For example, there is no radius at the crotch when seen in a view along the flow pattern, as the neck extends to the face of the body. To comply with the intent of the code, the areas A_f and A_m are interpreted as shown in the cross-section (Figure 3). Using these areas, the primary membrane stress is then calculated.

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

Body Analysis

As an alternate method of determining the primary membrane stress, an equivalent analysis for primary membrane stress is applied to an area away from the trunnions. In these areas, the metal area and fluid area are as shown in Figure 4. Since the depth of the metal area is equal to the depth of the fluid area, the ratio A_f/A_m is equivalent to the mean radius of the body over the thickness of the body shell, R_m/H_9. The primary membrane stress through this section is then:

$$P_{m'} = (R_m/H_9 + .5)\ p_s$$

4. Secondary Stresses

A. Body Primary plus secondary stress due to internal pressure: Paragraph NB-3545.2(a) of Section III of the code defines the formulas used in calculating this stress.

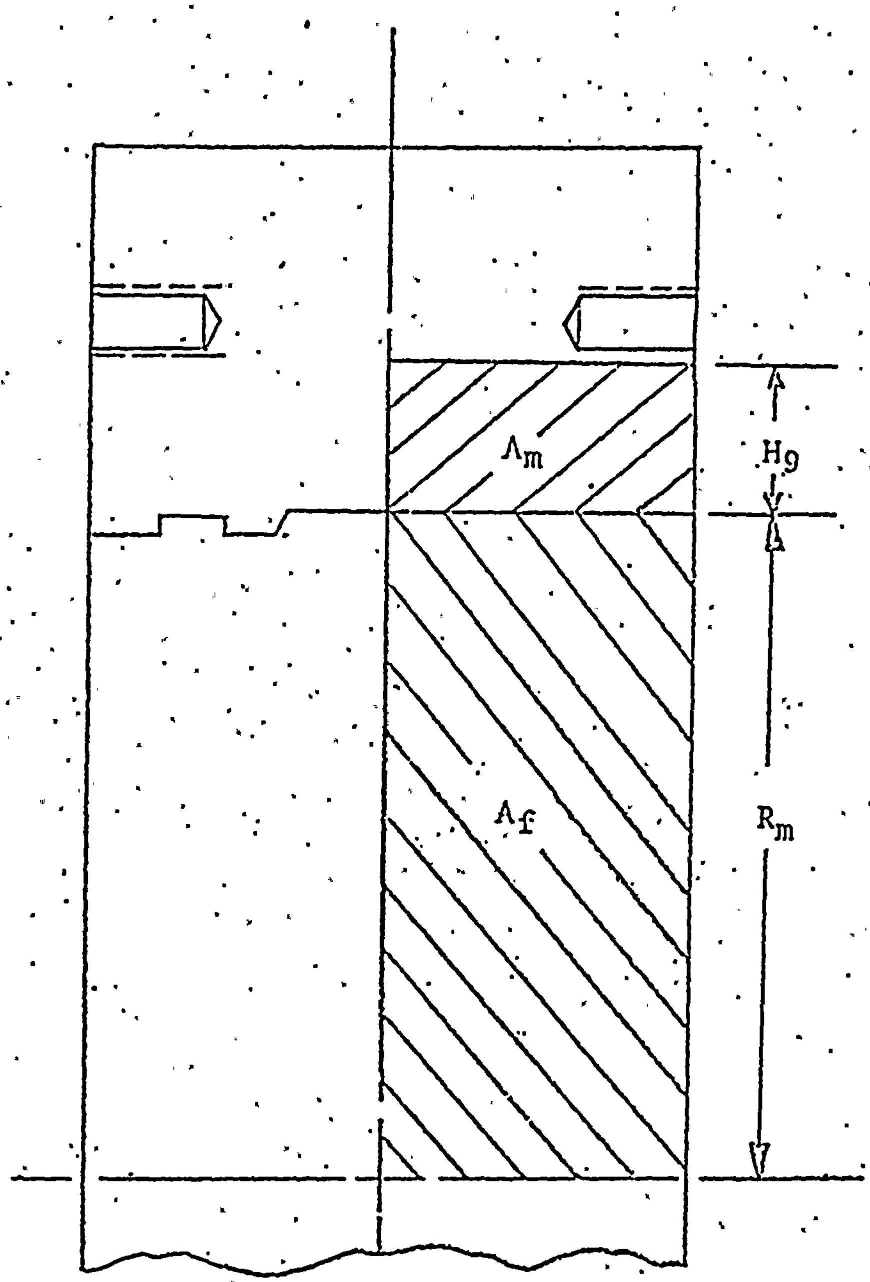
$$Q_p = C_p \left[\frac{r_i}{t_e} + .5 \right] p_s$$

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

$$P_{ed} = \frac{F_d S}{G_d} \quad \text{(Direct or Axial Load Effect)}$$

$$P_{eb} = \frac{C_b F_b S}{G_b} \quad \text{(Bending Load Effect)}$$

$$P_{et} = \frac{2 F_b S}{G_t} \quad \text{(Torsional Load Effect)}$$



PRESSURE AREA ANALYSIS

CROSS-SECTION IN BODY

Figure 4

Body Analysis

C. Thermal secondary stress: Paragraph 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 Paragraph NB-3545.2 and is simply the sum of the three previous secondary stresses.

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

5. Valve Fatigue Requirements

Paragraph NB-3543.3 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 2,000 is assumed. The allowable stress can then be found from Figure I-9.1 for carbon steel. This then gives an allowable stress of 65,000 psi.

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

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

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 NRS butterfly valves. The disc is designed to provide a structurally sound pressure retaining component while providing the least interference to the flow.

Primary Bending Stress

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

Combined bending stress in disc:

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

Where:

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

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

SHAFT ANALYSIS

The shaft is analyzed in accordance with Paragraph NB-
6.3 of Section III of the Code. The shaft loading is a com-
ation of seismic, pressure and operating loads. Maximum
rsional loading is either a combination of seating and bearing
rque or bearing and dynamic torque. Columnar stress is not
nsidered in the shaft loading due to its' negligible effect
the stress levels. Figure 2 shows the banjo assembly with the
rough shaft.

Shaft stresses due to pressure, seismic and operating loads:

$$S(5) = \frac{S(6)}{2} + \frac{(S(6)^2 + 4\ S(7)^2)^{1/2}}{2}$$

where

$$S(6) = (S(8)^2 + S(9)^2)^{1/2}$$ = Combined bending stress, psi

$$S(8) = \frac{(\pi R_4{}^2 P_s + W_2 g_x) .25\ B_1 R_5}{\pi\ .25\ R_5{}^4}$$ = Bending tensile stress due to pressure and seismic loads along x axis, psi.

$$S(9) = \frac{.25 W_2 g_y\ B_1 R_5}{.25\ \pi\ R_5{}^4}$$ = Bending tensile stress due to seismic loads along y axis, psi

$$S(7) = (S(10)^2 + S(11)^2)^{1/2}$$ = Combined shear stress, psi

$$S(10) = \frac{T_8 R_5}{.5 \pi R_5{}^4}$$ = Torsional shear stress, psi

$$S(11) = 1.333 \left[\frac{.5 \pi R_4{}^2 P_s + .5 W_2 (g_x{}^2 + g_y{}^2)^{1/2}}{\pi R_5{}^2} \right]$$ = Direct shear stress, psi

lso worthy of attention is the torsional shear stress at the
educed cross-section where the pin passes through the shaft.

