COMMONWEALTH EDISON COMPANY
CALCULATION TITLE PAGE

CALCULATION NO. BRW 96-015

PAGE NO.: 1

■ SAFETY RELATED ☐ REGULATORY RELATED ☐ NON-SAFETY RELATED

CALCULATION TITLE:

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

STATION/UNIT: Braidwood & Byron/1&2

SYSTEM ABBREVIATION: SI

EQUIPMENT NO.: (IF APPL.)

1SI8802A
1SI8802B
2SI8802A
2SI8802B

PROJECT NO.: (IF APPL.)

N/A

REV: 0 STATUS: QA SERIAL NO. OR CHRON NO. DATE: / /96

PREPARED BY: R. C. Bedford /R. C. Bedford DATE: 2/12/96

REVISION SUMMARY:

Initial issue.

REVIEWED BY: J D Tolar 2-12-96 /J. D. Tolar

REVIEW METHOD: Detailed review

COMMENTS (C OR NC): NC

APPROVED BY: Bruce J Acas 2/13/96 / Bruce J. Acas

COMMONWEALTH EDISON COMPANY

CALCULATION REVISION PAGE

CALCULATION NO. BRW 96-015 PAGE NO.: 2

REV: QA SERIAL NO. OR CHRON NO. DATE:________
STATUS:

PREPARED BY:____________________ DATE:________

REVISION SUMMARY:

REVIEWED BY:____________________ DATE:________

REVIEW METHOD: COMMENTS (C OR NC):_____

REV: QA SERIAL NO. OR CHRON NO. DATE:________
STATUS:

PREPARED BY:____________________ DATE:________

REVISION SUMMARY:

REVIEWED BY:____________________ DATE:________

REVIEW METHOD: COMMENTS (C OR NC):_____

Exhibit D
NEP-12-02
Revision 0

COMMONWEALTH EDISON COMPANY

CALCULATION TABLE OF CONTENTS

	PROJECT NO. N/A	
CALCULATION NO. BRW 96-015	REV. NO. 0	PAGE NO. 3
DESCRIPTION	PAGE NO.	SUB-PAGE NO.

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.

<u>Pressure Locking Component of Force Required to Open the Valve</u>

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.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

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:

(seat load) x [(seat mu) cos(seat angle) - sin(seat angle)] x 2 (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 Capability Evaluation. In calculating MGC_{Open} , Motor Torque, Motor Temperature Factor , Degraded Voltage, Pullout Efficiency, and an Application Factor of 1.0 are utilized. For additional conservatism, a degraded Stem Factor at a Coefficient of Friction (COF) of 0.20 is used.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

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 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

III. ASSUMPTION (con't)

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 0.485 will be used for the calculations as a conservative measure based on design open valve factors for these valves. Byron's Rising Stem MOV Data Sheets listed in reference 3 indicate an open valve factor of 0.485. Braidwood's Rising Stem MOV Data Sheets listed in reference 3 indicate an open valve factor of 0.598, however, this open valve factor was increased from the design value of 0.485 based on MOV White Paper WP-166, Low Differential Pressure Load Testing and Setup. Due to the low design closing differential pressure (33 psid), the closed valve factor was increased. This also over conservatively increased the open valve factor. Pressure locking is a high loading condition and, as such, the open design value of 0.598 is overly conservative. Based on tested valve factors tabulated in section VI indicating an average valve factor of 0.23 this open valve factor is very conservative.

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.

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

III. ASSUMPTION (con't)

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	Byron Station
1SI8802A, dated 06/27/94	1SI8802A, dated 09/14/94
1SI8802B, dated 01/06/96	1SI8802B, dated 09/14/94
2SI8802A, dated 06/27/94	2SI8802A, dated 03/24/95
2SI8802B, dated 06/27/94	2SI8802B, dated 03/16/95

3. Rising Stem MOV Data Sheets for :

Braidwood Station	Byron Station
1SI8802A, dated 08/10/95	1SI8802A, dated 08/05/94
1SI8802B, dated 08/10/95	1SI8802B, dated 08/08/94
2SI8802A, dated 08/10/95	2SI8802A, dated 08/08/94
2SI8802B, dated 08/10/95	2SI8802B, dated 08/08/94

4. MOV White Paper WP-134 Rev. 0, EPRIs MOV Testing Program Measured Valve Factors.

V. REFERENCES (cont)

5. Mechanical Engineering Design Forth Edition, Shigley and Mitchell

6. MOV White Paper 000, MOV Program Technical Guidance, Revision 2

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

VI. CALCULATIONS

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

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,

VI. CALCULATIONS

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.

Valve Factor (open) = (O10 thrust - avg open run load ÷ (0.7854 * (stem dia)^2 * close line pressure - open line pressure)) / 0.7854 * (mean seat dia)^2 * differential pressure

Valve Factor Data Summary Table

STATION	VALVE	VOTES TEST #	TEST DATE	O10 THRUST	DIFF PRESS	LINE PRESS	OPEN LINE PRESS	OPEN RUN LOAD	OPEN VALVE FACTOR	REFERENCE
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

VI. CALCULATIONS

INPUTS:

Bonnet Pressure	$P_{bonnet} := 1400 \cdot psi$	Assumption 3
Upstream Pressure	$P_{up} := 890 \cdot psi$	Assumption 3
Downstream Pressure	$P_{down} := 0 \cdot psi$	Assumption 3
Disk Thickness	$t := 1.02 \cdot in$	Attachment A
Seat Radius	$a := 2.001 \cdot in$	Attachment A
Effective Hub Radius	$b := 1.056 \cdot in$	Attachment A
Hub Length	$L := 0.60 \cdot in$	Attachment A
Seat Angle	$theta := 7 \cdot deg$	Reference 3
Poisson's Ratio (disk)	$v := .3$	Typical of Stainless Steel
Mod. of Elast. (disk)	$E := 27.6 \cdot 10^6 \cdot psi$	Attachment B, 200 F
Static Pullout Force	$F_{po} := 6180 \cdot lbf$	Reference 2, Assumption 6
Open Valve Factor	$VF := .485$	Reference 3, Assumption 5
Stem Diameter	$D_{stem} := 1.25 \cdot in$	Reference 3

