57 57

Nuclear Valve Division
Borg-Warner Corporation
7500 Tyrone Ave., Van Nuys, California 91409

BORG WARNER Energy Equipment

REPORT NO. NSR 401JBB3-4
PAGE 1 of 53
DATE 7 FEB 79
REVISION A
DATE 9 APR 79

APPROVED
DUKE POWER CO.
DATE: JUL 13 1979
S. K. BLACKLEY, JR.
CHIEF ENGINEER
By: MECHANICAL DIVISION

DOCUMENT CONTROL DATE
JUN 22 1979
DUKE POWER COMPANY
DESIGN ENGINEERING

Nuclear Safety Related

DIV	STATUS	INIT.	DATE
CIVIL			
MECH	AA	CDB	6-19-79
ELECT			

SEISMIC ANALYSIS OF 4 INCH, 150 LB CARBON STEEL MOTOR OPERATOR GATE VALVE

NOT SUITABLE FOR MICROFILM

FOR
DUKE POWER COMPANY
CATAWBA NUCLEAR STATION
UNITS 1 AND 2

BY

NUCLEAR VALVE DIVISION
BORG-WARNER CORPORATION
7500 Tyrone Avenue
Van Nuys, California 91409

CNM 1205.00-0669

Nuclear Valve Division

BORG WARNER Energy Equipment

REPORT NO. NSR401JBB3-004

PAGE 1.1

REVISION A

DATE 9 APR 79

CHANGE RECORD SHEET

REVISIONS						
DATE	PAGES AFFECTED			DESCRIPTION	APPROVED BY	REV LTR
	REVISED	ADDED	REMOVED			
APR 9 1979		1.1		Added Change Record Sheet	[signature]	A
	1			Revised title page changed from N/C to Rev. A		A
		Attach 1		Added Attachment 1 Static Deflection test results	[signature]	A

Prepared by: Steven M. Brown
S. M. Brown
Engineering Test Lab Supervisor

Approved by: Liwei Chen 040979
Liwei Chen
Project Engineer

The following seismic analysis report has been prepared in accordance with the design requirements prescribed in Duke Power Company, Catawba Nuclear Station, Units 1 and 2, Specification No. CNS-1205.00-6, Addendum 6, December, 1978, ASME Section III Carbon Steel Gate, Globe and Check Valve.

VALVE IDENTIFICATION

BORG-WARNER CO. N.V.D. PART NO	DUKE POWER CO. ITEM NO.	MILL POWER SUPPLY CO. LIST NO.
401JBB3-4	02B-400	CN-0150-12

Prepared by: Shou-jen Wang 2-7-79
Shou-Jen Wang
Analytical Engineer

Reviewed by: Liwei Chen 020779
Liwei Chen
Project Engineer

Approved by: RE Holder
R. E. Holder
Manager, Advanced Design

Nuclear Valve Division
Borg-Warner Corporation
7500 Tyrone Ave., Van Nuys, California 91409

TABLE OF CONTENTS

NOMENCLATURE

A	=	area (in^2)
a	=	angle (degree)
b	=	angle (degree)
C	=	pull (kip)
d	=	diameter of a circle (in)
d	=	deflection (10^{-4} in)
E	=	modulus of elasticity (ksi)
F	=	thrust (kip)
F	=	flexibility (in/lb)
f	=	natural frequency (Hz)
g	=	acceleration (in/sec^2)
I	=	moment of inertia (in^4)
J	=	polar moment of inertia (in^4)
L	=	length (in)
M	=	moment (in-kip)
M	=	mass (lb)
m	=	weight (lb)
m	=	reciprocal of Poisson's ratio
P	=	pressure (ksi)
PI	=	3.14159265, ratio of circumference to diameter of a circle
R	=	reaction force (kip)
r	=	radius of circle (in)
S	=	shear force (kip)

s = stress (ksi)

SLF = seismic load factor

SUM = summation

T = torque (ft-lb, or in-kip)

t = thickness (in)

v = mode shape

w = natural frequency (radian per sec.)

x,y,z = space coordinates in cartesian system

x = distance (in)

y = distance (in)

Z = section modulus (in^3)

[F] = flexibility matrix

[I] = unit matrix

[M] = mass matrix

[v] = mode shape

superscript:

T = transpose of a matrix

subscript:

a = axial

b = bending, bolt

i = inside, internal

i = row index of matrix

j = column index of matrix

m = meridian

max = maximal value

o = outside

p = piping

r = radial

s = shear

t = tangential

y = yoke

1,2,3 = sectional conditions

1. INTRODUCTION:

The seismic and operability analysis report of the following valves has been performed for the designs detailed in Nuclear Valve Division of Borg-Warner Corporation drawing numbers, as shown. Design specifications are prescribed in Duke Power Company, Catawba Station, Units 1 and 2, Specification No. CNS-1205.00-6, Addendum 6, December, 1978, ASME Section III Carbon Steel Gate, Globe and Check Valve.

The valve assembly is built to the criteria of ASME, B & PV Code, Section III, Class 2 or 3 Nuclear Valves and Duke Power Company Safety Class B or C.

VALVE SIZE (INCH)	PRESSURE RATING (LB)	MATERIAL	VALVE TYPE	OPERATOR	N.V.D. ASSEMBLY DWG. NO.
4	150	CARBON STEEL	MOTOR	GATE	401JBB3-4

In accordance with the design requirements, the valve(s) and appurtenance(s) shall be qualified by the procedures and guidelines of the Duke Power Company, Seismic Design Manual. Basically, two modes of operation are considered: Upset Mode and Faulted Mode. For Faulted Mode (Safe Shutdown Earthquake), a seismic load factor (SLF) of 3.0 g shall be applied in each of two orthogonal horizontal directions in combination with a SLF of 2.0 g in the vertical direction, all action simultaneously. The Upset Mode (Operational Basic Earthquake) is similar to Faulted Mode, except that the SLF values shall be taken as 8/15 of the respective values of the Safe Shutdown Earthquake.

Seismic Design Manual, Section 4.2 outlines a procedure for qualifying rigid system by the "equivalent static analysis" method . The method consists of performing a static structural analysis of the equipment under equivalent static forces conservatively representing the actual dynamic loadings. Seismic forces on each component of the equipment are obtained by concentrating its mass at its center of gravity and multiplying by the appropriate seismic load factors (SLF). Rigid systems are defined as systems which have no natural frequency less than 33 cycles per second.

All the values used within this analysis are the actual dimensions taken from the detail prints. In all cases, the values are greater

1. INTRODUCTION:

than respective d_m and t_m values required by ASME Boiler & Pressure Vessel Code, Table NC 3511-1.

Standard engineering practice shall be used to determine the maximum stress conditions in all portions of the equipment. It shall be demonstrated that the maximum stresses meet the acceptance criteria for the selected valve assembly materials defined in Section 6.0 of the Seismic Design Manual.

NVD has used the 100°F pressure rating of the applicable valve pressure class along with faulted mode loading when calculating stress levels. As a conservative approach, upset allowable stress limits are compared to faulted loading, thus substantiating the design under all faulted and upset conditions.

The ASME Class 2 valve design criteria is based upon the rules of ASME, B & PV Code, NC-3200 and standard engineering practices. The allowable stress limits of 1.0 s_m and 1.5 s_m shall be taken for the primary membrane (P_M) and (local) primary membrane plus primary bending ((P_M or P_L) + P_B) stress categories, respectively, for pressure boundry components . The design stress intensity values s_m for Class 1 components are cited in Table I-1.1 through I-1.3, ASME B & PV Code. For non-pressure boundary components, the stress limits are taken as 0.6 s_y (AISC allowable working stress limit) for upset mode and 0.9 s_y for emergency and faulted modes.

An idealized structure system shall be modeled to simulate the vibratory mode of the valve assembly. The calculated minimum natural frequency of vibration shall be examined to satisfy the specification limit of 33 cycles per second.

2. SUMMARY:

ASSEMBLY DRAWING NO.

