COMMONWEALTH EDISON COMPANY
CALCULATION TITLE PAGE

CALCULATION NO. QDC-1000-M-0077 PROJECT NO.: N/A PAGE NO.: 1

■ SAFETY RELATED ☐ REGULATORY RELATED ☐ NON-SAFETY RELATED

CALCULATION TITLE:
Pressure Locking Calculation for RHR System Valve MOV 1-1001-29B

STATION/UNIT: Quad Cities/1 SYSTEM ABBREVIATION: LPCI

EQUIPMENT NO.: (IF APPL.)
MOV 1-1001-29B

REV: *0* STATUS: APPROVED QA SERIAL NO. OR CHRON NO. N/A DATE: ______

PREPARED BY: *K. Higgins* [signature] DATE: 11/9/95

REVISION SUMMARY: *Original Issue*
DO ANY ASSUMPTIONS IN THIS CALCULATION REQUIRE LATER VERIFICATION YES ☐ NO ■

REVIEWED BY: *J. Kelly* [signature] 11-9-95

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

APPROVED BY: [signature] W.C. PORTER 11/9/95

COMMONWEALTH EDISON COMPANY
CALCULATION REVISION PAGE

CALCULATION NO. QDC-1000-M-0077 PAGE NO.: 2

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

PREPARED BY:________________________________ DATE:____________

REVISION SUMMARY:

REVIEWED BY:____________________________ DATE:____________

REVIEW METHOD: COMMENTS (C OR NC):______

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

PREPARED BY:________________________________ DATE:____________

REVISION SUMMARY:

REVIEWED BY:____________________________ DATE:____________

REVIEW METHOD: COMMENTS (C OR NC):______

COMMONWEALTH EDISON COMPANY

CALCULATION TABLE OF CONTENTS

	PROJECT NO. N/A	
CALCULATION NO. QDC-1000-M-0077	REV. NO. 0	PAGE NO. 3
DESCRIPTION	PAGE NO.	SUB-PAGE NO.

I. PURPOSE/OBJECTIVE

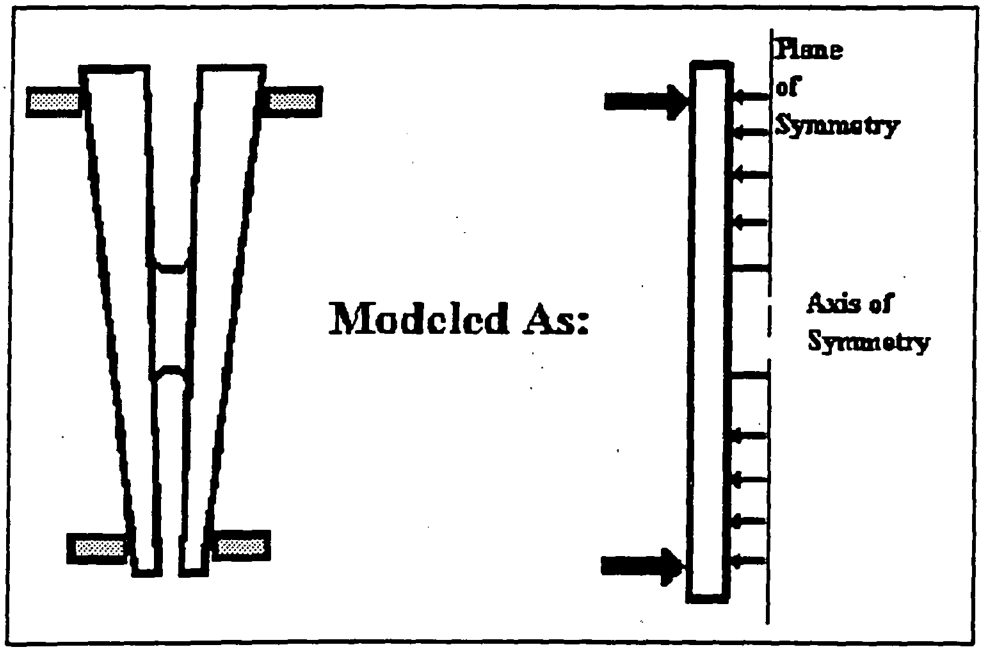
Valve MOV 1-1001-29B, which is installed in the RHR System at Quad Cities, has been determined to be susceptible to the pressure locking phenomena. The purpose of this calculation is to determine that the valve will open when called upon by design during a pressure locking event by calculating the force necessary to overcome pressure locking and the available Motor/Gearing Capability (MGC) of the MOV.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

