White Paper
MOV-WP-156

AISI TYPE 1018 MOTOR PINION KEY TORQUE LIMITS AND MOTOR PINION KEY INSPECTION/INSTALLATION GUIDELINES

Draft Position

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1.0 PURPOSE

This position paper identifies GL 89-10 motor operated valve actuator configurations which could experience stresses at the motor pinion key equal to or in excess of the stresses that failed AISI type 1018 motor pinion keys during the KALSI Engineering MOV thrust extension testing.

In addition, this position paper provides guidelines on the inspection and installation of motor pinion keys.

2.0 BACKGROUND

The NRC has issued a number of Information Notices regarding actuators with motor pinion key failures. These failures appear to involve type 1018 keys in high speed high inertia configurations. Table 1 summarizes the concerns reported in these information notices (References 6, 7, 8, and 9). Inadequate information is provided to develop criteria that allow identification of the actuator configurations susceptible to the key failures.

Limitorque had previously indicated that only the actuators size SMB-3 or 4 were susceptible to failure of 1018 keys. KALSI testing, on the other hand, provides specific information on the torque and number of cycles which failed a 1018 key (Reference 2). KALSI testing resulted in two consecutive failures of a 1018 keys after 200 cycles at 104% of the Limitorque actuator torque rating.

3.0 POSITION

Although, the key failure mechanism experienced at KALSI generated stresses which failed two keys supplied by Limitorque, the estimated shear stress level (Section 4.3) is insufficient to cause failure. This implies that the keys experienced stress concentrations or combined loads (bending, impact, and shear).

The shear stress calculated provides a screening for identifying actuator configurations which can generate stresses, at the pinion key, of the same magnitude as those imposed during the KALSI test. This methodology accounts for changes in motor shaft diameter and key size between actuator sizes which affect the shear stress. However, does not account for dimensional differences which could affect stress concentrations, bending or impact loads.

3.1 AISI Motor Pinion Key Actuator Torque Limits

AISI type 1018 motor pinion keys are not appropriate for certain actuator configurations based on the Information Notice's summarized in Table 1 and the motor pinion key failures at KALSI. This White Paper establishes torque limits, as a percent of the existing torque rating for Limitorque actuators with AISI type 1018 motor pinion keys.

Attachments A and B provide torque limits as a percent of the Limitorque actuator torque rating by actuator and overall gear ratio for gate/butterfly valves and globe valves, respectively. The percent of MATR represents the percent of the Limitorque actuator torque rating which results in a shear stress equal to the shear stress experienced during the KALSI testing. For example, a SMB-4 actuator with a 27:1 OAR would experience stresses at the motor pinion key of the same order of magnitude experienced at KALSI with a actuator output greater than 35% of the Limitorque actuator torque rating.

Attachments A and B provide limits for 100, 200, and 2000 cycles. Attachment D provides a worksheet for calculating the limit for actuator configurations not included in Attachment A or B. Actuators set up to generate maximum torque's greater than the 2000 cycle limits are of concern and shall be dispositioned.

Two approaches are recommended:

1. Systematic replacement of all AISI type 1018 motor pinion keys. These replacements could be scheduled during routine maintenance and prioritized based on valve importance and the as left maximum torque. Valves for which the existing maximum torque is exceeding the 2000 cycle limits should be given a higher priority.

2. Identify and disposition all valves where the as-left maximum torque exceeds the 2000 cycle limits provided in Attachment A. Valves can be dispositioned as follows:

   a. Verification that 4140 key material is used.
   b. Justification of continued operation based on not exceeding a limited cycle limit, combined with scheduling replacement of existing key with 4140 key material, or
   c. Immediate replacement.

3.2 Actuator Motor Stall Events

Motor stall events can overtorque Limitorque actuators and are typically evaluated to verify the integrity of Limitorque actuators. The motor pinion key should be included in the post evaluation of motor stall events. The 100 cycle limits provided in Attachment A or B can be used as an initial screening of the torque capability of 1018 keys under stall conditions.

3.3 Motor Pinion Key Inspection/Installation Guidelines

Attachment C provides guidelines on the proper inspection and installation of motor pinion keys. These procedures or equivalent station procedures shall be used for the installation of replacement 4140 keys in Limitorque actuators. Use of this procedure will preclude failures similar to those experienced by several Nuclear stations. See References 10, 11, and 12.

4.0 JUSTIFICATION

4.1 Assumptions/Engineering Judgment

KALSI Engineering verified that the failed keys were AISI type 1018 via testing (Reference 4). KALSI confirmed that the failed motor pinion keys were interference fit.

Only shear stress is used in this position paper to simplify the derivations. This is judged acceptable based on the analysis being comparative. In other words this analysis identifies actuator configurations which could impose shear stress on the motor pinion key in access of the shear stress imposed on the keys during the KALSI testing.

4.2 Technical Inputs

1. Motor shaft and key specifications (see Table 2, Reference 1)
2. Actuator design information; actuator size, OAR, efficiency, Limitorque actuator torque rating (Reference 5).

3. KALSI thrust extension testing results (Reference 2):

Actuator	Motor, ft-lb	OAR	Max. Torque 2000 cycles	Stall Torque 5 Cycles
000	5	68.4	117%	127%
00	25	43.6	96%	174%
0	40	61.7	104%	201%
1	60	77.3	141%	194%

4.3 Methods

During the KALSI thrust extension testing, an AISI type 1018 key failed in a SMB-0 actuator after approximately 200 cycles at 104% of the Limitorque actuator torque rating. A 1018 key was reinstalled and failed again after approximately 200 cycles at 104% of the Limitorque actuator torque rating

These failures establish a definitive actuator torque and configuration which failed AISI type 1018 key material. Other failures summarized in Table 1 confirm that key failure is an issue and indicate that action is warranted.

The stress level at the motor pinion key is a function of the overall gear ratio, gear efficiency, key slot configuration, and key size. The shear stress experienced during the KALSI test is estimated below. This 200 cycle shear stress is then used as a baseline number to identify actuator configurations which impose higher stresses.

NOTE: KALSI testing was performed such that each cycle included applying the full test torque in both directions. This is representative for gate and globe valves which have a fairly substantial unseating torque. However, globe valves typically have small unseating torque's. Therefore, the critical stress for the configuration which failed at KALSI will be consider as 400 cycles for globe valves.

The shear stress level is reduced using Equation 4 to establish an acceptable baseline stress level for continuous duty (2000 cycles). This extrapolation is based on ASME Design Fatigue Curves for carbon, low alloy, and high tensile steels (Reference 3). Equation 4 also provides a basis for extrapolating to higher torque levels based on reduced cycles.

The following steps calculate the shear stress for the configuration and torque which failed the key at KALSI. This stress is then utilized to calculate a torque limit for each actuator configuration installed in CECo stations using Equation 5. The torque limits are converted to a percentage of the Limitorque actuator torque rating using Equation 6.

1. Calculate the torque at the motor using standard Limitorque equation;

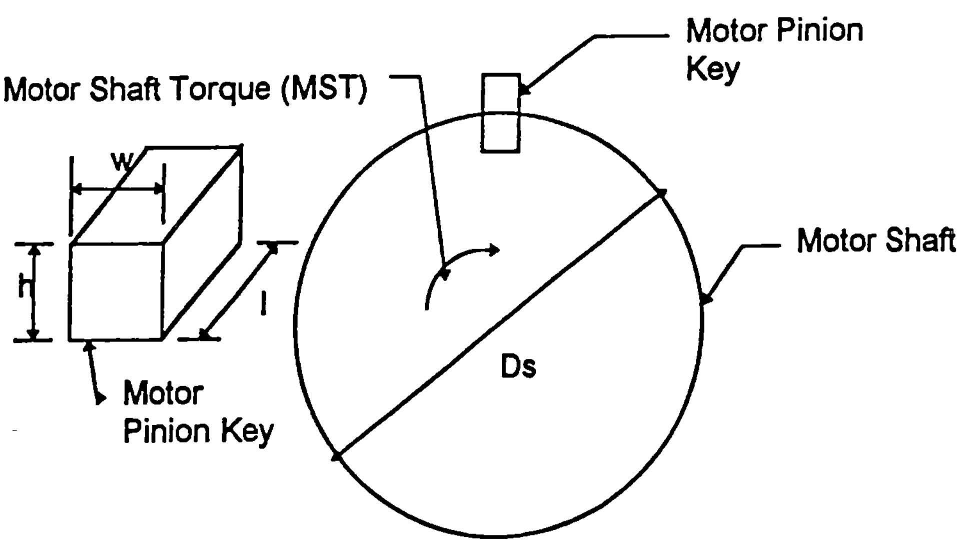
$$\text{Motor Shaft Torque} = \frac{\text{Actuator Torque}}{(\text{EFF})(\text{OAR})} \qquad \textbf{Eq. 1}$$

where.

ATR	=	actuator torque
MST	=	motor shaft torque
OAR	=	the actuator overall gear ratio
EFF	=	the gearing efficiency, for the purposes of this calculation the run efficiency will be utilized.

2. Establish an acceptable stress level for the key.

The following diagram illustrates the motor shaft, pinion key and pinion gear configuration with the associated forces:



The motor shaft applies a torque to the motor pinion gear through the motor pinion key. The stress imposed on the key is assumed to be shear only and is calculated as follows:

$$\sigma_S = \frac{MST}{\frac{D_s}{2} w l} \qquad \text{Eq. 2}$$

Substituting equation 1 into equation 2 and including a conversion factor of 12 for converting ft-lb to in-lb results in the following equation:

$$\sigma_s = \frac{24(AT)}{D_s w l (OAR)(EFF)} \qquad \text{Eq. 3}$$

The actuator which failed at KALSI was a SMB-0 with a 61.7 OAR (Reference 2). The motor pinion keys failed after approximately 200 cycles at 104% of the Limitorque actuator torque rating (500 ft-lb). The motor pinion key specifications for the SMB-0 are as follows per Table 2, Reference 1:

Ds=0.6245   w=0.125   l=0.875

The run efficiency per Reference 4 is 0.50. Substituting this data into Eq. 3 yields the critical shear stress for gate an butterfly valves at 200 cycles and a globe valve at 400 cycles:

$$\sigma_{\text{shear critical}} = \frac{24 \times 1.04 \times 500}{0.6245 \times 0.125 \times 0.875 \times 61.7 \times .50} \approx 6000 \text{ psi}$$

Reference 3 provides fatigue curves for carbon, low alloy, high tensile steel. The slope of the referenced fatigue curve is used in Tables 3 and 4 to extrapolate the critical stress to 100 and 2000 cycles.

Equation 3 can be rearranged to calculate the actuator torque limit based on the critical stress (from Table 3) for several different cycles ($AT_{\text{Limit, x cycles}}$):

$$(AT)_{\text{Limit, x cycles}} = \frac{\sigma_{\text{s, x cycles}} D_s w l (OAR)(EFF)}{24} \qquad \text{Eq. 4}$$

The actuator torque limit as a function of the Limitorque actuator torque rating is determined using the following equation:

$$\% MATR = \frac{AT_{\text{Limit, x cycles}}}{\text{MATR}} \qquad \textbf{Eq. 5}$$

where, MATR = Motor Limitorque actuator torque rating

4.4 Calculations

Equation 4 and 5 were utilized to calculate the actuator torque limit and % of the Limitorque actuator torque rating for AISI type 1018 key material with different expected cycles. Equation 4 has been applied to the Commonwealth Edison GL 89-10 population using Microsoft Access software. Motor shaft and key specifications were linked with the CECo design basis database (Reference 5) to provide the required inputs. The results for each actuator were divided by the Limitorque actuator torque rating to show the limits as a percent of this rating. The calculation was performed 100, 200, and 2000 cycles.

A worksheet is included in Attachment D for performing calculations at alternative cycles or for actuator configurations not provided in Attachment A or B. This fatigue analysis shall not be applied at cycles less than 100.

The calculations were performed using a Query in Microsoft Access.

6.0 REFERENCES

1. Motor shaft and key specifications provided by Quad Cities Station via a Memorandum of Telephone Conversation dated .
2. KALSI Limitorque Actuator Thrust Extension Report.
3. ASME 1992 Section III, Division 1, Appendices, Figure I-9.1
4. KALSI Engineering transmittal of "Failed Key Material Analysis," from Daniel Alvarez to John Kelly, dated June 16, 1994, Chron No. .
5. GL 89-10 MOV Database, CECo_X database, 1287202 bytes, date 4/5/94
6. Information Notice 81-08, Repetitive Failures of Limitorque Operator SMB-4 Motor-to-Shaft Key Failure.
7. Information Notice 81-08, Defective Motor Shaft Keys in Limitorque Motor Actuators.
8. Information Notice 90-37, Sheared Pinion Gear-to-Shaft Keys in Limitorque Motor Actuators
9. NETS TI 94-03, Potential Motor Key Failure in Limitorque Actuators at Clinton Station, 01/11/94.
10. Potential 10CFR Part 21 Issues Regarding Motor Pinion Gear and Key Movement", NETS letter from Y. Lassere to R.L. Bax, dated 3/24/93.
11. Limitorque Maintenance Update 92-2.
12. Motor Pinion Gear Key Inspection Criteria and Installation Guidelines for Limitorque Actuators", NETS letter form Y. Lassere to R.L. Bax, dated 3/16/93.

TABLE 1
LIMITORQUE MOTOR PINION KEY
NRC INFORMATION NOTICES

The following industry notices have been issued over the past several years regarding motor pinion key failure.

Document	Summary of Concerns
IEN 81-08, Reference (a)	Failure of grade AISI type 1018 motor pinion keys in actuators with 150 ft-lb or greater motors. Limitorque recommended replacement with 4140 keys.
IEN 88-44, Reference (b)	Motor pinion keys with resulphurized steel resulting in degraded structural properties and failure. Applies to actuators/motors (SMB-00 and up with 25 ft-lb motors or greater) and keys supplied prior to September 1983. Limitorque admitted to a lack of material control for keys prior to this date. It is not clear how Limitorque isolated the problem to specific actuators.
IEN 90-37, Reference (c)	Failure of grade AISI type 1018 motor pinion keys in SMB-0 actuators with 25 ft-lb motors on high speed (10 second closure) butterfly valves. Hardness check confirmed that the keys met ASTM-AISI type 1018 hardness standards.
TI-94-03, Reference (d)	Failure of AISI type 1018 motor pinion key in a SB-3 with a 80 ft-lb motor. Clinton attributes the failure to high impact/inertia loads caused during MOVAT's calibration (backseating valve into a load cell). Failure is believed to have occurred several years ago, however, the actuator continued to operate.

Limitorque Motor Shaft and Key Specifications

Table 2

Actuator Model Number*	Motor Shaft Diameter	Width	Height	Length
SMB-5	1.875	0.5	0.5	3.25
SMB-4	1.2495	0.25	0.25	1.625
SMB-3	1.062	0.25	0.25	1.625
SMB-2	0.937	0.25	0.25	1.375
SMB-1	0.7495	0.1875	0.1875	1.375
SMB-000	0.3745	0.0938	0.0938	0.6875
SMB-00	0.6245	0.125	0.125	0.875
SMB-0	0.6245	0.125	0.125	0.875

*SMB, SB, and SBD actuators of the same size have the same key dimensions. Only SMB actuators are listed.

Table 3
MOV Cycles Extrapolation
Gate and Butterfly Valves

MOV Cycles	Critical Stress, psi
100	7600
200	6000
500	4050
1000	3200
2000	2500

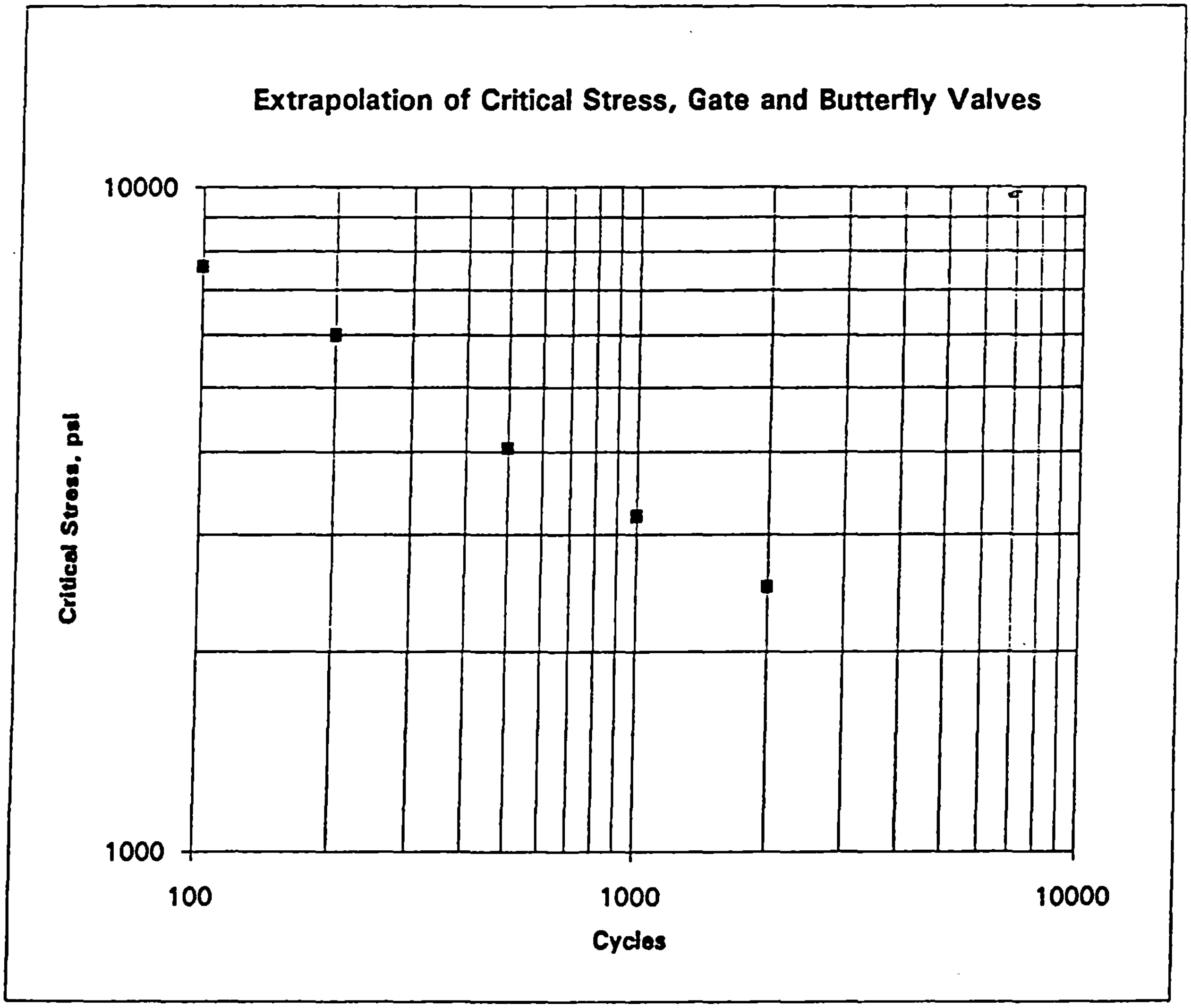
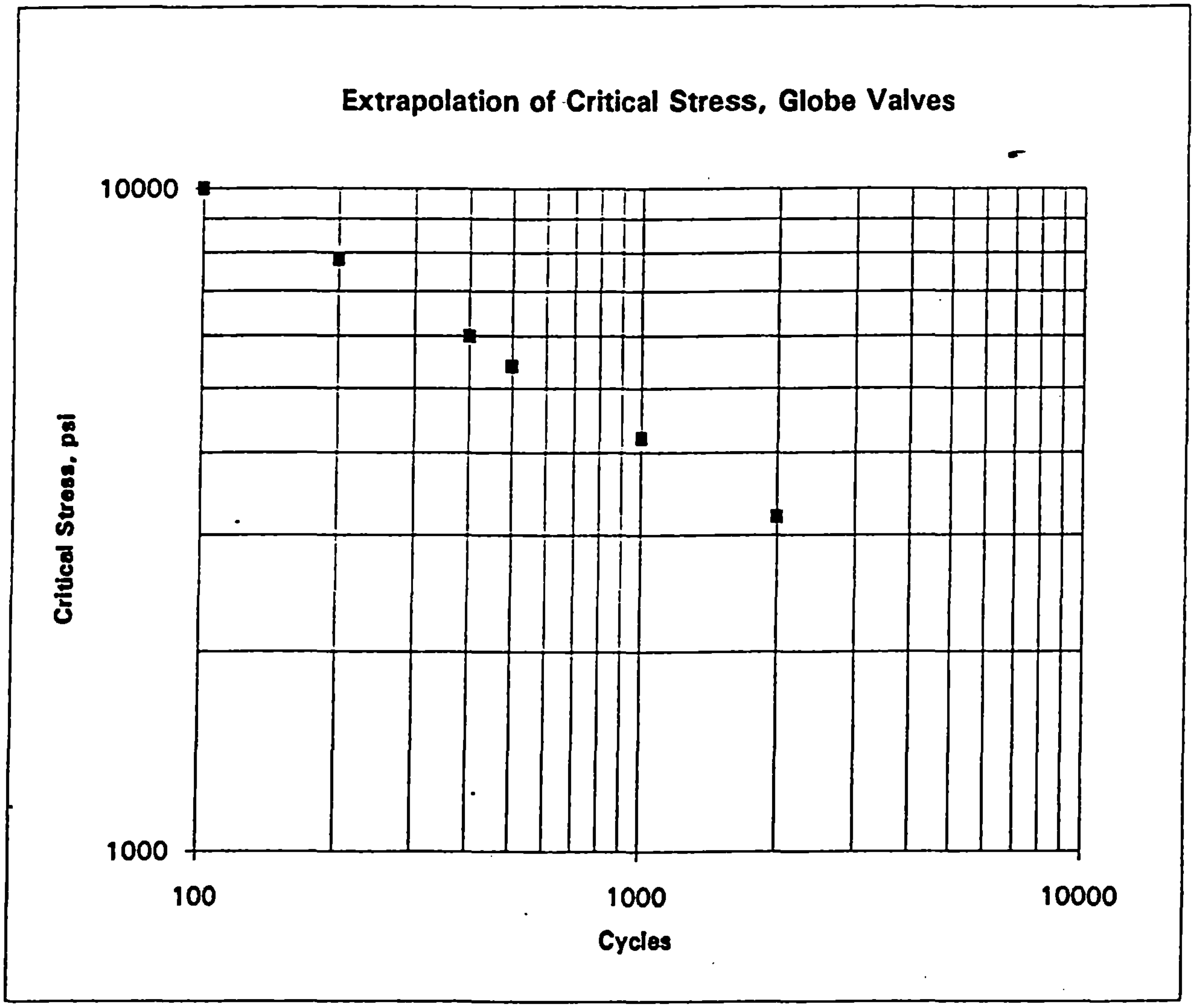


Table 4
MOV Cycles Extrapolation
Globe Valves

MOV Cycles	Critical Stress, psi
100	10000
200	7800
400	6000
500	5400
1000	4200
2000	3200



AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

ATE AND BUTTERFLY VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-5	214.7	50	1800	20000	518%	409%	170%
SMB-5	228.8	55	3600	20000	607%	479%	200%
SMB-5	230.2	55	3600	20000	611%	482%	201%
SMB-4	27	70	1900	7500	41%	32%	13%
SMB-4	43.2	70	3600	7500	65%	51%	21%
SMB-4	46.13	70	3600	7500	69%	55%	23%
SMB-4	48.4	70	3600	7500	73%	57%	24%
SMB-4	48.45	70	3600	7500	73%	57%	24%
SMB-4	51.8	55	1900	7500	61%	48%	20%
SMB-4	92.1	60	3600	7500	118%	94%	39%
SMB-4	92.12	55	1900	7500	109%	86%	36%
SMB-4	124.9	55	1800	7500	147%	116%	48%
SMB-4	131.8	50	1800	5100	208%	164%	68%
SMB-3	34.56	70	1900	4200	79%	62%	26%
SMB-3	34.6	70	1900	4200	79%	62%	26%
SMB-3	37.28	70	1900	4200	85%	67%	28%
SMB-3	46.7	60	3600	4200	91%	72%	30%
SMB-3	46.8	60	3600	4200	91%	72%	30%
SMB-3	61.5	55	3600	4200	110%	87%	36%
SMB-3	66	50	1900	4200	107%	85%	35%
SMB-3	70.9	55	3600	4200	127%	100%	42%
SMB-3	70.93	50	1900	4200	115%	91%	38%
SMB-3	76.26	55	3600	4200	136%	108%	45%
SMB-3	85.5	55	3600	4200	153%	121%	50%
SMB-3	88.56	50	1900	4200	144%	114%	47%
SMB-3	88.6	50	1800	4200	144%	114%	47%
SMB-3	106	45	1800	3300	197%	156%	65%
SMB-3	106.0	45	1900	3300	198%	156%	65%
SMB-3	123.1	45	1800	3300	229%	181%	75%
SMB-3	132.8	45	1900	3300	247%	195%	81%
SMB-3	132.8	45	1900	3300	247%	195%	81%
SMB-3	138.4	45	1800	2800	304%	240%	100%
SMB-3	186.4	45	1800	2800	409%	323%	135%

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AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

ATE AND BUTTERFLY VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-2	29.44	60	3600	1800	100%	79%	33%
SMB-2	31.2	60	3600	1800	106%	84%	35%
SMB-2	36.99	55	1800	1800	115%	91%	38%
SMB-2	49.5	50	1800	1800	140%	111%	46%
SMB-2	63	50	1800	1800	178%	141%	59%
SMB-2	63.26	50	1800	1800	179%	141%	59%
SMB-2	63.3	55	3600	1800	197%	156%	65%
SMB-2	67.4	50	1900	1800	191%	151%	63%
SMB-2	72	50	1800	1800	204%	161%	67%
SMB-2	72.01	50	1800	1800	204%	161%	67%
SMB-2	76.9	50	1800	1800	218%	172%	72%
SMB-2	76.99	50	1800	1800	218%	172%	72%
SMB-2	77	50	1710	1800	218%	172%	72%
SMB-2	90	45	1725	1250	330%	261%	109%
SMB-2	140	45	1900	1250	514%	406%	169%
SMB-1	24.2	65	3600	850	113%	89%	37%
SMB-1	25.65	65	1800	850	120%	95%	39%
SMB-1	40.15	60	3600	850	173%	137%	57%
SMB-1	40.2	60	3600	850	174%	137%	57%
SMB-1	42.5	50	1900	850	153%	121%	50%
SMB-1	45	55	3600	850	178%	141%	59%
SMB-1	53.41	50	1900	850	192%	152%	63%
SMB-1	63	50	1800	250	771%	609%	254%
SMB-1	63.9	55	3600	850	253%	200%	83%
SMB-1	63.92	50	1800	850	230%	182%	76%
SMB-1	72.4	50	1800	850	261%	206%	86%
SMB-1	72.42	55	3600	850	287%	226%	94%
SMB-1	77.25	50	1900	850	278%	220%	91%
SMB-1	77.3	50	1900	850	278%	220%	92%
SMB-1	82.6	50	1900	850	297%	235%	98%
SMB-1	88.4	50	1800	850	318%	251%	105%
SMB-1	92.4	45	1800	850	299%	236%	98%
SMB-1	103.7	45	1800	850	336%	265%	111%
SMB-1	109.9	12	1800	850	95%	75%	31%
SMB-1	124.1	45	1800	850	402%	317%	132%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