DESCRIPTION OF VALVE SECTION	MATERIAL SPECIFICATION	ALLOWABLE UPSET MODE (KSI)	CALCULATED FAULTED MODE (KSI)
BODY, MAIN RUN	SA182 F316	27.00	14.01
BODY, NECK	SA182 F316	27.00	3.50
YOKE, LEGS	SA182 F316	18.00	7.04
BOLT, BONNET	A564 TY 630	115.00	22.39
FLANGE, BONNET	SA182 F316	27.00	17.82

DESCRIPTION OF VALVE SECTION	REQUIRED MINIMUM FREQUENCY (CYCLE/SEC)	CALCULATED NATURAL FREQUENCY (CYCLE/SEC)
VALVE ASSEMBLY	33.00	42.52

3. VALVE ASSEMBLY:

The valve assembly has the general configuration which can be represented in a simplified sketch as shown in Figure 3-1. The approximate center of mass for each component is referenced with respect to the valve body main run. An evaluation of the structural integrity is made by reviewing the state of stress at critical sections as designated in Figure 3-1.

A conservative estimate of the component dead load weight and basic physical dimensions is obtained from the drawings and tabulated in Table 3-1.

As required in the seismic design specification, an evaluation shall consider two modes of operation - upset mode and faulted mode. In this stress analysis, a conservative approach is taken by qualifying the structure to the maximum design acceleration and using the acceptance limits for the upset mode. This maximum acceleration is based upon the seismic load factors for the faulted mode, and expressed as

$$g_{max} = [\ SLF_H + SLF_H + (1.0\ g + SLF_V\ )]$$

$$g_{max} = (\ 3.0\ g + 3.0\ g + 3.0\ g\ )\ = 5.20\ g$$

Since the structure may assume any arbitrary position, the maximum design acceleration shall be applied simultaneously along the orthogonal axes of the valve assembly.

In addition, the valve shall consider the maximum operational stem thrust (F_{st}) and torque (T_q) imposed by the operator.

$$T_q = (F_{st})(\text{stem factor})$$

where stem factor is determined basically in accordance with the stem thread dimension assuming the coefficient of friction being 0.15. For the case of manual operated valve, a 150 lb (max.) operator pull(C) is considered to be applied at the rim of the handwheel.

3. VALVE ASSEMBLY:

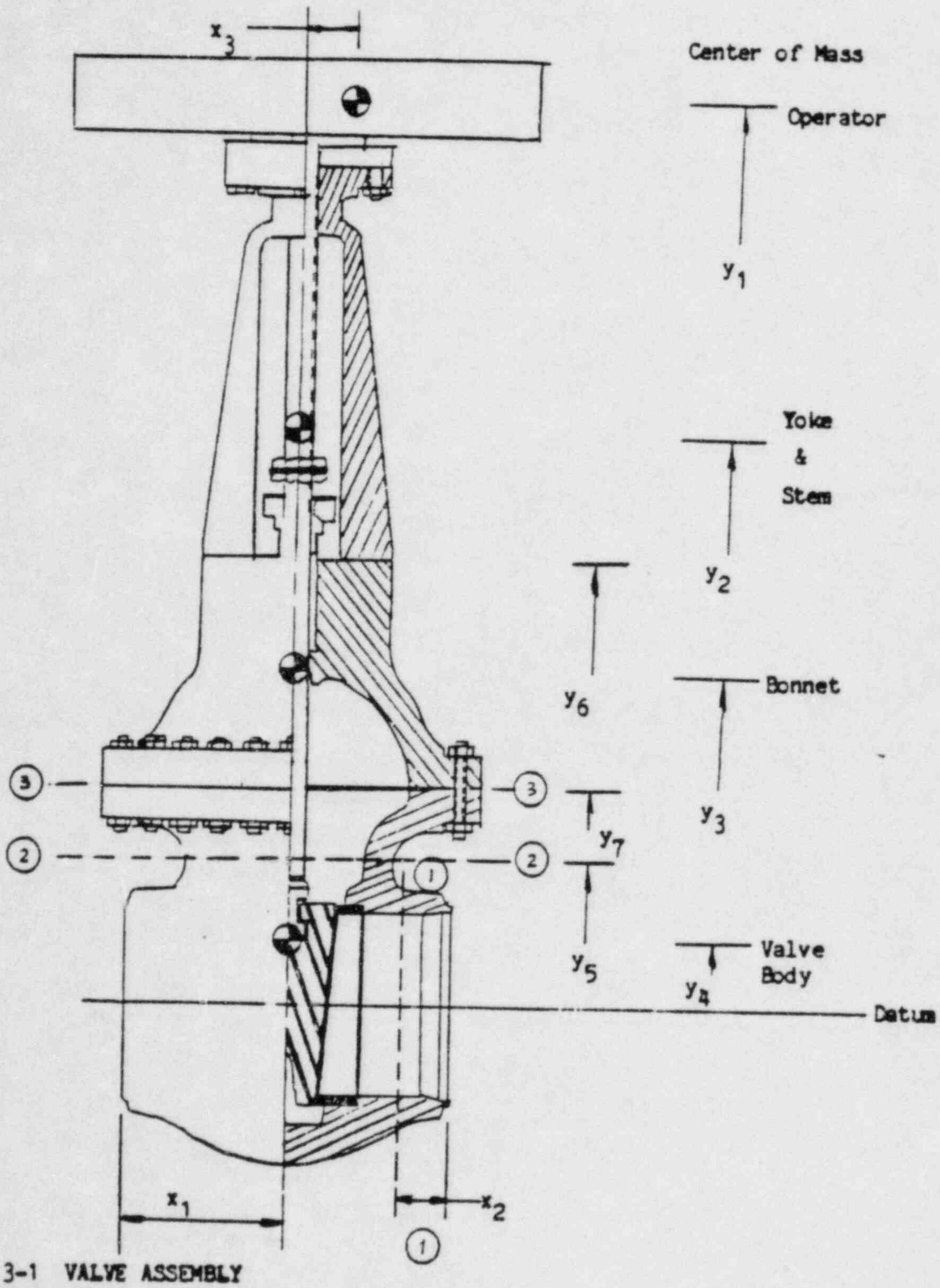


FIGURE 3-1 VALVE ASSEMBLY

3. VALVE ASSEMBLY:

TABLE 3-1 COMPONENT LOADS AND PHYSICAL DIMENSIONS

m_1	(lb)	206.000
m_2	(lb)	50.000
m_3	(lb)	38.000
m_4	(lb)	95.000
m_5	(lb)	8.000
m_{total}	(lb)	397.000
y_1	(in)	30.100
y_2	(in)	14.750
y_3	(in)	7.600
y_4	(in)	1.500
y_5	(in)	3.000
y_6	(in)	11.850
y_7	(in)	6.000
x_1	(in)	6.000
x_2	(in)	1.500
x_3	(in)	.750
z_3	(in)	2.813
T_q	(ft-lb)	19.200
F_{st}	(kip)	2.250
C	(kip)	0.000

3. VALVE ASSEMBLY:

3.1 Valve Body:

An evaluation of the maximum body stresses shall be made for the zones denoted on the preceding valve body sketch. The valve body cross sections are located in the main run and neck.

3.1.1 Loading:

A measure of the maximum force resultants at each zone is obtained by applying the maximum accelerations g_{max} along the orthogonal axes, simultaneously.

3.1.1.1 Plane 1-1: (Valve Body - main run)

Assuming simple support conditions at the weld ends as shown in Figure 3-2.1, the force resultants at the crotch region can be derived as follows:

$$M_{max} = T_{max} = \text{net external moment and torque}$$

$$= (m_1y_1+m_2y_2+m_3y_3+m_4y_4$$

$$+(m_1x_3,\ m_1z_3)_{max})\ g_{max}$$

$$R = \text{simple support reaction}$$

$$= M_{max}/2x_1 + (m_1+m_2+m_3+m_4+m_5)g_{max}/2$$

$$F_1 = \text{thrust} = (m_1+m_2+m_3+m_4+m_5)g_{max}/2$$

$$S_1 = \text{shear} = R + C$$

$$M_1 = \text{moment} = Rx_2$$

$$T_1 = \text{torque} = T_{max}/2$$

The numerical values are tabulated in Table 3-2.1.

