

CALCULATION TITLE PAGE



BRAIDWOOD STATION

Unit 1 Unit 2 Unit 0

CALCULATION NO: BRW 96-015

DESCRIPTION CODE: M14

DISCIPLINE CODE: Mechanical

SYSTEM CODE: SI

Title: Verification of Capability for Braidwood and Byron 1(2)SI8802A & B Valves Susceptible to Pressure Locking

Safety Related

Augmented Quality

Non-Safety Related

REFERENCE NUMBERS: (C011 Panel)

Type	Number	Type	Number
AEDV			
PROG			
RNID			
DCAT			
SSYS			

COMPONENT EPN: (C014 Panel)

DOCUMENT NUMBERS: (C012 Panel) (Calc. References)

EPN	Component Type	Doc Type	Sub Type	Document Number
1SI8802A	L05			
1SI8802B	L05			
2SI8802A	L05			
2SI8802B	L05			

REMARKS:

REV. No.	REVISING ORGANIZATION	APPROVED (Print & Sign)	DATE
1	Site Engineering	<i>Bruce J. Acas</i>	11/28/97



CALCULATION REVISION PAGE

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REVISION SUMMARIES

REV: 1

REVISION SUMMARY: Revised to reflect lower (tested) valve factor and increased margin between the predicted pressure locking thrust and the motor gearing capability. Revised pages 1, 2, 3, 6, 7, 8, 9, 11, 14, A3 and A4.

Electronic Calculation Data Files: (Program Name, Version, File Name ext/size/date/hour/:min)

Prepared By:
(Print/Sign)

R.C. BEDFORD / *RC Bedford*

Date: 11/25/97

Reviewed By:
(Print/Sign)

J. Tolar / *J Tolar*

Date: 11-25-97

Type of Review: Detailed Alternate Test

Do any ASSUMPTIONS / ENGINEERING JUDGMENTS required later verification? Yes No
Tracked By: _____

REV:

REVISION SUMMARY:

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Date:

Reviewed By:
(Print/Sign)

Date:

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I. PURPOSE/OBJECTIVE

The purpose of this calculation is to verify the capability of certain MOVs which have been determined to be susceptible to the pressure locking phenomena. The MOVs are installed in the Safety Injection system at Braidwood and Byron Stations.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

The methodology for calculating the thrust required to open the MOVs under the pressure locking scenario is based on the Reference 1 (Roark's) engineering handbook. This methodology has been verified in accordance with a test performed on a similar valve at Braidwood Station and is documented in Reference 7. The methodology determines the total force required to open the valve under a pressure locking scenario by solving for the four components to this required force. The four components of the force are the Pressure Locking Component, the Static Unseating Component, the Piston Effect Component, and the "Reverse Piston Effect" component. These components are determined using the following steps.

Pressure Locking Component of Force Required to Open the Valve

The valve disc is modeled as two plates attached at the center by a hub which is concentric with the valve disc. A plane of symmetry is assumed between the valve discs. This plane of symmetry is considered fixed in the analysis.

The pressure force is assumed to act uniformly upon the inner surface of the disc between the hub diameter and the outer disc diameter. The outer edge of the disc is assumed to be unimpeded and allowed to deflect away from the pressure force. In addition, the disc hub is allowed to stretch. The total displacement at the outer edge of the valve disc due to shear and bending and due to hub stretch are calculated using the reference 1 equations.

An evenly distributed force is assumed to act between the valve seat and the outer edge of the valve disc. This force acts to deflect the outer diameter of the valve disc inward and to compress the disc hub. The pressure force is reacted to by an increase in this contact force between the valve disc and seats. The valve body seats are conservatively assumed to be fixed. Therefore, the deflection due to the known pressure load must be balanced by the deflection due to the unknown seat load. The deflection due to the pressure force is first calculated. Then, the reference 1 equations are used to determine the contact force between the seat and disc which results in a deflection which is equal and opposite to the deflection due to the pressure force.

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Pressure Locking Component of Force Required to Open the Valve (Cont.)

The coefficient of friction between the seat and disc is determined based on best available data. When DP test data is available, the friction coefficient is based on the measured close valve factor. Otherwise, the seat friction coefficient is based on the nominal valve factor from DP testing of similar valves. The stem force required to overcome the contact load between the seat and disc which opposes the pressure force is equal to:

$$(\text{seat load}) \times [(\text{seat } \mu) \cos(\text{seat angle}) - \sin(\text{seat angle})] \times 2 \text{ (for two disc faces).}$$

Static Unseating Force

The static unseating force represents the open packing load and pullout force due to wedging of the valve disc during closure. These loads are superimposed on the loads due to the pressure forces which occur during pressure locking. The value for this load is based on static test data for the MOVs.

Piston Effect

The piston effect due to valve internal pressure exceeding outside pressure is calculated using the standard industry equation. This force assists movement of the valve stem in the open direction.

"Reverse Piston Effect"

The reverse piston effect is the term used in this calculation to refer to the pressure force acting downward against the valve disc. This force is equal to the differential pressure across the valve disc times the area of the valve disc times the sine of the seat angle times 2 (for two disc faces).

Total Force Required to Overcome Pressure Locking

As mentioned previously, the total stem force (tension) required to overcome pressure locking is the sum of the four components discussed above. All of the terms are positive with the exception of the piston effect component.

Next the Open Motor Gearing Capability (MGC_{Open}) is calculated using the Standard Limitorque Equation and modified by MOV White Paper 125, Installed Motor

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Capability Evaluation. In calculating MGC_{Open} , Motor Torque, Motor Temperature Factor, Degraded Voltage Factor, Pullout Efficiency, and Overall Actuator Ratio are utilized. For additional conservatism, a degraded Stem Factor at a Coefficient of Friction (COF) of 0.20 is used.

