COMMONWEALTH EDISON COMPANY
CALCULATION TITLE PAGE

CALCULATION NO. QDC-1400-M-0080 PROJECT NO.: N/A PAGE NO.: 1

■ SAFETY RELATED ☐ REGULATORY RELATED ☐ NON-SAFETY RELATED

CALCULATION TITLE:
Pressure Locking Calculation for Core Spray System Valve MOV 1-1402-25A

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

EQUIPMENT NO.: (IF APPL.)
MOV 1-1402-25A

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 ☐ SEE ASSUM. 4

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

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

APPROVED BY: [signature] W O Porter 11/9/95

COMMONWEALTH EDISON COMPANY
CALCULATION REVISION PAGE

CALCULATION NO. QDC-1400-M-0080 PAGE NO.: 2

REV: QA SERIAL NO. OR CHRON NO. DATE:______________
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REVIEW METHOD: COMMENTS (C OR NC):______

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

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REVIEWED BY:________________________________ DATE:______________

REVIEW METHOD: COMMENTS (C OR NC):______

COMMONWEALTH EDISON COMPANY

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I. PURPOSE/OBJECTIVE

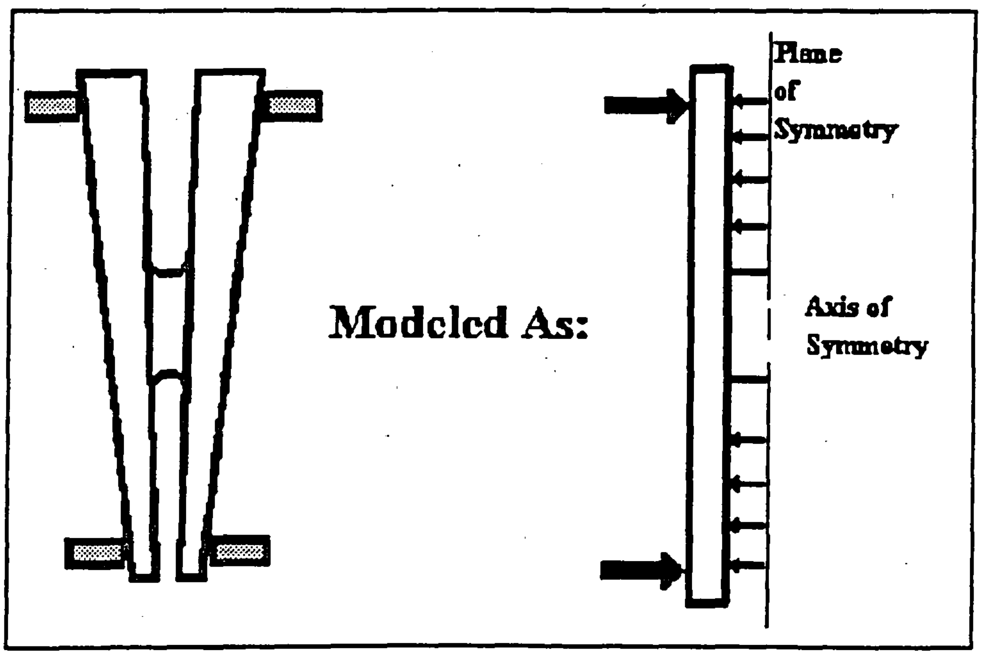
Valve MOV 1-1402-25A, which is installed in the Core Spray System at Quad Cities, has been determined to be susceptible to the pressure locking phenomena. The purpose of this calculation is to determine that the valve will open when called upon by design during a pressure locking event by calculating the force necessary to overcome pressure locking and the available Motor/Gearing Capability (MGC) of the MOV.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

