LOFT TECHNICAL REPORT LTR 112-120
MARCH 3, 1978

MASTER

DESIGN STRESS ANALYSIS OF 12-INCH BLOWDOWN VALVE

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DEPARTMENT OF ENERGY

IDAHO OPERATIONS OFFICE UNDER CONTRACT EY-76-C-07-1570

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"Design Stress Analysis of 12-inch Blowdown Valve"			

REMARKS:

This LTR is a report to determine the structural adequacy of the 12-inch austenetic stainless steel blowdown valve in accordance with Article NB-3500, ASME Boiler and Pressure Vessel Code, Section III (1971).

Based on the following calculation, this valve is considered to be structurally adequate in accordance with Article NB-3500 (Class I Valves) for the design conditions as set forth in Aerojet Nuclear Company Spec. ANC-60070 Rev. C.

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SUMMARY

This LTR is a report to determine the structural adequacy of the 12-inch austenetic stainless steel blowdown valve in accordance with Article NB-3500, ASME Boiler and Pressure Vessel Code, Section III (1971).

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DESIGN STRESS ANALYSIS

OF

12" BLOW DOWN VALVE

PER

ASME BOILER & PRESSURE VESSEL CODE

SECTION III

ARTICLE NB-3560

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JUN 25 1973
AEROJET NUCLEAR COMPANY
E.C. CONTRACT NO. AT (10-1)-1375
P.O. NO. S-7283
SPEC. NO. 60070
EQUIP. ITEM NO.
NO. 060202

James C. Quinn
12/6/72

REVISED 6/12/73

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TABLE OF CONTENTS

	<u>PAGE</u>
Forward-----	2
Conclusion-----	4
Material & Physical Data-----	5
Design Specifications-----	5

Valve Body Design

Minimum Wall Thickness-----	6
Primary Pressure Rating-----	7
Standard Calculation Pressure-----	7
Primary Membrane Stresses-----	8
Secondary Stresses due to Internal Pressure-----	9 & 9A
Secondary Stresses due to Pipe Reaction-----	9 & 9A
Secondary Stresses due to Thermal Effect-----	11 & 11A
Fatigue Requirements-----	13 & 13A
Cyclic Stresses-----	14 & 14A
Flange Inlet End Valve-----	17
Flange Outlet End Valve-----	19
Flange Outlet End-----	21

Other Significant Stresses

Body Studs & Nuts-----	23
Poppet-----	26
Web-----	28
Cone-----	30
Tables and Charts-----	33

Forward

This is a report to determine the structural adequacy of the pressure retaining and structural parts of a 12" Austenetic Stainless Steel Blowdown Valve. The analysis is in accordance with the design rules of Article NB-3500 of ASME - Boiler and Pressure Vessel Code - Section III (1971). The valve will be mounted horizontal, supported by attached lines and other supporting structures, and will be used to repeatedly simulate rapid pipe rupture and coolant release in an experimental pressurized water reactor (PWR). The design pressure and temperature of the valve is 2500 psia @ 650°F. The valve will be cycled a maximum of 260 times during a 30 year design life at an operating pressure of 2250 psia and temperature of 300°F prior to opening and 610°F after opening.

The loads used in the analysis of the Body flanges to develop an equivalent Pressure (Peq.) as required by the Code were found in Table III (Piping Loads) of Aerojet Nuclear Spec. ANC-60070 Rev. B. However, in accordance with Aerojet Spec., these loads were not combined as required by the Code, but were considered singly acting in conjunction with the load due to internal pressure. Leakage of the flange joint is the normal failure criterion and since the Torsional Moment and the Lateral force do not produce any reduction in gasket load, they were not used in the development of (Peq.). Likewise, in accordance with the Spec. ANC-60070 Rev. C, the Seismic and Blowdown Test acceleration loads were not applied during this consideration.

The materials selected for the wetted sections of the valve have been reviewed for compatibility with the water chemistry shown on Table V, in Aerojet Nuclear Company Specification ANC-60070 Rev. "C", and were found to be acceptable. There does not appear to be anything in the water chemistry that would effect the function of the valve.

The methods of calculations are in accordance with the applicable requirements of the following Codes and Standards.

ASME - Boiler & Pressure Vessel Code Section III Nuclear Vessels (1971)
and Summer 1972 Addenda.

ASME - Boiler & Pressure Vessel Code Section VIII Pressure Vessels
Div. 1 (1971) and Summer 1972 Addenda.

ANSI - B31.7 - Nuclear Power Piping (1969)

Conclusion

Based on the following calculation, this valve is considered to be structurally adequate in accordance with Article NB-3500 (Class I Valves) for the design conditions as set forth in Aerojet Nuclear Company Spec.

ANC-60070 Rev. C.

MATERIAL & PHYSICAL DATA

Body Inlet & Outlet	ASME-SA-182 Gr 316
Poppet	ASME-SB-166
Web	ASME-SA-182 Gr 316
Cone	ASME-SA-182 Gr 316
Body Bolting	ASME-SA-453 Gr 660

Design Stress Intensity Values (Sm) from ASME Sec. III Appendix I

DESCRIPTION	MATERIAL DESIGNATION	ULTIMATE STRENGTH S_u	YIELD STRENGTH S_y	Sm-psi			
				100°F	500°F	650°F	
Stainless Steel Forged	-316	75,000	30,000	20,000	17,900	16,600	
Inconel	N Cr Fe	80,000	35,000	23,300	23,300	23,300	
Stainless Steel Bolting	GR-660	130,000	85,000	28,300	27,100	26,900	

DESIGN SPECIFICATION

Design Pressure	2500 psi
Design Temperature	650°F

Operating Conditions	
Operating Pressure	2250 psi
Operating Temperature	610°F (After opening Va.)
Operating Temperature	300°F (Prior to opening Va.)

VALVE BODY DESIGN CALCULATIONS

REF. ASME SEC. III NB-3500
 CONSOLIDATED CONTROLS DWG. #J-72382 REVE

REF. P_1, P_2 ; TAB - NB-3531-5
 t_1, t_2 ; TAB - NB-3542.1

3531-6 (FLANGED
 END VALV

I) MINIMUM WALL THICKNESS (t_m)

REF NB-3543(b)

$$a) \quad t_{m1} = t_1 + \left(\frac{P_D - P_1}{P_2 - P_1} \right) (t_2 - t_1)$$

$$t_{m1} = 1.213 + \left[\frac{2500 - 1550}{2580 - 1550} \right] (2.075 - 1.213)$$

$$t_{m1} = 1.213 + (.922)(.863) = 2.008$$

CORROSION ALLOWANCE = .010"

$$t_{m1} = 2.008 + .010 = 2.018"$$

RECOMMENDED MIN. THICKNESS $2\frac{1}{8}"$ ~ DWG = $2\frac{3}{16}"$: .0

$$b) \quad t_{m2} = 1.77 + \left[\frac{2500 - 1550}{2580 - 1550} \right] (3.06 - 1.77)$$

$$= 1.77 + (.922)(1.29)$$

$$= 2.96$$

CORROSION ALLOWANCE = .010

$$\therefore t_{m2} = 2.96 + .010 = 2.970$$

RECOMMENDED MIN. THICKNESS $3"$ ~ DWG = $3\frac{1}{8}"$: .0

$d_1 = 10\frac{1}{8}$
 $P_1 = 1550$
 $P_2 = 2580$
 $P_D = 2500$
 $t_1 = 1.213$
 $t_2 = 2.075$

$d_2 = 16"$
 $t_1 = 1.77$
 $t_2 = 3.06$

P_1 } SAME
 P_2 } AS
 P_D } ABOVE

II PRIMARY PRESSURE RATING (P_R)

REF NB-3543(C)

$$P_R = P_{R1} + \left[\frac{P_d - P_1}{P_2 - P_1} \right] (P_{R2} - P_{R1})$$

$$P_{R1} = 900$$

$$P_{R2} = 1500$$

$$P_R = 900 + \left[\frac{2500 - 1550}{2580 - 1550} \right] (1500 - 900)$$

$$\left. \begin{array}{l} P_d \\ P_1 \\ P_2 \end{array} \right\} \begin{array}{l} \text{SAME} \\ \text{AS} \\ \text{ABOVE} \end{array}$$

$$P_R = 900 + (.922)(600)$$

$$P_R = 1453.2 \quad \text{USE } 1455 \text{ psi}$$

III STANDARD CALCULATION PRESSURE (P_S)