MGC_{Open} is compared to the Total Force Required to Overcome Pressure Locking, and a percent margin is calculated to show positive margin/capability. There is no acceptance criteria for this calculation.

III. ASSUMPTIONS

1. The valve disc is assumed to act as two ideal discs connected by a hub. The equations in reference 1 are assumed to conservatively model the actual load due to pressure forces.
2. Assumed pressure locking scenario for the 1(2)SI8802A&B Safety Injection Pump Discharge Hot Leg Isolation Valves. These valves are normally closed and must open during transfer from the cold leg to the hot leg recirculation phase of Emergency Core Cooling. During this transfer the applicable Safety Injection pump is shut down one at a time, the crosstie isolation valve (SI8821) is closed and then the applicable SI8802 valve is opened. In this scenario the pump pressure would potentially be trapped in the bonnet causing a pressure locking phenomenon to occur when the pump was shutdown. Two cases are assumed for this scenario: (1) Both Safety Injection pumps are initially running and (2) Only one Safety Injection pump is operating.
3. Based on Pre-Operational Test data for Braidwood and Byron units 1 & 2 (Reference 9 & 15) if both safety injection pumps are operating in the Emergency Core Cooling mode the discharge pressure is approximately 1400 psig (highest value from reference 9 & 15 testing corresponding to Byron unit 2). When the transfer from cold leg recirculation to hot leg recirculation takes place, one pump is shut down and the valve is subjected to the discharge pressure of one pump of approximately 890 psig (lowest value from Byron unit 2 testing). When the crosstie valve (SI8821) is closed this pressure is trapped in the system due to the pump discharge check valve. This yields a pressure locking average differential pressure of 955 psid as summarized in this calculation (pg 11). If only one pump is operating then the valve would be subject to a discharge pressure of approximately 920 psig (highest value from reference 9 & 15 testing corresponding to Byron unit 2). This yields a pressure locking average differential pressure of 920 psig. Therefore, this calculation will address the most limiting case of two pump operation. Downstream pressure in this scenario is assumed

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to be zero. It is assumed that the pumps were operating at their most efficient point (new pumps, no degradation) during this testing.

4. The 1(2)SI8802A&B Safety Injection Pump Discharge Hot Leg Isolation Valves are normally closed and subject to bonnet pressurization via Reactor Coolant System (RCS) pressure isolation valve leakage. Under a Loss of Coolant Accident (LOCA) these valves would be required to be opened in approximately 8.5 hours for the hot leg recirculation phase of Emergency Core Cooling. It is assumed that over this 8.5 hours prior to these valves having to open, that the RCS pressure which was potentially trapped in the valve bonnet would leak down to the pressure specified in assumption #3. This assumption was verified during the special test listed in reference (7). This test indicated that at a torque switch setting providing a similar maximum closing force as the SI8802 valves (less than 1400 lbs) the leakage rate averaged greater than 300 psig per minute between 2000 and 700 psig. This indicates that in less than 10 minutes the pressure would leak down to the point specified in assumption #3.
5. The coefficient of friction between the valve seat and disc is assumed to be the same under pressure locking conditions as it is under differential pressure conditions. The SI8802 valves were not differential pressure tested at Braidwood, however, they were at Byron Station. Similar valves at Braidwood Station were differential pressure tested in the SI system and open valve factors for these and the Byron valves have been tabulated in section VI (with the exception of Byron valve 1SI8802A for which the data was determined to be suspect). An open valve factor of 0.30 will be used for the calculations based on bounding the tested valve factors for these valves. Based on tested open valve factors tabulated in section VI, a valve factor of 0.30 is conservative with respect to the highest tested valve factor of 0.28 and the average valve factor of 0.23.
6. The valve unseating force is conservatively assumed to be the maximum unseating force for all of the valves listed in reference 2. This maximum opening value does not include equipment tolerances or extrapolation, rather this value is assumed to encompass these factors based on the grouping. The degraded voltage is conservatively assumed to be the lowest voltage from each of the valves listed in reference 2. Both of these assumptions ensure the calculation is conservative and bounds all operating conditions.
7. The calculation of motor gearing capability is performed at a degraded stem factor corresponding to a coefficient of friction of 0.20. This coefficient of friction bounds the degraded value for each of the subject valves listed in reference 3. This value is conservative.

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8. The disk hub radius is assumed to be equal to the effective radius of the hub due to the section not being circular in cross section. This effective radius is calculated in Attachment A.
9. For valve factor calculations, the open valve line pressure for all valves is assumed to be equal to the open valve line pressure obtained in SPP 93-034 (800 psig). These valves were differential pressure tested with similar system configurations. This was the only test at Braidwood in which this open line pressure data was taken.
10. For calculation of motor gearing capability, the temperature factor is taken from the Braidwood Rising Stem Data Sheets listed in reference 3. The Byron Data Sheets are the older revision and the application factor was reduced by the temperature factor. Byron valve 1SI8802A has a 34.1 OAR which produces a higher motor gearing capability. This is conservative.

