Commonwealth Edison Company
1400 Opus Place
Downers Grove, IL 60515

ComEd

May 24, 1996

U. S. Nuclear Regulatory Commission
Washington, D. C. 20555
Attn: Document Control Desk

Subject: Braidwood Station Units 1 and 2
Byron Station Units 1 and 2
Dresden Station Units 2 and 3
LaSalle County Station Units 1 and 2
Quad Cities Station Units 1 and 2
Zion Station Units 1 and 2

Commonwealth Edison (ComEd) Response to NRC Staff Request for Additional Information (RAI) for Generic Letter 95-07, "PRESSURE LOCKING AND THERMAL BINDING OF SAFETY-RELATED POWER-OPERATED GATE VALVES," dated August 17, 1995.

NRC Docket Nos. 50-454 and 50-455
NRC Docket Nos. 50-456 and 50-457
NRC Docket Nos. 50-237 and 50-249
NRC Docket Nos. 50-373 and 50-374
NRC Docket Nos. 50-254 and 50-265
NRC Docket Nos. 50-295 and 50-304

References:

(a) NRC Generic Letter (GL) 95-07, "PRESSURE LOCKING AND THERMAL BINDING OF SAFETY-RELATED POWER-OPERATED GATE VALVES," dated August 17, 1995.

(b) P. Piet (ComEd) letter to U. S. NRC, dated February 13, 1996, ComEd Response to GL 95-07.

(c) Teleconference between ComEd (P. Piet, B. Bunte) and members of the NRC staff (C. Shiraki, et. al.), dated March 8, 1996.

(d) C. Shiraki (NRC staff) letter to D. Farrar (ComEd), dated April 2, 1996, NRC Request for Additional Information (RAI) regarding GL 95-07.

In Reference (a), the NRC staff issued GL 95-07 that requested licensees ensure that safety-related power-operated gate valves susceptible to pressure locking or thermal binding are capable of performing their safety function and are within the current licensing bases of the facility. In Reference (b), ComEd submitted its 180-day response to GL 95-07 for each of our six facilities. ComEd discussed various issues regarding GL 95-07 during the Reference (c) teleconference with members of the NRC staff. The NRC staff issued the Reference (d) RAI to complete its review of ComEd's program that addresses concerns raised by GL 95-07. In

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Reference (d), the NRC staff requested the following additional information: *(1) the thrust prediction methodology (including the method for predicting actuator output capability), (2) the test procedures (including information specific to each test valve sufficient to perform the pressure locking calculations), (3) the test results (including the method for interpreting diagnostic equipment data), (4) the information regarding the diagnostic equipment used during testing (including calibration methods and diagnostic uncertainties), and (5) any limitations or conditions placed on the use of the methodology (i.e., valve size, type, temperature, pressure, etc.).* It should be noted that a response period was not specified within the NRC staff's RAI.

To respond to the Reference (d) RAI, ComEd is providing the following information:

- Attachment 1 includes a report that is being presented at the upcoming ASME/NRC Pump and Valve Symposium. This report provides a detailed description of the methodology used to predict the pressure locking unseating load.

  For performing operability assessments only, actuator capability calculations using the following process have been performed:

  - Determine motor torque at static unseating using motor power measurements
  - Calculate available motor torque at degraded voltage[1]
  - Calculate pressure locking unseating load using Attachment 1 methodology
  - Calculate motor torque required under pressure locking by multiplying the static unseating torque by the pressure locking thrust to static unseating thrust ratio

  Attachment 1 also includes a summary of the pressure locking test results.

- Attachments 2, 3 and 4 are the test procedures used to obtain pressure locking test data for the Crane, Westinghouse, and Borg-Warner valves. Attachments 2, 3 and 4 also provide the calibration ID, the last calibration date prior to the test and the schedule calibration for the test equipment. The equipment is calibrated with standards which are traceable to NIST standards.

- Attachments 5, 6 and 7 are examples of the pressure locking MathCad models used to calculate the expected pressure locking thrust for these valves. These example MathCad worksheets include all valve internals dimensional information required to use the ComEd pressure locking model.

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1 ComEd White Paper 125, "Installed MOV Motor Gearing Capability," Revision 2, dated October 4, 1995. This report contains the proprietary methodology used to predict the actuator capability for design calculations and has been previously presented to and reviewed by members of the NRC staff (NRR and RIII).

- Attachments 8, 9 and 10 provide key pressure and thrust measurements taken for each of the MOV strokes during the pressure locking tests. These included static strokes, hydro-pump DP test strokes, and pressure locking test strokes for the MOVs.

- Attachment 1 discusses the diagnostic test equipment used during the testing. The equipment used is standard MOV test equipment (VOTES and MPM) used by the ComEd MOV program. Test equipment inaccuracy has not been included in the values tabulated in Attachments 1, 8, 9 and 10. The ComEd pressure locking calculation methodology provides a best-estimate of the pressure locking unseating load. The ComEd test program's purpose is to test the accuracy of the ComEd pressure locking model. The measured pressure locking loads were compared to the predicted loads. The amount of variation between the predicted and measured pressure locking load forms the basis for the margin that is required with respect to predicted pressure locking loads. The accuracy of the methodology is graphically demonstrated in Attachment 1 (a comparison of predicted pressure locking unseating loads to measured pressure locking unseating loads).

  When using the ComEd methodology to predict the pressure locking unseating load, users are required to justify that sufficient margin is available for uncertainty in static unseating load, seat friction coefficient and stem factor. The ComEd NES overview inspection process for MOVs and a NES Calculation Metrics Review process will be used to ensure proper use of the pressure locking calculation methodology by ComEd Stations.

- Based on a review of the test data, ComEd has placed one limitation on use of the pressure locking methodology. The methodology should not be used when the calculated bonnet pressure exceeds the pressure rating of the affected MOV. Testing of the Borg-Warner valve and the Crane valve suggests that the methodology may not always be conservative under these conditions.

ComEd anticipates the opportunity to review test data collected by INEL and other licensees. Other limitations may be placed on the methodology pending review of the INEL and other test data.

I affirm that the content of this transmittal is true and correct to the best of my knowledge, information and belief.

Please direct any questions concerning this response to this office.

Sincerely,

John B Hosmer

John B. Hosmer
Vice President

Signed before me on this 24th day,

of May, 1996.
Notary Public

Mary Jo Yack

OFFICIAL SEAL
MARY JO YACK
NOTARY PUBLIC STATE OF ILLINOIS
MY COMMISSION EXPIRES: 11/29/97

Attachments

cc: H. Miller, Regional Administrator - RIII
R. Capra, Director of Directorate III-2, NRR
G. Dick, Byron Project Manager, NRR
R. Assa, Braidwood Project Manager, NRR
J. Stang, Dresden Project Manager, NRR
D. Skay, LaSalle Project Manager, NRR
R. Pulsifer, Quad Cities Project Manager, NRR
C. Shiraki, Zion Project Manager, NRR
C. Phillips, Senior Resident Inspector (Braidwood)
H. Peterson, Senior Resident Inspector (Byron)
C. Vanderniet, Senior Resident Inspector (Dresden)
P. Brochman, Senior Resident Inspector (LaSalle)
C. Miller, Senior Resident Inspector (Quad Cities)
R. Westberg, Senior Resident Inspector (Zion)

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RESPONSE TO NRC STAFF REQUEST FOR
ADDITIONAL INFORMATION (RAI)
FOR GENERIC LTR 95-07

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Attachment 1

ComEd Response to NRC Request for Additional Information on ComEd Pressure Locking Testing

COMMONWEALTH EDISON COMPANY PRESSURE LOCKING TEST REPORT

Brian D. Bunte, P.E.

Commonwealth Edison Company

John F. Kelly, P.E.

VECTRA Technologies, Inc.

ABSTRACT

Pressure Locking is a phenomena which can cause the unseating thrust for a gate valve to increase dramatically from its typical static unseating thrust. This can result in the valve actuator having insufficient capability to open the valve. In addition, this can result in valve damage in cases where the actuator capability exceeds the valve structural limits. For these reasons, a proper understanding of the conditions which may cause pressure locking and thermal binding, as well as a methodology for predicting the unseating thrust for a pressure locked or thermally bound valve, are necessary.