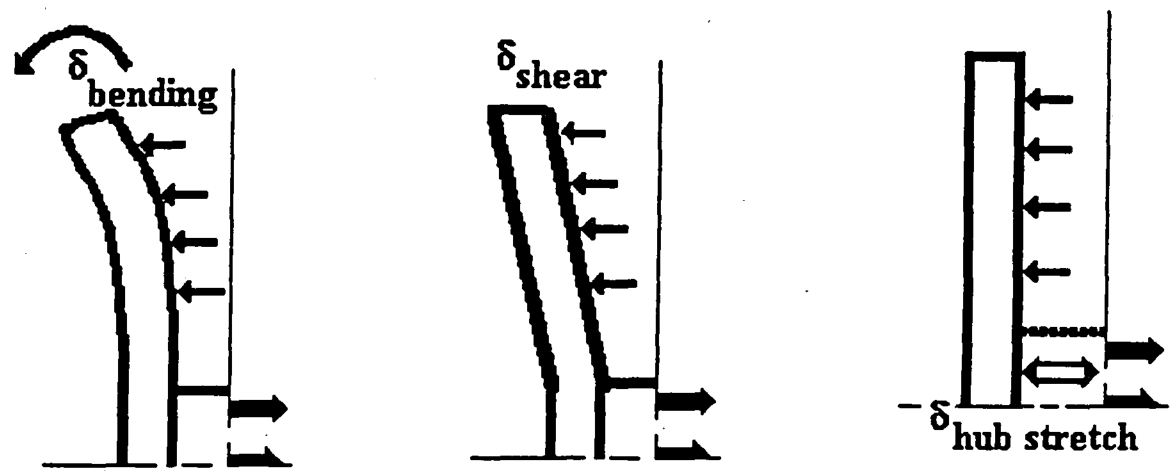
The methodology for calculating the thrust required to open the MOVs under the pressure locking scenario is based on the Reference 1 (Roark's) engineering handbook. This methodology has been previously applied by ComEd in the References 2 and 10 calculations. The methodology determines the total force required to open the valve under a pressure locking scenario by solving for the four components to this required force. The four components of the force are the Pressure Locking Component, the static unseating component, the piston effect component, and the "reverse piston effect" component. These components are determined using the following steps.

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

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



The pressure force is assumed to act uniformly upon the inner surface of the disk between the hub diameter and the outer disk diameter. The outer edge of the disk is assumed to be unimpeded and allowed to deflect away from the pressure force. In addition, the disk hub is allowed to stretch. The total displacement at the outer edge of the valve disk 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 disk. This force acts to deflect the outer diameter of the valve disk inward and to compress the disk hub. The pressure force is reacted to by an increase in this contact force between the valve disk 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 disk which results in a deflection which is equal and opposite to the deflection due to the pressure force. Because the disk thickness varies, the average thickness is used for purposes of this calculation.

The coefficient of friction between the seat and disk is determined based on the open valve factor from a DP test. The stem force required to overcome the contact load between the seat and disk which opposes the pressure force is equal to:

(seat load) x [(seat mu) cos(seat angle) - sin(seat angle)] x 2 (for two disk faces).

Static Unseating Force

The static unseating force represent the open packing load and pullout force due to wedging of the valve disk 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.

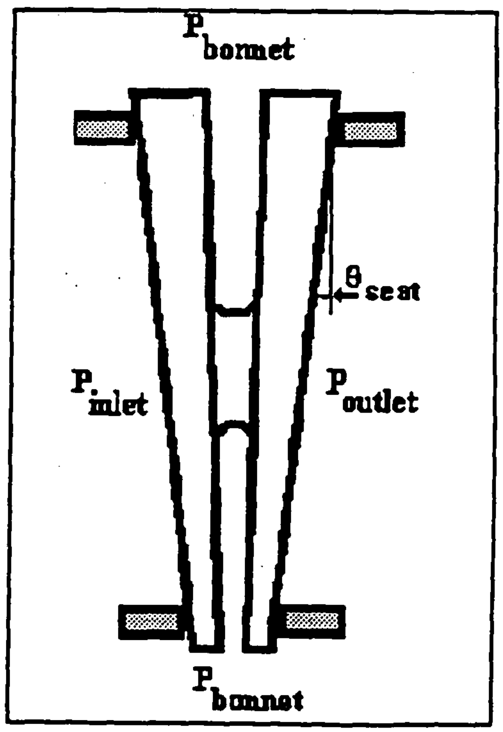
Piston Effect

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.

