CALCULATION TITLE PAGE

ComEd

BRAIDWOOD STATION

[x] Unit 1 [x] Unit 2 [ ] Unit 0

CALCULATION NO: 95-111

DESCRIPTION CODE: M14
DISCIPLINE CODE: Mechanical
SYSTEM CODE: RY

Title: Verification of Capability for Braidwood and Byron 3" 1(2)RY8000A & B Valves Susceptible to Pressure Locking

[x] Safety Related [ ] Augmented Quality [ ] Non-Safety Related

REFERENCE NUMBERS: (C011 Panel)

Type	Number	Type	Number
AEDV			
PROG			
RNID			
DCAT			
SSYS			

COMPONENT EPN: (C014 Panel)

EPN	Component Type
1RY8000A	L05
1RY8000B	L05
2RY8000A	L05
2RY8000B	L05

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

Doc Type	Sub Type	Document Number

REMARKS:

REV. No.	REVISING ORGANIZATION	APPROVED (Print & Sign)	DATE
1	Site Engineering	Bruce J. Acus [signature]	11/28/97

ComEd

CALCULATION REVISION PAGE

CALCULATION NO. 95-111 **PAGE NO.:** 2

REVISION SUMMARIES

REV: 1

REVISION SUMMARY: Revised calculation to incorporate enhanced motor gearing capability methodology in accordance with MOV White Paper WP-125 Revision 2. Updated the valve factor in accordance with the Margin Review Calculation sheets (per White Paper WP-172 revision 0). Utilizing this method for calculating motor gearing capability increases the margin between the predicted pressure locking thrust and actuator capability. Revised pages 1, 2, 3, 6, 7, 8, 10, 13, A3 and A4.

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

Prepared By: (Print/Sign) R.C BEDFORD / R.C. Bedford	**Date:** 11/25/97
Reviewed By: (Print/Sign) J. Tolar / J Tolar	**Date:** 11-25-97

Type of Review: [x] Detailed [ ] Alternate [ ] Test

Do any ASSUMPTIONS / ENGINEERING JUDGMENTS required later verification? [ ] Yes [x] No
Tracked By: ______________________

REV:

REVISION SUMMARY:

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

Prepared By: (Print/Sign)	**Date:**
Reviewed By: (Print/Sign)	**Date:**

Type of Review: [ ] Detailed [ ] Alternate [ ] Test

Do any ASSUMPTIONS / ENGINEERING JUDGMENTS required later verification? [ ] Yes [ ] No
Tracked By: ______________________

CALCULATION TABLE OF CONTENTS

CALCULATION NO. 95-111	REV. NO. 1	PAGE NO. 3
SECTION:	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 Pressurizer 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.

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 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)RY8000A&B Power Operated Relief Valve (PORV) Block valves. These valves are normally open in modes 1, 2, and 3, however, Technical Specification 3/4.4.4 allows one or both block valves to be closed due to excessive PORV seat leakage. One of the two block valves may be required to be opened in response to a Steam Generator Tube Rupture event as directed by the Emergency Operating Procedures. The potential exists that these valves could be closed and Reactor Coolant System (RCS) pressure could be trapped in the bonnet. Assuming that these valves would have to be opened under a design basis operating condition the pressure across the bonnet and upstream disc would be 2235 (operating RCS pressure) or less depending on how low the upstream (RCS) pressure would drop. Although it is not expected that these valves would have to be opened if RCS pressure dropped it is assumed that the upstream pressure is 350 psig (pressure at which the Residual Heat Removal system is placed in shutdown cooling). It is not expected that the bonnet pressure would increase above the pressure in the RCS due to RCS or ambient temperature conditions.

3. 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. These valves could not be differential pressure tested. An open valve factor of 0.420 (conservatively set equal to closed) will be

used for 1(2)RY8000A&B based on the assumed nominal value contained in the Margin Review Calculation Sheets for these valves (Reference 2).

4. 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 3. Both of these assumptions ensure the calculation is conservative and bounds all operating conditions.

