

COMMONWEALTH EDISON COMPANY
CALCULATION TITLE PAGE

CALCULATION NO. 95-111		PAGE NO.: 1
<input checked="" type="checkbox"/> SAFETY RELATED	<input type="checkbox"/> REGULATORY RELATED	<input type="checkbox"/> NON-SAFETY RELATED
<u>CALCULATION TITLE:</u> Verification of Capability for Braidwood and Byron 3" 1(2)RY8000A & B Valves Susceptible to Pressure Locking		
STATION/UNIT: Braidwood & Byron/1&2	SYSTEM ABBREVIATION: RY	
EQUIPMENT NO.: (IF APPL.) 1RY8000A 1RY8000B 2RY8000A 2RY8000B	PROJECT NO.: (IF APPL.) N/A	
REV: 0	STATUS: QA SERIAL NO. OR CHRON NO.	DATE: <u>1</u> / <u>1</u> /96
PREPARED BY: <u>R.C. Bedford</u>	<u>/R. C. Bedford</u>	DATE: <u>2</u> / <u>12</u> /96
REVISION SUMMARY: Initial issue.		
REVIEWED BY: <u>J.D. Tolar</u>	<u>2-12-96</u>	<u>J. D. Tolar</u>
REVIEW METHOD: <u>Detailed review</u>	COMMENTS (C OR NC): <u>NC</u>	
APPROVED BY: <u>Bruce J. Acas</u> <u>2/13/96</u> <u>Bruce J. Acas</u>		

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CALCULATION REVISION PAGE

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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 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.

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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. In calculating MGC_{Open} , Motor Start Torque, Motor Temperature Factor, Degraded Voltage, Pullout Efficiency, and an Application Factor of 0.90 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.

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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 0.523 will be used for 1(2)RY8000A&B based on the maximum assumed value contained in the Rising Stem MOV Data Sheets for these valves.
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

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III. ASSUMPTIONS (con't)

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 and application factor are 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

1RY8000A, dated 01/06/96
 1RY8000B, dated 01/06/96
 2RY8000A, dated 01/26/95
 2RY8000B, dated 01/26/95

Byron Station

1RY8000A, dated 10/20/94
 1RY8000B, dated 10/20/94
 2RY8000A, dated 06/23/94
 2RY8000B, dated 06/23/94

3. Rising Stem MOV Data Sheets for :

Braidwood Station

1RY8000A, dated 11/28/94
 1RY8000B, dated 11/28/94
 2RY8000A, dated 11/28/94
 2RY8000B, dated 11/28/94

Byron Station

1RY8000A, dated 04/06/94
 1RY8000B, dated 04/06/94
 2RY8000A, dated 04/06/94
 2RY8000B, dated 04/06/94

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V. REFERENCES (con't)

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 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

VI. CALCULATIONS

MathCad 5.0+ calculations of the following for each of the three groups 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.

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VI. CALCULATIONS

INPUTS:

Bonnet Pressure	$P_{\text{bonnet}} := 2235 \cdot \text{psi}$	Assumption 5
Upstream Pressure	$P_{\text{up}} := 350 \cdot \text{psi}$	Assumption 5
Downstream Pressure	$P_{\text{down}} := 0 \cdot \text{psi}$	Assumption 5
Disk Thickness	$t := 1.02 \cdot \text{in}$	Attachment A
Seat Radius	$a := 1.60937 \cdot \text{in}$	Attachment A
Effective Hub Radius	$b := 1.056 \cdot \text{in}$	Attachment A
Hub Length	$L := 0.60 \cdot \text{in}$	Attachment A
Seat Angle	$\theta := 7 \cdot \text{deg}$	Reference 3
Poisson's Ratio (disk)	$\nu := .3$	Typical of Stainless Steel
Mod. of Elast. (disk)	$E := 27.0 \cdot 10^6 \cdot \text{psi}$	Attachment B (300 F)
Static Pullout Force	$F_{\text{po}} := 5933 \cdot \text{lbf}$	Reference 2 / Assumption 4
Open Valve Factor	$VF := .523$	Reference 3
Stem Diameter	$D_{\text{stem}} := 1.25 \cdot \text{in}$	Reference 3

PRESSURE FORCE CALCULATIONS

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

$$\mu := VF \cdot \frac{\cos(\theta)}{1 - VF \cdot \sin(\theta)} \quad \mu = 0.554$$

Average DP across disks:

$$DP_{\text{avg}} := P_{\text{bonnet}} - \frac{P_{\text{up}} + P_{\text{down}}}{2} \quad DP_{\text{avg}} = 2060 \cdot \text{psi}$$

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

$$D := \frac{E \cdot (t)^3}{12 \cdot (1 - \nu^2)} \quad D = 2.624 \cdot 10^6 \cdot \text{lbf} \cdot \text{in}$$

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

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VI. CALCULATIONS

Geometry Factors: (Reference 1, Table 24)

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

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

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

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

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

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

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

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

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

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

$$Q_b := \frac{DP_{avg}}{2 \cdot b} \cdot (a^2 - b^2) \quad Q_b = 1438.621 \cdot \frac{\text{lbf}}{\text{in}}$$

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

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

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VI. CALCULATIONS

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

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

$$K_{sa} = -0.08198$$

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

$$y_{sq} = -4.1293 \cdot 10^{-5} \text{ in}$$

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

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

$$P_{force} = 9545.336 \text{ lbf}$$

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

$$y_{stretch} = 3.0274 \cdot 10^{-5} \text{ in}$$

Total Deflection due to pressure forces:

$$y_q := y_{sq} + y_{stretch}$$

$$y_q = -8.1591 \cdot 10^{-5} \text{ in}$$

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

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

(per lbf/in)

$$y_{sw} = -7.6824 \cdot 10^{-3} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_{bw} = -2.5057 \cdot 10^{-8} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_{compr} = 3.2071 \cdot 10^{-8} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_w = -1.3395 \cdot 10^{-7} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

$$F_s = 6159.3 \text{ lbf}$$

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VI. CALCULATIONS

UNSEATING FORCES

F_{packing} is included in measured static pullout Force

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

$$F_{\text{piston}} = 2742.76 \cdot \text{lbf}$$

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

$$F_{\text{vert}} = 4085.58 \cdot \text{lbf}$$

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

$$F_{\text{preslock}} = 5277.76 \cdot \text{lbf}$$

$$F_{\text{po}} = 5933 \cdot \text{lbf}$$

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

$$F_{\text{total}} = 12554 \cdot \text{lbf}$$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	MR := 15.0-ft·lbf	Reference 3
Temperature Factor:	Tf := 0.834	Reference 3, Assumption 8
Degraded Voltage:	DV := 408-volt	Reference 3 / Assumption 4
Under Voltage Factor:	n := 2.0	Reference 6
Overall Gear Ratio	OAR := 52.2	Reference 3
Pullout Efficiency	POE := 0.40	Reference 3
Application Factor	AF := 0.90	Reference 3, Assumption 8
Stem Factor @ $\mu=0.20$	SF := 0.0140-ft· $\frac{\text{lbf}}{\text{lbf}}$	Reference 3 / Assumption 6

CALCULATIONS:

$$\text{MGC}_{\text{Open}} := \frac{\left(\frac{\text{DV}}{460 \cdot \text{volt}}\right)^n \cdot \text{MR} \cdot \text{OAR} \cdot \text{Tf} \cdot \text{POE} \cdot \text{AF}}{\text{SF}} \quad (\text{Reference 6})$$

$$\text{MGC}_{\text{Open}} = 13210 \cdot \text{lbf}$$

$$F_{\text{total}} = 12554 \cdot \text{lbf}$$

$$\text{MGC}_{\text{Margin}} := \frac{\text{MGC}_{\text{Open}} - F_{\text{total}}}{F_{\text{total}}}$$

$$\text{MGC}_{\text{Margin}} = 5.2 \cdot \%$$

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VII. SUMMARY AND CONCLUSIONS

(FINAL)

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. LIMITATIONS

None.