Shaft Analysis

$$S(12) = S(10)\left[\frac{\dfrac{\pi R_5{}^4}{2}}{\dfrac{\pi R_5{}^4}{2} - \dfrac{D_2D_3{}^3}{12} - \dfrac{D_3D_2{}^3}{12}}\right]$$

DISC PIN ANALYSIS

As seen in Figure 2, there is one through shaft and one disc pin. The pin is subject to seismic and torsional loads.

Combined shear stress in top disc pin:

$$S(13) = (S(14)^2+S(15)^2)^{\frac{1}{2}}$$

Direct stress on disc pin due to seismic loads:

$$S(14) = \frac{W_7 g_z}{2N_1(.785)D_3^2}$$

Torsional shear stress in disc pin:

$$S(15) = \frac{T_8-.5U_5}{2N_1R_5.785D_3^2}$$

Bearing stress on disc pin:

$$S(16) = \frac{T_8-.5U_5}{2R_5K_2D_3N_1}$$

Where:

$$U_4 = .785(2R_4)^2P_0U_3R_5$$

$$U_5 = U_4+W_2g_xU_3R_5$$

P_0 = Actual Shut-Off Pressure

SHAFT BEARING ANALYSIS

The sleeve bearings in the trunnion (Figure 2) are subjected to both seismic and pressure loads.

$$S(17) = \frac{\pi P_d R_4{}^2 + W_2 (g_x{}^2 + g_y{}^2)^{\frac{1}{2}}}{2 L_5 D_2}$$ = Compressive stress on shaft bearing, psi

COVER CAP ANALYSIS

Figure 6 shows the bottom trunnion assembly, including the cover cap and cover cap bolts.

1. Cover cap bolt stresses:

The cover cap 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:

$$S(18) = \frac{W_2 \sqrt{g_x{}^2+g_y{}^2+g_z{}^2} + \pi P_s R_6{}^2}{4\ L_3\ 2.83\ D_6}$$

Shear tear out of bolt heads through cover cap, psi:

$$S(19) = \frac{W_2 \sqrt{g_x{}^2+g_y{}^2+g_z{}^2} + \pi P_s R_6{}^2}{4\ T_1\ 5.2\ D_6}$$

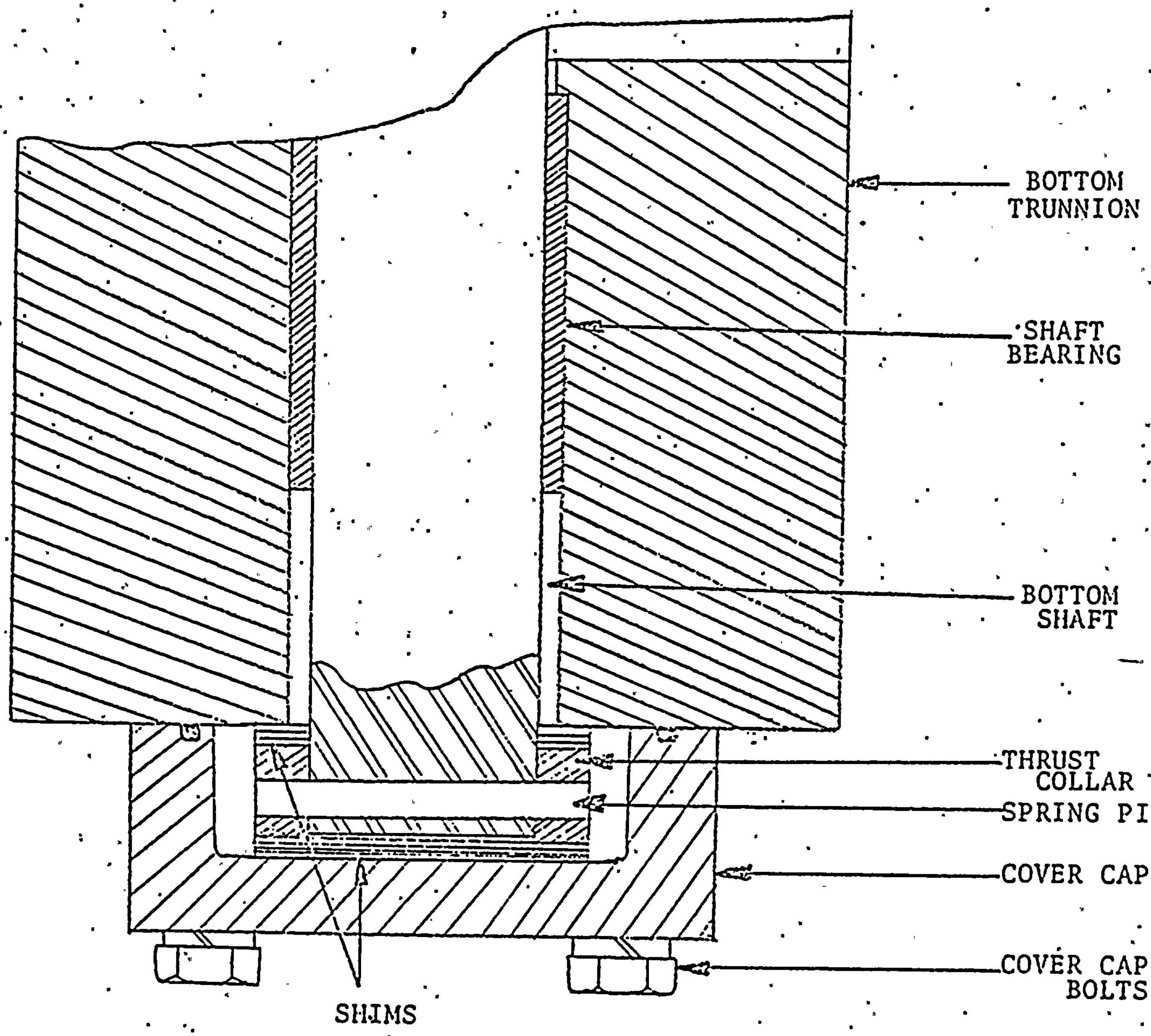
Combined stress in bolts, psi:

$$S(20) = \frac{S(22)}{2} + \frac{(S(22)^2 + 4S(21)^2)^{1/2}}{2}$$

Where:

$$S(21) = \frac{.25\ W_2 \sqrt{g_x{}^2+g_y{}^2+g_z{}^2}\ (D_2 + .66\ (D_4-D_2))}{.707\ H_3\ 4\ A_4}$$

= Shear Stress in Bolts Due to Torsional Load.