PRESSURE FORCE CALCULATIONS

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

$$mu := VF \cdot \frac{\cos(theta)}{1 - VF \cdot \sin(theta)} \qquad mu = 0.512$$

Average DP across disks:

$$DPavg := P_{bonnet} - \frac{P_{up} + P_{down}}{2} \qquad DPavg = 955 \cdot psi$$

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

$$D := \frac{E \cdot (t)^3}{12 \cdot \left(1 - v^2\right)} \qquad D = 2.682 \cdot 10^6 \cdot lbf \cdot in$$

$$G := \frac{E}{2 \cdot (1 + v)} \qquad G = 1.062 \cdot 10^7 \cdot psi$$

VI. CALCULATIONS

Geometry Factors: (Reference 1, Table 24)

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

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

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

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

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

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

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

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

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

$$M_{rb} := \frac{-DPavg\cdot a^2}{C_8}\cdot\left[\frac{C_9}{2\cdot a\cdot b}\cdot\left(a^2 - b^2\right) - L_{17}\right] \qquad M_{rb} = -579.387\cdot lbf$$

$$Q_b := \frac{DPavg}{2\cdot b}\cdot\left(a^2 - b^2\right) \qquad Q_b = 1306.281\cdot\frac{lbf}{in}$$

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

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

VI. CALCULATIONS

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

$$K_{sa} := -0.3\cdot\left[2\cdot\ln\left(\frac{a}{b}\right) - 1 + \left(\frac{b}{a}\right)^2\right] \qquad K_{sa} = -0.16705$$

$$y_{sq} := \frac{K_{sa}\cdot DPavg\cdot a^2}{t\cdot G} \qquad y_{sq} = -5.8993\cdot 10^{-5}\cdot in$$

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

$$P_{force} := 3.1416\cdot\left(a^2 - b^2\right)\cdot DPavg \qquad P_{force} = 8667.254\cdot lbf$$

$$y_{stretch} := \frac{P_{force}}{3.1416\cdot b^2}\cdot\frac{L}{(2\cdot E)} \qquad y_{stretch} = 2.6891\cdot 10^{-5}\cdot in$$

Total Deflection due to pressure forces:

$$y_q := y_{bq} + y_{sq} - y_{stretch} \qquad y_q = -0.0001\cdot in$$

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

$$y_{sw} := -\left[\frac{1.2\cdot\left(\frac{a}{a}\right)\cdot\ln\left(\frac{a}{b}\right)\cdot a}{t\cdot G}\right] \qquad y_{sw} = -1.4174\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{bw} := -\left(\frac{a^3}{D}\right)\cdot\left[\left(\frac{C_2}{C_8}\right)\cdot\left[\left(\frac{a\cdot C_9}{b}\right) - L_9\right] - \left[\left(\frac{a}{b}\right)\cdot C_3\right] + L_3\right] \qquad y_{bw} = -1.2635\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{compr} := \frac{2\cdot a\cdot\pi}{3.1416\cdot b^2}\cdot\frac{L}{(2\cdot E)} \qquad y_{compr} = 3.9009\cdot 10^{-8}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_w := y_{bw} + y_{sw} - y_{compr} \qquad y_w = -3.071\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$F_s := 2\cdot\pi\cdot a\cdot\frac{y_q}{y_w} \qquad F_s = 5114\cdot lbf$$

VI. CALCULATIONS

UNSEATING FORCES

$F_{packing}$ is included in measured static pullout Force

$F_{piston} := \frac{\pi}{4} \cdot D_{stem}^{2} \cdot P_{bonnet}$ $\quad F_{piston} = 1718.1 \cdot lbf$

$F_{vert} := \left[\pi \cdot \left(a^2\right)\right] \cdot \sin(theta) \cdot \left(2 \cdot P_{bonnet} - P_{up} - P_{down}\right)$ $\quad F_{vert} = 2928 \cdot lbf$

$F_{preslock} := 2 \cdot F_s \cdot (mu \cdot \cos(theta) - \sin(theta))$ $\quad F_{preslock} = 3947.4 \cdot lbf$

$F_{po} = 6180 \cdot lbf$

$F_{total} := -F_{piston} + F_{vert} + F_{preslock} + F_{po}$ $\quad F_{total} = 11337 \cdot lbf$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	$MT := 16.97 \cdot ft \cdot lbf$	Reference 3, 11
Temperature Factor:	$Tf := 0.98$	Reference 3, Assumption 10
Degraded Voltage:	$DV := 409 \cdot volt$	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
Application Factor	$AF := 1.0$	Reference 11 sets AF to 1.0
Stem Factor @ $\mu = 0.20$	$SF := 0.0140 \cdot ft \cdot \frac{lbf}{lbf}$	Reference 3, Assumption 7

CALCULATIONS:

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

$MGC_{Open} = 11536 \cdot lbf$ $\quad F_{total} = 11337 \cdot lbf$

$MGC_{Margin} := \frac{MGC_{Open} - F_{total}}{F_{total}}$ $\quad MGC_{Margin} = 1.7 \cdot \%$