IX. ATTACHMENTS

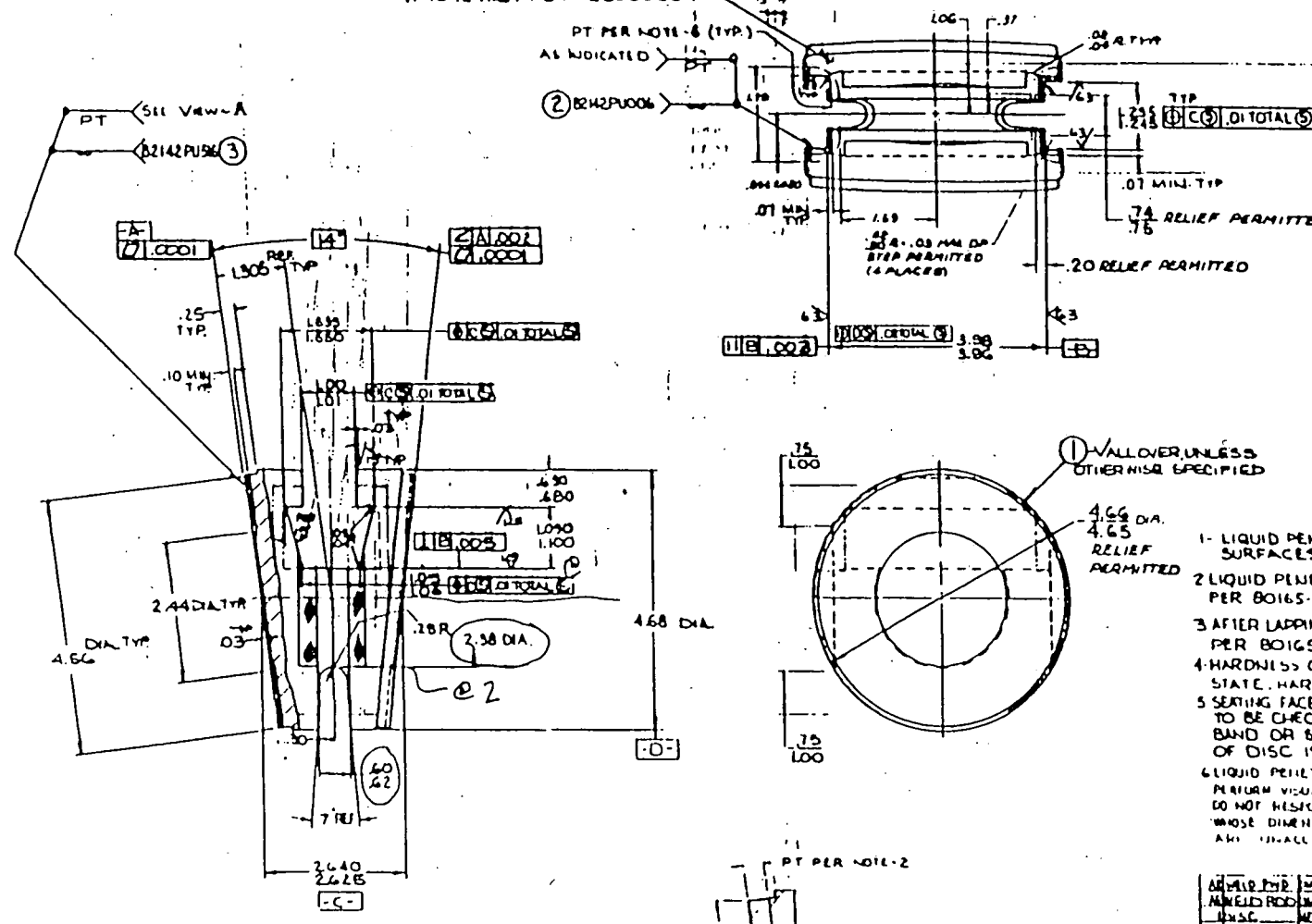
- (A) Westinghouse Drawing # 934D225 (Disc)
 Hand Sketch of Disc Dimensions provided for clarity
 Record of Conversation dated 01/03/96
 Record of Conversation dated 02/12/96

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

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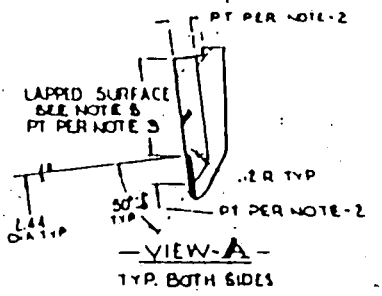
MARK SERIAL NO., PART NO. & HEAT NO.
 .1 TO .2 HIGH PER AE 598620-1



STRENGTH
 9 ZONE D4 REF TO NOTE A DELIBERATE
 82142PUSH WAS 82142PUSH; NOTE C
 REVISED
 DAN J. J. S.
 10 7/8 11.1 (SELECTION) WAS 934D225-10H,
 REVISED NOTE 1, ZONE D-A APPROX
 12 R TYP, ZONE D-B 2.440/1.125 WAS
 1.635/1.140, ZONE A-S 2.440/1.125 WAS
 1.635/1.140, VIEW A APPROX .18 TYP,
 80% TYP AND 2.44 DIA TYP.
 ZONE C-S 1.635/1.660 WAS 1.64/1.65
 WIDTH .48