5. The bonnet pressure is assumed to be the operating RCS pressure of 2235 psig. The downstream side of the valve is vented to the Pressurizer Relief Tank which is assumed to be at 0 psig. The upstream pressure is assumed to be 350 psig (pressure at which the Residual Heat Removal system is placed in shutdown cooling). This upstream pressure is conservative based UFSAR section 15.6.3, Steam Generator Tube Rupture, which specifies a low pressure of approximately 1400 psig.

6. 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 (reference 3). This value is conservative.

7. 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.

8. 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.

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
1RY8000A, dated 06/06/97	1RY8000A, dated 01/06/97
1RY8000B, dated 06/06/97	1RY8000B, dated 01/06/97
2RY8000A, dated 05/25/96	2RY8000A, dated 11/19/97
2RY8000B, dated 05/21/96	2RY8000B, dated 11/19/97

3. Rising Stem MOV Data Sheets for :

Braidwood Station	Byron Station
1RY8000A, dated 03/24/97	1RY8000A, dated 05/14/96
1RY8000B, dated 03/24/97	1RY8000B, dated 04/01/96
2RY8000A, dated 09/16/97	2RY8000A, dated 08/31/96
2RY8000B, dated 09/16/97	2RY8000B, dated 08/31/96

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. MOV White Paper 125 Revision 2, Installed Motor Capability Evaluation.

10. MOV White Paper 172 Revision 0, Miscellaneous Valve Factors.

VI. CALCULATIONS

MathCad 5.0+ calculations of the following for the group of valves listed:

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.

VI. CALCULATIONS

INPUTS:

Bonnet Pressure	$P_{bonnet} := 2235 \cdot psi$	Assumption 5
Upstream Pressure	$P_{up} := 350 \cdot psi$	Assumption 5
Downstream Pressure	$P_{down} := 0 \cdot psi$	Assumption 5
Disk Thickness	$t := 1.02 \cdot in$	Attachment A
Seat Radius	$a := 1.60937 \cdot in$	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)	$E := 27.0 \cdot 10^6 \cdot psi$	Attachment B (300 F)
Static Pullout Force	$F_{po} := 8090 \cdot lbf$	Reference 2 / Assumption 4
Open Valve Factor	$VF := .42$	Reference 2, 10
Stem Diameter	$D_{stem} := 1.25 \cdot in$	Reference 3

PRESSURE FORCE CALCULATIONS

Coefficient of friction between disk and seat: (Reference 4)

$$mu := VF \cdot \frac{\cos(theta)}{1 - VF \cdot \sin(theta)} \qquad mu = 0.439$$

Average DP across disks:

$$DPavg := P_{bonnet} - \frac{P_{up} + P_{down}}{2} \qquad DPavg = 2060 \cdot psi$$

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

$$D := \frac{E \cdot (t)^3}{12 \cdot \left(1 - v^2\right)} \qquad D = 2.624 \cdot 10^6 \cdot lbf \cdot in$$

$$G := \frac{E}{2 \cdot (1 + v)} \qquad G = 1.038 \cdot 10^7 \cdot psi$$

VI. CALCULATIONS

Geometry Factors: (Reference 1, Table 24)

$$C_2 := \frac{1}{4}\left[1 - \left(\frac{b}{a}\right)^2 \cdot \left(1 + 2\cdot\ln\left(\frac{a}{b}\right)\right)\right] \qquad C_2 = 0.05166$$

$$C_3 := \frac{b}{4\cdot a}\left[\left[\left(\frac{b}{a}\right)^2 + 1\right]\cdot\ln\left(\frac{a}{b}\right) + \left(\frac{b}{a}\right)^2 - 1\right] \qquad C_3 = 0.00546$$

$$C_8 := \frac{1}{2}\left[1 + \nu + (1 - \nu)\cdot\left(\frac{b}{a}\right)^2\right] \qquad C_8 = 0.80069$$

$$C_9 := \frac{b}{a}\left[\frac{1+\nu}{2}\cdot\ln\left(\frac{a}{b}\right) + \frac{1-\nu}{4}\left[1 - \left(\frac{b}{a}\right)^2\right]\right] \qquad C_9 = 0.2451$$