This report discusses the primary mechanisms which cause pressure locking. These include sudden depressurization of piping adjacent to the valve and pressurization of fluid trapped in the valve bonnet due to heat transfer. This report provides a methodology for calculating the unseating thrust for a valve which is pressure locked. This report provides test data which demonstrates the accuracy of the calculation methodology.

DESCRIPTION OF PRESSURE LOCKING PHENOMENA

Pressure locking occurs when the bonnet cavity pressure of a gate valve exceeds the pressure on <u>both</u> sides of the valve disk. The two primary mechanisms that exist for pressure locking of gate valves are described below:

<u>SUDDEN DEPRESSURIZATION</u>: This pressure locking mechanism occurs when a valve is pressurized from one side. Leakage past the valve seat will cause the fluid in the gate valve bonnet to pressurize to the pressure of the high pressure side of the valve disk. Depending on the leak-tightness of the valve seats, this pressurization process may take seconds or hours; however, it is extremely unlikely that the valve seat will be sufficiently leak tight to prevent this process from eventually occurring. If the source of pressure is suddenly removed, then pressure in the bonnet valve will remain trapped. If the valve is called upon to open before the bonnet pressure has decayed to the line pressure, then a pressure locking event occurs.

The time needed for the bonnet pressure to decay is dependent on several factors including leak tightness of valve seats and packing. In addition, when the bonnet fluid is at a high temperature or contains large amounts of air, the bonnet pressure decays much more slowly due to the pressurizer effect. Apparent cases of pressure locking occurring up to a day after the pressure source is removed have been recorded. However, test data presented later in this report suggests that the bonnet pressure is likely to decay within one hour of the sudden depressurization event occurring. This type of pressure locking is likely to occur when pumps adjacent to closed valves are shut off or when an event such as a LOCA causes pressure on one side of a valve to suddenly drop off.

When the initial differential pressure across the valve disk is sufficient to unseat the high pressure side disk from its seat, then the bonnet pressure following a sudden depressurization event is less than the bonnet pressure at the start of the event. The maximum pressure which can be trapped in the valve bonnet can be calculated by determining the differential pressure at which the valve disk will come back into contact with the valve seat. Until the disk to seat contact is re-established, the bonnet pressure will follow the upstream side pressure. This calculation has been developed by ComEd, but is not provided in this report due to constraints on length.

<u>THERMALLY INDUCED PRESSURE RISE IN BONNET</u>: This pressure locking mechanism occurs when the valve bonnet cavity of a gate valve is filled with liquid that contains little or no air. If a heat source is applied to fluid in the valve bonnet cavity, then expansion of the fluid can cause pressure in the valve bonnet to dramatically increase. The heat source can be fluid in piping adjacent to the valve or external environmental conditions as might be encountered following a high energy line break. Pressurization rates of 20 psi/°F to 60 psi/°F have been recorded during special testing. However, pressurization rates of this nature require the following conditions to exist:

- the valve seats and packing must be very leak tight
- the heat source must provide a high heat transfer rate to the bonnet cavity fluid
- no air can exist in the valve bonnet cavity, or the temperature rise in the valve bonnet cavity must be sufficient to cause the expanding fluid to collapse the air bubbles before the high pressurization rate can be achieved.

PRESSURE LOCKING CALCULATION METHODOLOGY

ASSUMPTIONS

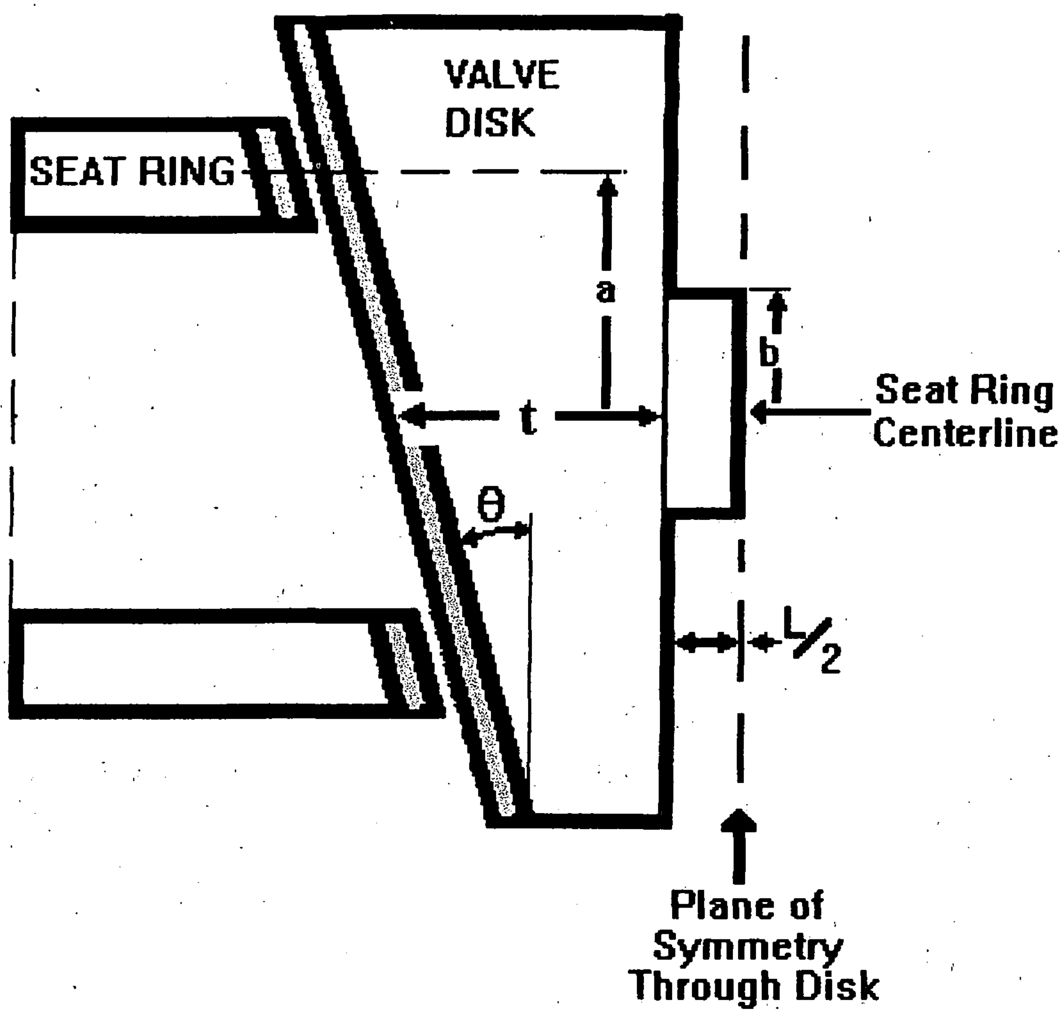
1. The valve disk is assumed to act as two ideal disks connected by a hub. The equations in reference 1 are assumed to conservatively model the actual load due to pressure forces.
2. The coefficient of friction between the valve seat and disk is assumed to be the same under pressure locking conditions as it is under DP conditions.

