Attachment 2 to
SL-2-81-749

ISOLATION/PURGE VALVE ANALYSIS FOR
8" MODEL 1200 BUTTERFLY VALVE

Project Site: St. Lucie Plant Unit 2

Customer: Florida Power & Light

Engineer: Ebasco Services, Inc.

Original Specification: FLO 2998.114 Rev. 3

Original Purchase Order: NY 422537

Original Pratt Job No.: D-0096-10

Valve Tag Nos.: I-FCV-25-20 & 21
I-FCV-25-26

General Arrangement Drawing: C-5531 Rev. 2

Cross Section Drawing: C-5530 Rev. 1

Prepared by: B. Tymcinrak
Date: Aug 5-1981

Reviewed by: T. J. Wrona
Date: 8-6-81

Certified by: Bomi M. Zarolia
Date: Aug. 6, 1981

BOMI M. ZAROLIA
30701
REGISTERED PROFESSIONAL ENGINEER OF
ILLINOIS

8109095316

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 furnished by customer/engineer or assumed to be worst case.

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

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

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

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

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

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

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

C. Sealing integrity can be evaluated as follows:

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

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

III. Method of Analysis

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

A. Torque calculation

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

Bearing friction torque is calculated from the following equation:

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

where

P =pressure differential, psia

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

U = bearing coefficient of friction

d = shaft diameter, in.

Fluid dynamic torque is calculated from the following equations:

For subsonic flow

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

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

For sonic flow

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

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

Where

T_D = fluid dynamic torque, in-lbs.

F_{RE} = Reynold number factor

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

P_1 = upstream static pressure at flow condition, psia

P_2 = downstream static pressure at flow condition, psia

D = disc diameter, in.

C_{T1} = subsonic torque coefficient

C_{T2} = sonic torque coefficient

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

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

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

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

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

The effect of valve closure during the transition from subsonic to sonic flow is to greatly amplify the resulting torques. In fact, the maximum dynamic torque occurs when initial sonic flow occurs 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-34577-WORST   TORQUE TABLE   4   8 / 4 / 81

JOB:FLOR.PWR:ST.LUCIE- P2-VARIABLE SIZE ADJUSTED(REYNLDS NO.FNCTN!)
SAT.STEAM/AIR MIXTURE WITH 1.4 LBS STEAM PER 1-LBS AIR
SPEC.GR.= .738255   MOL.WT.= 21.3872   KAPA(ISENT.EXP.)= 1.19775   R= 72.1972
GAS CONSTANT-CALC.
SONIC SPEED(MOVING MIXTR.)= 1292.39   FEET/SEC AT 200   DEG.

CRIT.CASE INLET VELOCITY IS 1.30492   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 .7051(48 IN)
WITH STMIX.)FIRST SONIC@ 72 DEG.V.A.)
ABSOL.MAX.TORQUE(FIRST SONIC)AT 72-68 DG.VLV.ANG.=   1246 IN-LBS @ 68 DEG.
MAX.TORQUE INCLUDES SIZE EFFECT(REYNOLDS NO.ETC)APPX. X 1.05066   FOR 7.981
INCH BASIC I.D. VALVE
ALL PRESSURES USED:STATIC(TAP)PRESS.-ABSOLUTE:P2 INCL.RECOVERY PRESS.
(TORQUE)CALC'S VALIDITY:P1/P2>1.07:

VALVE TYPE: 8"-1200:2.95/5.75   CLASS 150
DISC SIZE: 7.2 INCHES   OFFSET ASYMMETRIC DISC
SHAFT DIA.:   1.125   INCHES
BEARING TYPE: BRONZE
SEATING FACTOR: 15
INLET PRESS.VAR.MAX.:   37.6   PSIA
OUTLET PRESSURE(P6):   20.63   PSIA (72 DEG. ACTUAL PRESS.ONLY(VAR.))
MAX.ANG.FLOW RATE: 12368.8   CFM: 19554.2   SCFM: 1074.95   LB/MIN
CRIT.SONIC FLOW-90DG: 1017.58   LB/MIN AT 20.4778   INLET PSIA
VALVE INLET DENSITY: 8.69077E-02   LB/FT^3-MIN. .110771   LB/FT^3-MAX.
FULL OPEN DELTA P: 13.343 PSI
SYSTEM CONDITIONS:
PIPE IN-PIPE-OUT -AND- AIR/STEAM MIXTURE SERVICE @ 200 DEG.F
MINIMUM 0.75 DIAM. PIPE DOWNSTREAM FROM CENT.LINE SHAFT.

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

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

ANGLE APPRX.	P1 PSIA	P2 PSIA	DELP PSI	PRESS. RATIO	FLOW (SCFM)	FLOW (LB/MIN)	TD ----INCHLBS--	TB+TH	- TD-TB-TH	TIME(LOC SEC.
90	29.5	16.16	13.34	0.548	19554	1074	406	2	373	2.[illegible]
85	30.4	17.06	13.36	0.561	23978	1318	540	8	502	3.1[illegible]
80	31.1	17.76	13.38	0 570	23681	1301	566	40	526	3.4[illegible]
75	31.8	18.37	13.40	[illegible]	22861	1256	998	70	927	3.6[illegible]
72	32.1	18.78	13.32	0[illegible]	20913	1149	1253	89	1164	3.7[illegible]
70	32.3	18.63	13.69	0.[illegible]6	20167	1108	1081	76	1005	3.[illegible]
65	32.8	17.97	14.80	0.548	17915	984	1022	72	950	[illegible]
60	33.1	17.29	15.84	0.522	15125	831	682	48	633	4.1[illegible]
55	33.4	16.56	16.83	0.496	12642	694	542	59	482	4.[illegible]
50	33.5	16.05	17.50	0.478	10275	564	379	69	310	4.[illegible]
45	33.6	15.65	17.95	0.466	10391	571	321	77	243	4.[illegible]
40	33.7	15.38	18.30	0.457	7283	400	236	86	150	4.[illegible]
35	33.9	15.09	18.81	0.445	5216	286	147	94	52	4.[illegible]
30	34.3	14.91	19.34	0.435	4130	227	87	105	-17	4.[illegible]
25	34.7	14.31	19.91	0.427	2962	162	63	114	-51	4.[illegible]
20	35.3	14.75	20.53	0.418	1827	100	51	122	-70	4.[illegible]
15	35.9	14.71	21.19	0.410	1022	56	26	130	-103	5.[illegible]
10	36.5	14.70	21.83	0.402	490	26	18	136	-118	5.[illegible]
5	37.1	14.70	22.44	0.396	147	8	14	142	-127	5.[illegible]
0	37.6	14.70	22.90	0.391	0	0	1086	131	955	5.[illegible]

SEATING + BEARING + HUB SEAL TORQUE (M/M)=   1217 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) =   1253 IN-LBS @ 70 DEG.

B. Valve Stress Analysis

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

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

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

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

where

S_{max} = maximum combined stress, psi

T_1 = direct tensile stress, psi

T_2 = tensile stress due to bending, psi

S_1 = direct shear stress, psi

S_2 = shear stress due to torsion, psi

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

C. Operator Analysis

Model: Bettis N721C-5R40

Rating: 7900 in-lbs (at full open and closed positions only)

Max. valve torque: 1253 in-lb.

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

The Bettis spring-opposed cylinder furnished was specifically designed for the requirements of the original order which was based on a valve operating torque of 1419 in-lbs.

Since the LOCA induced torque derived in this analysis is less than both the figure originally considered and the maximum rating, it is concluded that the Bettis model furnished is structurally suitable to withstand combined LOCA and seismic loads.

IV. Conclusion

The calculated maximum torque of 1253 in-lbs. resulting from LOCA conditions is less than the torque which was used for the original seismic stress analysis (copy included on attachment 2).

Since the stress levels in the analysis are below the design allowable, it is concluded that the valve assembly is structurally suitable for LOCA induced forces combined with seismic and other loads.

ATTACHMENT 1A

PRATT PROPOSAL LETTER

TELEPHONE [illegible]

D-34577

PRATT

HENRY PRATT COMPANY

401 SOUTH HIGHLAND AVENUE · AURORA, ILLINOIS 60507

February 16, 1981

EBASCO Services, Incorporated
Two Rector Street
New York, NY 10006

Attention: S. K. Sinha

Subject: Florida Power & Light/St. Lucie Unit 2
Containment 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:

I.

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) 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.

   D. Maximum and minimum delay times from LOCA to initiation 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.

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 valves 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 $16,500. ($6,500 for 8" valve and $10,000 for 48" valve.)

4. The completion of this analysis is projected to be 8 weeks after receipt of P.O. 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 30 days. The terms of payment will be net 30 days.

II. As a secondary matter, Pratt will re-evaluate, at no additional cost, the unit 1 analysis previously submitted and reissue as required.

This re-evaluation is contingent on receipt of additional information defined in I.1.B, I.1.C, and I.1.D, above for the 48" unit 1 valves.

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

T. J. Wrona

T. J. Wrona, Manager
Contract and Proposal Engineering

/kk

cc: G. L. Beane

ATTACHMENT 1B

CUSTOMER/ENGINEER RESPONSE
TO REQUEST FOR INFORMATION

D-34577

05-5C/5-80

EBASCO SERVICES INCORPORATED

Two World Trade Center New York N Y 10046

PURCHASE CONTRACT NO. NY-422537

DATE OF CONTRACT June 27, 1975

SUPPLEMENT NO. 26

DATE May 15, 1981

PAGE 1

TO Henry Pratt Company
c/o Heyward Incorporated
2105 Park Avenue, Suite 6
Orange Park, Florida 32073

CREDIT TERMS APPROVED DATE 5-28-81

RECEIVED MAY 26 CUST. SERVICE

FLORIDA POWER & LIGHT COMPANY
ST LUCIE PLANT
1983 - 890 MW EXTENSION - UNIT NO. 2
BUTTERFLY VALVES AND ACCESSORIES

Gentlemen:

This Supplement is issued to authorize Seller to perform the Containment Isolation Purge Valve Analysis in accordance with the data attached hereto and detailed below:

ITEM NO	QTY	DESCRIPTION OF SCOPE	ITEM PRICE
-	Lot	Seller shall perform an analysis of the suitability of the 8" Containment Isolation Purge Valves (Item No. 63A, Seller's item D0096-10) to withstand Aerodynamic LOCA conditions in accordance with the attached data.	$6,500.00

NOTE: This price is firm through report submittal and not subject to any price adjustment.

REPORT SUBMITTAL — Seller shall submit the certified report for Purchaser's review not later than July 17, 1981 addressed as follows:

Florida Power & Light Company
c/o Ebasco Services Inc., Agent
Two World Trade Center
New York, N Y 10048
Attn: K N Chow
Supervising Mechanical/
Nuclear Engineer

LOGGED 5/26 AE RDN
CREDIT TO 5/27 FROM 5/28
A.E. TO 5/28 FROM
Short Order TO FROM
TO: ORDER ENTRY

EBASCO SERVICES INCORPORATED

Two World Trade Center New York, N Y 10048

PURCHASE CONTRACT
NO. NY-422537

DATE June 27, 1975

SUPPLEMENT NO. 26

DATE: May 15, 1981

PREVIOUS TOTAL CONTRACT PRICE $2,019,236.00

INCREASED BY THIS SUPPLEMENT (FIRM) . . . . $ 6,500.00

PRESENT TOTAL CONTRACT PRICE $2,025,736.00

Except as expressly modified herein, all terms and conditions of this Contract remain unchanged and shall also apply to this supplement.

This supplement is being issued in accordance with the terms of Contract No. NY-422537 and is being furnished you in duplicate. If this supplement is acceptable to you, please so indicate in the space provided below and return the original to us within five (5) days.