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

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

$$L_{11} := \frac{1}{64}\left[1 + 4\cdot\left(\frac{b}{a}\right)^2 - 5\cdot\left(\frac{b}{a}\right)^4 - 4\cdot\left(\frac{b}{a}\right)^2\cdot\left[2 + \left(\frac{b}{a}\right)^2\right]\cdot\ln\left(\frac{a}{b}\right)\right] \qquad L_{11} = 0.00049$$

$$L_{17} := \frac{1}{4}\left[1 - \frac{1-\nu}{4}\cdot\left[1 - \left(\frac{b}{a}\right)^4\right] - \left(\frac{b}{a}\right)^2\cdot\left[1 + (1+\nu)\cdot\ln\left(\frac{a}{b}\right)\right]\right] \qquad L_{17} = 0.04777$$

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

$$M_{rb} := \frac{-DPavg\cdot a^2}{C_8}\left[\frac{C_9}{2\cdot a\cdot b}\cdot\left(a^2 - b^2\right) - L_{17}\right] \qquad M_{rb} = -390.43\cdot lbf$$

$$Q_b := \frac{DPavg}{2\cdot b}\cdot\left(a^2 - b^2\right) \qquad Q_b = 1438.621\cdot\frac{lbf}{in}$$

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

$$y_{bq} := M_{rb}\cdot\frac{a^2}{D}\cdot C_2 + Q_b\cdot\frac{a^3}{D}\cdot C_3 - \frac{DPavg\cdot a^4}{D}\cdot L_{11} \qquad y_{bq} = -1.0025\cdot 10^{-5}\cdot in$$

VI. CALCULATIONS

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

$$K_{sa} := -0.3 \cdot \left[2 \cdot \ln\left(\frac{a}{b}\right) - 1 + \left(\frac{b}{a}\right)^2 \right] \qquad K_{sa} = -0.08198$$

$$y_{sq} := \frac{K_{sa} \cdot DPavg \cdot a^2}{t \cdot G} \qquad y_{sq} = -4.1293 \cdot 10^{-5} \cdot in$$

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

$$P_{force} := 3.1416 \cdot \left(a^2 - b^2\right) \cdot DPavg \qquad P_{force} = 9545.336 \cdot lbf$$

$$y_{stretch} := \frac{P_{force}}{3.1416 \cdot b^2} \cdot \frac{L}{(2 \cdot E)} \qquad y_{stretch} = 3.0274 \cdot 10^{-5} \cdot in$$

Total Deflection due to pressure forces:

$$y_q := y_{bq} + y_{sq} - y_{stretch} \qquad y_q = -8.1591 \cdot 10^{-5} \cdot in$$

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

$$y_{sw} := -\left[\frac{1.2 \cdot \left(\frac{a}{a}\right) \cdot \ln\left(\frac{a}{b}\right) \cdot a}{t \cdot G} \right] \qquad y_{sw} = -7.6824 \cdot 10^{-8} \cdot \frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{bw} := -\left(\frac{a^3}{D}\right) \cdot \left[\left(\frac{C_2}{C_8}\right) \cdot \left[\left(\frac{a \cdot C_9}{b}\right) - L_9 \right] - \left[\left(\frac{a}{b}\right) \cdot C_3 \right] + L_3 \right] \qquad y_{bw} = -2.5057 \cdot 10^{-8} \cdot \frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{compr} := \frac{2 \cdot a \cdot \pi}{3.1416 \cdot b^2} \cdot \frac{L}{(2 \cdot E)} \qquad y_{compr} = 3.2071 \cdot 10^{-8} \cdot \frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_w := y_{bw} + y_{sw} - y_{compr} \qquad y_w = -1.3395 \cdot 10^{-7} \cdot \frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$F_s := 2 \cdot \pi \cdot a \cdot \frac{y_q}{y_w} \qquad F_s = 6159.3 \cdot lbf$$

VI. CALCULATIONS

UNSEATING FORCES

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

$F_{piston} := \frac{\pi}{4} \cdot D_{stem}^{2} \cdot P_{bonnet}$ $\qquad F_{piston} = 2742.76 \cdot lbf$