BOTTOM TRUNNION AND THRUST BEARING ASSEMBLY

Figure 6

Cover Cap Analysis

$$S(22) = \frac{W_2\sqrt{g_x^2+g_y^2+g_z^2} + \pi P_s R_6^2}{4 A_3}$$ = Tensile Stress in Bolts Due to Seismic And Pressure Loads, psi

2. Cover cap stresses:

The combined stress in the covercap is calculated using the following formulas:

$$S(23) = \frac{S(24) + S(25)}{2} + \frac{((S(24) + S(25))^2 + 4S(26)^2)^{\frac{1}{2}}}{2}$$

Where:

$$S(24) = \frac{3(.785\ (D_4 + .25)^2\ P_s + W_2 g_z)}{4\ \pi\ T_4^2}$$ = Radial Stress

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

$$S(26) = \frac{.785\ (D_4 + .25)^2\ P_s + W_2 g_z}{\pi\ (D_4 + .25)\ T_4}$$ = Shear Stress

THRUST BEARING ANALYSIS

As seen in figure 6, 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 and pressure loads:

$$S(27) = \frac{W_2 \sqrt{g_x^2+g_y^2+g_z^2} + \pi P_s R_5^2}{.785 (D_4^2-(D_2+.25)^2)}$$

2. Shear load on thrust collar spring pin due to seismic, pressure and torsional loads:

$$S(28) = \left[(W_2 g_z + \pi P_s R_5^2)^2 + \left(\frac{.25 W_2 g_z (D_2+.0833+.66 (D_4-D_2))}{R_5}\right)^2\right]^{1/2}$$

3. Bearing stress of spring pin on thrust collar:

$$S(29) = \frac{((W_2 g_z + \pi P_s R_5^2)^2 + (.25 W_2 g_z)^2)^{1/2}}{D_5 (D_4-D_2)}$$

4. Shear tear out of spring pin through bottom of shaft:

$$S(31) = \frac{W_2 g_z + \pi P_s R_5^2}{2D_2 T_2}$$

OPERATOR MOUNTING ANALYSIS

The operator mounting consists of the top trunnion, the bonnet, the operator housing, and the bolt connections. The elements of the assembly are shown in Figure 7.

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.

$$S(32) = \frac{\dfrac{F_z + W_4\sqrt{g_x{}^2+g_y{}^2+g_z{}^2}}{4} + \dfrac{\overline{\overline{M}}_x(J_2+H_2)}{2J_2{}^2+2(J_2+H_2)^2} + \dfrac{\overline{\overline{M}}_y\,(J_1+H_2)}{2J_1{}^2+2(J_1+H_2)^2}}{.9\pi\, L_4 D_7}$$

(b) Bearing stress on tapped holes in trunnion.

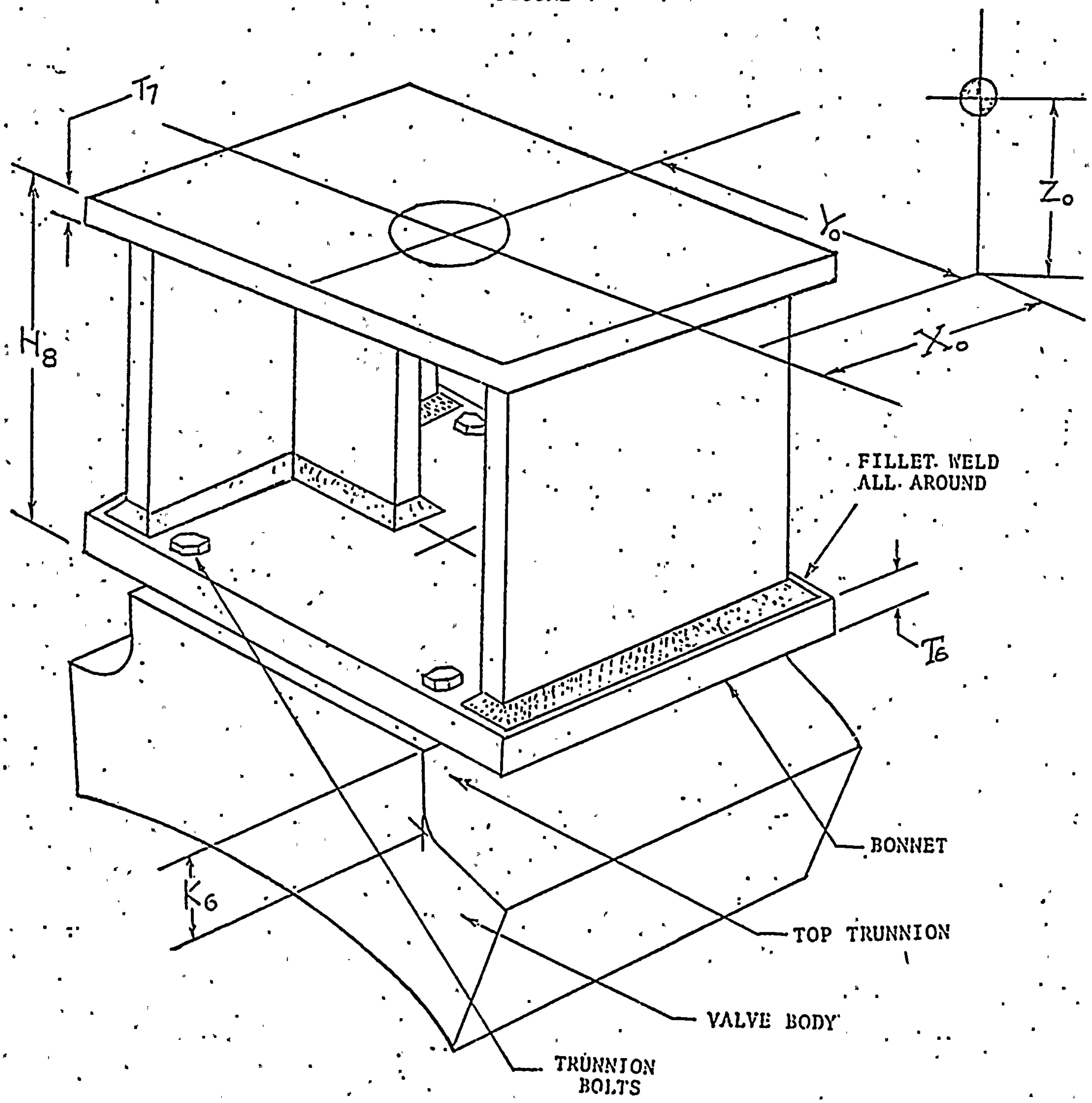
$$S(33) = \frac{\dfrac{M_z + T_8}{4(.707\ H_2)} + \dfrac{(F_x{}^2+F_y{}^2)^{1/2}}{4} + \dfrac{W_4\ (g_x{}^2+g_y{}^2)^{1/2}}{4}}{D_7 L_4}$$

(c) Bearing stress on through hole in bonnet.

$$S(34) = \frac{\dfrac{M_z + T_8}{4(.707\ H_2)} + \dfrac{(F_x{}^2+F_y{}^2)^{1/2}}{4} + \dfrac{W_4\ (g_x{}^2+g_y{}^2)^{1/2}}{4}}{D_7 T_6}$$

TOP TRUNNION MOUNTING

FIGURE 7



Operator Mounting Analysis

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

$$S(35) = \frac{\dfrac{F_z+W_4 g_z}{4} + \dfrac{\overline{\overline{M_x}}(J_2+H_2)}{2J_2^2+2(J_2+H_2)^2} + \dfrac{\overline{\overline{M_y}}(J_1+H_2)}{2J_1^2+2(J_1+H_2)^2}}{5.2\ D_7 T_6}$$

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

$$S(36) = \frac{S(37)+S(38)}{2} + \frac{((S(37)+S(38))^2+4(S(39)+S(40))^2)^{\frac{1}{2}}}{2}$$