TE AND BUTTERFLY VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-000	27.99	60	1800	90	143%	113%	47%
SMB-000	30.6	60	1700	90	156%	123%	51%
SMB-000	33.5	50	1800	90	142%	112%	47%
SMB-000	36.5	50	1800	90	155%	122%	51%
SMB-000	37.3	50	1900	90	158%	125%	52%
SMB-000	40	50	1800	90	170%	134%	56%
SMB-000	43.75	50	1900	90	186%	147%	61%
SMB-000	43.8	50	1900	90	186%	147%	61%
SMB-000	45.5	50	1700	90	193%	153%	64%
SMB-000	47.85	50	1800	90	203%	161%	67%
SMB-000	47.9	50	1800	90	204%	161%	67%
SMB-000	52	50	1900	90	221%	174%	73%
SMB-000	54.4	50	1800	90	231%	182%	76%
SMB-000	57	50	1800	90	242%	191%	80%
SMB-000	57.1	50	1800	90	243%	192%	80%
SMB-000	62.5	50	1800	90	266%	210%	87%
SMB-000	68	50	1800	90	289%	228%	95%
SMB-000	68.4	50	1800	90	291%	229%	96%
SMB-000	70.7	50	1800	90	300%	237%	99%
SMB-000	75	50	1800	90	319%	252%	105%
SMB-000	82	50	1800	90	348%	275%	115%
SMB-000	82.1	50	1800	90	349%	275%	115%
SMB-000	90.5	50	1900	90	385%	304%	126%
SMB-000	90.6	50	1800	90	385%	304%	127%
SMB-000	100	50	1800	90	425%	335%	140%
SMB-000	136	45	1800	90	520%	411%	171%
SMB-00	23	50	1900	250	99%	79%	33%
SMB-00	26.3	50	1900	250	114%	90%	37%
SMB-00	28.2	50	1900	250	122%	96%	40%
SMB-00	31.9	60	3600	250	166%	131%	54%
SMB-00	32	50	1800	250	138%	109%	46%
SMB-00	34	60	3600	250	176%	139%	58%
SMB-00	34.1	50	1800	250	148%	116%	49%
SMB-00	36.2	50	1900	250	157%	124%	52%
SMB-00	38.6	50	1800	250	167%	132%	55%
SMB-00	46.4	50	1800	250	201%	158%	66%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

GATE AND BUTTERFLY VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-00	46.8	50	1800	250	202%	160%	67%
SMB-00	49	50	1700	250	212%	167%	70%
SMB-00	49.3	50	1800	250	213%	168%	70%
SMB-00	52.2	55	3600	250	248%	196%	82%
SMB-00	55.8	50	1800	250	241%	191%	79%
SMB-00	59.4	50	1900	250	257%	203%	85%
SMB-00	63	50	1700	250	273%	215%	90%
SMB-00	67.5	50	1800	250	292%	231%	96%
SMB-00	72	50	1800	250	311%	246%	102%
SMB-00	76.9	50	1750	250	333%	263%	109%
SMB-00	77	50	1900	250	333%	263%	110%
SMB-00	82	50	1900	250	355%	280%	117%
SMB-00	87.8	50	1800	250	380%	300%	125%
SMB-00	94	50	1800	250	407%	321%	134%
SMB-00	101.3	50	1800	250	438%	346%	144%
SMB-00	106.4	50	1800	250	460%	363%	151%
SMB-00	107	50	1800	250	463%	365%	152%
SMB-00	109	50	1800	250	472%	372%	155%
SMB-0	29.6	55	3600	500	70%	56%	23%
SMB-0	34.96	55	1900	500	83%	66%	27%
SMB-0	39.1	55	1800	500	93%	73%	31%
SMB-0	41.3	55	1800	500	98%	78%	32%
SMB-0	41.33	55	1800	500	98%	78%	32%
SMB-0	43.69	50	1800	500	95%	75%	31%
SMB-0	43.7	50	1800	500	95%	75%	31%
SMB-0	46.25	50	1800	500	100%	79%	33%
SMB-0	46.3	50	1800	500	100%	79%	33%
SMB-0	54	50	1700	500	117%	92%	38%
SMB-0	58.1	50	1800	500	126%	99%	41%
SMB-0	61.64	50	1800	500	133%	105%	44%
SMB-0	69.56	50	1800	500	150%	119%	49%
SMB-0	69.6	50	1800	500	151%	119%	50%
SMB-0	78.8	50	1900	500	170%	135%	56%
SMB-0	78.81	50	1900	500	170%	135%	56%
SMB-0	79	50	3600	500	171%	135%	56%
SMB-0	84.06	50	3600	500	182%	144%	60%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

GATE AND BUTTERFLY VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-0	89.8	50	1800	500	194%	153%	64%
SMB-0	96	50	1800	500	208%	164%	68%
SMB-0	96.2	50	1800	500	208%	164%	68%
SMB-0	109	45	1800	500	212%	168%	70%
SMB-0	123.5	45	1800	500	240%	190%	79%
SMB-0	140.8	45	1800	500	274%	216%	90%
SMB-0	150.8	45	1700	500	294%	232%	97%
SBD-4	107.5	55	1800	7500	127%	100%	42%
SBD-3	38.3	70	3600	4200	87%	69%	29%
SBD-3	38.34	60	3600	4200	75%	59%	25%
SBD-3	43.87	60	3600	4200	86%	68%	28%
SBD-2	31.15	60	3600	1800	106%	84%	35%
SBD-00	28.2	60	3600	250	146%	116%	48%
SBD-00	34.1	60	3600	250	177%	140%	58%
SBD-00	41	60	3600	250	213%	168%	70%
SBD-00	41.1	60	3600	250	213%	168%	70%
SB-4	124.9	55	1800	7500	147%	116%	48%
SB-3	61.5	55	3600	4200	110%	87%	36%
SB-2	27.79	60	3600	1800	94%	75%	31%
SB-2	29.4	60	3600	1800	100%	79%	33%
SB-2	72.01	55	3600	1800	224%	177%	74%
SB-2	150	45	1800	1250	551%	435%	181%
SB-00	23	60	3600	250	119%	94%	39%
SB-00	26.3	60	3600	250	137%	108%	45%
SB-00	28.2	60	3600	250	146%	116%	48%
SB-00	30	60	3600	250	156%	123%	51%
SB-00	31.9	60	3600	250	166%	131%	54%
SB-00	34.1	60	3600	250	177%	140%	58%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

ATE AND BUTTERFLY VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SB-00	36.2	60	3600	250	188%	148%	62%
SB-00	38.6	60	3600	250	200%	158%	66%
SB-00	41	60	3600	250	213%	168%	70%
SB-00	52.2	55	3600	250	248%	196%	82%
SB-00	55.8	55	3600	250	266%	210%	87%
SB-0	26.4	55	3400	500	63%	50%	21%
SB-0	36.2	55	3600	500	86%	68%	28%
SB-0	41.33	55	3600	500	98%	78%	32%
SB-0	54	50	3600	500	117%	92%	38%
SB-0	54.83	50	3600	500	119%	94%	39%
SB-0	65.45	50	3600	500	142%	112%	47%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

GLOBE VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-5	214.7	50	1800	20000	681%	532%	218%
SMB-5	228.8	55	3600	20000	799%	623%	256%
SMB-5	230.2	55	3600	20000	804%	627%	257%
SMB-4	27	70	1900	7500	53%	42%	17%
SMB-4	43.2	70	3600	7500	85%	67%	27%
SMB-4	46.13	70	3600	7500	91%	71%	29%
SMB-4	48.4	70	3600	7500	96%	75%	31%
SMB-4	48.45	70	3600	7500	96%	75%	31%
SMB-4	51.8	55	1900	7500	80%	63%	26%
SMB-4	92.1	60	3600	7500	156%	122%	50%
SMB-4	92.12	55	1900	7500	143%	111%	46%
SMB-4	124.9	55	1800	7500	194%	151%	62%
SMB-4	131.8	50	1800	5100	273%	213%	87%
SMB-3	34.56	70	1900	4200	104%	81%	33%
SMB-3	34.6	70	1900	4200	104%	81%	33%
SMB-3	37.28	70	1900	4200	112%	87%	36%
SMB-3	46.7	60	3600	4200	120%	94%	38%
SMB-3	46.8	60	3600	4200	120%	94%	38%
SMB-3	61.5	55	3600	4200	145%	113%	46%
SMB-3	66	50	1900	4200	141%	110%	45%
SMB-3	70.9	55	3600	4200	167%	130%	53%
SMB-3	70.93	50	1900	4200	152%	118%	49%
SMB-3	76.26	55	3600	4200	180%	140%	57%
SMB-3	85.5	55	3600	4200	201%	157%	64%
SMB-3	88.56	50	1900	4200	190%	148%	61%
SMB-3	88.6	50	1800	4200	190%	148%	61%
SMB-3	106	45	1800	3300	260%	203%	83%
SMB-3	106.0	45	1900	3300	260%	203%	83%
SMB-3	123.1	45	1800	3300	302%	235%	97%
SMB-3	132.8	45	1900	3300	326%	254%	104%
SMB-3	132.8	45	1900	3300	326%	254%	104%
SMB-3	138.4	45	1800	2800	400%	312%	128%

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AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

GLOBE VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-3	186.4	45	1800	2800	539%	420%	172%
SMB-2	29.44	60	3600	1800	132%	103%	42%
SMB-2	31.2	60	3600	1800	140%	109%	45%
SMB-2	36.99	55	1800	1800	152%	118%	49%
SMB-2	49.5	50	1800	1800	185%	144%	59%
SMB-2	63	50	1800	1800	235%	183%	75%
SMB-2	63.26	50	1800	1800	236%	184%	75%
SMB-2	63.3	55	3600	1800	260%	202%	83%
SMB-2	67.4	50	1900	1800	251%	196%	80%
SMB-2	72	50	1800	1800	268%	209%	86%
SMB-2	72.01	50	1800	1800	268%	209%	86%
SMB-2	76.9	50	1800	1800	287%	224%	92%
SMB-2	76.99	50	1800	1800	287%	224%	92%
SMB-2	77	50	1710	1800	287%	224%	92%
SMB-2	90	45	1725	1250	435%	339%	139%
SMB-2	140	45	1900	1250	676%	528%	216%
SMB-1	24.2	65	3600	850	149%	116%	48%
SMB-1	25.65	65	1800	850	158%	123%	51%
SMB-1	40.15	60	3600	850	228%	178%	73%
SMB-1	40.2	60	3600	850	228%	178%	73%
SMB-1	42.5	50	1900	850	201%	157%	64%
SMB-1	45	55	3600	850	234%	183%	75%
SMB-1	53.41	50	1900	850	253%	197%	81%
SMB-1	63	50	1800	250	1014%	791%	325%
SMB-1	63.9	55	3600	850	333%	260%	107%
SMB-1	63.92	50	1800	850	303%	236%	97%
SMB-1	72.4	50	1800	850	343%	267%	110%
SMB-1	72.42	55	3600	850	377%	294%	121%
SMB-1	77.25	50	1900	850	366%	285%	117%
SMB-1	77.3	50	1900	850	366%	286%	117%
SMB-1	82.6	50	1900	850	391%	305%	125%
SMB-1	88.4	50	1800	850	419%	327%	134%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

GLOBE VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-1	92.4	45	1800	850	394%	307%	126%
SMB-1	103.7	45	1800	850	442%	345%	141%
SMB-1	109.9	12	1800	850	125%	97%	40%
SMB-1	124.1	45	1800	850	529%	413%	169%
SMB-000	27.99	60	1800	90	188%	146%	60%
SMB-000	30.6	60	1700	90	205%	160%	66%
SMB-000	33.5	50	1800	90	187%	146%	60%
SMB-000	36.5	50	1800	90	204%	159%	65%
SMB-000	37.3	50	1900	90	209%	163%	67%
SMB-000	40	50	1800	90	224%	174%	72%
SMB-000	43.75	50	1900	90	245%	191%	78%
SMB-000	43.8	50	1900	90	245%	191%	78%
SMB-000	45.5	50	1700	90	254%	198%	81%
SMB-000	47.85	50	1800	90	268%	209%	86%
SMB-000	47.9	50	1800	90	268%	209%	86%
SMB-000	52	50	1900	90	291%	227%	93%
SMB-000	54.4	50	1800	90	304%	237%	97%
SMB-000	57	50	1800	90	319%	249%	102%
SMB-000	57.1	50	1800	90	319%	249%	102%
SMB-000	62.5	50	1800	90	349%	273%	112%
SMB-000	68	50	1800	90	380%	297%	122%
SMB-000	68.4	50	1800	90	382%	298%	122%
SMB-000	70.7	50	1800	90	395%	308%	126%
SMB-000	75	50	1800	90	419%	327%	134%
SMB-000	82	50	1800	90	458%	358%	147%
SMB-000	82.1	50	1800	90	459%	358%	147%
SMB-000	90.5	50	1900	90	506%	395%	162%
SMB-000	90.6	50	1800	90	506%	395%	162%
SMB-000	100	50	1800	90	559%	436%	179%
SMB-000	136	45	1800	90	684%	534%	219%
SMB-00	23	50	1900	250	131%	102%	42%
SMB-00	26.3	50	1900	250	150%	117%	48%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

GLOBE VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-00	28.2	50	1900	250	161%	125%	51%
SMB-00	31.9	60	3600	250	218%	170%	70%
SMB-00	32	50	1800	250	182%	142%	58%
SMB-00	34	60	3600	250	232%	181%	74%
SMB-00	34.1	50	1800	250	194%	151%	62%
SMB-00	36.2	50	1900	250	206%	161%	66%
SMB-00	38.6	50	1800	250	220%	171%	70%
SMB-00	46.4	50	1800	250	264%	206%	85%
SMB-00	46.8	50	1800	250	266%	208%	85%
SMB-00	49	50	1700	250	279%	218%	89%
SMB-00	49.3	50	1800	250	281%	219%	90%
SMB-00	52.2	55	3600	250	327%	255%	105%
SMB-00	55.8	50	1800	250	318%	248%	102%
SMB-00	59.4	50	1900	250	338%	264%	108%
SMB-00	63	50	1700	250	359%	280%	115%
SMB-00	67.5	50	1800	250	384%	300%	123%
SMB-00	72	50	1800	250	410%	320%	131%
SMB-00	76.9	50	1750	250	438%	341%	140%
SMB-00	77	50	1900	250	438%	342%	140%
SMB-00	82	50	1900	250	467%	364%	149%
SMB-00	87.8	50	1800	250	500%	390%	160%
SMB-00	94	50	1800	250	535%	417%	171%
SMB-00	101.3	50	1800	250	577%	450%	185%
SMB-00	106.4	50	1800	250	606%	472%	194%
SMB-00	107	50	1800	250	609%	475%	195%
SMB-00	109	50	1800	250	620%	484%	199%
SMB-0	29.6	55	3600	500	93%	72%	30%
SMB-0	34.96	55	1900	500	109%	85%	35%
SMB-0	39.1	55	1800	500	122%	95%	39%
SMB-0	41.3	55	1800	500	129%	101%	41%
SMB-0	41.33	55	1800	500	129%	101%	41%
SMB-0	43.69	50	1800	500	124%	97%	40%
SMB-0	43.7	50	1800	500	124%	97%	40%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

LOBE VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SMB-0	46.25	50	1800	500	132%	103%	42%
SMB-0	46.3	50	1800	500	132%	103%	42%
SMB-0	54	50	1700	500	154%	120%	49%
SMB-0	58.1	50	1800	500	165%	129%	53%
SMB-0	61.64	50	1800	500	175%	137%	56%
SMB-0	69.56	50	1800	500	198%	154%	63%
SMB-0	69.6	50	1800	500	198%	155%	63%
SMB-0	78.8	50	1900	500	224%	175%	72%
SMB-0	78.81	50	1900	500	224%	175%	72%
SMB-0	79	50	3600	500	225%	175%	72%
SMB-0	84.06	50	3600	500	239%	187%	77%
SMB-0	89.8	50	1800	500	256%	199%	82%
SMB-0	96	50	1800	500	273%	213%	87%
SMB-0	96.2	50	1800	500	274%	214%	88%
SMB-0	109	45	1800	500	279%	218%	89%
SMB-0	123.5	45	1800	500	316%	247%	101%
SMB-0	140.8	45	1800	500	361%	281%	115%
SMB-0	150.8	45	1700	500	386%	301%	124%
SBD-4	107.5	55	1800	7500	167%	130%	53%
SBD-3	38.3	70	3600	4200	115%	90%	37%
SBD-3	38.34	60	3600	4200	98%	77%	32%
SBD-3	43.87	60	3600	4200	113%	88%	36%
SBD-2	31.15	60	3600	1800	139%	109%	45%
SBD-00	28.2	60	3600	250	193%	150%	62%
SBD-00	34.1	60	3600	250	233%	182%	75%
SBD-00	41	60	3600	250	280%	218%	90%
SBD-00	41.1	60	3600	250	281%	219%	90%
SB-4	124.9	55	1800	7500	194%	151%	62%

AISI Type 1018 Motor Pinion Key Strength Limits as a Percent of the Actuator Torque Rating

GLOBE VALVES

ACTUATOR MODEL NO	OAR	RUN EFF	MOTOR RPM	MATR	%MATR, 100 Cycle Limit	%MATR, 200 Cycle Limit	%MATR, 2000 Cycle Limit
SB-3	61.5	55	3600	4200	145%	113%	46%
SB-2	27.79	60	3600	1800	124%	97%	40%
SB-2	29.4	60	3600	1800	132%	103%	42%
SB-2	72.01	55	3600	1800	295%	230%	94%
SB-2	150	45	1800	1250	725%	565%	232%
SB-00	23	60	3600	250	157%	123%	50%
SB-00	26.3	60	3600	250	180%	140%	57%
SB-00	28.2	60	3600	250	193%	150%	62%
SB-00	30	60	3600	250	205%	160%	66%
SB-00	31.9	60	3600	250	218%	170%	70%
SB-00	34.1	60	3600	250	233%	182%	75%
SB-00	36.2	60	3600	250	247%	193%	79%
SB-00	38.6	60	3600	250	264%	206%	84%
SB-00	41	60	3600	250	280%	218%	90%
SB-00	52.2	55	3600	250	327%	255%	105%
SB-00	55.8	55	3600	250	349%	273%	112%
SB-0	26.4	55	3400	500	83%	64%	26%
SB-0	36.2	55	3600	500	113%	88%	36%
SB-0	41.33	55	3600	500	129%	101%	41%
SB-0	54	50	3600	500	154%	120%	49%
SB-0	54.83	50	3600	500	156%	122%	50%
SB-0	65.45	50	3600	500	186%	145%	60%

ATTACHMENT C

MOTOR PINION KEY INSPECTION/INSTALLATION GUIDELINES

Guideline for Installing Motor Pinion Gear to Motor Shaft

NOTE

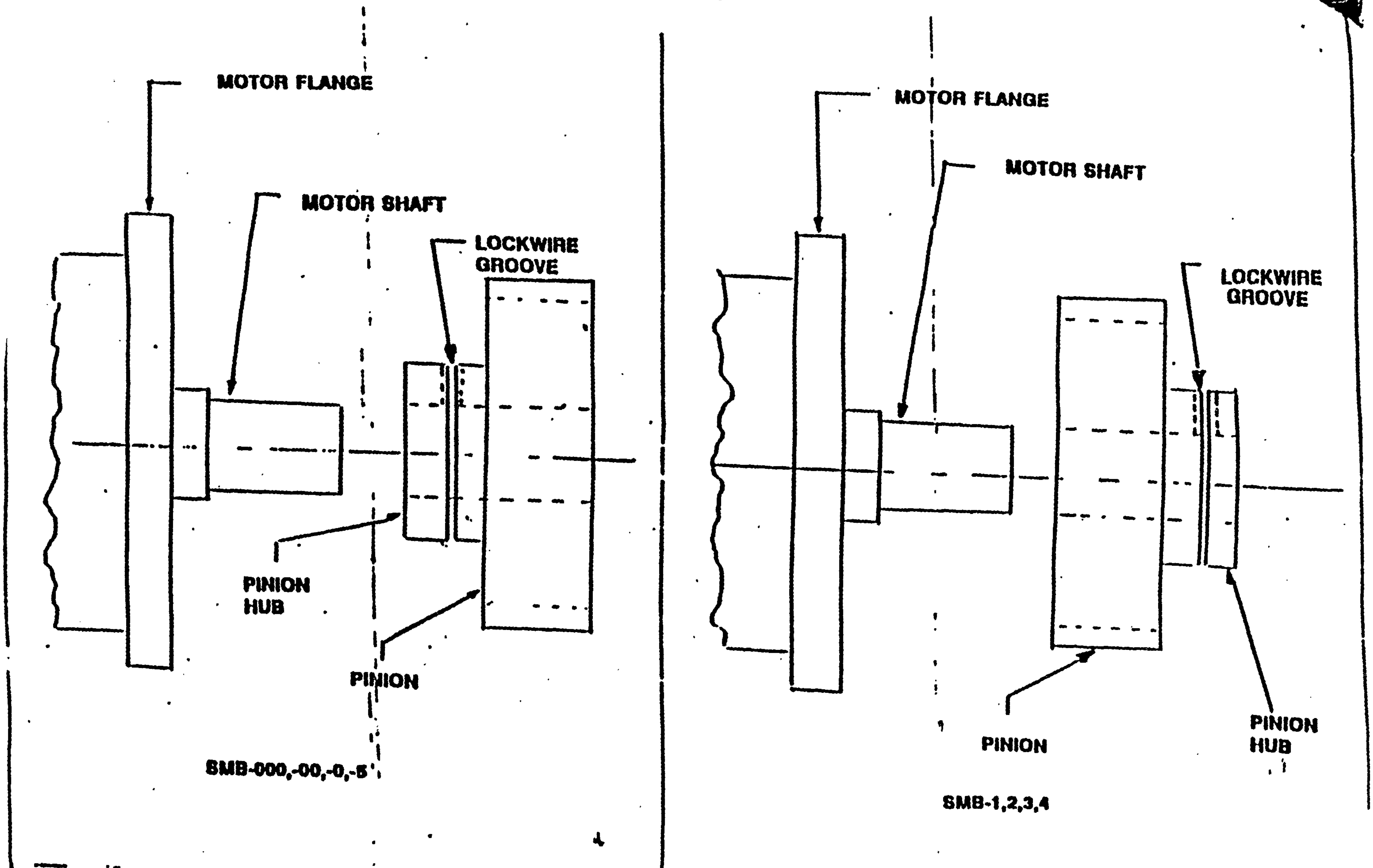
In step f) the final installation of the motor pinion, shaft key, and the set screw (with Loctite) should not be performed until the set screw dimple depth (on the motor shaft) is verified to ensure proper installation of the set screw.

f) Install motor pinion, shaft key, and set screw as follows:

- Install the motor pinion and the shaft key on the motor shaft verifying proper orientation (see figure 2 for proper orientation)
- Install set screw and tighten set screw enough to score the surface on the motor shaft
- Loosen set screw and remove pinion and shaft key from the motor shaft
- Using a drill bit (size equal to major diameter of set screw), spot drill at the scored mark on the motor shaft (center punch the score mark first may facilitate the drilling). At a minimum, drill until the full tip of the bit is below the outer diameter of the shaft
- Install the motor pinion and shaft key on the motor shaft verifying proper orientation
- Verify the depth of the dimple on the motor shaft such that with the set screw installed, the top of the set screw is flush with the bottom of the lockwire groove on the motor pinion (or the motor pinion if no lock wire groove is present). It may be necessary to remove the pinion and redrill to either deepen the dimple or enlarge the dimple to accept the full diameter of the set screw).
- Remove the set screw, motor pinion, and shaft key. Apply Loctite 242 to the set screw, threaded hole on the motor pinion, and shaft key and key slot (NOTE: if excessive clearance exists between the motor pinion and the motor shaft, Loctite 242 should also be applied to the shaft to reduce the clearance (see step I b) recommendation)).
- Install the motor pinion and the shaft key on the motor shaft verifying proper orientation. Install the set screw and tighten securely. Wipe off excess Loctite from the motor shaft.
- The head of the set screw should be flush with the bottom of the lockwire groove on the motor pinion gear (or the motor pinion if no lockwire groove is present), and its end should be tightly set in the drilled dimple on the motor shaft.

g) Lockwire the set screw in place. Ensure the twisted part of the wire is not directly above the set screw.
(with the exception of SMB-000, Lockwire should be used on all actuator supplied with lockwire groove)

Figure 2



LIMITORQUE CORPORATION

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ATTACHMENT D

AISI TYPE 1018 MOTOR PINION KEY TORQUE LIMIT WORKSHEET

Using the following equation, actuator design information, Table 1, and Table 2 or 3, the AISI type 1018 motor pinion key limit can be calculated for any actuator configuration and cycle limit as follows:

$$(AT)_{\text{Limit, x cycles}} = \frac{\sigma_{\text{s, x cycles}} D_s w l (OAR)(EFF)}{24} \qquad \textbf{Eq. 4}$$

where,

$\sigma_{\text{s, x cycles}}$: critical stress at x cycles (_______) from Table 2 (gate or butterfly valves or Table 3 (globe valves).

D_s,w,l: motor shaft diameter, pinion key width, and pinion key length from Table 1.

OAR, EFF: actuator overall gear ratio and run efficiency from the MOV Rising Stem Datasheet.