-NOTES-

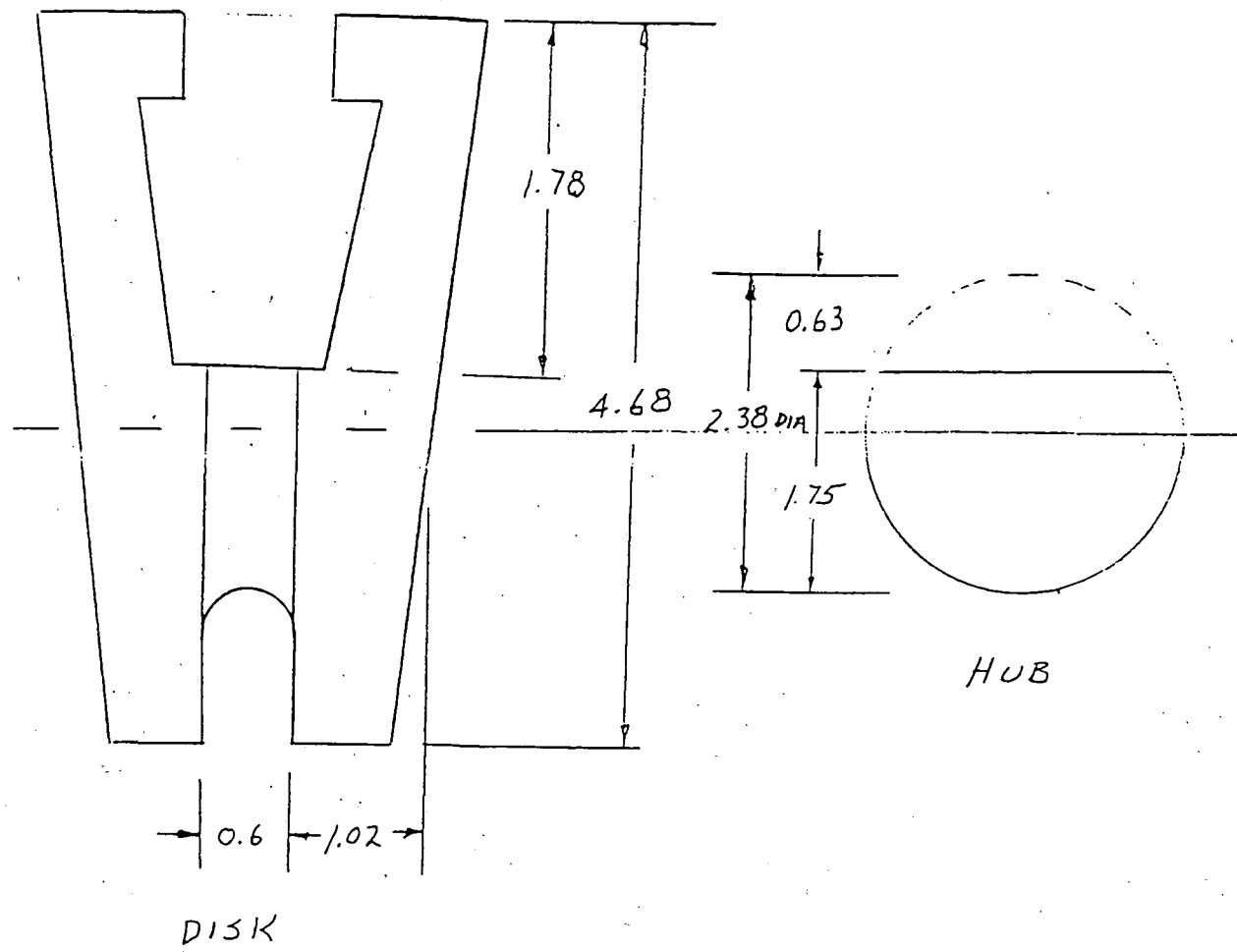
- 1- LIQUID PENETRANT EXAMINE ALL FINAL MACHINED SURFACES PER 80165-2, CL. A.A.
- 2- LIQUID PENETRANT EXAMINE INDICATED SURFACES PER 80165-2, CL. OJK.
- 3- AFTER LAPPING INDICATED SURFACES LIQUID PENETRANT PER 80165-2, CL. O.A.
- 4- HARDNESS OF STELLITE TO BE CHECKED IN ROUGH MACHINE STATE. HARDNESS TO BE 50 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 1/8 MIN. WIDTH AROUND ENTIRE FACE OF DISC IS OBTAINED.
- 6- LIQUID PENETRANT EXAMINE INDICATED AREA PER 80165-2, CL. I-4K. PERFORM VISUAL EXAMINATION FOR OPEN TYPE DISCONTINUITIES WHICH DO NOT RESPOND TO PENETRANT EXAMINATION. VISUAL DISCONTINUITIES WHOSE DIMENSIONS ARE GREATER THAN 2/64 INCH IN ANY DIRECTION ARE UNACCEPTABLE.



STELLITE 316 SST		316 SST	
Westinghouse Electric Corporation			
DISC-314-1500, 4-900			
GATE VALVE			
DISC			
D 04808	934D225		

ATTACHMENT A
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Disk Dimensions



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 pg 1-7 (Ref 8)

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 = 0.60in.

t = 1.02

Record of Conversation

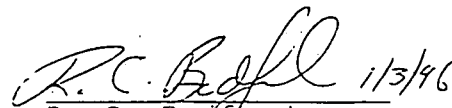
Per conversation with T. Matty of Westinghouse on 01/03/96 at 1345 (Phone 412-374-6401) the following seat ring dimensions were obtained for the listed valves:

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


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

Materials	Modulus of Elasticity $E = \text{Value Given} \times 10^6$ psi, for Temp., °F, of											
	-325	-200	-100	70	200	300	400	500	600	700	800	900
Carbon steels with C \leq 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- $\frac{1}{2}$ Mo
Mn- $\frac{1}{2}$ Mo Mn-V

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

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

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

2 $\frac{1}{2}$ 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-13Ni-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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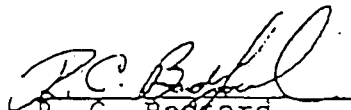
Concur

T. Matty / *Matty* 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.


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Braidwood Station

Concur


T. Matty
Westinghouse

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CALCULATION TITLE PAGE

CALCULATION NO. 95-111

PAGE NO.: 1

SAFETY RELATED REGULATORY RELATED NON-SAFETY RELATED

CALCULATION TITLE:

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

STATION/UNIT: Braidwood & Byron/1&2

SYSTEM ABBREVIATION: RY

EQUIPMENT NO.: (IF APPL.)

PROJECT NO.: (IF APPL.)

1RY8000A
1RY8000B
2RY8000A
2RY8000B

N/A

REV: 0 STATUS: QA SERIAL NO. OR CHRON NO. DATE: 1 / 196

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. Hoas 2/13/96 / Bruce J. Hoas

COMMONWEALTH EDISON COMPANY
CALCULATION REVISION PAGE

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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

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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 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.

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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. In calculating MGC_{Open} , Motor Start Torque, Motor Temperature Factor, Degraded Voltage, Pullout Efficiency, and an Application Factor of 0.90 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.

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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 0.523 will be used for 1(2)RY8000A&B based on the maximum assumed value contained in the Rising Stem MOV Data Sheets for these valves.
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

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III. ASSUMPTIONS (con't) :

- 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 and application factor are 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

1RY8000A, dated 01/06/96
 1RY8000B, dated 01/06/96
 2RY8000A, dated 01/26/95
 2RY8000B, dated 01/26/95

Byron Station.

1RY8000A, dated 10/20/94
 1RY8000B, dated 10/20/94
 2RY8000A, dated 06/23/94
 2RY8000B, dated 06/23/94

3. Rising Stem MOV Data Sheets for :

Braidwood Station

1RY8000A, dated 11/28/94
 1RY8000B, dated 11/28/94
 2RY8000A, dated 11/28/94
 2RY8000B, dated 11/28/94

Byron Station

1RY8000A, dated 04/06/94
 1RY8000B, dated 04/06/94
 2RY8000A, dated 04/06/94
 2RY8000B, dated 04/06/94

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V. REFERENCES (con't)

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 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

VI. CALCULATIONS

MathCad 5.0+ calculations of the following for each of the three groups 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.