$$F_{pistoneffect} = \frac{\pi}{4} \times D^2_{stem} \times \left(P_{bonnet} - P_{atm}\right)$$

"Reverse Piston Effect"(F_{vert})

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

Determination of Motor Gearing Capability

Next the motor gearing capability available to overcome static unseating forces is determined using the statically measured stem factor, the pullout efficiency, the temperature factor, and the ComEd motor test data (for breakdown torque and voltage factor), Reference 5.

$$MGC_{open} = \frac{MR_{breakdown} \times TempFactor \times OAR \times Eff_{pullout} \times \left(\frac{Voltage_{available}}{Voltage_{rated}}\right)^{Exponent}}{Stem\ Factor}$$

Determination of Open Valve Factor

The open valve factor is calculated by based on the open DP load. This load is determined by using the equation below: The O10 thrust is measured in the region of the trace during which the valve disk is sliding on the valve seat (prior to flow initiation). This thrust is based on the O4 zero since the valve is effectively closed at O10. The open running thrust is measured at the end of the open stroke and is referenced to the C3 zero since the valve is nearly fully open at the point at which the open running load is measured. The Line Pressure adjustment term in the equation accounts for the fact that the piston effect decreases during the opening valve stroke.

$$VF_{open} = \frac{O10_{thrust} - Running_{thrust} + \frac{\pi}{4}D^2_{stem}(O10_{linepressure} - Running_{linepressure})}{DP \times \frac{\pi}{4}D^2_{seat}}$$

Enhanced Capability Evaluation

The enhanced capability evaluation uses the measured overall MOV efficiency to predict the required motor torque during the pressure lock condition. This overall MOV efficiency is determined by dividing the actual motor torque during the latest static test by the measured pullout thrust. This ratio (MOV efficiency) is then multiplied by the estimated pressure lock pullout force to determine the required motor torque during pressure lock pullout. The available motor torque is set equal to the motor breakdown torque from the ComEd motor test data, Reference 5.

Acceptance Criteria

The margin between one time structural limit and required thrust is calculated. A margin of 15% or greater is required. This margin accounts for equipment inaccuracy and degradation.

The margin between MGC and required thrust is calculated. A margin of 20% or greater is required. This margin accounts for equipment inaccuracy and degradation.

If the enhanced capability evaluation is applied, a margin of 40% or greater is required. This margin accounts for voltage, current, and thrust measurement inaccuracies from the static test and a possible reduction in overall MOV efficiency from the static condition to the estimated pullout condition.

III. ASSUMPTIONS

1. The valve disk is assumed to act as two ideal disks connected by a hub. The equations in reference 1 are assumed to conservatively model the actual load due to pressure forces. This assumption is considered conservative since inspection of the disk drawings show large fillets between the disk hub and seats which should make the valve disk stiffer than assumed in the reference 1 equations.

2. The coefficient of friction between the valve seat and disk is assumed to be the same under pressure locking conditions as it is under DP conditions. This assumption (in combination with assumption 1) is considered to be justified based on bench marking of the calculation against ComEd and EPRI pressure locking test data for similar flex-wedge gate valves.

3. The upstream, downstream, and bonnet pressure values are based on a scenario in which the valve bonnet is pressurized to reactor pressure (1020 psia) by leakage past adjacent check valves. A LOCA occurs which causes the reactor pressure to drop off to 325 psig. The LPCI pump comes up to speed and the subject valve receives a signal to open simultaneously. Total dynamic head for the RHR pump in LPCI mode is 307 psig. The pressure values are based on a review of the UFSAR for Quad Cities. See reference 6.