Substituting this information into Equation 4:

$$(AT)_{\text{Limit, x cycles}} = \frac{(\_\_\_\ psi)(\_\_\ in)(\_\_\ in)(\_\_\ in)(\_\_)(\_\_)}{24}$$

$$(AT)_{\text{Limit, \_\_\_\_}} = \_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\ \text{ft} - \text{lb}$$

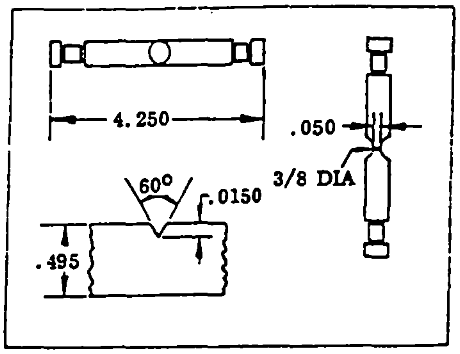
MPR ASSOCIATES INC. ENGINEERS

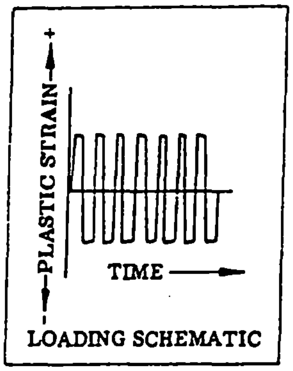
Attachment B to
MPR Enclosure Dated
November 4, 1994

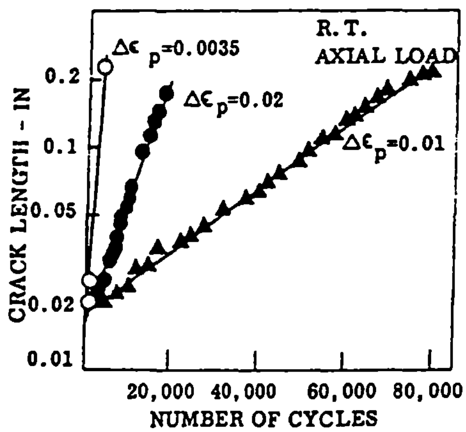
Low Cycle Fatigue Data for AISI Type 1018 Material from:

Metals and Ceramics Information Center, Battelle Columbus Laboratories, "Structural Alloys Handbook," Volume 1, 1988 Edition.

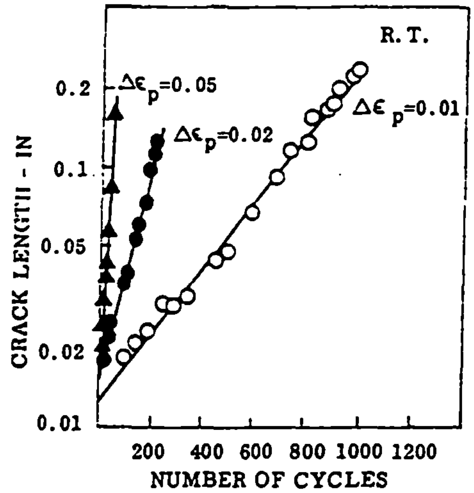
FATIGUE





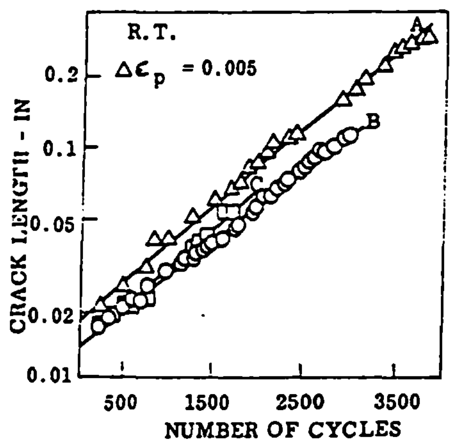


EFFECT OF PLASTIC STRAIN AMOUNTS ON LOW CYCLE FATIGUE PROPERTIES OF 1018 STEEL (2)



CURVES A, B AND C INDICATE VARIOUS EXTENSONOMETER POSITIONS ALONG THE SPECIMEN

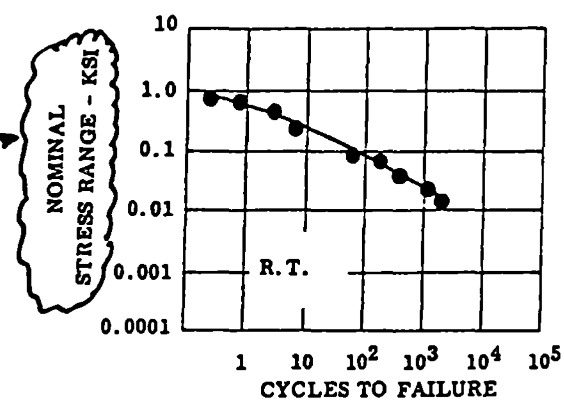
ϵ_p - PLASTIC STRAIN



EFFECT OF MEASUREMENT TECHNIQUE ON LOW CYCLE FATIGUE RESULTS OF 1018 STEEL (2)

Should be Strain Range

ROV



LOW CYCLE AXIAL FATIGUE LIFE OF 1018 STEEL (3)

ATTACHMENT

3

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MPR
ASSOCIATES INC.
ENGINEERS

November 7, 1994

Mr. Paul Dietz
Commonwealth Edison Company
1400 Opus Place
Downers Grove, IL 60515

Subject: Review of CECo WP-125, "Installed Motor Capability Evaluation", Revision 1, Draft 7/11/94.

Dear Mr. Dietz:

Enclosed is a draft report documenting our review of the subject CECo position paper. We have discussed our comments by phone with Mr. Ivo Garza. Our recommendations are summarized below:

- Some aspects of the position need to be clarified to ensure that there is no potential ambiguity in interpretation. The detailed comments in this regard are discussed in the enclosure.

- The technical justification for the position needs to be improved through the use of test data. The specific areas where data and comparisons are needed are discussed in the enclosure. It appears the use of motor test data obtained by Commonwealth Edison and actuator test data obtained by Texas Utilities could provide a considerable amount of the needed information.

As discussed with Mr. Garza, we understand that WP-125 is being significantly revised based on recent MOV test results and our comments. The enclosure can be used to assist in the revision process.

Please call if you have any questions or comments on the enclosed report.

Sincerely,

E. C. Wenzinger, Jr.

E. C. Wenzinger, Jr.

Enclosure

cc: Ivo Garza, CECo

The stated position needs to be clarified in the following areas.

- The purpose should be revised to clearly limit the application to operability determinations for marginal MOVs which do not satisfy the original Limitorque design equations.

- The position should be revised to clearly define the approaches for application of the position to torque controlled MOVs and to limit controlled MOVs.

- The conditions for use of an Application Factor of 1.0 need to be stated more clearly.

- Although not presently mentioned in the white paper, we understand that the author's intent is to restrict the use of running efficiency in the opening direction to when the rated motor starting torque is used. If the adjusted torque is used for the opening direction, the interpolated efficiency must be applied. These clarifications need to be implemented.

- The position should be revised to note that the "Thrust Effect" (identified in a recent Texas Utilities paper) is accounted for by the use of a stem factor determined by MOV test results based on spring pack displacement measurements.

- The position on stem factor needs to be revised to include references to other White Papers or documents which define the methods for determining stem factors.

Based on another utility's actuator testing, evidence exists that the original Limitorque design calculation does not contain significant margin for some specific actuator applications. Testing performed by Texas Utilities determined that the presence of stem thrust significantly reduces the efficiency of motor torque to actuator output torque conversion as compared to that obtained from a torque stand test. The test results indicate that this effect was not considered in the original Limitorque design calculation. Additionally, recent Limitorque allowances (reductions in perceived original design calculation conservatism) for evaluating in-service valves are called into question by this utility's test results. The position justification provided in the white paper takes credit for the recent Limitorque allowances and defines additional reductions in perceived design margins. In light of Texas Utilities test results, such a position is difficult to defend without supporting actuator testing.

Based on the Texas Utilities test results, the conservatism in some portions of the Limitorque design calculation can be offset by non-conservatism in the same calculation for some specific valve applications. If the White Paper 125 approach is pursued, the potentially offsetting effects need to be addressed. For example, the use of a higher torque in conjunction with a less conservative application factor appears to be removing a conservatism that is offsetting the lack of use of a degraded voltage factor (in the voltage range above 90% rated voltage).

Some specific valve/actuator applications may have significant margin if designed using the Limitorque original design equation. However, the information currently available indicates that actuator testing is necessary to determine the margin available for specific valve applications. Ideally, representative actuators could be tested to stall conditions with a representative stem thrust load applied. Separate torque stand and in situ testing similar to that performed by Texas Utilities could also be performed. It may be possible to use at least some of the Texas Utilities data as part of the validation of White Paper 125.

REVIEW APPROACH

The MPR review approach is as follows:

- Review the white paper purpose to ensure that it is unambiguously and completely stated.
- Review the statement of position to ensure that it:
  - addresses the purpose;
  - is unambiguous and complete; and
  - includes all appropriate restrictions and limitations with regard to its use.
- Review the technical justification to ensure that it:
  - logically presents a case which defends the stated position;
  - makes proper technical use of the theory and data which are referenced;
  - adheres to appropriate requirements of codes, standards and regulations which are referenced;
  - provides a sufficient technical basis for the stated position; and
  - is written in a way to provide a convincing justification.

As part of the review of the justification, comparisons to other data or approaches (e.g., EPRI data or models) are made, which may not have been considered in writing the justification.

RESULTS OF REVIEW INCLUDING DETAILED COMMENTS

Purpose of WP-125

The stated purpose of White Paper 125 is as follows:

> "The purpose of this paper is to address how and when an AC or DC motor operated valve (MOV) motor capability can be evaluated using full motor torque and other motor variables. This position is interim pending resolution of Limitorque motor rating and capability, and actuator efficiency questions. This position is to be applied to present and future capability calculations are to be used to support testing and to establish an interim torque switch setting until a modification to enhance margin can be completed.
>
> The capability calculations discussed in this white paper are for establishing testing thrust windows as opposed to determining the maximum allowable torque switch settings based on structural limits.
>
> Full motor capability calculations can be used to establish a testing thrust window which considers all the phenomena of concern identified in Limitorque Maintenance Update 89-1. That is, degraded voltage, proper use of diagnostic test equipment, degraded stem to stem nut coefficients of friction, and inertia effects are considered. In addition, temperature effects on the available motor torque shall be considered. The established motor capability determined through the use of full motor capability calculations preserves the overall margin for a MOV until such time that final modifications can be implemented."

Specific comments are as follows:

- The stated purpose should be clarified to state that this interim position is intended only for use to justify the operability of specific MOV applications which have less than the original design margin between required stem thrust and the operator thrust capability as determined by the standard Limitorque equations. As written, the position may be interpreted to be generally applicable for defining the thrust window for testing of all MOVs.

- It is stated that the full motor capability calculations will be used to establish testing thrust windows. It should be clarified that this white paper does not address consideration of other potentially limiting factors such as the valve thrust and torque limits.

- It is stated that "The established motor capability determined through the use of full motor capability calculations preserves the overall margin for a MOV ..." This position involves justification of specific MOV applications which impose on the original design margin defined by the Limitorque original design equations. The term "overall margin" should be carefully defined or the statement should be deleted. We understand that the intent of this statement is to indicate that a positive margin is maintained rather than to quantify the margin.

Technical Position of WP-125

The stated position of White Paper 125 is as follows:

"When calculating full motor capability, the following changes in the variables of the MOV motor capability calculation sizing equation can be considered: Note, use only the changes required to calculate an adequate thrust window to support testing.

- The application factor, which is typically set to 0.90 or less, is taken to be 0.95. If temperature compensation and an under voltage factor of less than one is included, the application factor becomes 1.0 (Ref. 8).
- The voltage supplied to the motor is assumed to be *degraded voltage*.
- The *start torque* of the motor is replaced by the start torque plus 75 percent of the difference between start torque and stall torque. See table 1 for AC and Porter Peerless DC motors.
- The *unit efficiency* to be used is:
- run efficiency if the actual test data indicates that the motor reaches full speed (steady state current indicates motor is a full speed) prior to disc movement in the opening direction.
- Use interpolation between running efficiency at nominal motor speed and pullout efficiency at zero rpm based on the rpm from the motor curve which corresponds to the maximum motor torque required up to and including hard seat contact. See Appendix A for details.
- pullout efficiency in both directions if the actual test data indicates disc movement prior to the motor reaching full speed.
- pullout efficiency in both directions if the valve control circuit permits the valve to be jogged in sequential operations.
- pullout efficiency for DC motors.
- The *stem factor* based on the design value or, if available, tested value, not degraded, coefficient of friction is used."

The stated position needs to be clarified in the following areas:

- The conditions for use of AF = 1 may be interpreted two possible ways:

  - Temperature compensation evaluated and degraded voltage factor $\leq$ 1.0; or

  - Temperature compensation $\leq$ 1.0 and degraded voltage factor $\leq$1.0

  We understand that the first is the intended interpretation. This should be clarified.

- The use of efficiency for open and close strokes needs to be clarified. We understand that the intent the white paper is as follows:

  - To permit use of running efficiency only when the rated motor starting torque is used. If the adjusted motor torque is used, use of the interpolated efficiency is permitted, but use of running efficiency is not permitted. (The present wording of the white paper permits use of adjusted motor torque in conjunction with running efficiency on opening strokes.)

  - To require use of pullout efficiency in the open direction when testing indicates disc movement prior to the motor reaching full speed. (The present wording of the white paper requires use of pullout efficiency for open and close strokes when testing indicates disc movement prior to the motor reaching full speed.)

  - To require use of pullout efficiency for all DC motor applications, regardless of how motor torque is determined.

- The position on stem factor is not sufficiently defined, because it does not refer to a specific method or source for obtaining a value of stem friction coefficient.

- The position on stem factor indicates that a non-degraded COF will be used while the justification in Section C.1.6 indicates that a degraded COF is used as the basis for setting the minimum target thrust window. This confusion is related to the need to clarify the conditions for application of the position to torque controlled versus limit controlled MOVs. We understand that the position is being revised to clearly defined the approach for each type of MOV.

<u>Texas Utilities Data Relevant to WP-125 Position</u>

At the Third NRC/ASME Symposium on Valve and Pump Testing, Texas Utilities presented their findings from a comparison of the design operator thrust capability (as determined by the standard Limitorque equations) to actuator stall testing and in situ testing results. Texas Utilities (TU) found that the measured performance of some specific actuators was less than that predicted by the design calculations, and that the performance of others exceeded the predictions. Specifically, TU torque stand testing showed from 0 to 40% average margin (depending on actuator size) between design calculations and test results. However, in situ testing identified an offsetting motor torque to stem torque conversion performance reduction (average of 10 to 23%, depending on actuator size) due to the presence of the stem thrust load. Additionally, their test results indicate that the use of running efficiency in the design calculation produces non-conservative results on a average basis. TU's results strongly suggest that on an average basis there is not a substantive margin between the design operator thrust capability calculation and actual actuator performance. In consideration of these findings, any general position which involves the reduction of perceived conservatisms in the Limitorque design operator thrust capability calculation will require significant test results to support the position.

<u>Overview and Examples of Limitorque Equations and WP-125 Approach</u>

The original Limitorque, revised Limitorque, and White Paper 125 approaches to calculating the actuator torque capability are evaluated in an example as follows. The original Limitorque torque capability equation defined by the Limitorque Selection Guide can be expressed as follows:

Actuator Torque Capability = DV * MTQ * OAR * AF * PE

Where, DV = Degraded Voltage Factor
(Degraded Voltage/Rated Voltage)**2 for AC Motors
(Degraded Voltage/Rated Voltage) for DC Motors
MTQ = Motor Rated Starting Torque, which is specified by Limitorque and is usually no greater than 90% of the motor rated stall torque (based on the vendor generic torque vs. speed curve)
OAR = Overall Gear Ratio
AF = Application Factor. Normally AF = 0.9
PE = Pullout Efficiency

The subsequent Limitorque allowances and clarifications include the following:

- DV is not applied if the motor terminal voltage is greater than 90% (Limitorque Technical Update 92-02).

- AF may be revised to 1.0 when motor terminal voltage is less than 90% (Limitorque Technical Update 93-03).

- The use of running efficiency (RE) instead of PE is permitted for applications involving a close safety function with no potential of the actuator stopping at any point during the closing stroke (Limitorque 9/17/92 letter to Cleveland Electric). Although not explicitly stated in the limitorque letter, we understand that this guidance was to be restricted to AC motors only.

- Temperature compensation is required whenever the rotor temperature is above 40 °C (Update 93-03).

White Paper 125 defines the following additional revisions:

- Revision of AF from 0.9 to 0.95

- Revision of MTQ to Rated Starting Torque plus 75% of the difference between rated stall torque and starting torque.

- For AC motor application, the use of RE for the opening stroke, as long as the motor is up to full speed before disc motion occurs.

- For AC motor closing stroke application, the use of an interpolation between PE and RE based on the rpm associated with the adjusted MTQ.

To illustrate the effects of these revisions to the original design equation, the following example actuator is evaluated:

- SMB-0-25 (Rated start torque = 25 ft-lbs.)
- Degraded Voltage slightly over 90% of rated voltage.
- Generic Motor Rated Stall Torque of 29.5 ft-lbs (White Paper 125, Table 1).
- OAR = 48
- PE = 40% (SEL 7)
- RE = 50% (SEL 7)
- SE = Stall Efficiency = 50% (SEL 7)
- Thrust Effect of 23% (TU Paper, Table 6a, ST Value for SMB-0) This is an average value - not worst case.

An actuator torque stand test to stall conditions at 90% voltage would be expected to produce a "best-estimate" output torque (measured on the drive sleeve) of:

(0.9)**2 * 29.5 * 48 * (SE=0.5) = 573.5 ft-lbs

An MOV test to stall conditions at 90% voltage would be expected to produce a "best-estimate" output torque (measured on the stem) of:

573.5/1.23 = 466.2 ft-lbs

Using the original Limitorque design calculation the actuator torque capability would be:

(0.9)**2 * 25 * 48 * 0.9 * 0.4 = 349.9 ft -lbs

producing a margin of 100 - (349.9/466.2) * 100 = +25%

Applying all of the applicable Limitorque allowances the actuator torque capability would be:

25 * 48 * 0.9 * 0.5 = 540 ft-lbs (closing)
25 * 48 * 0.9 * 0.4 = 432 ft-lbs (opening)

producing a margin of 100 - (540/466.2) * 100 = -16% (closing)
producing a margin of 100 - (432/466.2) * 100 = +7% (opening)

Without the "Thrust Effect", the margin would be:

100 - (540/573.5) * 100 = +6% (closing)
100 - (432/573.5) * 100 = +25% (opening)

Applying all of the White Paper 125 methods, the actuator torque capability would be:

For opening, (28.4) * 48 * (0.95) * 0.5 = 647.5 ft-lbs

producing a margin of 100 - (647/466.2) * 100 = -39%

We understand that it is not the intent of the white paper to use running efficiency with the increased torque value; however, the white paper does not include this restriction and has been in use for some time. If starting torque was used the margin would be:

25 * 48 * 0.95 * 0.5 = 570 ft-lbs

producing a margin of 100 - (570/466.2) * 100 = -22%

We also understand that the thrust effect is intended to be accounted for in the stem factor; however, the white paper is not clear on how this is accomplished for specific MOV applications. If starting torque is used and the "Thrust Effect" is not applied, the margin for the intended White Paper 125 approach in the opening direction would be:

100 - (570/573.5) * 100 = +0.6%

For closing including the thrust effect, 28.4 * 48 * 0.95 * (0.4) = 518 ft-lbs

producing a margin of 100 - (518/466.2) * 100 = -11%

Without the "Thrust Effect", the margin would be:

100 - (518/573.5) * 100 = +9.6%

For other actuators the WP-125 method includes a interpolated efficiency closer to running efficiency which would cause the closing direction to have negative margin even without consideration of the thrust effect.

Technical Justification of WP-125

Detailed comments concerning the information included in the Justification section of the white paper are provided below. See the white paper (Attachment 1) for the stated justifications.

Motor Capability Calculation

- Both the equation for Total Thrust and the equation for Full Motor Torque include terms for temperature degradation (TF and TC, respectively). These terms appear to be redundant and their use should be clarified.

Motor Torque for Analysis

- The position changes the normal use of starting torque for the motor torque term by adding 75% of the difference between the rated starting torque and the rated stall torque. In Section C.1.1, it is stated that "The motor torque for analysis has been conservatively estimated by reducing the stall torque to obtain a 25% margin." Since the stall torque may be only 10% greater (and in some cases less than 10%) than the starting torque, this statement may be interpreted as misleading. This change could be re-stated as a minimum operability margin of about 2.5% has been defined as compared with the nominal minimum design margin of about 10%. This change shows that the margin is small and it is difficult to characterize as "conservative".

- In Section C.1.1, the statement referenced from SEL-3 page 3 of 4 is a generalized statement used in a different context within the SEL. This reference does not provide significant support for the position that locked rotor torque rating can be reliably obtained in actual MOV performance.

- In Section C.1.1, it is indicated that separate EPRI documents clearly describe the stall capacity as exceeding 110% of the motor start torque. Although the specific references listed support the position that stall capacity normally exceeds 110% of the motor start torque, the references do not state or indicate that the stall capacity always exceeds 110% of the motor start torque. Accordingly, the referenced EPRI documents are consistent with the CECo findings that in some cases the stall capacity is less than 10% greater than the motor start torque. The words in this section should be revised to clarify the understanding of the EPRI statements.

- The reference for the quote at the bottom of page 5 should be changed from number 2 to 6.

- In Section C.1.1, the reference to "Reference 4, Section 7, page 7-15" should be clarified.

- Although vendor quoted generic speed-torque curves are typically conservative, this position should be supported by sufficient motor test results to conclude that the actual stall torque for a specific motor will always meet or exceed the rated stall torque on the manufacturers generic speed-torque curves. Texas Utilities testing found some random instances where motors did not achieve the rated stall torque. In Section C.1.1, it is noted that CECo testing of eight DC motors found that Peerless DC motors meet or exceed the stall torque values documented on the generic motor curves. The results of CECo AC motor testing should also be discussed in this section, and references to the specific test results and evaluations to support these conclusions need to be provided.

- The information in Table 1 should be clarified as follows:

  - A column should be added to define the specific motor vendor and model.

  - A note should be added to define the start torque as the Limitorque rated motor start torque.

  - A note should be added to indicate whether the source of the stall torque is the vendor's generic rating or CECo test results.

Application Factor (AF)

- No detailed technical basis has been provided for assuming an AF = 0.95 in place of the typical value of 0.90.

- As noted in Section C.1.2, Limitorque uses the AF to account for variances of motor start torque and pullout efficiency at varying voltage levels and various actuator speeds and conditions. This reference is not complete since it does not include the additional Limitorque statements that ".9 should be used in most cases. .8 should be used if the motor is 900 rpm or the actuator is an SB sized for line temperatures above 900 F." No comments have been made in the white paper for the other considerations that impact AF such as 900 RPM motors or high temperature applications using SB type actuators. Although these conditions are not normally expected in nuclear applications, the restriction needs to be mentioned to avoid potential misapplication.

- Since Limitorque allows AF = 1 when the degraded voltage ratio is applied, one may conclude that the value of AF is related to the degraded voltage effect and not simply a term providing additional design margin. If so, using an AF = 0.95 may be conservative when full rated voltage is available, and may be non-conservative when only 91% of rated voltage is available and the degraded voltage ratio is taken as DV = 1.

Degraded Voltage

- Similar to the standard method, the position should be clarified to indicate the intent to only use the degraded voltage ratio (DV) when the calculated degraded voltage is less than 90% of the rated voltage. However, we understand that CECo is reconsidering the elimination of the DV factor between 90% and 100% of rated voltage in light of recent motor test results.

- Above 90% rated voltage the limitorque-recommended use of AF = 0.9 and the fact that the starting torque is usually no greater than 90% of the rated stall torque may be compensate for not using the degraded voltage factor. Since the White Paper 125 position is using a calculated torque greater than starting torque, it appears this may be eliminating one of these compensating factors, and contributing to potentially non-conservative calculations.

Unit Efficiency

- The Texas Utilities (TU) test results indicated that the use of running efficiency in the design calculation produces reasonable results in torque stand testing, but produce non-conservative results on a average basis in MOV tests. Additionally, in some cases the use of pull out efficiency may not be conservative in MOV tests. These conclusions are due to the offsetting "Thrust Effect" identified by TU. We understand CECo intends to cover thrust effect through the use of higher stem factors. This needs to be clearly explained.

- A 04/30/93 NRC letter states that in a Limitorque 9/17/92 letter to Cleveland Electric, Limitorque indicates that run efficiency can be substituted for pullout efficiency where the application involves a close safety function with no potential of the actuator stopping at any point during the closing stroke. Also, the TU paper indicates that the manufacturer has stated that utilities may use the running efficiency when evaluating the capabilities of actuators already installed in power plants. No specific statements were identified in the NRC letter or TU paper concerning the opening stroke. Accordingly, the White Paper needs to provide explicit justification for using RE on open strokes.

- The efficiency interpolation factor where calculated is based on relatively high overall gear ratios. OARs in this range produce self-locking gear sets. The capability of actuators with non-locking gear sets to reach full speed before the stem nut engages should be addressed.

Stem Factor

- The position statement in Section B for stem factor states that "not degraded coefficient of friction is used." The justification in Section C.1.6 states that "Lubrication degradation of the stem coefficient of friction is considered in setting the minimum target thrust window." This confusion is related to the need to clarify the conditions for application of the position to torque controlled versus limit controlled MOVs. We understand that the position is being revised to clearly define the approach for each type of MOV. Specific information should be added concerning the magnitude and basis for the stem factor degradation that will be assumed for setting the minimum target thrust window.

- Since the stem factor is being used to compensate for the "Thrust Effect", specific criteria and bases need to be defined for determining the stem/stem-nut coefficient of friction for MOVs which have not been tested. The use of test results from other similar MOVs may be complicated by the following factors:

    - Differences in stem diameter or threads per inch affect the stem thread lead angle which in turn affects the calculated thread COF from test data. The thrust affect for a given actuator size and gear set will result in different changes in COF for different lead angles.

    - In a static MOV test, the "Rate of Loading Effect" could mask the "Thrust Effect" giving the appearance that neither effect is present. The "ROL effect" in a static test reduces the stem thread coefficient of friction (COF) as compared to a dynamic test, and the "Thrust Effect" would increase the calculated COF. Both static and dynamic tests of a valve are required to separate out the change in COF due to the potential presence of the ROL effect.

    - EPRI dynamic testing of MOVs has found that the difference in COF between static and dynamic tests of ROL sensitive MOVs is dependent upon the magnitude of the running load at the point of initial disk to seat contact. ROL sensitive MOVs which have low dynamic test running loads behave essentially the same as a static test. As a result, test results from ROL sensitive MOVs should not be used as the only basis for developing a COF for a valve which has not been tested.

When representative test data is not available, the use of the CECo design COF values for calculating operator thrust capability may not account for the "Thrust Effect". An EPRI survey of CECo personnel performed by MPR Associates found that CECo uses the following design COF values:

0.2 Non-Westinghouse valves
0.175 Non-Westinghouse valves lubricated on a 18 month cycle
0.15 Westinghouse valves
0.125 Westinghouse valves lubricated on an 18 month cycle

From results of EPRI dynamic testing of MOVs with torque measured directly from the stem, the average value of COF for good performing stem-nut lubricants was about 0.125. Accordingly, the use of a design COF value of 0.125 for an MOV which has not been tested may not account for the upper bound of actual test results or the thrust effect.

We understand that references to other White Papers will be included to define the methods for determining stem factors. Additionally, considerations for the dimensional differences and the ROL effect will be reviewed in light of thrust effect compensation.

- The approach for inaccuracies in test results also needs to be addressed.