Where

$$S(37) = \frac{F_z+W_4 g_z}{4\ A_5} = \text{Direct Tensile Stress, psi}$$

$$S(38) = \frac{\dfrac{\overline{\overline{M_x}}(J_2+H_2)}{2J_2^2+2(J_2+H_2)^2} + \dfrac{\overline{\overline{M_y}}(J_1+H_2)}{2J_1^2+2(J_1+H_2)^2}}{A_5} = \text{Tensile stress due to extended mass bending moment, psi}$$

$$S(39) = \frac{(F_x^2+F_y^2)^{\frac{1}{2}}+W_4(g_x^2+g_y^2)^{\frac{1}{2}}}{4\ A_6} = \text{Direct shear stress, psi}$$

$$S(40) = \frac{(M_z+T_8)}{(.707\ H_2)4\ A_6} = \text{Shear stress due to torsional load, psi}$$

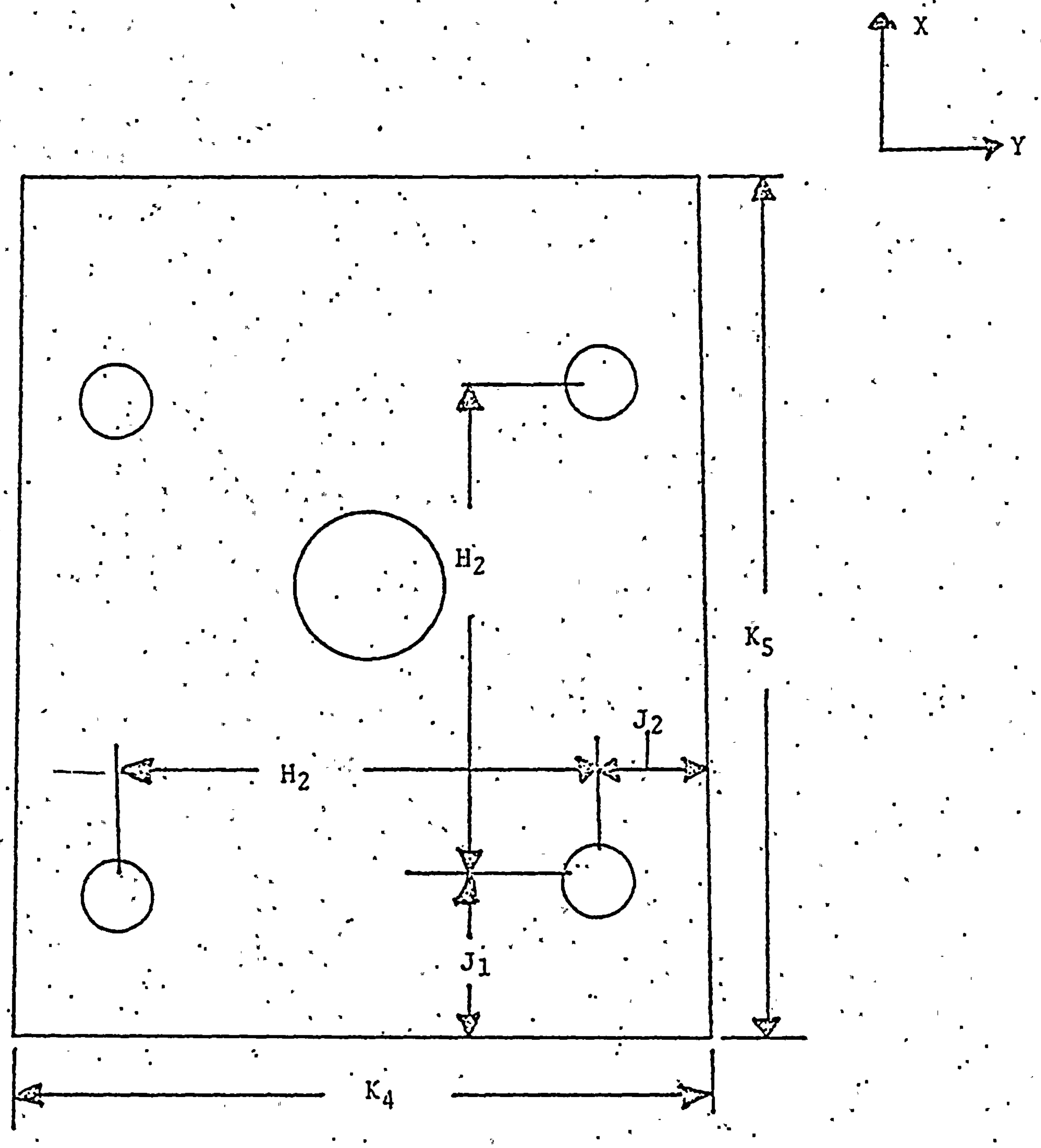
f. Shear tear out of operator bolt head through hole in bonnet.

$$S(41) = \frac{\dfrac{F_z}{N_2} + \dfrac{M_x(J_4+H_4)}{2J_4^2+2(J_4+H_4)^2} + \dfrac{M_y(J_3+H_4)}{2J_3^2+2(J_3+H_4)^2}}{5.2\ D_8 T_7}$$

g. Bearing stress on tapped holes in bonnet.

$$S(42) = \frac{\dfrac{M_z+T_8}{(.707\ H_4)N_2} + \dfrac{(F_x^2+F_y^2)^{\frac{1}{2}}}{N_2}}{D_8 T_7}$$

52



TOP TRUNNION BOLTING

Figure 8

Operator Mounting Analysis

h. Combined stress in operator bolts (See Fig. 9)

$$S(43) = \frac{S(44)+S(45)}{2} + \frac{((S(44)+S(45))^2+4(S(46)+S(47))^2)^{1/2}}{2}$$

Where

$$S(44) = \frac{F_z}{N_2A_7} = \text{Direct tensile stress, psi}$$

$$S(45) = \frac{\dfrac{M_x(J_4+H_4)}{2J_4{}^2+2(J_4+H_4)^2} + \dfrac{M_y(J_3+H_4)}{2J_3{}^2+2(J_3+H_4)^2}}{A_7} = \text{Tensile stress due to bending, psi}$$

$$S(46) = \frac{(F_x{}^2+F_y{}^2)^{1/2}}{N_2A_8} = \text{Direct shear stress}$$

$$S(47) = \frac{M_z+T_8}{(.707H_4)N_2A_8} = \text{Shear stress due to torsion, psi}$$