IV. DESIGN INPUTS

1. Valve Disk Geometry information is based on the Reference 4, Faxes from the Crane-Aloyco Valve Company. (Attachment A)

2. Motor Data is taken from the Reference 5 report and RSMDS, Reference 7.

3. Static and DP diagnostic test data is taken from the most recent diagnostic tests.

V. REFERENCES

1. Sixth Edition of Roark's Formulas for Stress and Strain

2. MPR Calculations 101-013-1, "Effect of Bonnet Pressure on Disc to Seat Contact Load", dated 3/23/95; and 101-013-4, "Estimate of Valve Unseating Force as Function of Bonnet Pressure", dated 3/23/95

3. NMAC Report NP-6660-D, " Application Guide For Motor Operated Valves"

4. Crane Telecopies from Dave Dwyer and Bruce Harry to Brian Bunte (ComEd) dated 5/3/95 and 6/16/95, Attachment A.

5. ComEd White Paper 125, MOV-WP-125, Rev. 2, 10/4/95

6. UFSAR Section 6.3.2.2.3.4, Tbl 6.3-5, Tbl 4.1-3 and Fig 6.3-8

7. ComEd Rising Stem MOV Data Sheet, 1-1001-29B, 02/06/95, 13:11

8. Thrust values are taken from static VOTES Test 7, performed 6/3/94 and DP VOTES Test 5, performed 11/12/93.

9. EMS Calculation CE-DR-030, "Pressure Locking Analysis of Dresden Motor Operated Valves", dated 6/13/95.

10. ComEd Calculation NED-M-MSD-182, "Verification of Operability for Dresden and Quad Cities Injection Valves Susceptible to Pressure Locking", dated June 22, 1995.

11. Thrust and Torque Calculation, OTC-240, Rev. 4, Attachment A.

VI. CALCULATIONS

The following is provided for MOV 1-1001-29B.

MathCad calculation of:

1) the pressure locking unseating force,
2) the available motor gearing capability to unseat while pressure locked
3) the enhanced capability

VI QCNPS Valve 1-1001-29B

INPUTS:

Bonnet Pressure	$P_{bonnet} := 1005 \cdot psi$	Reference 6
Upstream Pressure	$P_{up} := 307 \cdot psi$	Ref. 6 & Assum. 3
Downstream Pressure	$P_{down} := 325 \cdot psi$	Ref. 6 & Assum. 3
Disk Thickness, Avg	$t := 2.75 \cdot in$	Reference 4
Seat Radius	$a := 6.385 \cdot in$	Reference 7
Hub Radius	$b := 2.125 \cdot in$	Reference 4
Hub Length	$L := 2.4375 \cdot in$	Reference 4
Seat Angle	$theta := 5 \cdot deg$	Reference 7
Poisson's Ratio (disk)	$v := 0.3$	Reference 1 & 11, Stain. Steel
Mod. of Elast. (disk)	$E := 27.6 \cdot 10^6 \cdot psi$	Reference 1 & 11, Stain. Steel
Static Pullout Force (Test 7)	$F_{po} := 15000 \cdot lbf$	Reference 8
O10 Thrust (DP test 5)	$O10 := 9429 \cdot lbf$	Reference 8
Open Run Thrust (DP)	$Run := 450 \cdot lbf$	Reference 8
DP	$DPtest := 256 \cdot psi$	Reference 8
LP (valve closed)	$LPclose := 256 \cdot psi$	Reference 8
LP (valve open)	$LPopen := 0 \cdot psi$	Reference 8
Stem Diameter	$D_{stem} := 3.0 \cdot in$	Reference 7

VALVE FACTOR CALCULATION

Valve Factor:

$$VF := \frac{(O10 - Run) + \frac{\pi}{4} \cdot D_{stem}^{2} \cdot (LPclose - LPopen)}{\pi \cdot (a)^2 \cdot DPtest} \qquad VF = 0.329$$

Coefficient of friction between disk and seat: (Reference 3)

$$mu := VF \cdot \frac{\cos(theta)}{1 - VF \cdot \sin(theta)} \qquad mu = 0.337$$

PRESSURE FORCE CALCULATIONS

Average DP across disks:

$$DPavg := P_{bonnet} - \frac{P_{up} + P_{down}}{2} \qquad DPavg = 689 \cdot psi$$

Disk Stiffness Constants (Reference 1, Table 24)

$$D := \frac{E \cdot (t)^3}{12 \cdot \left(1 - \nu^2\right)} \qquad D = 5.256 \cdot 10^7 \cdot \mathrm{lbf \cdot in}$$