3. VALVE ASSEMBLY:

3.1.1.2 Plane 2-2: (Valve Body - neck)

At plane 2-2, the force resultants, determined by assuming the neck weight is one third of the total valve body weight (Figure 3-2.2), and can be expressed as:

$$F_2 = \text{thrust} = F_{st} + (m_1+m_2+m_3+m_4/3 + m_5)\ g_{max}$$

$$S_2 = \text{shear} = (m_1+m_2+m_3+m_4/3)\ g_{max} + C$$

$$M_2 = \text{moment} = [m_1(y_1-y_5)+m_2(y_2-y_5)+m_3(y_3-y_5) + (1/3)m_4(y_3-y_5)]\ g_{max}$$

$$T_2 = \text{torque} = m_1(x_3,\ z_3)_{max} g_{max}$$

The numerical values are tabulated in Table 3-2.2.

3. VALVE ASSEMBLY:

3.1.1.3 Plane 3-3:

The force resultants at plane 3-3 (see Figure 3-2.3) are derived as follows:

$$F_3 = \text{thrust} = F_{st} + (m_1+m_2+m_3+m_5)g_{max}$$

$$S_3 = \text{shear} = (m_1+m_2+m_3)\ g_{max} + C$$

$$M_3 = \text{moment} = [m_1(y_1-y_7)+m_2(y_2-y_7)$$
$$+m_3(y_3-y_7)]\ g_{max}$$

$$T_3 = \text{torque} = m_1(x_3,\ z_3)_{max}g_{max}$$

The numerical values are tabulated in Table 3-2.3.

3. VALVE ASSEMBLY:

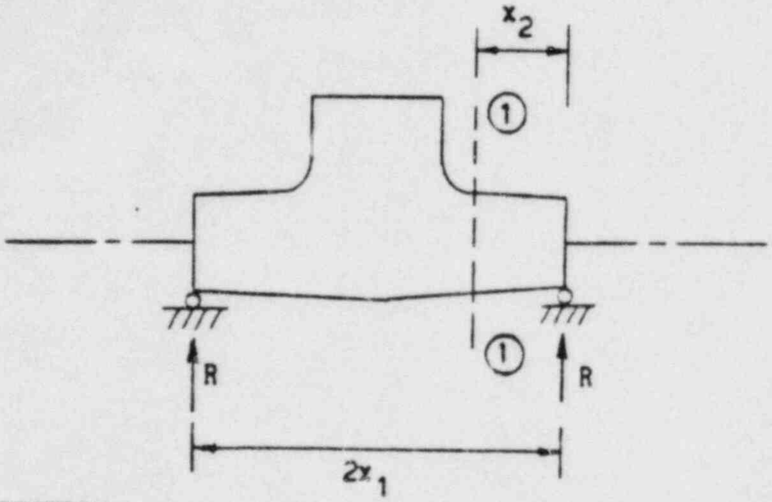


FIGURE 3-2.1 MAIN RUN SECTION

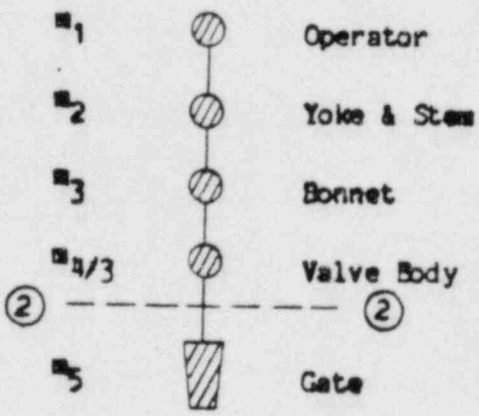


FIGURE 3-2.2 NECK SECTION

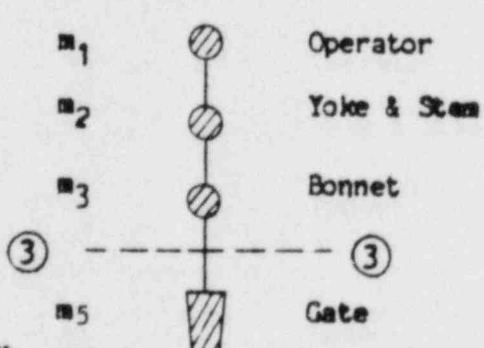


FIGURE 3-2.3 THREAD RELIEF SECTION

3. VALVE ASSEMBLY:

TABLE 3-2.1 SEISMIC LOAD RESULTANTS AT PLANE 1-1

F_1	(kip)	1.032
S_1	(kip)	4.477
M_1	(in-kip)	6.715
T_1	(in-kip)	20.667

TABLE 3-2.2 SEISMIC LOAD RESULTANTS AT PLANE 2-2

F_2	(kip)	3.985
S_2	(kip)	1.693
M_2	(in-kip)	33.751
T_2	(in-kip)	3.013

TABLE 3-2.3 SEISMIC LOAD RESULTANTS AT PLANE 3-3

F_3	(kip)	3.820
S_3	(kip)	1.529
M_3	(in-kip)	28.407
T_3	(in-kip)	3.013

3. VALVE ASSEMBLY:

3.1.2 Cross Sectional Properties:

For the critical zones on the valve body, the cross section area and inertia properties are computed as follows, (Figure 3-3)

$$A = \text{Area} = [(r_o)^2 - (r_i)^2](PI)$$

$$I = I_{max} = I_{min} = [(r_o)^4 - (r_i)^4](PI/4)$$

$$J = 2\ I_{min}$$

where r_o = outside radius

r_i = inside radius

All the numerical values are tabulated in Table 3-3.

3.1.3 Stresses:

3.1.3.1 Due to Seismic Loads:

Seismic stresses are derived for the valve body, by using the following expressions

$$s_a = \text{axial stress} = F/A + M\ r_o/I$$

$$s_s = \text{shear stress} = S/A + T\ r_o/J$$

where F,S,M,T are force resultants (see Section 3.1.1, Table 3-2)

and A,I,J are cross section properties (see Section 3.1.2, Table 3-3).

All the numerical values are tabulated in Table 3-4.

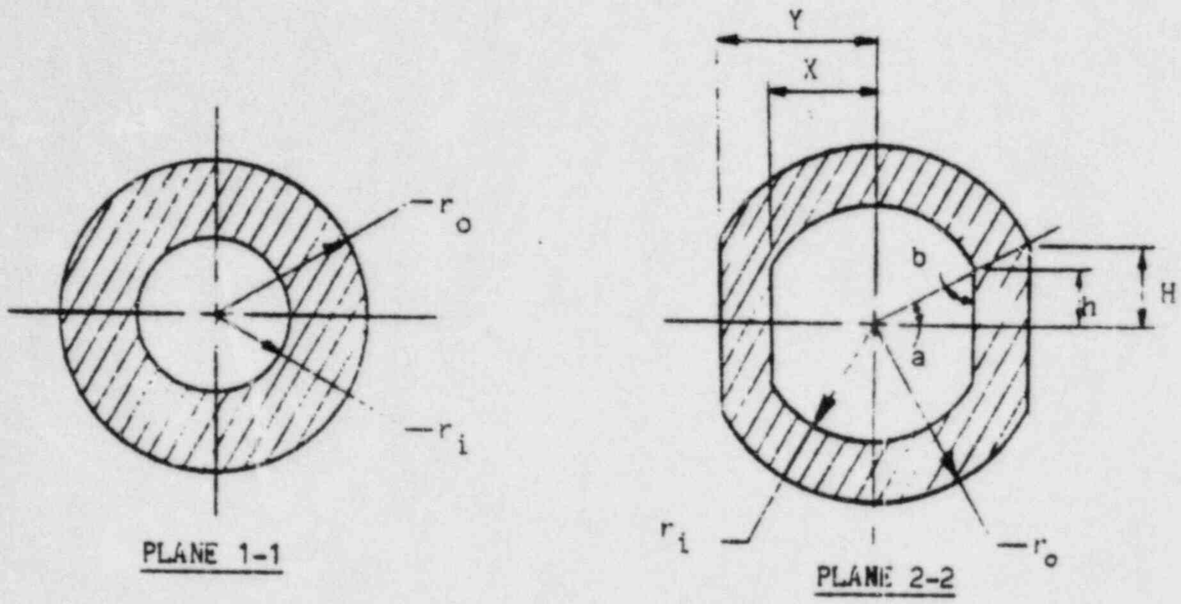


FIGURE 3-3 CROSS SECTIONS OF VALVE BODY

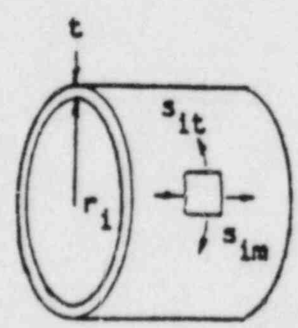


FIGURE 3-4.1 MEMBRANE STRESS AT PLANE 1-1

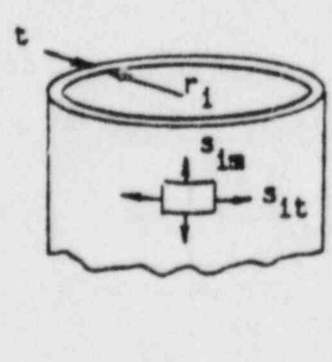


FIGURE 3-4.2 MEMBRANE STRESS AT PLANE 2-2

3. VALVE ASSEMBLY:

TABLE 3-3.1 CROSS SECTIONAL PROPERTIES AT PLANE 1-1

r_o (in)	2.560
r_i (in)	1.990
I_o (in^4)	21.416
A_o (in^2)	8.148
J_o (in^4)	42.831