P_S FOR A FLANGED END VALVE
WILL BE USED IN THESE CALCULATIONS
REF NB-3531

$$\text{WHEN } P_R = 1455 \sim P_S = 3000 \quad \text{REF FIG NB-3545.1-1}$$

PRIMARY MEMBRANE STRESS DUE TO INTERNAL PRESSURE

REF NB-3545.1

$$a) P_m = \left(\frac{A_s}{A_m} + 0.5 \right) P_s$$

$$A_s = 283 \text{ sq in.}$$

$$A_m = 144 \text{ sq in.}$$

AREAS OBTAINED
WITH PLANIMETER
FROM FULL SCALE
LAYOUT

$$P_m = \left(\frac{283}{144} + 0.5 \right) 3000$$

$$= (1.965 + 0.5) 3000$$

$$= (2.465) (3000)$$

$$= 7396 \text{ psi} \quad \underline{\text{OK}}$$

ALLOWABLE IS
 S_m @ 500°F FOR
BODY MATERIAL

$$S_m = 17,900$$

PRIMARY MEMBRANE STRESS DUE TO INTERNAL PRESSURE

REF. APPENDIX F
SECTION II
91 A-2221
PER
CUSTOMER
REQUEST

$$a) \quad S = \frac{P R}{t} + \frac{P}{2}$$

$$= \frac{(3000)(5.06)}{2.125} + \frac{3000}{2}$$

WHERE

P = INT. PRESS.
R = INT. RADIUS
t = WALL THICKNESS

$$= 7144 + 1500 = 8644 \text{ psi} \approx \text{OK}$$

ALLOWABLE
 S_m @ 500°F
FOR BODY MAT
 $S_m = 17,900$

$$b) \quad S = \frac{(3000)(8)}{3} + \frac{3000}{2}$$

$$S = 8000 + 1500 = 9500 \text{ psi} \approx \text{OK}$$

VALVE BODY SECONDARY STRESS

REF NB-3545.2

I) PRIMARY PLUS SECONDARY STRESSES DUE TO INTERNAL PRESSURE, PIPE REACTION & THERMAL EFFECTS

$$S_m = Q_p + P_{ed} + 2 Q_{T2}$$

$$\text{ALLOWABLE} = 3S_m @ 500^\circ\text{F}$$

$$3S_m = 53,700$$

a) PRIMARY PLUS SECONDARY STRESS DUE TO INTERNAL PRESSURE

$$Q_p = C_p \left(\frac{r_i}{t_e} + 0.5 \right) P_s$$

REF. NB-3545.2(a)(1)

$$C_p = 3$$

$$= 3 \left(\frac{5.06}{2.125} + 0.5 \right) 3000 = 25,930 \text{ psi}$$

b) SECONDARY STRESS DUE TO PIPE REACTION

REF NB-3545.2(b)

$$P_{ed}, P_{eb} \text{ or } P_{et}$$

$$\text{ALLOWABLE CONSIDERED INDIVIDUALLY IS}$$

$$1.5 S_m @ 500^\circ\text{F} = 26,850 \text{ psi}$$

VALVE BODY SECONDARY STRESS

REF NB-3545.2

I) PRIMARY PLUS SECONDARY STRESSES DUE TO INTERNAL PRESSURE, PIPE REACTION & THERMAL EFFECTS

$$S_m = Q_P + P_{ed} + 2 Q_{T2}$$

ALLOWABLE = $3S_m @ 5$
 $3S_m = 53,700$

a) PRIMARY PLUS SECONDARY STRESS DUE TO INTERNAL PRESSURE

$$Q_P = C_P \left(\frac{r_i}{t_e} + 0.5 \right) P_s$$

REF. NB-3545.2(a)

$$C_P = 3$$

$$= 3 \left(\frac{9.5}{3} + 0.5 \right) 3000 = 33,000$$

b) SECONDARY STRESS DUE TO PIPE REACTION

REF NB-3545.2(b)

P_{ed} , P_{eb} or P_{et}

ALLOWABLE CONSIDERED INDIVIDUALLY IS
 $1.5 S_m @ 500^\circ F = 26,850$

1) DIRECT OR AXIAL EFFECT

$$P_{ed} = \frac{(F_d)(S)}{G_d}$$

$$= \frac{(23)(19,900)}{84.6}$$

$$= 5410 \text{ psi} \leq 26,850 \text{ psi} \quad \text{OK}$$

REF. NB-3545.2(b)(1)
 $S = S_y$ OF PIPING MATL
 @ 500°F
 $S = 19,900 \text{ psi}$
 $F_d = 23$ REF FIG.
 NB-3545.2.3

$G_d = 84.6$ METAL
 AREA @ A-A

2) BENDING LOAD EFFECT

$$P_{eb} = \frac{(C_b)(F_b)(S)}{G_b}$$

$$= \frac{(1)(115)(19,900)}{228}$$

$$= 10,050 \text{ psi} \leq 26,850 \text{ psi} \quad \text{OK}$$

REF NB-3545.2(b)(1)
 $F_b = 115$
 REF FIG. NB-3545.2-4
 $C_b = 1$
 REF FIG. NB-3545.2-6
 $\frac{t_e}{\bar{r}} = \frac{2\frac{1}{8}}{6\frac{1}{8}} = .347$
 $G_b = 228$ SECTION
 MODULUS @ A-A

3) TORISIONAL LOAD EFFECT

$$P_{et} = \frac{(2)(F_b)(S)}{G_t}$$

$$= \frac{(2)(115)(19,900)}{456}$$

$$= 10,050 \leq 26,850 \text{ psi} \quad \text{OK}$$

REF. NB-3545.2(b)(1)

$G_t = 456$ - POLAR
 SECTION MODULUS
 @ A-A.

$G_d = \text{METAL AREA @ A-A}$

REF NB-3545.2(b)(4)

$$G_d = \frac{\pi}{4}(d_o^2 - d_i^2) = (.7854)(14.5^2 - 10.125^2) = 84.6$$

$G_b = \text{SECTION MODULUS @ A-A}$

REF NB-3545.2(b)(5)

$$G_b = \frac{\pi}{32} \left(\frac{d_o^4 - d_i^4}{d_o} \right) = .0982 \left[\frac{14.5^4 - 10.125^4}{14.5} \right] = 228$$

$G_t = \text{POLAR SECTION MODULUS @ A-A}$ REF. NB-3545.2(b)(6)

$$= \frac{\pi}{16} \left[\frac{d_o^4 - d_i^4}{d_o} \right] = .1964(2320) = 456$$

1) DIRECT OR AXIAL EFFECT

$$P_{ed} = \frac{(F_d)(S)}{G_d}$$

$$= \frac{(35)(30,000)}{179}$$

$$= 5866 \leq 26,850 \text{ psi OK}$$

REF. NB-3545.2(b)
S = SY OF PIPING MA
@ 500°F
S = 30,000 psi
F_d = 35 REF FIG.
NB-3545.

G_d = 179 META
AREA @ A-A

2) BENDING LOAD EFFECT

$$P_{eb} = \frac{(C_b)(F_b)(S)}{G_b}$$

$$= \frac{(1)(290)(30,000)}{753}$$

$$= 11554 \text{ psi} \leq 26,850 \text{ psi OK}$$

REF NB-3545.2(b)
F_b = 290
REF FIG. NB-3545.

C_b = 1
REF FIG. NB-3545.2

$$\frac{t_e}{\bar{r}} = \frac{3}{0.5} = .316$$

G_b = 753 SECTION
MODULUS @ A-A

3) TORISIONAL LOAD EFFECT

$$P_{et} = \frac{(2)(F_b)(S)}{G_t}$$

$$= \frac{(2)(290)(30,000)}{1506}$$

$$= 11,554 \leq 26,850 \text{ psi OK}$$

REF. NB-3545.2(b)

G_t = 1506 POLAR
SECTION MODULUS
@ A-A.

G_d = METAL AREA @ A-A

REF NB-3545.2(b)

$$G_d = \frac{\pi}{4}(d_o^2 - d_i^2) = (.7854)(22^2 - 16^2) = 179$$

G_b = SECTION MODULUS @ A-A

REF NB-3545.2(b)

$$G_b = \frac{\pi}{32} \left(\frac{d_o^4 - d_i^4}{d_o} \right) = .0982 \left[\frac{22^4 - 16^4}{22} \right] = 753$$

G_t = POLAR SECTION MODULUS @ A-A REF. NB-3545.2

$$= \frac{\pi}{16} \left[\frac{d_o^4 - d_i^4}{d_o} \right] = .1964(7669) = 1506$$

c) SECONDARY STRESS DUE TO THERMAL EFFECT
REF NB. 3545.2(C)

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

REF NB-3545.2(C)

- 1) STRESS COMPONENT FROM THROUGH WALL TEMPERATURE GRADIENT

REF NB-3545.2(C)(1)

$$Q_{T1} = 7500$$

SEE SUMMARY OF STRESS INDICIES & FACTORS

- 2) STRESS COMPONENT FROM THROUGH WALL THICKNESS VARIATION

REF NB-3545.2(C)(2)

$$\begin{aligned} Q_{T2} &= (C_6)(C_2)(\Delta T_2) \\ &= (253)(.46)(24) \\ &= 2793 \end{aligned}$$

SEE SUMMARY OF STRESS INDICIES & FACTORS

- 3) STRESS COMPONENT

REF NB-3545.2(C)(3)

$$\begin{aligned} Q_{T3} &= (C_6)(C_3)(\Delta T_2) \\ &= (253)(.65)(24) \\ &= 3947 \end{aligned}$$

c) SECONDARY STRESS DUE TO THERMAL EFFECT
REF NB. 3545.2(C)

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

REF NB-3545.2(C)

1) STRESS COMPONENT FROM THROUGH WALL TEMPERATURE GRADIENT

REF NB-3545.2(C)(1)

$$Q_{T1} = 7500$$

SEE SUMMARY OF STRESS INDICES & FACTORS

2) STRESS COMPONENT FROM THROUGH WALL THICKNESS VARIATION

REF NB-3545.2(C)(2)

$$\begin{aligned} Q_{T2} &= (C_6)(C_2)(\Delta T_2) \\ &= (253)(.46)(4.4) \\ &= 512 \end{aligned}$$