DESIGN INPUTS

The following design inputs are used in calculating the force required to unseat a pressure locked MOV:

- Design Basis Pressure Conditions at the time of the pressure locking event. This includes the upstream ($\mathbf{P_{up}}$), downstream ($\mathbf{P_{down}}$), and bonnet pressure ($\mathbf{P_{bonnet}}$).
- Valve Disk Geometry. This includes the hub radius (**b**), hub length (**L**), mean seat radius (**a**), seat angle (θ), and average disk thickness (**t**). Figure 1 below is provided for further clarification. When the hub cross-section is not circular (e.g. many Westinghouse gate valve designs), then an effective hub radius which corresponds to a circle of equal area to the hub cross-sectional area should be used.
- Valve Disk Material Properties. This includes the modulus of elasticity (**E**) and the Poisson's ratio (ν) for the disk base material.
- Valve Stem Diameter ($\mathbf{D_{stem}}$)
- Static Unseating Thrust ($\mathbf{F_{po}}$)
- Coefficient of Friction between Disk and Seat (μ)

FIGURE 1



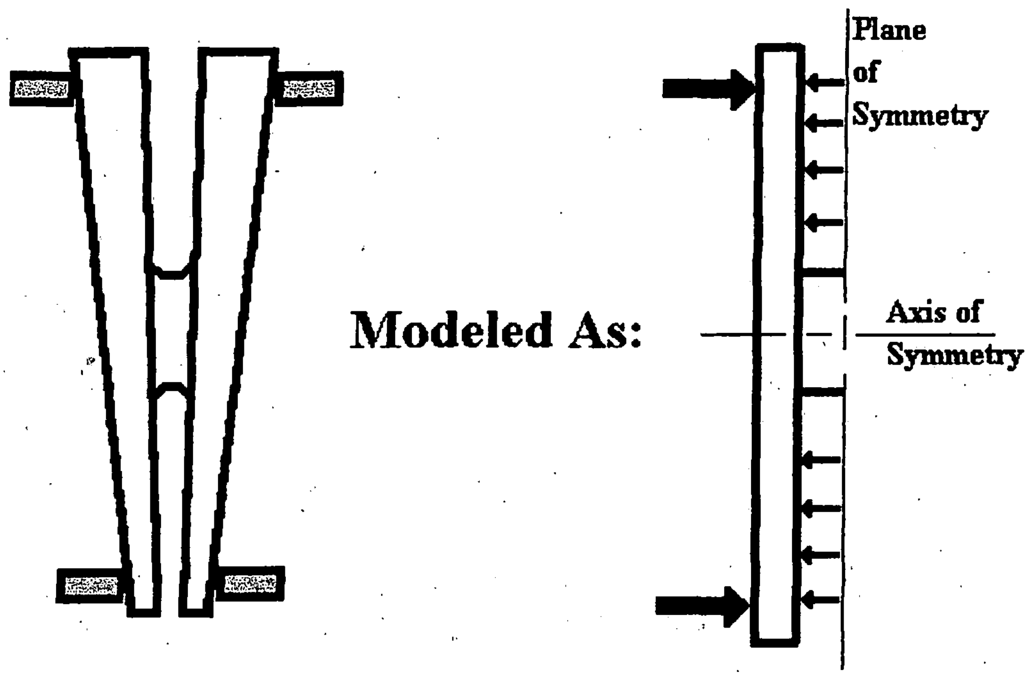
CALCULATIONS

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 is based in part on calculations developed by MPR Associates (Reference 2). The methodology determines the total force required to open the valve under a pressure locking scenario by calculating 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 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.

FIGURE 2



Based on this geometry, the following constants are calculated using the Reference 1 equations:

Average DP Across Disk

$$DP_{avg} = P_{bonnet} - \frac{P_{up} + P_{down}}{2} \qquad \textbf{(1)}$$

Disk Stiffness Constants

(*Reference* 1, *Table* 24)

$$D = \frac{E \times t^3}{12 \times (1 - \nu^2)} \qquad \textbf{(2)}$$

$$G = \frac{E}{2 \times (1 + \nu)} \qquad \textbf{(3)}$$

Geometry Factors

(*Reference* 1, *Table* 24)

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

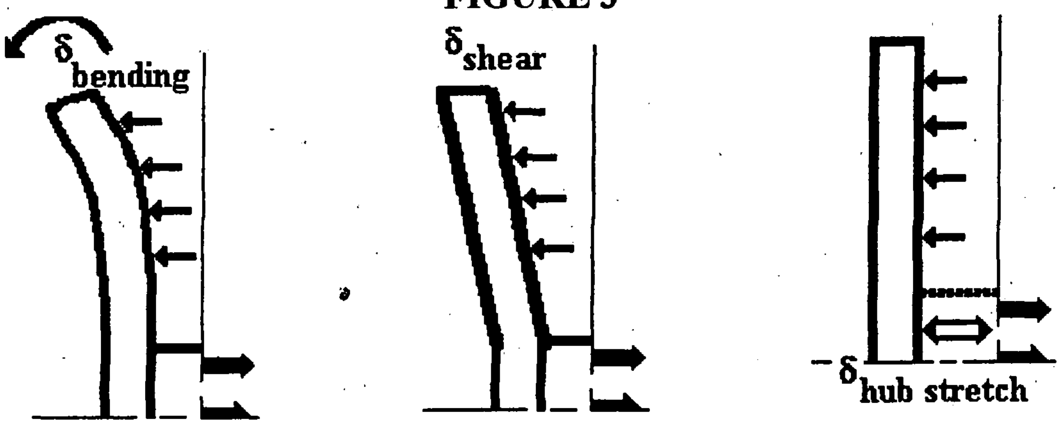
$$C_3 = \frac{b}{4a}\left\{\left[\left(\frac{b}{a}\right)^2 + 1\right]\ln\left(\frac{a}{b}\right) + \left(\frac{b}{a}\right)^2 - 1\right\} \qquad \textbf{(5)}$$

$$C_8 = \frac{1}{2}\left[1 + \nu + (1 - \nu)\left(\frac{b}{a}\right)^2\right] \qquad \textbf{(6)}$$

$$C_9 = \frac{b}{a}\left\{\frac{1 + \nu}{2}\ln\left(\frac{a}{b}\right) + \frac{1 - \nu}{4}\left[1 - \left(\frac{b}{a}\right)^2\right]\right\} \qquad \textbf{(7)}$$

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.

FIGURE 3



Additional Geometry Factors

(*Reference* 1, *Table* 24) $$L_{11} = \frac{1}{64}\left\{1 + 4\left(\frac{r_0}{a}\right)^2 - \left(\frac{r_0}{a}\right)^4 - \left(\frac{r_0}{a}\right)^2\left[2 + \left(\frac{r_0}{a}\right)^2\right]\ln\left(\frac{a}{r_0}\right)\right\} \quad (8)$$

($r_0 = b$ *for Case* 2*L*) $$L_{17} = \frac{1}{4}\left\{1 - \frac{1-\nu}{4}\left[1 - \left(\frac{r_0}{a}\right)^4\right] - \left(\frac{r_0}{a}\right)^2\left[1 + (1+\nu)\ln\left(\frac{a}{r_0}\right)\right]\right\} \quad (9)$$

Moment Factors

(*Reference* 1, *Table* 24, *Case* 2*L*) $$M_{rb} = \frac{-DPavg \times s^2}{C_8}\left[\frac{C_9}{2 \times a \times b}\left(a^2 - r_0^2\right) - L_{17}\right] \quad (10)$$

($r_0 = b$ *for Case* 2*L*) $$Q_b = \frac{DPavg}{2 \times b}\left(a^2 - r_0^2\right) \quad (11)$$

Deflection from pressure / bending

(*Reference* 1, *Table* 24, *Case* 2*L*) $$y_{bq} = M_{rb}\frac{a^2}{D}C_2 + Qb\frac{a^3}{D}C_3 - \frac{DPavg \times a^4}{D}L_{11} \quad (12)$$

Deflection from pressure / shear

$(Reference\,1, Table\,25, Case\,2L)$ $$Ksa = -0.3\left[2\ln\left(\frac{a}{b}\right) - 1:\left(\frac{r_0}{a}\right)^2\left(1 - 2\ln\left(\frac{r_0}{b}\right)\right)\right] \quad \textbf{(13)}$$

$(r_0 = b\ for\ Case\,2L)$ $$y_{sq} = \frac{K_{sa} \times DPavg \times a^2}{t \times G} \quad \textbf{(14)}$$

Deflection from pressure / hub stretch

$$P_{force} = \pi\left(a^2 - b^2\right)DPavg \quad \textbf{(15)}$$

$$y_{stretch} = \frac{-P_{force}}{\pi \times b^2}\frac{L}{2 \times E} \quad \textbf{(16)}$$

Total Deflection due to pressure

$$y_q = y_{bq} + y_{sq} + y_{stretch} \quad \textbf{(17)}$$

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 was previously calculated. The Reference 1 equations are now 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. This is done by first calculating the amount deflection created by a unit load of seat contact force (w = 1 lbf/in). The equilibrium contact load is then determined by dividing the deflection caused by the unit contact load into the previously calculated deflection due to the pressure force. The equations are provided below:

Additional Geometry Factors

(*Reference* 1, *Table* 24, *Case* 1*L*) $$L_3 = \frac{r_0}{4 \times a}\left\{\left[\left(\frac{r_0}{a}\right)^2 + 1\right]\ln\left(\frac{a}{r_0}\right) + \left(\frac{r_0}{a}\right)^2 - 1\right\} \quad (18)$$

(*for Case* 1*L*, $r_0 = a$, $\therefore L_3 = L_9 = 0$) $$L_9 = \frac{r_0}{a}\left\{\frac{1+\nu}{2}\ln\left(\frac{a}{r_0}\right) + \frac{1-\nu}{4}\left[1 - \left(\frac{r_0}{a}\right)^2\right]\right\} \quad (19)$$

Deflection from seat load / bending $(r_0 = a)$

(*Reference* 1, *Table* 24, *Case* 1*L*, $w = 1$) $$y_{bw} = -\frac{a^3}{D}\left[\frac{C_2}{C_8}\left(\frac{r_0 \times C_9}{b} - L_9\right) - \frac{r_0 \times C_9}{b} + L_3\right] \quad (20)$$

Deflection from seat load / shear $(r_0 = a)$

(*Reference* 1, *Table* 25, *Case* 1*L*, $w = 1$) $$K_{sa} = -1.2\,\frac{r_0}{a}\,\ln\left(\frac{r_0}{b}\right) \quad (21)$$

$$y_{sw} = K_{sa}\,\frac{a}{t \times G} \quad (22)$$

Deflection from seat load / hub compr.

$w = 1$, $\therefore$ *Compressive force* $= 2 \times \pi \times a$ $$y_{compr} = -\frac{2 \times \pi \times a}{\pi \times b^2}\left(\frac{L/2}{E}\right) \quad (23)$$

Total Deflection from unit seat load

$(w = 1)$ $$y_w = y_{bw} + y_{sw} + y_{compr} \quad (24)$$

Therefore, the equilibrium contact load distribution (lbf/in) and the corresponding load applied to each seat is calculated using the relationship below:

$$w_{equalibrium} = {}^{y_q}\!/_{y_w}\text{, where } y_w \text{ is calculated for } w = 1 \tag{25}$$

$$Load\ per\ seat = 2 \times \pi \times a \times \frac{y_q}{y_w} \tag{26}$$

Several methods may be used to determine an appropriate seat to disk friction coefficient. Using this friction coefficient and a force balance on the disk to seat interface, the following equation is derived for calculating the stem force required to overcome the increased contact load between the seat and disk:

$$F_{pres\,lock} = \left(2 \times \pi \times a \times \frac{y_a}{y_w}\right) \times \left[\mu \times \cos(\theta) - \sin(\theta)\right] \times 2 \tag{27}$$

where the last 2 corresponds to the number of seats

<u>Static Unseating Force (F_{static})</u>

The static unseating force results from 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 (F_{piston})

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.

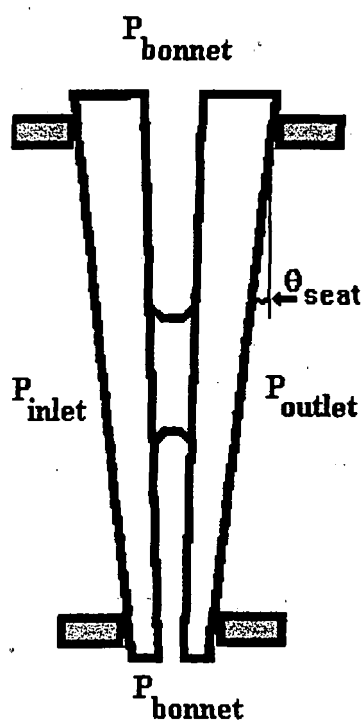
$$F_{piston\,effect} = \frac{\pi}{4} \times D_{stem}^{2} \times \left(P_{bonnet} - P_{atm} \right) \tag{28}$$

"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 calculated as follows:

$$F_{vert} = \left[\pi \times a^2 \times \left(2 \times P_{bonnet} - P_{inlet} - P_{outlet} \right) \right] \times \sin(\theta) \tag{29}$$

FIGURE 3



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.

$$F_{total} = F_{pres\,lock} + F_{static} + F_{vert} - F_{piston} \tag{30}$$

DESCRIPTION OF TEST VALVES

ORIGIN

The three test valves were obtained from different sources. The Crane valve is a test valve located at Quad Cities Station. The Westinghouse valve was obtained through the Westinghouse Owners Group. The Borg-Warner valve was obtained from Arizona Public Service.

PAST SERVICE AND TEST HISTORY

The Crane valve is a spare valve which was subjected to blowdown testing at Wyle Laboratories in Huntsville, Alabama. The Westinghouse valve is a test valve which was subjected to limited testing at South Texas Project. The Borg-Warner valve was a spare valve which had not been subjected to previous testing other than that performed at the vendor prior to delivery.

MATERIALS

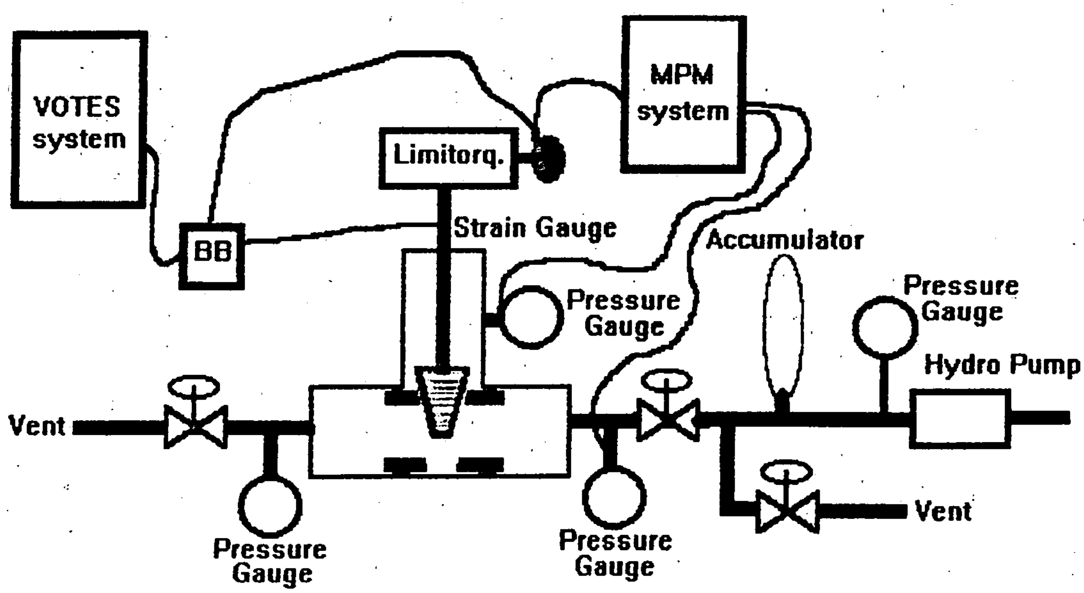
The Crane valve is a carbon steel valve (Model 783-U) which was modified during blowdown testing to contain a stainless steel valve disk and malcolmized guide rail (similar to the Model 783-UL valve design). The Westinghouse valve and Borg-Warner valve were stainless steel valve designs.

DESCRIPTION OF TEST APPARATUS

INSTRUMENTATION AND DATA ACQUISITION SYSTEMS

The figure below shows the basic test setup used for the pressure locking tests. A VOTES data acquisition system and a Motor Power Monitor (MPM) data acquisition system were used to collect stem thrust, actuator torque, and motor power data. In addition, on-line pressure data was collected during the Westinghouse and Borg-Warner valve tests. A hydrostatic test pump and accumulator were used as the pressure source during pressure locking tests and hydropump DP tests:

FIGURE 4



VALVE ORIENTATIONS

For the Crane test, the valve was laid on its side with the stem slightly below horizontal. This configuration was used to ensure that no air pockets would be trapped within the valve body when it was filled with water.