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

Geometry Factors: (Reference 1, Table 24)

$$C_2 := \frac{1}{4} \cdot \left[1 - \left(\frac{b}{a}\right)^2 \cdot \left(1 + 2 \cdot \ln\left(\frac{a}{b}\right)\right)\right] \qquad C_2 = 0.161$$

$$C_3 := \frac{b}{4 \cdot a} \cdot \left[\left[\left(\frac{b}{a}\right)^2 + 1\right] \cdot \ln\left(\frac{a}{b}\right) + \left(\frac{b}{a}\right)^2 - 1\right] \qquad C_3 = 0.028$$

$$C_8 := \frac{1}{2} \cdot \left[1 + \nu + (1 - \nu) \cdot \left(\frac{b}{a}\right)^2\right] \qquad C_8 = 0.689$$

$$C_9 := \frac{b}{a} \cdot \left[\frac{1 + \nu}{2} \cdot \ln\left(\frac{a}{b}\right) + \frac{1 - \nu}{4} \cdot \left[1 - \left(\frac{b}{a}\right)^2\right]\right] \qquad C_9 = 0.29$$

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

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

$$L_{11} := \frac{1}{64} \cdot \left[1 + 4 \cdot \left(\frac{b}{a}\right)^2 - 5 \cdot \left(\frac{b}{a}\right)^4 - 4 \cdot \left(\frac{b}{a}\right)^2 \cdot \left[2 + \left(\frac{b}{a}\right)^2\right] \cdot \ln\left(\frac{a}{b}\right)\right] \qquad L_{11} = 0.006$$

$$L_{17} := \frac{1}{4} \cdot \left[1 - \frac{1 - \nu}{4} \cdot \left[1 - \left(\frac{b}{a}\right)^4\right] - \left(\frac{b}{a}\right)^2 \cdot \left[1 + (1 + \nu) \cdot \ln\left(\frac{a}{b}\right)\right]\right] \qquad L_{17} = 0.139$$

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

$$M_{rb} := \frac{-DPavg \cdot a^2}{C_8} \cdot \left[\frac{C_9}{2 \cdot a \cdot b} \cdot \left(a^2 - b^2\right) - L_{17}\right] \qquad M_{rb} = -1.01 \cdot 10^4 \cdot \mathrm{lbf}$$

$$Q_b := \frac{DPavg}{2 \cdot b} \cdot \left(a^2 - b^2\right) \qquad Q_b = 5.877 \cdot 10^3 \cdot \frac{\mathrm{lbf}}{\mathrm{in}}$$

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

$$y_{bq} := M_{rb}\cdot\frac{a^2}{D}\cdot C_2 + Q_b\cdot\frac{a^3}{D}\cdot C_3 - \frac{DPavg\cdot a^4}{D}\cdot L_{11} \qquad y_{bq} = -5.783\cdot 10^{-4}\cdot in$$

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

$$K_{sa} := -0.3\cdot\left[2\cdot\ln\left(\frac{a}{b}\right) - 1 + \left(\frac{b}{a}\right)^2\right] \qquad K_{sa} = -0.393$$

$$y_{sq} := \frac{K_{sa}\cdot DPavg\cdot a^2}{t\cdot G} \qquad y_{sq} = -3.785\cdot 10^{-4}\cdot in$$

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

$$P_{force} := 3.1416\cdot\left(a^2 - b^2\right)\cdot DPavg \qquad P_{force} = 7.847\cdot 10^{4}\cdot lbf$$

$$y_{stretch} := \frac{P_{force}}{3.1416\cdot b^2}\cdot\frac{L}{(2\cdot E)} \qquad y_{stretch} = 2.443\cdot 10^{-4}\cdot in$$

Total Deflection due to pressure forces:

$$y_q := y_{bq} + y_{sq} - y_{stretch} \qquad y_q = -0.001\cdot in$$

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

$$y_{sw} := -\left[\frac{1.2\cdot\left(\frac{a}{a}\right)\cdot\ln\left(\frac{a}{b}\right)\cdot a}{t\cdot G}\right] \qquad y_{sw} = -2.888\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{bw} := -\left(\frac{a^3}{D}\right)\cdot\left[\left(\frac{C_2}{C_8}\right)\cdot\left[\left(\frac{a\cdot C_9}{b}\right) - L_9\right] - \left[\left(\frac{a}{b}\right)\cdot C_3\right] + L_3\right] \qquad y_{bw} = -5.983\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{compr} := \frac{2\cdot a\cdot\pi}{3.1416\cdot b^2}\cdot\frac{L}{(2\cdot E)} \qquad y_{compr} = 1.249\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_w := y_{bw} + y_{sw} - y_{compr} \qquad y_w = -1.012\cdot 10^{-6}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

in

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

$F_s := 2 \cdot \pi \cdot a \cdot \frac{y_q}{y_w}$ $\qquad F_s = 4.762 \cdot 10^4 \cdot lbf$