SEE SUMMARY OF STRESS INDICES & FACTORS

3) STRESS COMPONENT

REF NB-3545.2(C)(3)

$$\begin{aligned} Q_{T3} &= (C_6)(C_3)(\Delta T_2) \\ &= (253)(.55)(4.4) \\ &= 612 \end{aligned}$$

THEREFORE SECONDARY STRESS (Q_T)

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

$$= 7500 + 2793$$

$$= 10,293 \text{ psi}$$

REF NB-3545.2 (C)

CODE DOES NOT GIVE
ALLOWABLE FOR Q_T
AND DOES NOT USE
IT IN THE FORMULAS
ANYMORE. HOWEVER
SINCE IT IS LESS
THAN S_m @ 500°F FOR
THE BODY MATERIAL
I CONSIDER IT OK.

$$\therefore S_m = Q_p + P_{ed} + 2Q_{T2}$$

REF NB-3545.2

$$= 25,930 + 5,410 + (2)(2793)$$

$$= 36,926 \text{ psi}$$

ALLOWABLE IS
 $3S_m$ @ 500°F = 53,700

THEREFORE SECONDARY STRESS (Q_T)

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

$$= 7500 + 512$$

$$= 8,012$$

REF NB-3545.2 (C)

CODE DOES NOT GIVE
ALLOWABLE FOR Q_T
AND DOES NOT USE
IT IN THE FORMULAS
ANYMORE. HOWEVER
SINCE IT IS LESS
THAN $S_m @ 500^\circ F$ FOR
THE BODY MATERIAL
I CONSIDER IT OK.

$$\therefore S_m = Q_p + P_{ed} + 2Q_{T2}$$

REF NB-3545.2

$$= 33,000 + 5,866 + (2)(512)$$

$$= 39,890$$

ALLOWABLE IS
 $3S_m @ 500^\circ F = 53,700$

FATIGUE REQUIREMENTREF NB-3545.3(a) &
NB-3553(a)

MAXIMUM NUMBER OF CYCLES

$$N_a \geq 2000$$

$$S_a = \text{LARGER OF } S_{p1} \text{ OR } S_{p2}$$

$$a) \quad S_{p1} = \frac{2}{3} Q_P + \frac{P_{eb}}{2} + Q_{T3} + 1.3 Q_{T1} \quad \text{REF NB-3545.3 a}$$

$$S_{p1} = \frac{2}{3}(25,930) + \frac{10,050}{2} + 3947 + 1.3(7500)$$

$$S_{p1} = 17,287 + 5,025 + 3,947 + 9,750$$

$$S_{p1} = 36,009$$

$$b) \quad S_{p2} = 0.4(Q_P) + \frac{K}{2}(P_{eb} + 2Q_{T3})$$

$$S_{p2} = 0.4(25,930) + \frac{2}{2}[10,050 + 2(3947)]$$

$$S_{p2} = 10,372 + 17,944$$

$$S_{p2} = 28,316$$

K = FATIGUE STRENGTH
REDUCTION
FACTOR & IS
CONSIDERED TO
BE 2

REF NB-3545.3(a)

$$\therefore S_a = S_{p1} = 36,009$$

$$\therefore N_a = 1.1 \times 10^5 \geq 2000$$

REF NB-3545.3(a).
FIG 1-9.2

FATIGUE REQUIREMENTREF NB-3545.3(a) &
NB-3553(a)

MAXIMUM NUMBER OF CYCLES

$$N_a \geq 2000$$

$$S_a = \text{LARGER OF } S_{p1} \text{ OR } S_{p2}$$

$$a) \quad S_{p1} = \frac{2}{3} Q_p + \frac{P_{eb}}{2} + Q_{T3} + 1.3 Q_{T1} \quad \text{REF NB-3545.3 a}$$

$$S_{p1} = \frac{2}{3}(33,000) + \frac{11,554}{2} + 612 + 1.3(7500)$$

$$S_{p1} = 22,000 + 5777 + 612 + 9750$$

$$S_{p1} = 38,139$$

$$b) \quad S_{p2} = 0.4(Q_p) + \frac{K}{2}(P_{eb} + 2Q_{T3})$$

$$S_{p2} = 0.4(33,000) + \frac{2}{2} [11,554 + 2(612)]$$

$$S_{p2} = 13,200 + 12,778$$

$$S_{p2} = 25,978$$

K = FATIGUE STRENGTH
REDUCTION
FACTOR & IS
CONSIDERED TO
BE 2

REF NB-3545.3(a)

$$\therefore S_a = S_{p1} = 38,139$$

$$\therefore N_a = 8 \times 10^4 \geq 2000$$

REF. NB-3545.3 (a)
FIG 1-9.2

CYCLIC STRESS CALCULATIONS

REF NB-3554

$$a) Q_p + P_{ed} + (C_1)(C_2)(C_4)(\Delta T_{f \text{ max}}) \leq 3 S_m \quad \text{REF NB-3554(a)}$$

$$25,930 + 5410 + (253)(.46)(.33)(310)$$

$$25930 + 5410 + 11,906$$

$$43,246 \leq 3 S_m = 53,700 \quad \text{OK}$$

SEE SUMMARY OF
STRESS INDICIES &
FACTORS

$$b) S_{m(\text{max})} = Q_p + P_{eb} + (C_1)(C_3)(C_4)(\Delta T_{f \text{ max}}) \quad \text{REF NB-3554 (b)}$$

$$S_{m(\text{max})} = 25,930 + 10,050 + (253)(.65)(.33)(310)$$

$$= 25,930 + 10,050 + 16,823$$

$$= 52,803 \leq 3 S_m = 53,700$$

$$c) S_i = \frac{4}{3} Q_p + P_{eb} + C_1(C_3 C_4 + C_5) \Delta T_{fi} \quad \text{REF NB-3554(c)}$$

$$S_i = \frac{4}{3} (25,930) + 10,050 + 253[(.65)(.33) + 1.2] 310$$

$$S_i = 34,573 + 10,050 + 110,940$$

$$S_i = 155,563$$

$$d) \text{ ALLOWABLE NUMBER OF CYCLES } (N_i)$$

$$S_a = \frac{S_i}{2} = \frac{155,563}{2}$$

REF FIG I-9.2

$$= 77,781$$

REF NB-3554(c)

$$\therefore N_i \approx 3.3 \times 10^3$$

REF FIG I-9.2

CYCLIC STRESS CALCULATIONS

REF NB-3554

$$a) Q_p + P_{ed} + (C_L)(C_2)(C_4)(\Delta T_{f \text{ max}}) \leq 3 S_m \quad \text{REF NB-3554($$

$$33,000 + 5866 + (253)(.46)(.18)(310)$$

$$33,000 + 5866 + 6,494$$

$$45,360 \leq 3 S_m = 53,700 \quad \text{OK}$$

SEE SUMMARY OF
STRESS INDICES &
FACTORS

$$b) S_{m(\text{max})} = Q_p + P_{eb} + (C_L)(C_3)(C_4)(\Delta T_{f \text{ max}}) \quad \text{REF NB-3554 (b)}$$

$$S_{m(\text{max})} = 33,000 + 11,554 + (253)(.55)(.18)(310)$$

$$= 33,000 + 11,554 + 7,765$$

$$= 52,318 \leq 3 S_m = 53,700$$

$$c) S_i = \frac{4}{3} Q_p + P_{eb} + C_L(C_3 C_4 + C_5) \Delta T_{fi} \quad \text{REF NB-3554(c)}$$

$$S_i = \frac{4}{3}(33,000) + 11,554 + 253[(.55)(.18) + 1.2]310$$

$$S_i = 44,000 + 11,554 + 101,881$$

$$S_i = 157,435$$

$$d) \text{ ALLOWABLE NUMBER OF CYCLES } (N_i)$$

$$S_a = \frac{S_i}{2} = \frac{157,435}{2}$$

REF FIG I-9.2

$$= 78,717$$

REF NB-3554(c)

$$\therefore N_i \approx 3.3 \times 10^3$$

REF FIG I-9.2

e) FATIGUE USAGE (I_t)

(I_t) MAY NOT BE GREATER THAN ONE

REF NB-3554 (b)
NB-3553 (a)

$$I_t = \sum \left(\frac{N_{ri}}{N_i} \right)$$

WHERE N_{ri} IS THE ESTIMATED
ACTUAL NUMBER OF CYCLES
@ ΔT_{fi} (310°F)

REF. NB 3553 (a)

$$I_t = \frac{260}{3.3 \times 10^3} = 78 \times 10^{-3}$$

$$= 0.078 \leq 1$$

THEREFORE THIS VALUE IS SUITABLE FOR THE
EXPECTED FATIGUE USAGE OF 260 CYCLES
WITH A ΔT OF 310°F

e) FATIGUE USAGE (I_t)

(I_t) MAY NOT BE GREATER THAN ONE

REF NB-3554(b)

NB-3553(u)

$$I_t = \sum \left(\frac{N_{ji}}{N_i} \right)$$

WHERE N_{ji} IS THE ESTIMATED
ACTUAL NUMBER OF CYCLES
(@ ΔT_{ji} (310°F))

REF. NB 3553(a)

$$I_t = \frac{260}{3.3 \times 10^3} = 78 \times 10^{-3}$$

$$= 0.078 \leq 1$$

THEREFORE THIS VALVE IS SUITABLE FOR THE
EXPECTED FATIGUE USAGE OF 260 CYCLES
WITH A ΔT OF 310°F

SUMMARY OF STRESS INDICIES & FACTORS.