TABLE 3-3.2 CROSS SECTIONAL PROPERTIES AT PLANE 2-2

r_o (in)	3.640
r_i (in)	2.810
I_o (in^4)	88.910
A_o (in^2)	16.819
J_o (in^4)	177.820

3. VALVE ASSEMBLY:

TABLE 3-4.1 SEISMIC STRESSES AT PLANE 1-1

s_a (ksi)	.929
s_s (ksi)	1.785

TABLE 3-4.2 SEISMIC STRESSES AT PLANE 2-2

s_a (ksi)	1.619
s_s (ksi)	.162

3. VALVE ASSEMBLY:

3.1.3.2 Due to Internal Pressure, Piping Rection and Thermal Effect:

Stress due to internal pressure, and piping reaction , thermal effect and other sustained or occasional loads on each plane can be obtained as follows

3.1.3.2.1 Plane 1-1: (Valve Body - main run)

A. Internal Pressure:

The membrane stresses due to internal pressure in the meridian, tangent and radial directions are calculated by following three equations respectively:

$s_{im} = P_s\, r_i/(2t)$

$s_{it} = P_s\, r_i/t$

$s_{ir} = -\, P_s/2$

where r_i = inside radius (see Table 3-3)

t = wall thickness = $r_o - r_i$ (see Table 3-3)

P_s = standard calculated pressure (or design pressure)

B. Piping Reaction:

The moment applied simultaneously in Plane 1-1 due to piping reaction is determined by:

$M_p = Z_p\, s$ (Bending moment)

$T_p = 1.2\, Z_p\, s$ (Torsional moment)

The stress (ksi) in meridian direction and shear stress due to piping reaction will be:

3. VALVE ASSEMBLY:

$s_p = M_p r_o/I$

$s_s = T_p r_o/J$

where $Z_p = I_p/ r_o p$
= section modulus of the pipe

and s is yield strength of the pipe .

C. Thermal Effect:

The thermal secondary stresses due to transient effect shall be calculated on the basis of a continuous ramp change in fluid temperature at 100°F/hour. Per ASME Code, NB-3545.2, the thermal secondary stress s_T can be determined by:

$s_T = s_{t1} + s_{t2}$

where s_{t1} = stress component resulting from wall temp. gradient (ASME III, Fig. NB-3545.2 (c)-2)

s_{t2} = stress component resulting from wall thickness variation
= $K_2 K_1 dT$

K_2 = product of Young's modulus and linear thermal expansion coefficient

K_1 = stress index due to structural discontinuity (ASME III, Fig. NB-3545.2(c)-3)

dT = maximum temperature difference between thick and thin walls (ASME III, Fig. NB-3545.2(c)-4)

For those valves with higher temperature change rates than 100°F/hour, the thermal secondary stress will be determined per NB-3554, where the value of dT in the above formula is obtained as the product of C_4 and

3. VALVE ASSEMBLY:

$dT_{fl(max)}$ which are the maximum magnitude of the difference in average wall temperatures and the maximum specified step change in fluid temperature respectively (see NB-3534).

D. Stress Intensities:

Stress intensities can then be calculated as:

$$s_m = s_a + s_{im} + s_p + s_T$$

$$s_t = s_{it} + s_T$$

$$s_1 = (s_m + s_t)/2 + [(s_m - s_t)^2/4 + s_s{}^2]^{1/2}$$

$$s_2 = (s_m + s_t)/2 - [(s_m - s_t)^2/4 + s_s{}^2]^{1/2}$$

$$s_3 = s_{ir}$$

$$s_{12} = s_1 - s_2$$

$$a_{23} = s_2 - s_3$$

$$s_{31} = s_3 - s_1$$

The stress intensities under both piping reaction moments (bending, torsional) are checked, and the worst condition which is due to bending moment in this case is tabulated in Table 3-5.1. From Table 3-5.1 the maximum stress satisfies the allowable limit of the specified material (see Summary), therefore the valve is assured of operability and successful performance during and after the seismic event and piping load.

In addition, the requirements for structural integrity are met by satisfying the acceptance criteria in Section 6.0 of Duke Power Co., Seismic Design Manual. It is checked that the piping loadings (bending and torsional moments) capability of the valve body is greater than the adjacent piping, i.e.

$$R = [Z_v\ s_v]/[Z_p\ s_p] > 1.2$$

3. VALVE ASSEMBLY:

3.1.3.2.2 Plane 2-2: (Valve Body - neck)

The membrane stresses are due to internal pressure in three orthogonal directions and are shown as follows: (Figure 3-4.2)

$$s_{im} = P_s\ r_i/(2t)$$

$$s_{it} = P_s\ r_i/t$$

$$s_{ir} = -\ P_s/2$$

where r_i = inside radius (see Table 3-3)

t = wall thickness = $r_o - r_i$ (see Table 3-3)

P_s = standard calculated pressure (or design pressure)

In addition, there is a stress due to a thrust which is caused by main-seat as:

$$s_{th} = F_{st}/[(PI)(r_o{}^2 - r_i{}^2)]$$

The thermal secondary stresses are the same as those determined in Plane 1-1 (see Section 3.1.3.2.2.C.)

Stress intensities are then as follows:

$$s_m = s_a + s_{im} + s_{th} + s_T$$

$$s_t = s_{it} + s_T$$

$$s_1 = (s_m+s_t)/2 + [(s_m-s_t)^2/4 + s_s{}^2]^{1/2}$$

$$s_2 = (s_m+s_t)/2 - [(s_m-s_t)^2/4 + s_s{}^2]^{1/2}$$

$$s_3 = s_{ir}$$

$$s_{12} = s_1 - s_2$$

$$s_{23} = s_2 - s_3$$

3. VALVE ASSEMBLY:

$$s_{31} = s_3 - s_1$$

The numerical values are tabulated in Table 3-5.2.

3. VALVE ASSEMBLY:

TABLE 3-5.1 PLANE 1-1

r_{op}	(in)	2.250
r_{ip}	(in)	2.024
Z_p	(in^3)	3.088
s_p	(ksi)	11.075
K_1		.500
K_2		.240
dT	(^oF)	1.000
s_T	(ksi)	1.120
P_s	(ksi)	.275
r_o	(in)	2.560
r_i	(in)	1.990
s_1	(ksi)	13.874
s_2	(ksi)	1.810
s_3	(ksi)	-.138
s_{12}	(ksi)	12.064
s_{23}	(ksi)	1.948
s_{31}	(ksi)	-14.012

3. VALVE ASSEMBLY:

TABLE 3-5.2 PLANE 2-2

P_s	(ksi)	.275
r_o	(in)	3.640
r_i	(in)	2.810
s_1	(ksi)	3.358
s_2	(ksi)	2.031
s_3	(ksi)	-.138
s_{12}	(ksi)	1.327
s_{23}	(ksi)	2.168
s_{31}	(ksi)	-3.496

3. VALVE ASSEMBLY:

3.2 Yoke:

As required in the Specification, the yoke is analyzed for the maximum seismic acceleration . An analysis of the yoke structure is performed with the yoke and operator mass located at each center of gravity as shown in Figure 3-5.