UNSEATING FORCES Reference 2

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

$F_{piston} := \frac{\pi}{4} \cdot D_{stem}{}^2 \cdot P_{bonnet}$ $\qquad F_{piston} = 7.104 \cdot 10^3 \cdot lbf$

$F_{vert} := \pi \cdot a^2 \cdot \sin(theta) \cdot \left(2 \cdot P_{bonnet} - P_{up} - P_{down}\right)$ $\qquad F_{vert} = 1.538 \cdot 10^4 \cdot lbf$

$F_{preslock} := 2 \cdot F_s \cdot (mu \cdot \cos(theta) - \sin(theta))$ $\qquad F_{preslock} = 2.372 \cdot 10^4 \cdot lbf$

$F_{total} := -F_{piston} + F_{vert} + F_{preslock} + F_{po}$ $\qquad F_{po} = 1.5 \cdot 10^4 \cdot lbf$

$F_{total} = 4.699 \cdot 10^4 \cdot lbf$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	$MR := 145.95 \cdot ft \cdot lbf$	Reference 5
Temperature Factor:	$Tf := .935$	Reference 7
Degraded Voltage:	$DV := 397 \cdot volt$	Reference 7
Under Voltage Factor:	$n := 2.119$	Reference 5
Stem Factor:	$SF := 0.0254 \cdot ft$	Reference 7
Overall Ratio:	$OAR := 46.13$	Reference 7
Pullout Efficiency:	$Eff_{po} := 0.65$	Reference 7

MOTOR / GEARING CAPABILITY CALCULATIONS:

$$MGC := MR \cdot Tf \cdot OAR \cdot Eff_{po} \cdot \frac{\left(\frac{DV}{460 \cdot volt}\right)^n}{SF} \qquad MGC = 1.179 \cdot 10^5 \cdot lbf$$

OPEN STRUCTURAL LIMIT: StructuralLimit := 104718·lbf **Reference 7**

OPEN LIMIT: Limit := (min((StructuralLimit MGC)))

MARGIN: $Margin := \frac{Limit - F_{total}}{F_{total}}$ Margin = 1.228

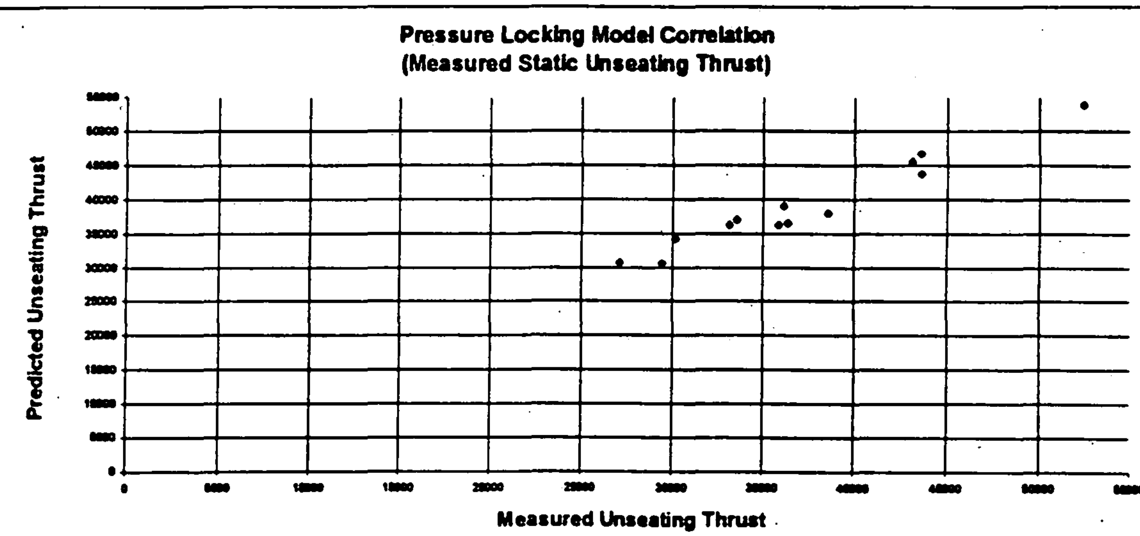
VII. COMPARISON OF MODEL OF OTHER SOURCES OF INFORMATION

Finite Element Calculation to Determine Seat Contact Force due to Bonnet Pressure

Results from the Reference 9 calculation demonstrate that the contact force between the seat and disk which is calculated using the MathCad model above is conservative and reasonably accurate (within 5%) for the 16" valve size modeled in this calculation. Once this Finite Element Analysis calculation is finalized, this calculation will be updated to provide the actual values.