CALCULATION NO. *BRW 96-015*	PROJECT NO. *N/A*	PAGE NO. *15*

VI. 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.

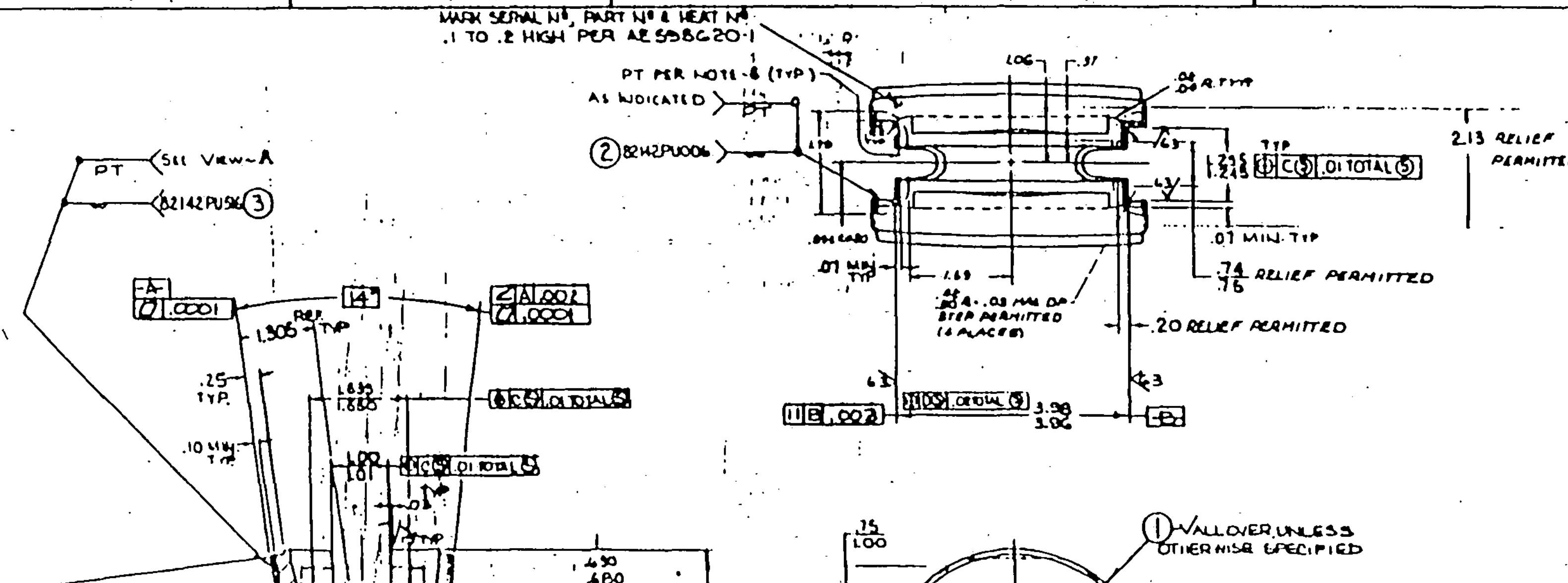
VI. LIMITATIONS

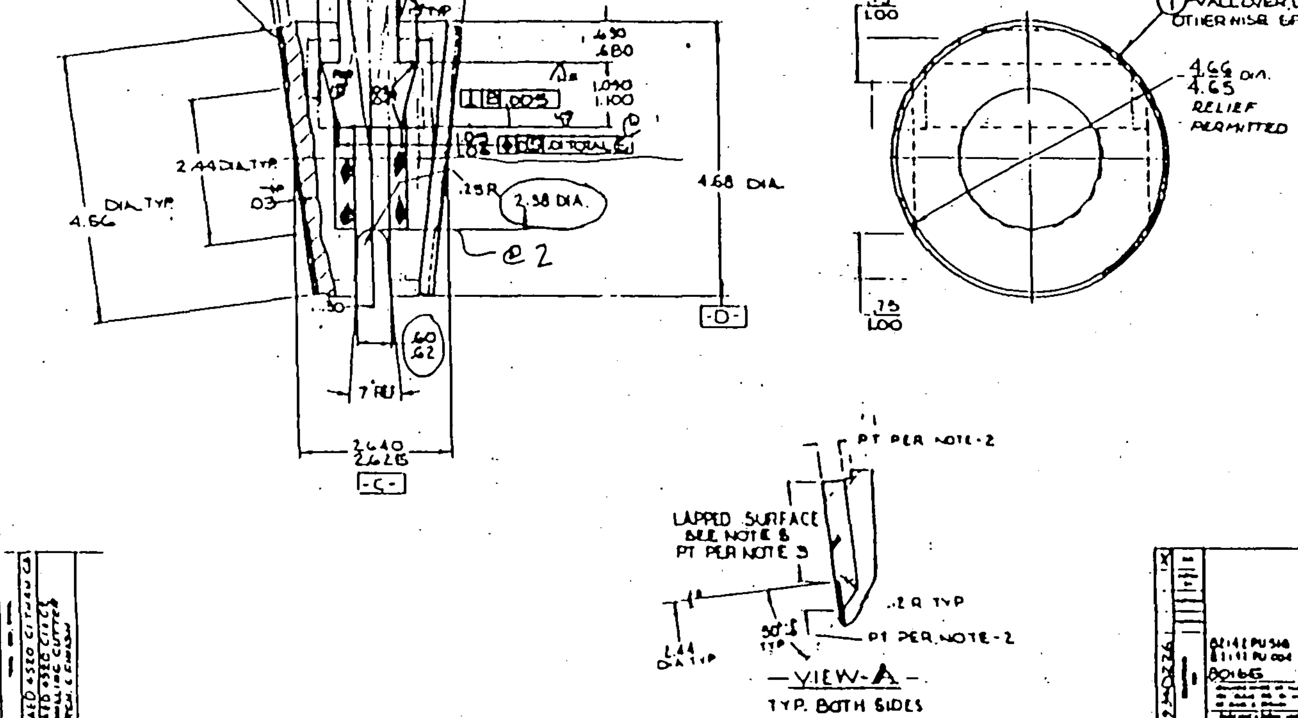
None.

IX. 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





NOTES

1. LIQUID PENETRANT EXAMINE ALL FINAL MACHINED SURFACES PER 80165-2, CL. AA.
2. LIQUID PENETRANT EXAMINE INDICATED SURFACES PER 80165-2, CL.OOK.
3. AFTER LAPPING INDICATED SURFACES LIQUID PENETRANT PER 80165-2, 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 80165-2, CL 1-4A. PERFORM VISUAL EXAMINATION FOR OPEN TYPE DISCONTINUITIES WHICH DO NOT RESPOND TO PENETRANT EXAMINATION. VISUAL DISCONTINUITIES WHOSE DIMENSIONS ARE GREATER THAN 3/64 INCH IN ANY DIRECTION ARE UNACCEPTABLE.