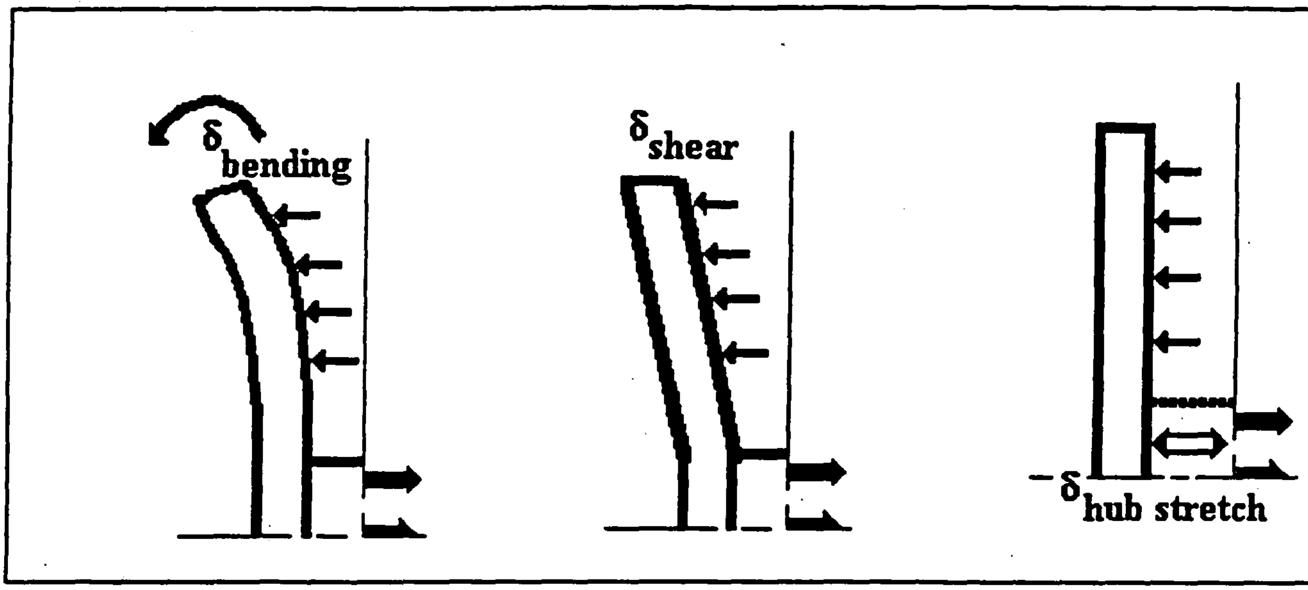
The methodology for calculating the thrust required to open the MOVs under the pressure locking scenario is based on the Reference 1 (Roark's) engineering handbook. This methodology has been previously applied by ComEd in the References 2 and 10 calculations. The methodology determines the total force 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 varies in thickness, the average thickness is used for purposes of the 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).

<u>Static Unseating Force</u>

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.

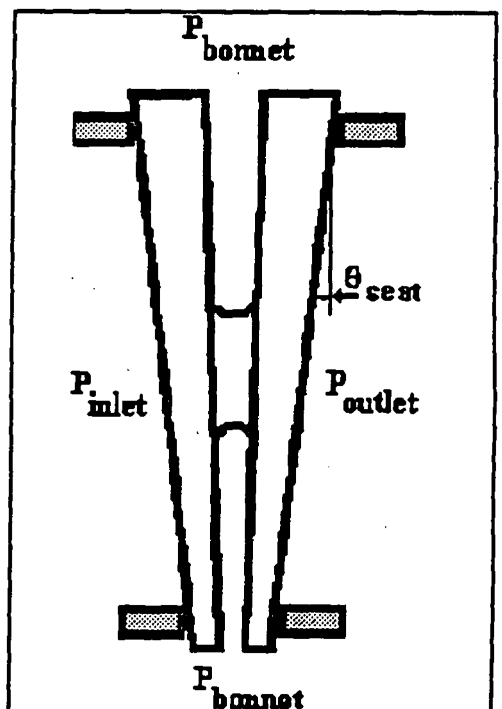
<u>Piston Effect</u>

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.

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

<u>"Reverse Piston Effect" (F_{vert})</u>

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.

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

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.