Very truly yours,

Accepted - Date ____________________
Henry Pratt Company

By________________________________

Title_______________________________

FLORIDA POWER & LIGHT COMPANY
EBASCO SERVICES INCORPORATED AGENT
W C Arent, Director of Purchasing

By________________________________
P J Pulgrano
Contract Administrator

PJP/yf

cc: Henry Pratt Co
401 South Highland Ave
Aurora, Illinois 60507
Attn: Mr J Sirovatka

cc: Henry Pratt Co
55 Washington St
East Orange, New Jersey 07017
Attn: Mr J Peirano

ATTACHMENT 1

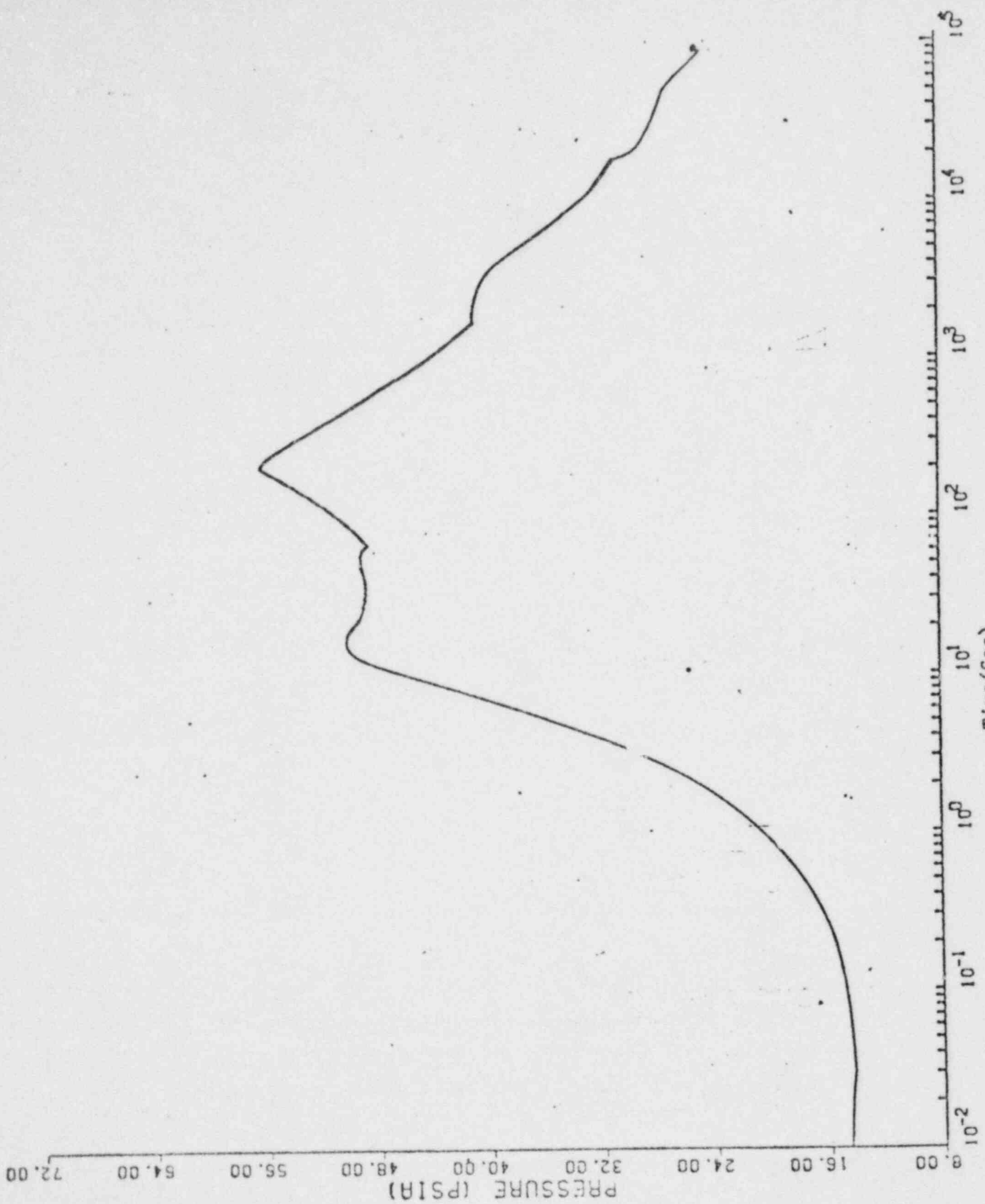
PURCHASE REQUISITION 91316

Information to be used for anlysis of 8" containment isolation purge valves.

1) Downstream Resistance - 24" W.C. at 2500 cfm
2) Maximum delay from LOCA to initiation of valve rotation - 2.95 sec.
3) Containment Pressure - Time Curve - Attachment 2
4) Containment Temperature - Time Curve - Attachment 3
5) Valve orientation - Plan View - Attachment 4 (valve Tag. No.I-FCV-25-20)
6) Valve orientation - Section View - Attachment 5 (valveTag. No. I-FCV-25-20)
7) Flow diagram - Attachment 6 (2 sheets)

Prepared: J.F. Rabon 4/30/81
Checked: R Raghavan 4/30/81

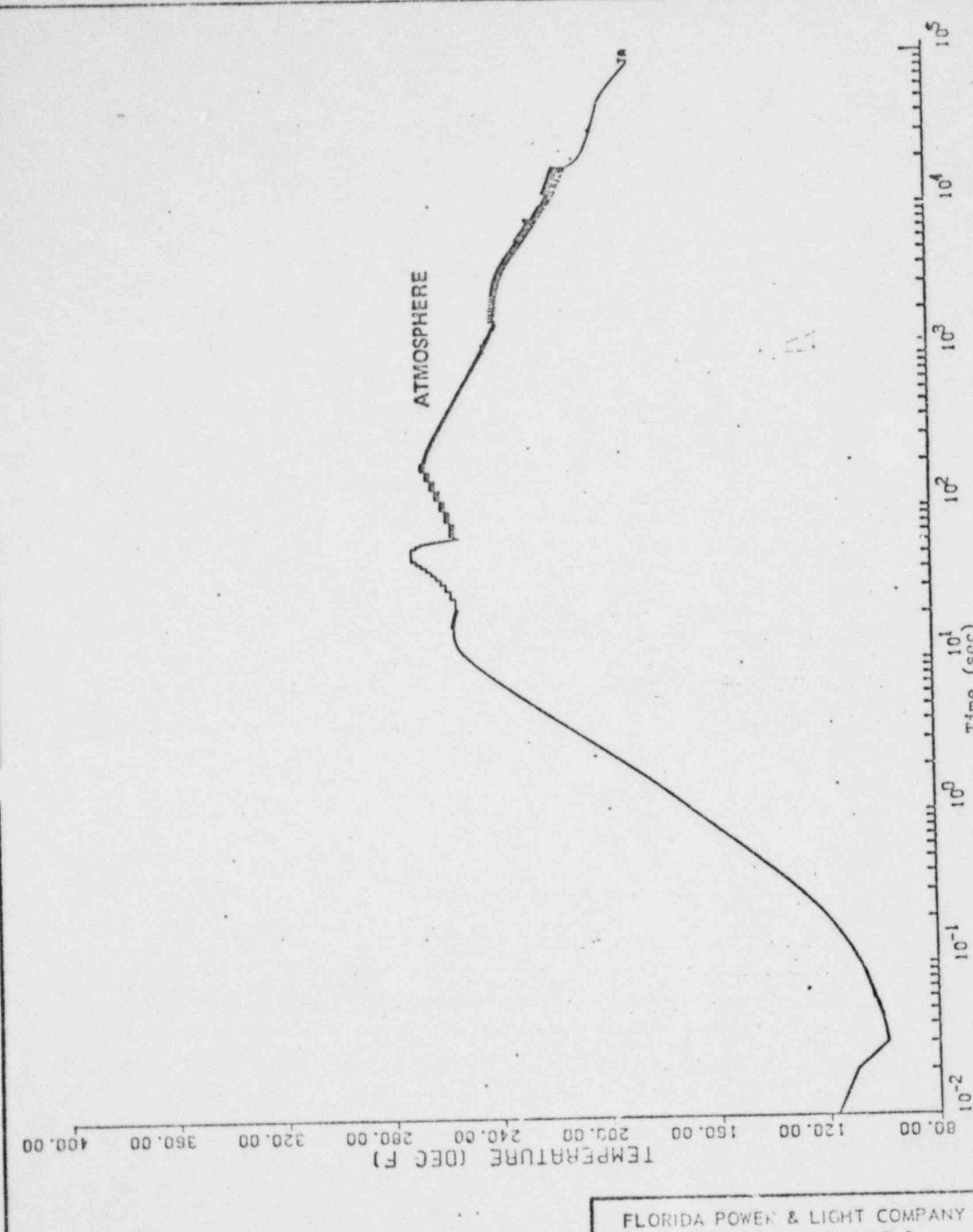
ATTACHMENT 2



FLORIDA POWER & LIGHT COMPANY
ST. LUCIE PLANT UNIT 2

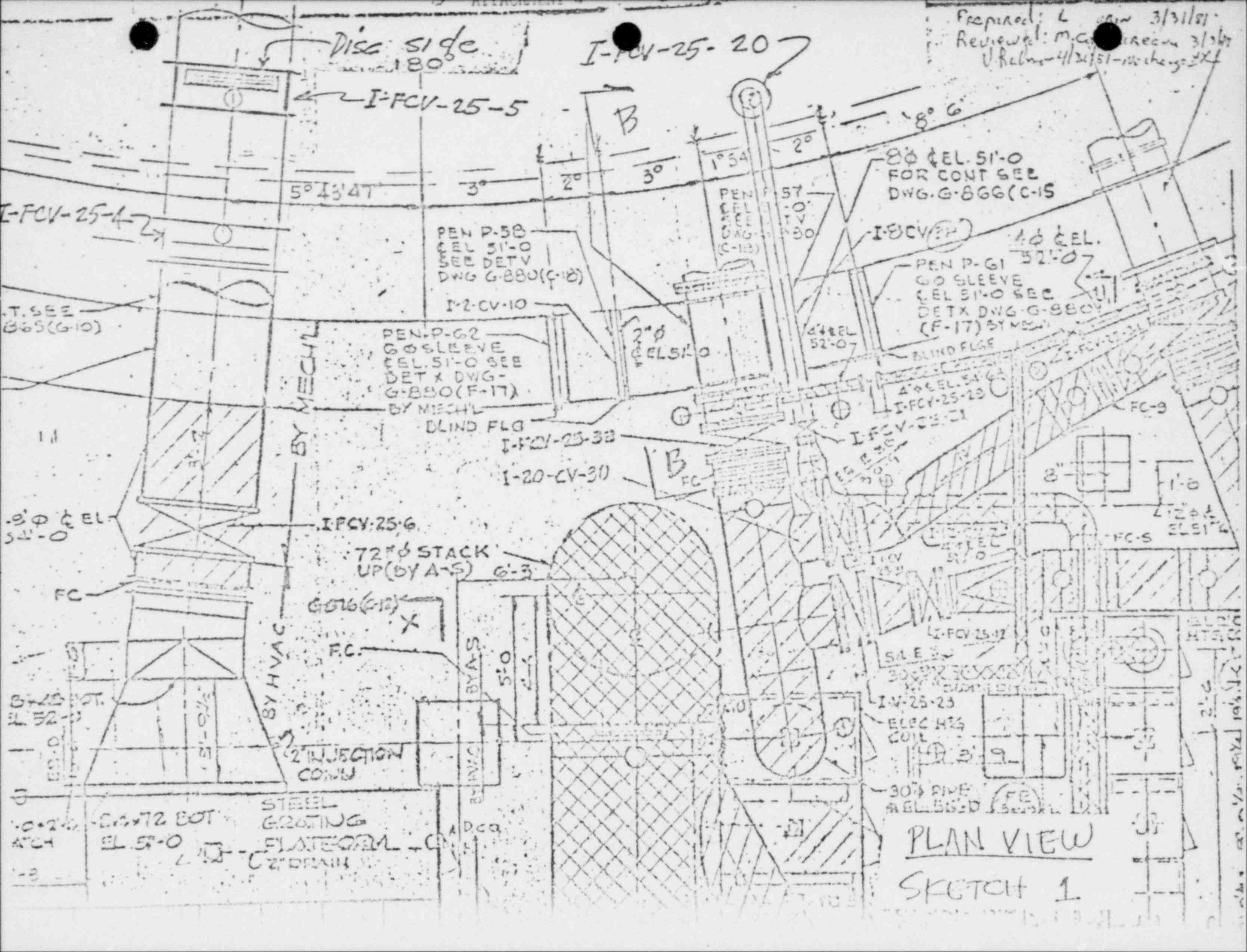
CONTAINMENT PRESSURE - 9.82 FT^2
DESLS (MAX. SAFETY INJECTION)

ATTACHMENT 3



FLORIDA POWER & LIGHT COMPANY
ST. LUCIE PLANT UNIT 2

CONTAINMENT TEMP.-9.82 FT^{2}DESLS
(MAXIMUM SAFETY INJECTION)