$F_{vert} := (\pi \cdot a^2) \cdot \sin(theta) \cdot (2 \cdot P_{bonnet} - P_{up} - P_{down})$ $\qquad F_{vert} = 4085.58 \cdot lbf$

$F_{preslock} := 2 \cdot F_s \cdot (mu \cdot \cos(theta) - \sin(theta))$ $\qquad F_{preslock} = 3870.67 \cdot lbf$

$F_{po} = 8090 \cdot lbf$

$F_{total} := -F_{piston} + F_{vert} + F_{preslock} + F_{po}$ $\qquad F_{total} = 13303 \cdot lbf$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	$MT := 16.97 \cdot ft \cdot lbf$	Reference 3, 9
Temperature Factor:	$Tf := 0.834$	Reference 3, Assumption 8
Degraded Voltage:	$DV := 408 \cdot volt$	Reference 3 / Assumption 4
Under Voltage Factor:	$n := 2.2769$	Reference 6, 9
Overall Gear Ratio	$OAR := 52.2$	Reference 3
Pullout Efficiency	$EFF := 0.40$	Reference 3
Stem Factor @ $\mu = 0.20$	$SF := 0.0140 \cdot ft \cdot \frac{lbf}{lbf}$	Reference 3 / Assumption 6

CALCULATIONS:

$$MGC_{Open} := \frac{\left(\frac{DV}{460 \cdot volt}\right)^{n} \cdot MT \cdot OAR \cdot Tf \cdot EFF}{SF}$$ (Reference 6, 9)

$MGC_{Open} = 16063 \cdot lbf$ $\qquad F_{total} = 13303 \cdot lbf$

$MGC_{Margin} := \frac{MGC_{Open} - F_{total}}{F_{total}}$ $\qquad MGC_{Margin} = 20.7 \cdot \%$