IV. DESIGN INPUTS

1. Valve Disk Geometry information is based on Westinghouse Drawing #934D225 Rev 10. (Attachment A)
2. Modulus of Elasticity - 1995 ASME Section II, Table TM-1 (Attachment B)

V. REFERENCES

1. Sixth Edition of Roark's Formulas for Stress and Strain
2. Margin Review Calculation Sheets for :

Braidwood Station

1SI8802A, dated 06/06/97
 1SI8802B, dated 06/06/97
 2SI8802A, dated 05/25/96
 2SI8802B, dated 05/25/96

Byron Station

1SI8802A, dated 03/08/96
 1SI8802B, dated 03/08/96
 2SI8802A, dated 03/08/96
 2SI8802B, dated 03/08/96

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2. Rising Stem MOV Data Sheets for:

Braidwood Station

1SI8802A, dated 03/14/97
 1SI8802B, dated 03/14/97
 2SI8802A, dated 03/14/97
 2SI8802B, dated 03/14/97

Byron Station

1SI8802A, dated 08/05/94
 1SI8802B, dated 08/08/94
 2SI8802A, dated 08/08/94
 2SI8802B, dated 08/08/94

4. MOV White Paper WP-134 Rev. 0, EPRIs MOV Testing Program Measured Valve Factors.
5. Mechanical Engineering Design Forth Edition, Shigley and Mitchell
6. MOV White Paper 000, MOV Program Technical Guidance, Revision 3
7. Special test of Westinghouse 4 inch valve, test procedure dated 09/12/95, results summarized in DOC ID #DG96-000078.
8. Marks' Standard Handbook for Mechanical Engineers Eighth Edition
9. Preoperational Tests BwPT-SI-12 Rev. 0 and BwPT-SI-52 Rev. 0, Section 9.8.
10. Byron Station NDIT No. BYR-96-002
11. MOV White Paper 125 Revision 2, Installed Motor Capability Evaluation.
12. Special Process Procedures (SPPs) 91-061, 92-021, 92-074, 93-034
13. Differential Pressure Test Reviews and Upgrades:
 - PI-15, Dated 11/24/93 (1SI8821B)
 - PI-15, Dated 12/28/93 (2SI8821A)
 - PI-15, Dated 02/14/94 (2SI8821B)
14. NES letter DOC ID # DG96-000079 regarding calculation of open valve factor from DP Test Data
15. Byron Station NDIT No. BYR-96-022

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VI. CALCULATIONS

Calculation of valve factors for similar differential pressure tested valves at Braidwood and Byron Stations.

Open valve factors are calculated below based on test data using the current methodology for calculating this value. Differential pressure and VOTES test data summarized in the below table is used as input into the open valve factor equation (Reference 14) to generate the open valve factors.

$$\text{Valve Factor (open)} = \frac{(\text{O10 thrust} - \text{avg open run load} + (0.7854 * (\text{stem dia})^2 * \text{close line pressure} - \text{open line pressure}))}{0.7854 * (\text{mean seat dia})^2 * \text{differential pressure}}$$

Valve Factor Data Summary Table

STATION	VALVE	VOTES	TEST	O10	DIFF	LINE	OPEN	OPEN	OPEN	REFERENCE
		TEST #	DATE	THRUS	PRESS	PRESS	LINE	OPEN	OPEN	
							PRESS	LOAD	FACTOR	
Byron	1SI8802B	7	10/4/91	4493	1545	1560	800	1089	0.22	Ref 10, Assum 9
Byron	2SI8802A	4	3/7/95	2315	1510	1510	800	192	0.16	Ref 10, Assum 9
Byron	2SI8802B	2	3/7/95	5444	1520	1520	800	1014	0.28	Ref 10, Assum 9
Braidwood	1SI8821A	5	3/29/94	4323	1537	1563	800	868	0.23	Ref 12, Assum 9
Braidwood	1SI8821B	1	9/21/92	3892	1457	1480	800	-217	0.27	Ref 12, 13, Assum 9
Braidwood	2SI8821A	10	10/14/91	1752	1479	1517	800	-158	0.15	Ref 12, 13, Assum 9
Braidwood	2SI8821B	12	3/22/93	4532	1477	1520	800	651	0.26	Ref 12, 13, Assum 9

MathCad 5.0+ calculations of the following for the SI8802 valve with the given assumptions:

- 1) The pressure locking unseating force,
- 2) The opening motor gearing capability,
- 3) The available margin between the pressure locking unseating force and the opening motor gearing capability.

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VI. CALCULATIONS

INPUTS:

Bonnet Pressure	$P_{\text{bonnet}} = 1400 \text{ psi}$	Assumption 3
Upstream Pressure	$P_{\text{up}} = 890 \text{ psi}$	Assumption 3
Downstream Pressure	$P_{\text{down}} = 0 \text{ psi}$	Assumption 3
Disk Thickness	$t = 1.02 \text{ in}$	Attachment A
Seat Radius	$a = 2.001 \text{ in}$	Attachment A
Effective Hub Radius	$b = 1.056 \text{ in}$	Attachment A
Hub Length	$L = 0.60 \text{ in}$	Attachment A
Seat Angle	$\theta = 7 \text{ deg}$	Reference 3
Poisson's Ratio (disk)	$\nu = .3$	Typical of Stainless Steel
Mod. of Elast. (disk)	$E = 27.6 \cdot 10^6 \text{ psi}$	Attachment B, 200 F
Static Pullout Force	$F_{\text{po}} = 6399 \text{ lbf}$	Reference 2, Assumption 6
Open Valve Factor	$VF = .30$	Assumption 5
Stem Diameter	$D_{\text{stem}} = 1.25 \text{ in}$	Reference 3

PRESSURE FORCE CALCULATIONS

Coefficient of friction between disk and seat: (Reference 14)

$$\mu = VF \cdot \frac{\cos(\theta)}{1 - VF \cdot \sin(\theta)} \quad \mu = 0.309$$