2. Bonnet Stresses

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

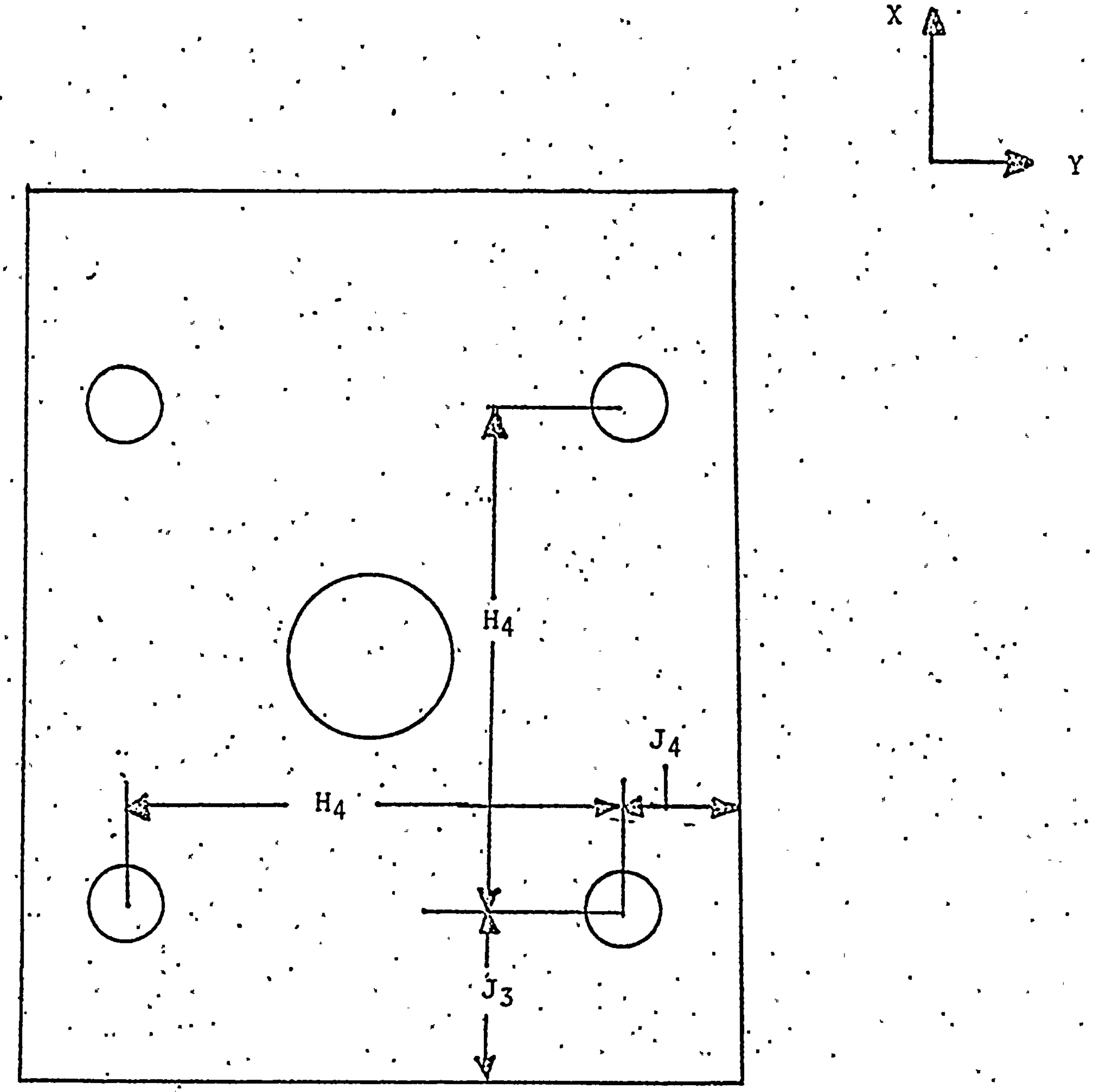
a. The maximum combined stress in the bonnet was calculated using the following formulas:

$$S(48) = \frac{S(49)+S(50)}{2} + \frac{((S(49)+S(50))^2+4(S(51)+S(52))^2)^{1/2}}{2}$$

= Combined stress in bonnet legs

$$S(49) + \frac{F_z+W_4g_z}{B_5} = \text{Direct tensile stress, psi}$$

571



BONNET BOLT PATTERN

Figure 9

Operator Mounting Analysis

$$S(50) = \frac{\overline{\overline{M}}_x B_8}{I_1} + \frac{\overline{\overline{M}}_y B_9}{I_2} = \text{Tensile stress due to bending moment, psi}$$

Where

$$S(51) = \frac{(F_x^{\,2}+F_y^{\,2})^{1/2} + W_4(g_x^{\,2}+g_y^{\,2})^{1/2}}{B_5} = \text{Direct shear stress, psi}$$

$$S(52) = \frac{T}{K_0} C_0 = \text{Shear stress in bonnet body due to torsional load, psi}$$

Where

T = Torque, in-lbs.
C_0 = Torsional constant for non-circular cross section
K_0 = Function of cross-section, $in.^4$

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

Bottom Bonnet Weld

$$S(53) = \frac{(S(54)^2 + 4S(55)^2)^{1/2}}{2} = \text{Combined shear stress in bottom weld, psi}$$

Where

$$S(54) = S(56) + S(57) = \text{Total tensile stress, psi}$$

$$S(56) = \frac{F_z + W_4 g_z}{U_1} = \text{Direct tensile stress, psi}$$

$$S(57) = \frac{\overline{\overline{M}}_x}{Z_1} + \frac{\overline{\overline{M}}_y}{Z_2} = \text{Bending tensile stress}$$

$$S(55) = S(58) + S(59) = \text{Total shear stress}$$

$$S(58) = \frac{(F_x^{\,2}+F_y^{\,2})^{1/2} + W_4(g_x^{\,2}+g_y^{\,2})^{1/2}}{U_1} = \text{Direct shear stress, psi}$$

Operator Mounting Analysis

$$S(59) = \frac{M_z + T_8}{Z_3} = \text{Torsional shear stress, psi}$$

Top Bonnet Weld

$$S(60) = \frac{(S(61)^2 + 4S(62)^2)^{1/2}}{2} = \text{Combined shear stress in top bonnet weld}$$

Where

$$S(61) = S(63) + S(64) = \text{Total tensile stress, psi}$$

$$S(63) = \frac{F_z}{U_2} = \text{Direct tensile stress, psi}$$

$$S(64) = \frac{M_x}{Z_1} + \frac{M_y}{Z_2} = \text{Bending tensile stress, psi}$$

$$S(62) = S(65) + S(66) = \text{Total shear stress, psi}$$

$$S(65) = \frac{(F_x^2 + F_y^2)^{1/2}}{U_2} = \text{Direct shear stress, psi}$$

$$S(66) = \frac{M_z + T_8}{Z_4} = \text{Torsional shear stress, psi}$$

c. 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.

Operator Mounting Analysis

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

$$S(67) = \frac{S(68)+S(69)}{2} + \frac{\left((S(68)+S(69))^2+4(S(70)+S(71))^2\right)^{1/2}}{2}$$

Where

$$S(68) = \frac{F_z+W_4g_z}{K_4K_5-.785B_2^2} = \text{Direct tensile stress, psi}$$

$$S(69) = \frac{(M_x+F_yK_6).5K_4}{.0833K_5K_4^3-\frac{\pi B_2^4}{64}} + \frac{(M_y+F_xK_6).5K_5}{.0833K_4K_5^3-\frac{\pi B_2^4}{64}} = \text{Bending tensile stress, psi}$$

$$S(70) = \frac{(F_x^2+F_y^2)^{1/2}+W_4(g_x^2+g_y^2)^{1/2}}{K_4K_5-.785B_2^2} = \text{Direct shear stress, psi}$$

$$S(71) = \frac{(M_z+T_8).5(K_4^2+K_5^2)^{1/2}}{.0833(K_4K_5^3+K_5K_4^3)-\frac{\pi B_2^4}{32}} = \text{Torsional shear stress, psi}$$

FREQUENCY ANALYSIS

A. Introduction

To calculate the natural frequency of the various components of the NRS 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

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

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:

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

Frequency Analysis

Where

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

C. Banjo Assembly

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

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

Where

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

D. Cover Cap Assembly

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

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

Where

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

E. Bonnet Assembly

Figure 7 shows the top trunnion assembly. The following 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.