REVISIONS

9	ZONE D4 REF TO NOTE A DELETED & 82142PU006 WAS 82142PU004; NOTE 6 REVISED DAH/JW/JSZ
10	ITEM 11.1 1306C55H01 WAS 934D234H01, REVISED NOTE 1, ZONE D-4 ADDED .12 R TYP, ZONE D-8 1.238/1.245 WAS 1.235/1.240, ZONE A-5 2.640/2.628 WAS 2.638/2.628, VIEW-A ADDED .12 R TYP, 50° TYP AND 2.44 DIA TYP. ZONE C-5 1.635/1.660 WAS 1.64/1.65 WJD/-

LIST OF MATERIALS

AE	WELD PMP		STELLITE 156				3
AE	WELD ROD		STELLITE 6				2
	DISC		316 SST	1306C55H01		1	1

82142PU56
82142PU006
80165

Westinghouse Electric Corporation

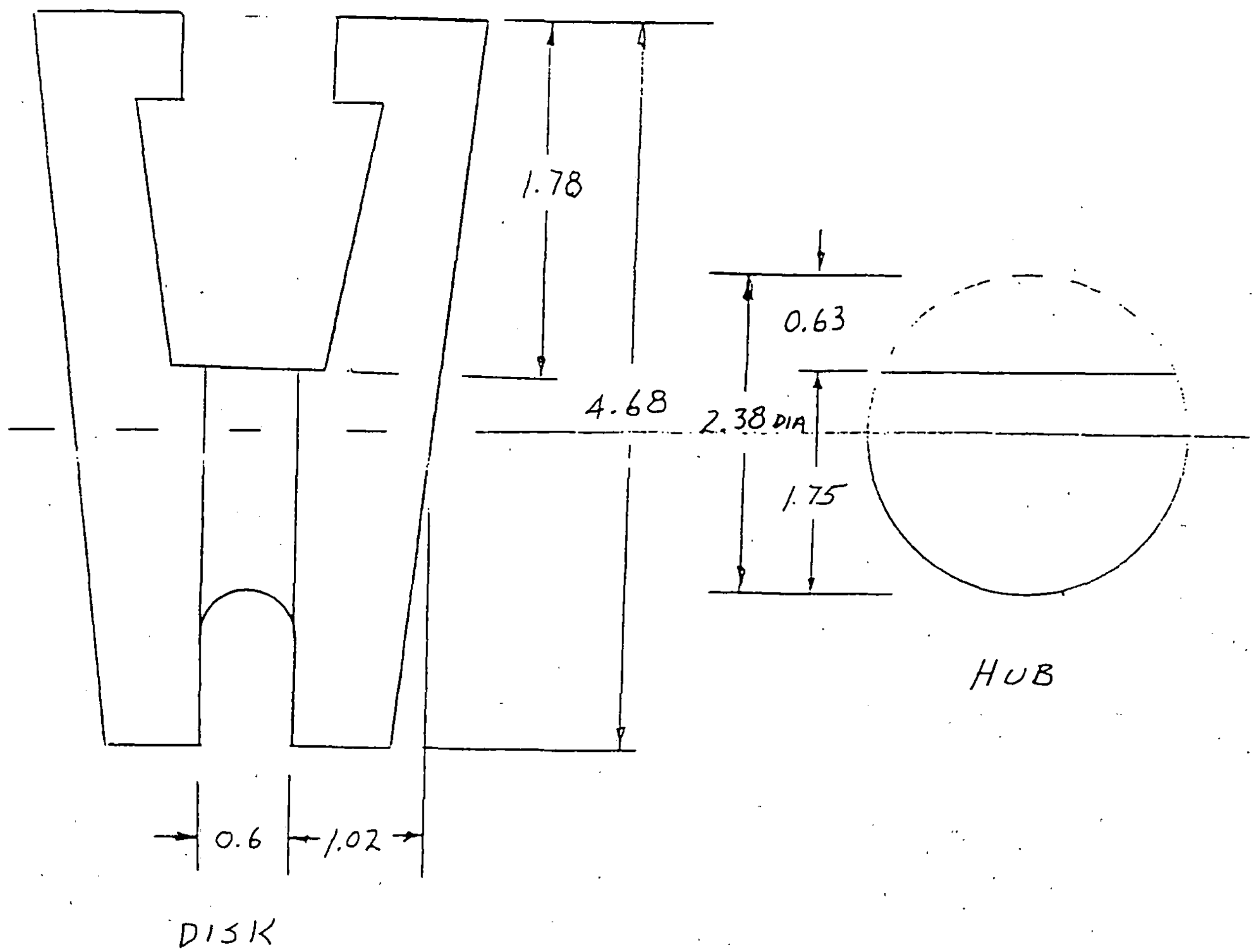
DISC-3/4-1500,4-900
GATE VALVE
DISC

D | 04808 | 934D225

N.T.S

ATTACHMENT A
CALC BRW 96-015 REV.

Disk Dimensions



Effective Radius of Hub Section

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

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

h/D = 0.63in/2.38in = .264 interpolation from table pg 1-7 (REF 8)

Area/Circle = 0.21108

Area of Missing Section = 0.21108 * 4.449in^2 = 0.939in^2

Area of Hub = 4.449 - 0.939 = 3.509in^2

Effective Area Diameter

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

Effective Hub Radius (b) = 2.114/2 = 1.056 in

L = 0.60in

t = 1.02

ATTACHMENT A
CALC BRW 96-015 REV.

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

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R. C. Bedford
MOV Programs
Braidwood Station

ATTACHMENT A 1-4
CALC BRW 96-015 REV. C

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.

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Braidwood Station

Calc #: 96-015, Rev. 0
ATTACHMENT B

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	$\boxed{27.6}$	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–½Mo	Mn–¼Mo
Mn–½Mo	Mn–V

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

¾Ni–½Mo–Cr–V	1Ni–½Cr–½Mo
½Ni–½Mo–V	¾Ni–1Mo–¾Cr
¾Ni–½Mo–⅓Cr–V	½Ni–½Cr–¼Mo–V
¾Cr–¾Ni–Cu–Al	2Ni–1Cu
¾Cr–½Ni–Cu	2½Ni
¾Ni–½Cu–Mo	3½Ni

(3) Material Group C consists of the following ½–2Cr steels:

½Cr–½Mo
1Cr–½Mo
1¼Cr–½Mo–Si
1¼Cr–½Mo
2Cr–½Mo

(4) Material Group D consists of the following 2¼–3Cr steels:

2¼Cr–1Mo
3Cr–1Mo

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

5Cr–½Mo
5Cr–½Mo–Si
5Cr–½Mo–Ti
7Cr–½Mo
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)

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

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Braidwood Station

Concur Teresa / Matty 2/27/96
T. Matty
Westinghouse

* Made up of Seat Bore 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.