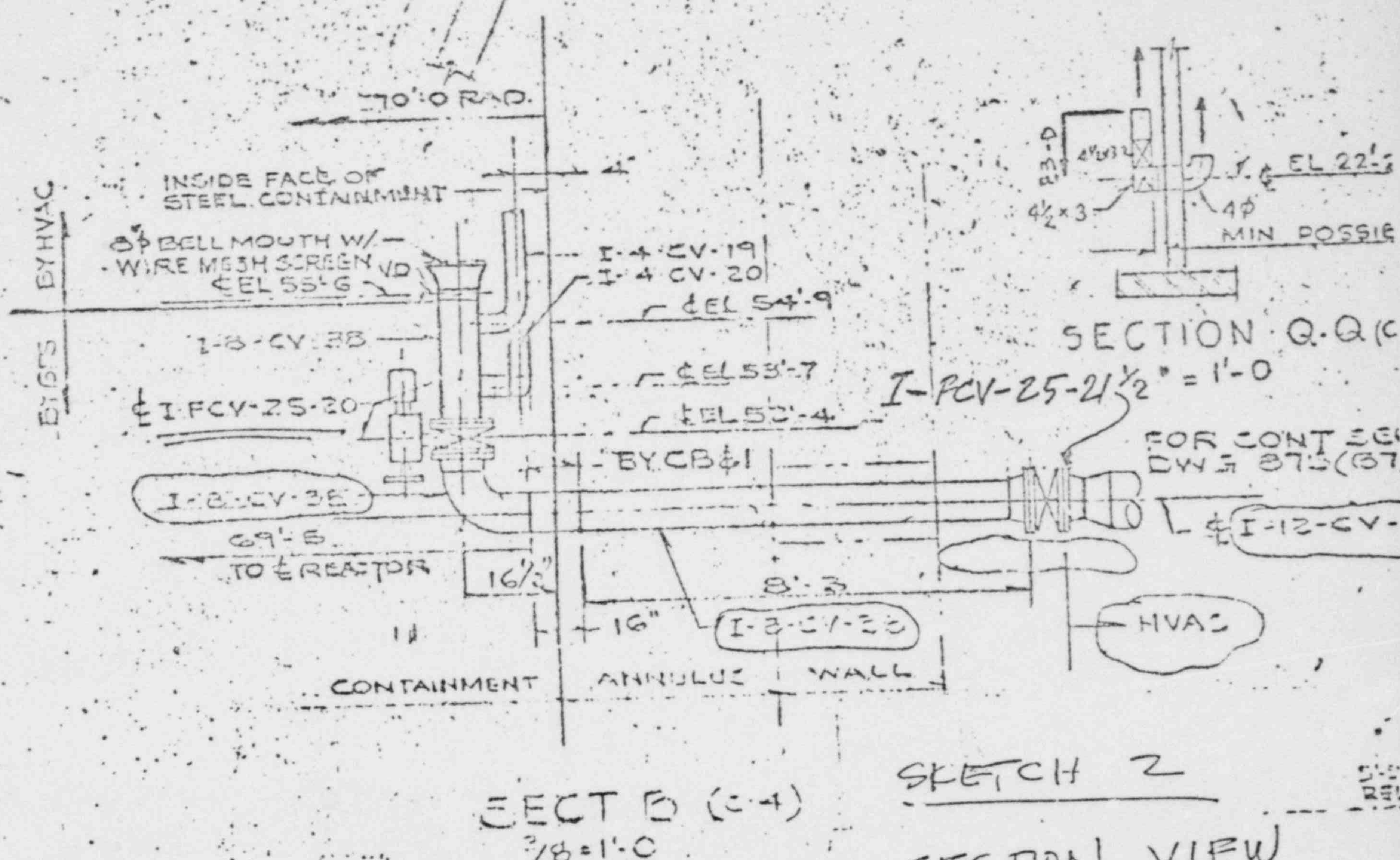
Disc side
180°
I-FCV-25-5
I-FCV-25-20
I-FCV-25-4
B
5° 43' 47"
3°
2°
3°
1° 54'
2°
8° 6'
8Ø ¢EL 51'-0
FOR CONT SEE
DWG G-866(C-15
4Ø ¢EL.
52'-0
PEN P-58
¢EL 51'-0
SEE DET V
DWG G-880(C-18)
I-2-CV-10
PEN P-62
6Ø SLEEVE
¢EL 51'-0 SEE
DET X DWG-
G-880(F-17)
BY MECH'L
BLIND FLG
2"Ø
¢EL 51'-0
PEN P-61
6Ø SLEEVE
¢EL 51'-0 SEE
DET X DWG G-880
(F-17) BY MECH'L
BLIND FLGE
4"Ø EL 52'-0
4"Ø EL 54'-6
I-FCV-25-29
I-FCV-25-21
I-FCV-25-38
I-20-CV-30
B
FC
FC-9
FC-5
8"
1'-0
BY MECH'L
I-FCV-25-6
72"Ø STACK
UP(BY A-S)
6'-3
FC
G-876(C-12)
X
F.C.
BY HVAC
BY A-S
5'-0
2" INJECTION
CONN
STEEL
GRATING
PLATFORM
2" DRAIN
EL 51'-0
I-FCV-25-17
I-V-25-23
ELEC HTG
COIL
30"Ø PIPE
FE
PLAN VIEW
SKETCH 1
Prepared: L 3/31/81
Reviewed: M.C. 3/31/81

Prepared L. Cirin 3/31/81
Reviewed M.C. Ilhaneo 3/31/81

SECTION P_1-P_1 (OPP.

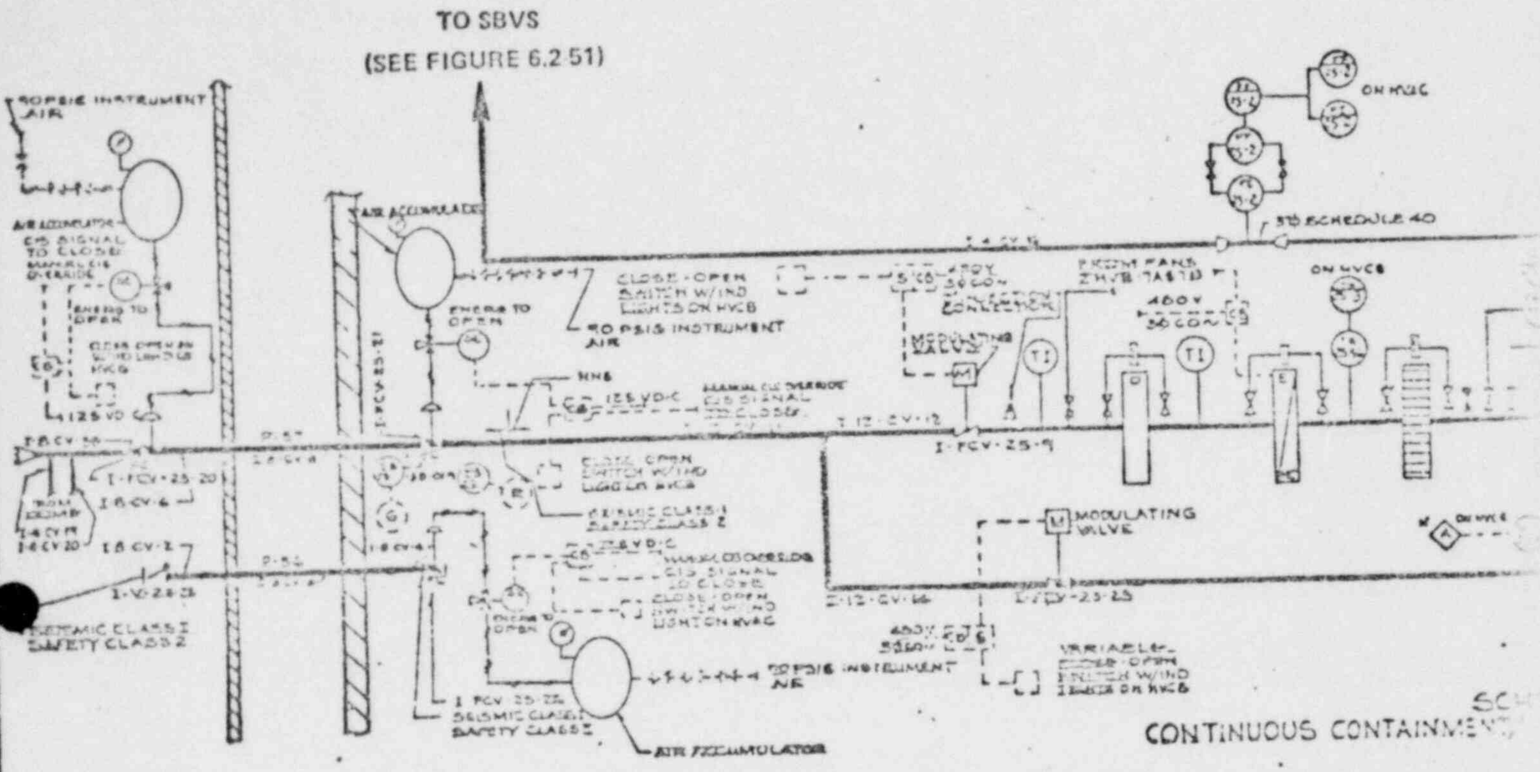
SECTION P_2-P_2 (E-

1/2" = 1'-0

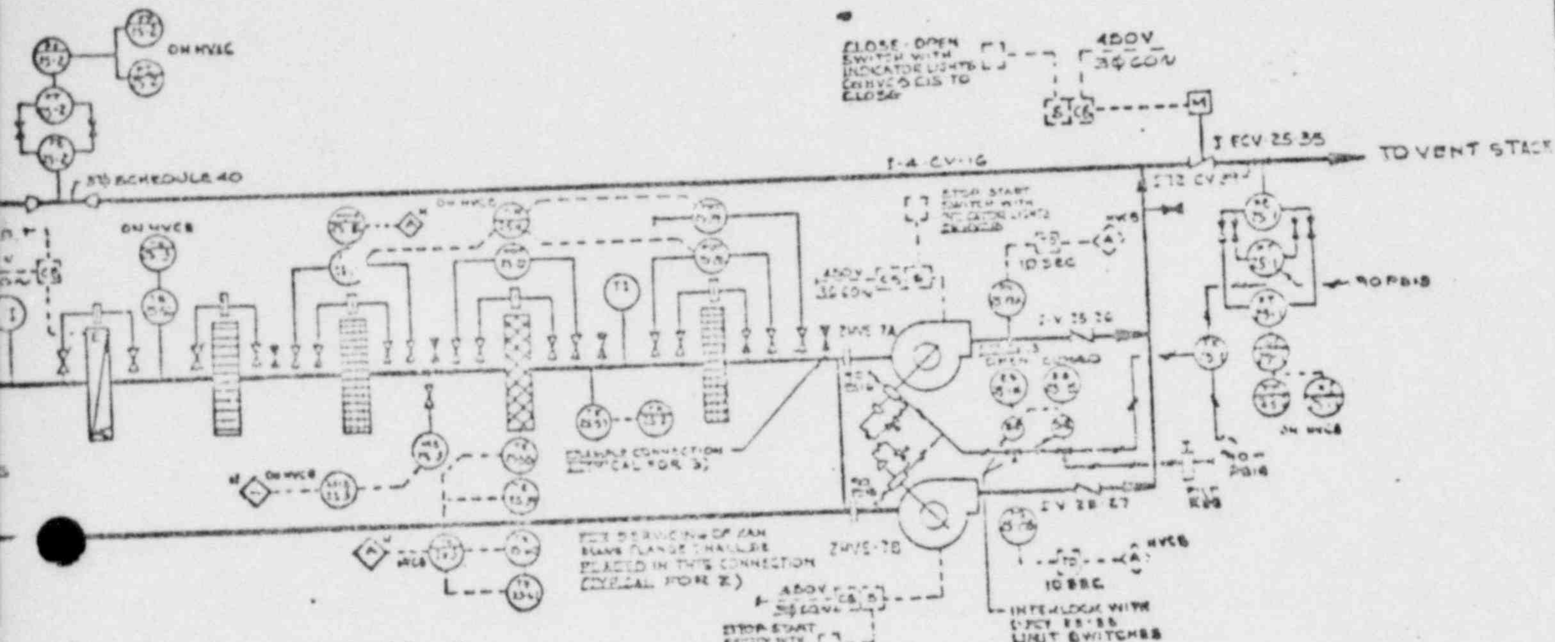


SKETCH 2

SECTION VIEW



SCH
CONTINUOUS CONTAINMENT



SCHEME - J
CONTINUOUS CONTAINMENT/HYDROGEN PURGE SYSTEM

AMENDMENT NO. 0 (12/80)

FLORIDA POWER & LIGHT COMPANY
ST. LUCIE PLANT UNIT 2

CONTINUOUS CONTAINMENT HYDROGEN
PURGE SYSTEM – P & ID
FIGURE 9.4-11

REF DWG: PART OF 2998-G-879 SO3 (REV. 5)

ATTACHMENT 2

Nuclear Purge Valve

Stress Analysis

JH. 5-71

Rev. 1
May 21, 1980

STRESS REPORT FOR

8" NRS/N72[illegible]-8R40

NUCLEAR CLASS 2

PER SECTION III

ASME BOILER AND PRESSURE VESSEL CODE

Project Site: St. Lucie Unit No. 2

Customer: Florida Power & Light Co.

Engineer: EBASCO

Specification: FLO 2998.114 Rev. 3

Purchase Order: NY 422[illegible]

Pratt Job No.: D0096-10 11

Valve Tag Nos.: I-FCV-25-20 & 21
I-FCV-25-26

General Arrangement Drawing: C-5531 Rev. 2

Cross Section Drawing: C-5530 Rev. 1

Prepared by: Rao N. Koza
Date: 5/21/80
Reviewed by: [illegible signature]
Date: 5-21-80
Certified by: Bomi M. Zarola
Date: 5/21/80

BOMI M. ZAROLIA
30701
REGISTERED
PROFESSIONAL
ENGINEER
OF
ILLINOIS

Form No. HPCo. 3.172

HENRY PRATT COMPANY

Design Review Record

Rev. 1
May 21, 1980

SITE St. Lucie Unit No. 2

CUSTOMER Florida Power & Light Co.