The Westinghouse valve was installed in a test stand with the stem upright. The valve bonnet was vented by bleeding air out of the packing leak-off line.

The Borg-Warner valve was installed in a special test stand which allowed pivoting the valve about its centerline. The valve stem could be put at any angle between upright and sloped downward at a 15 degree angle in either direction. To remove air from the valve bonnet, the valve was rotated on its side and rocked up and down as it filled with water.

DESCRIPTION OF TEST METHODS

STATIC BASELINE TESTS

The test process started with static test strokes to verify the proper installation of the data acquisition systems and to measure static unseating load magnitude and repeatability.

LOCAL LEAK RATE TESTS

Local leak rate tests of the valves were performed to measure seat tightness. These tests were performed at multiple torque switch settings in some cases.

DP TESTS

DP Tests in the open direction were performed by pressurizing the valve from one side with the hydropump and then stroking the valve open. Test data indicates that the differential pressure was maintained across the valve disk while the disk slid across the valve seat. The purpose of the DP tests was to precondition the valve seats and disks and to monitor the seat-to-disk friction coefficient. The DP tests were performed until a stable friction coefficient was achieved.

PAIRED STATIC / PRESSURE LOCKING TESTS

A series of pressure locking tests was performed for each valve. Inlet pressure, outlet pressure, bonnet pressure, and static seating force were varied during these tests. Static baseline tests to measure the static unseating load were performed between the pressure locking tests. The closure strokes for the static tests were performed at the same initial conditions (pressure and seating force) as the closure strokes prior to the pressure locking tests so that the change in unseating load due to pressure locking could be accurately determined.

BONNET DEPRESSURIZATION TESTS

To measure the seat tightness, bonnet depressurization rate tests were performed. The entire valve assembly (including the valve bonnet) was pressurized while in the closed position. Then the upstream and downstream pressure were vented. The bonnet pressure as a function of time was measured.

THERMALLY INDUCED BONNET PRESSURIZATION TESTS

To measure the potential for pressure locking due to bonnet fluid heat-up, thermally induced bonnet pressurization rate tests were performed on the Westinghouse and Borg-Warner valves. After venting air from the valve bonnet cavity, each valve was closed while filled with water at approximately 100 psig. The valve bonnet was then heated using an outside heat source. The pressure of the fluid in the valve bonnet was measured directly. The temperature of fluid in the valve bonnet for the Borg-Warner valve and the temperature of the outside of the valve bonnet for the Westinghouse valve were measured. Initial pressurization rates between 0.5 and 2.0 psi/degree F were measured. Much higher ultimate pressurization rates were witnessed during the Borg-Warner tests. The data from this testing is not presented in this report, but is available from ComEd upon request.

PRESSURE LOCKING TEST DATA

The following table provides the pressure locking test results comparing the measured pressure locking unseating load to the predicted pressure locking unseating load:

TABLE 1

Valve	Test #	TSS	Static Unseating Thrust	Bonnet Pressure	Predicted Increase	Measured Increase	Percent Conservatism (Non-Cons.)	Notes
Crane 10"	6	1	25000	650	5103	4539	-2%	6
Crane 10"	7	1	25000	850	7213	8191	4%	6
Crane 10"	9	1	26000	1040	9421	11500	8%	6
Crane 10"	10	1	26000	1040	9922	12140	9%	6
Crane 10"	13	1	28000	1195	19462	22140	10%	
Crane 10"	14	1	28000	1375	22974	25480	9%	
Crane 10"	15	1	28000	1375	23126	25480	8%	
Crane 10"	34	2.5	38000	655	6243	5796	-1%	6
Crane 10"	35	2.5	38000	655	5142	5796	2%	6
Crane 10"	38	2.5	37500	1055	13164	13870	2%	6
Crane 10"	39	2.5	37500	1055	13065	13870	2%	6
Crane 10"	42	2.5	40000	1365	30028	29190	-2%	
Crane 10"	43	2.5	40000	1165	30428	24913	-14%	5
Crane 10"	46	2.5	40000	1575	32231	33680	4%	
Crane 10"	47	2.5	40000	1575	31931	33680	4%	
Crane 10"	50	2.5	40000	1775	37749	37950	1%	3,4
West. 4"	30	2	1450	496	1537.6	1555	-1%	
West. 4"	31	2	1450	514	1593.4	1538	2%	
West. 4"	33	2	900	1000	3100	3007	2%	
West. 4"	35	2	900	1000	3100	2990	3%	
West. 4"	37	2	50	1500	4650	4775	-3%	
West. 4"	39	2	50	1500	4650	4672	0%	
West. 4"	42	2	-400	2000	6200	5989	4%	
West. 4"	44	2	-400	2000	6200	6126	1%	
Borg-W. 10"	43	2	16935	205	5691	8532	4%	1
Borg-W. 10"	48	1	7882	209	5802	7386	19%	1
Borg-W. 10"	50	1	7782	402	11160	13004	16%	1
Borg-W. 10"	52	1	7906	630	17489	18799	23%	1
Borg-W. 10"	54	1	7882	694	19265	20514	23%	1
Borg-W. 10"	56	1	5023	919	25511	36849	-164%	1,2
Borg-W. 10"	74	2	17477	208	6225	10167	-2%	1
Borg-W. 10"	75	2	17477	213	6375	10765	-5%	1
Borg-W. 10"	77	2	17751	391	11703	16155	-5%	1
Borg-W. 10"	78	2	17751	402	12032	16853	-7%	1
Borg-W. 10"	80	2	17949	467	13977	22172	-26%	1,2
Borg-W. 10"	81	2	17949	219	6555	10591	-2%	1
Borg-W. 10"	83	2	17700	110	3292	7757	-5%	1
Borg-W. 10"	84	2	17700	55	1646	5171	0%	1
Borg-W. 10"	86	2	17352	0	0	3628	0%	3
Borg-W. 10"	95	1	8000	0	0	3132	0%	3
Borg-W. 10"	96	1	8000	557	16671	19035	9%	1
Borg-W. 10"	97	1	8000	504	15085	18189	0%	1

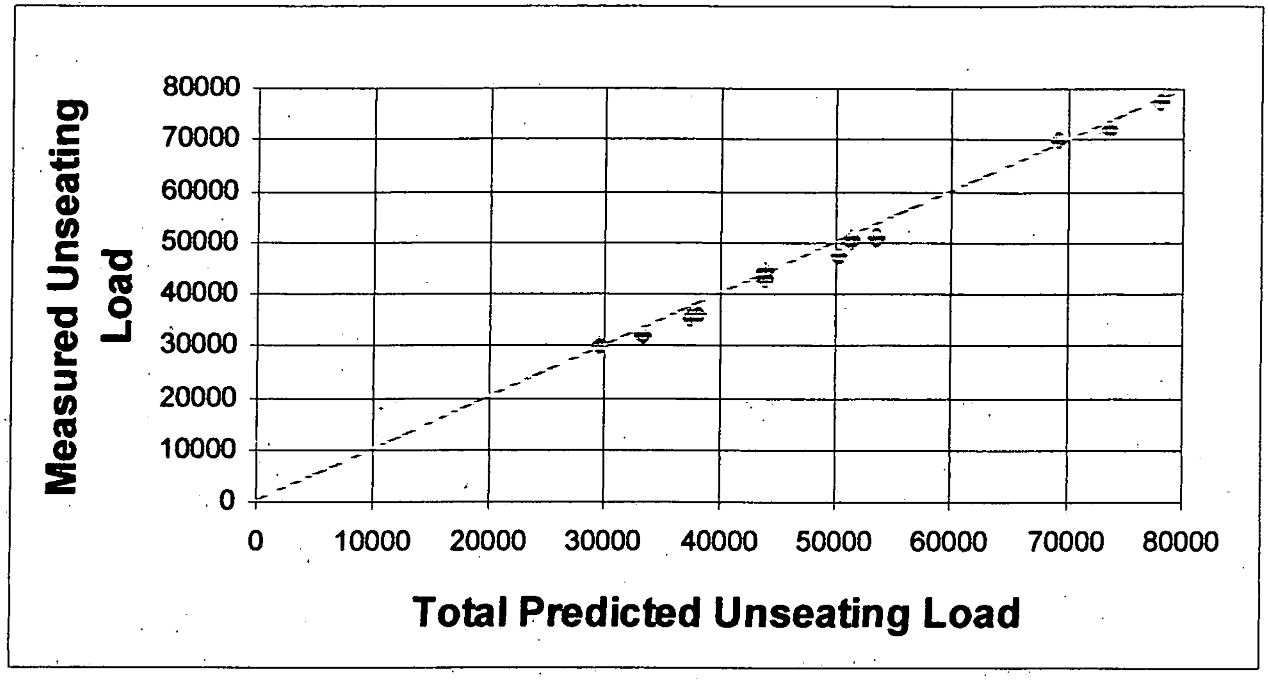
NOTES:

1. The percent conservatism values are calculated after a "memory effect" of 3100 lbf (at TSS=1) or 3500 lbf (at TSS=2) is added to the predicted pressure locking load. Testing indicated that the process of applying and then relieving pressure against one side of the closed valve was sufficient to cause the unseating force to increase by these amounts, even when no pressure was captured in the valve bonnet. This effect was only noted for the Borg-Warner test valve.
2. When bonnet pressure significantly exceeds the pressure class rating of the test valve, the pressure locking calculation methodology appears to become non-conservative.
3. Tests 86 and 95 were performed to quantify the "memory effect" for the Borg-Warner valve. These tests were performed like a pressure locking test in that high pressure (~ 600 psig) was put against one side of the valve disk and then bled off. However, any pressure that entered the valve bonnet was relieved prior to the opening stroke.
4. The AC motor for the test valve stalled during this test and the valve did not fully unseat. Test data suggests that open valve motion was initiated prior to the stall. Consequently, the measured increase due to pressure locking is believed to be correct.
5. The pressure data for this test is questionable and is being evaluated at this time.
6. The upstream and downstream pressure during these tests was approximately 350 psig. This was done to approximate the LPCI and LPCS injection valve pressure conditions which could exist in the event of a LOCA.

Graphs 1 through 6 provide the data in Table 1 for the three test valves. The total measured unseating load versus the total predicted unseating load and the pressure related portion of the measured load versus the predicted pressure related portion of the unseating load are plotted for each valve.

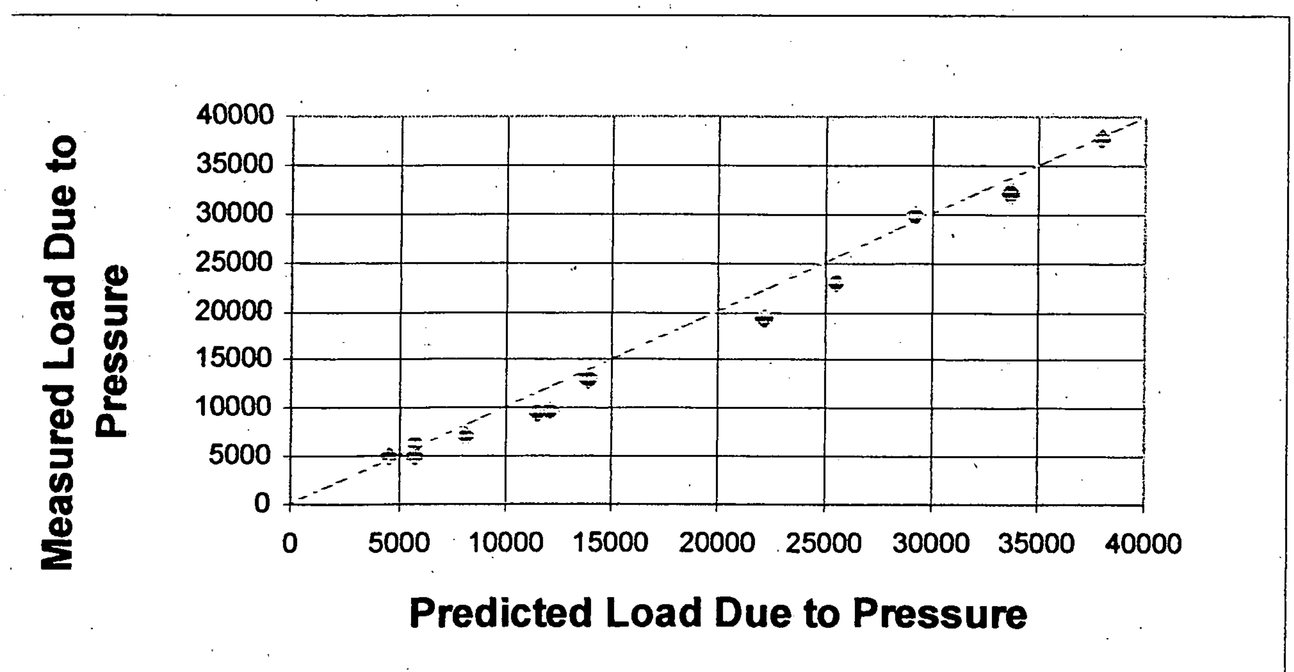
GRAPH 1

Predicted Unseating Thrust Versus Measured Pressure Locking Unseating Force for Crane Valve