Attachment A

White Paper 125

Revision 1 Draft 7/11/94

Installed Motor Capability Evaluation

Interim Position

Commonwealth Edison Company

Corporate MOV Program Support

Prepared by: ______________________
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White Paper 125

Installed Motor Capability Evaluation
Interim Position

A. Purpose:

The purpose of this paper is to address how and when an AC or DC motor operated valve (MOV) motor capability can be evaluated using full motor torque and other motor variables. This position is interim pending resolution of Limitorque motor rating and capability, and actuator efficiency questions. This position is to be applied to present and future capability calculations. These full motor capacity calculations are to be used to support testing and to establish an interim torque switch setting until a modification to enhance margin can be completed.

The capability calculations discussed in this white paper are for establishing testing thrust windows as opposed to determining the maximum allowable torque switch settings based on structural limits.

Full motor capability calculations can be used to establish a testing thrust window which considers all the phenomena of concern identified in Limitorque Maintenance Update 89-1. That is, degraded voltage, proper use of diagnostic test equipment, degraded stem to stem nut coefficients of friction, and inertia effects are considered. In addition, temperature effects on the available motor torque shall be considered. The established motor capability determined through the use of full motor capability calculations preserves the overall margin for a MOV until such time that final modifications can be implemented.

B. Position:

When calculating full motor capability, the following changes in the variables of the MOV motor capability calculation sizing equation can be considered: Note, use only the changes required to calculate an adequate thrust window to support testing.

- The *application factor*, which is typically set to 0.90 or less, is taken to be 0.95. If temperature compensation and an under voltage factor of less than one is included, the application factor becomes 1.0 (Ref. 8).

- The voltage supplied to the motor is assumed to be *degraded voltage*.

- The *start torque* of the motor is replaced by the start torque plus 75 percent of the difference between start torque and stall torque. See table 1 for AC and Porter Peerless DC motors.

- The *unit efficiency* to be used is:

    - run efficiency if the actual test data indicates that the motor reaches full speed (steady state current indicates motor is at full speed) prior to disc movement in the openning direction.

    - Use interpolation between running efficiency at nominal motor speed and pullout efficiency at zero rpm based on the rpm from the motor curve which corresponds to the maximum motor torque required up to and including hard seat contact. See Appendix A for details.

    - pullout efficiency in both directions if the actual test data indicates disc movement prior to the motor reaching full speed

    - pullout efficiency in both directions if the valve control circuit permits the valve to be jogged in sequential operations

    - pullout efficiency for DC motors

- The *stem factor* based on the design value or ,if available, tested value, not degraded, coefficient of friction is used.

C. Justification:

1. Motor Capability Calculation for Analysis

The motor/gearing capacity is determined from the equation provided in the Limitorque Selection Guides:

$$\text{Total Thrust} = \frac{MT * OAR * EF * AF * DV * TF}{FS}$$

where,

MT	=	Motor Torque, ft-lbs
OAR	=	Unit Ratio, dimensionless
EF	=	Unit Efficiency, dimensionless
AF	=	Application Factor, dimensionless
DV	=	Degraded Voltage Ratio, dimensionless (Ratio of degraded to rated voltage, squared for AC, simple ratio for DC)
FS	=	Stem Factor, ft
TF	=	Temperature degradation Factor, dimensionless

As noted previously, motor torque, gear efficiency, terminal voltage, and application factor are maximized to perform the overload analysis. In the case of the normal sizing analysis, overload analysis, or full motor capability analysis, the total thrust resulting from the motor/gearing is calculated using the same equation, only the values of the variables differ. The specific justification for the value of each factor is presented below. The overall conservatism of this methodology is further justified by comparison with the published stall torque values from Texas Utilililies MOV test program Reference 10. The details of this comparison is given in Appendix B.

1.1 Motor Torque for Analysis (MT)

The motor manufacturer has issued generic curves of motor speed versus torque that provide torque values from the zero torque or full speed condition to the motor stall or zero speed condition. The motor torque for analysis has been conservatively estimated by reducing the stall torque to obtain a 25 percent margin.

$$\text{Full Motor Torque} = \left[.25 \cdot \text{StartTorque} + .75 \cdot \text{StallTorque}\right] TC$$

TC is the temperature factor per Reference 8.

The Limitorque Selection Guides in performing overload analysis would use either the stall torque or 110 percent of the motor start torque. As shown in table 1, the motor stall capacity does not always exceed 110 percent of the motor start torque and that for some motors it would be non-conservative to use 110 percent of start torque in calculations for operability. The Limitorque Selection Guide SEL-3 (Ref. 1, page 3 of 4), states:

> "Limitorque motors will produce whatever torque is demanded <u>up to and including the locked rotor torque rating;"</u>

The generic motor curves themselves represent expected though not guaranteed motor performance. In Reference 2, it states,

> "However, motor stall torque is better estimated by using the motor curves. If the correct motor curve can be identified, the speed vs. torque curve can provide a generic stall torque value for a specific design motor."

In separate EPRI documents, the stall capacity of the motors provided with Limitorque actuators are clearly described as exceeding 110 percent of motor start torque. From Reference 3, Section 3.4.1, Page 3-24:

> "The rated starting torque of the motor is usually 65 percent to 90 percent of the motor stall torque."

From the section for DC motors, Section 3.4.3, Page 3-27:

> "The rated starting torque (10 ft-lb) is 63 percent of the locked-rotor torque (16 ft-lb). This margin is larger than in an AC motor."

From Reference 4, Section 7, Page 7-15,

> "Motor Stall Torque
>
> These values normally exceed nominal motor ratings by as much as 40 percent for AC motors (120 percent for DC motors) at rated voltages."

CECo has also independently performed testing of DC motors and has found that the Peerless DC motors meet or exceed the stall torque values documented on the manufacturers generic motor curves. The testing has also demonstrated that the voltage ratio relationship for degraded voltage is valid at voltages as low as 10 percent of the motor's rated voltage, (Reference 5).

1.2 Application Factor (AF)

The application factor is described in Reference 7 (Page 5) as accounting for:

> "variances of the Motor Start Torque and the Pullout Efficiency at varying voltage levels and various actuator speeds and conditions. It also makes allowances for any special application considerations."

The application factor is purely a term used to provide design margin for effects or phenomena not explicitly defined.

In using full motor capacity calculations for interim testing and operability assessment all of this additional design margin need not be provided, i.e., the application factor is to provide margin for sizing purposes. Limitorque Technical Update 93-03 (Reference 8) allows the complete removal of the Application Factor if the motor torque is derated by the appropriate temperature compensation factor and an under voltage factor based on a motor voltage of less than 90%.

1.3 Degraded Voltage (DV)

In the same way that the sizing calculations are performed, the degraded voltage term is used in the full motor capability equations. The motor terminal voltage is conservatively calculated at locked rotor conditions. The ratio of the motor terminal voltage to rated voltage at degraded voltage conditions is squared for AC motors and is used as a simple ratio for DC motors. A degraded voltage term is not included for motor voltages down to 90% of rated. No deviation is taken with respect to the standard industry equation for sizing for the degraded voltage term. However, this is distinct from the Limitorque stall equation which uses rated voltage, i.e, the ratio of voltages is 1.0. Therefore, Limitorque's caution as expressed in the Maintenance Updates 89-1 and 92-1 with respect to degraded voltage conditions, have been encompassed.

1.4 Overall Gear Ratio (OAR)

The gear ratios are physical constants related to the actual physical dimensions of the gears. There are no adjustments taken for the gear ratios.

1.5 Unit Efficiency (EF)

The unit efficiency to use in the full motor capability calculations is based on the specific gear set, the motor rpm, and the actuator model.

In the opening stroke, run efficiency is used for valves that operate over the full open stroke because the valve control circuitry seals-in the open signal. If the valve can be mid-positioned or throttled or if the valve is otherwise known to have a drive train without a lost motion gear set, the pull-out efficiency is conservatively used. Test data has shown that in the beginning of the opening stroke, the motor and gearing attains full speed before the stem nut engages the stem threads to pull the disk out of the seat. See Table 2. This phenomenon is physically attributed to the gears being engaged on one face after the closing stroke, where upon the gears must turn to take up the backlash prior to pullout. In addition, the motor and gearing are not loaded until the stem is placed in tension from its compressed state. The stem nut clearance provides another gap where the motor runs essentially unloaded. Furthermore, in the case of almost all gate valves, loading does not occur until the hammer blow feature has impacted and engaged the drive sleeve and the T-head clearance between the stem and the disc is taken-up. For mid-position or throttle valves, backlash in the drivetrain may not always be available, because the valve can be partially stroked in the same direction multiple times. The motor and gearing can be loaded as soon as the motor starts to turn.

In the closing stroke, interpolation between pullout and run efficiency is used for valves with sealed-in circuits and lost motion actuators. Both References 4, page 7-14, and 9, page 1932 show that the efficiency of the worm is a function of the worm speed. The biggest decreases in efficiency coming from the smallest diameter worms running at the lowest rpm's. The decrease is noticable although not appreciable for small changes in rpm about 900 and 1800 rpm for SMB-000 actuators. Therefore, it is prudent to use less than running efficiency. References 4 and 9 also show that the efficiency versus gear speed is concave downward. Appendix A provides details on the method for interpolating efficiency.

1.6 Stem Factor (FS)

The stem factor is directly related to the coefficient of friction between the stem and stem nut. Specific analytical equations are available for calculating the stem factor for given coefficients of friction for a given stem thread.

The lubrication frequency until the time that a modification is scheduled to restore the margins to the actuator shall be considered when determining the friction coefficients to be used for calculating the minimum target thrust window. When possible, consider increased lubrication frequency. Lubrication degradation of the stem coefficient of friction is considered in setting the minimum target thrust window. This assures that the actuator will provide more than the minimum required thrust under degraded conditions prior to torque switch trip. The stem factor used for calculating the full motor capability should be the tested stem factor. Lubrication degradation will correspondingly decrease both the thrust at torque switch trip and the motor gearing thrust. Therefore, when comparing motor capacity to torque switch setting, stem factor degradation is not appropriate.

D. References

1. Selection Procedures for Nuclear Actuators, Limitorque Corporation, June 6, 1979.

2. Limitorque Maintenance Update 89-1, Limitorque Corporation.

3. EPRI Report NP-6660-D, Research Project 2814-6, "Application Guide for Motor-Operated Valves in Nuclear Power Plants," March 1990.

4. EPRI Report, "Technical Repair Guidelines for the Limitorque Model SMB-000 Valve Actuator."

5. GDS Associations Calculation MSC-GN-001, "Study for Degraded Voltage Impact on DC Motor Starting Capability," Rev. 1, August 13, 1992.

6. Limitorque Maintenance Update 92-1, Limitorque Corporation.

7. Limitorque Technical Update 92-02, Limitorque Corporation.

8. Limitorque Technical Update 93-03, Limitorque Corporation.

9. Green, Robert E. Editor, "Machinery's Handbook", Edition 24, Industrial Press Inc., New York, 1992.

10. Black, Bill R., "Results of the Motor-Operated Valve Engineering and Test Program" from Proceedings of the Third NRC/ASME Symposium on Valve and Pump Testing, published in NUREG/CP-0137.

[illegible]					
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1800	2	2.6	0.6 (30%)	0.45 (23%)	2.45
1800	2	2.8	0.8 (40%)	0.6 (30%)	2.6
1800	5	5.6	0.6 (12%)	0.45 (9%)	5.45
1800	5	6.5	1.5 (30%)	1.1 (22%)	6.1
1800	10	10.9	0.9 (9%)	0.7 (7%)	10.7
1800	15	17.3	2.3 (15%)	1.7 (11%)	16.7
1800	25	29.5	4.5 (18%)	3.4 (14%)	28.4
1800	40	49	9 (22.5%)	6.7 (17%)	46.7
1800	60 - (56 frame)	66	6 (10%)	4.5 (8%)	64.5
1800	60	81	21 (35%)	16 (27%)	76.0
1800	80	93	13 (16.3%)	9.7 (12%)	89.7
1800	100	114	14 (14%)	10 (10%)	110
1800	150	182	32 (21%)	24 (16%)	174
1800	200	235	35 (17.5%)	26 (13%)	226
1800	250	295	45 (18%)	34 (13%)	284
1800	300	335	35 (11.6%)	26 (8.7%)	326
1800	350	405	55 (15.7%)	41 (12%)	391
3600	2	2.7	0.7 (35%)	0.5 (26%)	2.5
3600	5	6.4	1.4 (28%)	1.0 (21%)	6.0
3600	5	7.4	2.4 (48%)	1.8 (36%)	6.8
3600	7.5	8.8	1.3 (17.3%)	1.0 (13%)	8.5
3600	10	12.4	2.4 (24%)	1.8 (18%)	11.8
3600	15	20	5 (33.3%)	3.7 (25%)	18.7
3600	25	29	4 (16%)	3.0 (12%)	28.0
3600	40	50	10 (25%)	7.5 (18%)	47.5
3600	60	85	25 (41.6%)	18 (31%)	78
3600	80	100	20 (25%)	15 (18%)	95

[illegible]					
RPM/VOLT	[illegible] Torque	[illegible] Torque	[illegible]	[illegible] Difference	[illegible] Torque for Analysis
3600	100	105	5 (5%)	3.7 (3.7%)	103.7
3600	150	172.5	22.5 (15%)	16 (11%)	166
3600	200	227	27 (13.5%)	20 (10%)	220
3600	250	295	45 (13.5%)	37 (13%)	287
3600	300	355	55 (18.3%)	41 (13%)	341
3600	400	477	77 (19%)	57 (14%)	457
250V	5	10.7	5.7 (114%)	4.2 (85%)	9.2
250V	7.5	15	7.5 (100%)	5.6 (75%)	13.1
250V	10	16	6 (60%)	4.5(45%)	14.5
250V	15	25	10 (66%)	7.5 (50%)	22.5
250V	25	40	15 (60%)	11.2 (45%)	36.2
250V	40	57	17 (42%)	12.7 (31%)	52.7
250V	60	82	22 (36%)	16.5 (27%)	76.5
250V	80	107	27 (33%)	20.2 (25%)	100
250V	100	120	20 (20%)	15 (15%)	115
250V	150	310	160 (106%)	120 (80%)	270
250V	200	530	330 (165%)	247 (123%)	447
125V	2	4.8	2.8 (140%)	2.1 (100%)	4.1
125V	5	10.7	5.7 (114%)	4.2 (85%)	9.2
125V	7.5	15	7.5 (100%)	5.6 (75%)	13.1
125V	10	16	6 (60%)	4.5 (45%)	14.5
125V	15	25	10 (66%)	7.5 (50%)	22.5
125V	25	40	15 (60%)	11.2 (45%)	36.2
125V	40	63	23 (57%)	17.2 (43%)	57.2
125V	60	84	24 (40%)	18 (30%)	78
125V	80	125	45 (56%)	33.7 (42%)	113
125V	100	108	8 (8%)	6 (6%)	106
125V	200	510	310 (155%)	232 (116%)	432

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
2-202-5B	Gate	AC	0	0.127	0.507	1.197	8 - 3/25/93
2-205-2-4	Gate	AC	0	0.080	0.368	0.754	3 - 4/9/93
2-1201-1	Gate	AC	0	0.060	0.654	1.214	9 - 4/2/93
2-1301-1	Gate	AC	0	0.079	0.953	1.403	11 - 5/2/93
2-1301-2	Gate	DC	0	0.121	0.768	1.035	9 - 5/1/93
2-1301-3	Gate	DC	0	0.122	0.128	2.613	17 - 5/1/93
2-1402-3A	Gate	AC	0	0.079	0.856	1.586	16 - 5/10/93
2-1501-5A	Gate	AC	0	0.076	0.605	2.155	3 - 4/3/93
2-1501-5D	Gate	AC	0	0.081	0.626	1.901	19 - 5/2/93
2-1501-19A	Gate	AC	0	0.074	0.664	1.413	5 - 2/5/93
2-2301-3	Gate	DC	0	0.159	0.248	0.561	10 - 4/13/93
2-2301-4	Gate	AC	0	0.071	0.767	1.4424	10 - 4/22/93
2-2301-8	Gate	DC	0	0.120	0.151	0.526	4 - 5/3/93
2-2301-9	Gate	DC	0	0.182	0.227	0.616	4 - 4/26/93
2-3706	Gate	AC	0	0.062	0.549	1.226	22 - 4/30/93
2-1301-4	Gate	AC	0	0.094	1.102	1.832	7 - 4/26/93
2-1402-3B	Gate	AC	0	0.079	0.849	1.594	16 - 5/10/93

2-2301-10	Globe	DC	0	0.093	0.101	1.143	8 - 4/23/93
2-2301-14	Globe	DC	0	0.084	0.096	0.506	10 - 5/11/93
2-1501-38B	Globe	AC	0	0.073	0.061	1.131	3 - 1/30/93
2-1501-21B	Globe	AC	0	0.134	2.1142	3.580	4 - 3/2/93
2-1402-38A	Globe	AC	0	0.070	0.125	0.531	10 - 3/1/93
2-1402-38B	Globe	AC	0	0.079	0.813	1.208	14 - 3/9/93
2-220-1	Globe	AC	0	0.074	0.864	1.436	7 - 4/29/93
2-220-2	Globe	DC	0	0.139	0.166	1.638	2 - 4/29/93

The purpose of this appendix is to provide a more accurate method to interpolate between running efficiency at running rpm and pullout efficiency at near stall conditions. The last page of this appendix gives the interpolation factor to be used to determine the actuator efficiency at the WP-MOV 125 motor rating.

The following evaluation of worm efficiencies and coefficient of friction is from Machinery's Handbook 23rd Edition pages 1925 through 1933. From Table 2 we can find the coefficient of friction for worm pitch velocities below 30 fps:

$$E = \frac{100 \cdot \tan(L_a)}{\tan(L_a + \mathrm{atan}(cof))}$$

Solving for cof we get:

$$cof = \tan\left(\mathrm{atan}\left(100 \cdot \frac{\tan(L_a)}{E}\right) - L_a\right)$$

For a lead angle of 5^{o}, the efficiencies for various speeds are found from Table 2: At 5 fpm E= 40%; at 10 fpm E=47% and at 20 fpm E=52%. Substitutuing these values into the above equation we can find the coefficient of friction at these velocities.

$$L_a := 5 \cdot \frac{\pi}{180}$$

$E := 40$ $\quad cof := \tan\left(\mathrm{atan}\left(100 \cdot \frac{\tan(L_a)}{E}\right) - L_a\right) \quad cof = 0.12877$

$E := 47$ $\quad cof := \tan\left(\mathrm{atan}\left(100 \cdot \frac{\tan(L_a)}{E}\right) - L_a\right) \quad cof = 0.09708$

$E := 52$ $\quad cof := \tan\left(\mathrm{atan}\left(100 \cdot \frac{\tan(L_a)}{E}\right) - L_a\right) \quad cof = 0.07959$

We can use these values with the values from Table 3 to fit an equation for coefficient of friction versus pitch velocity:

$i := 0 .. 62$

V (fpm)	COF
5	.129
10	.097
20	.08
30	.073
40	.070
50	.066
60	.062
70	.060
80	.058
90	.056
100	.054
110	.052
120	.051
130	.050
140	.049
150	.048
160	.047
170	.046
180	.045
190	.044
200	.043
225	.041
250	.040
275	.038
300	.036
325	.035
350	.034
375	.033
400	.033
425	.032
450	.031

$$v = \begin{bmatrix} 475 \\ 500 \\ 550 \\ 600 \\ 650 \\ 700 \\ 750 \\ 800 \\ 850 \\ 900 \\ 950 \\ 1000 \\ 1100 \\ 1200 \\ 1300 \\ 1400 \\ 1500 \\ 1600 \\ 1700 \\ 1800 \\ 1900 \\ 2000 \\ 2100 \\ 2200 \\ 2300 \\ 2400 \\ 2500 \\ 2600 \\ 2700 \\ 2800 \\ 2900 \\ 3000 \end{bmatrix} \qquad cof := \begin{bmatrix} .030 \\ .030 \\ .028 \\ .027 \\ .026 \\ .026 \\ .025 \\ .024 \\ .023 \\ .023 \\ .022 \\ .022 \\ .021 \\ .020 \\ .019 \\ .019 \\ .018 \\ .0175 \\ .0170 \\ .0165 \\ .0165 \\ .0160 \\ .0160 \\ .0155 \\ .0150 \\ .0150 \\ .0150 \\ .0145 \\ .0145 \\ .0140 \\ .0140 \\ .0140 \end{bmatrix}$$

We will curve fit these values to provide an equation for coefficient of friction. We will use a power fit with a shift in both the coefficient of friction and in the velocity. The actual formula doesn't matter as long as the fit is good.

$\text{cofmin} := -.022 \quad \text{vshift} := -.20$

$\text{lncof}_i := \ln(\text{cof}_i - \text{cofmin}) \quad \text{lnv}_i := \ln(v_i + \text{vshift})$

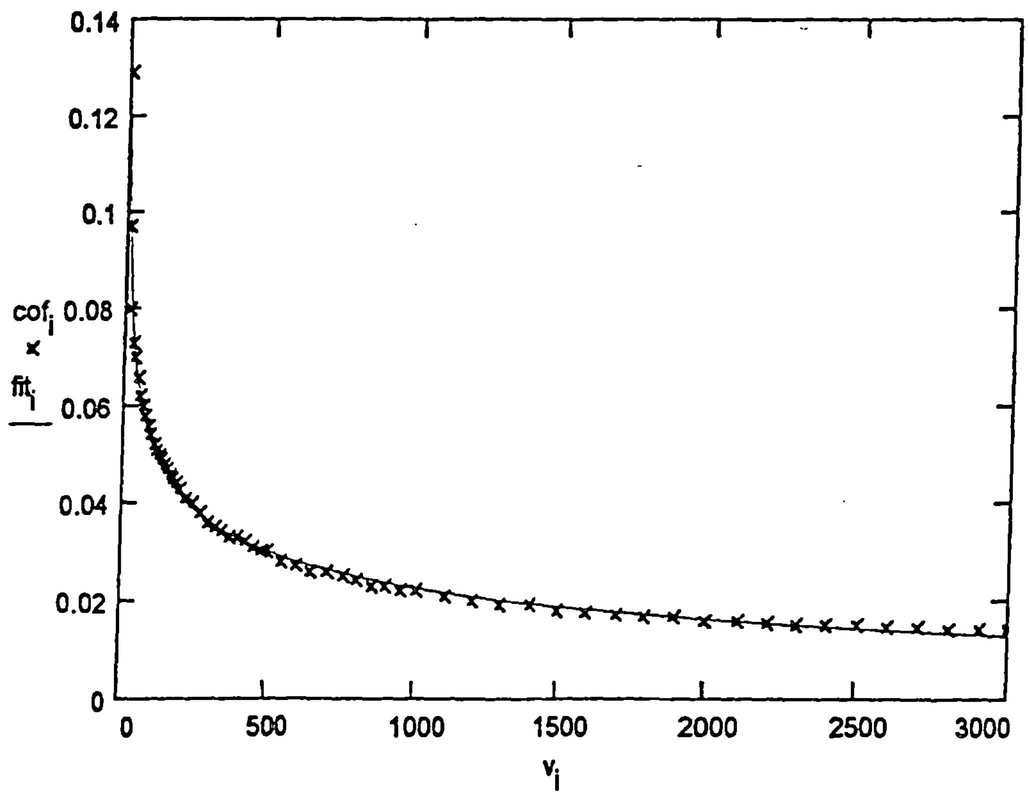
$\text{fit}_i := e^{\text{intercept}(\text{lnv},\,\text{lncof})} \cdot (v_i - \text{vshift})^{\text{slope}(\text{lnv},\,\text{lncof})} + \text{cofmin}$

The correlation factor is: $r := \text{corr}(\text{lnv}, \text{lncof}) \quad r = -0.99837$

This indicates a very good fit. We can also calculate the maximum error in this fit:

$\text{difference}_i := \left|\dfrac{\text{fit}_i - \text{cof}_i}{\text{cof}_i}\right| \quad \text{maxerr} := \max(\text{difference}) \quad \text{maxerr} = 0.0826$

This indicates that our maximum error is about 8%. The values and the curve fit is graphed below:



Now that we have a good fit for the coefficient of friction versus worm speed, we can calculate the efficiencies of the worm and worm gear set in the various Limitorque actuators. We concentrate on the worm gear set, since the remaining spur gears and bearings run at much higher efficiencies (~98 - 99%) and would not influence the change in efficiency versus the rpm. The equation for efficiency is:

$$E = \frac{100 \cdot \tan(L_a)}{\tan(L_a + \text{atan}(\text{cof}))}$$

We need to find the lead angle of the worms in order to use this equation. If we want to find the efficiency versus rpm, we will also need to find the pitch diameter. Ataached are the measured values of worm pitch, worm thread OD and worm thread height. The pitch diameter is then the worm OD minus the thread height. The tan of the lead angle is the lead aivided by pi times the pitch diameter. The velocity will be the rpm's multiplied by the pitch diameter and pi.