ENGINEER EBASCO

PURCHASE ORDER NO. NY422537

DESIGN SPEC. NO. FLO 2298.114 Rev. 3

PRATT PROD. ORDER D-0096-10 & 11

Valve Size, Type & Class

8" NRS, Class 2

INTERNAL DESIGN REVIEW AND CHECKING OF STRESS REPORTS

Stress Report No. D-0096-10 & 11 Date 2/12/80

General Arrangement Drawing No. C-5531 Rev. 2

Cross Section Drawing No. C-5530 Rev. 1

Conforms to Design Specification [X]

Conforms to ASME Codes [X]

Prepared By Rao N. Kaza Date 5/21/80

Approved By T. J. Vrona Date 5-21-80

Management Review of Internal Design Review

Approved A. K. Wilson Date 5-22-80
Vice President, and
Manager of Engineering

Form No. HPCo. 8/QAP-16-2.2

Stress Report For

8" NRS/N721C-SR40

Nuclear Class 2

Per Section III

ASME Boiler and Pressure Vessel Code

Project Site St. Lucie Unit No. 2

Customer Florida Power & Light Co.

Engineer EBASCO

Specification FLO 2998.114 Rev. 3

Purchase Order NY 422537

Pratt Job Number D0096-10 & 11

Valve Tag Numbers I-FCV-25-20 & 21

I-FCV-25-26

General Arrangement Drawing C-5531 Rev. 1

Cross-Section Drawing C-5530 Rev. 0

Prepared by: Rao N. Kaza

Date: 2/12/80

Reviewed by: [illegible]

Date: 2/12/80

Certified by: [illegible]

Date: 2/12/80

HENRY PRATT COMPANY

Design Review Record

Site St. Lucie Unit No. 2

Customer Florida Power & Light Co.

Engineer EBASCO

Purchase Order No. NY422537

Design Spec. No. FLO 2298.114 Rev. 3

Pratt Prod. Order D0096-10 & 11

Valve Size, Type & Class:

8" NRS, Class 2

Internal Design Review and Checking of Stress Reports

Stress Report No. D0096-10 & 11 Date 2/12/80

General Arrangement Drawing No. C-5531 Rev. 1

Cross Section Drawing No. C-5530 Rev. 0

Conforms to Design Specification ☒

Conforms to ASME Codes ☒

Prepared by Rao N. Kaza Date 2/12/80

Approved by R.J. Krone Date 2/12/80

Management Review of Internal Design Review

Approved A.K. Wilson Date 2/15/80

Position Vice President - Mgr of Engineering

Form No. HPCo. 8/QAP-16-2.2

SUMMARY DATA

Valve size and type: 8" NRS

Operator: N721C-SR40

Pratt Job No: D0096-10 & 11

Standard Calculation Pressure (Design Pressure)	285	psig
Operating Pressure (Maximum)	150	psig
Acceleration Levels (Design)	3	gx
	3	gy
	3	gz
Acceleration Levels (Specified)	3	gx
	3	gy
	2	gz
Specified Design Pressure	-3 to 65	psig
Minimum Valve Wall Thickness	1.03	inches
Code Required Wall Thickness	.31	inches

Table 1	STRESS LEVELS FOR VALVE COMPONENTS						
Component	Code Ref. Paragraph	Name & Symbol		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. 70	652	Sm 17500
		Primary membrane stress in body	P_m'	36	ASME SA-516 Gr. 70	1051	Sm 17500
	NB-3545.2	Primary plus secondary stress due to internal pressure	Q_p	36	ASME SA-516 Gr. 70	3740	Sm 17500
	NB-3545.2	Pipe Reaction Stresses Axial Load	Ped	36	ASME SA-516 Gr. 70	2789	1.5Sm 26250
		Bending Load	Peb	36		4999	
		Torsional Load	Pet	36		4999	
	NB-3545.2	Thermal Secondary Stress	Q_t	38	ASME SA-516 Gr. 70	1105	Sm 17500
	NB-3545.2	Primary plus secondary stress	S_n	38	ASME SA-516 Gr. 70	8948	3Sm 52500
	NB-3545.3	Normal duty fatigue stress Na $\geq$ 2000	S_p	38	ASME SA-516 Gr. 70	6744	Sm 17500
Disc	NB-3546.2	Combined bending stress in disc	S(1)	39	ASME SA-516	2934	1.5Sm 26250
	NB-3546.2	Shear tear out of shaft thru disc	S(4)	41	ASME SA-516 Gr. 70	1442	.6Sm 10500

Table 1	STRESS LEVELS FOR VALVE COMPONENTS						
Component	Code Ref. Paragraph	Name & Symbol		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	17350	Sm 33700
	NB-3546.3	Torsional shear stress at reduced pin cross-section	S(12)	43	ASME SA-564 Type 630 Cond. H-1150	9218	.6Sm 20220
Disc Pin	NB-3546.3	Combined shear stress in top disc pin	S(13)	44	ASME SA-320 Gr. B8M	5609	.6Sm 8160
	NB-3546.3	Bearing stress on top pin	S(16)	44	ASME SA-320 Gr. B8M	4043	Sm 13600
Shaft Bearing		Compressive stress on shaft bearing	S(17)	45	ASTM B-438 Gr. 1 Type II	3651	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.70	943	.6Sm 10500
	NB-3546.1	Shear tear out of cover cap bolt head thru cover cap	S(19)	46	ASME SA-516 Gr.70	202	.6Sm [illegible]00
	NB-3546.1	Combined stress in cover cap bolts	S(20)	46	ASME SA-193 Gr.B7	6010	Sm 25000
		Combined stress in cover cap	S(23)	46	ASME SA-516 Gr.70	2713	Sm 17500

Table 1	STRESS LEVELS FOR VALVE COMPONENTS						
Component	Code Ref. Paragraph	Name & Symbol		Ref. Page	Material	Stress Level, psi	Allowable Stress Level, psi
Thrust Bearing		Bearing stress on thrust collar	S(27)	49	SAE-660	243	Sm 8800
		Shear load on thrust collar spring pin	S(28)	49	AISI 420	357	Pm 1540#
		Bearing stress of spring pin on thrust collar	S(29)	49	SAE-660	1612	Sm 8800
		Shear tear out of spring pin thru bottom of shaft	S(31)	49	ASME SA-564 Type 630 Cond. H-1150	910	.6Sm 20220

Table 1	STRESS LEVELS FOR VALVE COMPONENTS						
Component	Code Ref. Paragraph	Name & Symbol		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. 70	4195	.6Sm 10500
		Bearing stress of trunnion bolt on tapped hole	S(33)	50	ASME SA-516 Gr. 70	5157	Sm 17500
		Bearing stress of trunnion bolt on thru hole in bonnet	S(34)	50	ASME A-36	5157	Sm 12600
		Shear tear out of trunnion bolt head thru bonnet	S(35)	52	ASME A-36	2272	.6Sm 7560
		Combined stress in trunnion bolt	S(36)	52	SAE Grade 8	25015	Sm 30000
		Shear tear out of operator bolt head thru hole in bonnet	S(41)	52	ASME A-36	792	.6Sm 7560
		Bearing stress of operator bolt on thru hole in bonnet	S(42)	52	ASME A-36	4173	Sm 12600
		Combined stress in operator bolt	S(43)	54	SAE Grade 8	12688	Sm 30000
		Combined stress in bo bonnet body	S(48)	54	ASME A-36	3486	Sm 12600

Table 1 STRESS LEVELS FOR VALVE COMPONENTS

Component	Code Ref. Paragraph	Name & Symbol		Ref. Page	Material	Stress Level, psi	Allowable Stress Level, psi
Operator Mounting Con't.		Combined shear stress in bottom bonnet welds	S(53)	56		1012	.6Sm 7200
		Combined shear stress in top bonnet welds	S(60)	57		585	.6Sm 7200
		Combined stress in trunnion body	S(67)	58	ASME SA-516 Gr. 70	1943	Sm 17500

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.70	17862
Banjo	F_{N2}	60	ASME SA-564 Type 630 Cond. H-1150	8683
Cover Cap	F_{N3}	60	ASME SA-516 Gr.70	3105
Bonnet	F_{N4}	61	ASME A-36	280

TABLE 3 **DIMENSIONAL DATA**

Job Number: D0096-10 & 11 Valve Size: 8" NRS

Operator Mounting: TEE BONNET Operator: N721C-SRAO-M3-12-5

A_f	12.62	C_3	.50	g	32.2
A_m	7.06	C_6	249	G_b	92.41
A_3	.078	C_7	1.40	G_d	44.10
A_4	.068	C_8	1.50	G_T	184.82
A_5	.142	C_9	.50	g_x	3
A_6	.126	d	7.981	g_y	3
A_7	.142	d_m	7.981	g_z	3
A_8	.126	D_1	8	H_2	2.5
B_1	.59	D_2	1.125	H_3	2.5
B_2	1.375	D_3	.498	H_4	3.188
B_3	N/A	D_4	2.0	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	N/A	D_7	.50	H_8	7.0
B_7	N/A	D_8	.50	H_9	1.484
B_8	3.5	D_9	N/A	I_1	60.79
B_9	2.5	E	30E6	I_2	10.45
C[illegible]	.30	F_b	15.4	I_3	6.86
C_b	1.00	F_d	4.1	I_4	7.64
C_p	3.00	F_x	1.63	I_5	505.85
C_o	1.02	F_y	.663	I_6	.079
C_2	.42	F_z	.663	I_7	N/A

J_1	1.25	M_z	5874	ΔT_2	1.0
J_2	.50	$\overline{M_x}$	11754	T_1	1.188
J_3	.906	$\overline{M_y}$	7206	T_2	.172
J_4	1.906	$\overline{\overline{M_x}}$	11754	T_3	.75
J_5	.906	$\overline{\overline{M_y}}$	7206	T_4	.374
J_6	1.906	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	.40	N_2	4	T_8	14.19
K_3	3.5	N_3	4.	U_1	7.5
K_4	3.5	P_d	275	U_2	7.5
K_5	5.0	P_r	150	U_3	.25
K_6	1.0	P_s	285	V_1	N/A
L_1	5.0	Q_{T1}	1000	V_2	N/A
L_2	N/A	r	1.235	V_3	N/A
L_3	.468	r_i	3.99	V_4	N/A
L_4	.50	r_2	1.0	V_5	N/A
L_5	1.375	R_4	3.6	V_6	N/A
L_6	N/A	R_5	.5625	V_7	N/A
L_7	N/A	R_m	4.73	V_8	N/A
L_8	.50	R_6	1.4	W_1	179
L_9	7.2	S	30,000	W_2	23
m	3.5	t_c	1.03	W_3	221
M_x	6882	t_m	.31	W_4	22
M_y	2334	T_c	1.484	W_6	.25

W_7	18
W_8	N/A
X_o	1.00
Y_o	7.86
Z_o	2.52
Z_1	21.46
Z_2	6.14
Z_3	23.76
Z_4	23.76
Z_7	N/A

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.

19

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	$W_3(g_y Z_o + g_z Y_o)$, operator extended mass seismic bending moment about the x axis, acting at the base of the operator, in-lbs.
My	$W_3(g_x Z_o + g_z X_o)$, operator extended mass seismic bending moment about the y axis, acting at the base of the operator, in-lbs.
Mz	$W_3(g_x Y_o + g_y X_o)$, operator extended mass seismic bending moment about the z axis, in-lbs.
$\overline{Mx}$	$Mx + F_y T_5$, operator extended mass seismic bending moment about the x axis, acting at the bottom of the adapter plate, in-lbs.
$\overline{My}$	$My + F_x T_5$, operator extended mass seismic bending moment about the y axis, acting at the bottom of the adapter plate, in-lbs.
$\overline{\overline{Mx}}$	$Mx + F_y(T_5 + H_8) + g_y W_4 K_3$, operator extended mass seismic bending moment about the x axis, acting at the base of the bonnet, in-lbs.
$\overline{\overline{My}}$	$My + F_x(T_5 + H_8) + g_x W_4 K_3$, 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.
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
P_r	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
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.
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, in.
Y_o	Eccentricity of center of gravity of operator extended mass along y axis, in.
Z_o	Eccentricity of center of gravity of operator extended mass along z axis, 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

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 belt 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, si
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

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.

Pages 20-26a, Stress Level Summary sheets, Frequency Analysis Summary sheets, 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.

19

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 mutuall rpendicular directions.

Seismic loads   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.