3.2.1 Loading:

The gross stresses in the yoke structure shall be investigated for the critical zones as shown on the sketch in Figure 3-5.

As a conservative approach, the maximum force resultants at the specified zone can be expressed as

$$F_y = \text{thrust} = F_{st} + (m_1+m_2)\, g_{max}$$

$$S_y = \text{shear} = (m_1+m_2)\, g_{max} + C$$

$$M_y = \text{moment} = [m_1(y_1-y_6)+m_2(y_2-y_6)]\, g_{max}$$
$$+ C\,(y_1-y6)$$

$$T_y = \text{torque} = T_q + (m_1x_3, m_1z_3)_{max} g_{max}$$

All the numerical values are tabulated in Table 3-6.

3. VALVE ASSEMBLY:

3.2.2 Cross Sectional Properties:

The yoke legs have the cross section as shown in Figure 3-6, and the cross sectional properties can be determined as follows:

A_{1A} = area = $2\ (a\ b + a_1 b_1)$

A_{sA} = shear area = $2\ A_{1A}/3$ (approximation)

$I_{1A} = 2[(ab^3) + (a_1 b_1{}^3)]/12$

Z_{1A} = section modulus = $I_{1A}/(b/2)$

For a bent type structure, each column has sectional properties as follows: (assuming that the neutral axis 2-2 lies at the middle of the larger area "a b")

$I_{2A} = ba^3/12 + (b_1 a_1{}^3/12 + a_1 b_1 d_1{}^2)$

$A_{2A} = A_{1A}/2$

$Z_{2A} = I_{2A}/(a/2)$

where

$d_1 = (a + a_1)/2$

The numerical values are tabulated in Table 3.7.

3. VALVE ASSEMBLY

FIGURE 3-5 YOKE

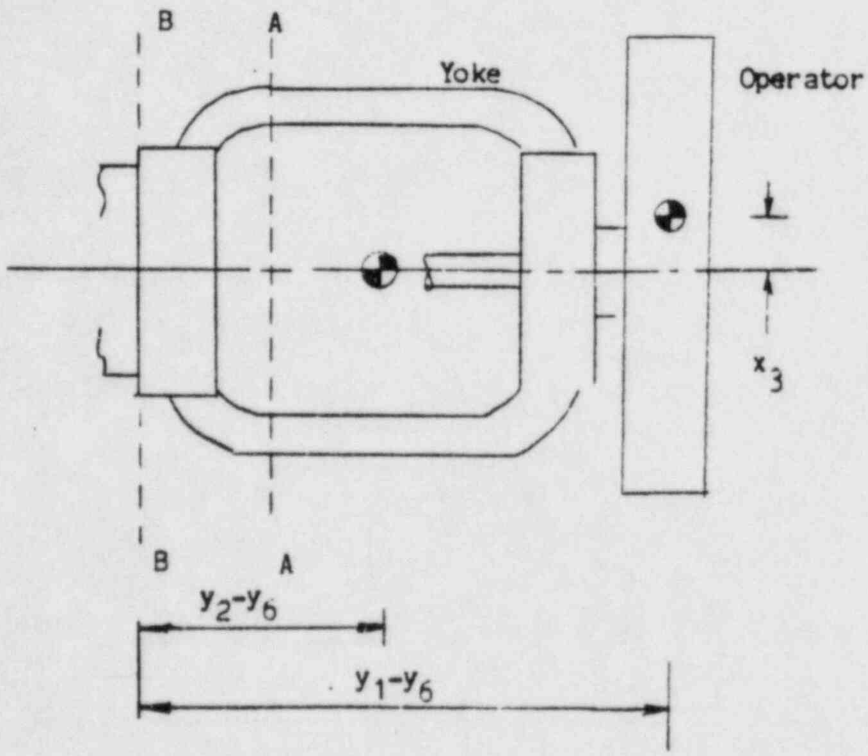
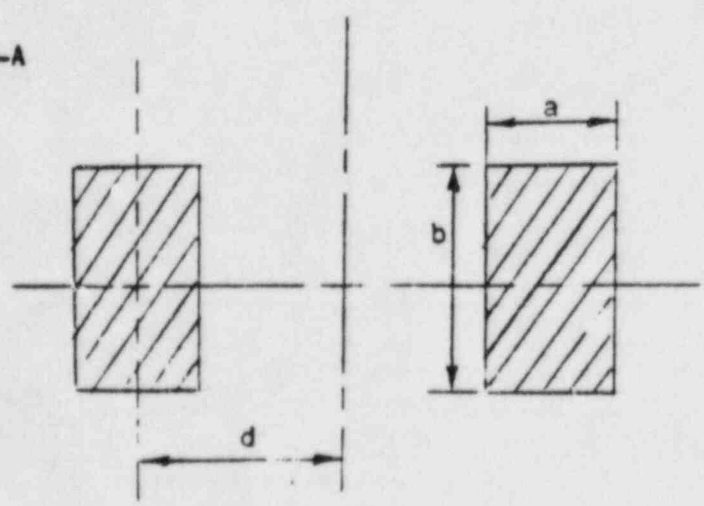


FIGURE 3-6 CROSS SECTION OF YOKE LEGS

PLANE A-A



3. VALVE ASSEMBLY:

TABLE 3-6 LOADING-YOKE

F_y	(kip)	3.581
S_y	(kip)	1.331
M_y	(in-kip)	20.303
T_y	(in-kip)	3.244

TABLE 3-7 CROSS SECTIONAL PROPERTIES--YOKE

a	(in)	1.050
b	(in)	3.620
a_1	(in)	0.000
b_1	(in)	0.000
I_{1A}	(in^4)	8.302
A_{1A}	(in^2)	7.602
A_{SA}	(in^2)	5.068
Z_{1A}	(in^3)	4.587
d	(in)	1.280
I_{2A}	(in^4)	.349
A_{2A}	(in^2)	3.801
Z_{2A}	(in^3)	.665

3. VALVE ASSEMBLY:

3.2.3 Stress:

3.2.3.1 Yoke:

Considering the yoke structure as a cantilever beam, the maximum stresses are derived by using the following expressions:

$$s_{a,bm} = F_y/A_{1A} + M_y/Z_{1A}$$

$$s_s = S_y/A_{sA} + T_y/(d)(A_{sA})$$

Another possible behavior is illustrated by the bent-type structure, where the two yoke legs are assumed to have rotational restraints at both ends. (See Figure 3-7)

The possible maximum corresponding stress can be expressed as:

$$s_{a,bt} = P_{max}/(A_{2A}) + M/(Z_{2A})$$

where

$$P_{max} = F_y/(2) + S_y\ h/(x)$$

$$M = V\ L/2$$

$$V = S_y/2$$

The maximum principal stresses s_1 and s_2 and shear stress $s_{s,max}$ for the yoke are obtained from the formula:

$$s_1 = (s_a)/2 + [(s_a)^2/4 + s_s{}^2]^{1/2}$$

$$s_2 = (s_a)/2 - [(s_a)^2/4 + s_s{}^2]^{1/2}$$

$$s_{s,max} = [(s_a)^2/4 + s_s{}^2]^{1/2}$$

where $s_a = (s_{a,bm}, s_{a,bt})_{max}$

The numerical values are tabulated in Table 3-8.