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R. C. Bedford
MOV Programs
Braidwood Station

Concur Terence Matty 2/29/96
T. Matty
Westinghouse

Appendix C

Byron Station Capability Calculations in support of GL 95-07 Evaluation

Appendix D

Braidwood Station Capability Calculations in support of GL 95-07 Evaluation

COMMONWEALTH EDISON COMPANY
CALCULATION TITLE PAGE

CALCULATION NO. BRW 96-015 PAGE NO.: 1.

■ SAFETY RELATED ☐ REGULATORY RELATED ☐ NON-SAFETY RELATED

CALCULATION TITLE:

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

STATION/UNIT: Braidwood & Byron/1&2 SYSTEM ABBREVIATION: SI

EQUIPMENT NO.: (IF APPL.)

1SI8802A
1SI8802B
2SI8802A
2SI8802B

PROJECT NO.: (IF APPL.)

N/A

REV: *0* STATUS: QA SERIAL NO. OR CHRON NO. DATE: / /96

PREPARED BY: R. C. Bedford /*R. C. Bedford* DATE: 2/12/96

REVISION SUMMARY:

Initial issue.

REVIEWED BY: J D Tolar 2-12-96 /*J. D. Tolar*

REVIEW METHOD: *Detailed review* COMMENTS (C OR NC): NC

APPROVED BY: Bruce J Acas 2/13/96 / Bruce J. Acas

COMMONWEALTH EDISON COMPANY

CALCULATION REVISION PAGE

CALCULATION NO. BRW 96-015	PAGE NO.: 2
REV: STATUS: QA SERIAL NO. OR CHRON NO.	DATE:__________
PREPARED BY:____________________ REVISION SUMMARY:	DATE:__________
REVIEWED BY:____________________ REVIEW METHOD:	DATE:__________ COMMENTS (C OR NC):_____
REV: STATUS: QA SERIAL NO. OR CHRON NO.	DATE:__________
PREPARED BY:____________________ REVISION SUMMARY:	DATE:__________
REVIEWED BY:____________________ REVIEW METHOD:	DATE:__________ COMMENTS (C OR NC):_____

COMMONWEALTH EDISON COMPANY

CALCULATION TABLE OF CONTENTS

	PROJECT NO. N/A	
CALCULATION NO. BRW 96-015	REV. NO. 0	PAGE NO. 3
DESCRIPTION	PAGE NO.	SUB-PAGE NO.
TITLE PAGE	1	
REVISION SUMMARY	2	
TABLE OF CONTENTS	3	
I. PURPOSE/OBJECTIVE	4	
II. METHODOLOGY AND ACCEPTANCE CRITERIA	4 - 6	
III. ASSUMPTIONS	6 - 8	
IV. DESIGN INPUT	8	
V. REFERENCES	8, 9	
VI. CALCULATIONS	9 - 14	
VII. SUMMARY AND CONCLUSIONS	15	
VIII. LIMITATIONS	15	
IX. ATTACHMENTS	15	
A) Disc Dimensions	A1-A4	
B) Modulus of Elasticity - 1995 ASME Section II, Table TM-1	B1	

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.

<u>Pressure Locking Component of Force Required to Open the Valve</u>

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.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

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:

(seat load) x [(seat mu) cos(seat angle) - sin(seat angle)] x 2 (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 Capability Evaluation. In calculating MGC_{Open} , Motor Torque, Motor Temperature Factor , Degraded Voltage, Pullout Efficiency, and an Application Factor of 1.0 are utilized. For additional conservatism, a degraded Stem Factor at a Coefficient of Friction (COF) of 0.20 is used.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

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 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

III. ASSUMPTION (con't)

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 0.485 will be used for the calculations as a conservative measure based on design open valve factors for these valves. Byron's Rising Stem MOV Data Sheets listed in reference 3 indicate an open valve factor of 0.485. Braidwood's Rising Stem MOV Data Sheets listed in reference 3 indicate an open valve factor of 0.598, however, this open valve factor was increased from the design value of 0.485 based on MOV White Paper WP-166, Low Differential Pressure Load Testing and Setup. Due to the low design closing differential pressure (33 psid), the closed valve factor was increased. This also over conservatively increased the open valve factor. Pressure locking is a high loading condition and, as such, the open design value of 0.598 is overly conservative. Based on tested valve factors tabulated in section VI indicating an average valve factor of 0.23 this open valve factor is very conservative.

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.

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

III. ASSUMPTION (con't)

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	Byron Station
1SI8802A, dated 06/27/94	1SI8802A, dated 09/14/94
1SI8802B, dated 01/06/96	1SI8802B, dated 09/14/94
2SI8802A, dated 06/27/94	2SI8802A, dated 03/24/95
2SI8802B, dated 06/27/94	2SI8802B, dated 03/16/95

3. Rising Stem MOV Data Sheets for :

Braidwood Station	Byron Station
1SI8802A, dated 08/10/95	1SI8802A, dated 08/05/94
1SI8802B, dated 08/10/95	1SI8802B, dated 08/08/94
2SI8802A, dated 08/10/95	2SI8802A, dated 08/08/94
2SI8802B, dated 08/10/95	2SI8802B, dated 08/08/94

4. MOV White Paper WP-134 Rev. 0, EPRIs MOV Testing Program Measured Valve Factors.

V. REFERENCES (cont)

5. Mechanical Engineering Design Forth Edition, Shigley and Mitchell

6. MOV White Paper 000, MOV Program Technical Guidance, Revision 2

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

VI. CALCULATIONS

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

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,

VI. CALCULATIONS

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.