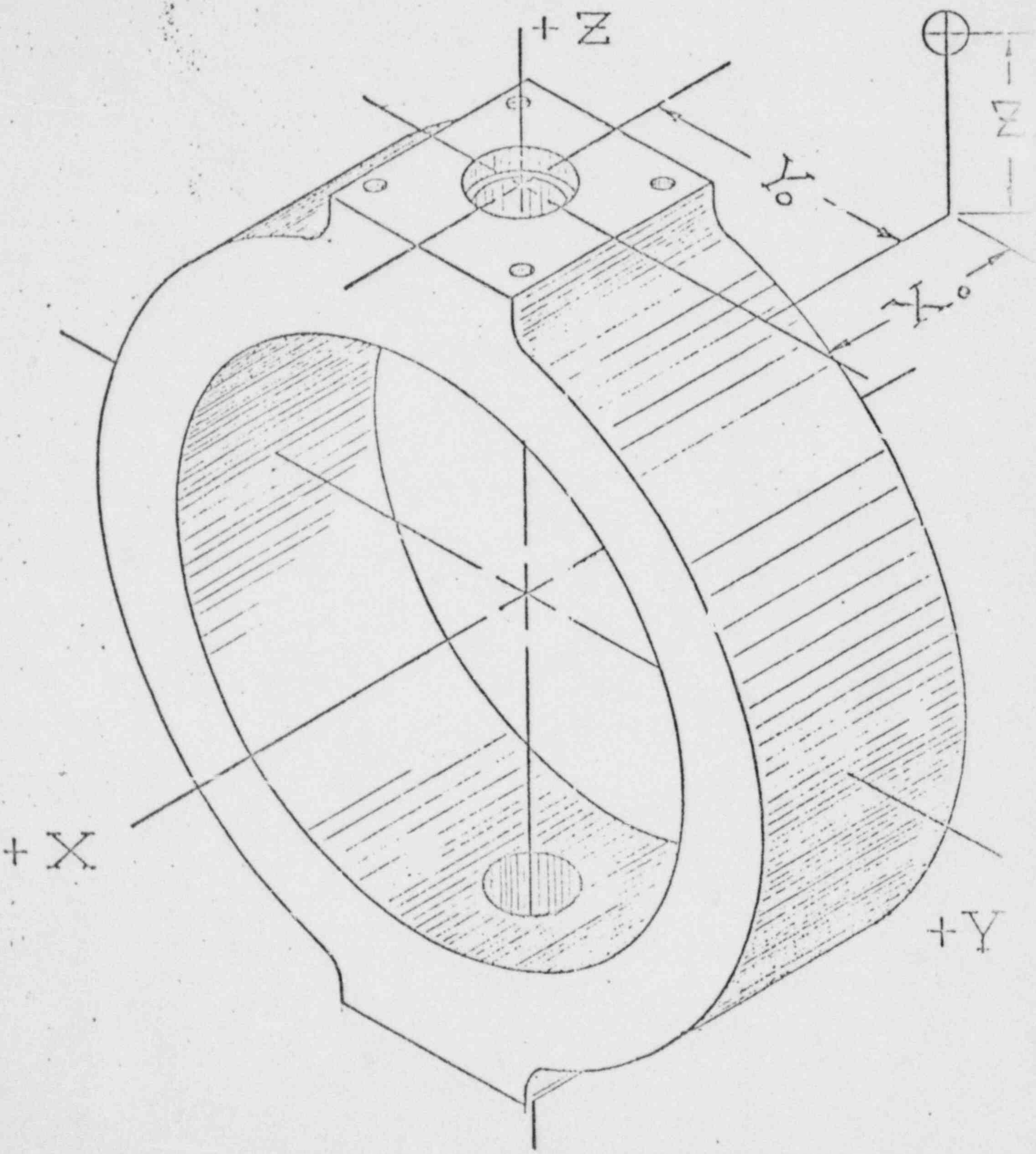


Figure 1 VALVE BODY SPATIAL ORIENTATION

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.

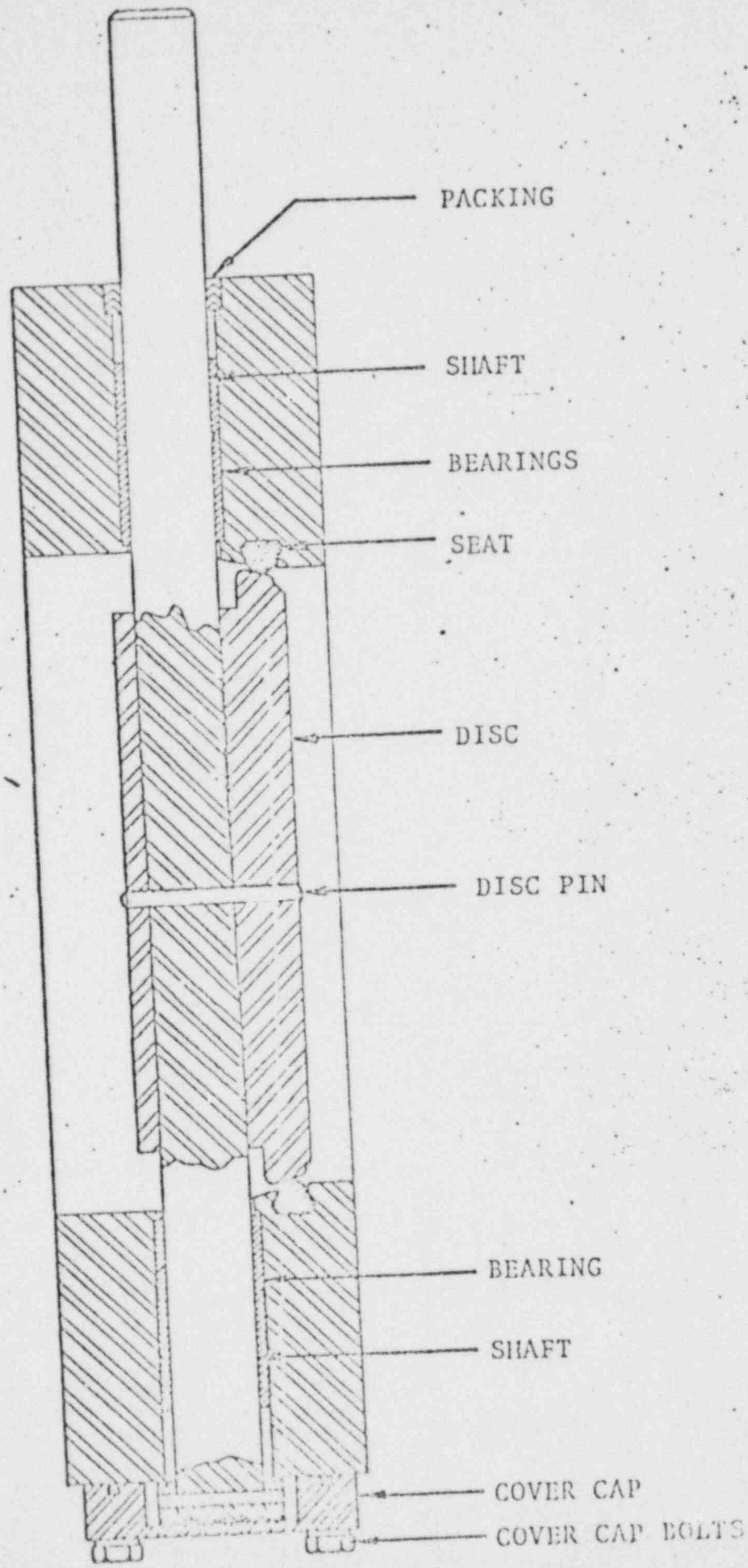


FIGURE 2 VALVE CROSS-SECTION

END CONNECTION ANALYSIS

1. 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

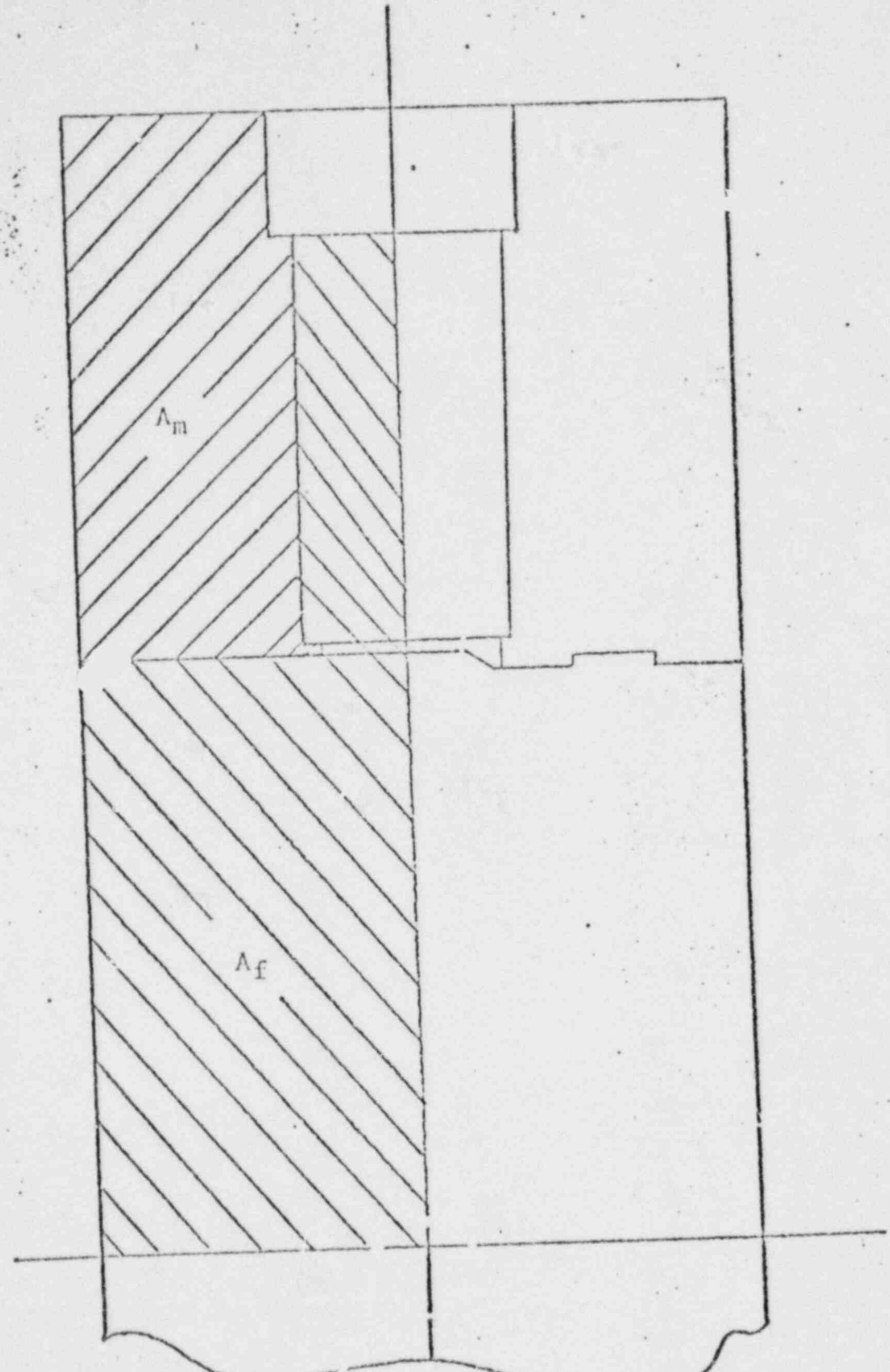
The body analysis consists of calculations as detailed in Paragraph NB-3540 of Section III of the code. Since Paragraph NB-3540 is primarily intended to control the design of high pressure, high temperature globe and gate valves, in some cases it is not possible to directly apply the equations as specified in the code. 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 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.

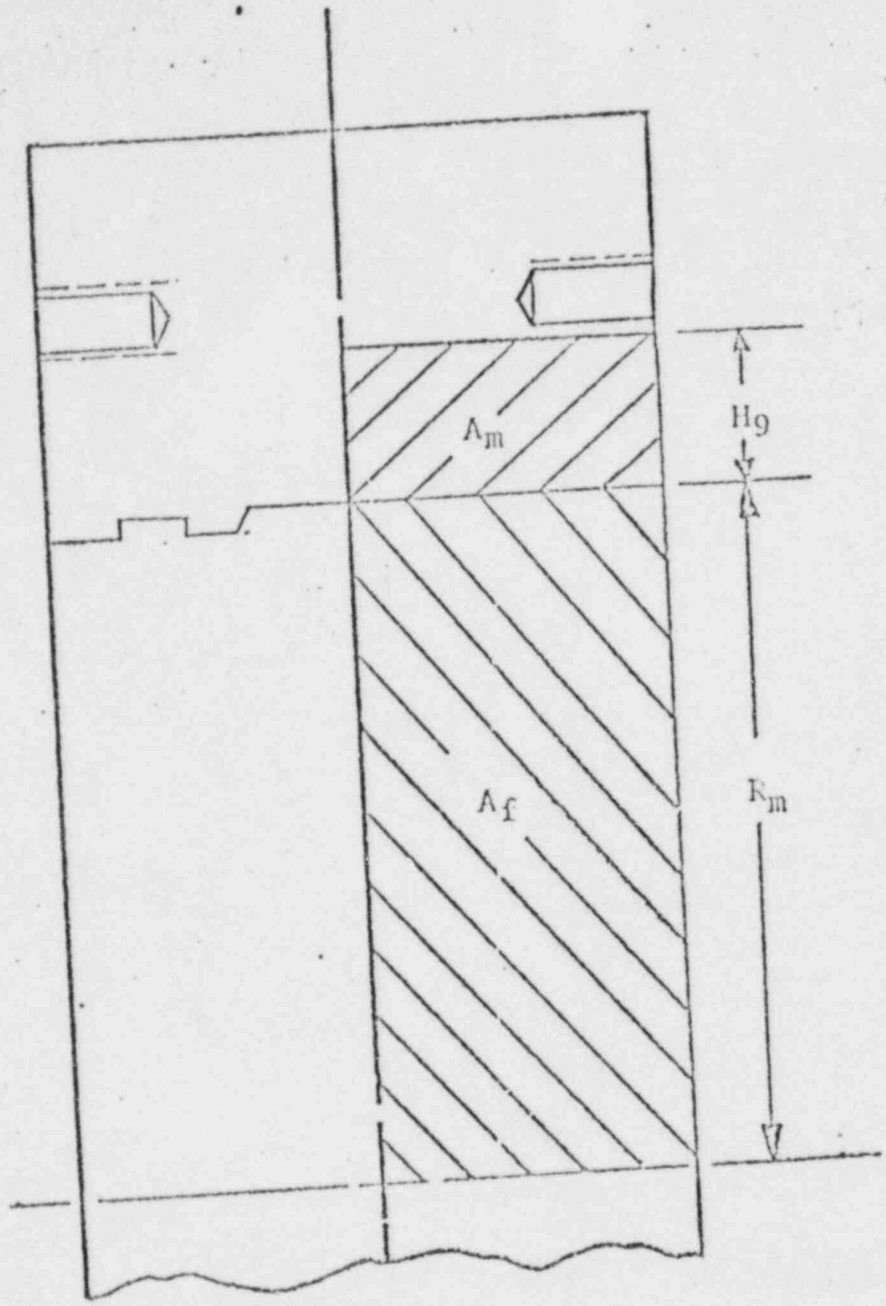
$$Q_p = C_p \left[\frac{r_i}{t_e} + .5 \right] p_s$$

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

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

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

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



PRESSURE AREA ANALYSIS

CROSS-SECTION IN BODY

Figure 4

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-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 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.