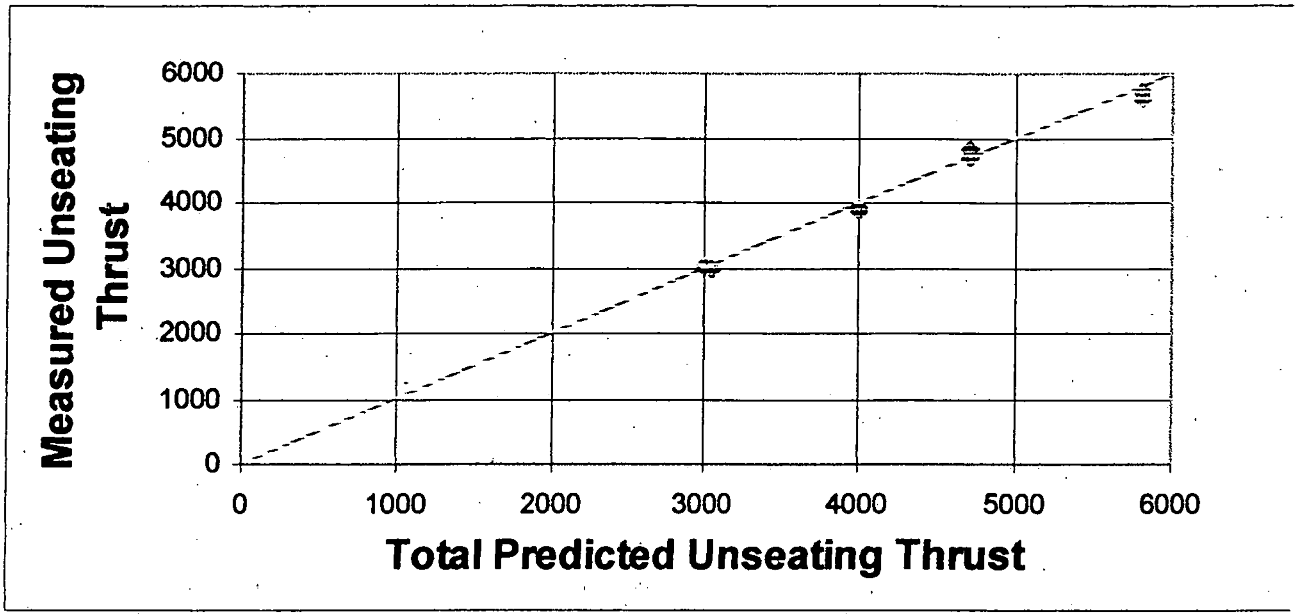


GRAPH 2

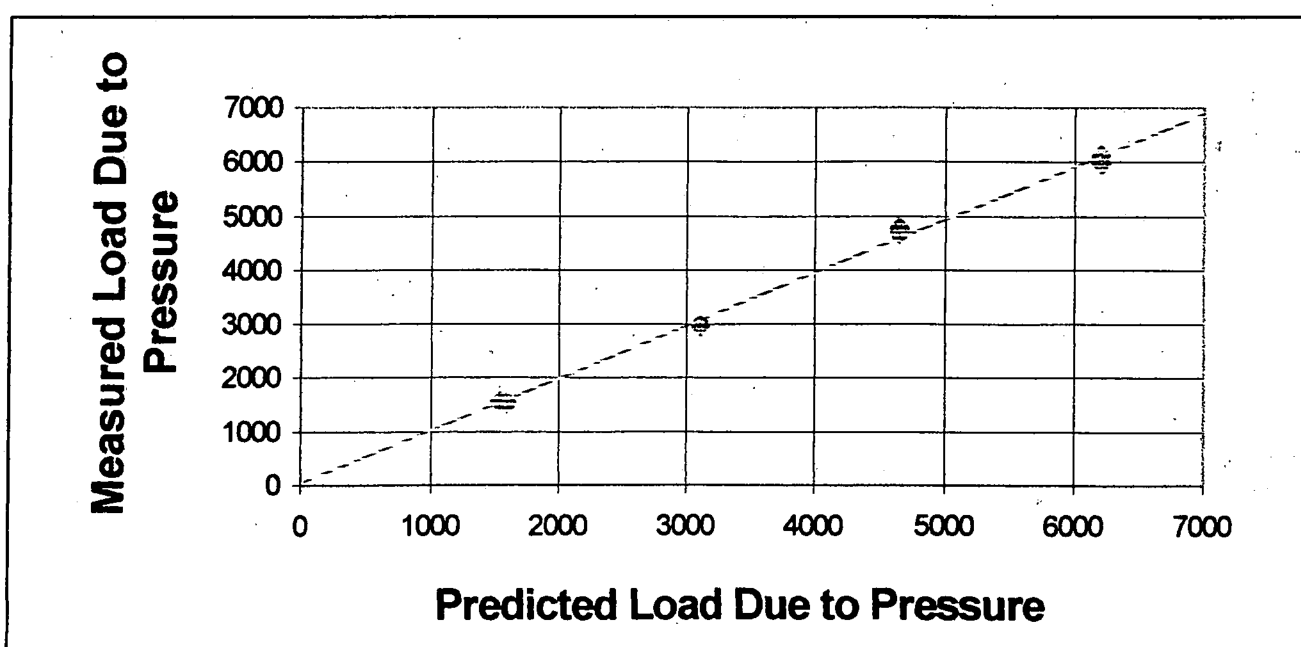
Predicted Versus Measured Portion of Pressure Thrust Due to Pressure Forces for Crane Valve



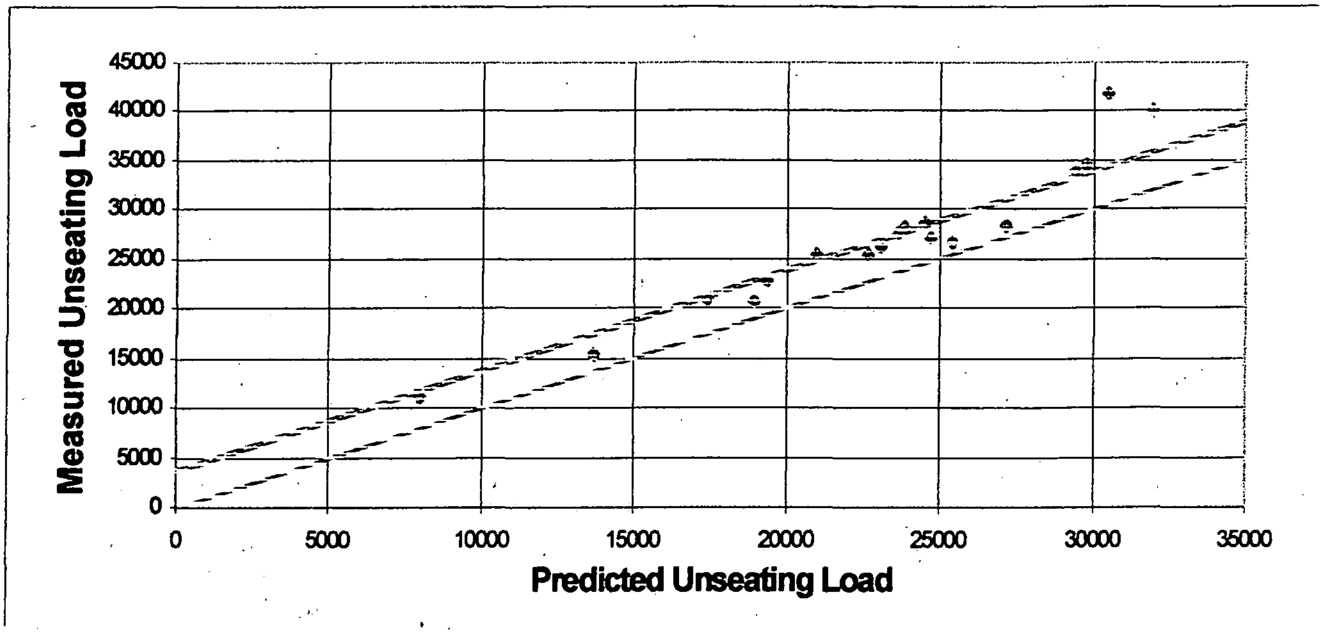
GRAPH 3
Predicted Unseating Thrust Versus Measured Pressure Locking Unseating Thrust for Westinghouse Valve



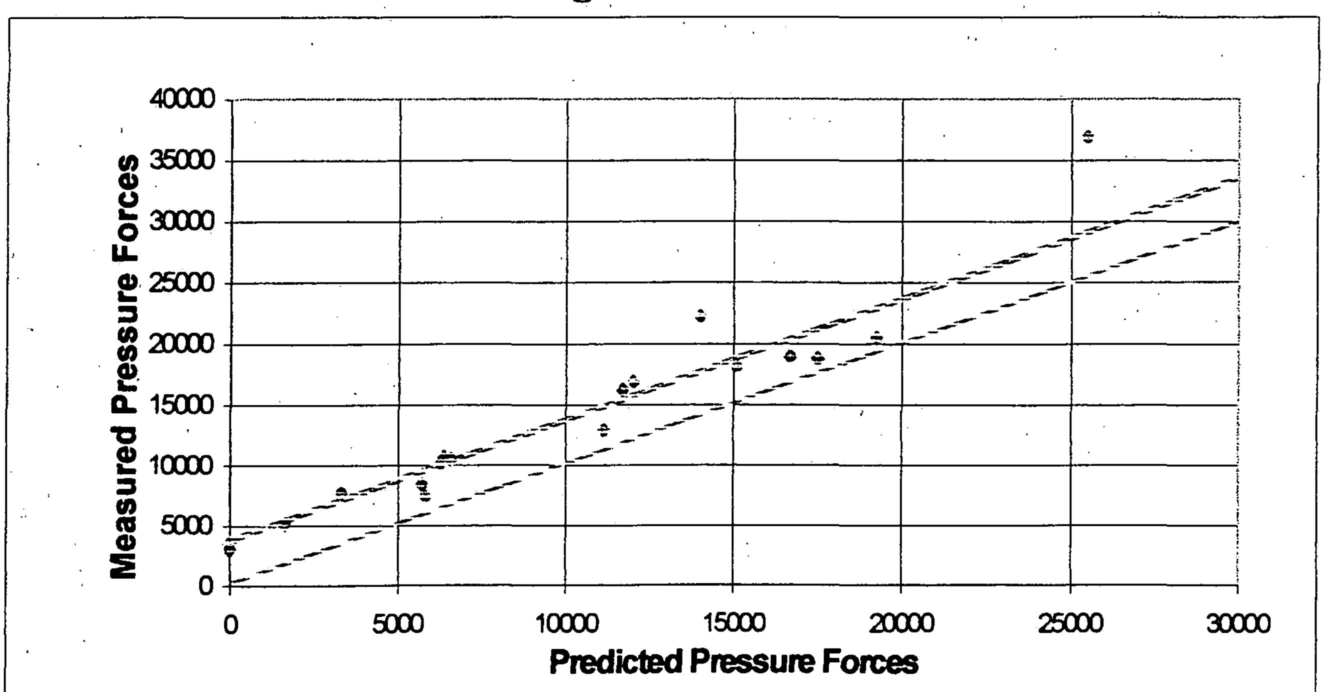
GRAPH 4
Predicted Versus Measured Portion of Unseating Thrust Due to Pressure Forces for Westinghouse Valve