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

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 by leakage past adjacent check valves. A LOCA occurs which causes the reactor pressure to drop off to 325 psig. The LPCS pump comes up to speed and the subject valve receives a signal to open simultaneously. Total dynamic head for the CS pump is approximately 380 psig. The pressure values are based on a review of the UFSAR for Quad Cities and the CS Pump Manual, C0048. See Reference 6.

4. A valve temperature of 185 °F is assumed for the purpose of this calculation. This temperature is taken from reference 12 and is based on a one time limitation of the disk ears. The valve temperature typically does not exceed 150 °F. The assumed temperature will require verification.

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 6.3.2.1.2, Fig 6.3-3, Tbl 4.1-3 and Core Spray Pump Manual C0048

7. ComEd Rising Stem MOV Data Sheet, 1-1402-25A, 03/03/95, 11:05

8. Thrust values are taken from static VOTES Test 37 performed 08/07/94 and DP VOTES Test 38 performed 08/12/94

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. Crane-Aloyco Valve Drawing CA00790.

12. EMS Letter, EMSP-95-324, 9-15-94

VI. CALCULATIONS

The following is provided for MOV 1-1402-25A.

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 1-1402-25A

INPUTS:

Bonnet Pressure	$P_{bonnet} := 1005 \cdot psi$	Reference 6
Upstream Pressure	$P_{up} := 380 \cdot psi$	Ref. 6 & Assum. 3
Downstream Pressure	$P_{down} := 325 \cdot psi$	Ref. 6 & Assum. 3
Disk Thickness, Avg	$t := 1.69 \cdot in$	Reference 4
Seat Radius	$a := 4.36 \cdot in$	Reference 7
Hub Radius	$b := 1.25 \cdot in$	Reference 4
Hub Length	$L := 1.625 \cdot in$	Reference 4
Seat Angle	$theta := 5 \cdot deg$	Reference 7
Poisson's Ratio (disk)	$v := 0.3$	Stain. Steel, Ref. 1 & 11
Mod. of Elast. (disk)	$E := 27.6 \cdot 10^6 \cdot psi$	Stain. Steel, Ref. 1 & 11
Static Pullout Force (Test 37)	$F_{po} := 39071 \cdot lbf$	Reference 8
O10 Thrust (DP test 38)	$O10 := 13082 \cdot lbf$	Reference 8, Att. B
Open Run Thrust (DP)	$Run := 504 \cdot lbf$	Reference 8, Att. B
DP	$DPtest := 350 \cdot psi$	Reference 8, Att. B
LP (valve closed)	$LPclose := 350 \cdot psi$	Reference 8, Att. B
LP (valve open)	$LPopen := 0 \cdot psi$	Reference 8, Att. B
Stem Diameter	$D_{stem} := 1.875 \cdot in$	Reference 7

VALVE FACTOR CALCULATION

Valve Factor:

$$VF := \frac{(O10 - Run) + \frac{\pi}{4} \cdot D_{stem}^{2} \cdot (LPclose - LPopen)}{\pi \cdot (a)^2 \cdot DPtest} \qquad VF = 0.648$$

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

$$mu := VF \cdot \frac{\cos(theta)}{1 - VF \cdot \sin(theta)} \qquad mu = 0.684$$

PRESSURE FORCE CALCULATIONS

Average DP across disks:

$$DPavg := P_{bonnet} - \frac{P_{up} + P_{down}}{2} \qquad DPavg = 652.5 \cdot psi$$

Disk Stiffness Constants (Reference 1, Table 24)

$$D := \frac{E\cdot(t)^3}{12\cdot\left(1-\nu^2\right)}$$ $D = 1.22\cdot10^7 \cdot lbf\cdot in$

$$G := \frac{E}{2\cdot(1+\nu)}$$ $G = 1.062\cdot10^7 \cdot psi$

Geometry Factors: (Reference 1, Table 24)

$$C_2 := \frac{1}{4}\cdot\left[1-\left(\frac{b}{a}\right)^2\cdot\left(1+2\cdot\ln\left(\frac{a}{b}\right)\right)\right]$$ $C_2 = 0.178$