Average DP across disks:

$$DP_{\text{avg}} = P_{\text{bonnet}} - \frac{P_{\text{up}} + P_{\text{down}}}{2} \quad DP_{\text{avg}} = 955 \text{ psi}$$

Disk Stiffness Constants (Reference 1, Table 24, Reference 5)

$$D = \frac{E \cdot (t)^3}{12 \cdot (1 - \nu^2)} \quad D = 2.682 \cdot 10^6 \text{ lbf in}^{-3}$$

$$G = \frac{E}{2 \cdot (1 + \nu)} \quad G = 1.062 \cdot 10^7 \text{ psi}$$

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VI. CALCULATIONS

Geometry Factors: (Reference 1, Table 24)

$$C_2 = \frac{1}{4} \left[1 - \left(\frac{b}{a} \right)^2 \cdot \left(1 + 2 \cdot \ln \left(\frac{a}{b} \right) \right) \right] \quad C_2 = 0.09137$$

$$C_3 = \frac{b}{4a} \left[\left[\left(\frac{b}{a} \right)^2 + 1 \right] \cdot \ln \left(\frac{a}{b} \right) + \left(\frac{b}{a} \right)^2 - 1 \right] \quad C_3 = 0.01262$$

$$C_8 = \frac{1}{2} \left[1 + v + (1 - v) \cdot \left(\frac{b}{a} \right)^2 \right] \quad C_8 = 0.74748$$

$$C_9 = \frac{b}{a} \left[\frac{1 + v}{2} \cdot \ln \left(\frac{a}{b} \right) + \frac{1 - v}{4} \left[1 - \left(\frac{b}{a} \right)^2 \right] \right] \quad C_9 = 0.28588$$

$$L_3 = \frac{a}{4a} \left[\left[\left(\frac{a}{a} \right)^2 + 1 \right] \cdot \ln \left(\frac{a}{a} \right) + \left(\frac{a}{a} \right)^2 - 1 \right] \quad L_3 = 0$$

$$L_9 = \frac{a}{a} \left[\frac{1 + v}{2} \cdot \ln \left(\frac{a}{a} \right) + \frac{1 - v}{4} \left[1 - \left(\frac{a}{a} \right)^2 \right] \right] \quad L_9 = 0$$

$$L_{11} = \frac{1}{64} \left[1 + 4 \cdot \left(\frac{b}{a} \right)^2 - 5 \cdot \left(\frac{b}{a} \right)^4 - 4 \cdot \left(\frac{b}{a} \right)^2 \cdot \left[2 + \left(\frac{b}{a} \right)^2 \right] \cdot \ln \left(\frac{a}{b} \right) \right] \quad L_{11} = 0.00162$$

$$L_{17} = \frac{1}{4} \left[1 - \frac{1 - v}{4} \left[1 - \left(\frac{b}{a} \right)^4 \right] - \left(\frac{b}{a} \right)^2 \cdot \left[1 + (1 + v) \cdot \ln \left(\frac{a}{b} \right) \right] \right] \quad L_{17} = 0.08216$$

Moment (Reference 1, Table 24, Case 2L)

$$M_{rb} = \frac{-DP_{avg} \cdot a^2}{C_8} \left[\frac{C_9}{2 \cdot a \cdot b} (a^2 - b^2) - L_{17} \right] \quad M_{rb} = -579.387 \cdot \text{lb} \cdot \text{ft}$$

$$Q_b = \frac{DP_{avg}}{2 \cdot b} (a^2 - b^2) \quad Q_b = 1306.281 \cdot \frac{\text{lb} \cdot \text{ft}}{\text{in}}$$

Deflection due to pressure and bending: (Reference 1, Table 24, Case 2L)

$$y_{bq} = M_{rb} \frac{a^2}{D} C_2 + Q_b \frac{a^3}{D} C_3 - \frac{DP_{avg} \cdot a^4}{D} L_{11} \quad y_{bq} = -3.9033 \cdot 10^{-5} \cdot \text{in}$$

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VI. CALCULATIONS

Deflection due to pressure and shear stress: (Reference 1, Table 25, Case 2L)

$$K_{sa} = -0.3 \left[2 \cdot \ln \left(\frac{a}{b} \right) - 1 + \left(\frac{b}{a} \right)^2 \right]$$

$$K_{sa} = -0.16705$$

$$y_{sq} = \frac{K_{sa} \cdot DP_{avg} \cdot a^2}{t \cdot G}$$

$$y_{sq} = -5.8993 \cdot 10^{-5} \text{ in}$$

Deflection due to hub stretch (from center of hub to disk): (Reference 5)

$$P_{force} = 3.1416 \cdot (a^2 - b^2) \cdot DP_{avg}$$

$$P_{force} = 8667.254 \text{ lbf}$$

$$y_{stretch} = \frac{P_{force} \cdot L}{3.1416 \cdot b^2 \cdot (2 \cdot E)}$$

$$y_{stretch} = 2.6891 \cdot 10^{-5} \text{ in}$$

Total Deflection due to pressure forces:

$$y_q = y_{bq} + y_{sq} - y_{stretch}$$

$$y_q = -0.0001 \text{ in}$$

Deflection due to seat contact force and shear stress (per lbf/in.): (Reference 1, Table 25, Case 1L)

$$y_{sw} = - \left[\frac{1.2 \cdot \left(\frac{a}{a} \right) \cdot \ln \left(\frac{a}{b} \right) \cdot a}{t \cdot G} \right]$$

$$y_{sw} = -1.4174 \cdot 10^{-7} \cdot \frac{\text{in}}{\frac{\text{lbf}}{\text{in}}}$$