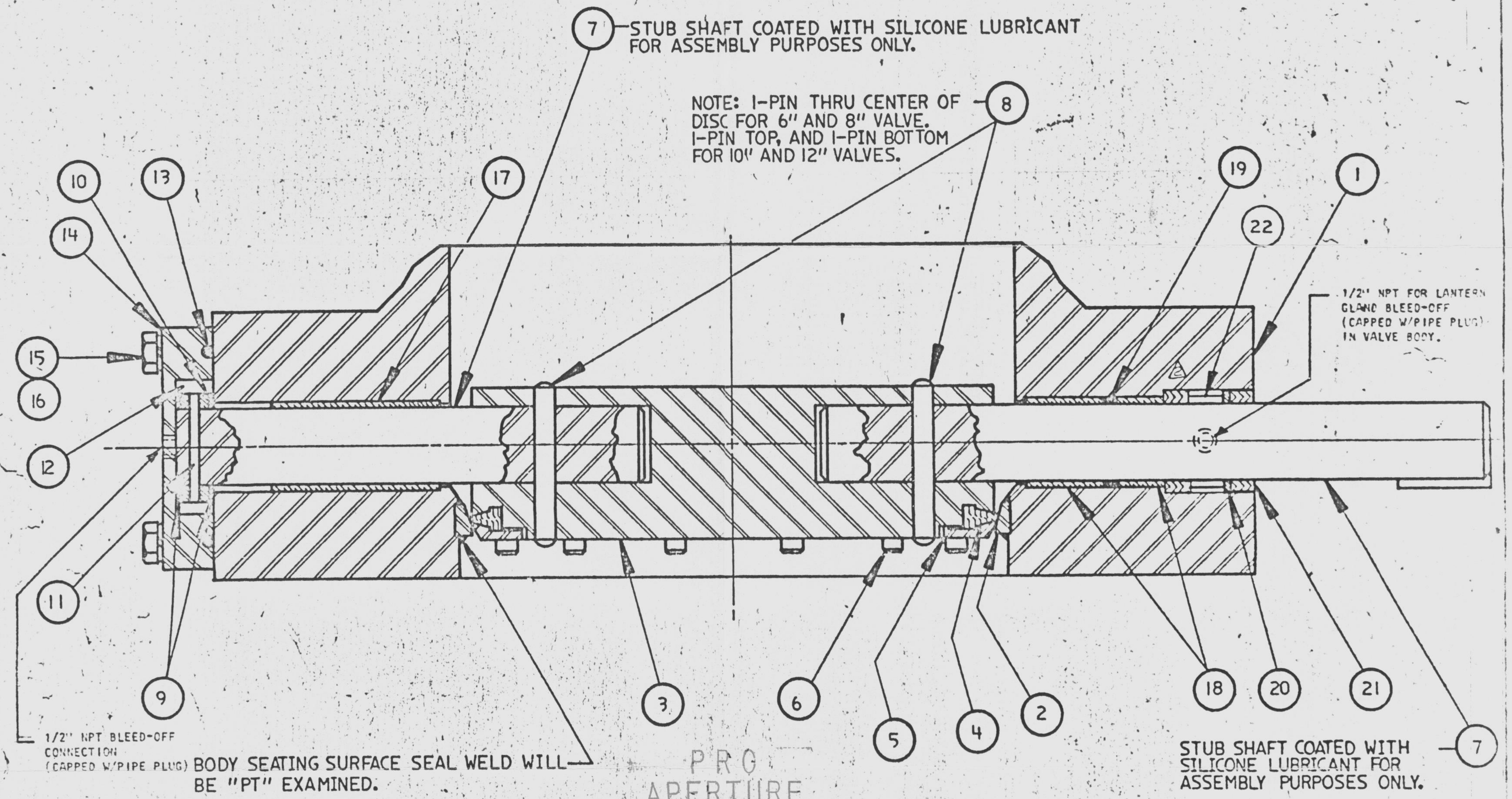
Natural frequency of bonnet:

$$F_{n4} = \left(\frac{9.8}{\Delta y_4}\right)^{1/2}$$

Where

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

	PARTS AND MATERIALS OF CONSTRUCTION		PARTS AND MATERIALS OF CONSTRUCTION
	1. BODY: MAT'L.; SA-516 GR. 55		12. GREASE: DOW CORNING 111
	2. SEATING SURFACE: MAT'L.; SA-479 TYPE 304		13. O-RING: MAT'L.; E.P.T.
	3. DISC: MAT'L.; SA-516 GR. 55	⚠	14. BOTTOM COVER: MAT'L.; SA-516 GR. 70
	4. SEAT: MAT'L.; E.P.T.		15. COVER BOLTS: MAT'L.; SA-193 GR. B-7
	5. CLAMP SEGMENT RING: MAT'L.; SA-285 GR. C		16. LOCKWASHER: MAT'L.; CARBON STEEL
	6. CLAMP SEGMENT SCREWS: MAT'L.; SA-193 GR. B-7		17. BOTTOM BEARING: MAT'L.; ASTM B-438 GR. 1 TYPE 2 BRZ.
	7. SHAFT: MAT'L.; SA-564 TYPE 630 COND. H1150		18. TOP BEARINGS: MAT'L.; ASTM B-438 GR. 1 TYPE 2 BRZ.
⚠	8. PINS: MAT'L.; SA-320 GR. B8M		19. SHAFT SEAL: MAT'L.; E.P.T.
	9. THRUST COLLAR SHIMS: MAT'L.; HARD BRASS	⚠	20. PACKING RETAINER RING: MAT'L.; SB-144 ALLOY 3B
	10. THRUST COLLAR: MAT'L.; SAE 660 BRONZE		21. PACKING: MAT'L.; E.P.T. V-RINGS
	11. THRUST COLLAR PIN: MAT'L.; AISI 420 STN. STL.		22 LANTERN GLAND RING: MAT'L; ASTM A-269



PRO APERTURE CARD

*MATERIAL AND NDE STANDARDS SHALL BE IN ACCORDANCE WITH ASME SECTION III CLASS 2 REQUIREMENTS.

CUSTOMER: BECHTEL POWER CORP.
CUSTOMER P. O. P8856-P-31-AC
PRATT ORDER NO.: D-0026-3
PROJECT: PENNSYLVANIA POWER & LIGHT CO., SUSQUEHANNA

ITEM NO.

1.17 UNIT-1 6"-HBB-BF-AO-5721

1.18 UNIT-2 6"-HBB-BF-AO-5721

REV.	DATE	BY	CH'D.	APP.	APP.	Q.A.
1	6-18-75	R.O.				
2	8-4-76	FPM		THW	A.I.	
3						
4						

HENRY PRATT COMPANY
AURORA, ILL.