GRAPH 5
Predicted Unseating Thrust Versus Measured Pressure Locking Unseating Thrust for Borg-Warner Valve



GRAPH 6
Predicted Versus Measured Portion of Unseating Thrust Due to Pressure Forces for Borg-Warner Valve



PRIMARY DIFFERENCES BETWEEN THE COMMONWEALTH EDISON PRESSURE LOCKING CALCULATION AND THE PRESSURE LOCKING CALCULATION METHOD PUBLISHED IN NUREG/CP-0146

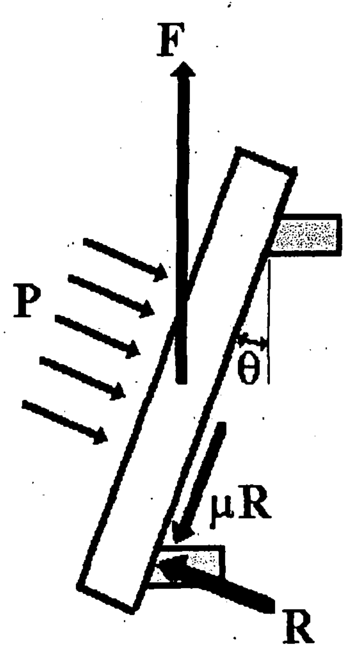
The ComEd methodology is based on calculating the contact load at the edge of the disk which results in an equal and opposite disk deflection to that caused by pressure trapped between the disks. The ComEd methodology differs in several ways from the methodology described in the Reference 4 NUREG.

- The NUREG Methodology ignores disk deflection due to hub elongation. This is non-conservative. For typical disk geometries, the expected impact of ignoring this effect is less than 5%.
- The NUREG Methodology is based on using Table 24 of Roark's equations for calculating forces in the disk. This table ignores disk deflection due to transverse shear stresses. Section 10.3 of Roark's Equations discusses the conditions under which deflection due to shear is negligible. For typical disk geometries the deflection due to shear is often not negligible. Table 25 of Roark's Equations provides the equations for calculating disk deflection due to shear. Ignoring deflection due to shear is non-conservative. For small valve sizes where the disk thickness to disk diameter aspect ratio is large (>0.3), ignoring shear may result in under predicting the disk to seat contact load by 10% or more.

The ComEd methodology treats the vertical pressure force on the disk separately from the pressure locking load caused by the increased contact load between the seat and disk. The NUREG methodology relies on use of the open disk factor for translating the increased seating contact force into an increased unseating load. The open disk factor is based on a free body diagram in which the disk hub is unloaded. This is not the case for pressure locking. The NUREG treatment of these two components to the pressure locking unseating load is non-conservative. This source of non-conservatism is generally much more significant than the other concerns mentioned above for the NUREG method and is the primary ComEd concern with the NUREG method.

The derivations on the following pages are provided to support the discussion above:

OPEN SEAT FACTOR DERIVATION (Opening a valve against a differential pressure)



F = Stem Force (tension)

P = Pressure Force

= DP x Seat Area

R = Seat Reaction Force

μR = Seat Friction Force

θ = Seat Angle

Disk Factor (VF) = F / P

(by definition)

Sum of forces in x-direction:

$$\sum F_x = P\cos\theta - R\cos\theta - \mu R\sin\theta \qquad \textbf{(31)}$$

$$R = P\frac{\cos\theta}{\cos\theta + \mu\sin\theta} \qquad \textbf{(32)}$$

Sum of forces in y-direction:

$$\sum F_y = F - P\sin\theta + R\sin\theta - \mu R\cos\theta \qquad \textbf{(33)}$$

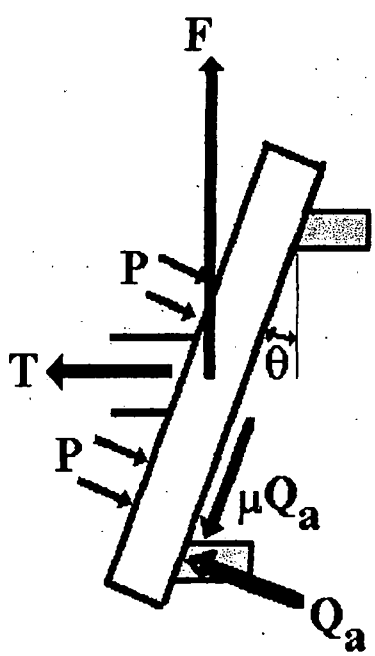
$$F = P\sin\theta - \left(P\frac{\cos\theta}{\cos\theta + \mu\sin\theta}\right)(\sin\theta - \mu\cos\theta)$$

$$F = P\left[\frac{\sin\theta(\cos\theta + \mu\sin\theta)}{\cos\theta + \mu\sin\theta} - \frac{\cos\theta(\sin\theta - \mu\cos\theta)}{\cos\theta + \mu\sin\theta}\right]$$

$$F/_P = \frac{\sin\theta\cos\theta + \mu\sin^2\theta - \cos\theta\sin\theta + \mu\cos^2\theta}{\cos\theta + \mu\sin\theta}$$

$$F/_P = \frac{\mu}{\cos\theta + \mu\sin\theta} \qquad \textbf{(34)}$$

PRESSURE LOCKING SUM OF FORCES



F = Stem Force (tension)

P = Pressure Force

= DP x Seat Area

Q_a = Seat Reaction Force

(calculated using Roark's)

μQ_a = Seat Friction Force

θ = Seat Angle

T = Disk Hub Tension

Note that the sum of the forces in the x-direction is different than for the seat factor case due to the hub tension force T. Consequently, the Q_a value is a typically a much lower portion of the P value under pressure locking than it is for the seat factor calculation.

(This is the benefit of using Roark's equations for calculating the seat load increase.) Therefore, the sum of the forces in the y-direction should be solved for directly from the free body diagram above, as follows:

$$\sum F_Y = F - \mu Q_a \cos\theta - P\sin\theta + Q_a \sin\theta \quad \textbf{(35)}$$

$$\therefore\ F = Q_a(\mu\cos\theta - \sin\theta) + P\sin\theta \quad \textbf{(36)}$$

The first term in the equation above is the pressure locking load term in the ComEd methodology. The second term in the equation above is the F_{vert} or reverse piston effect term in the ComEd methodology. The ComEd method adds these two terms to the static unseating load and then subtracts the stem rejection load to get the predicted unseating load under pressure locking conditions.

Rather than use these equations, the NUREG method applies the open seat factor to the Q_a value. Because of the relationship in equation 37 below, the NUREG method substantially under predicts the vertical pressure force portion of the required thrust.

$$Qa < P\cos\theta / (\cos\theta + \mu\sin\theta) \quad \textbf{(37)}$$

XVI. REFERENCES

1. Young, W. C., 1989, *Sixth Edition of Roark's Formulas for Stress and Strain*, McGraw-Hill Inc

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. Electric Power Research Institute, Nuclear Maintenance Applications Center, 1990, *Application Guide For Motor-Operated Valves in Nuclear Power Plants*, EPRI/NMAC Report NP-6660-D, March

4. Smith, D.E., 1994, "Calculation to Predict the Required Thrust to Open a Flexible Wedge Gate Valve Subjected to Pressure Locking", *Proceedings of the Workshop on Gate Valve Pressure Locking and Thermal Binding*, , NUREG/CP-0146, July, 1995

Attachment 2

ComEd Response to NRC Request for Additional Information on ComEd Pressure Locking Testing