(per lbf/in)

Deflection due to seat contact force and bending (per lbf/in.): (Reference 1, Table 24, Case 1L)

$$y_{bw} = - \left(\frac{a^3}{D} \right) \cdot \left[\left(\frac{C_2}{C_8} \right) \cdot \left[\left(\frac{a \cdot C_9}{b} \right) - L_9 \right] - \left[\left(\frac{a}{b} \right) \cdot C_3 \right] + L_3 \right]$$

$$y_{bw} = -1.2635 \cdot 10^{-7} \cdot \frac{\text{in}}{\frac{\text{lbf}}{\text{in}}}$$

(per lbf/in)

Deflection due to hub compression (per lbf/in), (from center of hub to disk): (Reference 5)

$$y_{compr} = \frac{2 \cdot a \cdot \pi \cdot L}{3.1416 \cdot b^2 \cdot (2 \cdot E)}$$

$$y_{compr} = 3.9009 \cdot 10^{-8} \cdot \frac{\text{in}}{\frac{\text{lbf}}{\text{in}}}$$

(per lbf/in)

Total deflection due to seat contact force (per lbf/in.):

$$y_w = y_{bw} + y_{sw} - y_{compr}$$

$$y_w = -3.071 \cdot 10^{-7} \cdot \frac{\text{in}}{\frac{\text{lbf}}{\text{in}}}$$

(per lbf/in)

Seat Contact Force for which deflection is equal to previously calculated deflection from pressure forces:

$$F_s = 2 \cdot \pi \cdot a \cdot \frac{y_q}{y_w}$$

$$F_s = 5114 \text{ lbf}$$

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VI. CALCULATIONS
UNSEATING FORCES F_{packing} is included in measured static pullout Force

$$F_{\text{piston}} = \frac{\pi}{4} \cdot D_{\text{stem}}^2 \cdot P_{\text{bonnet}}$$

$$F_{\text{piston}} = 1718.1 \cdot \text{lbf}$$

$$F_{\text{vert}} = \left[\pi \cdot (a^2) \right] \cdot \sin(\theta) \cdot (2 \cdot P_{\text{bonnet}} - P_{\text{up}} - P_{\text{down}})$$

$$F_{\text{vert}} = 2928 \cdot \text{lbf}$$

$$F_{\text{preslock}} = 2 \cdot F_s \cdot (\mu \cdot \cos(\theta) - \sin(\theta))$$

$$F_{\text{preslock}} = 1891.1 \cdot \text{lbf}$$

$$F_{\text{po}} = 6399 \cdot \text{lbf}$$

$$F_{\text{total}} = F_{\text{piston}} + F_{\text{vert}} + F_{\text{preslock}} + F_{\text{po}}$$

$$F_{\text{total}} = 9500 \cdot \text{lbf}$$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	MT = 16.97 ft-lbf	Reference 3, 11
Temperature Factor:	Tf = 0.98	Reference 3, Assumption 10
Degraded Voltage:	DV = 409 volt	Reference 2, 3, Assumption 6
Under Voltage Factor:	n = 2.2769	Reference 11
Overall Gear Ratio	OAR = 28.2	Reference 3, Assumption 10
Pullout Efficiency	EFF = 0.45	Reference 3
Stem Factor @ $\mu = 0.20$	SF = 0.0140 ft $\frac{\text{lbf}}{\text{lbf}}$	Reference 3, Assumption 7

CALCULATIONS:

$$MGC_{\text{Open}} = \frac{\left(\frac{DV}{460 \text{ volt}} \right)^n \cdot MT \cdot OAR \cdot Tf \cdot EFF}{SF} \quad (\text{Reference 6, 11})$$

$$MGC_{\text{Open}} = 11536 \cdot \text{lbf}$$

$$F_{\text{total}} = 9500 \cdot \text{lbf}$$

$$MGC_{\text{Margin}} = \frac{MGC_{\text{Open}} - F_{\text{total}}}{F_{\text{total}}}$$

$$MGC_{\text{Margin}} = 21.4\%$$

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VII. SUMMARY AND CONCLUSIONS

The results of the calculation indicate that with all the indicated conservatism inherent in the inputs, the 1(2)SI8802A&B Safety Injection Pump Discharge Hot Leg Isolation Valves have positive margin under the assumed pressure locking scenario. Therefore, pressure locking is not considered a concern for the subject MOVs. This calculation is being used as an input into the operability assessment (Attachment C) for PIF #'s 456-201-95-022600 and 454-200-95-0003.