$$T_{e1} = 4\frac{7}{16} \quad , \quad T_{e2} = 5\frac{3}{16} \quad , \quad t_e = 2\frac{1}{8} \quad \bar{\lambda} = 6.06''$$

$$C_2 = .46 \quad \text{WHEN} \quad \frac{T_{e2}}{t_e} = 2.44 \quad \& \quad \frac{\bar{\lambda}}{t_e} = 2.85 \quad \text{SEE FIG. VII THIS REPORT}$$

$$C_3 = .65 \quad \text{WHEN} \quad \frac{T_{e1}}{T_{e2}} = 2.09 \quad \& \quad \frac{\bar{\lambda}}{t_e} = 2.85 \quad \text{SEE FIG. VIII THIS REPORT}$$

$$C_4 = .33 \quad \text{WHEN} \quad \frac{T_{e1}}{t_e} = 2.09 \quad \text{SEE FIG. IX THIS REPORT}$$

$$C_5 = 1.2 \quad \text{WHEN} \quad T_{e1} = 4.44 \quad \text{SEE FIG. IX THIS REPORT}$$

$$C_6 = 253 = (\text{YOUNG'S MODULUS @ } 500^{\circ}\text{F}) (\text{COEFF. THERMAL EXPAN. @ } 500^{\circ}\text{F})$$

REF. TABLE 1-5.C

$$\Delta T_2 = 24 \quad \text{WHEN} \quad \frac{T_{e1}}{t_e} = 2.09 \quad \& \quad \frac{T_{e2}}{t_e} = 2.44 \quad \text{SEE FIG. X THIS REPORT}$$

$$Q_{T1} = 7500 \quad \text{WHEN} \quad T_{e1} = 4.44 \quad \text{SEE FIG. VI THIS REPORT}$$

$$Q_{T2} = C_6 C_2 \Delta T_2 = 2793$$

$$Q_{T3} = C_6 C_3 \Delta T_2 = 3947$$

$$S_m = 17,900 \quad \therefore 3 S_m = 53,700 \text{ psi}$$

$$Q_p = 25,930 \text{ psi} \quad \text{SEE PAGE 9 (11)}$$

$$P_{ed} = 5410 \text{ psi} \quad \text{SEE PAGE 10 (13)}$$

$$P_{eb} = 10,050 \text{ psi} \quad \text{SEE PAGE 10 (13)}$$

$$P_{et} = 10,050 \text{ psi} \quad \text{SEE PAGE 10 (13)}$$

$$\Delta T_{(max.)} = 310^{\circ}\text{F}$$

SUMMARY OF STRESS INDICIES & FACTORS

LTR-112-120

$$T_{e1} = 4\frac{7}{16} \quad T_{e2} = 5\frac{3}{16} \quad t_e = 3 \quad \bar{r} = 9.5$$

$$C_2 = .46 \quad \text{WHEN } \frac{T_{e2}}{t_e} = 1.73 \quad \& \quad \frac{\bar{r}}{t_e} = 3.17$$

SEE FIG VII

$$C_3 = .55 \quad \text{WHEN } \frac{T_{e1}}{t_e} = 1.48 \quad \& \quad \frac{\bar{r}}{t_e} = 3.17$$

SEE FIG VIII

$$C_4 = .18 \quad \text{WHEN } \frac{T_{e1}}{t_e} = 1.48$$

SEE FIG IX

$$C_5 = 1.2 \quad \text{WHEN } T_{e1} = 4.44$$

SEE FIG IX

$$C_6 = 253 = (\text{YOUNG'S MODULUS @ } 500^\circ\text{F})(\text{COEF. THERMAL EXPAN. @ } 500^\circ\text{F})$$

$$(26.1 \times 10^6)(.0501 \text{ IN/FT})(1/12 \text{ IN})(1/430^\circ\text{F}) = 253$$

$$\Delta T_2 = 4.4 \quad \text{WHEN } \frac{T_{e1}}{t_e} = 1.48 \quad \& \quad \frac{T_{e2}}{t_e} = 1.73$$

SEE FIG X

$$Q_{T1} = 7500 \text{ WHEN } T_{e1} = 4.44$$

SEE FIG VI

$$Q_{T2} = C_6 C_2 \Delta T_2 = 512$$

$$Q_{T3} = C_6 C_3 \Delta T_2 = 612$$

$$S_m = 17,900 \quad 3S_m = 53,700$$

$$Q_p = 33,000 \text{ PSI}$$

SEE PAGE 9A (12)

$$P_{ed} = 5866 \text{ PSI}$$

SEE PAGE 10A (14)

$$P_{eb} = 11554 \text{ PSI}$$

SEE PAGE 10A (14)

$$P_{et} = 11554 \text{ PSI}$$

SEE PAGE 10A (14)

$$\Delta T_{f(\text{max.})} = 310^\circ\text{F}$$

Body Joint - Ref. ASME Nuclear Power Plant Component
 Section III NB-3546.1 and ASME Boiler & Pressure Vessel Code
 Section VIII Paragraphs UA-47, UA-49, UA-50 and UA-51.

$$P_{eq} = \text{LARGER OF } \frac{16 M}{\pi G^3} \text{ OR } \frac{4 F}{\pi G^2} \quad \text{WHERE } M = 500,000 \text{ IN}^2$$

$$F = 160,000 \text{ IN}^2 \quad \text{PER CUSTOMER SPEC.}$$

$$= \frac{(16)(5.0 \times 10^5)}{\pi (5.59 \times 10^3)} = 456 \text{ psi}$$

$$= \frac{(4)(1.6 \times 10^5)}{\pi (3.14 \times 10^2)} = 650 \text{ psi} \leftarrow \text{USE THIS VALUE FOR } P_{eq}$$

DESIGN CONDITIONS					GASKET	
** FLANGE DESIGN PRESSURE $P_{FD} = P_D + P_{EQ}$				3150	OD	18.5
DESIGN PRESSURE AT 650°F P_D				2500		
EQUIVALENT PRESSURE P_{EQ}				650	ID	17
					G	17 3/4
					*b	.31
ALLOWABLE STRESS	FLANGE	$S_{mo} = 16,600$ psi AT 650°F			*y	4500
		$S_{ma} = 20,000$ psi AT 100°F				
	BOLTING	$S_{mo} = 26,900$ psi AT 650°F			*m	3.0
		$S_{ma} = 28,000$ psi AT 100°F				
* BOLTING						
$H_D = 2b \times 3.14 G m P_{FD} =$		326,717	SIZE BOLTS	2 1/2	GREATER OF W_{mi}/S_{mo} OR $W_{mi}/S_{ma} = A_m = 41.1$	
$H = .785 G^2 P_{FD} =$		779,070	No. BOLTS	16	$W_{ma} = \left(\frac{A_D + A_m}{2} \right) S_{ma} = 1,570,062$	
$W_{mi} = H_D + H =$		1,105,787	TOTAL BOLT AREA $A_B = 71.04$		TENSILE STRESS $\sigma = W_{mi}/A_B = 15,566$	
$W_{m2} = \pi b G m =$		77,790				
* FLANGE MOMENTS						
OPERATING LOAD	LOAD		LEVER ARM		MOMENT	
	$H_D = .785 B^2 P_{FD} =$		633,024	$h_D = R + 0.5 g_i =$	4.343	$M_D = H_D h_D = 2,749,223$
	$H_G =$		326,717	$h_G = \frac{1}{2}(C - G) =$	5.0	$M_G = H_G h_G = 1,633,585$
	$H_T = H - H_D =$		146,046	$h_T = \frac{1}{2}(R + g_i + h_G) =$	5.437	$M_T = H_T h_T = 794,052$
						$M_o = \sum M = 5,176,860$
BOLT LOAD	$W_{mn} = 1,570,062$		$h_T =$	5.0	$M_a = W_{mn} h_T = 7,850,313$	
* SHAPE CONSTANTS					* OTHER FACTORS	
$K_2 A/B =$	2.03	$g/g_o =$	1	$f =$	1	$t = 7"$
$T =$	1.495	$h_o = \sqrt{B g_o} =$	6.99	$e = F/h_o =$	.130	$\frac{t^3}{d} = .90$
$Z =$	1.64	$h/h_o =$	~	$d = \frac{U}{V} h o g_o^2 =$	379.6	$\frac{te+1}{T} = 1.28$
$Y =$	2.90	$F =$	.909			$L = \frac{te+1}{T} + \frac{t^3}{d} = 2.18$
$U =$	3.19	$V =$	.550			
* FLANGE STRESSES						
OPERATING LOAD	LONGITUDINAL ** HUB (USE $B_o B$) $S_H = \frac{f M_o}{L g_o^2 B} + \frac{P_D B}{4 g_o} = 15,851 + 3268 = 19,119$				$1.5 S_{mo} = 24,900$	
	RADIAL $S_R = \frac{(1.33 te + 1) M_o}{L t^2 B} = 6695$				$1.5 S_{mo} = "$	
	TANGENTIAL $S_T = \frac{Y M_o}{t^2 B} - Z S_R = 19,149 - 10,980 = 8169$				$1.5 S_{mo} = "$	
SO% OVERLOAD ON BOLTING	LONGITUDINAL HUB (USE $B_o B$) $S_H = \frac{f M_a}{L g_o^2 B} = 24,036$				$1.5 S_{ma} = 30,000$	
	RADIAL $S_R = \frac{(1.33 te + 1) M_a}{L t^2 B} = 10,152$				$1.5 S_{ma} = "$	
	TANGENTIAL $S_T = \frac{Y M_a}{t^2 B} - Z S_R = 12,388$				$1.5 S_{ma} = "$	

Body Joint - Ref. ASME Nuclear Power Plant Component
 Section III NB-3546.1 and ASME Boiler & Pressure Vessel Code
 Section VIII Paragraphs UA-47, UA-49, UA-50 and UA-51.