3. VALVE ASSEMBLY:

S_y = Shear
F_y = Thrust

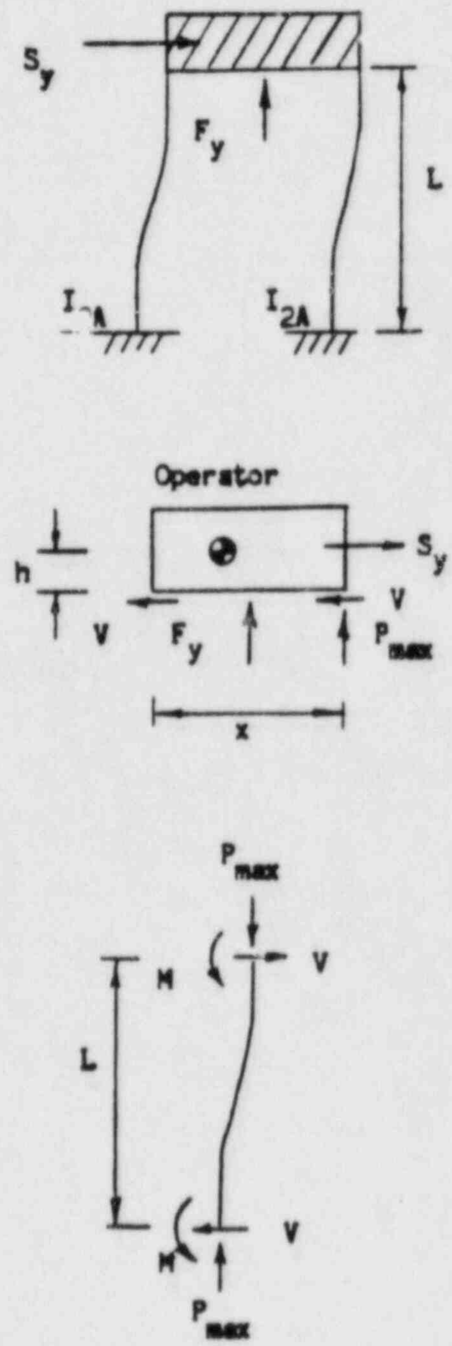


FIGURE 3-7 BENT TYPE STRUCTURE OF YOKE

3. VALVE ASSEMBLY:

TABLE 3-8 STRESSES--YOKE STRUCTURE

L	(in)	12.390
h	(in)	6.250
x	(in)	7.500
$s_{a,bm}$	(ksi)	4.898
s_s	(ksi)	.763
$s_{a,bt}$	(ksi)	6.962
s_1	(ksi)	7.044
s_2	(ksi)	-.083
$s_{s,max}$	(ksi)	3.564

3. VALVE ASSEMBLY:

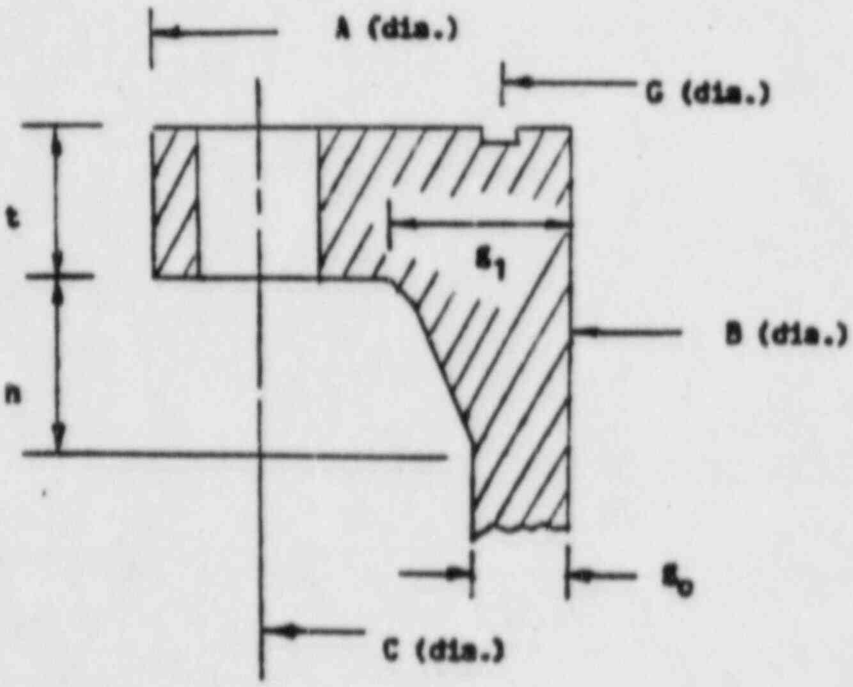


FIGURE 3-9 FLANGE (Ref: ASME B & PV Code, Figure XI-3120-1)

3. VALVE ASSEMBLY:

3.3 Bolted Body-Bonnet Joint:

3.3.1 Loading:

The criteria for the body to bonnet flange joint is governed by the rule NB-3647.1 and NC-3647.1, and Article XI-3000, Subsection NA, Section III of the ASME, Boiler and Pressure Vessel Code. A yoke with operator constitutes an extended structure and the seismic forces impose a moment and thrust on the bolted flange joint. Numerical values for these components are derived at Plane 3-3 (see Section 3.1.1.3), and resummarized in Table 3-10.

3.3.2 Stress:

3.3.2.1 Bolt Stress:

The fundamental dimensions of the flange, as shown in Figure 3-9, are tabulated in Table 3-11, (refer to ASME, Boiler and Pressure Vessel Code, Figure XI-3120-1). Some other parameters and dimensions which are defined as follows are also tabulated in the same table.

gasket parameters:

$b_o = N/2$

$b = (b_o)^{0.5} / 2$ if $b_o > 0.25"$

$b = b_o$ otherwise

where $N = (d_o - d_i)/2$

d_o = outside diameter of gasket

d_i = inside diameter of gasket

moment arms: (per Table XI-3230-1, ASME Code)

$R = (C-B)/2 - g_1$

$h_D = R + 0.5\, g_1$

$h_G = (C-G)/2$

$h_T = (R + g_1 + h_G)/2$

All the notations used in this section are in accordance with those defined in Section XI-3130, ASME Code, Subsection NA.

3. VALVE ASSEMBLY:

The flange design pressure per NB-3647.1 and NC-3647.1 is:

$$P_{FD} = P_s + 16\, M_3/((PI)G^3) + 4\, F_3/((PI)G^2)$$

where P_s = design pressure

Flange design bolt load (W) then is computed by the procedures of XI-3221.1:

$$W = 0.785\, G^2\, P_{FD} + 2b(3.14\, G\, m\, P_{FD})$$

where m = 3.0 per Table XI-3221.1

For a total bolt cross sectional area A_b, the average bolt stress is:

$$s_b = W/A_b$$

The numerical values are tabulated in Table 3-11.

3. VALVE ASSEMBLY:

TABLE 3-10 LOADING--PLANE 3-3

F_3	(kip)	3.820
S_3	(kip)	1.529
M_3	(in-kip)	28.407
T_3	(in-kip)	3.013

3. VALVE ASSEMBLY:

TABLE 3-11 BOLT STRESSES

g_1	(in)	1.125
g_2	(in)	.400
B	(in)	5.500
G	(in)	6.713
t	(in)	1.250
A	(in)	10.625
C	(in)	9.250
h	(in)	.818
N	(in)	.573
b		.268
R	(in)	.750
h_g	(in)	1.268
h_D	(in)	1.313
h_T	(in)	1.572
P_s	(psi)	275.000
A_b	(in^2)	2.663
P_{FD}	(psi)	861.182
W	(kip)	59.631
s_b	(ksi)	22.393

3. VALVE ASSEMBLY:
3.3.2.2 Flange Stress:

The flange moment (M_o) is computed by the procedures of XI-3230 as shown in the following formula:

$$M_o = M_D + M_T + M_G$$

where

$$M_D = H_D h_D, \quad M_T = H_T h_T, \quad M_G = H_G h_G$$

$$H_D = 0.785\, B^2\, P_{FD}$$

$$H = 0.785\, G^2\, P_{FD}$$

$$H_G = W - H$$

$$H_T = H - H_D$$

Stresses in the body flange and neck shall be determined for the design conditions per XI-3240. The method of analysis utilizes a number of parameters which are prescribed in the ASME Boiler and Pressure Vessel Code. On Figure XI-3240-6, the parameter f is a function of the ratios g_1/g_o and h/h_o. Likewise, Figures XI-3240-1 through XI-3240-3 provide the other parameters (T, Y, F, Z, U, V) required for the computation.

Other parameters are:

$$d = U\, h_o\, g_o^2/V, \quad e = F/h_o$$

$$L = (te + 1)/T + t^3/d$$

Finally, the maximum stresses are

Longitudinal hub stress:

$$s_H = (fM_o)/(L\, g_1^2 B) + (P_s)B/(4\, g_o)$$

Radial flange stress:

$$s_R = (1.33\, te + 1)\, M_o/(Lt^2 B)$$

Tangential flange stress:

$$s_T = Y\, M_o/(t^2 B) - Z\, s_R$$

All the numerical values are tabulated in Table 3-12.