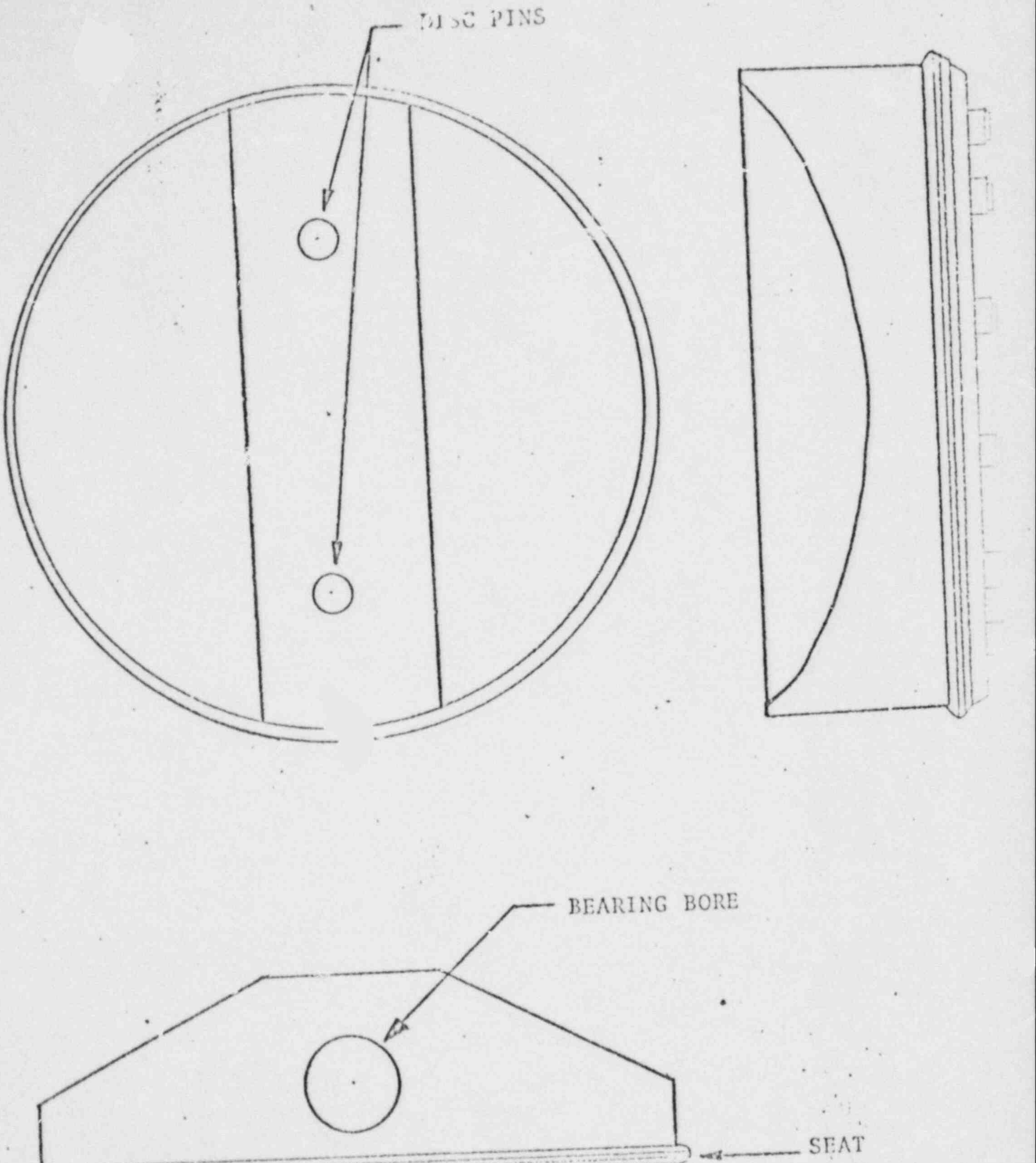
Combined bending stress in disc:

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

Where:

$$S(2) = \frac{.90413\ P_5 R_4{}^3 C_7}{I}$$ = Bending stress due to moment along shaft axis, psi

$$S(3) = \frac{.6666\ P_5 R_4{}^3 C_8}{I_3}$$ = Bending stress due to moment about shaft axis, psi



NRS DISC

Figure 5

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.

$$S(6) = \frac{\pi P_s R_4^2 + W_2 \sqrt{g_x^2 + g_y^2 + g_z^2}}{2L_9 \left(K_2 + \frac{D_2}{2} (1 - \text{SIN } 45^\circ)\right)}$$ = 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:

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

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

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

Shaft Analysis

$$S(12) = S(10)\left[\frac{\dfrac{\pi R_5{}^4}{2}}{\dfrac{\pi R_5{}^4}{2} - \dfrac{D_2 D_3{}^3}{12} - \dfrac{D_3 D_2{}^3}{12}}\right]$$

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.

$$S(17) = \frac{\pi P_d R_4{}^2 + W_2 (g_x{}^2 + g_y{}^2)^{1/2}}{2\ L_5 D_2}$$ = 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 tap 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:

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

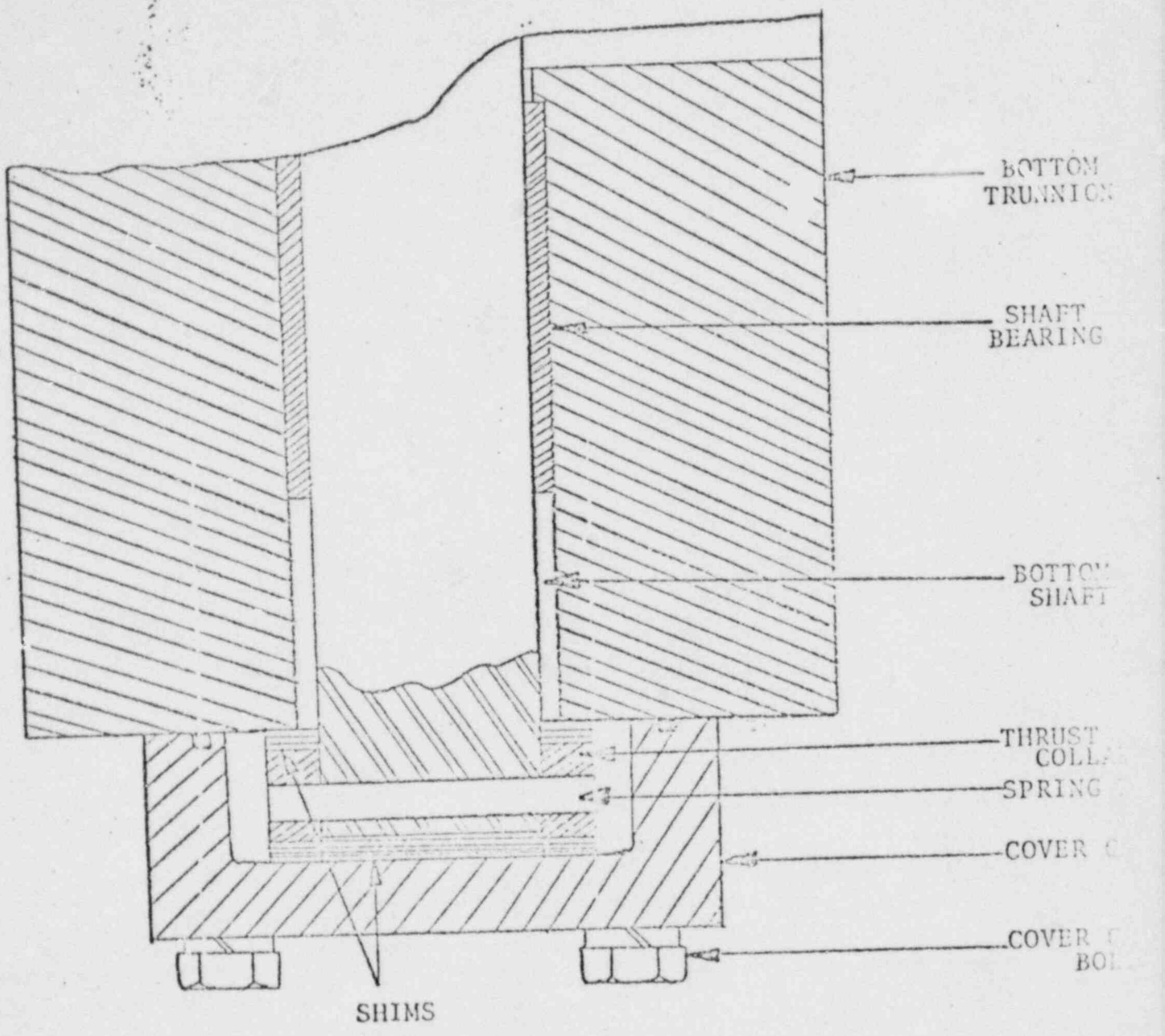
Combined stress in bolts, psi:

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

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

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

2. Cover cap stresses:

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

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

Where:

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

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

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

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:

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

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:

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

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.

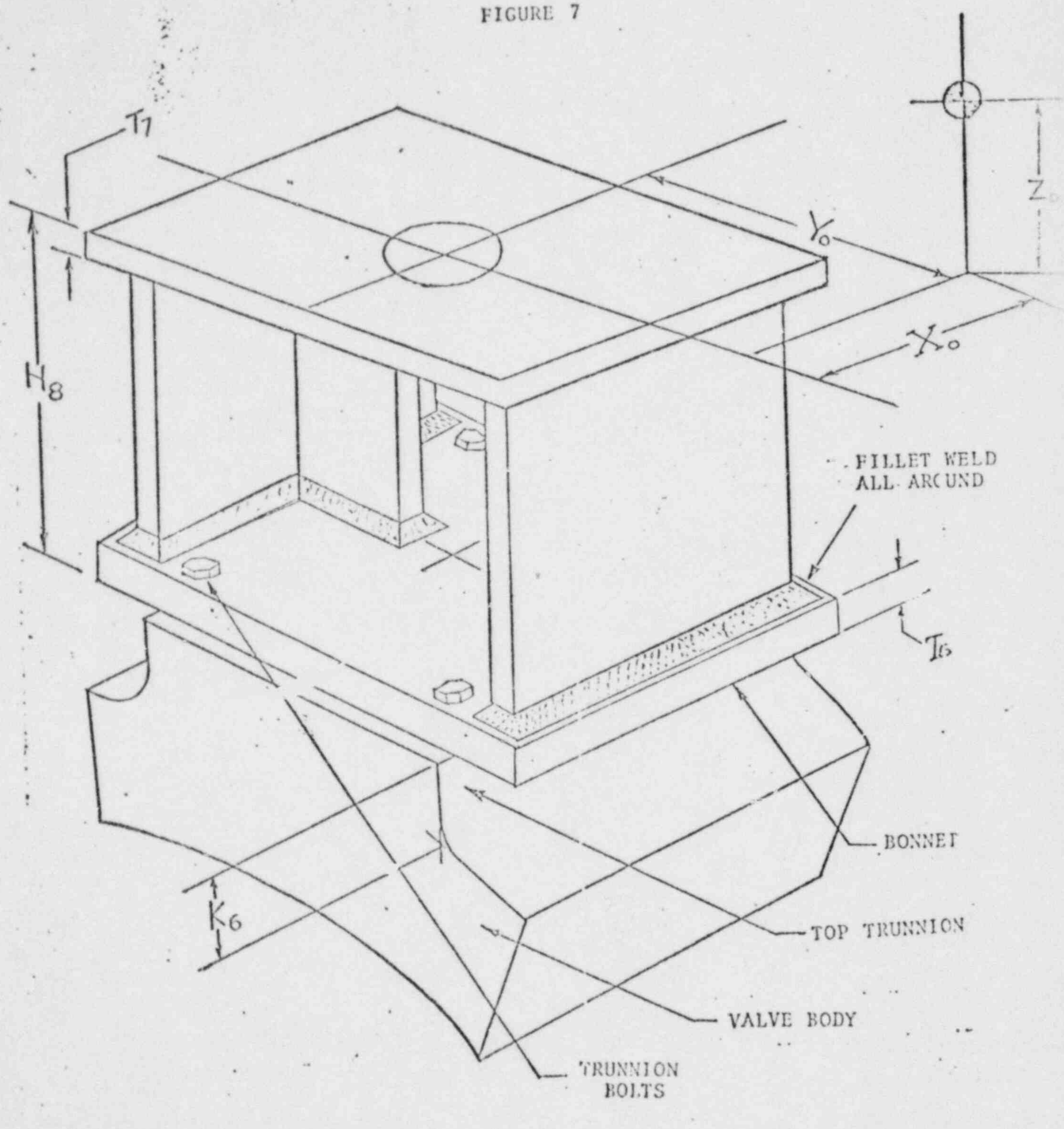
$$S(33) = \frac{\dfrac{M_z + T_8}{4(.707\ H_2)} + \dfrac{(F_x^2+F_y^2)^{1/2}}{4} + \dfrac{W_4\ (g_x^2+g_y^2)^{1/2}}{4}}{D_7 L_4}$$