Comparison to Actual Test Data

The reference 10 calculation demonstrates that the MathCad model calculates the pressure locking unseating thrust for the 6" flex-wedge Velan gate valve with good accuracy. In addition, pressure locking testing performed on a 10" Crane 900# Class flex-wedge gate valve at Quad Cities on July 21, 1995 indicates that the model accurately and conservatively predicts that pressure locking unseating force. A graph showing the initial results of this testing is provided below. Further testing is scheduled for later this year. The results of the entire test sequence will be formally documented in a ComEd report after all testing is completed.



VIII. SUMMARY AND CONCLUSIONS

This calculation has determined that the force required to unseat MOV 1-1001-29B (F_{total}) is 38,800 lbf and that the Motor/Gearing Capability (MGC) is 117,900 lbf. The Open Structural Limit for MOV 1-1001-29B, taken from the RSMDS, is the Weak Link value of 104,718 lbf (calculated at 575 °F design temp) . The margin was determined by finding the difference between the limiting open force, in this case the weak link value, and the unseating force, then dividing the resultant value by the total unseating force to produced a margin of 122.8%.

The calculated margin of 122.8% is greater than the 15% minimum margin requirement, therefore, a pressure locking event will not prevent this valve from performing its design function.

IX. ATTACHMENTS

Telecopy from Dave Dwyer (Crane-Aloyco Valves) to Brian Bunte (ComEd) dated 5/3/95, Page A-1.

Telecopy from Bruce Harry (Crane-Aloyco Valves) to Brian Bunte (ComEd) dated 6/16/95, Page A-2.

CRANE VALVES Nuclear Operations

FAX TRANSMITTAL

TO: BRIAN BUNTE PHONE: ______ FAX: [illegible]

FROM: David H. Dwyer, Project Engineer PHONE: (815) 740 - 7511 FAX: (815) 727 - 4246

SUBJECT: DISC DIMENSIONS DATE: 5/3/95

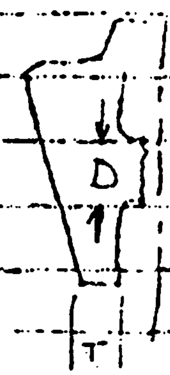
REFFRENCE: ______ TOTAL PAGES: 1

MESSAGE:

BRIAN - THE FOLLOWING ARE APPROXIMATE NOMINAL DIMENSIONS FOR THE 10" + 16" VALVES DISCS WE DISCUSSED THIS MORNING. ALL DIMENSIONS IN INCHES

SIZE	HUB DIA (D)	PLATE THICKNESS (T) MIN	PLATE THICKNESS (T) MAX
10	2.5	1 5/16	2 5/16
16	4.25	2 3/16	3 5/16

PLATE THICKNESS IS GREATEST AT TOP [illegible]



REGARDS

[illegible signature]

Att. I.D. A Sht 1 of 2
Calc. No. QDC-[illegible]-M Rev. 0

0077

CRANE®

CRANE VALVES NUCLEAR OPERATIONS
TELECOPIER TRANSMITTAL

104 North Chicago Street, Joliet, IL 60431

DATE: 6-16-95

TO: BRIAN BUNTE FROM: BRUCE HARRY

TITLE: SR. ENG. TITLE: DEV. ENG.

COMPANY: CECO

PHONE: 708-663-3824 PHONE: 815-740-7570

FAX: 708-663-7181 FAX: (815) 727-4246

TOTAL PAGES: 1

SUBJECT: HUB DIM. FOR 10" AND 16" FIG 783 FLEXIBLE WEDGES.

THE LENGTHS OF THE FLEXIBLE WEDGE INTERNAL HUBS ARE 1 5/8" AND 2 7/16" FOR THE 10" AND 16" SIZES, RESPECTIVELY.

(FINAL)

Att. I.D. A Sht 2 of 2
Calc. No. QDC-1000-M Rev. 0

0077