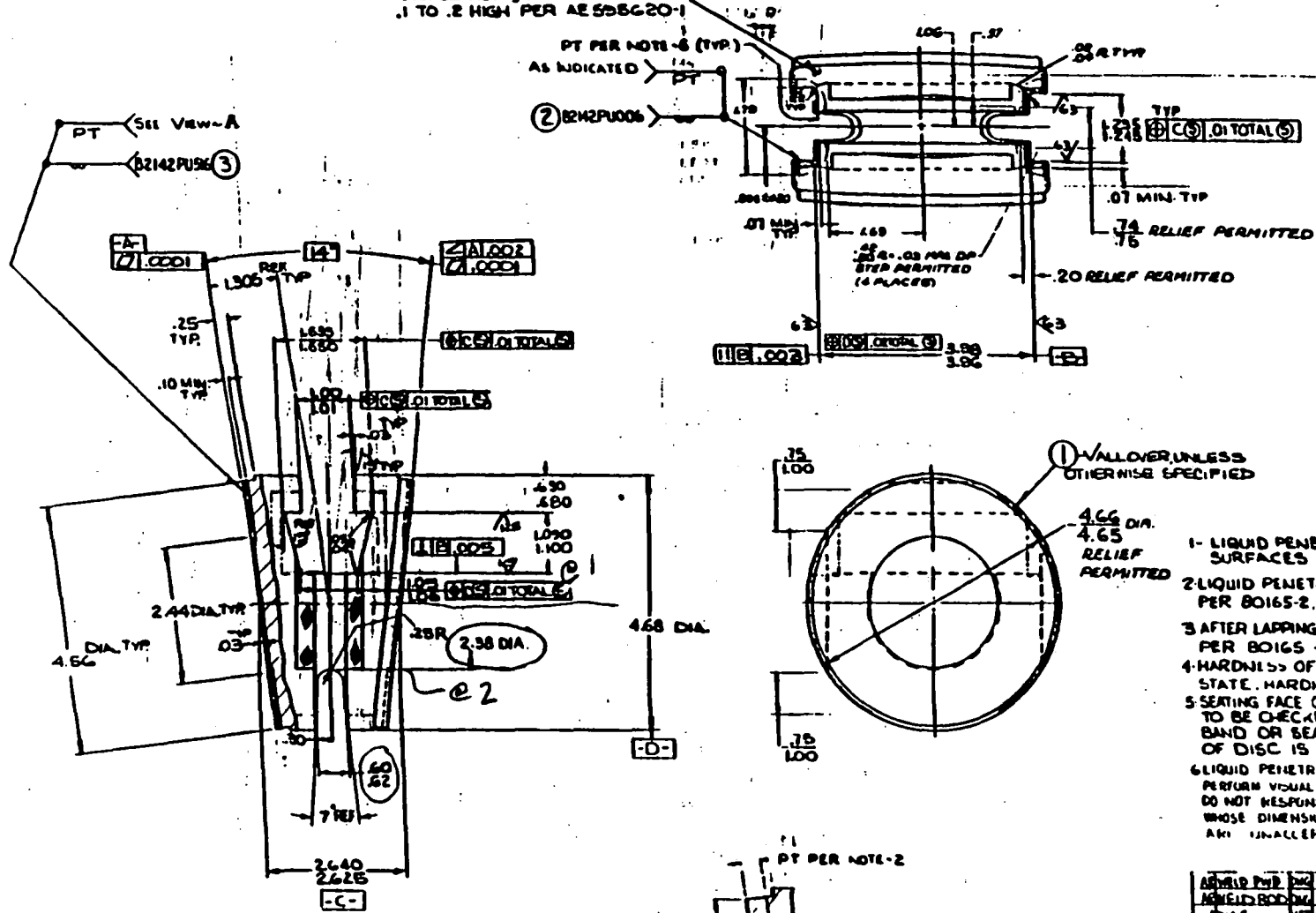
VIII. ATTACHMENTS

- (A) Westinghouse Drawing # 934D225 (Disc)
Hand Sketch of Disc Dimensions provided for clarity
Record of Conversation dated 01/03/96
Record of Conversation dated 02/12/96

- (B) Modulus of Elasticity - 1995 ASME Section II, Table TM-1

REVISION NO. 1

MARK SERIAL NO., PART NO. & HEAT NO.
.1 TO .2 HIGH PER AE 555G 20-1



REVISIONS	
1	Zone D4 Ref to Note A Deleted & B2142PU006 was B2142PU006; Note G
2	Revised Drawing
3	Zone D4 Ref to Note A Deleted & B2142PU006 was B2142PU006; Note G
4	Revised Drawing
5	Zone D4 Ref to Note A Deleted & B2142PU006 was B2142PU006; Note G
6	Revised Drawing

-NOTES-

- 1- LIQUID PENETRANT EXAMINE ALL FINAL MACHINED SURFACES PER BOIGS-2, CL. AA.
- 2- LIQUID PENETRANT EXAMINE INDICATED SURFACES PER BOIGS-2, CL. OOK.
- 3- AFTER LAPPING INDICATED SURFACES, LIQUID PENETRANT PER BOIGS -2, CL. OA.
- 4- HARDNESS OF STELLITE TO BE CHECKED IN ROUGH MACHINE STATE. HARDNESS TO BE 38 Rc MIN.
- 5- SEATING FACE OF DISC TO HAVE A LAPPED FINISH. THESE SURFACES TO BE CHECKED BY "BLUE-IN" METHOD. LAP UNTIL A CONTINUOUS BAND OR BEAT OF .08 MIN. WIDTH AROUND ENTIRE FACE OF DISC IS OBTAINED.
- 6- LIQUID PENETRANT EXAMINE INDICATED AREA PER BOIGS-2, CL. 1-4H. PERFORM VISUAL EXAMINATION FOR OPEN TYPE DISCONTINUITIES WHICH DO NOT RESPOND TO PENETRANT EXAMINATION. VISUAL DISCONTINUITIES WHOSE DIMENSIONS ARE GREATER THAN .254 INCH IN ANY DIRECTION ARE UNACCEPTABLE.