3. VALVE ASSEMBLY:

TABLE 3-12 FLANGE STRESSES

f	2.200
F	.780
V	.170
U	3.422
T	1.532
Y	3.114
Z	1.732
d	4.777
e	.526
L	1.490
M_o (in-kip)	79.581
s_H (ksi)	17.821
s_R (ksi)	11.646
s_T (ksi)	8.666

3. VALVE ASSEMBLY:

3.3.2.3 Dome Shape Bonnet:

The required minimum thickness of the head and flange of a dome shape bonnet with bolting flange is determined by the rules of NC-3326.2, Section III of the ASME, Boiler and Pressure Vessel Code. A calculation is performed to show that the sufficient wall thickness in N.V.D. design satisfies the requirements.

Head thickness (minimum)

$$t_h = (5\ P\ L)/(6\ s_A)$$

Flange thickness (minimum)

$$t_f = Q + [1.875\ M_o\ (C+B)/[s_A\ B(7C-5B)]]^{0.5}$$

where $Q = [(P\ L)/(4\ s_A)][(C+B)/(3C-B)]$

P = design pressure

L = inside radius of the head

s = allowable stress of the material

C = bolt circle diameter

B = inside diameter of flange

M_o = total moment (see Table 3-12)

The numerical values are tabulated in Table 3-13.

3. VALVE ASSEMBLY:

TABLE 3-13 DOME SHAPE BONNET

P	(ksi)	.275
L	(in)	2.700
C	(in)	9.250
B	(in)	5.400
s_A	(ksi)	18.000
M_o	(in-kip)	79.581
t_h	(in)	.034
t_f	(in)	.779
$t_{h,design}$	(in)	.600
$t_{f,design}$	(in)	1.250

4. MAXIMUM DEFLECTION OF OPERATOR:

The deflection of the operator and the valve assembly is computed by using the static analysis of the finite element computer program SAP6. The structural model is presented in Section 5 of this report. The corresponding values of acceleration in X, Y and Z directions as specified in the specification are used in the computer program, assuming acting simultaneously. Table 4-1 tabulated the results of the displacement in X, Y and Z directions obtained from SAP6 output printout, and the resultant maximum displacement of each nodal point.

The maximum displacements shown in the table indicate that they are sufficiently small and will not limit the operation of the stem and operator. A review was made to assure that machining tolerances do adequately accommodate the movements.

TABLE 4-1 DEFLECTIONS

NODE	DX(IN)	DY(IN)	DZ(IN)	DMAX(IN)
20	.0093	-.0018	.0132	.0162
19	.0082	.0001	.0129	.0153
18	.0068	.0001	.0084	.0108
17	.0063	.0001	.0066	.0091
16	.0063	-.0003	.0060	.0087
15	.0063	.0005	.0073	.0096
14	.0035	-.0002	.0034	.0049
13	.0035	.0004	.0040	.0054
12	.0012	-.0003	.0020	.0024
11	.0012	.0004	.0021	.0025
10	.0012	-.0002	.0020	.0024
9	.0012	.0003	.0021	.0024
8	.0012	.0001	.0021	.0024
7	.0007	.0001	.0013	.0015
6	.0002	.0001	.0005	.0005
5	.0000	.0001	.0001	.0001
4	.0000	-.0000	.0000	.0000
3	.0000	.0001	.0000	.0001

5. NATURAL FREQUENCY:

The natural frequency for the valve assembly is determined by idealizing the structure as a multi-mass system interconnected by structural members as denoted in Figure 5-1. A finite element computer program, SAP6 -- Structural Analysis Program for static and dynamic analysis, is used to calculate both the deflections and natural frequencies. Details for the use and interpretation of the SAP6 computer program are available in the User Manual, by SAP Users Group, Civil Engineering Department, University of Southern California, Nov. 1977.

In the structural model, 20 nodal points are assigned to represent various components of the valve assembly, along with 13 concentrated lumped masses. The structural characteristics are described by means of geometrical properties for 10 designated cross sections. These quantities are calculated for the beam members (MTYP=2) on the dynamic model. The whole system is considered to be restrained only at the two weld ends of the body main run, and have six degrees of freedom at each other nodal point (3 translational, 3 rotational). The description of the nodal points is shown below:

Nodal Point	Mass Point	Description
1,2		weld ends of body main run
3,4	a,b	valve body main run
5	c	valve body center + gate/disc/seat + 1/5 stem
6	d	valve body neck + 1/5 stem
7	e	valve body neck flange + bonnet + 1/5 stem
8		center of yoke-body/bonnet joint
9,10		dummy nodes
11,12	f,g	yoke flange + 1/5 stem +clamps/bolts
13,14	h,i	yoke "leg"
15,16	j,k	yoke top + 1/5 stem
17		center of yoke top
18	l	adapter/yoke-operator joint
19		dummy node
20	m	C.G. of operator

5. NATURAL FREQUENCY:

Due to the fact that the geometry varies throughout its lengh, each structural member is defined by the minimum geometrical properties for the segment. Solutions derived with this approximation will result in the lowest natural frequency as a conservative approach. The dummy nodes 9 & 10 are justified to account for the curvature effect of the yoke leg, and dummy node 19 is to serve the transaction from node 18 (adapter or yoke-operator joint) to node 20 (center of gravity of operator). For shell type yoke design, the equivalent cross section properties to beams are used (beams 8 & 9). The mass moments of inertia for all mass points except operator are computed by an engineering approximation of the actual mass distribution and the radius of gyration. Slight discrepancies in data will have negligible effects on the final computer results. The mass moment of inertia of the operator, however, is provided directly from the supplier.

Table 5-2 through Table 5-4 list the input data of the nodal point coordinates, the geometrical properties of the structural members as well as the masses and mass moments of inertia. The first three modes of the natural frequency are tabulated in Table 5-1.