$j := 0..5 \qquad k := 0..100$

	$pitch_j :=$	$OD_j :=$	$height_j :=$	$ratio_j :=$	$motorratio_j :=$
SMB-000	.1738	.80	.125	50	2
SMB-00	.2604	1.13	.188	45	2.42
SMB-0	.3906	1.61	.297	37	2.6
SMB-1	.2604	1.88	.172	66	2.6
SMB-2	.3482	2.28	.281	60	2.182
SMB-3	.7188	3.06	.516	41	2.33

We will substitute the above values into the equations for worm geometry:

$pitchdia_j := OD_j - height_j \qquad rpm_k := k \cdot 36$

$$L_{a_j} := \operatorname{atan}\left(\frac{pitch_j}{pitchdia_j \cdot \pi}\right) \qquad velocity_{j,k} := \frac{rpm_k \cdot pitchdia_j \cdot \pi}{12 \cdot motorratio_j}$$

$$fit_{j,k} := e^{intercept(lnv,\,lncof)} \cdot \left(velocity_{j,k} - vshift\right)^{slope(lnv,\,lncof)} + cofmin$$

Efficiency is then:

$$E_{j,k} := \frac{100 \cdot \tan\left(L_{a_j}\right)}{\tan\left(L_{a_j} + \operatorname{atan}\left(fit_{j,k}\right)\right)}$$

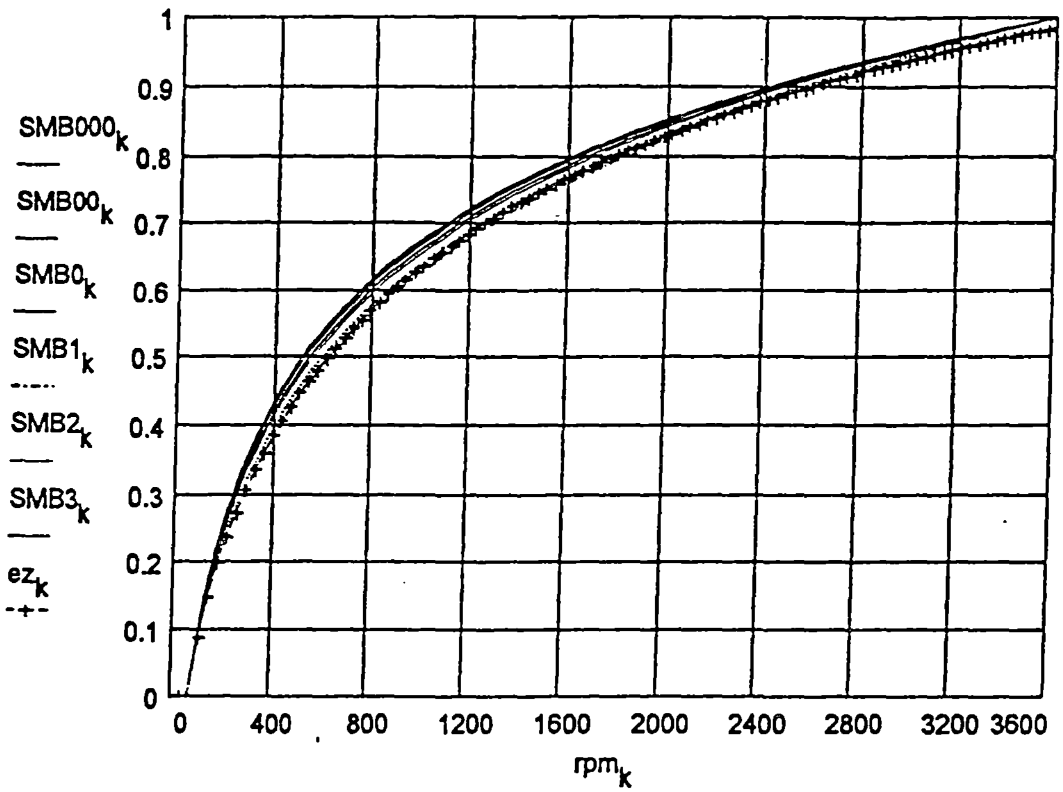
Without estimating the efficiencies for the rest of the actuator we could not use this number for actuator efficiency. What we want is a method to be used to interpolate between the running and pullout efficiencies published by Limitorque. Therefore, we will normalize the efficiency between the pullout and run values:

For 3600 rpm motor:

$$N_{j,k} := \frac{E_{j,k} - E_{j,2}}{E_{j,100} - E_{j,2}}$$

$SMB000_k := N_{0,k}$ $\quad$ $SMB00_k := N_{1,k}$ $\quad$ $SMB0_k := N_{2,k}$

$SMB1_k := N_{3,k}$ $\quad$ $SMB2_k := N_{4,k}$ $\quad$ $SMB3_k := N_{5,k}$

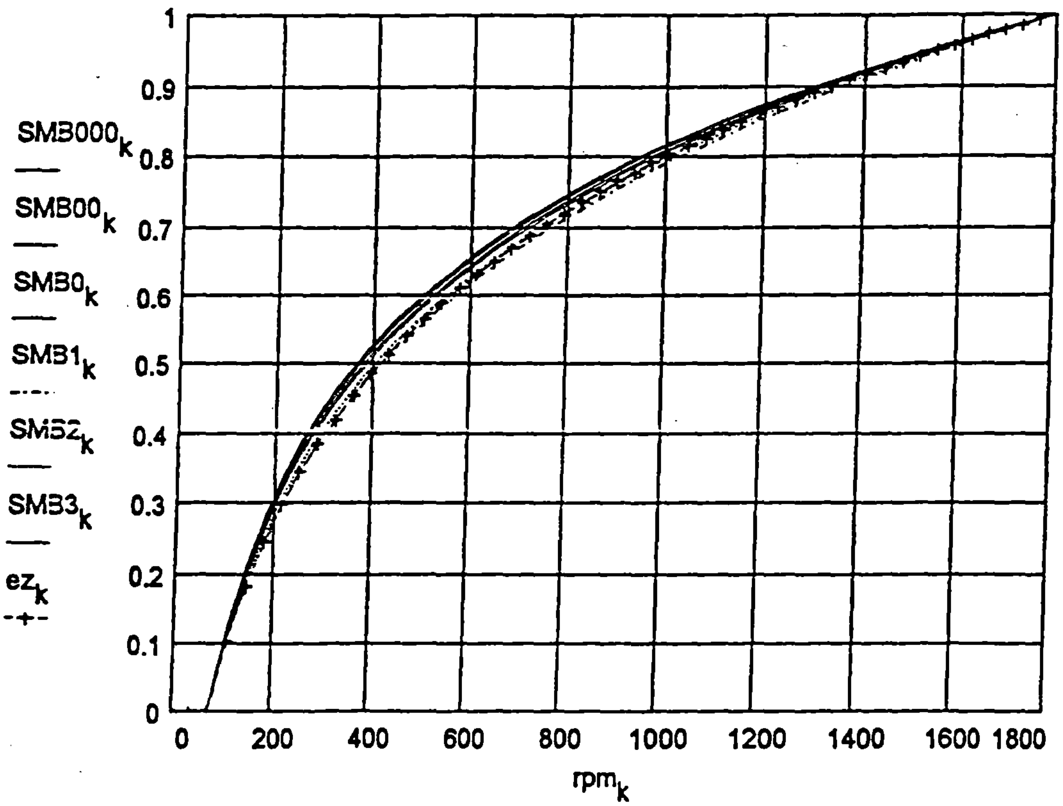


For 1800 rpm motor:

$$N_{j,k} := \frac{E_{j,k} - E_{j,2}}{E_{j,50} - E_{j,2}}$$

$SMB000_k := N_{0,k}$ $\quad SMB00_k := N_{1,k}$ $\quad SMB0_k := N_{2,k}$

$SMB1_k := N_{3,k}$ $\quad SMB2_k := N_{4,k}$ $\quad SMB3_k := N_{5,k}$



There is conservatism in the above values in that the pullout efficiency used was at 72 rpm rather than zero. Using a lower value would tend to over estimate the coefficient of friction, under estimate the efficiency at pullout and result in a higher normalized efficiency. At 72 rpm the worm velocities in the above actuators would be aproximately 5 fpm or above which is within the range of friction factors reported by Machinery's Handbook. By sticking within this range, we are assured that our friction factors are correct.

The graphs above show that the normalized efficiencies are grouped close enough so that they can be estimated by a single curve. For conservatism we use the curve for the SMB-1 actuator. On both the 3600 rpm and 1800 rpm curves above we have graphed a curve labelled ez_k. The formula for thes curves are as follows:

For the 3600 rpm motors:

$$ez3600_q := \exp\left[.218898 \cdot \ln\left(\frac{rpm3600_q - 72}{3456}\right) - .050263 - .0680836 \cdot \left(\ln\left(\frac{rpm3600_q - 72}{3456}\right)\right)^2\right]$$

For the 1800 rpm motors:

$$ez1800_p := \exp\left[.31568 \cdot \ln\left(\frac{rpm1800_p - 72}{1656}\right) - .000013885 - .0700982 \cdot \left(\ln\left(\frac{rpm1800_p - 72}{1656}\right)\right)^2\right]$$

The actuator efficiency can then be found using this interpolation factor with the following formula:

$$efficiency = (1 - ez) \cdot efficiency_{pullout} + ez \cdot efficiency_{running}$$

For ease of use we can tabulate these interpolation factors for the motors in this white paper. The motor rpm's at the white paper torques are tabulated below:

The motor speed, in rpm, at the white paper torques are:

1800 RPM			3600 RPM		
$Trq1800_p$	$rpm1800_p$	$ez1800_p$	$Trq3600_q$	$rpm3600_q$	$ez3600_q$
2.45	1480	0.94831	2.5	560	0.47727
2.6	1400	0.9295	6.0	2720	0.89281
5.45	1340	0.91458	6.8	2400	0.86297
6.1	1020	0.82044	8.5	130	0.12462
10.7	1380	0.92461	11.8	160	0.17016
16.7	1190	0.87385	18.7	80	0.02053
28.4	40	0	28.0	160	0.17016
46.7	1080	0.84029	47.5	440	0.41391
64.5	100	0.08588	78	500	0.4473
76.0	840	0.7526	95	150	0.15588
89.7	140	0.17864	103.7	440	0.41391
110	880	0.76902	166	420	0.40189
174	775	0.72473	220	200	0.22063
226	175	0.24231	287	160	0.17016
284	150	0.1981	341	120	0.10735
326	750	0.71332	457	160	0.17016
391	575	0.62145			

The following is an example of how to use the above table:

Suppose that we have an 1800 rpm, 5 ft-lb, 56 frame motor in an actuator which has a 40% pullout ar 50% running efficiency. Table 1 of WP-MOV 125 shows that we can use 6.1 ft-lb of motor torque. Tr above table shows that our interpolation factor would be .82044. The efficiency of the actuator could be found as:

$Efficiency_{run} := 50$ $\quad$ $Efficiency_{pullout} := 40$ $\quad$ $InterpolationFactor := .82044$

$$Efficiency := Efficiency_{pullout} + \left(Efficiency_{run} - Efficiency_{pullout}\right) \cdot InterpolationFactor$$

$Efficiency = 48.2044$

APPENDIX B
COMPARISON WITH TU PUBLISHED DATA

In order to incorporate the latest information into our design and evaluation process, we have reviewed industry literature to determine if there is any test data which might affect the conclusions of this white paper. Reference 10 describes testing performed at TU which provides data on the output torque of various actuator and motor combinations at 80% voltage. This gives us a chance to test our methodology against actual test data. The report describes the 15 ft-lb, 3400 rpm motor and SMB-00 actuator as being the only combination at TU which failed to produce the torque predicted by the equation used at TU. The stall torque capability of this actuator with various gear ratios is provided in Table 1 of Reference 10. The only data which is real, is the measured stall capability. The remaining data is comparisons of the measured value to the value which would be predicted using TU's methodology. What is of interest to us is how well our method predicts the measured values. The values in the table are all for 15 ft-lb motors operating at 80% voltage. The OAR of the actuators tested varied from 23:1 to 36.3:1. From the methods outlined in this paper, we would use a motor torque of 18.7 ft-lb, an efficiency of 50*.02 + 40*.98 = 40.2%, and an undervoltage factor of .64. If the motor was operating at design ambient conditions during the test we would use an application foactor of 1, otherwise we would use .95. To provide a conservative comparison we will use an application factor of 1. The actuator output would then be 18.7*.64*.402*OAR = 4.81*OAR for AF=1 and 4.57*OAR for AF = .95. The following table summarizes the predicted and the measured torque.

OAR	WP-125 AF=.95	WP-125 AF=1	Measured Closed	Measured Open
23.0:1	105.1	110.6	119.0	108.0
23.0:1	105.1	110.6	123.1	112.8
23.0:1	105.1	110.6	131.7	118.4
23.0:1	105.1	110.6	129.9	119.7
23.0:1	105.1	110.6	135.2	124.0
23.0:1	105.1	110.6	140.6	126.8
23.0:1	105.1	110.6	-----	131.9
23.0:1	105.1	110.6	134.5	135.0
23.0:1	105.1	110.6	-----	138.5

OAR	WP-125 AF=.95	WP-125 AF=1	Measured Closed	Measured Open
23.0:1	105.1	110.6	-----	147.9
30.0:1	137.1	144.3	172.9	155.6
30.0:1	137.1	144.3	-----	160.6
31.9:1	145.8	153.4	173.1	-----
31.9:1	145.8	153.4	171.6	161.4
31.9:1	145.8	153.4	-----	184.4
31.9:1	145.8	153.4	213.6	195.7
34.1	155.4	164.0	-----	185.0
34.1	155.4	164.0	-----	202.0
36.3:1	165.9	174.6	229.5	-----
36.3:1	165.9	174.6	198.0	-----
36.3:1	165.9	174.6	193.2	187.3
36.3:1	165.9	174.6	-----	193.0
36.3:1	165.9	174.6	200.0	199.0
36.3:1	165.9	174.6	233.9	216.6
Normalized Average	4.57	4.81	5.754	5.510
Normalized SD	-----	-----	.428	.431

In order to provide a means to make a single statement on the usefulness of our methodology we need to combine the data into a single sampling. Therefore, we have normalized the data by dividing the actuator torque by the OAR. The results are in the last two rows. When both the opening and closing data are combined, we have an average of 5.617 ± .448. Based on this data we 96% confidence that WP-125 with an AF = 1 predicts a conservative actuator torque. If we use the worst case, actuator torque in the opening direction, we have 95% confidence that the WP-125 methodology conservatively predicts the actuator torque. Keep in mind that the above actuator configuration is the worst case tested at TU, i.e. the only example that TU tested which did not produce the torque predicted by standard industry equations. Since, with 95% confidence, WP-125 methods conservatively calculate the actuator torque for the worst case configuration, we have more than 95% confidence that it will conservatively predict the actuator torque for other actuator configurations.

ATTACHMENT

4

MPR
ASSOCIATES INC.
ENGINEERS

November 15, 1994

Mr. Paul Dietz
Commonwealth Edison Company
1400 Opus Place
Downers Grove, IL 60515

Subject: Review of Commonwealth Edison Company White Paper MOV-WP 122, "Limitorque Operator Thrust and Torque Rating Limits," Interim Position, Rev. 1 Draft, July 11, 1994.

Dear Mr. Dietz:

Enclosed is a report of our review of the subject Commonwealth Edison White Paper. We are in general agreement with the methodology presented in the White Paper and consider that it represents an appropriate and useful interim approach to extend the thrust ratings of certain Limitorque valve operators and to account for overtorquing of those operators. Our major comments and recommendations can be summarized as follows:

- The purpose of this white paper should clarify that the methodology of the white paper is not intended to justify routine overtorquing of operators, i.e., the torque ratings are not being increased by this white paper. Higher torque ratings may eventually be justified by the additional test and evaluation work (at Kalsi) that we understand is in progress. However, the current knowledge of the reasons for the torque-related failures in the initial Kalsi tests does not support general, routine overtorquing of Limitorque valve operators.

- The white paper does not account for the difference between the torque required to increase the stem load and the torque required to release the stem load when the stem travel direction reverses (i.e., the difference between the tightening and loosening torque). This difference affects the correlation of the Kalsi test data to actual valve operation. The assumption in the white paper (that the two torques are the same) tends to overestimate the severity of the Kalsi tests and, therefore, is not conservative. It is recommended the effect of the difference between the tightening and loosening torques be considered in the methodology.

- For the SMB-000, SMB-00, and SMB-1 valve operators, a method of accounting for periods of high torque separate from the wedging portion of the stroke is proposed. This method has only a limited theoretical basis and has no experimental basis. It is recommended that this methodology not be used until its technical bases are improved.

The enclosed report provides more detailed discussion of these major comments as well as other more minor comments. We have discussed by telephone the majority of our comments with Mr. Ivo Garza. Please do not hesitate to call if you have any questions or comments regarding the enclosed report.

Sincerely,

Dwight H. Harrison

Enclosure

cc: Ivo Garza, Commonwealth Edison Co. (w/enclosure)

MPR ASSOCIATES INC. ENGINEERS

Enclosure to
MPR letter of
November 15, 1994

REVIEW REPORT FOR REVIEW OF COMMONWEALTH EDISON WHITE PAPER MOV-WP 122

OVERVIEW

This review report documents the approach and conclusions of an independent review of the Commonwealth Edison Company White Paper MOV-WP 122 "Limitorque Operator Thrust and Torque Rating Limits." This review was conducted using the Revision 1 Draft version dated July 11, 1994. A copy of this version is included as Attachment A to this report.

SCOPE OF MOV-WP 122

This white paper describes how the results of operator testing performed by Kalsi Engineering, Inc. should be utilized to determine the service life of Limitorque valve operators when they are operated above the published thrust ratings and how to adjust their life for operation in excess of the published torque rating. This white paper also provides the basis for the derating of the service life of SMB-000 valve operators with torque between 58 and 99 lb-ft to less than the Limitorque rated service life when AISI 8620 worm material is used.

CONCLUSIONS AND RECOMMENDATIONS

The white paper presents an approach that appears appropriate and useable for the operation of certain specified types of Limitorque valve operators at thrusts above their rating. It utilizes tests of actual operators in a reasonably conservative methodology. It also provides a means to evaluate the effect on the remaining operator life of past operation at greater than rated torque. We have some detailed comments on the methods proposed by the white paper and the material included in the white paper. Those comments and recommendations are presented in the body of this report. The major comments are as follows:

- The purpose of this white paper should clarify that the methodology of the white paper is not intended to justify routine overtorquing of operators, i.e., the torque ratings are not being increased by this white paper. Higher torque ratings may eventually be justified by the additional test and evaluation work (at Kalsi) that we under stand is in progress. However, the current knowledge of the reasons for the torque-related failures in the initial Kalsi tests does not support general, routine overtorquing of Limitorque valve operators.

- The white paper does not account for the difference between the torque required to increase the stem load and the torque required to release the stem load when the stem travel direction reverses (i.e., the difference between the tightening and loosening torque). This difference affects the correlation of the Kalsi test data to actual valve operation. The assumption in the white paper (that the two torques are the same) tends to overestimate the severity of the Kalsi tests and, therefore, is not conservative. It is recommended the effect of the difference between the tightening and loosening torques be considered in the methodology.

- For the SMB-000, SMB-00, and SMB-1 valve operators, a method of accounting for periods of high torque separate from the wedging portion of the stroke is proposed. This method has only a limited theoretical basis and has no experimental basis. It is recommended that this methodology not be used until its technical bases are improved.

REVIEW APPROACH

The MPR review approach is as follows:

- Review the white paper purpose to ensure that it is clearly and completely stated.

- Review the statement of position to ensure that it:

  - addresses the purpose;
  - is clear and complete; and
  - includes all appropriate restrictions and limitations with regard to its use.

- Review the technical justification to ensure that it:

  - logically presents a case with defends the stated position;
  - makes proper technical use of the theory and data which are referenced;
  - adheres to appropriate requirements of codes, standards, and regulations which are referenced;
  - does not exclude references to key data or requirements;
  - provides a sufficient technical basis for the stated position; and
  - is written in a way to provide a convincing justification.

As part of the review of the justification, comparisons to other data or approaches (e.g., EPRI data or models) that may not have been considered in writing the justification are made.

RESULTS OF REVIEW INCLUDING DETAILED COMMENTS

1. <u>Purpose of MOV-WP 122</u>

The purpose stated for the white paper is as follows:

"The purpose of this white paper is to describe how the Kalsi Report (Kalsi) may be applied to determine the appropriate service life of Limitorque actuators when operated above the published thrust or torque ratings. It includes both the guidelines for Engineering to use for determining the target thrust windows using Kalsi and the requirements that the station must follow when using a target thrust window based on Kalsi.

These guidelines are based on the Kalsi Report, <u>Thrust Rating Increase of Limitorque Actuators</u>, which documents Phase I of the Kalsi test program. This White Paper shall be used on an Interim basis until the completion of Phase II portion of the Kalsi testing program, which will examine Limitorque *torque* rating increases in greater detail.

This position paper may decrease the allowable service life for SMB-000 actuators with between 58 and 99 ft-lb of torque to a value less than the 2000 cycles published by Limitorque. This applies to Pre '88 actuators with 8620 worm material. This derating does not affect those actuators with the replacement 4320 worm material. See Section 4.0 for the methodology."

1.1 <u>Comment</u>. Based on telephone discussions with Mr. Ivo Garza of Commonwealth Edison, we understand that it is not the intent that this white paper be used to justify routine torquing of any MOVs to greater than the Limitorque rating. That is, the white paper is to be used as a means to evaluate and account for past instances of overtorquing, not as a means to justify normal overtorquing. Although this is implied in the limitations, the limitations also imply that valve operators may remain in service with the torque in excess of the Limitorque rating (see comment below). It is recommended that the purpose of the white paper specifically state that this white paper is not intended to permit routine overtorquing of valve operators.

2. <u>Technical Position of MOV-WP 122</u>

2.1 Thrust Limit Position (Section 3.0)

The position of the white paper is that the thrust of a Limitorque operator of the type covered by the white paper may be increased above 100 percent of the published thrust rating using a design fatigue curve from Figure 1 of the Kalsi report (Reference 1). This Figure is included as Attachment B to this report.

2.1.1 Comment: The thrust limit position is based on the detailed justification in the referenced Kalsi report and, therefore, the thrust limit position appears to be adequately stated and substantiated.

2.2 Torque Limit Position (Section 4.0)

The position of the white paper is that past operation at torque in excess of the published rating may be accounted for by reducing the design lifetime to account for the damage incurred during the overtorquings. The damage itself is estimated by applying the results of the Kalsi tests to determine an allowable number of cycles at higher than rated torque. In the case of SMB-000, SMB-00, and SMB-1 valve operators the higher than rated torque is based on actual failures and the application of appropriate safety factors to account for the available data. In the case of SMB-0 valve operators, no failures were experienced; however, a method to estimate the fatigue life is developed based on the demonstrated capability of the operator to withstand 4000 cycles at slightly over the rated torque and to withstand a limited number of stall torque cycles. This position is limited in applicability to operators of the type and worm ratio of those in the Kalsi tests.

2.2.1 Comment: The Kalsi report states explicitly that no increase in the torque rating above the published rating can be justified at this time. The white paper infers (in Section 5.0) that operators have been reset to be within the torque ratings; however, this is not stated as a general condition or limitation of this white paper. In fact, one of the limitations for the torque position implies that operators may remain in service with a torque in excess of the Limitorque published values, if the effects of dP loading on the required torque is considered. The white paper gives no guidance on how the dP loading is to the considered, however. We understand from telephone discussions with Mr. Ivo Garza of Commonwealth that it is not intended that this white paper be used to justify routine overtorquing; however, it appears to us that the current text of the white paper is somewhat ambiguous as to whether the revised design life estimates would allow overtorquing of current valve operators as a normal practice. We suggest that the text be edited to assure that the intent of the white paper is clear in this regard. Note that if the white paper were to be used to justify routine overtorquing, the justification would need to be expanded to discuss specifically the reasoning in the Kalsi report and to show that enough is known to permit routine overtorquing.

2.3 Position on Calculating the Remaining Life for Overtorqued Valve Operators (Section 5.0)

This position defines the amount of damage that has accumulated from previous overtorquing and reduces the remaining allowable number of cycles to ensure that the cumulative damage will not reach 1.0 in those remaining cycles at 100 percent of the rated torque.

2.3.1 Comment: We have no comments on this section; however, see the comments on Appendix A in Section 3.8.1 of this report.

3. <u>Technical Justification of MOV-WP 122</u>

3.1 Technical Justification for Thrust Position (Section 6.0)

The technical justification for the thrust position is found in the Kalsi report. No added justification is provided in the white paper.

3.1.1 Comment: Since the Kalsi recommendations are being followed for the operators (except for the derating of certain SMB-000 operators), no additional justification beyond that in the Kalsi report is needed.

3.2 Justification for Housing Cover Bolt Torque (Section 7.0)

The housing cover bolt position is based on the results presented in Kalsi Engineering Reports 1752C and 1759C.

3.2.1 Comments: We have no comments on the justification for this position.

3.3 Technical Justification for Torque Limit Extension (Section 8.0)

The technical basis for the determination of the damage from overtorquing is covered in detail in the white paper. This includes:

- Developing a method to estimate the effective number of cycles of stress that occurred in the ramp loading used in the Kalsi tests;

- Estimating the effective stress amplitude in the torque-related components that failed in the Kalsi tests by using the observed number of cycles and the generic fatigue curve;

- Developing safety factors to apply to the observed failures;

- Developing a method to account for the effect on the number of worm cycles of the rate of loading (or stiffness) of the application of the load;

- Developing a method to determine the effective number of cycles and effective stress amplitude for in-service valve operators; and
- Developing a method to reduce the remaining valve operator life to account for the past overtorquing events.

The method to derive an equivalent lifetime for the SMB-0 operators that experienced no failures in the Kalsi tests that parallels the method of the other valve operators is presented in detail.

3.3.1 Comment: The method and degree of explanation is generally adequate; however, we have number of detailed technical comments as discussed individually in connection with Appendices A and B where the details of the application of the method are presented. See Sections 3.7.1 and 3.8.1 of this report.

3.4 Technical Justification for SMB-0 Overtorque (Section 9.0)

The design capability of an SMB-0 valve operator to withstand overtorque is based on the stall torque and the Kalsi tests. The method accounts for the increased number of loading cycles than may occur if the actual valve is not as "stiff" as the Kalsi test rig.

3.4.1 Comments:

a. The basic premise of the justification is that the <u>design</u> life curve for the SMB-0 valve operators can be bounded by two points: 4000 cycles at 104% rated torque and 5 cycles at 201% rated torque. The equation for the interpolation is referred to in the text; however, it was not provided in the draft version that was reviewed. It is noted that no safety factor is directly applied to these values, i.e., they are treated as design values. The presumption is that substantially more torque or cycles would have been necessary for a failure to have occurred. There is, however, some margin that arises because the Kalsi tests probably involved more loaded cycles than an actual valve. This introduces a factor of 2 to 4 in the number of cycles, depending on the difference between the tightening and the loosening torques. That is, the number of cycles in the Kalsi tests does not directly translate to actual valve cycles. Although the actual margin in the method of MOV-WP 122 cannot be quantified, engineering judgement would indicate that it is comparable to that which would be obtained if actual failure data were available. Accordingly, the method appears to be technically appropriate.

b. The justification goes into detail to establish a stiffness for the test fixture at the high torque condition. It would appear that the stiffness is approximately the reciprocal of the R_0 value that is used elsewhere. Using the parameters in the Kalsi report for the load (48,000 lbs), the stiffness of the spring pack (500,000 lb/in), the worm ratio (37), and the lead (0.200 inch) gives a calculated R_0 of 17.76 or a stiffness of 5.6 percent of the loading per worm turn. The use of 6.3 percent as the limiting "stiffness" would be conservative and appears technically appropriate.

c. See comment 3.7.1.i in connection with the SMB-0 worksheet.

3.5 Technical Justification for Estimating Remaining Life (Section 10)

The estimates for the remaining life are based on the cumulative damage theory (Miner's Rule) which is consistent with the method used in the ASME Code.

3.5.1 Comments: We have no comments on this justification.

3.6 Technical Justification for Reducing Service Life of Pre-1988 SMB-000 Actuators (Section 11)

The fatigue life of SMB-000 operators with AISI 8620 worm material is reduced based on the results of the Kalsi test of one such operator. That test had a service life well below what would be expected if the operator's actual design life were 2000 cycles at 100 percent rated torque.