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VI. CALCULATIONS

INPUTS:

Bonnet Pressure	$P_{\text{bonnet}} := 2235 \cdot \text{psi}$	Assumption 5
Upstream Pressure	$P_{\text{up}} := 350 \cdot \text{psi}$	Assumption 5
Downstream Pressure	$P_{\text{down}} := 0 \cdot \text{psi}$	Assumption 5
Disk Thickness	$t := 1.02 \cdot \text{in}$	Attachment A
Seat Radius	$a := 1.60937 \cdot \text{in}$	Attachment A
Effective Hub Radius	$b := 1.056 \cdot \text{in}$	Attachment A
Hub Length	$L := 0.60 \cdot \text{in}$	Attachment A
Seat Angle	$\theta := 7 \cdot \text{deg}$	Reference 3
Poisson's Ratio (disk)	$\nu := .3$	Typical of Stainless Steel
Mod. of Elast. (disk)	$E := 27.0 \cdot 10^6 \cdot \text{psi}$	Attachment B (300 F)
Static Pullout Force	$F_{\text{po}} := 5933 \cdot \text{lbf}$	Reference 2 / Assumption 4
Open Valve Factor	$VF := .523$	Reference 3
Stem Diameter	$D_{\text{stem}} := 1.25 \cdot \text{in}$	Reference 3

PRESSURE FORCE CALCULATIONS

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

$$\mu := VF \cdot \frac{\cos(\theta)}{1 - VF \cdot \sin(\theta)} \quad \mu = 0.554$$

Average DP across disks:

$$DP_{\text{avg}} := P_{\text{bonnet}} - \frac{P_{\text{up}} + P_{\text{down}}}{2} \quad DP_{\text{avg}} = 2060 \cdot \text{psi}$$

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

$$D := \frac{E \cdot (t)^3}{12 \cdot (1 - \nu^2)} \quad D = 2.624 \cdot 10^6 \cdot \text{lbf} \cdot \text{in}$$

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

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VI. CALCULATIONS

Geometry Factors: (Reference 1, Table 24)

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

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

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

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

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

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

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

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

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

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

$$Q_b := \frac{DP_{avg}}{2 \cdot b} \cdot (a^2 - b^2) \quad Q_b = 1438.621 \cdot \frac{\text{lbf}}{\text{in}}$$

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

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

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VI. CALCULATIONS

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

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

$$K_{sa} = -0.08198$$

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

$$y_{sq} = -4.1293 \cdot 10^{-5} \cdot \text{in}$$

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

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

$$P_{force} = 9545.336 \cdot \text{lbf}$$

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

$$y_{stretch} = 3.0274 \cdot 10^{-5} \cdot \text{in}$$

Total Deflection due to pressure forces:

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

$$y_q = -8.1591 \cdot 10^{-5} \cdot \text{in}$$

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

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

(per lbf/in)

$$y_{sw} = -7.6824 \cdot 10^{-8} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_{bw} = -2.5057 \cdot 10^{-8} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_{compr} = 3.2071 \cdot 10^{-8} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_w = -1.3395 \cdot 10^{-7} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

$$F_s = 6159.3 \cdot \text{lbf}$$

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CALCULATION NO. 95-111

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VI. CALCULATIONS

UNSEATING FORCES

F_{packing} is included in measured static pullout Force

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

$$F_{\text{piston}} = 2742.76 \cdot \text{lbf}$$

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

$$F_{\text{vert}} = 4085.58 \cdot \text{lbf}$$

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

$$F_{\text{preslock}} = 5277.76 \cdot \text{lbf}$$

$$F_{\text{po}} = 5933 \cdot \text{lbf}$$

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

$$F_{\text{total}} = 12554 \cdot \text{lbf}$$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	MR := 15.0 ft·lbf	Reference 3
Temperature Factor:	Tf := 0.834	Reference 3, Assumption 8
Degraded Voltage:	DV := 408·volt	Reference 3 / Assumption 4
Under Voltage Factor:	n := 2.0	Reference 6
Overall Gear Ratio	OAR := 52.2	Reference 3
Pullout Efficiency	POE := 0.40	Reference 3
Application Factor	AF := 0.90	Reference 3, Assumption 8
Stem Factor @ $\mu=0.20$	SF := 0.0140·ft· $\frac{\text{lbf}}{\text{lbf}}$	Reference 3 / Assumption 6

CALCULATIONS:

$$\text{MGC}_{\text{Open}} := \frac{\left(\frac{\text{DV}}{460 \cdot \text{volt}}\right)^n \cdot \text{MR} \cdot \text{OAR} \cdot \text{Tf} \cdot \text{POE} \cdot \text{AF}}{\text{SF}} \quad (\text{Reference 6})$$

$$\text{MGC}_{\text{Open}} = 13210 \cdot \text{lbf}$$

$$F_{\text{total}} = 12554 \cdot \text{lbf}$$

$$\text{MGC}_{\text{Margin}} := \frac{\text{MGC}_{\text{Open}} - F_{\text{total}}}{F_{\text{total}}}$$

$$\text{MGC}_{\text{Margin}} = 5.2 \cdot \%$$

REVISION NO.

0

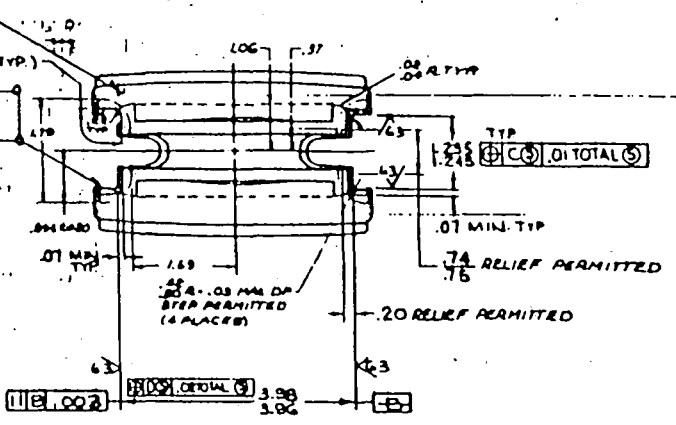
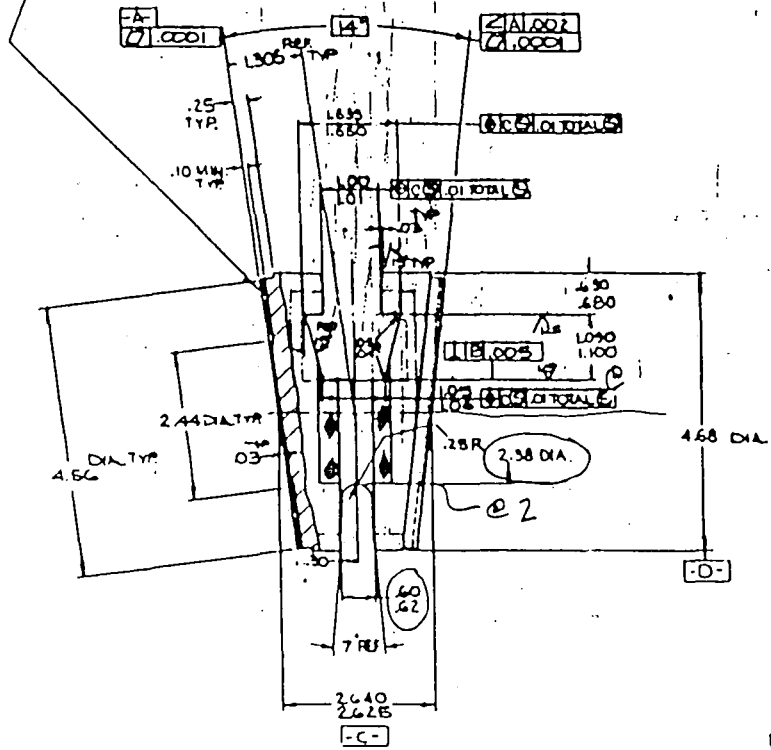
CALCULATION NO. 95-111	PROJECT NO. N/A	PAGE NO. 13		
VII. SUMMARY AND CONCLUSIONS				<i>(FINAL)</i>
<p>The results of the calculation indicate that with all the indicated conservatism inherent in the inputs, the 1(2)RY8000A&B PORV Block Valves have positive margin under the assumed pressure locking scenario. 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.</p>				
VIII. LIMITATIONS				
None.				
IX. ATTACHMENTS				
<p>(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</p>				
(B) Modulus of Elasticity - 1995 ASME Section II, Table TM-1				
REVISION NO.	0			