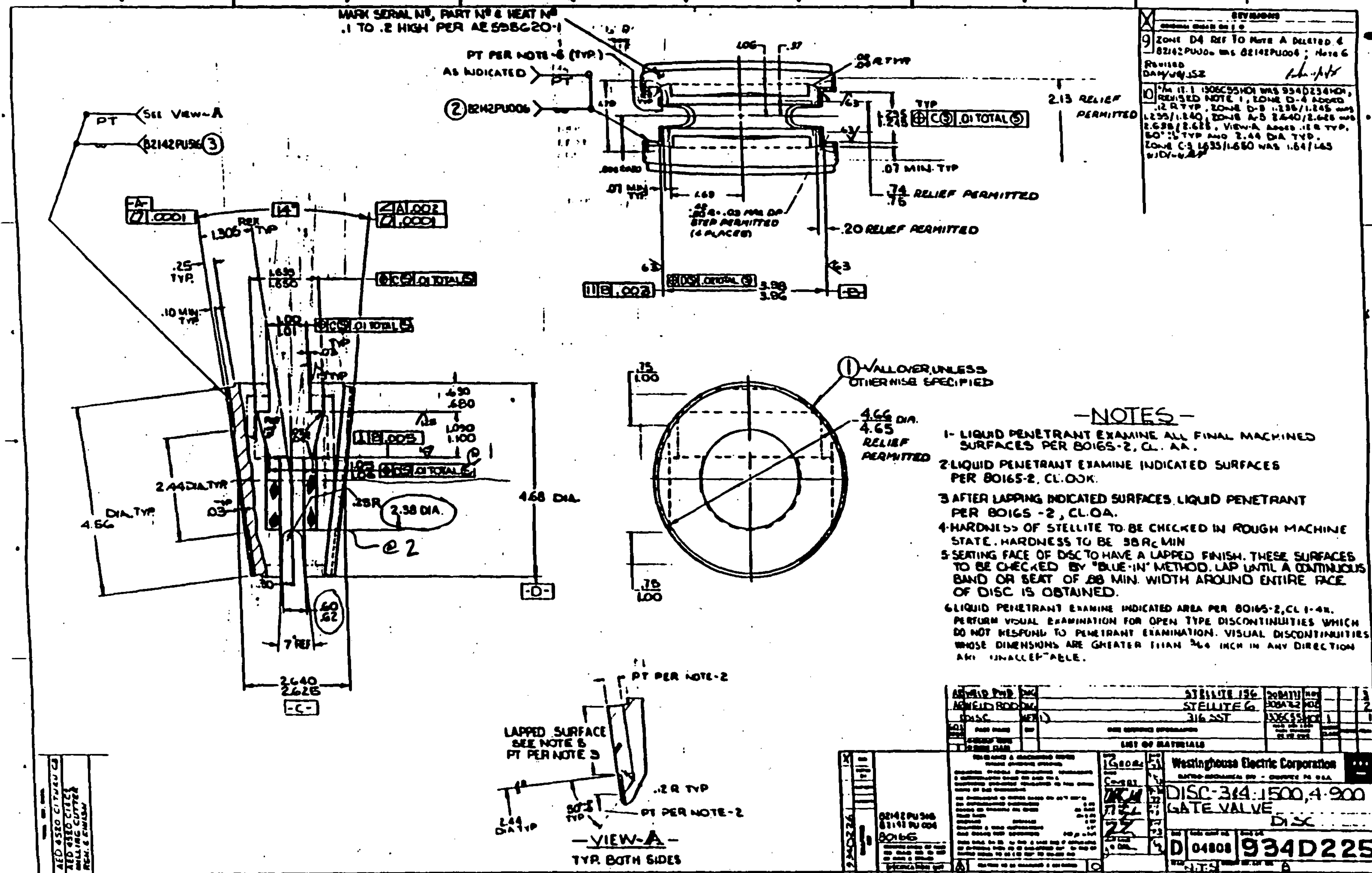
VII. SUMMARY AND CONCLUSIONS

The results of the calculation indicate that with all the indicated conservatism inherent in the inputs, the 1(2)RY8000A&B PORV Block 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

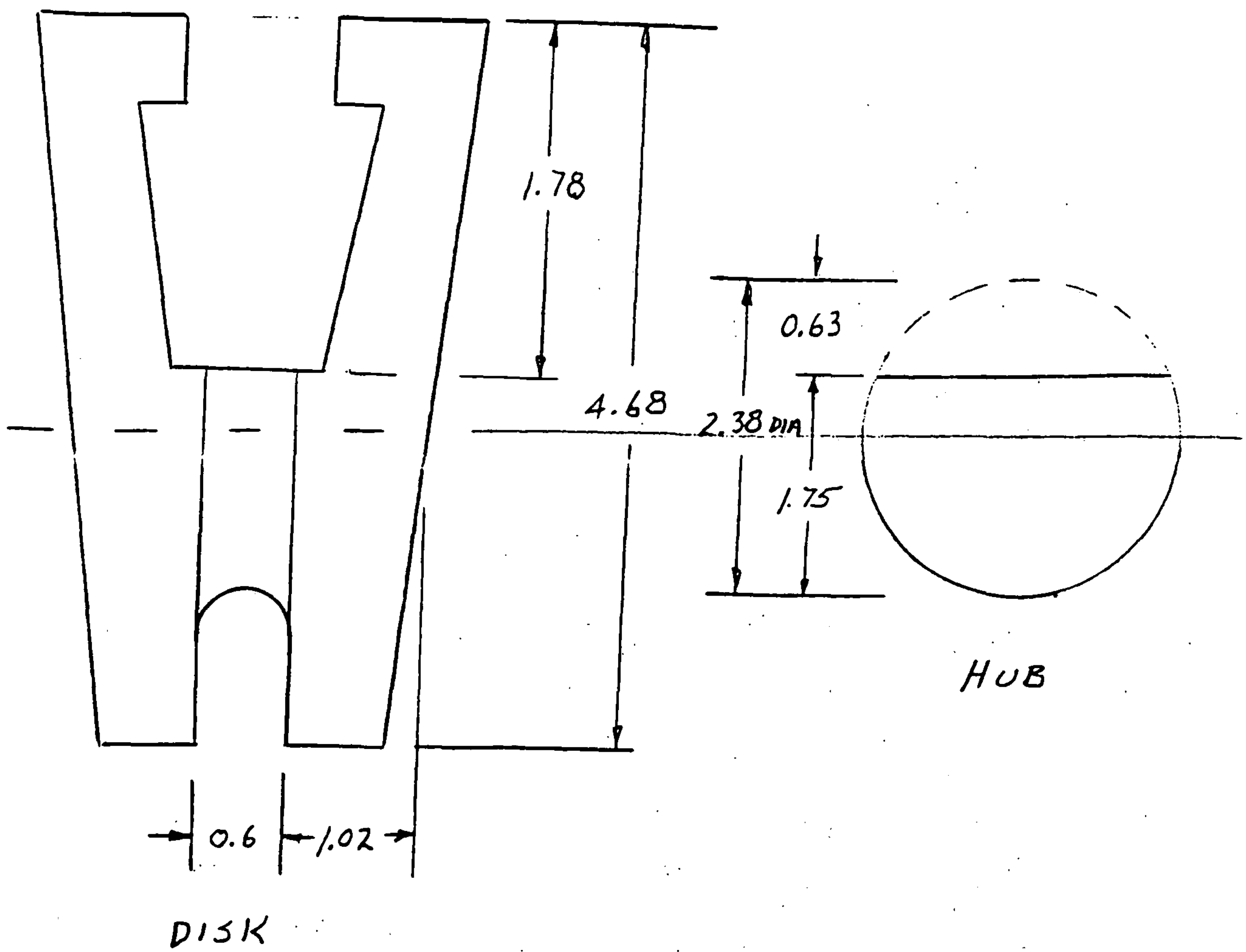
(A) Westinghouse Drawing # 934D225 (Disc)
Hand Sketch of Disc Dimensions provided for clarity
Record of Conversation dated 01/03/96
Record of Conversation dated 02/12/96