DESIGN CONDITIONS						GASKET	
** FLANGE DESIGN PRESSURE $P_{FD} = P_D + P_{eq}$					2500	OD	18.5
DESIGN PRESSURE AT 650 °F P_D					2500		
EQUIVALENT PRESSURE P_{eq}					0	ID	17
						G	17 3/4
						*b	.31
						*y	4500
ALLOWABLE STRESS	FLANGE	$S_{mo} = 16,600$ psi AT 650 °F				*m	3.0
		$S_{ma} = 20,000$ psi AT 100 °F					
	BOLTING	$S_{mo} = 26,900$ psi AT 650 °F					
		$S_{ma} = 28,000$ psi AT 100 °F					
* BOLTING							
$H_D = 2b \times 3.14 G m P_{FD} =$		259,299	SIZE BOLTS	2 1/2	GREATER OF W_{m1}/S_{mo} OR $W_{m1}/S_{ma} = A_m =$		
$H = .785 G^2 P_{FD} =$		619,000	No. BOLTS	16	$W_{ma} = \frac{(A_B + A_m)}{2} S_{ma} =$		
$W_{m1} = H_D + H =$		878,299	TOTAL BOLT AREA	$A_B = 71.04$	TENSILE STRESS $\sigma = W_{m1}/A_B =$		
$W_{m2} = \pi b G m =$		77,790			12,363		
* FLANGE MOMENTS							
OPERATING LOAD	LOAD		LEVER ARM		MOMENT		
	$H_D = .785 B^2 P_D =$		340,000	$h_D = R + 0.5 g_1 =$		5.75	$M_D = H_D h_D =$
	$H_G =$		259,299	$h_G = \frac{1}{2}(C - G) =$		5.00	$M_G = H_G h_G =$
	$H_T = H - H_D =$		279,000	$h_T = \frac{1}{2}(R + g_1 + h_G) =$		6.156	$M_T = H_T h_T =$
							$M_o = \sum M =$
BOLT LOAD	$W_{ma} =$		1,451,667	$h_G =$		5.00	$M_a = W_{ma} h_G =$
							7,258,336
* SHAPE CONSTANTS				* OTHER FACTORS			
$K = A/B =$	2.47	$g/g_o =$	1	$f =$	1	$t =$	5.75
$T =$	1.35	$h_o = \sqrt{B g_o} =$	6.42	$e = F/h_o =$	.1415	$t+1 =$	1.815
$Z =$	1.395	$h/h_o =$	-	$d = \frac{U}{V} h_o g_o^2 =$	286.5	$\frac{t+1}{T} =$	1.342
$Y =$	2.28	$F =$	.909				
$U =$	2.51	$V =$	.550				
* FLANGE STRESSES						ALLOWABLE **	
OPERATING LOAD	LONGITUDINAL ** HUB (USE B_o OR B_i) $S_H = \frac{f M_o}{L g_o^2 B} + \frac{P_D B}{4 g_o} =$					19,413 + 2625 =	22,038
	RADIAL $S_R = \frac{(1.33 t e + 1) M_o}{L t^2 B} =$					11,949	
	TANGENTIAL $S_T = \frac{Y M_o}{t^2 B} - Z S_R =$					26,147 - 16,669 =	9478
50% OVERLOAD ON BOLTING	LONGITUDINAL HUB (USE B_o OR B_i) $S_H = \frac{f M_a}{L g_o^2 B} =$					28,314	
	RADIAL $S_R = \frac{(1.33 t e + 1) M_a}{L t^2 B} =$					17,428	
	TANGENTIAL $S_T = \frac{Y M_a}{t^2 B} - Z S_R =$					13,823	

FLANGE - OUTLET END LFR-112-120
 CONSOLIDATED CONTROLS DWG
 J-72383 REV C

Body Joint - Ref. ASME Nuclear Power Plant Component
 Section III NB-3546.1 and ASME Boiler & Pressure Vessel Code
 Section VIII Paragraphs UA-47, UA-49, UA-50 and UA-51.

DESIGN CONDITIONS					GASKET		
** FLANGE DESIGN PRESSURE $P_{FD} = P_D + P_{eq}$				2500	OD	17	
DESIGN PRESSURE AT °F P_D				2500			
EQUIVALENT PRESSURE P_{eq}				0	ID	15	
					G	16	
					*b	.354	
ALLOWABLE STRESS	FLANGE	$S_{mo} = 16,600$ psi AT 650°F				*y	4500
		$S_{ma} = 20,000$ psi AT 100°F					
	BOLTING	$S_{mo} = 26,900$ psi AT 650°F				*m	3.0
		$S_{ma} = 28,800$ psi AT 100°F					
* BOLTING							
$H_g = 26 \times 3.14 G m P_{FD} = 266,910$		SIZE BOLTS	2"	GREATER OF W_{m1}/S_{mo} OR $W_{m2}/S_{ma} = A_m = 28.6$			
$H = .785 G^2 P_{FD} = 503,000$		No. BOLTS	16	$W_{ma} = \left(\frac{A_B + A_m}{2} \right) S_{ma} = 1,045,745$			
$W_{m1} = H_g + H = 769,910$				TENSILE STRESS $\sigma = W_{m1}/A_B = 17,498$			
$W_{m2} = \pi b G n = 80,073$		TOTAL BOLT AREA $A_B = 44.0$					
* FLANGE MOMENTS							
OPERATING LOAD	LOAD		LEVER ARM		MOMENT		
	$H_D = .785 B^2 P_D = 196,000$		$h_D = R + 0.5 g_1 = 4.563$		$M_D = H_D h_D = 895,000$		
	$H_g = 266,910$		$h_g = \frac{1}{2}(C - G) = 3.25$		$M_g = H_g h_g = 867,458$		
	$H_T = H - H_D = 307,000$		$h_T = \frac{1}{2}(R + g_1 + h_g) = 4.75$		$M_T = H_T h_T = 1,460,000$		
					$M_o = \sum M = 3,222,458$		
BOLT LOAD	$W_{ma} = 1,045,745$		$h_g = 3.25$		$M_u = W_{ma} h_g = 3,398,671$		
* SHAPE CONSTANTS				* OTHER FACTORS			
$K = A/B = 2.65$	$g/g_o = 1$	$f = 1$	$t = 4.875$		$\frac{t^3}{d} = .417$		
$T = 1.30$	$h_o = \sqrt{B g_o} = 5.8$	$e = F/h_o = .156$	$t e + 1 = 1.77$		$L = \frac{t e + 1}{T} + \frac{t^3}{d} = 1.777$		
$Z = 1.33$	$h/h_o = \sim$	$d = \frac{U}{V} h o g_o^2 = 278$	$\frac{t e + 1}{T} = 1.36$				
$Y = 2.12$	$F = .909$						
$U = 2.32$	$V = .550$						
* FLANGE STRESSES					ALLOWABLE **		
OPERATING LOAD	LONGITUDINAL ** HUB (USE $B_o B_i$) $S_H = \frac{f M_o}{L g_1^2 B} + \frac{P_D g}{4 g_o} = 15,993 + 1850 = 17,843$				$1.5 S_{ho} = 24,900$		
	RADIAL $S_R = \frac{(1.33 t e + 1) M_o}{L t^2 B} = 15,445$				$1.5 S_{mo} = "$		
	TANGENTIAL $S_T = \frac{Y M_o}{t^2 B} - Z S_R = 28,746 - 20,542 = 8,204$				$1.5 S_{mo} = "$		
50% OVERLOAD ON BOLTING	LONGITUDINAL HUB (USE $B_o B_i$) $S_H = \frac{f M_a}{L g_1^2 B} = 16,798$				$1.5 S_{na} = 30,000$		
	RADIAL $S_R = \frac{(1.33 t e + 1) M_a}{L t^2 B} = 16,224$				$1.5 S_{na} = "$		
	TANGENTIAL $S_T = \frac{Y M_a}{t^2 B} - Z S_R = 8,618$				$1.5 S_{na} = "$		

STUD STRESS INTENSITIES

FORMULAS USED

$$\sigma = \frac{F}{AN} = \frac{F}{\frac{\pi}{4} D^2 N} = \frac{4F}{\pi D^2 N}$$

$$\tau = \frac{T_s}{Z_p N} = \frac{F T_F}{\frac{\pi}{16} D^3 N} = \frac{16 F T_F}{\pi D^3 N}$$

$$S = 2 \tau_{MAX} = 2 \sqrt{(\sigma/2)^2 + \tau^2} = \sqrt{\sigma^2 + 4\tau^2}$$

a) BODY STUDS (2 1/2" - 8 UN)

1) TENSILE

$$\sigma = \frac{(4)(W_{m1})}{\pi D^2 N} = \frac{(4)(1,105,787)}{\pi (2.346)^2 (16)} = 15,988 \text{ PSI}$$