3.6.1 Comments:

a. The justification refers to a calculation that would give a service life of 162 strokes at 99 lb-ft of torque. The calculation is not further summarized in the white paper nor is it referenced. We agree that the Kalsi tests indicate that 2000 cycles at 99 lb-ft may be optimistic as a design value for the Pre-1988 SMB-000 valve operators. The 755 test cycles in the Kalsi tests had effectively more than the 755 ramp loadings to failure at a torque of 105 lb-ft by a factor of two to four. If a safety factor of 5.24 is applied and the effective number of ramps in the Kalsi tests were only two this would still imply a design life of about 300 single ramps. Although this is significantly less than 2000 cycles, is also greater than the value of 162 cycles that is quoted in the justification. The source of the 162 stroke value should be more completely identified.

b. The justification includes the assumption that the Limitorque rating is based on test data equivalent to the Kalsi tests. The method then uses

2000 cycles as if it were a data point, combines it with the approximately 800 cycle Kalsi point to get an average value of 1400 cycles. A factor of two is then used to get an allowable number of 700 cycles. This justification appears weak, particularly the use of a relatively small safety factor and the need to the use the Limitorque rating as if it were a data point. It would appear better to base the limit on the one clearly applicable Kalsi test and use the normal ASME Code safety factor of 5.24. The effect on the Kalsi test result interpretation of the reduction of torque during loosening should also be included.

c. The justification refers to a "survey of typical valves" to establish justification for the statement that valves with less than 58 lb-ft of torque should not be a concern. No details of the survey's content or results are given, nor is it referenced. It would appear that essentially the same conclusion would be reached by simply allowing the correlation method of Appendix B to be applied. For torques less than 58 lb-ft, the maximum stress and the number of cycles above the endurance torque would be such that little or no cumulative damage would occur. It would appear more appropriate to base the lower limit on the Kalsi test, unless further documentation of the "survey" is included in the white paper.

3.7 Worksheet for Determining the Effect of Torque on Allowable Number of Valve Cycles (Appendix A)

This appendix provides a detailed procedure to use the expressions obtained in Appendix B and apply them to a particular valve. Some of these comments depend on the comments on Appendix B in Section 3.8.1 of this report.

3.7.1 Comments:

a. The changes to the values of R_0 and the addition of separate values for SMB-000 (4320 worm) that are identified in the comments on Appendix B (see Section 3.8.1 of this report) will affect the values of the "torque at the endurance limit" in the table on page 13. That torque is the torque in the Kalsi test times the ratio of 38,500 over S_0. We calculate the revised values to be as follows:

- SMB-000 (8620 worm): Value unchanged from 44.6 lb-ft;
- SMB-000 (4320 worm): New value of 55.0 lb-ft;
- SMB-00: 133.8 lb-ft, instead of 167 lb-ft; and
- SMB-1: 502.9 lb-ft, instead of 650 lb-ft.

If the effect of the reduction in torque during a loosening cycle is considered there will be additional changes to these values because of the changes in S_0. The revised values would be as follows:

- SMB-000 (8620 worm): 37.25 lb-ft;
- SMB-000 (4320 worm): 47.41 lb-ft;
- SMB-00: 113.36 lb-ft; and
- SMB-1: 411.40 lb-ft.

b. Step 17) refers to the "allowable number of worm cycles." It appears that the value calculated is the allowable number of loading ramps, i.e., the number of valve cycles (for most valves). In particular, the slope of the torque curve calculated in step 14) is also equal to the quantity $TORQUE_{MEASURED}/n$ as used in Appendix B. When this quantity is divided by $(TORQUE_{MEASURED} - TORQUE_{ENDURANCE})^3$ in step 16), the result is identically the expression in Appendix B on page 22 for L_{Design}, the design life for a ramp loading that contains n cycles, i.e., a ramp with n worm rotations. Consequently, step 17) should state allowable "valve" loading cycles, not allowable "worm" cycles. Note that this comment also affects step 21), see the comment below.

Mr. Ivo Garza of Commonwealth Edison has confirmed that step 17) requires modification.

c. The values in the table in step 17) should be modified as discussed above in connection with Appendix B.

d. In steps 18) through 20) additional cycles are defined to account for other times during a stroke when the operator experienced torque above the endurance torque. It appears that the assumption is being made that the damage per worm turn for these other times will be the same as the average damage per worm turn during the main loading ramp. There is no justification of this procedure in the white paper. The cumulative damage for portions of the stroke when there was significant torque could be estimated in a fashion analogous to that used to estimate the damage during a ramp. However, to do so would require knowledge of the magnitude of the torque as well as the extent of the stroke over which it was experienced. This could easily become rather complicated and cumbersome. It may be better for the time being to limit the method in the white paper to cases for which the endurance torque is exceeded only during the final wedging. If accounting for these instances of high torque during the stroke is required, a more detailed methodology that considers the magnitude as well as the number of turns needs to be worked out and justified and the appropriate guidance provided in this white paper.

e. In step 21) the value from step 17) is divided by the number of worm turns above the "endurance" torque as determined in step 20). As commented above, step 17) is in terms of number of valve loading ramps, not the number of worm turns; consequently, it would be inappropriate to perform an additional division to get an allowable number of valve cycles.

f. It would appear that the output of this appendix is intended to be the amount of cumulative damage (in terms of the reciprocal, the allowable number of cycles) that is associated with the particular valve cycle (torque-thrust versus time) that has been analyzed. That is, the valve has experienced damage from the cycle analyzed of 1/N of its life in the one stroke that is analyzed in the Appendix. It is then necessary to sum up the total damage from all overtorquing events for the valve, determine the amount of life that has been expended, and then suitably reduce the life of the particular valve operator. The guidance (see section 5.0) provided for these final, but essential, parts of reaching a conclusion on a particular valve is based on the assumption that all the overtorquing cycles were the same, which may not be the case. It would appear that a more detailed outline of the remaining steps of the process should be included -- either in this appendix or in an additional appendix.

g. In step 15) the maximum torque is to be "adjusted for measurement accuracy." No details of how to accomplish this adjustment are indicated. It would appear that to assure consistency of application some guidance as to the source and application of the accuracy adjustment should be provided. For example, is the intent to cover random as well as bias or known calibration errors? This comment also applies to step 17) of the worksheet for SMB-0.

h. In the second part of the appendix a worksheet for SMB-0 is presented. That worksheet requires a value of "Endurance Torque." It is not evident where this value is obtained, since the method used to define the quantity for the other types of valve operators does not apply to the SMB-0 type where no failures were experienced.

i. In the worksheet for SMB-0 operators the sources of the values in steps 15) and 17) are not indicated nor are they clearly covered in the justification in Section 9.0. It appears that Section 9.0 needs to be expanded to explain the sources of the values used in these steps.

3.8 Correlation of Equivalent Maximum Stress to Actuator Torque and Allowable Lifetime (Appendix B)

This appendix discusses in detail the method to establish the lifetime for the various types of valve operators for which failures were observed (SMB-000, SMB-00, and SMB-1).

3.8.1 Comments:

These comments are presented approximately in the order the items appear in the appendix, not in the order of their importance.

a. The source of the equation for the fatigue curve for low alloy steel should be stated and its applicability to the torque limited components, such as the worm, should be discussed.

b. The expression given for the allowable number of cycles, N, as a function of the alternating stress, S, is valid only for values of $S > 38500$ psi. That condition should be stated.

c. In the integral expression for the cumulative damage, the terms R_o, S_o, and S_e should be defined. Based on the use of these terms we believe that their definitions are as follows:

   - R_o is the total number of worm turns during one loading ramp, i.e., from the beginning of the ramp when thrust is zero, until the end when it is a maximum.

   - S_o is the apparent alternating stress intensity at the peak of the ramp load.

   - S_e is the "endurance" stress intensity, i.e., the stress below which there is no accumulation of fatigue damage.

d. The integral expression for the cumulative damage is an approximation for a summation of discrete cycles. An equivalent expression can be derived for the case of discrete cycles. We have done that and concluded that only if the number of worm rotations under load is very small (e.g., 2 or 3) is there a substantial error involved in using the integral relation. For the specific cases of actual failures that are analyzed in the white paper, the differences in the effective stress amplitude have been calculated. These differences are small -- less than four percent. Because of the other approximations in the method, these differences are not judged to be significant and, therefore, the added complexity of the summation-type expressions is not considered warranted. That is, we conclude it is satisfactory to use the simpler integral expression.

e. In the development of the apparent alternating stress from the results of the Kalsi tests, it is assumed that the torque experiences four equivalent ramps in the course of one valve operator cycle. That is, there is one tightening ramp and one loosening ramp in each of two directions for a total of four ramps. The torque required from the operator during the loosening portion of the ramp is not the same as that during the tightening, however. The amount of this difference depends on the stem-to-stem nut friction coefficient and stem thread parameters and is different for each valve operator. The effect of this difference is to reduce the effective number of cycles (worm rotations) under significant load for the loosening ramp as contrasted to the tightening ramp for which the cumulative damage relations are derived. That is, the number of significant cycles in the Kalsi test is not as large as is assumed by the use of the factor of four introduced in Appendix B.

The method used for the tightening ramp can be applied to the loosening ramp by recognizing that the amplitude of the ramp is less than that for the tightening ramp by a factor F, where F is determined from the relations for the tightening and loosening of a screw thread. We have evaluated this factor using the thread parameters given in the Kalsi report and with a friction coefficient of 0.1 as measured in the vast majority of the tests. On this basis the values for S_0 for the three operators with failures changes significantly, as follows:

- SMB-000(8620 worm): 108,500 psi instead of 90,900 psi
- SMB-000(4320 worm): 85,300 psi instead of 73,500 psi
- SMB-00: 81,500 psi instead of 71,000 psi
- SMB-1: 112,300 psi instead of 95,600 psi

This method, i.e., with the alternating stress reduced in the loosening portion of the cycle, is both more realistic and is more conservative than the factor of four used in Appendix B. The method in Appendix B would appear to overestimate the severity of the Kalsi tests and, therefore, would underestimate the damage from one loading ramp. We recommend that the method in Appendix B be changed to account for the reduced torque when the thrust load is loosened.

f. In the table in the middle of page 21, it appears that the value of S_0 has a typographical error: 98,893 should be 90,893 psi. The value in the table is not a solution of the relation just above the table.

g. The values of the number of worm turns in one loading ramp, R_0, for the SMB-00 and SMB-1 actuators appear to be in error by a factor of four. It appears that the values given are for the total number of loaded turns in all the four ramps that make up one valve cycle in the Kalsi tests. We calculate the following values of R_0 (based on the lead, maximum load, spring constant, and worm ratio in the Kalsi report) for the three actuators:

- SMB-000: 31.94 turns (checks the Appendix B value);
- SMB-00: 21.0 turns, instead of 83.3 turns; and
- SMB-1: 15.3 turns, instead of 61.2.

If these values of R_0 are used, the values of S_0 will also change for two of the actuators (without adjustment for the loosening portion of the cycle as discussed in comment e, above):

- SMB-00: 70,973 psi, instead of 57,637 psi and
- SMB-1: 95,564 psi, instead of 71,069 psi.

Note that this change will affect subsequent values in the appendix as well. In a telephone discussion with Mr. Ivo Garza of Commonwealth, he indicated that this was also noted by other reviewers and is being corrected.

h. The value of S_0 for an SMB-000 actuator is calculated in Appendix B entirely from the first observed failure where the worm was AISI 8620 material and there were 755 valve cycles to failure. For the SMB-000 actuator with an AISI 4320 worm material there were two failures: one at 2458 and the other at 1648 valve cycles. The fatigue lifetime for a SMB-000 actuator with an AISI 4320 worm is, in effect, obtained in the Appendix by taking the estimate obtained from the first test (with the 8620 worm) and multiplying it by the ratio of the average cycles of the two failures (2053) to the single failure (755) for a ratio of 3.1. It is not evident why the two failures of the AISI 4320 worms are not simply treated as a separate case with their own equivalent stress amplitude value. If this were done, the apparent alternating stress for the 4320 worm would be 73,467 psi (instead of about 91,000 psi). The appendix does not justify why the 8620 case is selected as the base. Unless this can be done (for example, by arguing that the fatigue curve applies the 8620 and not to 4320), it would appear more appropriate to treat the different worm material as a separate case. We note that in Appendix A it is treated separately for the purpose of estimating the damage from overtorquing.

Note that in telephone discussions with Mr. Ivo Garza of Commonwealth Edison he indicated that the use of separate S_0 values for the two worm materials of the SMB-000 actuators is being considered.

h. The changes in the values of R_0 (as discussed above in comment 6) will affect the values of the "combined factors" as presented in the table at the bottom of page 22 (and used in Appendix A). Also, if the SMB-000 actuator with an AISI 4320 worm is treated as a separate case, another row needs to be added to the table. Our calculations indicate that the values would become:

- SMB-000 (4320 worm): 6.791×10^7
- SMB-00: 3.336×10^8
- SMB-1: 5.999×10^9

If the lower torque during the loosening cycle is accounted for in the analysis, these values will be further changed to the following:

- SMB-000 (8620 worm): 2.732×10^7
- SMB-000 (4320 worm): 5.042×10^7
- SMB-00: 2.529×10^8
- SMB-1: 4.344×10^9

Note that in Appendix A these lower factors will, in effect, increase the amount of damage for every ramp loading experienced by about 40 percent compared to the factors in the current version of MOV-WP 122.

MPR
ASSOCIATES INC.
ENGINEERS

Attachment A to
MPR Enclosure of
November 15, 1994

White Paper
MOV-WP 122

Limitorque Operator Thrust and Torque Rating Limits

Interim Position

Rev. 1 Draft, July 1994

Commonwealth Edison Company

Corporate MOV Program Support

White Paper
MOV-WP 122

Limitorque Operator Thrust and Torque Rating Limits

Interim Position

Rev. 1 Draft, July 11, 1994

Commonwealth Edison Company

Corporate MOV Program Support

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Pilot Site MOV Project Manager

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Limitorque Operator Thrust and Torque Rating Limits

1.0 PURPOSE

The purpose of this white paper is to describe how the Kalsi Report (Kalsi) may be applied to determine the appropriate service life of Limitorque actuators when operated above the published thrust or torque ratings. It includes both the guidelines for Engineering to use for determining the target thrust windows using Kalsi and the requirements that the station must follow when using a target thrust window based on Kalsi.

These guidelines are based on the Kalsi Report, Thrust Rating Increase of Limitorque Actuators, which documents Phase I of the Kalsi test program. This White Paper shall be used on an Interim bases until the completion of the Phase II portion of the Kalsi testing program, which will examine Limitorque *torque* rating increases in greater detail.

This position paper may decrease the allowable service life for SMB-000 actuators with between 58 and 99 ft-lb of torque to a value less than the 2000 cycles published by Limitorque. This applies to Pre '88 actuators with 8620 worm material. This derating does not affect those actuators with the replacement 4310 worm material. See section 4.0 for the methodology.

2.0 LIMITATIONS

The following limitations apply to both thrust and torque limit extensions:

- The original Kalsi results apply only to the SMB-000, SMB-00, SMB-0, and SMB-1 operators. The draft Kalsi report for SB compensating spring components (Reference 12.7) extends the results to SB-000, SB-00, SB-0, and SB-1 actuators. These extensions specifically do not apply to SBD actuators.

- Application of the Kalsi Report may derate an operator's allowable number of cycles below the design life of 2000 cycles. Therefore, it is important to ensure that the remaining number of allowable cycles will not be exhausted prior to replacing parts or reaching the end of the operator's service life.

- The accuracy of the diagnostic equipment used to measure the maximum thrust and torque must be taken into account in the set-up calculation and in the actual measurements.

- An evaluation of the actuator mounting bolts should be performed to ensure that the bolt material and torque pre-load values are adequate to accommodate the increased thrust and torque loads *in combination with* the valve-specific seismic loads.

- The housing cover and mounting bolts should be tightened using standard craft practice.

- Diagnostic testing should be performed following any tightening of the upper housing cover bolts to ensure that there is no binding of the actuator.

The increased Limitorque operator thrust capabilities may be used provided that the following requirements and limitations are satisfied:

- When determining the remaining number of allowable cycles for an operator, it is necessary to consider the number of cycles experienced by the operator *before* application of the new thrust limits. Miner's Rule can then be used to assess cumulative fatigue damage:

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \ldots + \frac{n_i}{N_i} = 1.0 \qquad \text{Eqn. 1}$$

where n_x = number of cycles experienced at thrust level x
N_x = number of cycles allowed at thrust level x

Example
Determine the remaining number of allowable cycles for an SMB-1 actuator whose thrust was increased from 100% to 180% of rated thrust after 1000 cycles.

Solution
Kalsi developed the thrust fatigue curve shown in Figure 1, which is described by the following equation:

$$T = 200, \qquad 1 \le N \le 762 \qquad \text{Eqn. 2}$$
$$T = 456.4 - 88.95(\log N), \qquad 763 \le N \le 4000$$

where T = Allowable Thrust Level
N = Allowable Number of Cycles.

Thus
$N_{100\%} = \log^{-1}\{(456.4 - 100)/88.95\} = 10{,}156$ cycles
$N_{180\%} = \log^{-1}\{(456.4 - 180)/88.95\} = 1280$ cycles

Using Eqn. 1
$n_2 = N_2(1 - n_1/N_1) = 1280(1- 1000/10,156) = \underline{1154}$ cycles

Note that in this example, the total allowable number of cycles, 1000 + 1154 = 2154, is greater than the design cycle life of 2000 cycles. Thus cycle counting should be unnecessary.

- The operator torque shall be verified to be less than the published Limitorque rating for 2000 cycles. If not, a torque limit extension shall be performed.

- If not already replaced, the SMB-0 motor pinion keys should be scheduled for replacement with 4140 material during next operator refurbishment.

The following limitations apply to torque limit extensions:

- These torque limit extensions do not consider the limitations imposed when low strength motor pinion key material is used in lieu of 4140 material. If an actuator has been torqued above the published limitorque ratings and it has a low strength steel motor pinion key installed, the motor pinion key must be evaluated separately. See Reference 12.6

- The operator thrust shall be verified to be less than the published Limitorque rating for 2000 cycles. If not, a thrust limit extension shall be performed.

- This method is intended primarily to evaluate over-torqued conditions which have occurred in the past. If the actuator is to remain in service with a torque in excess of the Limitorque published values, the effects of dP loading on the required torque must be considered

- The remaining life of the actuator shall be calculated in accordance with section 5. The torque bearing components in the actuator shall be replaced prior to the end of the remaining life. The torque bearing components are the worm gear, the worm, the worm shaft, the worm shaft bearings, the worm shaft gear, the motor pinion gear and the motor pinion gear key.

- This procedure is only applicable to SMB-000 actuators with a 50:1 worm ratio, SMB-00 actuators with a 45:1 worm ratio, SMB-0 actuators with a 37:1 worm ratio and SMB-1 actuators with a 34:1 worm ratio. Note that there are two different materials used in SMB-000 worms. Prior to 1988 the worm material was 8620 steel. After that time the material was upgraded to 4320. If any doubt about the worm material exists, use the formula for 8620 material.

- The worm shaft bushings in SMB-1 actuators which have overtorqued should be inspected for wear at the next available opportunity.

3.0 POSITION - Thrust Limit Extension

The actual Limitorque operator thrust limit may be increased above 100 percent of the published rated thrust according to the curve shown in Figure 1. This figure was developed using the method described in ASME Section III, Div. 1, App. II, Art. II-1000. The method provides a way to apply safety margins to the number of test cycles and test load to arrive at corresponding design values that can be used in actual service. In this case, the method applies a safety factor of 5.24 for cycles (with test and design loads equal) and 1.47 for load (with test and design number of cycles equal). Highlights from Figure 1 are given below:

- up to 763 cycles allowed at 200 percent of rated load
- 2000 cycles allowed at 162 percent of rated load
- a maximum of 4000 cycles allowed at 136 percent of rated load.

4.0 POSITION - Torque Limit Extension

Kalsi documents fatigue failure of torque-affected parts in SMB-000, SMB-00 and SMB-1 actuators. These failures occurred prior to completing the cycle testing and with fairly low stem factors. Therefore, with more common stem factors one would expect that actuators which were overthrusted would also need to be reviewed for overtorque. The following methods may be used to find the limited lifetime for actuators which have exceeded their Limitorque published ratings. This lifetime is intended to allow maintenance on valves to be scheduled as appropriate. This method is not intended to allow unlimited actuator lifetime. This method requires that the torque during the over-torqued condition be measured or estimated. Also, due to the conservative safety factors (4 - 5) that are used in this methodology, this method should not be construed to limit the life of actuators currently set within the Limitorque published ratings to less than 2000 cycles except for SMB-000 actuators with 8620 worm material.

Kalsi Phase I test data provides information required to determine the reduced life for a limited number of actuator configurations based on torques above the nominal Limitorque rating. This is done by comparing the mean time between failures to the fatigue curve for the failed part. This curve is shown in Figure 14 of the Kalsi test report. This allows the equivalent stress in the part to be found. The equivalent stress in the limiting component under the actual field conditions can be found by multiplying the stress under test conditions by the torque measured in the field divided by the torque measured during the test. The allowable number of fatigue cycles is then found from the fatigue curve. This value is divided by a safety factor which varies from 4 to 5.2, depending on the number of replicate tests, to find the design allowable fatigue cycles. For more accuracy, the equation for the fatigue curve is used rather than the plot itself. The procedure for SMB-000, SMB-00, SMB-1 actuators is shown in Appendix A.

The Kalsi testing erodes our confidence that the Limitorque allowable service life of SMB-000 actuators made prior to 1988 (8620 worm material) is really 2000 cycles at 99 ft-lb of torque. The actual service life depends on the stiffness of the valve and spring pack, and on the overall ratio of the actuator and a single number is not appropriate for all SMB-000 actuators. For SMB-000 actuators with more than 99 ft-lb of maximum torque, this paper should already be applied and a conservative service life should be applied. For pre '88 SMB-000 actuators with between 58 and 99 ft-lb of maximum torque it is prudent to determine the service life of the actuator using a methodology which is more conservative than the Limitorque published value. For SMB-000 actuators with 8620 worm material and with a maximum torque of between 58 and 99 ft-lb of torque, Appendix A shall be completed with the factor in step 17 being 1.67×10^{8}. Prior to exceeding the calculated service life for these valves, the 8620 worm shall be replaced.

Torque related failures in components which would be installed in nuclear power plants were not found in SMB-0 actuators throughout the course of the Kalsi testing. The tested actuator ran at 104% torque for 4000 cycles. Using this point and the additional five over-torques to 200% during the Kalsi testing, we can infer a torque limit for the SMB-0 actuator. The following method is to be used to find the design life of SMB-0 actuators with a 37:1 worm ratio at torque values higher than the Limitorque values.

DETERMINE THE STIFFNESS OF THE OVER-TORQUED ACTUATOR AND VALVE

From the VOTES diagnostic test, determine the maximum torque as a percentage of rated torque. Then subtract the torque at C11, expressed as percentage of rated torque, from this. Next determine the number of worm cycles experienced during the testing between C11 and the maximum torque, by using the following formula:

$$\mathit{worm\ cycles} = \frac{\mathit{motor\ speed}\ (RPM)}{60\,\frac{sec}{min}}\ \frac{\mathit{worm\ gear\ ratio}}{OAR}\ \mathit{loaded\ time}\ (sec)$$

Divide the difference in torque between C11 and the maximum value by the number of worm cycles to find the percent increase in torque per worm cycle. This is a measure of how stiff the valve and actuator are.

COMPARE TO KALSI TEST
The Kalsi test fixture had a stiffness of 6.3% per worm turn. If the valve that we have tested has a stiffness less than 6.3% per worm cycle, then the testing at Kalsi would not bound our valve and we would have to derate the allowable cycles calculated below. If our valve has a stiffness greater than 6.3% per worm turn, then the following calculation is conservative.

DETERMINE ALLOWABLE STROKES
The number of allowable strokes can be determined by drawing a straight line on semi-log paper between 200% torque at 5 cycles and 100% torque at 4000 cycles. The equation for this line would be:

$$\textit{Allowable cycles} = \frac{3{,}200{,}000}{\exp[0.066846\ (\%\ \textit{of rated torque})]}$$

If the valve/actuator stiffness is more than 6.3% per worm turn then the above gives the allowable number of strokes. If not, the allowable number of strokes must be derated.

DERATE FOR STIFFNESS (IF REQUIRED)
If the valve/actuator stiffness is less than 6.3% per worm turn the allowable number of strokes must be derated by the following formula:

$$\textit{Derated cycles} = \textit{Allowable cycles}\ \frac{\textit{valve/actuator stiffness}}{6.3\%\ \textit{per worm turn}}$$

5.0 POSITION - Calculation of Remaining Life for Over-Torqued Actuators

This position is applicable for determining when actuator restoration is required for actuators which are found to have been over-torqued in the past but which have been reset to within the Limitorque ratings. The allowable number of cycles is found using the above methods. Miner's rule is used to find the remaining cycles at 100% torque.

$$n_2 = N_2(1 - n_1 / N_1)$$

- n_1 is the actual number of cycles at the higher torque
- N_1 is the allowable number of cycles at the higher torque. This value is found from the calculations used to establish the extended torque rating.
- N_2 is 2,000 strokes, from the original Limitorque ratings.
- n_2 is the allowable number of strokes remaining at 100% torque rating.

6.0 JUSTIFICATION - Thrust Limit Extension

The justification for this position is provided in References 12.1 and 12.2 below. The justification for the thrust extension of SB actuators is given in Reference 12.7 below.

7.0 JUSTIFICATION - Housing Cover Bolt Torque

During the Kalsi test program, bolt pre-load torque values were selected to prevent joint separation during 270% to 280% thrust overload. The higher bolt torque minimizes the potential for fastener failures due to alternating stress as the joint is loaded, since the aim of the Kalsi report is to focus on actuator components. Thus the torque values provided in the Kalsi report and in Limitorque Technical Update 92-01 may be overly restrictive based on plant-specific thrust overload and seismic requirements. Bolt torque values affect the fatigue life of the ***bolts*** and not the fatigue life or wear of the actuator components.

Both the housing cover and actuator base joints are loaded in the valve closing direction. In a rigid joint, such as the actuator base to valve yoke connection, only loads in excess of the bolt pre-load provide significant contribution to the bolt alternating stress. However, for a flexible joint, such as the upper housing cover to upper housing with an intervening gasket, the bolts will see significant alternating stress with little sensitivity to the bolt pre-load. Thus as long as the upper housing cover bolts have some preload, i.e., are not loose, there should be no significant impact on the bolt fatigue life between a light preload and the maximum preload values specified in Appendix A of the Kalsi report.

Kalsi Engineering Report 1752C documents testing and analysis to demonstrate the insensitivity of the bolt preload on fatigue life for the SMB-000 with new and old style housing covers. The SMB-000 actuator fasteners are subjected to the most severe alternating stress magnitude in this series of actuators (from SMB-000 through SMB-1). A subsequent report, 1759C, documents the comparative stresses of fasteners used for this series of actuators and provides recommendations for upper housing cover and actuator mounting bolt torque values.