MARK SERIAL NO., PART NO. & HEAT NO.
 .1 TO .2 HIGH PER AE598620-1

PT PER NOTE-6 (TYP.)
 AS INDICATED

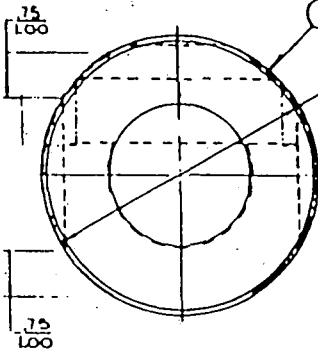
② 82142P006

PT (SEE VIEW-A)
 ③ 82142P006



2.13 RELIEF PERMITTED

① VALLOVER, UNLESS OTHERWISE SPECIFIED



4.66 DIA.
 4.65 DIA.
 RELIEF PERMITTED

-NOTES-

- LIQUID PENETRANT EXAMINE ALL FINAL MACHINED SURFACES PER BOIGS-2, CL. AA.
- LIQUID PENETRANT EXAMINE INDICATED SURFACES PER BOIGS-2, CL. OOK.
- AFTER LAPPING INDICATED SURFACES, LIQUID PENETRANT PER BOIGS -2, CL. O.A.
- HARDNESS OF STELLITE TO BE CHECKED IN ROUGH MACHINE STATE. HARDNESS TO BE 30 Rc MIN.
- 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 200 MIN. WIDTH AROUND ENTIRE FACE OF DISC IS OBTAINED.
- LIQUID PENETRANT EXAMINE INDICATED AREA PER BOIGS-2, CL. I-4X. PERFORM VISUAL EXAMINATION FOR OPEN TYPE DISCONTINUITIES WHICH DO NOT RESPOND TO PENETRANT EXAMINATION. VISUAL DISCONTINUITIES WHOSE DIMENSIONS ARE GREATER THAN 3/4 INCH IN ANY DIRECTION ARE UNACCEPTABLE.

PT PER NOTE-2

LAPPED SURFACE
 SEE NOTE 5
 PT PER NOTE 3

2.2 R TYP
 PT PER NOTE-2

-VIEW-A-
 TYP. BOTH SIDES

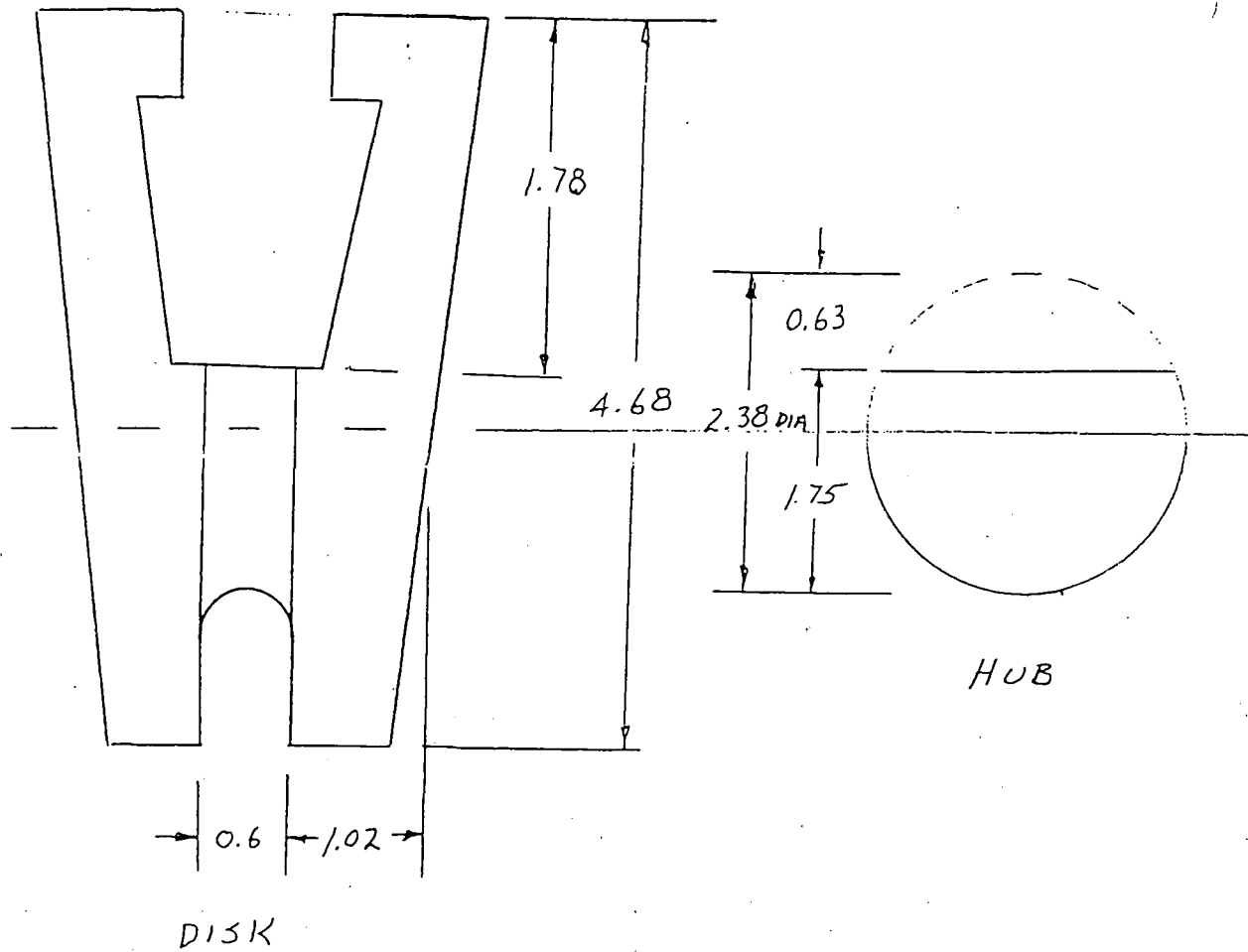
877 SIGNS
 9 ZONE D4 REF TO NOTE A DELETED & 82142P006 WAS 82142P004; NOTE C REVISED DAM/W/J52
 10 7A 11.1 150625501 WAS 934D225401; REVISED NOTE 1, ZONE D-4 ADDED; 2.2 R TYP, ZONE D-8 1.238/1.245 WAS 1.235/1.240; ZONE A-5 2.640/2.645 WAS 2.638/2.642. VIEW-A ADDED. 1.8 R TYP. 80°-1/2° TYP AND 2.44 DIA TYP. ZONE C3 1.635/1.650 WAS 1.64/1.645 8/10/66

ALSO SEE DRAWING
 TO BE USED
 MILLING CUTTER
 REVERSE SIDE

ARMED PWD	316	STELLITE 316	PERMITS	30613	300
WELDED ROD	316	STELLITE G.	316 SST	30613	300
DISC	316	316 SST	30613	300	300
LIST OF MATERIALS					
Westinghouse Electric Corporation					
DISC-314-1500,4-900					
GATE VALVE					
DISC					
D	04808	934D225			

ATTACHMENT A 111
 CAL 95-111 REV 0

Disk Dimensions



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 pg 1-7 (Ref 8)

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 = 0.60in

t = 1.02

Record of Conversation

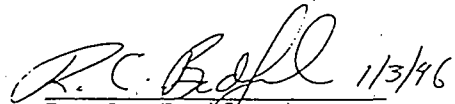
Per conversation with T. Matty of Westinghouse on 01/03/96 at 1345 (Phone 412-374-6401) the following seat ring dimensions were obtained for the listed valves:

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


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Braidwood Station

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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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-¹/₂Mo Mn-¹/₂Mo
Mn-¹/₂Mo Mn-V

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

¹/₂Ni-¹/₂Mo-Cr-V 1Ni-¹/₂Cr-¹/₂Mo
¹/₂Ni-¹/₂Mo-V ¹/₂Ni-1Mo-¹/₂Cr
¹/₂Ni-¹/₂Mo-¹/₂Cr-V ¹/₂Ni-¹/₂Cr-¹/₂Mo-V
¹/₂Cr-¹/₂Ni-Cu-Al 2Ni-1Cu
¹/₂Cr-¹/₂Ni-Cu 2¹/₂Ni
¹/₂Ni-¹/₂Cu-Mo 3¹/₂Ni

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

¹/₂Cr-¹/₂Mo
1Cr-¹/₂Mo
1¹/₂Cr-¹/₂Mo-Si
1¹/₂Cr-¹/₂Mo
2Cr-¹/₂Mo

(4) Material Group D consists of the following 2¹/₂-3Cr steels:

2¹/₂Cr-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-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

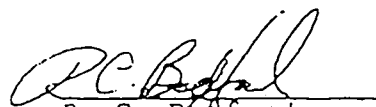
Per conversation with T. Matty of Westinghouse on 01/03/96 at 1345 (Phone 412-374-6401) the following seat ring dimensions were obtained for the listed valves:

1/2RY8000A&B 3 inch valves

Seat ring inside diameter 2.6875 in *
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Seat ring inside diameter 3.5075 in *
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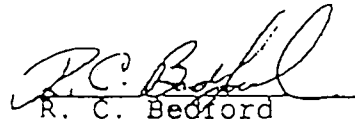
Concur

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

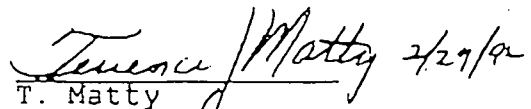
* Made up of Seat Bore plus .0625 for chamfers

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


T. Matty
Westinghouse

COMMONWEALTH EDISON COMPANY
CALCULATION TITLE PAGE

CALCULATION NO. 95-111

PAGE NO.: 1

SAFETY RELATED REGULATORY RELATED NON-SAFETY RELATED

CALCULATION TITLE:

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

STATION/UNIT: Braidwood & Byron/1&2

SYSTEM ABBREVIATION: RY

EQUIPMENT NO.: (IF APPL.)

PROJECT NO.: (IF APPL.)

1RY8000A
1RY8000B
2RY8000A
2RY8000B

N/A

REV: 0 STATUS: QA SERIAL NO. OR CHRON NO. DATE: 1 / 196

PREPARED BY: RC 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. 95-111		PAGE NO.: 2
REV: STATUS:	QA SERIAL NO. OR CHRON NO.	DATE: _____
PREPARED BY: _____	DATE: _____	
REVISION SUMMARY:		
REVIEWED BY: _____	DATE: _____	
REVIEW METHOD:	COMMENTS (C OR NC): _____	
REV: STATUS:	QA SERIAL NO. OR CHRON NO.	DATE: _____
PREPARED BY: _____	DATE: _____	
REVISION SUMMARY:		
REVIEWED BY: _____	DATE: _____	
REVIEW METHOD:	COMMENTS (C OR NC): _____	

COMMONWEALTH EDISON COMPANY
 CALCULATION TABLE OF CONTENTS

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IX. ATTACHMENTS	13	
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 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.

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PROJECT NO. N/A

PAGE NO. 5

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. In calculating MGC_{Open} , Motor Start Torque, Motor Temperature Factor, Degraded Voltage, Pullout Efficiency, and an Application Factor of 0.90 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.

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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 0.523 will be used for 1(2)RY8000A&B based on the maximum assumed value contained in the Rising Stem MOV Data Sheets for these valves.
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

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0

CALCULATION NO. 95-111

PROJECT NO. N/A

PAGE NO. 7

III. ASSUMPTIONS (con't)

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 and application factor are 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

1RY8000A, dated 01/06/96
 1RY8000B, dated 01/06/96
 2RY8000A, dated 01/26/95
 2RY8000B, dated 01/26/95

Byron Station

1RY8000A, dated 10/20/94
 1RY8000B, dated 10/20/94
 2RY8000A, dated 06/23/94
 2RY8000B, dated 06/23/94

3. Rising Stem MOV Data Sheets for :

Braidwood Station

1RY8000A, dated 11/28/94
 1RY8000B, dated 11/28/94
 2RY8000A, dated 11/28/94
 2RY8000B, dated 11/28/94

Byron Station

1RY8000A, dated 04/06/94
 1RY8000B, dated 04/06/94
 2RY8000A, dated 04/06/94
 2RY8000B, dated 04/06/94

REVISION NO.

0

CALCULATION NO. 95-111

PROJECT NO. N/A

PAGE NO. 8

V. REFERENCES (con't)

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 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

VI. CALCULATIONS

MathCad 5.0+ calculations of the following for each of the three groups 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.

REVISION NO.

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VI. CALCULATIONS

INPUTS:

Bonnet Pressure	$P_{\text{bonnet}} := 2235 \cdot \text{psi}$	Assumption 5
Upstream Pressure	$P_{\text{up}} := 350 \cdot \text{psi}$	Assumption 5
Downstream Pressure	$P_{\text{down}} := 0 \cdot \text{psi}$	Assumption 5
Disk Thickness	$t := 1.02 \cdot \text{in}$	Attachment A
Seat Radius	$a := 1.60937 \cdot \text{in}$	Attachment A
Effective Hub Radius	$b := 1.056 \cdot \text{in}$	Attachment A
Hub Length	$L := 0.60 \cdot \text{in}$	Attachment A
Seat Angle	$\theta := 7 \cdot \text{deg}$	Reference 3
Poisson's Ratio (disk)	$\nu := .3$	Typical of Stainless Steel
Mod. of Elast. (disk)	$E := 27.0 \cdot 10^6 \cdot \text{psi}$	Attachment B (300 F)
Static Pullout Force	$F_{\text{po}} := 5933 \cdot \text{lbf}$	Reference 2 / Assumption 4
Open Valve Factor	$VF := .523$	Reference 3
Stem Diameter	$D_{\text{stem}} := 1.25 \cdot \text{in}$	Reference 3

PRESSURE FORCE CALCULATIONS

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

$$\mu := VF \cdot \frac{\cos(\theta)}{1 - VF \cdot \sin(\theta)} \quad \mu = 0.554$$

Average DP across disks:

$$DP_{\text{avg}} := P_{\text{bonnet}} - \frac{P_{\text{up}} + P_{\text{down}}}{2} \quad DP_{\text{avg}} = 2060 \cdot \text{psi}$$

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

$$D := \frac{E \cdot (t)^3}{12 \cdot (1 - \nu^2)} \quad D = 2.624 \cdot 10^6 \cdot \text{lbf} \cdot \text{in}$$

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

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VI. CALCULATIONS

Geometry Factors: (Reference 1, Table 24)

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

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

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

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

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

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

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

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

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

$$M_{rb} := \frac{-DP_{avg} \cdot a^2}{C_8} \left[\frac{C_9}{2 \cdot a \cdot b} \cdot (a^2 - b^2) - L_{17} \right] \quad M_{rb} = -390.43 \cdot \text{lb} \cdot \text{f}$$

$$Q_b := \frac{DP_{avg} \cdot (a^2 - b^2)}{2 \cdot b} \quad Q_b = 1438.621 \cdot \frac{\text{lb} \cdot \text{f}}{\text{in}}$$