2) TORSIONAL SHEAR

$$\tau = \frac{16 W_{m1} T_F}{\pi D^3 N} = \frac{16 (1,105,787) (.2277)}{\pi (2.346)^3 16} = 6207 \text{ PSI}$$

$$\therefore S = \sqrt{(15,988)^2 + 4(6207)^2} = 20,242$$

ALLOWABLE: $2S_m @ 500^\circ\text{F}$
 $2S_m = 54,200 \text{ psi}$
 REF USAS B31.7 ART 105.2.6

F = APPLIED FORCE
 A = STUD ROOT AREA
 D = STUD ROOT DIA
 N = NO. OF STUDS
 T_s = TORQUE
 Z_p = POLAR SECTION MODULUS
 T_F = THREAD FACTOR
 M = .15

CALCULATION OF NUT TORQUES

LTR-112-120

NUT TORQUE (T_N)

$$T_N = F (T_F + N_F)$$

$F = F_0$ = AXIAL FORCE DUE TO FLANGE DESIGN PRESSURE

$F = F_s$ = AXIAL FORCE DUE TO AN ASSUMED STRESS

T_F = THREAD FACTOR

N_F = NUT CONTACT FACTOR

a) AXIAL FORCE (F_0) ~ $P_{FD} = 3150 \text{ psi}$

BODY STUD NUT

$$F_0 = \frac{W_{mi}}{N} = \frac{1,105,787}{16} = 69,112 \#$$

b) AXIAL FORCE (F_s) REQUIRED TO PRODUCE A STRESS EQUAL TO 50% OF THE YIELD STRENGTH (S_Y) AT 500°F

BODY STUD

$$F_s = .5 S_Y A$$

$$S_Y @ 500^\circ\text{F} = 81,500 \text{ psi}$$

$$A = \frac{\pi}{4} D^2 = \frac{\pi}{4} (2.346)^2 = 4.32$$

$$F_s = (.5)(81,500)(4.32) = 176,146 \#$$

c) THREAD FACTOR *

$$T_F = \frac{d}{2} \left[\frac{\pi d \mu \sec \beta + L}{\pi d - \mu L \sec \beta} \right]$$

d = MEAN THREAD CONTACT DIA.

μ = COEFF. OF FRICTION

β = THD ANGLE

L = THD LEAD

LOCATION	THREAD	β	d	L	T_F	
					$\mu = .15$	$\mu = .20$
BODY	2½-8UN	30°	2.392	.125	.2277	.2972

* REF ELEMENTS OF MACHINE DESIGN
KIMBALL & BARR FIRST EDITION
EQN. 12 PAGE 162

d) NUT CONTACT FACTOR (N_F)

$$N_F = \frac{D_o + D_i}{4} \mu$$

D_o = O.D NUT CONTACT
 D_i = I.D NUT CONTACT

LOCATION	D_o HEX FLATS	D_i THD DIA	N_F	
			$\mu = .15$	$\mu = .20$
BODY	3 7/8	2 1/2	.2390	.3188

e) NUT TORQUES

$$T_N = F (T_F + N_F)$$

	NUT TORQUES IN-LB					
	AXIAL FORCE (F_o)		AXIAL FORCE (F_s)		RECOMMENDED NUT TORQUE	
	$\mu = .15$	$\mu = .20$	$\mu = .15$	$\mu = .20$	MIN.	MAX.
BODY	32,255	42,573	82,080	109,000	50,000	55,000

POPPET

LTR-112-120

CONSOLIDATED CONTROLS DWG. E-72385 REV C

1) PRIMARY MEMBRANE STRESS

REF. NB-3324.1 (CYLINDRICAL SHELL)

$$P_m = \frac{P_s R}{t_m} + 0.5 P_s$$

$$P_m = \frac{(3000)(5.064)}{.863} + 0.5(3000)$$

$$P_m = 17,614 + 1500 = 19,114 \text{ OK}$$

ALLOWABLE = S_m
REF NB-3546.2
 $S_m = 23,300 \text{ psi}$

2) BEARING STRESS ON SEAT

$$S_b = \frac{L_i}{A_c} \quad \text{WHERE} \quad \begin{array}{l} L_i = \text{LOAD ON SEAT} \\ A_c = \text{CONTACT SEAT AREA} \end{array}$$

a) SEAT CONTACT AREA

$$A_c = \frac{\pi D_o^2}{4} - \frac{\pi D_i^2}{4}$$

$$= \frac{\pi}{4} (10.206^2 - 10.126^2)$$

$$= \frac{\pi}{4} (104.16 - 102.54) = 1.276 \text{ SQ IN.}$$

b) LOAD ON SEAT @ DESIGN PRESSURE

$$L_1 = P_d [\text{PISTON AREA} - \text{STEM AREA}]$$

$$= 2500 \left[\frac{\pi}{4} (4.75)^2 - \frac{\pi}{4} (2.25)^2 \right]$$

$$= 2500 [17.72 - 3.98]$$

$$= 34,360 \#$$

∴ BEARING STRESS

$$S_b = \frac{L_1}{A_c} = \frac{34,360}{1.276}$$

$$S_b = 26,982 \text{ PSI}$$

 $S_b = S_y$ OF BASE MATERIAL

REF NB-3227.1

$$S_y = 35,000 > 26,982 \quad \underline{\text{OK}}$$

WEB

CONSOLIDATED CONTROLS

LTR-112-120

DWG.

J-72386 REV C

SHEET 1 of 2

- 1) STRESS IN FLAT PLATE, EDGES FIXED, UNIFORM LOAD OVER CONCENTRIC CIRCULAR AREA.

REF. FORMULA FOR STRESS & STRAIN
"ROARK" ~ TABLE X CASE # 7

$$S_r = \frac{3W}{2\pi t^2} \left[1 - \frac{r_0^2}{2a^2} \right]$$

$$W = P_0 (A)$$

$$A = \frac{\pi}{4} \left[(\text{PISTON DIA})^2 - (\text{STEM DIA})^2 \right]$$

$$A = \frac{\pi}{4} (10.07^2 - 2.247^2) = 75.68$$

$$\therefore W = (2500)(75.68) = 189,194 \#$$

$$S_r = \frac{(3)(189,194)}{2\pi(5.5)^2} \left[1 - \frac{(5.03)^2}{2(9.25)^2} \right]$$

$$S_r = 2986 (.852) = 2545 \text{ psi}$$

ALLOWABLE: S_m

OK

$$S_m = 16,600$$

2) SHEAR STRESS @ PORTS THRU WEB

$$S_s = \frac{\text{LOAD ON WEB}}{\text{METAL AREA}}$$

$$S_s = \frac{L}{A}$$

$$L = P_b \left[\left(\text{AREA @ ID OF VALVE} \right) - \left(\text{FLOW AREA THRU WEB} \right) \right]$$

$$L = 2500 \left[\frac{\pi 16^2}{4} - 80 \text{ sq in} \right]$$

$$L = 302,655 \#$$

NOTE: FLOW AREA THRU WEB OBTAINED FROM FULL SIZE LAYOUT WITH A PLANIMETER.

TOTAL METAL AREA

$$\text{METAL LENGTH} = (3") (4 \text{ PLACES}) = 12"$$

$$\text{HOLE LENGTH} = (3)(5/8") + 2(7/8") + 2(1 5/8") = 6 7/8"$$

$$\therefore \text{RETANGULAR METAL LENGTH} = 12" - 6 7/8" = 5 1/8"$$

$$\text{RETANGULAR METAL AREA} = (5 1/8)(6) = 30 3/4 \text{ sq in}$$

$$\begin{aligned} \text{METAL REMOVED FOR FLOW CONTOUR} &\approx 3 \text{ sq in / WEB} \\ &= (3)(4) = 12 \text{ sq in} \end{aligned}$$

$$\therefore \text{TOTAL METAL AREA} = (30 3/4 - 12) = 18 3/4 \text{ sq in.}$$

$$\therefore S_s = \frac{L}{A} = \frac{302,655}{18.75}$$

$$S_s = 16,142 \text{ psi}$$

OK

$$\text{ALLOWABLE} = S_m$$

$$\begin{aligned} S_m &= 16,600 \\ \text{REF NB-3546.3} \end{aligned}$$

CONE

LTR--112-120

CONSOLIDATED CONTROLS DWG E-72384 Rev C

8 1/4" BOLT CIRCLE
12 - 3/4-16 UNF CAP SCREWS

$$P_0 = 2500 \text{ psi}$$

$$P = 10,000 \text{ psi for } .002 \text{ sec.}$$

- 1) TENSILE STRESS IN CAP SCREWS DUE TO THE GREATER OF THE REQUIRED BOLT LOADS DUE TO DESIGN CONDITIONS OR REQUIRED BOLT LOAD FOR GASKET SEATING.

a) GASKET DATA

$$OD = 5.52$$

$$ID = 4.75$$

$$G = 5.135$$

$$b = .193$$

$$y = 4500$$

$$m = 3.00$$

b) ALLOWABLE STRESSES

$$S_{mo} = 26,900 \text{ psi @ } 650^\circ \text{F}$$

$$S_{ma} = 28,300 \text{ psi @ } 100^\circ \text{F}$$

c) GASKET LOAD

$$H_g = 2 \pi b G m P_0$$

$$= (2) \pi (.193) (5.135) (3.00) (2500)$$

$$= 46,702 \text{ \#}$$

d) HYDROSTATIC END FORCE

$$\begin{aligned}
 H &= (.785)(G^2)(P_0) \\
 &= (.785)(5.135)^2(2500) \\
 &= 51,748 \#
 \end{aligned}$$

e) REQUIRED BOLT LOAD FOR DESIGN COND.