(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



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

$$S(35) = \frac{\dfrac{F_z+W_4g_z}{4} + \dfrac{\overline{\overline{M_x}}(J_2+H_2)}{2J_2{}^2+2(J_2+H_2)^2} + \dfrac{\overline{M_y}(J_1+H_2)}{2J_1{}^2+2(J_1+H_2)^2}}{5.2\ D_7T_6}$$

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

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

Where

$$S(37) = \frac{F_z+W_4g_z}{4\ A_5} = \text{Direct Tensile Stress, psi}$$

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

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

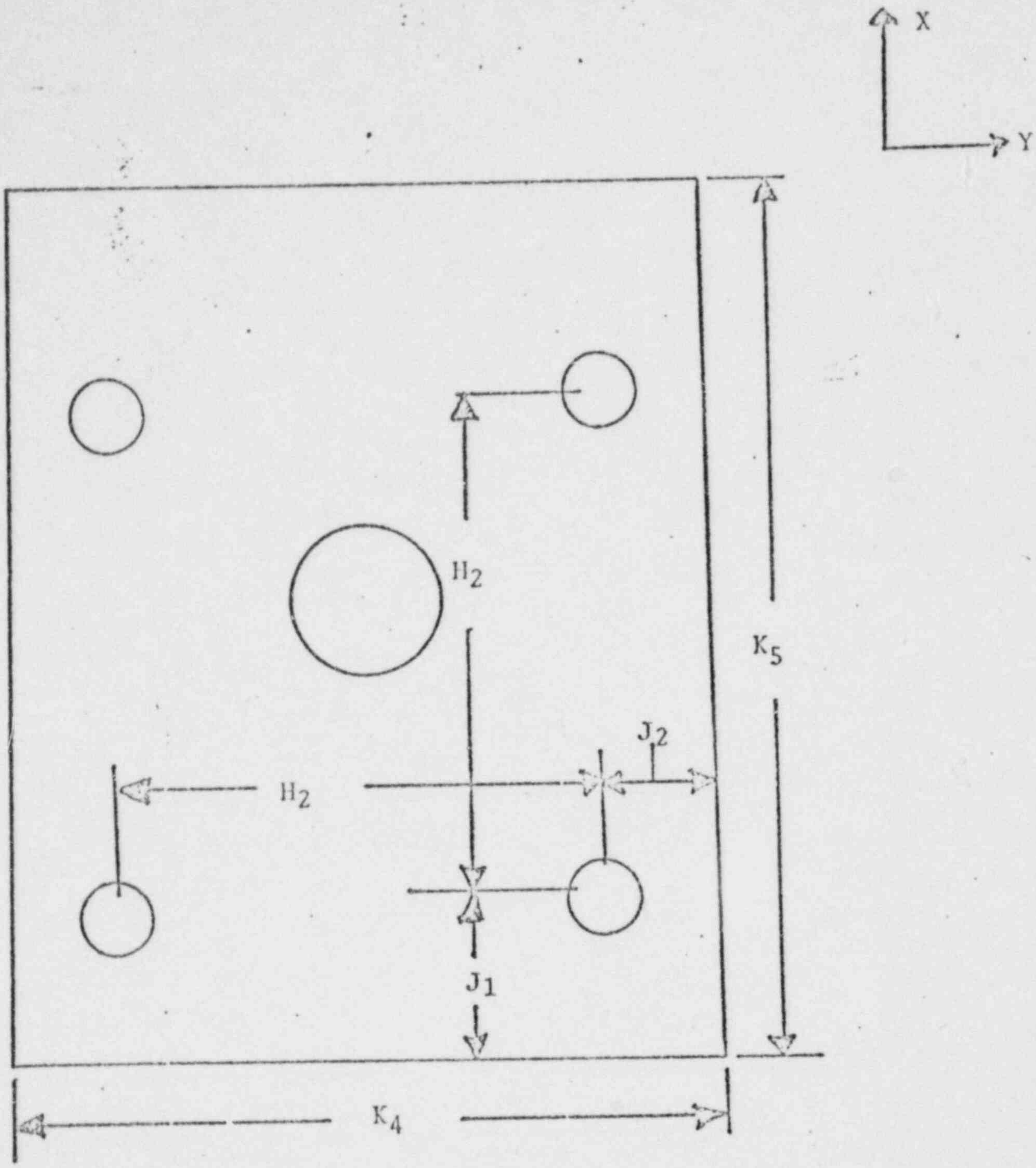
$$S(40) = \frac{(M_z+T_8)}{(.707\ H_2)4\ A_6}$$ = Shear stress due torsional load,

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

$$S(41) = \frac{\dfrac{F_z}{N_2} + \dfrac{M_x(J_4+H_4)}{2J_4{}^2+2(J_4+H_4)^2} + \dfrac{M_y(J_3+H_4)}{2J_3{}^2+2(J_3+H_4)^2}}{5.2\ D_8T_7}$$

g. Bearing stress on tapped holes in bonnet.

$$S(42) = \frac{\dfrac{M_z+T_8}{(.707\ H_4)N_2} + \dfrac{(F_x{}^2+F_y{}^2)^{1/2}}{N_2}}{D_8T_7}$$



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

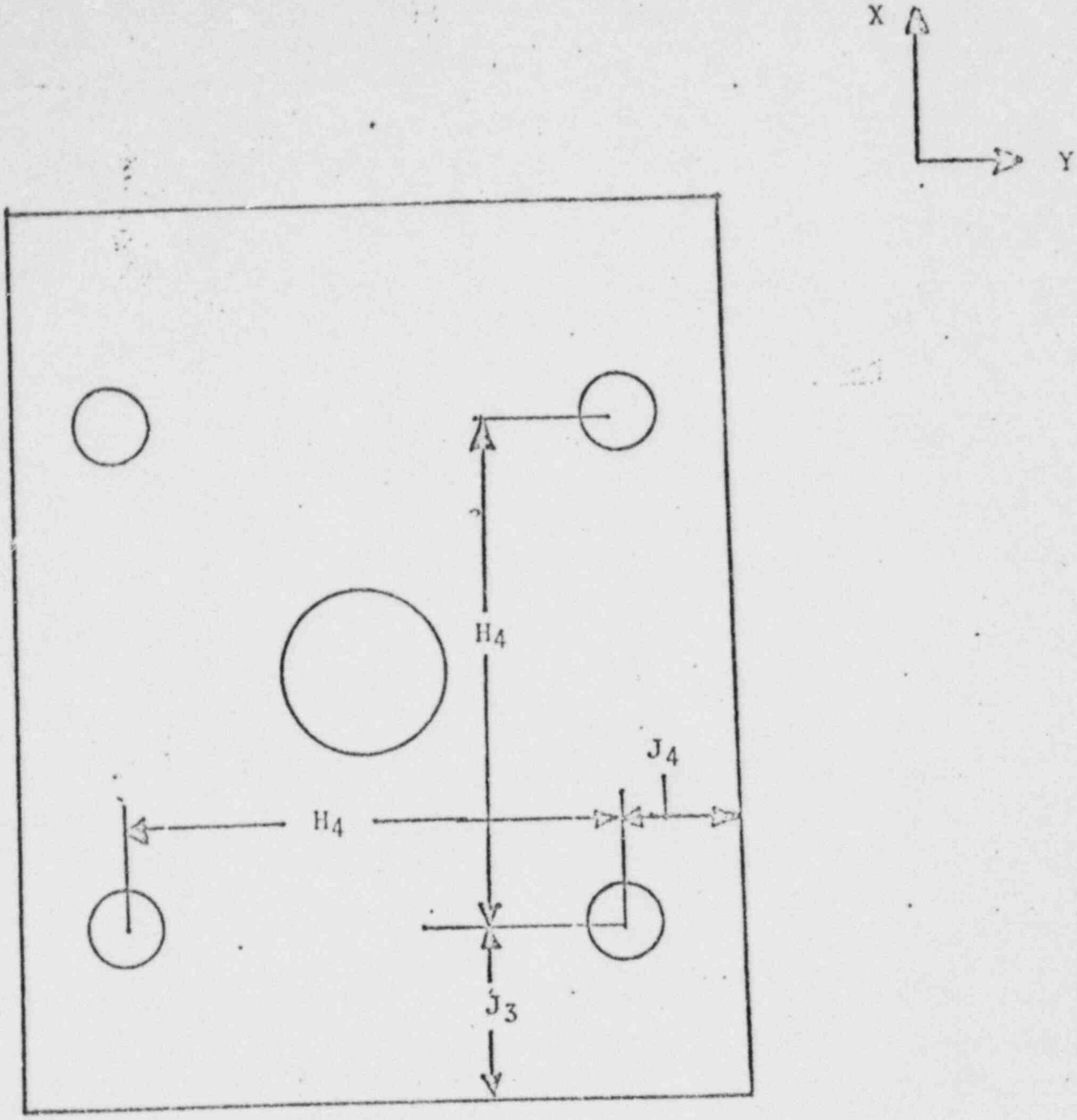
The bonnet stresses are calculated with the assumption that loading is through the bolt connections as previously defined.

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

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



BONNET BOLT PATTERN

Figure 9

Operator Mounting Analysis

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

Where

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

$$S(52) = \frac{T}{K_0}\,C_0$$ = 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

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

Where

$S(54) = S(56) + S(57)$ = Total tensile stress, psi

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

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

$S(55) = S(58) + S(59)$ = Total shear stress

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

Operator Mounting Analysis

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

Top Bonnet Weld

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

Where

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

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

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

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

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

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

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:

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

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:

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

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

GENERAL ARRANGEMENT AND
CROSS-SECTION DRAWINGS

Core Performance Branch Meeeting 9/2/81

1. FPL commits to implementing the following rod bowing penalties:

Burnup (gigawatt-days per metric ton of uranium	Departure from Nucleate Boiling Ratio Penalty (percent)
0-2.4	0
2.4-5	3.0
5-10	7.1
10-15	10.3
15-20	12.9
20-25	15.3
25-30	17.4
30-35	19.4
35-40	21.2

These penalties will be implemented in the technical specifications and the tech spec bases will be explained accordingly.

These penalties were applied becasue it is expected that the St. Lucie 2 fuel will experience rod bowing equal to that predicted for SONGS 2 & 3. This is because of the similarity of spacer grid span lengths and fuel rod cladding dimensions.

2. FPL submits the Supplemental ECCS Analysis (NUREG-0630) attached.

3. FPL will submit best available analysis for seismic plus LOCA loads on the fuel by September 1981: Results using approved methods will be submitted by May 1982. This approach follows that suggested in NUREG-0609 and is the same approach as that taken for the WSES-3 Safety Evaluation Report.

4. FPL agrees with NRC that the following items should be included in a confirmatory Safety Evaluation Report item entitled, "Miscellaneous Fuel Design Documentation".

   (1) Neutron source rod and incore instrumentation assembly documentation.

   (2) CEA fretting and axial growth documentation.

   (3) Mechanical fracturing documentation.

   (4) Documentation for stress analyses for fuel rods, poison rods, and CEAs; strain analyses for fuel assemblies, fuel rods, poison rods, and CEAs; strain fatigue analyses for fuel assemblies and fuel rods; and poison rod internal presure.

   (5) Documentation for non-LOCA transient core coolability.

FPL will submit documentation for items (1), (2), (3), and (5), September 11, 1981. If documentation for item (4) cannot be provided by that date, a schedule emphasizing the earliest possible submittal will be provided.

5. FPL will provide a revised response to question 492.1 to adequately identify the methodology used to assure the thermal hydraulic design of future reloads are bounded by the existing safety analysis. See revised response attached to these minutes.

6. FPL will implement in the technical specifications a 1% penalty for grid spacing differences relative to experimental data supporting the CE-1 DNB correlation. This penalty by itself would result in a change in the 95/95 NBR from 1.19 to 1.20. The setpoint analysis supporting the technical specifications will be performed such that the safety analyses remain valid.

7. FPL will provide a revised response to question 492.13 to indicate reactor trip on low coolant flow, based on steam generator ΔP measurements. The revised response is attached.

Question No.