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

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

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VI. CALCULATIONS

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

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

$$K_{sa} = -0.08198$$

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

$$y_{sq} = -4.1293 \cdot 10^{-5} \cdot \text{in}$$

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

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

$$P_{force} = 9545.336 \cdot \text{lb}f$$

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

$$y_{stretch} = 3.0274 \cdot 10^{-5} \cdot \text{in}$$

Total Deflection due to pressure forces:

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

$$y_q = -8.1591 \cdot 10^{-5} \cdot \text{in}$$

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

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

(per lbf/in)

$$y_{sw} = -7.6824 \cdot 10^{-8} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_{bw} = -2.5057 \cdot 10^{-8} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_{compr} = 3.2071 \cdot 10^{-8} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

(per lbf/in)

$$y_w = -1.3395 \cdot 10^{-7} \cdot \frac{\text{in}}{\left(\frac{\text{lbf}}{\text{in}}\right)}$$

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

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

$$F_s = 6159.3 \cdot \text{lb}f$$

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VI. CALCULATIONS

UNSEATING FORCES

 F_{packing} is included in measured static pullout Force

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

$$F_{\text{piston}} = 2742.76 \cdot \text{lbf}$$

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

$$F_{\text{vert}} = 4085.58 \cdot \text{lbf}$$

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

$$F_{\text{preslock}} = 5277.76 \cdot \text{lbf}$$

$$F_{\text{po}} = 5933 \cdot \text{lbf}$$

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

$$F_{\text{total}} = 12554 \cdot \text{lbf}$$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	MR := 15.0-ft·lbf	Reference 3
Temperature Factor:	Tf := 0.834	Reference 3, Assumption 8
Degraded Voltage:	DV := 408-volt	Reference 3 / Assumption 4
Under Voltage Factor:	n := 2.0	Reference 6
Overall Gear Ratio	OAR := 52.2	Reference 3
Pullout Efficiency	POE := 0.40	Reference 3
Application Factor	AF := 0.90	Reference 3, Assumption 8
Stem Factor @ $\mu=0.20$	SF := 0.0140-ft $\frac{\text{lbf}}{\text{lbf}}$	Reference 3 / Assumption 6

CALCULATIONS:

$$\text{MGC}_{\text{Open}} := \frac{\left(\frac{\text{DV}}{460 \cdot \text{volt}}\right)^n \cdot \text{MR} \cdot \text{OAR} \cdot \text{Tf} \cdot \text{POE} \cdot \text{AF}}{\text{SF}} \quad (\text{Reference 6})$$

$$\text{MGC}_{\text{Open}} = 13210 \cdot \text{lbf} \quad F_{\text{total}} = 12554 \cdot \text{lbf}$$

$$\text{MGC}_{\text{Margin}} := \frac{\text{MGC}_{\text{Open}} - F_{\text{total}}}{F_{\text{total}}} \quad \text{MGC}_{\text{Margin}} = 5.2 \cdot \%$$

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CALCULATION NO. 95-111

PROJECT NO. N/A

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VII. SUMMARY AND CONCLUSIONS

(FINAL)

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. 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

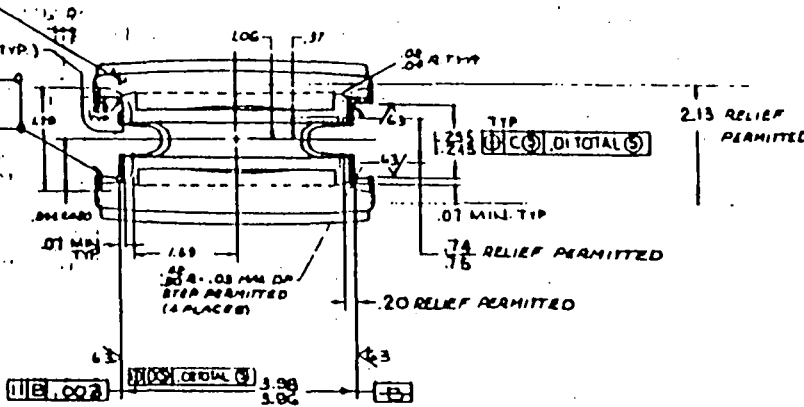
REVISION NO.

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MARK SERIAL NO., PART NO. & HEAT NO.
 .1 TO .2 HIGH PER AE 5986G20-1

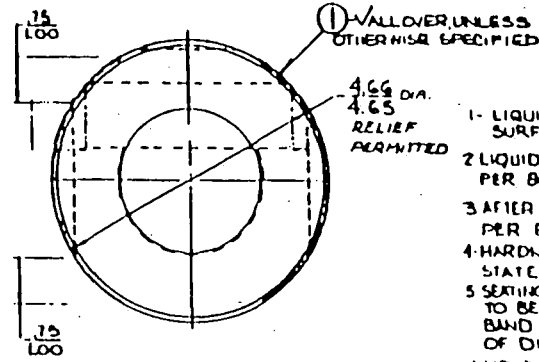
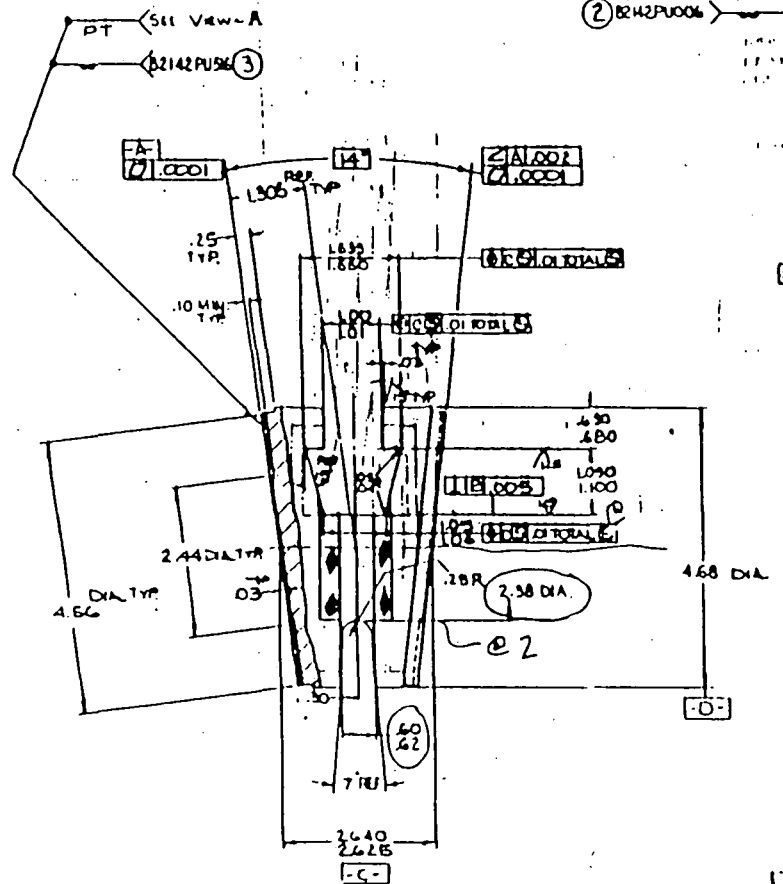
PT PER NOTE-6 (TYP)
 AS INDICATED