$$\begin{aligned}
 W_{m1} &= H_G + H \\
 &= 46,702 + 51,748 \\
 &= 98,450 \#
 \end{aligned}$$

f) REQUIRED BOLT LOAD FOR GASKET SEATING

$$\begin{aligned}
 W_{m2} &= \pi(b)(G)(y) \\
 &= \pi(.193)(5.135)(4500) \\
 &= 14,011
 \end{aligned}$$

g) CAP SCREW DATA

(12) 3/4" - 16 UNC

TENSILE AREA = .373 sq in / SCREW

TOTAL TENSILE AREA = (12)(.373)

$$A_s = 4.48$$

$$\therefore \underline{\underline{TENSILE STRESS}} = \frac{W_{m1}}{A_s}$$

$$= \frac{98,450}{4.48}$$

$$= 21,975 \text{ psi} \quad \underline{\text{OK}}$$

ALLOWABLE $S_m @ 65$
 $S_m = 26,900$

- 2) TENSILE STRESS IN CAP SCREWS DUE TO A SURGE OF PRESSURE OF 10,000 psi FOR APPROXIMATELY .002 SEC.

$$\sigma = \frac{\text{LOAD ON CAP SCREWS}}{A_s}$$

$$\begin{aligned}\text{LOAD} &= (P)(\text{AREA @ CONE GASKET}) \\ &= (10,000) \left(\frac{\pi}{4} \right) (4.750)^2 \\ &= 177,205 \# \end{aligned}$$

$$\text{AREA OF CAP SCREWS} = 4.48 \text{ sq in.}$$

$$\sigma = \frac{177,205}{4.48} = 39,555 \text{ psi}$$

THIS STRESS IS CONSIDERED ADEQUATE DUE TO THE EXTREMELY SHORT DURATION OF TIME (.002 SEC) THE CAP SCREWS ARE EXPOSED TO IT. ALSO IT IS LESS THAN THE 85,000 psi YIELD STRENGTH OF THE MATERIAL AND IS LESS THAN THE STRESS RUPTURE PROPERTIES OF 56,000 psi @ 1200°F FOR 100 HRS.

REF, ASME - BOILER & PRESSURE VESSEL CODE SECTION II
PAGE 386 ~ SA-453.

LTR-112-120

TABLE NB-3542-1
VALVE BODY MINIMUM WALL THICKNESS

Inside Diameter d_m , In.	Primary Pressure Rating P_r	Minimum Wall Thickness, t_m - In.						
		150 lb.	300 lb.	400 lb.	600 lb.	900 lb.	1500 lb.	2500 lb.
1		.25	.25	.25	.25	.28	.28	.44
2		.25	.25	.25	.25	.46	.46	.79
3		.25	.28	.31	.31	.42	.66	1.14
4		.25	.31	.38	.38	.51	.83	1.47
5		.28	.38	.44	.44	.63	1.02	1.81
6		.28	.38	.44	.50	.74	1.21	2.15
7		.30	.41	.50	.57	.83	1.41	2.51
8		.31	.44	.56	.63	.93	1.59	2.83
9		.33	.47	.63	.70	1.03	1.76	3.17
10		.34	.50	.69	.77	1.13	1.94	3.51
11		.36	.53	.72	.85	1.24	2.12	3.85
12		.38	.56	.75	.92	1.35	2.31	4.19
13		.40	.61	.81	.97	1.46	2.50	4.52
14		.42	.65	.84	1.03	1.56	2.69	4.86
15		.43	.68	.88	1.11	1.67	2.88	5.20
16		.45	.71	.91	1.18	1.77	3.06	5.54
17		.46	.75	.94	1.25	1.86	3.24	5.88
18		.48	.78	1.00	1.31	1.96	3.42	6.22
19		.50	.81	1.07	1.39	2.07	3.61	6.55
20		.51	.84	1.10	1.46	2.17	3.79	6.89
21		.53	.88	1.13	1.53	2.28	3.97	7.23
22		.54	.91	1.17	1.59	2.38	4.15	7.57
23		.56	.94	1.20	1.66	2.48	4.33	7.91
24		.57	.97	1.24	1.72	2.59	4.51	8.25
25		.59	1.00	1.28	1.79	2.69	4.69	8.59
26		.61	1.04	1.32	1.85	2.79	4.87	8.92
27		.62	1.07	1.36	1.91	2.89	5.05	9.26
28		.64	1.10	1.39	1.98	2.99	5.24	9.60
29		.66	1.14	1.43	2.04	3.09	5.42	9.94
30		.67	1.17	1.47	2.11	3.19	5.60	10.28
31		.69	1.20	1.51	2.17	3.30	5.78	10.62
32		.71	1.23	1.54	2.23	3.40	5.96	10.95
33		.72	1.27	1.58	2.30	3.50	6.14	11.29
34		.74	1.30	1.62	2.36	3.60	6.32	11.63
35		.75	1.33	1.65	2.43	3.70	6.50	11.97
36		.77	1.37	1.69	2.49	3.80	6.68	12.31
37		.79	1.40	1.73	2.55	3.90	6.87	12.65
38		.80	1.43	1.77	2.62	4.01	7.05	12.98
39		.82	1.47	1.81	2.68	4.11	7.23	13.32
40		.84	1.50	1.84	2.75	4.21	7.41	13.66
41		.85	1.53	1.88	2.81	4.31	7.59	14.00
42		.87	1.56	1.92	2.88	4.41	7.77	14.34
43		.88	1.60	1.96	2.94	4.51	7.95	14.68
44		.90	1.63	1.99	3.00	4.61	8.13	15.01
45		.92	1.66	2.03	3.07	4.72	8.32	15.35
46		.93	1.70	2.07	3.13	4.82	8.50	15.69
47		.95	1.73	2.11	3.20	4.92	8.68	16.03
48		.97	1.76	2.14	3.26	5.02	8.86	16.37
49		.98	1.80	2.18	3.32	5.12	9.04	16.71
50		1.00	1.83	2.22	3.39	5.22	9.22	17.04

LTR-112-120

FIG. NB-3545.1-1 STANDARD CALCULATION PRESSURE VS. PRIMARY-PRESSURE RATING FOR FLANGED-END VALVES

↑
1455

FIG. NB-3545.2-3

NB-3000-DESIGN

FIG. NB-3545.2-3

LTR-112-120

FIG III

FIG. NB-3545.2-4

$d_i = 16$
 $F_b = 290$

SEE FIG NB-3545.2-5

172-112-120

FIG. NB-3545.2-6 SECTION III - NB-REQUIREMENTS FOR CLASS I COMPONENTS

FIG. NB-3545.26

LTR-112-120

FIG. NB-3545.2(c)-2 SECTION III - NB-REQUIREMENTS FOR CLASS 1 COMPONENTS

FIG. NB-3545.2(c)-2 THERMAL STRESS, Q_T , VS. WALL THICKNESS
(Based on the Equivalent Linear Gradient for a 500F Fluid Temperature Rise at 100F/HR)

Fig VI

FIG. NB-3545.2(c)-3 THERMAL STRESS INDEX VERSUS THICKNESS CONTINUITY RUN/BRANCH

FIG. NB-3546.2(d)-5 THERMAL STRESS INDEX VERSUS THICKNESS CONTINUITY RUN/BRANCH

Fig. NB-3545.2(c)-5

Fig. NB-3545.2(c)-6

FIG IX

FIG. NB-3545.2(c)-4 MAXIMUM TEMPERATURE DIFFERENCE BETWEEN AVERAGE TEMPERATURE OF THE THICK (T_{e1}) WALL AND THE AVERAGE TEMPERATURE OF THE THIN (t_e) WALL AT 100°F/HOUR FOR FERRITIC AND AUSTENITIC MATERIALS

17R-112-120

FIG. 19-2

SECTION III - APPENDIX I-STRESS TABLES

FIG. 19-2 DESIGN FATIGUE CURVE FOR AUSTENITIC STEELS NICKEL-CHROMIUM-IRON ALLOY, NICKEL-IRON-CHROMIUM ALLOY AND NICKEL-COPPER ALLOY

VALUE OF S_n , PSI

17R-112-120

The table below lists many commonly used gasket materials and indicates the facings with which they may be used. The suggested design values of m and y have generally proved satisfactory in actual serv-

ice when using the effective gasket seating width b given in Design Table 3. The design values and other details given in this table are suggested only and are not mandatory.