492.1 "Standard format and content of Safety Analysis Reports, Regulatory Guide 1.70, states that in Chapter 4 of the SAR "... the applicant should provide an evaluation and supporting information to establish the capability of the reactor to perform its safety functions throughout its design lifetime under all normal operation modes..."

Are the analyses presented in Section 4.4 representative of the initial core only or have future cycles been analyzed? Provide a discussion of how power distributions for future cycles are considered in FSAR analyses. Is there any assurance that St. Lucie 2 can operate at the licensed power level without excessive DNB trips throughout future cycles? Will revisions to the design methodology be required in order to maintain sufficient thermal margin?

Response

The St. Lucie Unit 2 FSAR documents the ability of the core design to meet performance and safety requirements for the expected plant lifetime to the extent possible, based on information available prior to actual operation. Radial power distribution predictions as a function of burnup for the first three cycles are shown in Figures 4.3-2a through 4.3-24. The maximum radial peaking factor in the DNB analyses of Section 4.4 is 1.55 which is at least 5% higher than the predictions reported in Section 4.3.

Also, the predicted CEA worths at hot full power and hot zero power conditions are, respectively 10.2% and 7.57% (Table 4.3-7). Corresponding valves used in safety analyses, assuming a stuck CEA, are less than 5.5% and 2.5%, respectively, for transients other than steam appropriate to the end of cycle 4 at hot full power and hot zero power conditions assuming a stuck CEA are 6.68% and 5.00% respectively (Section 15.0.3.2.3).

The minimum allowable reactor core flow rate is 369,947 gpm (Table 4.4-1). This flow rate is assured over plant lifetime by periodic measurements required by technical specifications. The St. Lucie Unit 2 technical specification will be similar to St. Lucie Unit 1 Technical Specification 3.2.5. This technical specification will require periodic (18-month) measurements to verify availablity of the flow rate assumed in the safety analyses and the core protection calculator system.

The complete set of Technical Specifications for St. Lucie Unit 2 will be based on the present plant design. Subsequent to plant startup and operation, core reload designs are evaluated based on the present plant design. Subsequent to plant startup and operation, core reload designs are evaluated based on operating data and specific reload core parameters. This evaluation includes thermal margin analyses and an assessment of the validity of the FASR safety analyses. It is possible the evaluation could lead to the implementation of revised analysis methodology (however, no such revisions are presently known). The technical specifications would than revised as necessary to reflect the current fuel cycle.

492.13 Provide a description of the instrumentation available and the surveillance requirements and procedures which would alert the reactor operator to an abnormal core flow or core pressure drop during steady-state operation.

We will require that the plant Technical Specifications include the requirements that the actual reactor coolant system total flowrate be greater than or equal to the value indicated by the core protection calculator system.

Response

St. Lucie II will have a Technical Specification similar to St. Lucie I Technical Specification 3.2.5, This Technical Specification will assure that reactor coolant flow rate is consistent with that assumed a) in the transient and accident analysis, and b) in the core protection calculator system. As stated in the Bases of the St. Lucie Unit I Technical Specifications: "The 18 month periodic measurement of the RCS total flow rate is adequate to detect flow degradation and ensure correlation of the flow indication channels with measured flow such that the indicated percent flow will provide sufficient verification of flow rate on a 12 hour basis."

As described in Section 7.2.1.1.2.3, the reactor coolant flow measurement signals are provided by summing the square root of the differential pressure differential pressure across each steam generator to provide indication of the total coolant flow through the reactor. This measurement of differential pressure (ΔP) is directly proportional to the actual flow. The low flow reactor trip is actuated directly by the summed ΔP signals.

No FSAR change is required.

Supplemental ECCS Analysis (NUREG-0630)

A supplemental analysis utilizing the material models of NUREG-0630 (Reference 1) has been performed. This supplemental analysis also utilized the heat transfer portion of C-E's alternate ECCS Evaluation Model which is described in Reference 2. The combination of the NUREG-0630 material models and the alternate heat transfer model provides results which are less limiting than the results in the St. Lucie Unit 2 FSAR which were obtained by using C-E's NRC-approved ECCS Evaluation Model (Reference 3).

For this analysis, the peak clad temperature decreased by 128°F and the peak local clad oxidation decreased by 10.62% from the corresponding values reported in Section 6.3.3 of the St. Lucie Unit 2 FSAR. Similar results were provided to NRC, in Reference 4 for another Combustion Engineering designed PWR. As in this previous analysis, this analysis illustrates the overall conservatism of the C-E flow blockage representation in its NRC approved ECCS Evaluation Model (Reference 3).

METHOD OF ANALYSIS

The analysis used the three material models of NUREG-0630. Specifically, the models predict cladding rupture temperature, cladding burst strain and fuel assembly flow blockage. In addition, the analysis utilized the heat transfer portion of the alternate ECCS Model for the calculation of steam heat transfer coefficients for locations at and above the blockage plane. All other portions of the calculation used C-E's NRC-approved ECCS Evaluation Model.

Figures 3, 8, and 16 in Reference 1 present the NRC recommended rupture temperature, rupture strain and reduction in fuel assembly flow area respectively. This analysis assumed a heating ramp rate of 0°C/sec. and utilized the appropriate values from these three figures. The 0°C/sec. heating ramp rate predicts the earliest rupture and the maximum burst strain and maximum flow area reduction. Although this introduces additional, unnecessary conservatism into the analysis, it was done to remain consistent with the previous analysis performed (Reference 4) and to expedite a response to this NRC Question.

Since clad rupture occured during reflood, the blowdown hydraulic transient is not sensitive to, and will not be effected by flow blockage modeling. Furthermore, calculation of reflood rates is based on the core average behavior and is not affected by local blockage. Therefore, the blowdown and reflood hydraulics calculated for the FSAR analysis remain applicable and were used in this study. The hot rod clad temperature and oxidation values were recalculated using the NUREG-0630 clad material models and the alternate steam cooling heat transfer models. Other input assumptions remain the same as described in Section 6.3.3 of the FSAR. The calculation described above was performed for the 1.0 DEG/PD* break, which is the limiting large break.

RESULTS

Table 1 summarizes the significant input parameters and results of this supplemental analysis. The calculated rupture strain is 90%, which corresponds to a flow blockage of 71%. These are the maximum values predicted by the NUREG-0630 models. As mentioned earlier, rupture is predicted during reflood. The rupture temperature of 1515°F is based on the 0°C/sec heating ramp rate curve. Use of a more representative heating ramp rate would calculate rupture at a higher temperature. The higher rupture temperature would result in a lower rupture strain and lower flow blockage than the maximum value calculated here.

As concluded previously in Reference 4, the combination of the improved heat transfer of C-E's alternate model with higher strain and flow blockage actually results in a significant decrease in both calculated peak clad temperature and peak clad oxidation. For this analysis, the peak clad temperature decreased by 128°F and the peak clad oxidation decreased by 10.62% from the corresponding results of the FSAR analysis presented in Section 6.3.3 and presented graphically in Figures 1 and 2.

The results of this study demonstrate that the ECCS analysis results presented in the St. Lucie Unit 2 FSAR comply with the acceptance criteria of 10CFR50.46 at an assumed peak linear heat rate of 13.0 kw/ft.

---

*DEG/PD - Double-Ended Guillotine/Pump Discharge

REFERENCES

1. D. A. Powers and R. O. Meyer, "Cladding Swelling and Rupture Models for LOCA Analysis . NRC Report NUREG-0630, April 1980.

2. Enclosure 1-P of Letter LD-78-069, from A. E. Scherer, C-E, to Dr. Denwood F. Ross, NRC, dated September 18, 1978.

3. "Calculative Methods for the C-E Large Break LOCA Evaluation Model", CENPD-132, August 1974 (Proprietary).

   "Updated Calculative Methods for the C-E Large Break LOCA Evaluation Model," CENPD-132, Supplement 1, August 1974 (Proprietary).

   "Calculational Methods for the C-E Large Break LOCA Evaluation Model", CENPD-132 , Supplement 2, July 1974 (Proprietary).

4. Response to NRC Question 231.34 San Onofre Nuclear Generating System Units 2 & 3, Final Safety Analysis Report.

TABLE I

I. INPUT PARAMETERS AND RESULTS OF THE ECCS SUPPLEMENTAL ANALYSIS

PARAMETER	SUPPLEMENTAL ANALYSIS
Rupture Strain Model	NUREG-0630 Models (1)
Steam Cooling Heat Transfer Model	C-E's Alternate Model (2)
Model for Remainder of Calculation	"Calculation Method for the C-E Large Break Evaluation Model"
Allowable Peak Linear Heat Generation Rate (kw/ft)	13.0
Rupture Strain (%)	90
Flow Blockage (%)	71
Hoop Stress at Rupture (KPSI)	5.63
Clad Temperature at Rupture (°F)	1515
Rupture Time	During Reflood

II. Comparison of Supplemental and FSAR Analysis Results

PARAMETER	SUPPLEMENTAL ANALYSIS	FSAR ANALYSIS
Peak Clad Temperature (°F)	1972	2093
Location	ABOVE BLOCKAGE	AT BLOCKAGE
Peak Local Clad Oxidation (%)	4.69	15.76
Location	ABOVE BLOCKAGE	AT BLOCKAGE

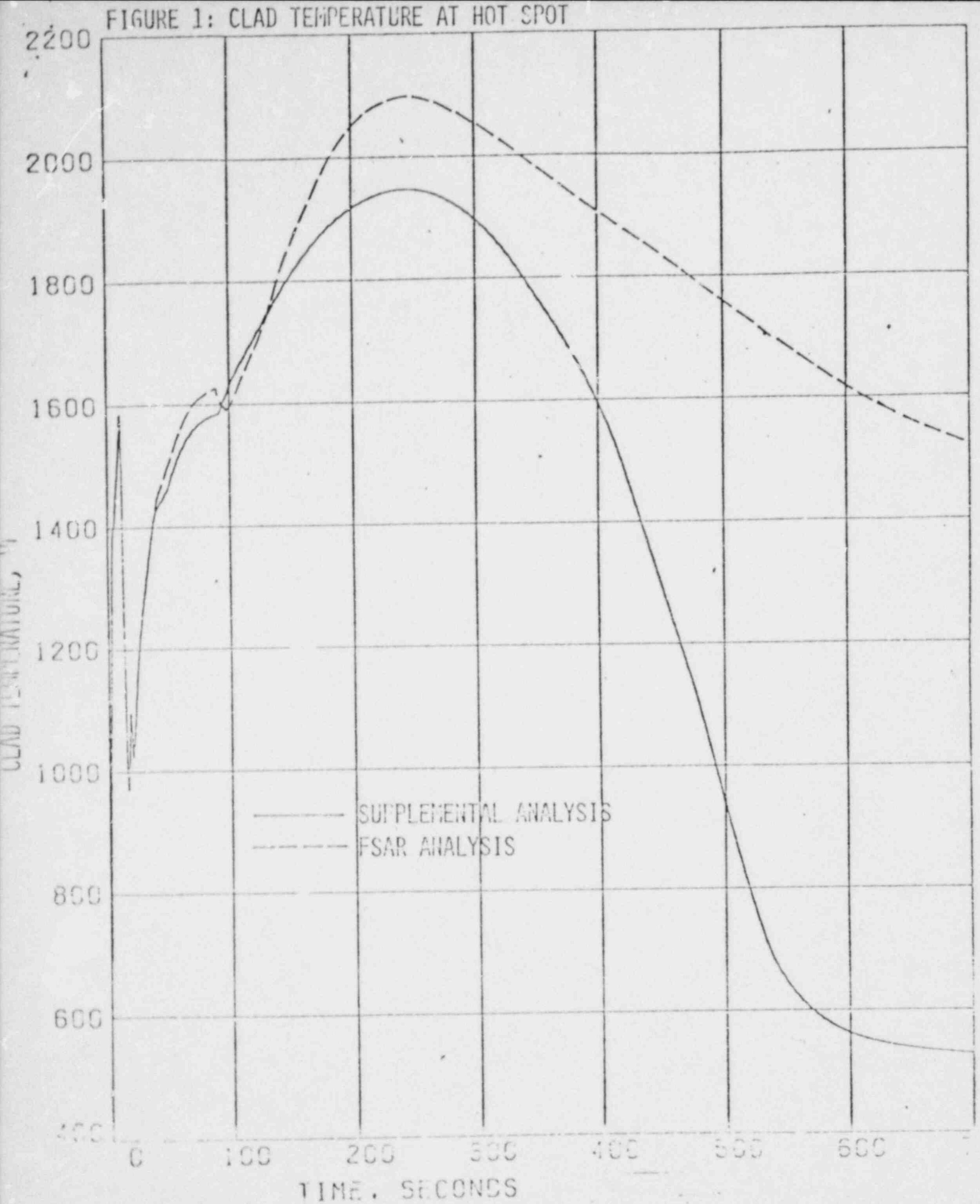


FIGURE 1: CLAD TEMPERATURE AT HOT SPOT
2200
2000
1800
1600
1400
1200
1000
800
600
0
100
200
300
400
500
600
TIME, SECONDS
SUPPLEMENTAL ANALYSIS
FSAR ANALYSIS

FIGURE 2: PEAK LOCAL CLAD OXIDATION