$$C_3 := \frac{b}{4\cdot a}\cdot\left[\left[\left(\frac{b}{a}\right)^2+1\right]\cdot\ln\left(\frac{a}{b}\right)+\left(\frac{b}{a}\right)^2-1\right]$$ $C_3 = 0.031$

$$C_8 := \frac{1}{2}\left[1+\nu+(1-\nu)\cdot\left(\frac{b}{a}\right)^2\right]$$ $C_8 = 0.679$

$$C_9 := \frac{b}{a}\left[\frac{1+\nu}{2}\cdot\ln\left(\frac{a}{b}\right)+\frac{1-\nu}{4}\cdot\left[1-\left(\frac{b}{a}\right)^2\right]\right]$$ $C_9 = 0.279$

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

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

$$L_{11} := \frac{1}{64}\cdot\left[1+4\cdot\left(\frac{b}{a}\right)^2-5\cdot\left(\frac{b}{a}\right)^4-4\cdot\left(\frac{b}{a}\right)^2\cdot\left[2+\left(\frac{b}{a}\right)^2\right]\cdot\ln\left(\frac{a}{b}\right)\right]$$ $L_{11} = 0.007$

$$L_{17} := \frac{1}{4}\cdot\left[1-\frac{1-\nu}{4}\cdot\left[1-\left(\frac{b}{a}\right)^4\right]-\left(\frac{b}{a}\right)^2\cdot\left[1+(1+\nu)\cdot\ln\left(\frac{a}{b}\right)\right]\right]$$ $L_{17} = 0.153$

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

$$M_{rb} := \frac{-DPavg\cdot a^2}{C_8}\cdot\left[\frac{C_9}{2\cdot a\cdot b}\cdot\left(a^2-b^2\right)-L_{17}\right]$$ $M_{rb} = -5.368\cdot10^3 \cdot lbf$

$$Q_b := \frac{DPavg}{2\cdot b}\cdot\left(a^2-b^2\right)$$ $Q_b = 4.554\cdot10^3 \cdot\frac{lbf}{in}$

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

$$y_{bq} := M_{rb}\cdot\frac{a^2}{D}\cdot C_2 + Q_b\cdot\frac{a^3}{D}\cdot C_3 - \frac{DPavg\cdot a^4}{D}\cdot L_{11}$$ $$y_{bq} = -6.597\cdot 10^{-4}\cdot in$$

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

$$K_{sa} := -0.3\cdot\left[2\cdot\ln\left(\frac{a}{b}\right) - 1 + \left(\frac{b}{a}\right)^2\right]$$ $$K_{sa} = -0.474$$

$$y_{sq} := \frac{K_{sa}\cdot DPavg\cdot a^2}{t\cdot G}$$ $$y_{sq} = -3.279\cdot 10^{-4}\cdot in$$

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

$$P_{force} := 3.1416\cdot\left(a^2 - b^2\right)\cdot DPavg$$ $$P_{force} = 3.576\cdot 10^{4}\cdot lbf$$

$$y_{stretch} := \frac{P_{force}}{3.1416\cdot b^2}\cdot\frac{L}{(2\cdot E)}$$ $$y_{stretch} = 2.145\cdot 10^{-4}\cdot in$$

Total Deflection due to pressure forces:

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

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

$$y_{sw} := -\left[\frac{1.2\cdot\left(\frac{a}{a}\right)\cdot\ln\left(\frac{a}{b}\right)\cdot a}{t\cdot G}\right]$$ $$y_{sw} = -3.644\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{bw} := -\left(\frac{a^3}{D}\right)\cdot\left[\left(\frac{C_2}{C_8}\right)\cdot\left[\left(\frac{a\cdot C_9}{b}\right) - L_9\right] - \left[\left(\frac{a}{b}\right)\cdot C_3\right] + L_3\right]$$ $$y_{bw} = -9.965\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_{compr} := \frac{2\cdot a\cdot\pi}{3.1416\cdot b^2}\cdot\frac{L}{(2\cdot E)}$$ $$y_{compr} = 1.643\cdot 10^{-7}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

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

$$y_w := y_{bw} + y_{sw} - y_{compr}$$ $$y_w = -1.525\cdot 10^{-6}\cdot\frac{in}{\left(\frac{lbf}{in}\right)}$$