(2) B2142P006



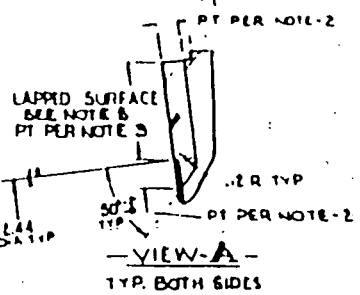
REVISIONS

9	ZONE D4 REF TO NOTE A DELETED & B2142P006 WAS B2142P004; NOTE C REVISED	DAN/JUL/52
10	7A II.1 INSPECTION WAS REDESIGNATION, REVISED NOTE 1, LOME D-4 ADDED. II.1 R.T.T.P. ZONE B D-8 II.187/1.148 WAS LESS/1.140, ZONE A5 1.640/1.658 WAS 1.638/1.648, VIEW A AREA II.1 R TYP. 20" II.1 TYP AND 2.44 DIA. TYP. LOME C3 1.633/1.640 WAS 1.641/1.645 B/W/D-2.1	JL/JUL/52



-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. O.A.
- 4- HARDNESS OF STELLITE TO BE CHECKED IN ROUGH MACHINE STATE. HARDNESS TO BE 30 R.C. 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. I-4X. PERFORM VISUAL EXAMINATION FOR OPEN TYPE DISCONTINUITIES WHICH DO NOT RESPOND TO PENETRANT EXAMINATION. VISUAL DISCONTINUITIES WHOSE DIMENSIONS ARE GREATER THAN $\frac{3}{16}$ INCH IN ANY DIRECTION ARE UNACCEPTABLE.



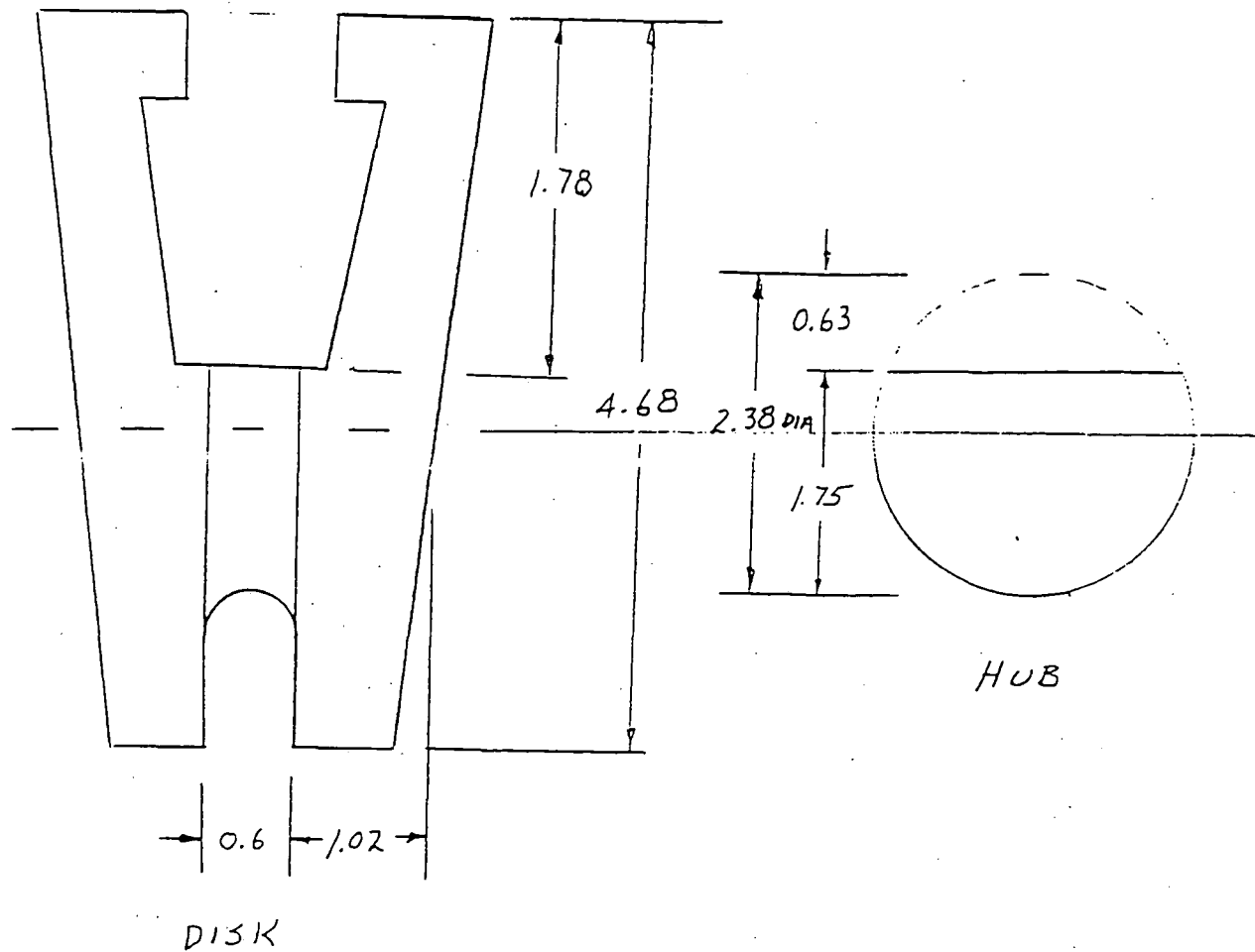
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APPROVED	WAS	3/18/52	308111	HW	3
WAS	WAS	3/18/52	308111	HW	3
WAS	WAS	3/18/52	308111	HW	3
WAS	WAS	3/18/52	308111	HW	3
WAS	WAS	3/18/52	308111	HW	3

2142P006		WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS
2142P004		WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS
B0165		WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS	WAS

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

2142 95-111 REV 0

Disk Dimensions

Effective Radius of Hub Section

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

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

$$h/D = 0.63\text{in}/2.38\text{in} = .264 \text{ interpolation from table pg 1-7 (Ref 8)}$$

$$\text{Area/Circle} = 0.21108$$

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

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

Effective Area Diameter

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

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

$$L = 0.60\text{in}$$

$$t = 1.02$$

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


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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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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 x 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-1/2Mo Mn-V

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

1/2Ni-1/2Mo-Cr-V 1Ni-1/2Cr-1/2Mo
1/2Ni-1/2Mo-V 1/2Ni-1Mo-1/2Cr
1/2Ni-1/2Mo-1/3Cr-V 1/2Ni-1/2Cr-1/2Mo-V
1/2Cr-1/2Ni-Cu-Al 2Ni-1Cu
1/2Cr-1/2Ni-Cu 2 1/2Ni
1/2Ni-1/2Cu-Mo 3 1/2Ni

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

1/2Cr-1/2Mo
1Cr-1/2Mo
1 1/2Cr-1/2Mo-Si
1 1/2Cr-1/2Mo
2Cr-1/2Mo

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

2 1/2Cr-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:

15Cr-8Ni 18Cr-10Ni-Cb
15Cr-8Ni-N 18Cr-13Ni-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

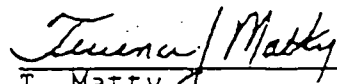
Seat ring inside diameter 2.6875 in *
 Seat ring outside diameter 3.75 in
 Mean seat ring diameter 3.21875 in

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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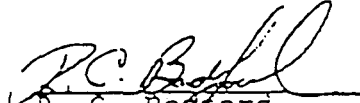
Concur

 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.


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Concur


T. Matty
Westinghouse