Valve Factor (open) = (O10 thrust - avg open run load + (0.7854 * (stem dia)^2 * close line pressure - open line pressure)) / 0.7854 * (mean seat dia)^2 * differential pressure

Valve Factor Data Summary Table

STATION	VALVE	VOTES TEST #	TEST DATE	O10 THRUST	DIFF PRESS	LINE PRESS	OPEN LINE PRESS	OPEN RUN LOAD	OPEN VALVE FACTOR	REFERENCE
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

VI. CALCULATIONS

INPUTS:

Bonnet Pressure	$P_{bonnet} := 1400 \cdot psi$	Assumption 3
Upstream Pressure	$P_{up} := 890 \cdot psi$	Assumption 3
Downstream Pressure	$P_{down} := 0 \cdot psi$	Assumption 3
Disk Thickness	$t := 1.02 \cdot in$	Attachment A
Seat Radius	$a := 2.001 \cdot in$	Attachment A
Effective Hub Radius	$b := 1.056 \cdot in$	Attachment A
Hub Length	$L := 0.60 \cdot in$	Attachment A
Seat Angle	$theta := 7 \cdot deg$	Reference 3
Poisson's Ratio (disk)	$v := .3$	Typical of Stainless Steel
Mod. of Elast. (disk)	$E := 27.6 \cdot 10^6 \cdot psi$	Attachment B, 200 F
Static Pullout Force	$F_{po} := 6180 \cdot lbf$	Reference 2, Assumption 6
Open Valve Factor	$VF := .485$	Reference 3, Assumption 5
Stem Diameter	$D_{stem} := 1.25 \cdot in$	Reference 3

PRESSURE FORCE CALCULATIONS

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

$$mu := VF \cdot \frac{\cos(theta)}{1 - VF \cdot \sin(theta)} \qquad mu = 0.512$$

Average DP across disks:

$$DPavg := P_{bonnet} - \frac{P_{up} + P_{down}}{2} \qquad DPavg = 955 \cdot psi$$

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

$$D := \frac{E \cdot (t)^3}{12 \cdot \left(1 - v^2\right)} \qquad D = 2.682 \cdot 10^6 \cdot lbf \cdot in$$

$$G := \frac{E}{2 \cdot (1 + v)} \qquad G = 1.062 \cdot 10^7 \cdot psi$$

CALCULATION NO. *BRW 96-015*	PROJECT NO. *N/A*	PAGE NO. *12*

VI. CALCULATIONS

Geometry Factors: (Reference 1, Table 24)

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

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

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

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

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

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

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

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

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

$$M_{rb} := \frac{-DPavg\cdot a^2}{C_8}\cdot\left[\frac{C_9}{2\cdot a\cdot b}\cdot\left(a^2 - b^2\right) - L_{17}\right] \qquad M_{rb} = -579.387\cdot lbf$$

$$Q_b := \frac{DPavg}{2\cdot b}\cdot\left(a^2 - b^2\right) \qquad Q_b = 1306.281\cdot\frac{lbf}{in}$$

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

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

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VI. CALCULATIONS

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

$$K_{sa} := -0.3\cdot\left[2\cdot\ln\left(\frac{a}{b}\right) - 1 + \left(\frac{b}{a}\right)^2\right] \qquad K_{sa} = -0.16705$$

$$y_{sq} := \frac{K_{sa}\cdot DPavg\cdot a^2}{t\cdot G} \qquad y_{sq} = -5.8993\cdot 10^{-5}\cdot in$$

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

$$P_{force} := 3.1416\cdot\left(a^2 - b^2\right)\cdot DPavg \qquad P_{force} = 8667.254\cdot lbf$$

$$y_{stretch} := \frac{P_{force}}{3.1416\cdot b^2}\cdot\frac{L}{(2\cdot E)} \qquad y_{stretch} = 2.6891\cdot 10^{-5}\cdot in$$

Total Deflection due to pressure forces:

$$y_q := y_{bq} + y_{sq} - y_{stretch} \qquad y_q = -0.0001\cdot in$$

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

$$y_{sw} := -\left[\frac{1.2\cdot\left(\frac{a}{a}\right)\cdot\ln\left(\frac{a}{b}\right)\cdot a}{t\cdot G}\right] \qquad y_{sw} = -1.4174\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{bw} := -\left(\frac{a^3}{D}\right)\cdot\left[\left(\frac{C_2}{C_8}\right)\cdot\left[\left(\frac{a\cdot C_9}{b}\right) - L_9\right] - \left[\left(\frac{a}{b}\right)\cdot C_3\right] + L_3\right] \qquad y_{bw} = -1.2635\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{compr} := \frac{2\cdot a\cdot\pi}{3.1416\cdot b^2}\cdot\frac{L}{(2\cdot E)} \qquad y_{compr} = 3.9009\cdot 10^{-8}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_w := y_{bw} + y_{sw} - y_{compr} \qquad y_w = -3.071\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$F_s := 2\cdot\pi\cdot a\cdot\frac{y_q}{y_w} \qquad F_s = 5114\cdot lbf$$

VI. CALCULATIONS

UNSEATING FORCES

$F_{packing}$ is included in measured static pullout Force

$F_{piston} := \frac{\pi}{4} \cdot D_{stem}^{2} \cdot P_{bonnet}$ $\qquad F_{piston} = 1718.1 \cdot lbf$

$F_{vert} := \left[\pi \cdot \left(a^{2}\right)\right] \cdot \sin(theta) \cdot \left(2 \cdot P_{bonnet} - P_{up} - P_{down}\right)$ $\qquad F_{vert} = 2928 \cdot lbf$

$F_{preslock} := 2 \cdot F_{s} \cdot (mu \cdot \cos(theta) - \sin(theta))$ $\qquad F_{preslock} = 3947.4 \cdot lbf$

$F_{po} = 6180 \cdot lbf$

$F_{total} := -F_{piston} + F_{vert} + F_{preslock} + F_{po}$ $\qquad F_{total} = 11337 \cdot lbf$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	$MT := 16.97 \cdot ft \cdot lbf$	Reference 3, 11
Temperature Factor:	$Tf := 0.98$	Reference 3, Assumption 10
Degraded Voltage:	$DV := 409 \cdot volt$	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
Application Factor	$AF := 1.0$	Reference 11 sets AF to 1.0
Stem Factor @ $\mu = 0.20$	$SF := 0.0140 \cdot ft \cdot \frac{lbf}{lbf}$	Reference 3, Assumption 7