8.0 JUSTIFICATION - Torque Limit Extension

The Kalsi Phase I test report does not develop generic guidelines for exceeding existing published torque limits. However, the report results can be useful on a case-by-case basis, as illustrated above. The fatigue curve for low alloy steel is well documented. Figure 14 from the Kalsi report is a reproduction of the curve from the criteria for the ASME Codes. (Reference 12.5) By comparing the lifetime of a tested part to the appropriate curve, the effective stress in the tested part can be found. This can be correlated to the torque applied to the actuator during the test to allow the effective stress and the lifetime for the limiting part to be found. The equivalent stress is dependent on the load history of the tested part. In the case of Kalsi's testing the part experienced a linear increase in load during each worm cycle. Without further analysis, this is the only load pattern qualified by Kalsi's testing. This procedure takes the linear loading into account by conservatively enveloping the VOTES torque trace with a linear loading curve.

In order to use the test data as design data, the conservatism prescribed in ASME Section III, Division 1, Appendix II also needs to be included. The ASME methodology allows for applying a correction factor to the number of test cycles or the test stress, or a portion of both correction factors. For simplicity, the correction factor will be applied to the number of test cycles. The correction factors are listed in II-1520(g). For our case, these correction factors are 1.0 except for the statistical variation factor which is:

$$K_{ss} = 1.470 - (0.044)(\text{number of replicate tests})$$

In our case the number of replicate tests could conservatively be represented by one less than the number of failures per actuator. This is 0 for the 8620 material in the SMB-00, 1 for the 4320 material in the SMB-000, 0 for the failure in the SMB-00, and 2 for the SMB-1. The correction factor for the number of cycles is:

$$K_n = K_s^{4.3}$$

In our case, K_n ranges from 4 for the SMB-1 to 5.24 for the SMB-000 and SMB-00. These safety factors are applied to the number of cycles allowed based the results of Kalsi testing.

This data could be read from Figure 14 in the Kalsi report, however for greater accuracy the maximum stress in the tested part and the allowable life inferred from the test will be calculated based on the equation for the fatigue curve.

These interpolations are shown in Appendix B. In addition Appendix C provides a list of facts, assumptions and conclusions used in arriving at this position.

9.0 JUSTIFICATION - SMB-0 Overtorque

During Kalsi testing of SMB-0 actuators with 37:1 worm gear ratios the only torque related failure which was found was the motor pinion key. This failure has already been addressed in the industry by replacement of the lower strength keys with higher strength keys. Failure of other torque related components was not experienced during 5 test strokes at 201% rated torque or during 4000 test strokes at 104% rated torque. Therefore, these two points can be taken as two minimum service life points. The allowable life of the SMB-0 actuators with 37:1 worm sets can the be interpolated between these two points on a semi-log basis. The equation for this interpolation is as shown above.

Kalsi Report 1707C Rev. 0 states that the differences between the test jig and the actual installation must be evaluated. By ensuring that the actual valve stiffness is higher than the test fixture stiffness, we ensure that the worm, worm shaft, bearings and worm gear teeth in the tested actuator received more severe loading than our actual actuator did. If our valve is not as stiff as the test fixture, it would receive a proportionally higher amount of fatigue cycles in the torque related components during each valve stroke. By derating the life of our actuator in proportion to the ratio of the actual valve stiffness to the test fixture stiffness we account for these extra cycles.

The value of 6.3% per worm turn for valve/actuator stiffness is from Kalsi test data. In order to develop 82,707 pounds of thrust at 1005 ft-lb of torque with a 500 kip per inch test rig spring pack, the valve stem would deflect the spring pack by
82,707 / 500,000 = .165414 inches.

With a .2 inch lead the drive sleeve would have to rotate .165414 / .2 = .82707 revolutions to move the stem .165414 inches. With a 37:1 worm ratio the worm would have to turn .82707 (37) = 30.60 turns to achieve the stated thrust.

In addition the worm would displace the spring pack to achieve the 1005 ft-lb torque. From the Limitorque manual the 0501-184 spring pack produces 500 ft-lb at a setting of 3.0 and 200 ft-lb at a setting of 1.5. From Limitorque training material, this corresponds to displacements of .321" and .1145", respectively. Extrapolating from these two points we find that the worm displaces .6686". From previous measurements, the worm has a pitch of 0.3906". The number of turns of the worm required to move it .6686" is .6686 / .3906 = 1.71. The total number of worm turns to achieve 201% torque is 1.71 + 30.60 = 32.31.

The stiffness of the test fixture is then 201% / 32.31 = 6.22 < 6.3% per worm turn.

10.0 JUSTIFICATION - Estimating Remaining Life

The method for calculating the extended ratings is justified above. The use of Miner's rule for fatigue life is a standard engineering practice. As an example it is used in the ASME piping code. The use of the 2000 cycles for the life of Limitorque actuators should be conservative. Data from Kalsi testing listed above supports higher torque ratings.

11.0 JUSTIFICATION - Reducing Service life for Pre '88 SMB-000 Actuators

The Kalsi testing erodes our confidence that the Limitorque allowable service life of SMB-000 actuators made prior to 1988 (8620 worm material) is really 2000 cycles at 99 ft-lb of torque. The actual service life depends on the stiffness of the valve and spring pack, and on the overall ratio of the actuator and a single number is not appropriate for all SMB-000 actuators. Calculation of the service life using .5 second seating time (somewhat conservative) shows that a typical actuator with a 100:1 OAR would be expected to have aproximately a 162 stroke service life at 99 ft-lb. Even without the 5.24 safety factor this would be less than one half of the service life assigned by Limitorque. Based on this, we are compelled to determine if the service life would be reduced below the Limitorque published values when the individual data for the susceptible valves is used. Using the above assumptions a survey of typical valves show that valves with less than 58 ft-lb of final torque should not be a concern. For SMB-000 actuators with more than 99 ft-lb of maximum torque, this paper should already be applied and a conservative service life should be applied.

For pre '88 SMB-000 actuators with between 58 and 99 ft-lb of maximum torque it is prudent to determine the service life of the actuator using a methodology which is more conservative than the Limitorque published value. Since we have two data points, the Limitorque rating and the Kalsi test data, each of which have a confidence attached to them and which should intersect to form a range of possible values, we need to determine how to weight these two data points to determine the most probable service life and then provide a reasonable safety factor on this life. If we assume that Limitorque tested at least the same number of SMB-000 actuators that Kalsi tested and did not derate the tested data with a safety factor, we can assign equal confidence to both of these ratings. This paper, without the safety factor would assign a service life of aproximately 800 cycles where Limitorque allows 2000 cycles; the average would be 1400 cycles. If we use this value and assign a safety factor of 2 our limit would be aproximately 700 cycle at 99 ft-lb for an actuator with .5 second seating time and a 100:1 OAR. This is 4.3 times higher than the value that would be calculated by this procedure for calculating reduced service life at higher output torques. Therefore, for SMB-000 actuators with 8620 worm material and with a maximum torque of between 58 and 99 ft-lb of torque, Appendix A shall be completed with the factor in step 17 being 1.67×10^8. Prior to exceeding the calculated service life for these valves, the 8620 worm shall be replaced. This position is more conservative than the current industry practice but is required in light of the data that has been received from the Kalsi testing.

12.0 REFERENCES

12.1 Thrust Rating Increase of Limitorque SMB-000, SMB-00, SMB-0, and SMB-1 Actuators, Document No. 1707C. Rev. 0, November 25, 1991, Kalsi Engineering, Inc.

12.2 Letter to A. Bert Davis, NRC Region III, from Mary Beth Depuydt, CECo Nuclear Licensing Administrator, dated February 18, 1993. Subject: LaSalle County Station Units 1 and 2, Response to Questions Engineering Report, Docket Nos. 50-373 and 50-374.

12.3 Kalsi-Limitorque Phase II Overload Testing, November 1992 Progress Report, Attachment 1, page 2.

12.4 Kalsi-Limitorque Phase II Overload Testing, December 1992 Progress Report, Standard Response to NRC Questions.

12.5 Criteria of the ASME Boiler & Pressure Vessel Code for Design by Analysis in Sections III and Viii, Division 2, 1969, The American Society of Mechanical Engineers, New York, NY.

12.6 WP_MOV_156 Revision C draft, dated June 30,1994

12.7 June 14, 1993 letter from G. A. Moran, Kalsi Engineering Inc. to S. Chiu, Texas Utility; and S. Chiu clarificationin 12-8-93 10:00 AM telecon with I. A. Garza confirming that the load applied to the SB components was 162% for 2000 cycles.

12.8 Offical transmittal of "Review and Comments on Commonwealth Edison Company's Methodology on Torque Rating Extension of Limitorque Actuators" from M. S. Kalsi, Kalsi Engineering to Ivo Garza, Commonwealth Ed, dated September 12, 1994. This was originally transmitted October 7, 1993.

12.9 Comments from Engineering and Management Specialists Inc. from Y. A. Patel to O. Shirani dated July 29, 1994.

Appendix A

Worksheet for Determining The Effect of Torque on the Allowable Number of Valve Cycles Using the Torque or Thrust Trace

Torque information from:

_____ torque trace

_____ thrust trace (assumed stem factor: ________)

ACTUATOR ENDURANCE LIMITS

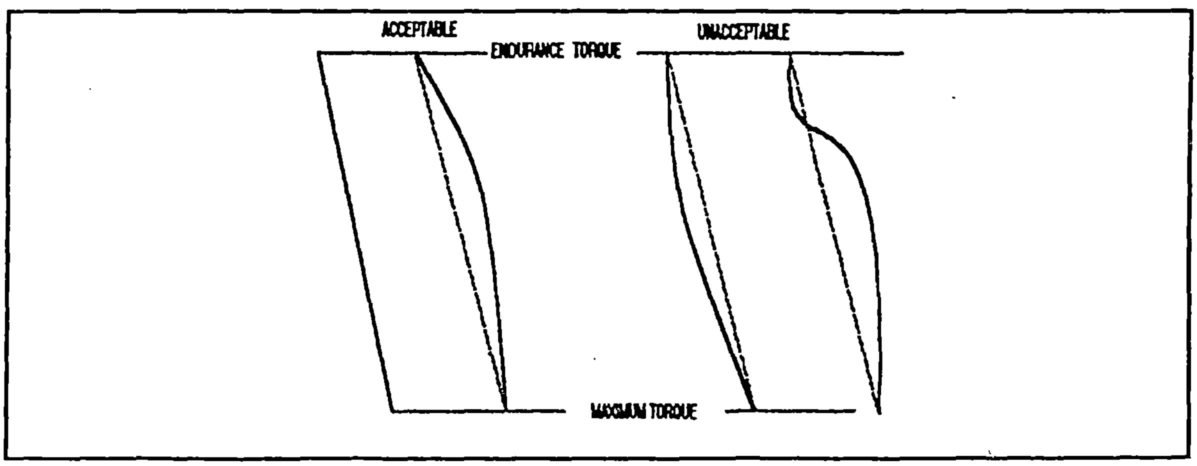
ACTUATOR	TORQUE ENDURANCE LIMIT
SMB-000	44.6 FT-LB
SMB-00	167 FT-LB
SMB-1	650 FT-LB

1) Obtain a hard copy of the VOTES trace.

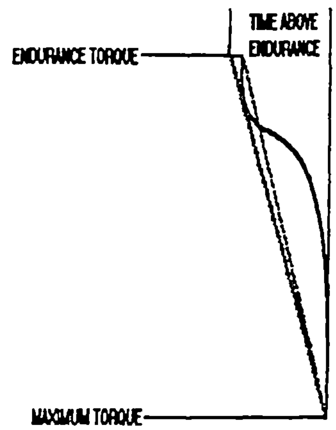
2) Note the endurance limit of the actuator from the above table. ___________ft-lb

3) On the VOTES trace identify the point where the torque exceeds the endurance limit. Draw a straight from this point on the curve to the point of maximum torque.

4) a) Does the trace go straight or concave downward to the point of maximum torque? See below for examples.



b) If the line does not go straight or concave downward as the examples on the left do, then draw a second line from the maximum torque point which just touches the left-most point on the VOTES trace. See the sketch below.



Note the time where this line crosses the endurance torque. ____________sec.

c) If the trace was straight or concave downward, note the time where the trace crosses the endurance torque. ____________sec.

5) Note, from the VOTES trace, the time where the torque is maximum. ____________sec.

6) Subtract the time in step 4 from the time in step 5 to find the time above the endurance limit. ____________ - ____________ = ____________ sec.

7) Note the overall actuator ratio ____________.

8) Note the worm / worm gear ratio. ____________.

9) Find the motor pinion / worm shaft gear ratio by dividing the overall actuator ratio by the worm / worm gear ratio. ____________ / ____________ = ____________.

10) Note the nominal motor speed. ____________ RPM

11) Find the worm speed by dividing the nominal motor speed by the motor / worm shaft gear ratio from step 9 and then by 60 seconds per minute.
____________ / ____________ / 60 = ____________RPS

12) Multiply the worm speed by the time above the endurance limit from step 6 to find the number of worm turns above the endurance limit.
_____________ X _____________ = _____________ turns

13) Note the maximum torque from the VOTES trace. _____________ ft-lb

14) Find the slope of the torque curve by subtracting the endurance torque from the maximum torque and then dividing by the number of worm turns from step 12.
(___________ - ___________) / _____________ = ft-lb/turn

15) Subtract the endurance torque from the maximum torque, adjusted for measurement accuracy and then cube the difference.
$(\_\_\_\_\_\_\_\_\_\_\_ - \_\_\_\_\_\_\_\_\_\_\_)^3$ = _____________

16) Divide the slope from step 14 by the value in step 15.
___________ / ___________ = ___________

17) Calculate the allowable number of worm cycles by multiplying the value in step 16 by the factor from the following table.
allowable worm cycles = ___________ X ___________ = ___________ cycles

ACTUATOR SIZE	STEP 17 FACTOR
SMB-000 (PRE '88)	3.88×10^7
SMB-000 (4320)	1.20×10^8
SMB-00	5.05×10^8
SMB-1	1.08×10^{10}

18) Review the remainder of the VOTES trace for the complete open-to-close-open stroke to determine if the actuator experienced any other time above the endurance torque. If so, note the cumulative additional time that the operated above the endurance torque.__________ sec.

19) Multiply this by the worm speed from step 11 to find the number of worm turns above the endurance limit. _____________ X ___________ = ___________ turns

20) Add this to the number of worm turns found in step 12.
__________ + __________ = ___________ worm turns per stroke

21) Divide the allowable number of worm turns from step 17 by the number of worm turns per stroke from step 20 to find the allowable number of valve cycles at the tested torque level. Allowable number of valve cycles:

____________ / ____________ = ____________

Comments ____

____________________ / ________ ____________________ / ________

Preparer Date Reviewer Date

WORKSHEET FOR SMB-0

Torque information from:

_____ torque trace

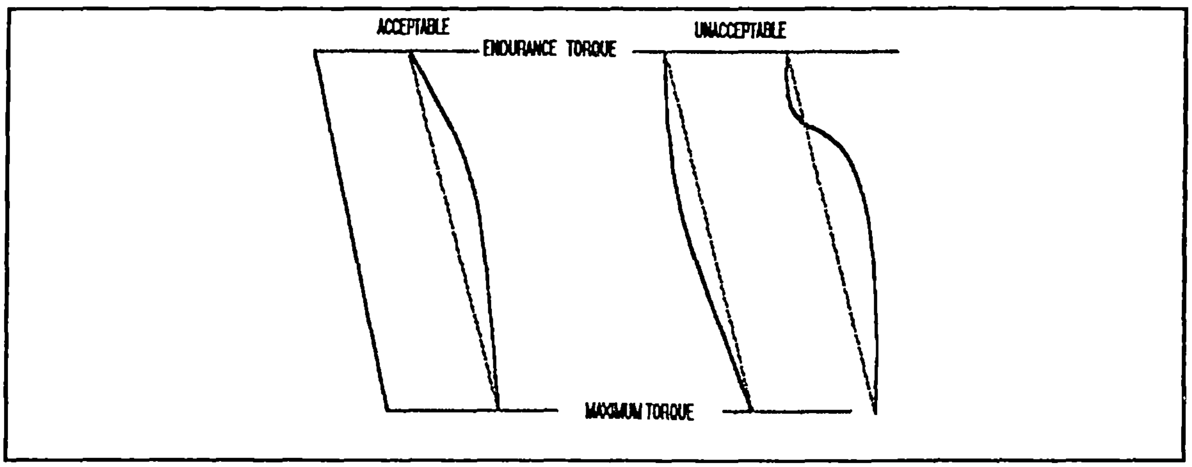
_____ thrust trace (assumed stem factor: _________)

1) Obtain a hard copy of the VOTES trace.

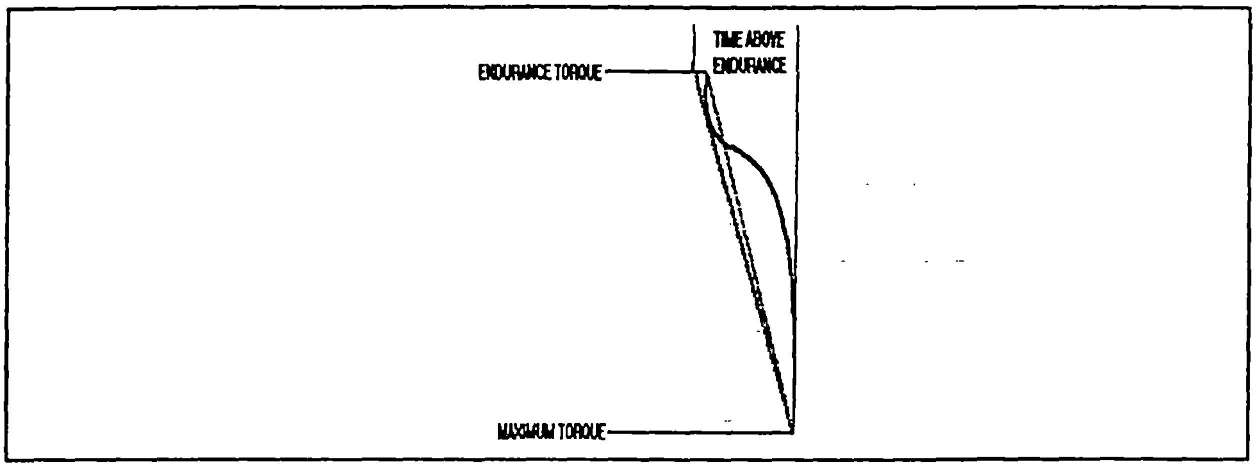
2) Note the value of C11 from the trace ___________ft-lb

3) On the VOTES trace draw a straight from C11 to the point of maximum torque. After C15 the motor slows down and this line does not need to follow the trace

4) a) Does the trace go straight or concave downward to the point of maximum torque? See below for examples.



b) If the line does not go straight or concave downward as the examples on the left do, then draw a second line from the maximum torque point which just touches the left-most point on the VOTES trace. See the sketch below.



Note the time where this line crosses C11. ___________sec.

c) If the trace was straight or concave downward, note the time where the trace crosses C11 . ___________sec.

5) Note, from the VOTES trace, the time where the torque is maximum. ___________sec.

6) Subtract the time in step 4 from the time in step 5 to find the time above the endurance limit. ___________ - ___________ = ___________ sec.

7) Note the overall actuator ratio ___________.

8) Note the worm / worm gear ratio. ___________.

9) Find the motor pinion / worm shaft gear ratio by dividing the overall actuator ratio by the worm / worm gear ratio. ___________ / ___________ = ___________.

10) Note the nominal motor speed. ___________ RPM

11) Find the worm speed by dividing the nominal motor speed by the motor / worm shaft gear ratio from step 9 and then by 60 seconds per minute.
___________ / ___________ / 60 = ___________RPS

12) Multiply the worm speed by the time above the endurance limit from step 6 to find the number of worm turns above the endurance limit.
_____________ X _____________ = _____________ turns

13) Note the maximum torque from the VOTES trace. _____________ ft-lb

14) Find the slope of the torque curve by subtracting the C11 torque from the maximum torque and then dividing by the number of worm turns from step 12.
(_____________ - _____________) / _____________ = ft-lb/turn

15) Divide the slope from step 14 by 500 ft-lb / 100%
_____________ / 5 = _____________ % / turn

16) Divide the slope in step 15 by 6.3 % / worm turn
_____________ / 6.3 = _____________

17) Calculate the nominal allowable cycles by inserting the maximum torque, adjusted for measurement accuracy, into the following equation:
$3.2x10^6$ / exp(.066846_____________ / 500) = _____________

18) If the value in step 16 is greater than 1 divide the nominal allowable cycles by the value from step 16 otherwise divide by one.
_____________ / _____________ = _____________ cycles.

Comments ____
____
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_____________ / _____ Preparer Date

_____________ / _____ Reviewer Date

Appendix B

CORRELATION OF EQUIVALENT MAXIMUM STRESS TO ACTUATOR TORQUE AND ALLOWABLE LIFETIME

The equation for the fatigue curve for low alloy steel is:

$$S = \frac{E}{4\sqrt{N}} \ln\left[\frac{100}{100-61.4}\right] + 38{,}500$$

Where S is the alternating stress, N is the allowable number of cycles to failure, and E is Young's modulus. Rearranging this to find the allowable number of cycles:

$$N = \frac{E^2}{16\,(S-38{,}500)^2}\left(\ln\left[\frac{100}{100-61.4}\right]\right)^2$$

The cumulative damage would be the number of cycles at a given load divided by N, the allowable cycles at that load. The total cumulative damage, CD, for a ramp loading is:

$$CD = \int_{S_e}^{S_0} \frac{R_0\,dS}{S_0 N}$$

During each stroke in the Kalsi test fixture there were four loading cycles so that we have to multiply the above value by four. If we do that and if we substitute in the expression for N we get:

$$CD = \frac{64\,R_0}{S_0 E^2\left(\ln\left[\frac{100}{100-61.4}\right]\right)^2}\int_{S_e}^{S_0}(S-38{,}500)^2\,dS$$

$$CD = \frac{64\,R_0}{3\,S_0 E^2\left(\ln\left[\frac{100}{100-61.4}\right]\right)^2}(S_0-38{,}500)^3$$

The term containing the endurance stress minus 38,500 is excluded from the above equation since it is insignificant. Inserting the value of Young's modulus, $E = 30X10^6$ and recognizing that S_e, the endurance limit is approximately equal to 38,500, we get:

$$CD = 2.62\times10^{-14}\,\frac{R_0}{S_0}\,(S_0-38{,}500)^3$$

Noting that the fatigue life, L, is one over the cumulative damage per stroke, CD, we can write the equation for worm life as:

$$L = \frac{1}{CD} = 3.82 x 10^{13} \frac{S_0}{R_0 (S_0 - 38,500)^3}$$

In the case of the Kalsi testing the life and the number of cycles per ramp is known. We can then rearrange the above equation to find the equivalent maximum stress in the failed part:

$$S_0 = 2.62 x 10^{-14} R_0 L (S_0 - 38,500)^3$$

The life of the limiting parts in the various actuators, the corresponding number of cycles per ramp and the equivalent maximum stress are tabulated below:

ACTUATOR	R_0	L	S_0 (psi)
SMB-000	32.0	755	98,893
SMB-00	83.3	3774	57,637
SMB-1	61.2	1285	71,069

From this result and the torque applied to the actuator during the test, we can calculate the equivalent maximum stress in the limiting part as a function of actuator torque. Since the stress in the part is proportional to the actuator torque we can use a simple ratio to find the stress in the limiting component at other actuator torques. The equivalent maximum stress at other torques would be:

$$S_{MAX} = S_0 \frac{TORQUE_{MEASURED}}{TORQUE_{TESTED}}$$

The torque which produces the endurance limit stress can also be found by the ratio of the endurance stress to the equivalent maximum stress multiplied by the tested stress:

$$TORQUE_{ENDURANCE} = \frac{38,500}{S_0} TORQUE_{TESTED}$$

If we substitute these values back into the equation for worm life we will be able to find the life of an actuator which is torqued to a given value.

The equation for cumulative damage was:

$$CD = \frac{16\,(4R_0)}{3\,S_{MAX}E^2\left(\ln\left[\frac{100}{100-61.4}\right]\right)^2}\,(S_{MAX}-38{,}500)^3$$

If we substitute in "n" as the number of cycles in the ramp from zero torque to maximum torque and the above expressions for stress in terms of measured actuator torque we get:

$$CD = \frac{16\,n\,S_0^{\,2}}{3\,TORQUE_{MEASURED}\,E^2\left(\ln\left[\frac{100}{100-61.4}\right]\right)^2}\;\frac{(TORQUE_{MEASURED}-TORQUE_{ENDURANCE})^3}{TORQUE_{TESTED}^{\,2}}$$

Recognizing that the allowable life is 1 over the cumulative damage, substituting in the value of Young's modulus and combining the constants we get:

$$L = 1.529\times10^{14}\left(\frac{TORQUE_{TESTED}}{S_0}\right)^2\frac{TORQUE_{MEASURED}}{n\,(TORQUE_{MEASURED}-TORQUE_{ENDURANCE})^3}$$

We need to reduce the allowable life by the safety factor discussed in the body of this paper:

$$L_{Design} = \frac{1.529\times10^{14}}{Safety\ Factor}\left(\frac{TORQUE_{TESTED}}{S_0}\right)^2\frac{TORQUE_{MEASURED}}{n\,(TORQUE_{MEASURED}-TORQUE_{ENDURANCE})^3}$$

We can combine the constants into one constant for each actuator to get:

$$L_{Design} = C\,\frac{TORQUE_{MEASURED}}{n\,(TORQUE_{MEASURED}-TORQUE_{ENDURANCE})^3}$$

The safety factor, tested torque divided by the S_0 squared and the combined constant for each actuator are tabulated below:

ACTUATOR	$TORQUE_{TESTED}$	$\frac{TORQUE^2_{TESTED}}{S_0^{\,2}}$	SAFETY FACTOR	COMBINED FACTOR
SMB-000	105	1.33×10^{-6}	5.242	3.88×10^{7}
SMB-00	240	1.73×10^{-5}	5.242	5.05×10^{8}
SMB-1	1200	2.85×10^{-4}	4.02	1.08×10^{10}

With everything else in the actuator being the same, the lifetime for the SMB-00 with the 4320 material would be the lifetime for the SMB-000 with the 8620 material multiplied by the difference in tested lifetime and the safety factor. Since there was one replicate test for the 4320 material the safety factor would be 4.600. The average life between failures during the Kalsi testing was 2,053 for the 4320 material compared to 755 for the 8620 material. The 4320 material would have a design life of 3.10 times the design life of the 8620 material. The combined factor for the SMB-000 with a 4320 worm would then be 1.20×10^8.