APPROVED PWD	STELLITE 136	304111	1
APPROVED RODING	STELLITE 6	304112	2
DISC	316 SST	304113	3

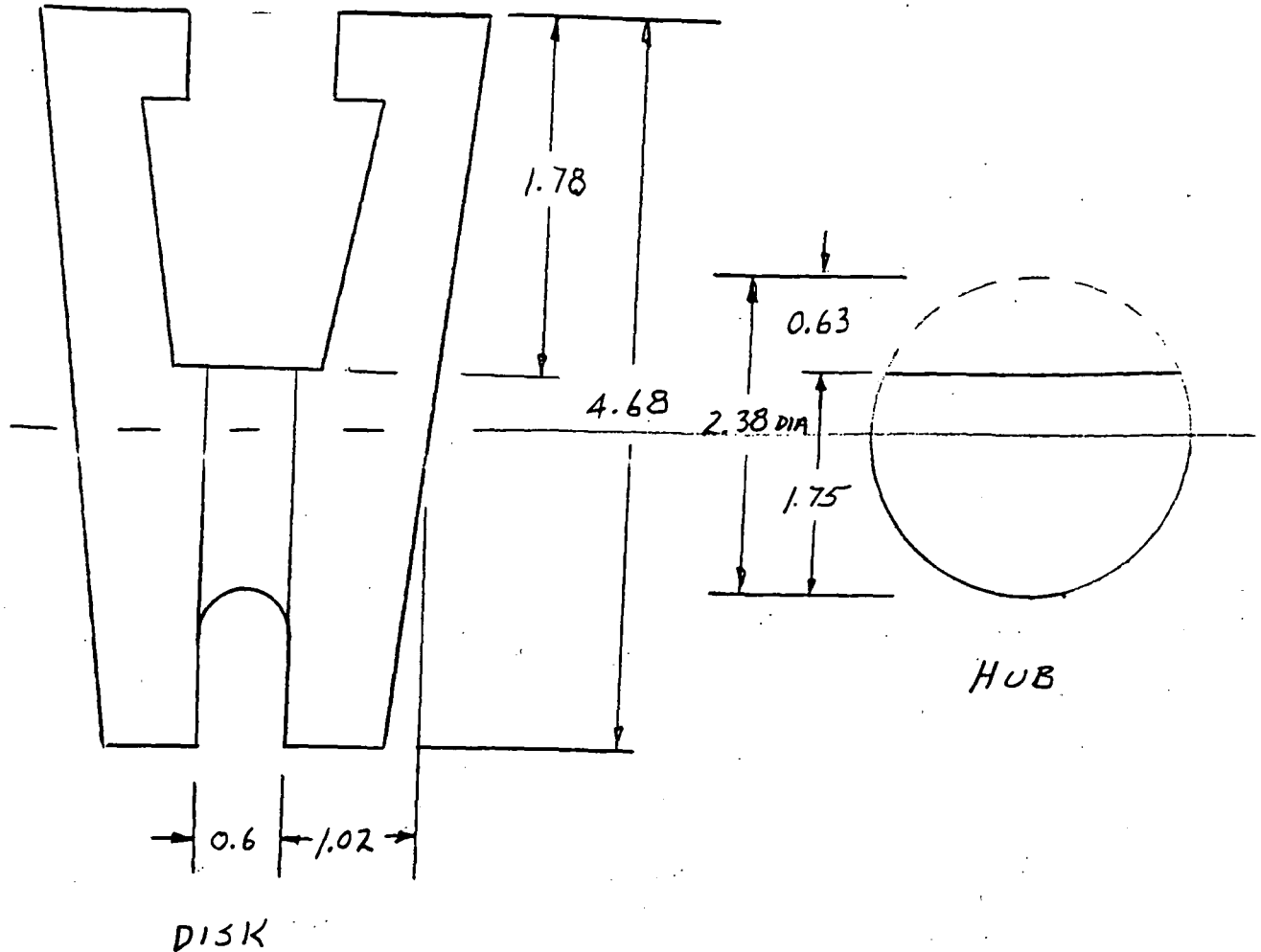
WESTINGHOUSE ELECTRIC CORPORATION	
DISC-314-1500, 4-900	
GATE VALVE DISC	
D 04808	934D225

AE0 550 CITYVIEW 5
TID 510 CITIES
MILLING CUTTER
R&L 55000

CALC. BRW 96.015 REV. 1

Disk Dimensions

ATTACHMENT 11 70.00
CALC. BRW 96 015 REV. 1



Effective Radius of Hub Section

Total Area = $\pi(2.38)^2/4 = 4.449 \text{ in}^2$

Area of Hub Section Missing (Reference 8 Segments of Circles h/D)

$h/D = 0.63\text{in}/2.38\text{in} = .264$ interpolation from table

Area/Circle = 0.21108

Area of Missing Section = $0.21108 * 4.449\text{in}^2 = 0.939\text{in}^2$

Area of Hub = $4.449 - 0.939 = 3.509\text{in}^2$

Effective Area Diameter

Area = $\pi*d^2/4$ $d = \sqrt{(3.509 * 4/\pi)} = 2.114\text{in}$

Effective Hub Radius (b) = $2.114/2 = 1.056 \text{ in}$

$L = 0.60\text{in}$

$t = 1.02$

Record of Conversation

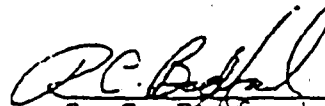
Per conversation with T. Matty of Westinghouse on 01/03/96 at 1345 (Phone 412-374-6401) the following seat ring dimensions were obtained for the listed valves:

1/2RY8000A&B 3 inch valves

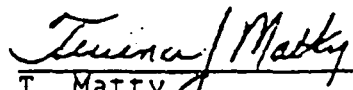
Seat ring inside diameter 2.6875 in *
Seat ring outside diameter 3.75 in
Mean seat ring diameter 3.21875 in

1/2SI8801A&B, 1/2SI8802A&B, 1/2SI8821A&B 4 inch valves

Seat ring inside diameter 3.5075 in *
Seat ring outside diameter 4.5 in
Mean seat ring diameter 4.0038 in


R. C. Bedford
MOV Programs
Braidwood Station

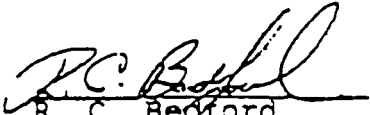
Concur

 2/27/91
T. Matty
Westinghouse

* Made up of Seat Bone plus .0625 for chamfers

Record of Conversation

Per conversation with T. Matty of Westinghouse on 02/12/96 at 0810 (Phone 412-374-6401) it was confirmed that valves 1(2)RY8000A&B, 1(2)SI8801A&B, 1(2)SI8802A&B and 1(2)SI8821A&B all contain discs manufactured from Westinghouse sub assembly drawing 934D225.