(per lbf/in)

in

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

$F_s := 2\cdot\pi\cdot a\cdot\frac{y_q}{y_w}$ $\qquad F_s = 2.159\cdot 10^4 \cdot lbf$

UNSEATING FORCES (Reference 2)

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

$F_{piston} := \frac{\pi}{4}\cdot D_{stem}^2\cdot P_{bonnet}$ $\qquad F_{piston} = 2.775\cdot 10^3 \cdot lbf$

$F_{vert} := \pi\cdot a^2\cdot \sin(theta)\cdot\left(2\cdot P_{bonnet} - P_{up} - P_{down}\right)$ $\qquad F_{vert} = 6.792\cdot 10^3 \cdot lbf$

$F_{preslock} := 2\cdot F_s\cdot(mu\cdot\cos(theta) - \sin(theta))$ $\qquad F_{preslock} = 2.567\cdot 10^4 \cdot lbf$

$F_{total} := -F_{piston} + F_{vert} + F_{preslock} + F_{po}$ $\qquad F_{po} = 3.907\cdot 10^4 \cdot lbf$

$F_{total} = 6.876\cdot 10^4 \cdot lbf$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	$MR := 73.5\cdot ft\cdot lbf$	Reference 5
Temperature Factor:	$Tf := 0.936$	Reference 7
Degraded Voltage:	$DV := 381\cdot volt$	Reference 7
Under Voltage Factor:	$n := 2.0875$	Reference 5
Stem Factor:	$SF := 0.0182\cdot ft$	Reference 7
Overall Ratio:	$OAR := 29.44$	Reference 7
Pullout Efficiency:	$Eff_{po} := 0.45$	Reference 7
Motor Torque at Pullout (from latest static test)	$MT_{po} := 18.65\cdot ft\cdot lbf$	Ref 8, motor 36, pf = 0.84 @ 16 A

MOTOR / GEARING CAPABILITY CALCULATIONS:

$MGC := MR\cdot Tf\cdot OAR\cdot Eff_{po}\cdot\frac{\left(\frac{DV}{460\cdot volt}\right)^n}{SF}$ $\qquad MGC = 3.379\cdot 10^4 \cdot lbf$

OPEN STRUCTURAL LIMIT: $StructuralLimit := 83505 \cdot lbf$ **(Reference 12, Assum. 4)**

$$MarginStructural := \frac{StructuralLimit - F_{total}}{F_{total}} \qquad MarginStructural = 0.214$$

OPEN LIMIT: $Limit := (min((StructuralLimit \;\; MGC)))$

MARGIN: $$Margin := \frac{Limit - F_{total}}{F_{total}} \qquad Margin = -0.509$$

ENHANCED CAPABILITY EVALUATION:

This enhanced capability evaluation uses the measured overall MOV efficiency to predict the required motor torque during the pressure lock condition (MTrequired). 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 used to estimate the required motor torque during pressure lock conditions. The resulting margin must be a minimum of 40% to account for measurement inaccuracies and possible efficiency losses or reduced stem/stem nut coefficient of friction at the estimated pressure lock loads.

$$MT_{required} := F_{total} \cdot \frac{MT_{po}}{F_{po}} \qquad MT_{required} = 32.82 \cdot ft \cdot lbf$$

$$MT_{avail.} := MR \cdot Tf \cdot \left(\frac{DV}{460 \cdot volt}\right)^{n} \qquad MT_{avail.} = 46.423 \cdot ft \cdot lbf$$

$$Margin := \frac{MT_{avail.} - MT_{required}}{MT_{required}} \qquad Margin = 41.446 \cdot \%$$

The above enhanced capability does not take advantage of the motor locked rotor torque (88.52 ft-lb) In addition, this valve actuates within a few minutes of accident initiation, resulting in little change to motor temperature at the time of opening, thereby, eliminating the temperature degradation factor.