CALCULATIONS:

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

$MGC_{Open} = 11536 \cdot lbf$ $\qquad F_{total} = 11337 \cdot lbf$

$$MGC_{Margin} := \frac{MGC_{Open} - F_{total}}{F_{total}}$$ $MGC_{Margin} = 1.7 \cdot \%$

VI. 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.

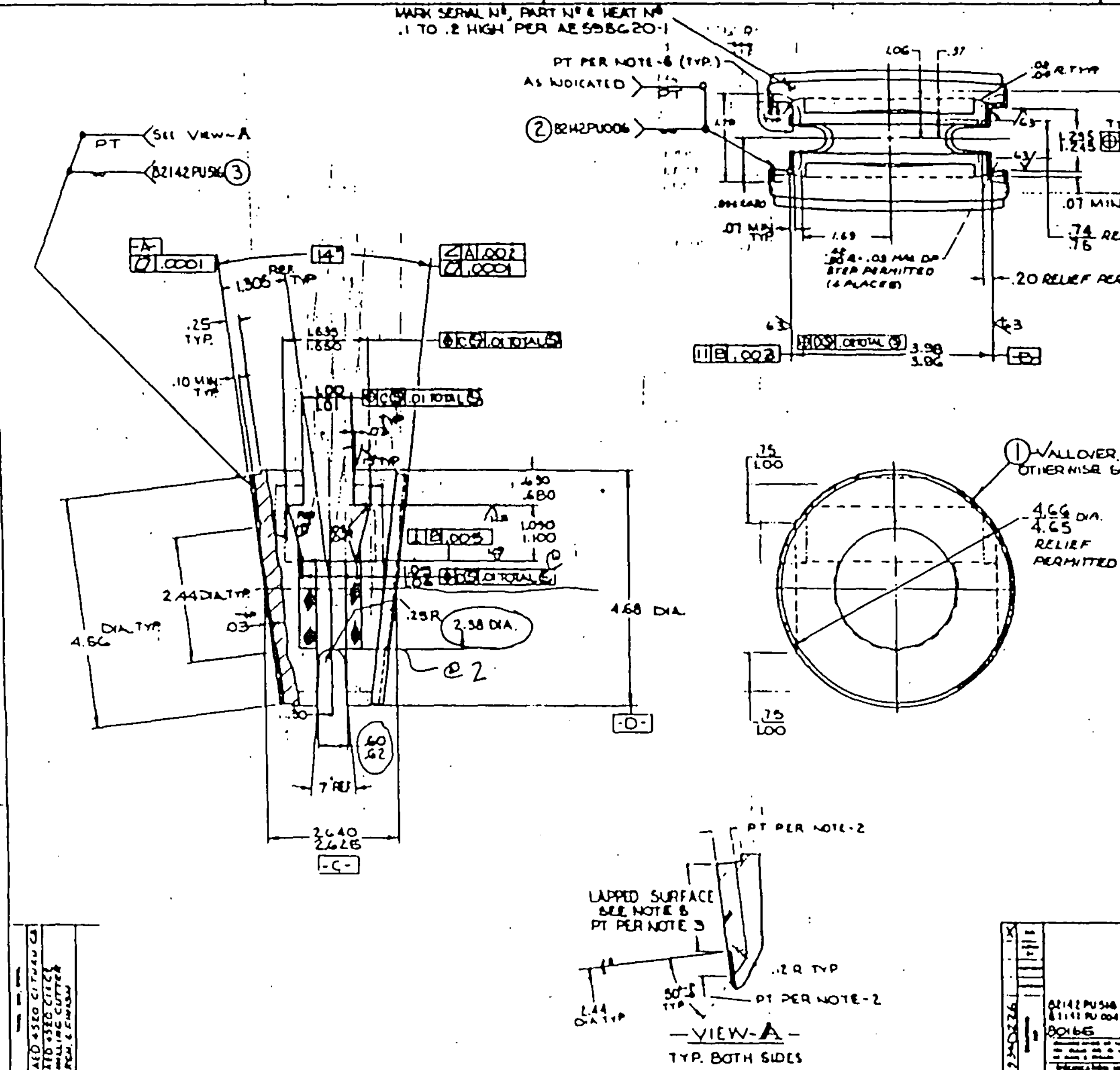
VI. LIMITATIONS

None.

IX. 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



-NOTES-

1- LIQUID PENETRANT EXAMINE ALL FINAL MACHINED SURFACES PER 80165-2, CL. AA.

2 LIQUID PENETRANT EXAMINE INDICATED SURFACES PER 80165-2, CL. OOK.

3 AFTER LAPPING INDICATED SURFACES, LIQUID PENETRANT PER 80165-2, 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 SEAT OF .08 MIN. WIDTH AROUND ENTIRE FACE OF DISC IS OBTAINED.

6 LIQUID PENETRANT EXAMINE INDICATED AREA PER 80165-2, CL 1-4K. PERFORM VISUAL EXAMINATION FOR OPEN TYPE DISCONTINUITIES WHICH DO NOT RESPOND TO PENETRANT EXAMINATION. VISUAL DISCONTINUITIES WHOSE DIMENSIONS ARE GREATER THAN 3/64 INCH IN ANY DIRECTION ARE UNACCEPTABLE.