R. C. Bedford
MOV Programs
Braidwood Station

Concur



T. Matty
Westinghouse

Table TM-1

1995 SECTION II

TABLE TM-1
MODULI OF ELASTICITY E OF FERROUS MATERIALS FOR GIVEN TEMPERATURES

Materials	Modulus of Elasticity E = Value Given × 10 ⁶ psi, for Temp., °F, of											
	-325	-200	-100	70	200	300	400	500	600	700	800	900
Carbon steels with C ≤ 0.30%	31.4	30.8	30.2	29.5	28.8	28.3	27.7	27.3	26.7	25.5	24.2	22.4
Carbon steels with C > 0.30%	31.2	30.6	30.0	29.3	28.6	28.1	27.5	27.1	26.5	25.3	24.0	22.3
Material Group A ¹	31.1	30.5	29.9	29.2	28.5	28.0	27.4	27.0	26.4	25.3	23.9	22.2
Material Group B ²	29.6	29.1	28.5	27.8	27.1	26.7	26.1	25.7	25.2	24.6	23.0	...
Material Group C ³	31.6	31.0	30.4	29.7	29.0	28.5	27.9	27.5	26.9	26.3	25.5	24.8
Material Group D ⁴	32.6	32.0	31.4	30.6	29.8	29.4	28.8	28.3	27.7	27.1	26.3	25.6
Material Group E ⁵	32.9	32.3	31.7	30.9	30.1	29.7	29.0	28.6	28.0	27.3	26.1	24.7
Material Group F ⁶	31.2	30.7	30.1	29.2	28.5	27.9	27.3	26.7	26.1	25.6	24.7	23.2
Material Group G ⁷	30.3	29.7	29.1	28.3	27.6	27.0	26.5	25.8	25.3	24.8	24.1	23.5

NOTES:

(1) Material Group A consists of the following carbon-molybdenum steels:

- C- $\frac{1}{2}$ Mo Mn- $\frac{1}{4}$ Mo
- Mn- $\frac{1}{2}$ Mo Mn-V

(2) Material Group B consists of the following Ni steels:

- $\frac{3}{4}$ Ni- $\frac{1}{2}$ Mo-Cr-V 1Ni- $\frac{1}{2}$ Cr- $\frac{1}{2}$ Mo
- $\frac{1}{2}$ Ni- $\frac{1}{2}$ Mo-V $\frac{3}{4}$ Ni-1Mo- $\frac{3}{4}$ Cr
- $\frac{3}{4}$ Ni- $\frac{1}{2}$ Mo- $\frac{1}{3}$ Cr-V $\frac{1}{2}$ Ni- $\frac{1}{2}$ Cr- $\frac{1}{4}$ Mo-V
- $\frac{3}{4}$ Cr- $\frac{3}{4}$ Ni-Cu-Al 2Ni-1Cu
- $\frac{3}{4}$ Cr- $\frac{1}{2}$ Ni-Cu 2 $\frac{1}{2}$ Ni
- $\frac{3}{4}$ Ni- $\frac{1}{2}$ Cu-Mo 3 $\frac{1}{2}$ Ni

(3) Material Group C consists of the following $\frac{1}{2}$ -2Cr steels:

- $\frac{1}{2}$ Cr- $\frac{1}{2}$ Mo
- 1Cr- $\frac{1}{2}$ Mo
- 1 $\frac{1}{2}$ Cr- $\frac{1}{2}$ Mo-Si
- 1 $\frac{1}{4}$ Cr- $\frac{1}{2}$ Mo
- 2Cr- $\frac{1}{2}$ Mo

(4) Material Group D consists of the following 2 $\frac{1}{4}$ -3Cr steels:

- 2 $\frac{1}{4}$ Cr-1Mo
- 3Cr-1Mo

(5) Material Group E consists of the following 5-9Cr steels:

- 5Cr- $\frac{1}{2}$ Mo
- 5Cr- $\frac{1}{2}$ Mo-Si
- 5Cr- $\frac{1}{2}$ Mo-Ti
- 7Cr- $\frac{1}{2}$ Mo
- 9Cr-Mo

(6) Material Group F consists of the following chromium steels:

- 12Cr-Al
- 13Cr
- 15Cr
- 17Cr

(7) Material Group G consists of the following austenitic steels:

- 18Cr-8Ni 18Cr-10Ni-Cb
- 18Cr-8Ni-N 18Cr-18Ni-2Si
- 16Cr-12Ni 20Cr-6Ni-9Mn
- 18Cr-13Ni-3Mo 22Cr-13Ni-5Mn
- 16Cr-12Ni-2Mo-N 23Cr-12Ni
- 18Cr-3Ni-13Mn 25Cr-20Ni
- 18Cr-10Ni-Ti

(Final)