FIGURE 5-1

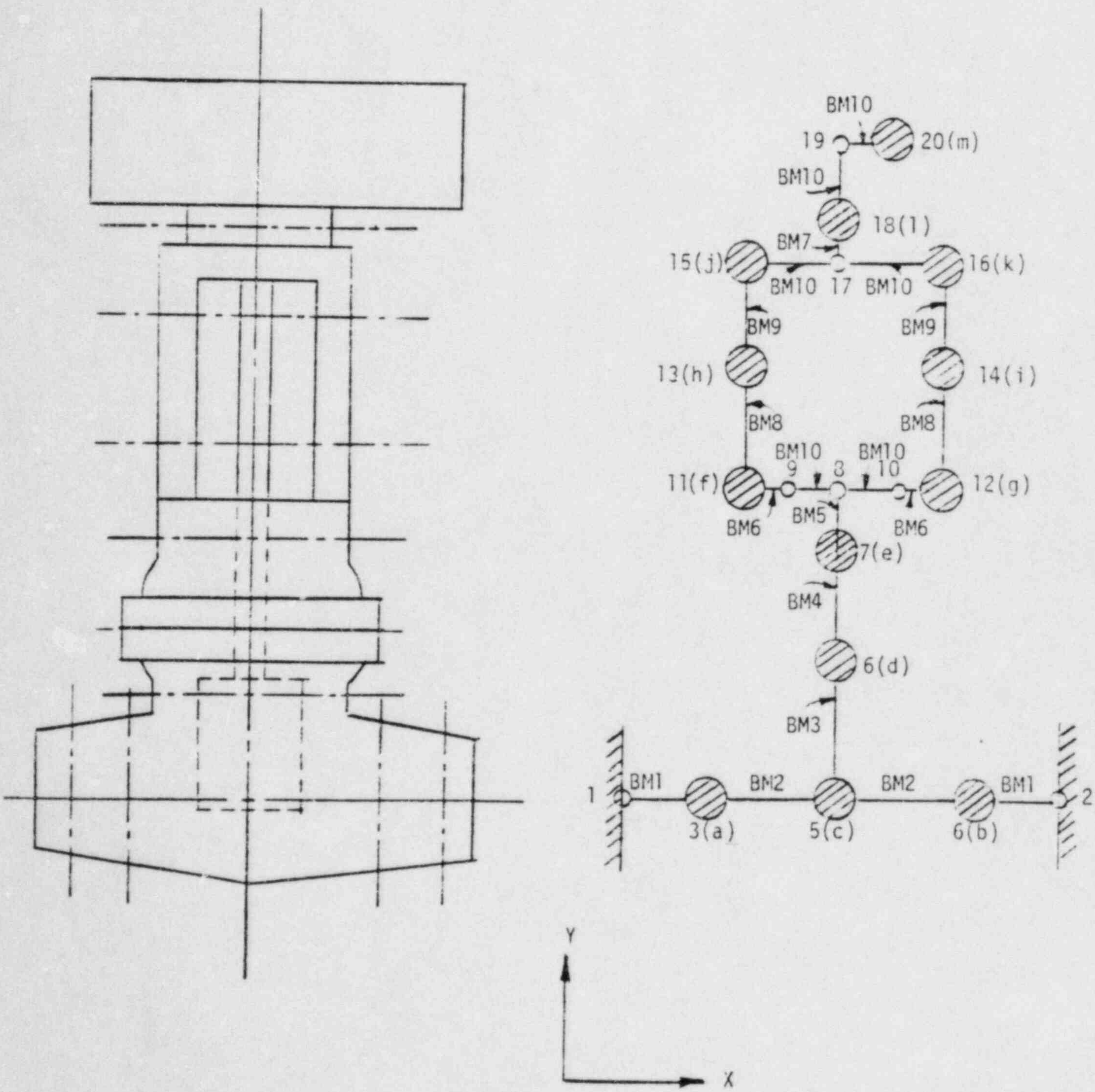

BM10
19
20(m)
BM10
18(1)
BM7
15(j)
17
16(k)
BM10
BM10
BM9
BM9
13(h)
14(i)
BM8
BM8
BM10
BM10
9
8
10
11(f)
12(g)
BM6
BM5
BM6
7(e)
BM4
6(d)
BM3
1
BM1
BM2
BM2
BM1
2
3(a)
5(c)
6(b)
Y
X

TABLE 5-1 NATURAL FREQUENCY

MODE 1	42.520
MODE 2	53.300
MODE 3	73.020

TABLE 5-2 NODAL POINTS

NODAL POINT	X-COORD.	Y COORD.	Z COORD.
1	-6.00	0.00	0.00
2	6.00	0.00	0.00
3	-3.98	0.00	0.00
4	3.98	0.00	0.00
5	0.00	0.00	0.00
6	0.00	3.00	0.00
7	0.00	8.00	0.00
8	0.00	11.85	0.00
9	-1.59	11.85	0.00
10	1.59	11.85	0.00
11	-2.11	11.85	0.00
12	2.11	11.85	0.00
13	-1.88	16.65	0.00
14	1.88	16.65	0.00
15	-1.66	21.45	0.00
16	1.66	21.45	0.00
17	0.00	21.45	0.00
18	0.00	23.85	0.00
19	0.00	30.10	0.00
20	-.75	30.10	2.81

TABLE 5-3 BEAM CROSS SECTIONAL PROPERTIES

BEAM NO.	AREA	TORSION J	MOMENT I1	MOMENT I2
1	8.15	42.83	21.416	21.416
2	8.15	42.83	21.416	21.416
3	16.82	177.82	88.910	88.910
4	23.84	278.68	139.340	139.340
5	20.46	78.14	39.072	39.072
6	1000.00	20000.00	10000.000	10000.000
7	44.40	330.76	165.381	165.381
8	3.27	1.04	3.068	.259
9	2.33	.53	1.550	.132
10	1000.00	20000.00	10000.000	10000.000

TABLE 5-4 MASS AND MASS MOMENT OF INERTIA

NODE	MASS	M.M.I-X	M.M.I-Y	M.M.I-Z
3	.037	0.000	.024	0.000
4	.037	0.000	.024	0.000
5	.128	0.000	.084	0.000
6	.044	0.000	.059	0.000
7	.098	0.000	.152	0.000
11	.022	0.000	.032	0.000
12	.022	0.000	.032	0.000
13	.018	0.000	.026	0.000
14	.018	0.000	.026	0.000
15	.021	0.000	.030	0.000
16	.021	0.000	.030	0.000
18	.008	0.000	.011	0.000
20	.534	14.356	36.052	29.980

Nuclear Valve Division
Borg-Warner Corporation
7500 Tyrone Ave., Van Nuys, California 91409

Energy Equipment

6. CONCLUSION:

The following valves have been evaluated and qualified in accordance with the ASME Boiler and Pressure Vessel Code, Section III, Nuclear Power Plant Components, Division 1, Subsection NC, 1974 and the requirements in Duke Power Company, Catawba Nuclear Station, Units 1 and 2, Specification No. CNS-1205.00-6, Addendum 6, December, 1978, ASME Section III Carbon Steel Gate, Globe and Check Valve.

Qualification is based upon the analytical procedure defined in Duke Power Company, Seismic Design Manual for rigid mechanical systems

In accordance with the design specifications, it has been demonstrated that the valve assembly satisfies the design criteria for stresses and deformations. A finite element computer program SAP VI has been used to evaluate the natural frequency of the valve assembly. With conservative quantities for mass and inertia, specification limit of 33 cycles per second, which classifies the component as a rigid system.

VALVE SIZE (INCH)	PRESSURE RATING (LB)	MATERIAL	VALVE TYPE	OPERATOR	N.V.D. ASSEMBLY DWG. NO.
4	150	CARBON STEEL	MOTOR	GATE	401JBB3-4

Nuclear Valve Division

Borg-Warner Corporation
7500 Tyrone Ave., Van Nuys, California 91409

Energy Equipment

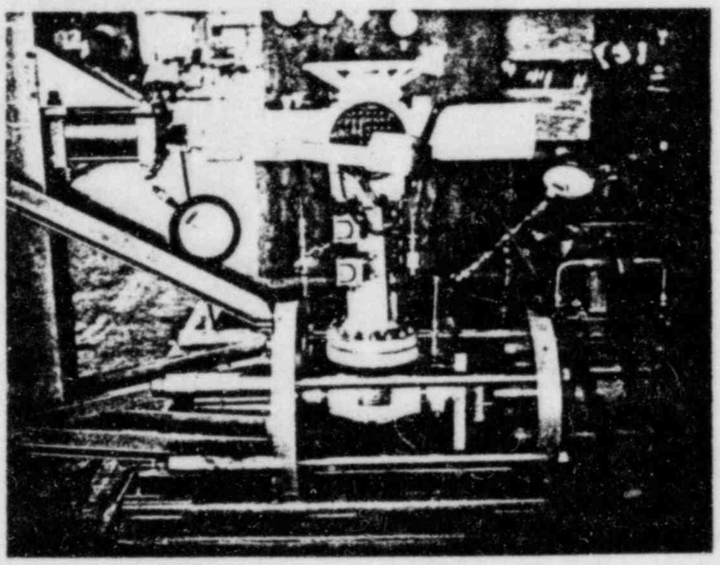
STATIC DEFLECTION TEST RESULTS

REPORT NO. NSR401JBB3-004

PAGE Attachment 1

REVISION A

ATTACHMENT TO REPORT NO. NSR401JBB3-004



TEST SETUP

Nuclear Valve Division
Borg-Warner Corporation
7500 Tyrone Ave., Van Nuys, California 91409

BORG WARNER Energy Equipment

REPORT NO. NSR401JBB3-004
PAGE Attachment 1
REVISION A

ATTACHMENT TO REPORT NO. NSR 401JBB3-004

The following active valve was tested per NVD Report No. 1681, as required by Duke Specification No. CNS 1205.00-6.

RESULTS OF TEST

NVD PART NO. 401JBB3-004 DUKE ITEM NO. 2B-400

SERIAL NO. 44645 DESCRIPTION 4" 150lb. GATE M.O.

DIRECTION OF LOAD X

MAGNITUDE OF LOAD 1150 LBS

SEAT LEAKAGE TEST PRESSURE 210 PSIG

TEST	CYCLE NUMBER		
	1	2	3
OPEN TIME (Seconds)	4.2	4.2	4.2
CLOSE TIME (Seconds)	4.4	4.4	4.4

SEAT LEAKAGE 0.1 cc L

SEAT LEAKAGE (Reverse) 0.3 cc R

Test Performed by: [signature]
Witnessed by: [signature]

Photograph of setup on next page.

Attachment 7

Seismic Analysis
and
Static Deflection Test
for
Valves VQ3B and VQ15B