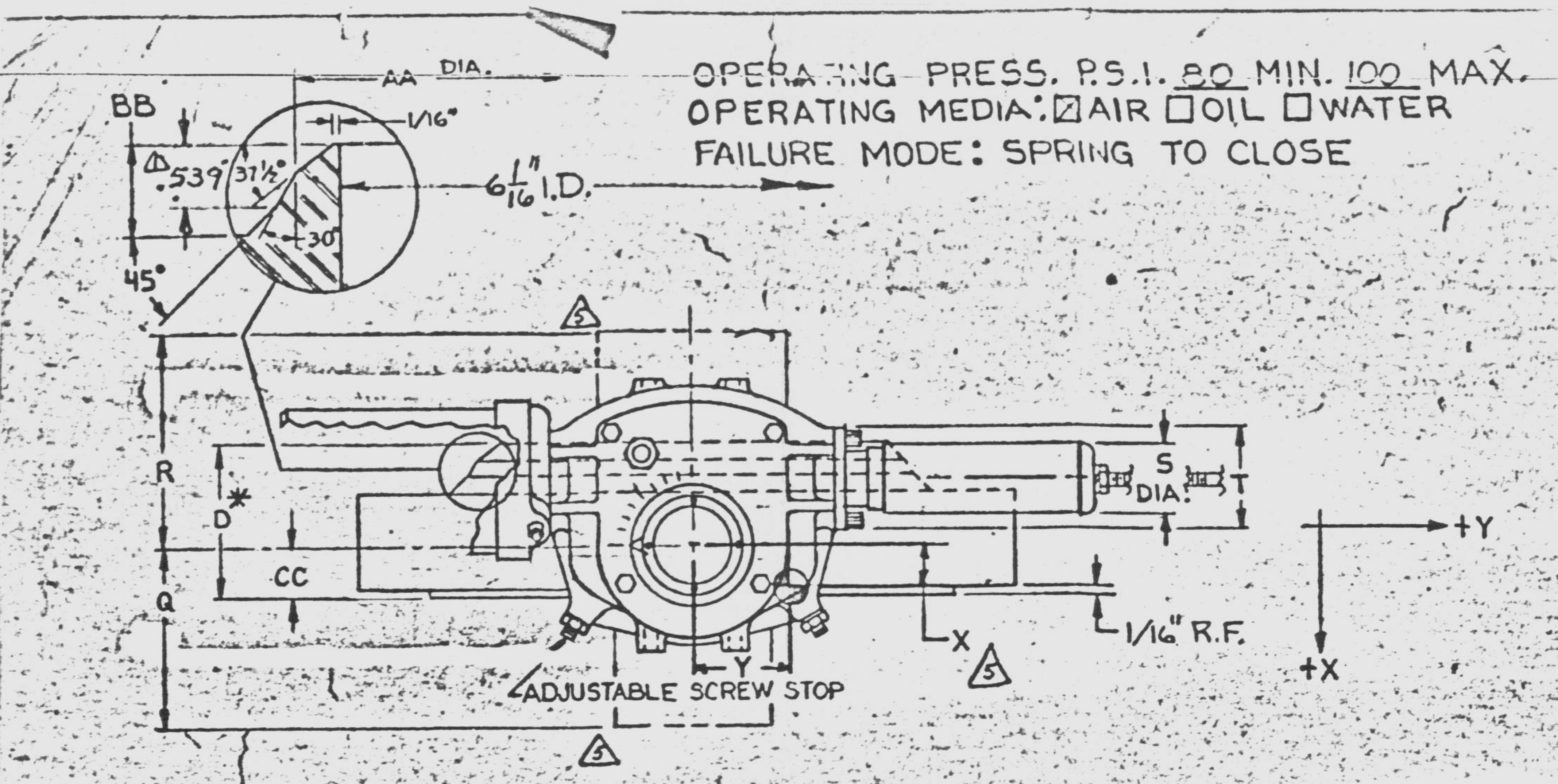
6" THRU 12" ASME SECTION III CLASS NRS VALVE CROSS SECT. AND MATERIALS LIST FLANGE X WELD END

SCALE NONE DATE 6-18-75
DRAWN BY R. O'C. CHECKED BY CYKO
APPROVED
Q.A. APPROVAL
DWG. NO. C-2988

8809160247-01

D

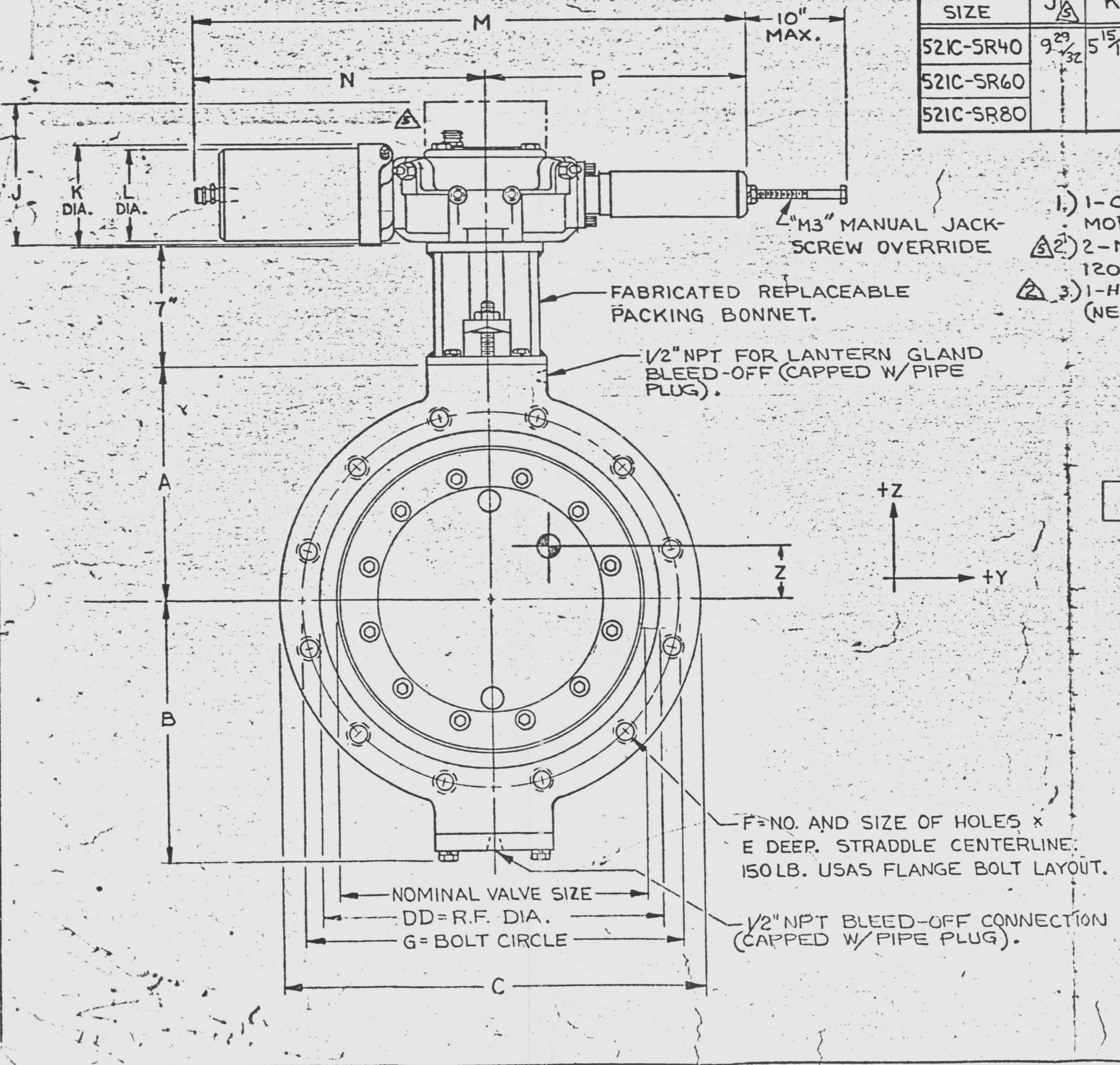
OPERATING PRESS. P.S.I. 80 MIN. 100 MAX.
OPERATING MEDIA: ☒AIR ☐OIL ☐WATER
FAILURE MODE: SPRING TO CLOSE