GASKET MATERIALS (UA-47.1)

DESIGN TABLE 2

Gasket Material		Gasket Factor m	Design Sealing Stress Min. y	Sketch of Section	Facing Limitations (See Design Table 3)	
Rubber without fabric or a high percentage of asbestos fiber Shore Durometer below 75 Shore Durometer 75 or higher		0.50 1.00	0 200		1a, 1b, 4, 6 only	Use Column II of Design Table 3
Asbestos with a suitable binder for the operating conditions	1/8" thick	2.00	1600
	1/16" thick	2.75	3700			
	1/32" thick	3.50	6500			
Solid "TEFLON"	1/8" thick	2.00	1600
	1/16" thick	2.75	3700			
	1/32" thick	3.50	6500			
Rubber with cotton fabric insertion		1.25	400		None	
Rubber with asbestos fabric insertion, with or without wire reinforcement	3-ply	2.25	2200
	2-ply	2.50	2900			
	1-ply	2.75	3700			
Vegetable fiber		1.75	1100		1a, 1b, 4, 6	
Spiral-wound metal, asbestos filled		Carbon Steel Stainless or Monel	2.50 3.00	2900 4500		1a only
Corrugated metal, asbestos inserted	Soft aluminum	2.50	2900
	Soft copper or brass	2.75	3700			
or Corrugated metal, jacketed asbestos filled	Iron or soft steel	3.00	4500
	Monel or 4-6% chrome	3.25	5500			
Stainless steels	Soft aluminum	2.75	3700
	Soft copper or brass	3.00	4500			
	Iron or soft steel	3.25	5500			
	Monel or 4-6% chrome	3.50	6500			
	Stainless steels	3.75	7600			
Flat metal jacketed asbestos filled	Soft aluminum	3.25	5500
	Soft copper or brass	3.50	6500			
	Iron or soft steel	3.75	7600			
	Monel	3.50	8000			
	4-6% chrome	3.75	9000			
	Stainless steels	3.75	9000			
Grooved metal	Soft aluminum	3.25	5500
	Soft copper or brass	3.50	6500			
	Iron or soft steel	3.75	7600			
	Monel or 4-6% chrome	3.75	9000			
	Stainless steels	4.25	10100			
Solid flat metal	Lead	2.00	1400
	Soft aluminum	4.00	8800			
	Soft copper or brass	4.75	13000			
	Iron or soft steel	5.50	18000			
	Monel or 4-6% chrome	6.00	21800			
	Stainless steels	6.50	26000			
Ring Joint	Iron or soft steel	5.50	18000
	Monel or 4-6% chrome	6.00	21800			
	Stainless steels	6.50	26000			

* The surface of a gasket having a lap should be against the smooth surface of the facing and not against the rabbet.

NOTE: The gasket factors listed apply only to joints in which the gasket is contained entirely within the edges of the bolt holes.

FIG XII

54

VALUES OF f (UA-51.6)

DESIGN CHART 5

$f = .1$ (MINIMUM)

→ $f = 1$ FOR HUBS OF UNIFORM THICKNESS ($h/g_0 = 1$)

$f = 1$ FOR LOOSE HUBBED FLANGES

FIG XIII

LTR-112-120

g_1/g_0

Fig XIV

FACTORS INVOLVING $K = A/B$

DESIGN TABLE 4 (Cont'd)

K	T	Z	Y	U
1.80	1.58	1.89	3.47	3.82
1.81	1.58	1.88	3.4	3.78
1.82	1.58	1.86	3.41	3.75
1.83	1.57	1.85	3.38	3.72
1.84	1.57	1.84	3.35	3.69
1.85	1.56	1.83	3.33	3.65
1.86	1.56	1.81	3.30	3.62
1.87	1.56	1.80	3.27	3.59
1.88	1.55	1.79	3.24	3.56
1.89	1.55	1.78	3.22	3.54
1.90	1.54	1.77	3.19	3.51
1.91	1.54	1.75	3.17	3.48
1.92	1.54	1.74	3.14	3.45
1.93	1.53	1.73	3.12	3.43
1.94	1.53	1.72	3.09	3.40
1.95	1.53	1.71	3.07	3.38
1.96	1.52	1.70	3.05	3.35
1.97	1.52	1.69	3.03	3.33
1.98	1.51	1.68	3.01	3.30
1.99	1.51	1.68	2.98	3.28
2.00	1.51	1.67	2.96	3.26
2.01	1.50	1.66	2.94	3.23
2.02	1.50	1.65	2.92	3.21
2.04	1.49	1.63	2.88	3.17
2.06	1.48	1.62	2.85	3.13
2.08	1.48	1.60	2.81	3.09
2.10	1.47	1.59	2.78	3.05
2.12	1.46	1.57	2.74	3.01
2.14	1.46	1.56	2.71	2.97
2.16	1.45	1.55	2.67	2.94
2.18	1.44	1.53	2.64	2.90
2.20	1.44	1.52	2.61	2.87
2.22	1.43	1.51	2.58	2.84
2.24	1.42	1.50	2.56	2.81
2.26	1.41	1.49	2.53	2.78

K	T	Z	Y	U
2.28	1.41	1.48	2.50	2.75
2.30	1.40	1.47	2.48	2.72
2.32	1.40	1.46	2.45	2.69
2.34	1.39	1.45	2.43	2.67
2.36	1.38	1.44	2.40	2.64
2.38	1.38	1.43	2.38	2.61
2.40	1.37	1.42	2.36	2.59
2.42	1.36	1.41	2.33	2.56
2.44	1.36	1.40	2.31	2.54
2.46	1.35	1.40	2.29	2.52
2.48	1.35	1.39	2.27	2.50
2.50	1.34	1.38	2.25	2.47
2.53	1.33	1.37	2.22	2.44
2.56	1.32	1.36	2.19	2.41
2.59	1.31	1.35	2.17	2.38
2.62	1.30	1.34	2.14	2.35
2.65	1.30	1.33	2.12	2.32
2.68	1.29	1.32	2.09	2.30
2.71	1.28	1.31	2.07	2.27
2.74	1.27	1.31	2.04	2.25
2.77	1.26	1.30	2.02	2.22
2.80	1.26	1.29	2.00	2.20
2.83	1.25	1.28	1.98	2.17
2.86	1.24	1.28	1.96	2.15
2.89	1.23	1.27	1.94	2.13
2.92	1.22	1.27	1.92	2.11
2.95	1.22	1.26	1.90	2.09
2.98	1.21	1.25	1.88	2.07
3.02	1.20	1.25	1.86	2.04
3.06	1.19	1.24	1.83	2.01
3.10	1.18	1.23	1.81	1.99
3.14	1.17	1.23	1.79	1.97
3.18	1.16	1.22	1.77	1.94
3.22	1.16	1.21	1.75	1.92
3.26	1.15	1.21	1.73	1.90

K	T	Z	Y	U
3.30	1.14	1.20	1.71	1.88
3.34	1.13	1.20	1.69	1.86
3.38	1.12	1.19	1.67	1.84
3.42	1.11	1.19	1.66	1.82
3.46	1.11	1.18	1.64	1.80
3.50	1.10	1.18	1.62	1.78
3.54	1.09	1.17	1.61	1.76
3.58	1.08	1.17	1.59	1.75
3.62	1.07	1.16	1.57	1.73
3.66	1.07	1.16	1.56	1.71
3.70	1.06	1.16	1.55	1.70
3.74	1.05	1.15	1.53	1.68
3.78	1.05	1.15	1.52	1.67
3.82	1.04	1.15	1.50	1.65
3.86	1.03	1.14	1.49	1.64
3.90	1.03	1.14	1.48	1.62
3.94	1.02	1.14	1.46	1.61
3.98	1.01	1.13	1.45	1.60
4.00	1.01	1.13	1.45	1.59
4.05	1.00	1.13	1.43	1.57
4.10	.99	1.13	1.42	1.56
4.15	.99	1.12	1.40	1.54
4.20	.98	1.12	1.39	1.53
4.25	.97	1.12	1.38	1.51
4.30	.96	1.11	1.36	1.50
4.35	.96	1.11	1.35	1.48
4.40	.95	1.11	1.34	1.47
4.45	.94	1.11	1.33	1.46
4.50	.94	1.10	1.31	1.44
4.55	.93	1.10	1.30	1.43
4.60	.92	1.10	1.29	1.42
4.65	.92	1.10	1.28	1.41
4.70	.91	1.09	1.27	1.39
4.75	.90	1.09	1.26	1.38
4.80	.90	1.09	1.25	1.37
4.85	.89	1.09	1.24	1.36
4.90	.88	1.09	1.23	1.35
4.95	.88	1.08	1.22	1.34
5.00	.87	1.08	1.21	1.33

2.03

EFFECTIVE GASKET WIDTH (UA-47.2)

Facing Sketch (Exaggerated)	Basic Seating Width b_o	
	Column I	Column II
1a	$\frac{N}{2}$	$\frac{N}{2}$
1b	$\frac{w+T}{2}$ $\frac{w+N}{4} \text{ max}$	$\frac{w+T}{2}$ $\frac{w+N}{4} \text{ max}$
2	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
3	$\frac{w}{2}$ $\frac{N}{4} \text{ min}$	$\frac{w+N}{4}$ $\frac{3N}{8} \text{ min}$
4	$\frac{3N}{8}$	$\frac{7N}{16}$

EFFECTIVE GASKET SEATING WIDTH b

$$b = b_o \text{ when } b_o \leq \frac{1}{4}$$

$$b = \frac{\sqrt{b_o}}{2}, \text{ when } b_o > \frac{1}{4}$$

DESIGN TABLE 3

Facing Sketch (Exaggerated)	Basic Seating Width b_o	
	Column I	Column II
5	$\frac{3N}{8}$	$\frac{7N}{16}$
6	$\frac{N}{4}$	$\frac{3N}{8}$
7	$\frac{N}{4}$	$\frac{3N}{8}$
8	$\frac{w}{8}$	

LOCATION OF GASKET LOAD REACTION