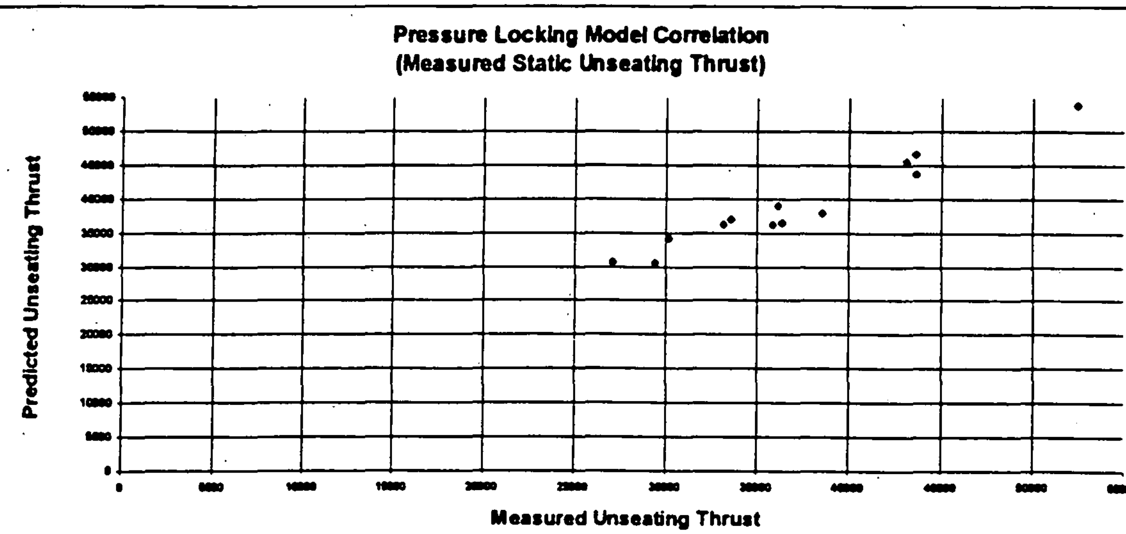
VII. COMPARISON OF MODEL OF OTHER SOURCES OF INFORMATION

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 1-1402-25A (F_{total}) is 69,940 lbf and that the Motor/Gearing Capability (MGC) is 33,790 lbf. The Open Structural Limit for MOV 1-1402-25A is the one time value of 83,505 lbf (Ref. 12 and Assum. 4). The margin was determined by finding the difference between the limiting open force, in this case the MGC value, and the unseating force, then dividing the resultant value by the total unseating force to produced a margin of -50.9%.

Based on the conservative MGC calculation insufficient margin exists for this valve. The enhanced capability evaluation was applied and indicates in excess of 41.45%.

The calculated margin of 41.5% is greater than the 40% minimum required margin, 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.

VOTES Test 38, 1-1402-25A, dated 08/12/94, pages B-1 and B-2.

CRANE VALVES Nuclear Operations

FAX TRANSMITTAL

TO BRIAN BUNTE PHONE ______ FAX 708-663-7195

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

SUBJECT DISC DIMENSIONS DATE 5/3/95

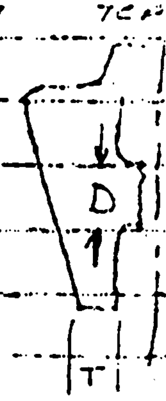
REFFRENCE ______ TOTAL PAGES 1

MESSAGE

BRIAN - THE FOLLOWING ARE APPROXIMATE NOMINAL DIMENSIONS FOR THE 10" + 16" VALVES DISCS WE ~~DIS~~ 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 3/16
16	4.25	2 3/16	3 5/16

PLATE THICKNESS IS GREATEST AT TOP OF DISC



REGARDS

[signature]

Att. I.D. A Sht 1 of 2
Calc. No. QDC-1400-M Rev. 0

0080

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.