WELD PAD		STELLITE 156			3
WELD ROD		STELLITE 6			2
DISC		316 SST			1

LIST OF MATERIALS

Westinghouse Electric Corporation

DISC-3/4-1500, 4-900 GATE VALVE DISC

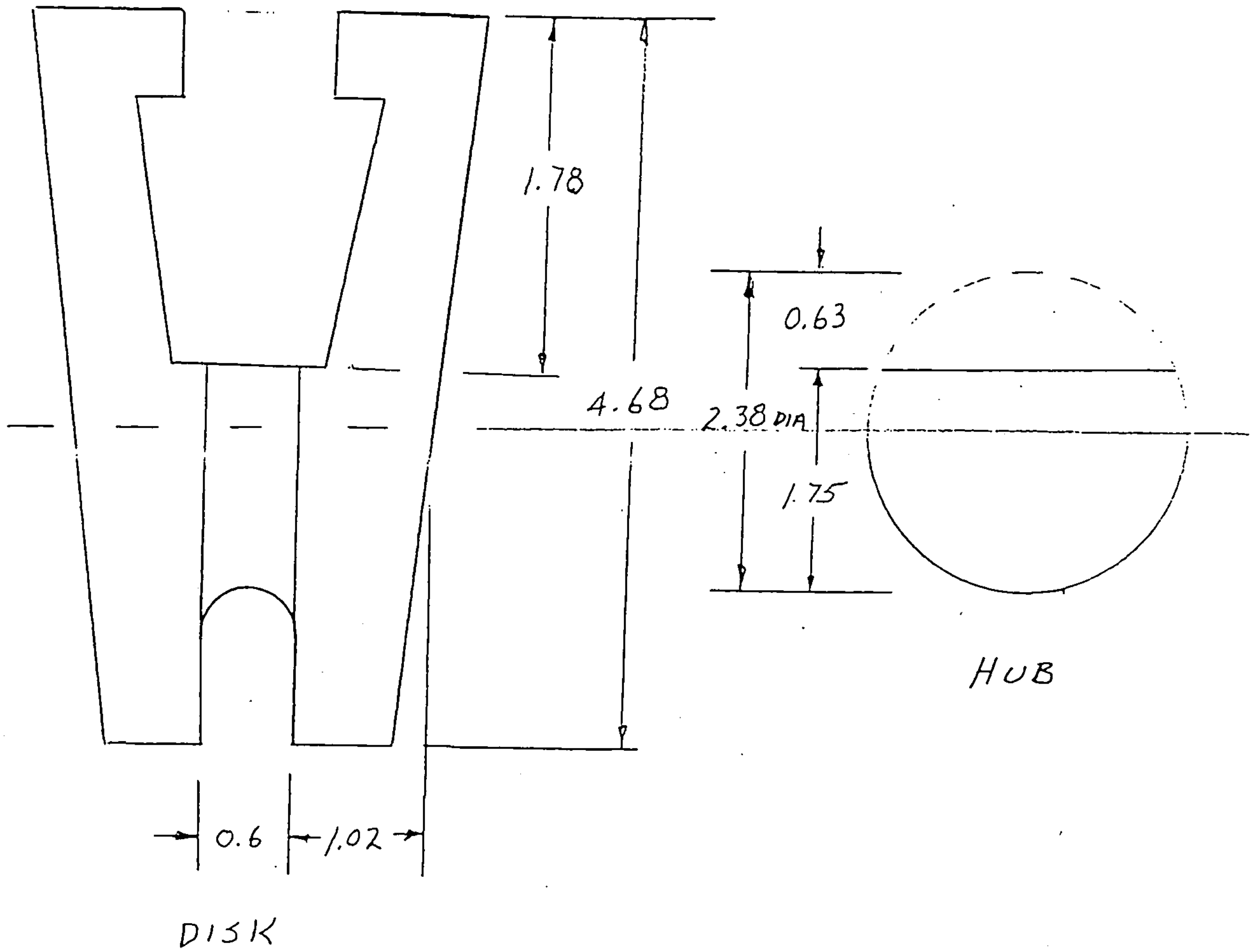
D 04808 934D225

N.T.S.

934D225

CALC BRW 96-015 REV.

Disk Dimensions



Effective Radius of Hub Section

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

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

h/D = 0.63in/2.38in = .264 interpolation from table pg 1-7 (REF 8)

Area/Circle = 0.21108

Area of Missing Section = $0.21108 * 4.449 in^2 = 0.939 in^2$

Area of Hub = $4.449 - 0.939 = 3.509 in^2$

Effective Area Diameter

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

Effective Hub Radius (b) = 2.114/2 = 1.056 in

L = 0.60in

t = 1.02

ATTACHMENT #7 ''
CALC BRW 96-015 REV.

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 1/3/96

R. C. Bedford
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Braidwood Station

ATTACHMENT A   -1-
CALC BRW 96-015 REV. C

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.

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Calc #: 96-015, Rev. 0
ATTACHMENT B,

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	$\boxed{27.6}$	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–$^1/_2$Mo	Mn–$^1/_4$Mo
Mn–$^1/_2$Mo	Mn–V

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

$^3/_4$Ni–$^1/_2$Mo–Cr–V	1Ni–$^1/_2$Cr–$^1/_2$Mo
$^1/_2$Ni–$^1/_2$Mo–V	$^3/_4$Ni–1Mo–$^3/_4$Cr
$^3/_4$Ni–$^1/_2$Mo–$^1/_3$Cr–V	$^1/_2$Ni–$^1/_2$Cr–$^1/_4$Mo–V
$^3/_4$Cr–$^3/_4$Ni–Cu–Al	2Ni–1Cu
$^3/_4$Cr–$^1/_2$Ni–Cu	2$^1/_2$Ni
$^3/_4$Ni–$^1/_2$Cu–Mo	3$^1/_2$Ni

(3) Material Group C consists of the following $^1/_2$–2Cr steels:

- $^1/_2$Cr–$^1/_2$Mo
- 1Cr–$^1/_2$Mo
- 1$^1/_4$Cr–$^1/_2$Mo–Si
- 1$^1/_4$Cr–$^1/_2$Mo
- 2Cr–$^1/_2$Mo

(4) Material Group D consists of the following 2$^1/_4$–3Cr steels:

- 2$^1/_4$Cr–1Mo
- 3Cr–1Mo

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

- 5Cr–$^1/_2$Mo
- 5Cr–$^1/_2$Mo–Si
- 5Cr–$^1/_2$Mo–Ti
- 7Cr–$^1/_2$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)

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:

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

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Concur T. Matty 2/27/91
T. Matty
Westinghouse

* Made up of Seat Bore 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.

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Concur Terence Matty 2/27/96
T. Matty
Westinghouse