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



CALC 95-111 REV. 1

Disk Dimensions



<u>Effective Radius of Hub Section</u>

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

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

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 T. 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.

R. C. Bedford
R. C. Bedford
MOV Programs
Braidwood Station

Concur Terrence J Matty 2/29/96
T. Matty
Westinghouse

Calc #: 95-111, Rev. 1
ATTACHMENT B1

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–1/2Mo
- Mn–1/2Mo
- Mn–1/4Mo
- Mn–V

(2) Material Group B consists of the following Ni steels:
- 3/4Ni–1/2Mo–Cr–V
- 1/2Ni–1/2Mo–V
- 3/4Ni–1/2Mo–1/3Cr–V
- 3/4Cr–3/4Ni–Cu–Al
- 3/4Cr–1/2Ni–Cu
- 3/4Ni–1/2Cu–Mo
- 1Ni–1/2Cr–1/2Mo
- 3/4Ni–1Mo–3/4Cr
- 1/2Ni–1/2Cr–1/4Mo–V
- 2Ni–1Cu
- 2 1/2Ni
- 3 1/2Ni

(3) Material Group C consists of the following 1/2–2Cr steels:
- 1/2Cr–1/2Mo
- 1Cr–1/2Mo
- 1 1/4Cr–1/2Mo–Si
- 1 1/4Cr–1/2Mo
- 2Cr–1/2Mo

(4) Material Group D consists of the following 2 1/4–3Cr steels:
- 2 1/4Cr–1Mo
- 3Cr–1Mo

(5) Material Group E consists of the following 5–9Cr steels:
- 5Cr–1/2Mo
- 5Cr–1/2Mo–Si
- 5Cr–1/2Mo–Ti
- 7Cr–1/2Mo
- 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–8Ni–N
- 16Cr–12Ni
- 18Cr–13Ni–3Mo
- 16Cr–12Ni–2Mo–N
- 18Cr–3Ni–13Mn
- 18Cr–10Ni–Ti
- 18Cr–10Ni–Cb
- 18Cr–18Ni–2Si
- 20Cr–6Ni–9Mn
- 22Cr–13Ni–5Mn
- 23Cr–12Ni
- 25Cr–20Ni

(Final)