Att. I.D. A Sht 2 of 2
Calc. No. 95-140-M Rev. 0

0080

Test: 38
8/12/94
4:1:54

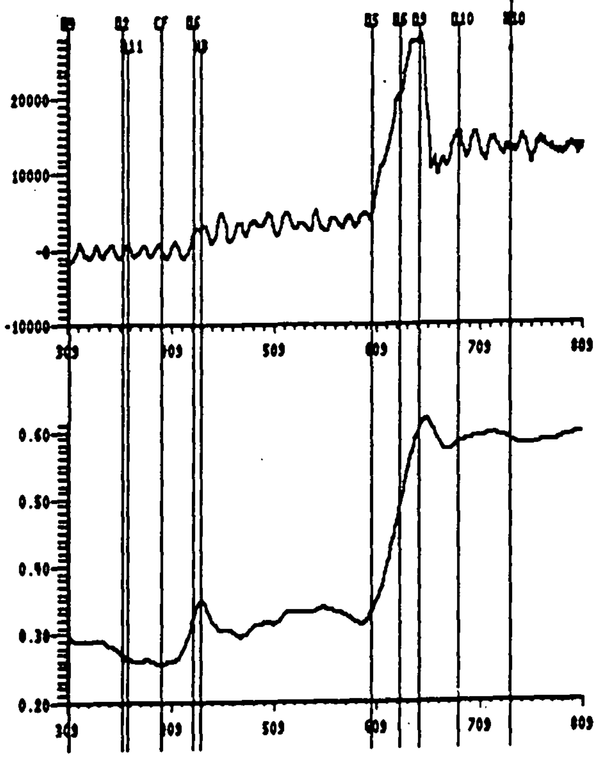
Tag: 1-1402-25A

VOTES SENSOR

Force = 13882
(lbs)
SPKS RWD

POWER FACTOR
pf = 0.585

Filter:
LowPass/Notch



Time in Seconds 0.738
Calibration Range: 484 to -74358 lbs.

Tc. Switch Setting Open/Close...............: 1.500/1.500
Limit Switch Rotor Adjustment (Y/N)...........: N
Flow (gpm) Start/Finish.......................: 6000/ 0
Upstream Pressure (psi) Start/Finish...........: 360/ 350
Downstream Pressure (psi) Start/Finish.........: 0/ 0

General Comments:
WR# Q05960; SYS:278217Q; CAL'D ON 8/7/94 W/C-CLAMP A1284 ON THREADS
DP TEST; NSO OPENED VALVE STOPPED THEN RESUMED OPENING VALVE; THEN FULLY CLOSED!
NEW SPRING PACK 0901-211&EXT GREASE RELIEF;AMPROBE=2V/200A/T2;Stem Fact=0.01050
Torque from tested spring pack @ TSS=1.5=320.6 '#; Apparent COF=0.05 used 0.08

Valve Information

Plant: QC1
Unit.: R13
Tag Number......: 1-1402-25A
Type............: GATE
Size............: 10"
Target Thrust...: 35000 lbs
Orientation.....: 45 DEG DOW
Location........: 623' RX B;MEZZ LVL
Stem Material...: A276T316
Stem Diameter...: 1.875 inches
Threads per Inch: 4.00
Threads per Rev.: 2
/Poisson Ratio.: 98.3 x 10E6 psi
erial #..: 12120
nsitivity 1.143E-0002 μv/v/lb

Valve Actuator

Actuator Type..: LIMITORQ
Size...........: SMB-2
Max Thrust Rate: 95200 lbs
Serial #.......: 88466A
Order #........: 332680
Worm Gear Teeth: 33
Gear Ratio.....: 29.4
Spring Pack # 0901-211

Actuator Motor

Voltage Type: AC
Volts.......: 460
Amp rating..: 11.40 amps
Nom. Speed..: 3405.00 rpm
Start torque: 60.00 ft-lb
Run Torque..: 12.00 ft-lb
Horse Power.: 7.80 h.p.