VALVE SIZE	A	B	C	D*	E	F	G	AA	BB	CC	DD
6	6½	8	11	6	1 3/16	8-¾-10	9½	6 25/32	1½	2½	8½
8											
10											
12											
14											
16											
18											
20											
24											

NOTE: ALL DIMENSIONS ARE SHOWN IN INCHES.
* D± 1/16" THRU 10" VALVES.
* D± 1/8" FOR 12" VALVES AND LARGER.
OVERALL DIMENSIONS ARE MAXIMUM
OTHERS ARE NOMINAL

OPERATOR SIZE	J	K	L	M	N	P	Q	R	S	T
521C-SR40	9 29/32	5 15/16	5 3/8	43½	28¾	14 13/16	10 7/8	10 1/32	2¼	3¾
521C-SR60										
521C-SR80										



ACCESSORIES
(MOUNTED ON OPERATOR)
1.) 1-CIRCLE SEAL SINGLE SOLENOID VALVE
MODEL S.V. 315 9101-1 120V.A.C. 60 HZ.
2.) 2-NAMCO EA-740-20000 LIMIT SWITCHES
120 V.A.C 10 AMP. D.P.D.T.
3.) 1-HOFFMAN JUNCTION BOX # A-606 CHNF
(NEMA IV).

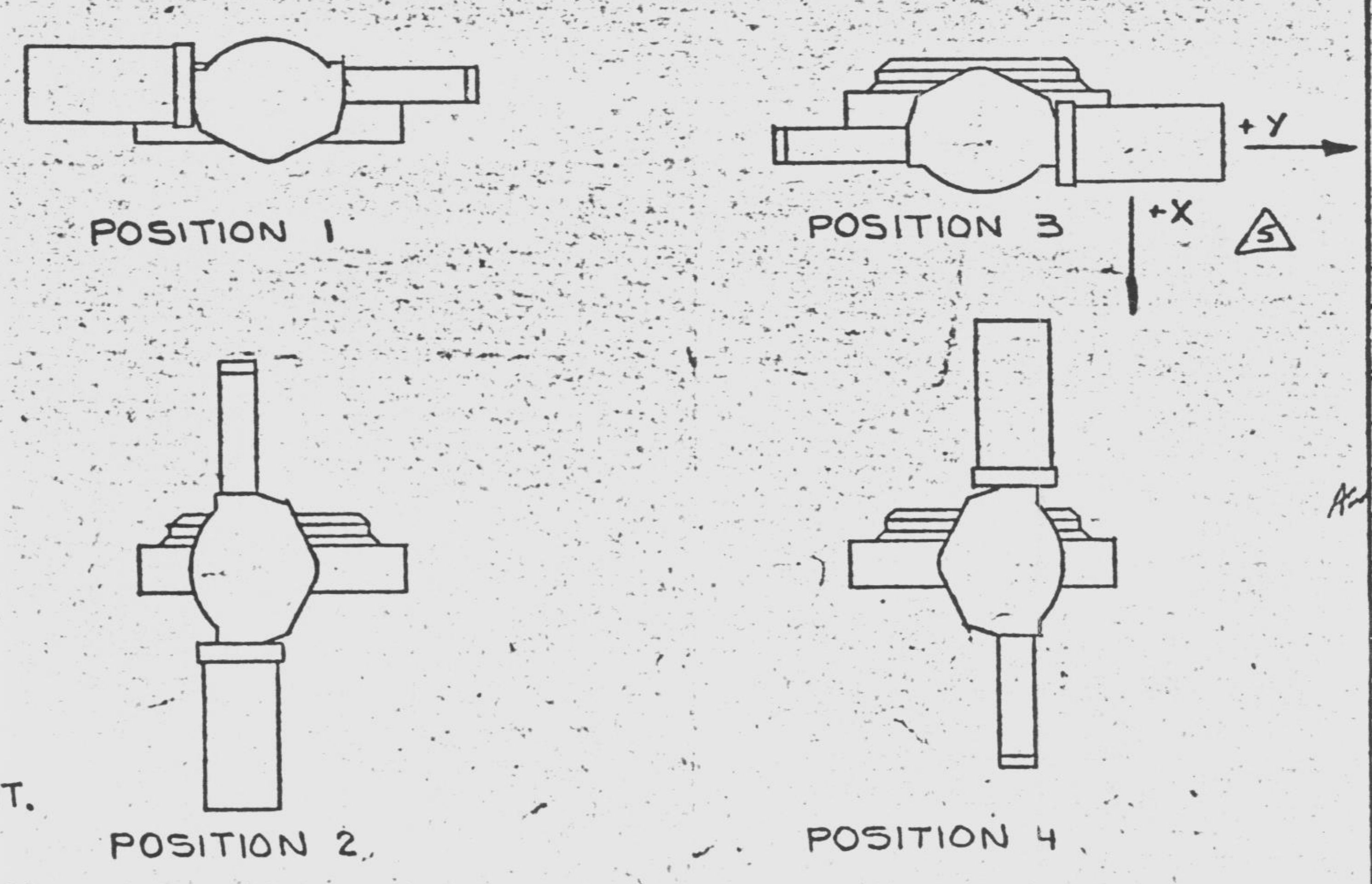
CUSTOMER: BECHTEL POWER CORP.
CUSTOMER P.O. P8856-P-31-AC
PRATT ORDER NO. D-0026-3
PROJECT: PENNSYLVANIA POWER & LIGHT CO., SUSQUEHANNA

PRATT ITEM	QUAN.	SIZE	OPER. POS.	OPERATOR DESCRIPTION	VALVE CROSS SECTION DRWG.	VALVE WT.	OPER. WT.	TAG	C.G. LOCATIONS X	Y	Z
3	2	6"	3	521C-SR60-M3	C-2988	165#	112#	*	.81	-3.58	7.85

NOTE: FOR BONNET CROSS-SECT. & PARTS + MAT'L. LIST SEE DWGS A-8721 & A-8722

* TAG:	ITEM NO
UNIT 1-6"HBB-BF-AO-5721	1.17
UNIT 2-6"HBB-BF-AO-5721	1.18

PAINT: EXTERIOR OF VALVE & OPERATOR
1 COAT OF CARBIT 4-E-2 PRIMER GOOD FOR 600°F
NOTE: FOR BONNET CROSS-SECT. & PARTS & MAT'L LIST SEE DWGS. A-8721 & A-8722.



FLANGE × WELD END

REV.	DATE	BY	APP	REV.	DATE	BY	APP
4	12-17-75			8			
3	10-8-75			7			
2	6-18-75			6			
1	11-21-77			5	4-8-76		

PRATT — HENRY PRATT COMPANY AURORA ILL.

GENERAL ARRANGEMENT
NUCLEAR N.R.S. VALVE W/BON.
& BETTIS SPRING-RETURN OPER.

SCALE NONE DATE 5-23-?
DRAWN BY G. DAVIS CHECKED BY
APPROVED
PART NO D-0026-3
DWG. NO C-2600

PRC APERTURE CARD

8206160247-02

D