Signal Conditioner Calibration Due Date 08/09/94

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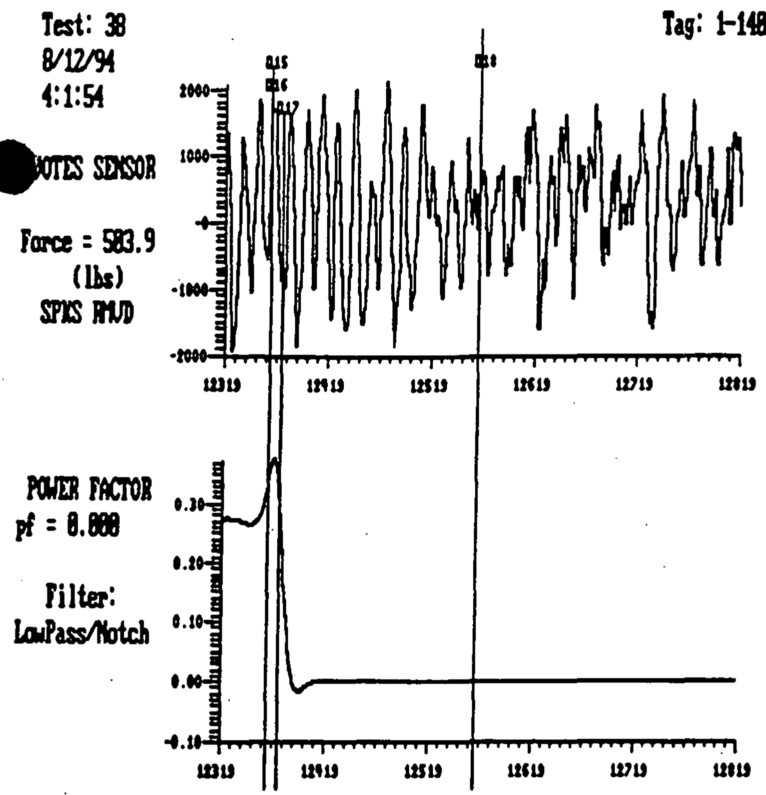
QDC-1400-M 0
0020



Time in Seconds 12.564
Calibration Range: 484 to -74358 lbs.

Tc. Switch Setting Open/Close...............: 1.500/1.500
Limit Switch Rotor Adjustment (Y/N)............: N
Flow (gpm) Start/Finish.........................: 6000/ 0
Upstream Pressure (psi) Start/Finish...........: 360/ 350
Downstream Pressure (psi) Start/Finish.........: 0/ 0
General Comments:
WR# Q05960; SYS:278217Q; CAL'D ON 8/7/94 W/C-CLAMP A1284 ON THREADS
DP TEST; NSO OPENED VALVE STOPPED THEN RESUMED OPENING VALVE; THEN FULLY CLOSED!
NEW SPRING PACK 0901-211&EXT GREASE RELIEF;AMPROBE=2V/200A/T2;Stem Fact=0.01050
Torque from tested spring pack @ TSS=1.5=320.6 '#; Apparent COF=0.05 used 0.08

Valve Information		Valve Actuator		Actuator Motor	
lant:	QC1	Actuator Type..:	LIMITORQ	Voltage Type:	AC
nit.:	R13	Size...........:	SMB-2	Volts.......:	460
ag Number......:	1-1402-25A	Max Thrust Rate:	95200 lbs	Amp rating..:	11.40 amps
ype............:	GATE	Serial #.......:	88466A	Nom. Speed..:	3405.00 rpm
ize............:	10"	Order #........:	332680	Start torque:	60.00 ft-lb
arget Thrust...:	35000 lbs	Worm Gear Teeth:	33	Run Torque..:	12.00 ft-lb
rientation.....:	45 DEG DOW	Gear Ratio.....:	29.4	Horse Power.:	7.80 h.p.
ocation........:	623' RX B;MEZZ LVL	Spring Pack #	0901-211		
tem Material...:	A276T316				
tem Diameter...:	1.875 inches				
hreads per Inch:	4.00				
hreads per Rev.:	2				
/Poisson Ratio.:	98.3 x 10E6 psi				
erial #..:	12120				
nsitivity	1.143E-0002 μv/v/lb				

Signal Conditioner Calibration Due Date 08/09/94

(FINAL)

Att. I.D. B Sht 2 of 2
Calc. No. QDC-1400-M Rev. 0
0080