We can double check this by finding the equivalent maximum stress that would produce a mean life of 2053 cycles.

$$S_0 - 2.62x10^{-14} R_0 2053 (S_0 - 38{,}500)^3$$

If we solve this for S_0 we find the equivalent maximum stress equals 73,466 psi. We know that the stress for this worm was really the same as it was for the 8620 material. We know that the fatigue life for alloy steel is a function of the ratio actual stress to tensile stress of the material. The equivalent maximum stress we calculate is normalized to the tensile stress for the material used to develop the fatigue curve. Putting these two pieces together we would expect the tensile strength for 4320 material would be 90,893/73,466 = 1.24 times the tensile strength of 8620 material. From the 24th Edition of Machinery's Handbook we find the tensile strength of 4320 to be 115,000 psi and the tensile strength of 8620 to be 91,750 psi. The ratio of these values is 1.25. This confirms that it is reasonable to expect the worm made from 4320 material to fail at 2053 cycles while the worm made from 8620 fails at 755 cycles.

APPENDIX C
INDUSTRY DATA AVAILABLE ON LIMITORQUE ACTUATOR RATINGS

FACTS:	USES:
SMB-000	
Ramp loading to 117% rated torque. A loading and unloading event occurs at each end of the cycle.	Used to determine which fatigue equations are appropriate. Used to normalize the stress in the failed part against the percent of rated torque. WP122.
128 total worm turns during the load events in each cycle.	Used to normalize the number of cycles actually seen during a valve stroke. WP122
8620 steel worm fails after 755 cycles. Worm and worm gear replaced.	Data point used in torque limit paper. WP122
5 stall cycles 127% rated torque were incurred.	Conservatively excluded when we determined load between failures.
Worm gear replaced due to wear after 1756 cycles.	Worm gear wear not shown as limiting component in WP122. Wear did not cause failure. WP122 requires replacement of all torque related components, including worm gear, before the end of the service life of the most limiting component, the worm.
4320 steel worm fails after 2458 cycles. Worm and worm gear replaced.	Data point used in torque limit paper. WP122
Torque switch spring failed after 3795 cycles.	Not shown as limiting component in WP122. Worm is much more limiting.
Worm gear replaced due to wear after 802 cycles.	Worm gear wear not shown as limiting component in WP122. Wear did not cause failure. WP122 requires replacement of all torque related

	components, including worm gear, before the end of the service life of the most limiting component, the worm.
4320 steel worm fails after 1648 cycles. Test concluded.	Data point used in torque limit paper. WP122

SMB-00

Ramp loading to 96% rated torque. A loading and unloading event occurs at each end of the cycle.	Used to determine which fatigue equations are appropriate. Used to normalize the stress in the failed part against the percent of rated torque. WP122.
333.2 total worm turns during the load events in each cycle.	Used to normalize the number of cycles actually seen during a valve stroke. WP122
5 stall cycles to 174% rated torque were incurred.	Conservatively excluded when we determined load between failures.
Housing crack detected using liquid penetrant test after 2000 cycles.	Not included in torque limit White Paper. The location of the cracks are stressed by thrust.
Two more housing cracks found during liquid penetrant testing after seismic qualification after 2081 cycles.	Not included in torque limit White Paper. The location of the cracks are stressed by thrust.
Worm failed after 3774 cycles. Worm and worm gear replaced.	Data point used in torque limit paper. WP122
Test concluded after 4076 cycles - no growth in the detected cracks.	No relevant data is extracted from this fact.

SMB-0

Ramp loading to 104% rated torque. A loading and unloading event occurs at	Used to determine which fatigue equations are appropriate. Used to

each end of the cycle.	normalize the stress in the failed part against the percent of rated torque. WP122.
71 total worm turns during the load events in each cycle. The first 650 cycles were performed with less stiff spring and 284 worm turns would have occurred during the load events in each of these cycles.	Used to normalize the slope of the stress per cycle ramp between the tested condition and the conditions experienced in the field. The additional number of loaded worm turns were conservatively excluded. WP122
Motor pinion gear key sheared after 161 cycles.	Included in White Paper 156 on the actions to be taken to address the 1018 carbon steel motor pinion key problem. Separared from WP 122 since the sress in the key is proportional to motor torque and not actuator torque. Motor torque may be a problem even if actuator torque is not, or motor torque may not be a problem even if actuator torque is.
Motor bearing failed after 163 cycles.	Not icluded in WP 122. Kalsi report attributed this to an assembly error.
Motor replaced after 395 cycles. Off line test showed motor to be satisfactory.	Not included in White Papers. Motor was subsequently found to be in working order. The stem lubrication was found to be the source of the lower than expected thrust. ComEd testing supports the coefficient of friction being used at ComEd.
Motor pinion gear key sheared after 290 cycles.	Included in White Paper 156 on the actions to be taken to address the 1018 carbon steel motor pinion key problem. Separared from WP 122 since the sress in the key is proportional to motor torque and not actuator torque. Motor torque may be a problem even if

	actuator torque is not, or motor torque may not be a problem even if actuator torque is.
Motor pinion gear key found cracked after 47 cycles. replaced with 4140 steel key.	Included in White Paper 156 on the actions to be taken to address the 1018 carbon steel motor pinion key problem. Separared from WP 122 since the sress in the key is proportional to motor torque and not actuator torque. Motor torque may be a problem even if actuator torque is not, or motor torque may not be a problem even if actuator torque is.
5 stall cycles to 201% rated torque were incurred.	White Paper 122 Uses this as the proven service life of the actuator. This conservatively ignores the the fact that the same actuator also endured 4000 strokes at 104% rated torque.
Crack detected by liquid penetrant testing in housing cover after 2500 cycles.	Not included in torque limit White Paper. The location of the cracks are stressed by thrust.
Test concluded with no growth in the housing cracks after 4001 cycles.	White Paper 122 uses this as the proven service life of the SMB-0 actuator. This conservatively ignores the fact that the same actuator also endured 5 strokes at 201% rated torque.
SMB-1	
Ramp loading to 141% rated torque. A loading and unloading event occurs at each end of the cycle.	Used to determine which fatigue equations are appropriate. Used to normalize the stress in the failed part against the percent of rated torque. WP122.
244.8 total worm turns during the load	Used to normalize the number of cycles

events in each cycle.	actually seen during a valve stroke. WP122
Worm shaft failure after 1974 cycles.	Data point used in torque limit paper. WP122
5 stall cycles to 194% rated torque were incurred.	Conservatively excluded when we determined load between failures.
Tripper pin discovered broken after 2374 cycles.	Not included as a limiting component in White Paper 122. Declutch tripper pin had longer life than the worm shaft. Cause of tripper pin failure does not appear to be related to actuator torque.
Worm shaft bushing replaced after 505 cycles.	Inspection requirements have been added to WP122.
Worm shaft bushing replaced after 232 cycles.	Inspection requirements have been added to WP122.
Worm shaft bushing replaced after 310 cycles.	Inspection requirements have been added to WP122.
Worm shaft failure after 1167 cycles.	Data point used in torque limit paper. WP122
Worm shaft bushing replaced after 173 cycles.	Inspection requirements have been added to WP122.
Worm shaft bushing replaced after 277 cycles.	Inspection requirements have been added to WP122.
Worm shaft bushing replaced after 226 cycles.	Inspection requirements have been added to WP122.
Worm shaft failure after 714 cycles.	Data point used in torque limit paper. WP122
Test concluded after 4000 cycles.	No relevant data extracted from this.

GENERIC

Additional loaded worm turns occurred during Kalsi testing. The method used to calculate loaded worm turns did not include the turns required to compress the spring pack.

These additional cycles were conservatively excluded for the SMB-000, 00 and 1 actuators. Due to the difference in methodology on the SMB-0 actuator these cycles were included in calculating the slope of the torque versus worm turn curve.

Seismic acceleration was applied to the actuators while under load.

Inclusion of seismic load testing is appropriate since the valves in the field may also have to undergo seismic loading when closed.

INPUT INTO WP-122

FACT:	CONCLUSION:
Ramp loadup used in destructive life testing.	Equation for fatigue life versus limiting part stress must be based on a ramp loading.
Actual loadup may not be linear ramp	Must envelope actual loadup with an equivalent linear ramp which will not under predict the number of cycles at load.
Multiple failures seen in torque related components during destructive testing.	The more repeated failures seen during testing, the more data we have to predict the service limits of the actuators. Use the failure data to correlate torque and service life.
Actuator torque failures not present in SMB-0 after 4001 cycles at 104% rated torque and an additional 5 cycles at 201% rated torque.	Use these two points as proven service life points. Interpolate the service life at intermediate torques based on these two points.
Slope of loadup ramp in destuctive testing may differ from that in the field.	Develop an equation to adjust for the differences in the slope.

FORMULAE:	SOURCE:
$S = \frac{E}{4\sqrt{n}} \ln\left(\frac{100}{100-61.4}\right) + 38,500$	ASME Div.2 Fatigue curve for Low-Alloy Steels, reprinted as Figure 14 in Kalsi report. This formula is a best fit of that curve. Note exact numbers aren't critical since the S-N curve is normalized to match the actual testing

$$CD = \sum \frac{n_i}{N_i}$$

Miner's rule

$$CD = \int_{S_e}^{S_0} \frac{R_0\, dS}{S_0 N}$$

Miner's Rule assuming no cmulative damage below the threshold stress (S_0) and ramp loading with a slope of R_0/S_0 revolutions per psi. This also assumes that the worm receives a full fatigue cycle per revolution.

$$L = \frac{1}{CD}$$

Derived: If one event causes a certain fraction (f) of the useful life to be used, then the structure could endure 1/f events without exceeding the useful life.

$$S_{torque} = Torque \frac{S_{test}}{Torque_{test}}$$

This assumes that the geometry of the part in question remains constant.

$$SafetyFactr = 1.470 - 0.044\, Replicates$$

From ASME, Section III, Division 1, Appendix II. This accounts for statistical variations in the test. Material and geometry between the test and the design must be the same.

ASSUMPTIONS:

JUSTIFICATION:

Ramp loading is experienced during loading in the field.	This is verified from the diagnostic trace. Where ramp loading is not seen, WP122 instructs the user to draw an enveloping ramp which provides more loaded cycles than the actuator actually experienced
The parts which failed during the Kalsi test and not other parts in the actuator can be used to limit the service life of the actuator.	Where torque related failures occurred, there were multiple failures, with the exception of the SMB-00. This indicates that the testing did identify the limiting component.
The geometry of the limiting component is the same in the test and in the field.	The limiting components would be the same provided that the worm and worm gear are the same. WP-122 is restricted to the worm gear ratios tested by Kalsi.
The stress levels in the limiting components at a given actuator torque value are the same in the field as they are in the test.	Again the worm and worm gear ratio must be the same.
ADDITIONAL DATA:	**USES:**
Discussion with Site support engineers revealed no actuator torque failures in the time that we have been diagnostic testing. Twisted worm shafts in actuators which were overtorqued but still operating have been noted. These were observed during inspections and recommended torque bearing parts replacements.	Provides confidence in the WP-122 torque limits.
Motor pinion key failures related to motor torque have been reported.	Provides confidence that WP-156 position to prioritize and replace 1018 motor pinion keys is correct.

COMPARISON OF TORQUE LIMITS WITH LIMITORQUE PUBLISHED VALUES

TYPICAL SMB-000 (8620 WORM)

Torque	WP122	LIMITORQUE
200%	4	1
110%	162	2000

TYPICAL SMB-000 (4320 WORM)

Torque	WP122	LIMITORQUE
200%	13	1
110%	502	2000

TYPICAL SMB-00

Torque	WP122	LIMITORQUE
200%	5	1
110%	454	2000

TYPICAL SMB-1

Torque	WP122	LIMITORQUE
200%	6	1
110%	1164	2000

TYPICAL SMB-0

Torque	WP122	LIMITORQUE
200%	5	1
110%	2050	2000

CONCLUSION:
At moderate overtorques WP-122 is typically more conservative than Limitorque's published values. The 2050 cycle allowable at 110% for the SMB-0 is based on seeing no actuator torque failures in the SMB-0 actuator during 1604 loading cycles in the Kalsi test stand. Limitorque has no values of cycle life between 110% and 200%. At 200% torque WP-122 predicts between 3 and 6 cycles than Limitorque's published value of 1. For the SMB-000 with the 4320 worm WP-122 limits the life at 200% torque to typically 13 cycles. The word typical must be used in these comparisons since Limitorques published values ignore difference in worm material, differences in valve stiffness and provide no consideration for the way in which the actuators are loaded.

Motor torque based failures of the motor pinion key are addressed in WP-156, which prioritzes the order in which 1018 motor pinion keys should be replaced.

MPR
ASSOCIATES INC.
ENGINEERS

Attachment B to
MPR Enclosure of
November 15, 1994

Allowable Thrust Fatigue Life Curve for Limitorque SMB-000, SMB-00, SMB-0, and SMB-1 Actuators

Figure 1 from
Thrust Rating Increase of
Limitorque SMB-000, SMB-00, SMB-0, and SMB-1 Actuators

Kalsi Engineering, Inc. Report 1707C, Rev. 0
November 25, 1991

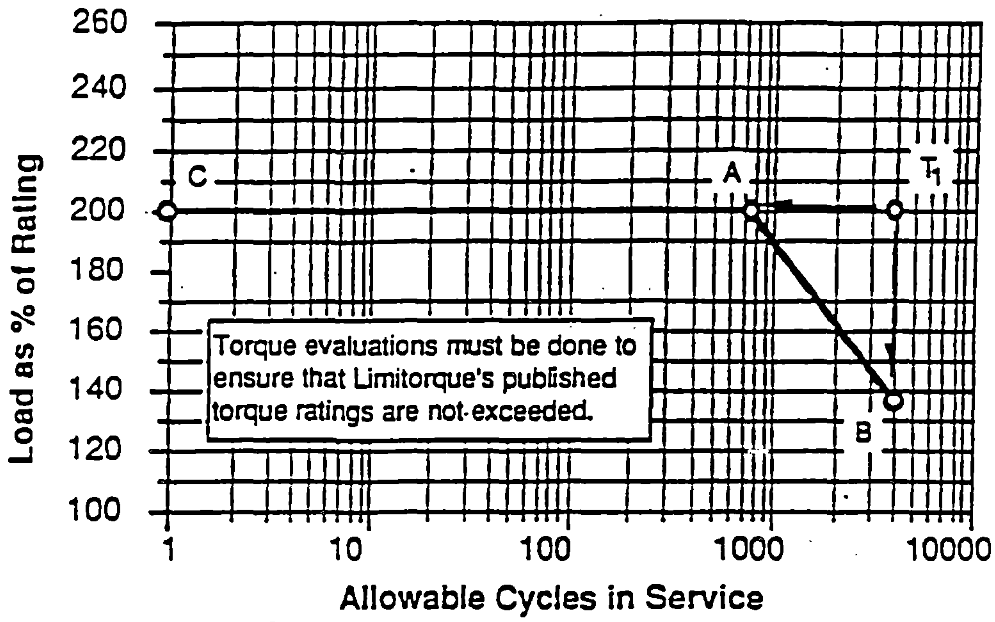

Figure 1
Allowable Thrust Fatigue Life Curve (C-A-B) for Limitorque SMB-000, SMB-00, SMB-0, and SMB-1 Actuators

ATTACHMENT

5

MPR
ASSOCIATES INC.
ENGINEERS

December 6, 1994

Mr. Paul Dietz
Commonwealth Edison Company
1400 Opus Place, Suite 400
Downers Grove, IL 60515

Subject: Review of White Paper WP-129, "MOV Design Margin Evaluation and Diagnostic Test Feedback Evaluation"

Enclosure: Review Report for Review of Commonwealth Edison WP-129

Dear Mr. Dietz:

Enclosed is a report of our review of the subject white paper. Based on our review of this white paper and our discussions with Mr. I. Garza and Mr. B. Bunte, we conclude that the December 15, 1993 version of the paper (which we reviewed) needs substantial upgrading and improvement to:

(1) reflect the way that MOV margin evaluations are actually being performed;

(2) clarify the definitions of terms and describe how values for the terms are determined;

(3) clarify the criteria which are used to determine which margin category an MOV belongs to; and

(4) justify the approach that the conservatisms in the "design parameters" used in the evaluations are sufficient to exclude considerations of random uncertainties in operability evaluations.

Based on our discussions with Mr. Garza and Mr. Bunte, it appears that a considerable amount of work has been done in the above areas, and we understand that the white paper is in the process of being revised. The information in the enclosure can be used to assist in the revision process.

Please call if you have any questions or comments.

Sincerely,

Mitchell Albers

Mitchell Albers

cc: I. Garza (w/encl.)
B. Bunite (w/encl.)

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MOV has not been tested. However, the method by which valve factor is determined from other data is not addressed. For example, use of data from an isolated valve test may be inappropriate (non-conservative) since considerable valve-to-valve variations are known to occur. Methods which consider a range of data such as those described in WP-154 and WP-160 should be specified.

- Figure 1 of WP-129 indicates that a risk-based approach is used for prioritizing the safety importance of each MOV included in the Generic Letter 89-10 scope. However, prioritization of MOVs based on risk importance is not discussed in WP-129. The white paper where this prioritization is justified should be referenced.

- The criteria by which the margin is classified as high, medium, low or none needs to be definitively stated. For example, WP-129 states, "A high margin valve would typically have greater than 35 percent margin to the minimum required thrust and greater than 10 percent margin to the maximum allowable thrust." It is not clear whether the 35 percent and 10 percent values are hard criteria for determining high margin or whether they are simply indicative yardsticks. Our understanding is that these are the criteria for use in WP-129. If so, they should be clearly stated to be the criteria.

- There are a total of twenty-one different design margin calculations identified in WP-129 (Section 4, items 4a through 4u). It is not clearly stated whether all of these twenty-one calculated design margin values need to satisfy specific criteria for the design margin to be acceptable, or whether (in some cases) only some of the calculated margin values need to meet criteria. The position needs to be clarified.

  Figure 1 of this review report illustrates the twenty-one different design margin calculations identified in WP-129. A diagram similar to Figure 1 should be included in the white paper to more clearly illustrate the set of design margin calculations described.

- The method described in WP-129 allows for determination of the operability and design margin of an MOV from which no test data have been obtained (i.e., neither static nor dynamic tests have been performed). This is accomplished by using default values for some parameters which are typically determined through testing. Because of the considerable uncertainties in quantifying operator output at control switch trip without a test (which are not addressed nor quantified in WP-129), we consider that this approach for assessing MOV operability without test data is not acceptable. We suggest that this option be deleted from the white paper or that, if it is retained, a justification be provided to show that the uncertainties are acceptable or are accounted for.

Technical Justification

- WP-129 recognizes that there are random sources of error associated with MOV operability determination, such as torque switch repeatability, diagnostic equipment accuracy, and spring pack testing uncertainty. It is CECO's position that allowance

- On page 3 of 16 reference is made to "Adequate Design Margin calculations" as determined by the "MOV target thrust window (TTW)." A source and definition of these terms should be provided.

Section B.2 (page 5 of 16) describes the method for determining an appropriate valve factor if it has not been verified by in situ testing. Specific comments on this section are:

- The document states that use of valve factors for similar valves tested at other utilities or by EPRI is acceptable, after adjustment for any differences in the methodology for determining valve factor. As opposed to saying "adjustments" we suggest the document indicate that valve factors should be determined in a manner consistent with the CECo method from WP-131.

- The document states that if a more appropriate value can not be determined, the following valve factors should be used:

    - for flexible wedge gate valves, at least 0.5
    - for double disc gate valves, at least 0.35
    - for globe valves, at least 1.1

  A basis for these values should be provided.

Section B.3 (page 5 of 16) describes methods for feedback of diagnostic test results to the MOV design margin evaluation. A specific comment on this section is:

- Three categories of testing status for an individual MOV are identified including: 1) no diagnostic test performed, 2) static test performed, and 3) dynamic and corresponding static test performed. The system operating conditions obtained during a "dynamic" test can range from near static conditions to near design basis conditions. An indication of what is considered acceptable "dynamic" test conditions (e.g., at least some percentage of design basis differential pressure is achieved) should be provided.

Section B.3.b (page 6 of 16) describes assumptions to be made when performing a design margin evaluation on an MOV that has been static tested only. Specific comments on this section are:

- Use of the measured value of stem coefficient of friction is specified, except that a value no less than 0.08 should be used. A basis for this minimum value of stem coefficient of friction should be provided.

- Use of appropriate, justified values for valve factor and rate-of-loading effect are specified. Additional description regarding the meaning of "appropriate" and "justified" should be provided.

- It is stated that the effect of random uncertainties is not included, but that this is compensated for by the conservatisms in the definition of the design parameters. Adequate justification is not provided to support this conclusion. Justification should include example margin calculations which show that this conclusion is supported.

- As mentioned in the detailed comments above, additional justification is needed for several parameter values used in the WP-129 method (e.g., valve factors of 0.5, 0.35 and 1.1, and stem friction coefficients of 0.15 and 0.08). The justification section (Section D) should be expanded to cover all of the areas of WP-129 where justification is needed.

White Paper 129

MOV Design Margin Evaluation

and

Diagnostic Test Feedback Evaluation

Commonwealth Edison Company

Corporate MOV Program Support

Prepared by: ______________________
Ivo A. Garza
MOV Program Support

Reviewed by: ______________________
Tim W. Schwallie
MOV Technical Expert

Concurred by: ______________________

Pilot Site
MOV Project Manager

Approved by: ______________________
Yves Lassere
MOV Project Manager

- static or dynamic test is performed and the as-left control switch setting is not within the target thrust window

- vendor information is received that changes any of the design assumptions such that the margin could be decreased

- dynamic or static test performed on a similar valve at another CECo station changes any of the design assumptions such that the margin could be decreased and that similar valve has not been statically or dynamically tested at the station

- if information is received from industry testing initiatives, such as Kalsi and EPRI, that changes any of the design assumptions such that the margin could be decreased

- operating conditions are changed such that the assumed voltage at the MOV motor terminal decreases, the assumed differential pressure increases, or the design ambient temperature increases

- the MOV hardware is modified or adjusted so that less thrust is available or more thrust may be required to operate the valve

- modifications to the system or operating procedures change the design basis operating conditions

MOV Design Margin is determined by calculating the difference or margin between the thrust associated with the current control switch setting and each of the following MOV design capabilities:

- minimum required thrust to close the valve

- maximum allowable thrust or torque to prevent motor damage, valve damage, or valve operator damage

The Design Margin also considers the difference between the thrust required to open the valve and the maximum allowable thrust or torque to prevent valve motor or actuator damage.

For operability evaluations the calculation for the minimum required thrust and the maximum allowable thrust or torque will not include an allowance for random uncertainties. In contrast, Adequate Design Margin calculations, as determined by the MOV target thrust window (TTW), include allowances for random uncertainties.

3a. No diagnostic test performed

The MOV Design Margin Evaluation is performed using the following assumptions and information:

- stem coefficient of friction - 0.15
- packing load - 1000 lb per stem diameter inch
- valve factor - appropriate, justified value
- control switch trip thrust/torque - based on generic spring curve or spring pack test results
- rate of loading - appropriate, justified value
- if a sister valve has been tested, values from that test should be used, as appropriate

3b. Static test performed

The MOV Design Margin Evaluation is performed using the following assumptions and information:

- stem coefficient of friction - measured value but no less than 0.08
- packing load - measured value
- control switch trip thrust/torque - measured value marked at control switch trip (C14)
- Maximum thrust/torque - measured value marked at C16
- Maximum pullout thrust - measured value marked at O9
- valve factor - appropriate, justified value
- rate of loading - appropriate, justified value

3c. Dynamic and corresponding static test performed

The MOV Design Margin Evaluation is performed using the following assumptions and information:

- stem coefficient of friction - measured value from dynamic test but no less than 0.08
- packing load - measured value from static test

- control switch trip thrust/torque - measured value from static test marked at control switch trip (C14)
- valve factor - calculated value from dynamic test and corresponding static test
- rate of loading - calculated value from dynamic test and corresponding static test

C. Discussion

1. Control Switch Setting Thrust

The thrust associated with the current torque switch setting can be determined but is not limited to the following methods: (Note: the methods are ordered by preference.)

a. results of dynamic test using Liberty test equipment

b. results of static test using Liberty test equipment

c. results of static test using MOVATs test equipment

d. calculated using torque switch setting, assumed stem coefficient of friction of 0.15, and results of a spring pack test

e. calculated using torque switch setting, assumed stem coefficient of friction of 0.15, and generic spring pack torque curve

Note: Spring pack testing uncertainty and spring pack curve uncertainty are assumed to be random with the bias being zero. Therefore, these uncertainties are not included in the operability evaluation.

For limit controlled valves, the thrust associated with current limit switch settings can be determined with the following methods: (Note: the methods are ordered by preference.)

a. results of dynamic test using Liberty test equipment

b. results of static test using Liberty test equipment

c. results of static test using MOVATs test equipment

d. calculated using limit switch setting, assuming generic stem nut deflection constants

2. Minimum Required Thrust

The minimum required thrust (MRT) is a function of the valve design. In the closing direction, MRT is a function of the thrust required to close the valve. The following are included in the calculation of MRT to close: (Note: the values are calculated using the methodology of the T^2 program updated for the latest position papers.)

- design differential pressure
- line pressure
- valve disc area
- packing load
- valve factor
- valve condition factor
- stem piston area

In the opening direction, MRT is a function of the thrust required to pull the valve out of the closed seat. The thrust associated with opening can be determined with the following methods: (Note: the methods are ordered according by preference.)

a. the largest result of static and dynamic testing using votes test equipment (i.e. the pull-out or 09 thrust)

b. calculated using the following equation from WP-107 (Reference 4)

Pull-out = 0.8 * Inertia * CSTT

CSTT = Control Switch Trip Thrust

Inertia Factor assumed

3. Maximum Allowable Thrust

The maximum allowable thrust (MAT) is a function of the motor's capability to generate torque and the valve and actuator's ability to transmit and absorb thrust and torque. The following limiting conditions are evaluated:

- seismic limit for the valve (the maximum closing thrust combined with the seismic thrust)

- actuator thrust limits
- actuator torque limits divided by stem factor
- valve structural limits (both opening and closing)
- motor degraded voltage thrust capability
- motor degraded temperature thrust capacity
- increased motor capability (interim position discussed in White Paper 125)
- motor thrust capability decreased due to motor brakes

4. Margin Calculations

The following calculations are performed to evaluate margin:

4a. Thrust to close

$$\frac{CSTT - MRT}{MRT} * 100 \quad \text{percent}$$

CSTT = control switch trip thrust

MRT = minimum required thrust

4b. Motor Gearing Capability to close valve

$$\frac{MGC - CSTT}{MGC} * 100 \quad \text{percent}$$

CSTT = control switch trip thrust

MGC = Motor Gearing Capability (All motor capability calculations include under-voltage effects)

4c. Decreased Motor Gearing Capability to close valve

```
DMGC - CSTT
---------- * 100  percent
   DMGC
```

CSTT = control switch trip thrust

DMGC = Motor Gearing Capability Decreased not only for voltage but also for ambient temperature

4d. Increased Motor Gearing Capability to close valve using White Paper - 125

```
IMGC - CSTT
----------- * 100  percent
   IMGC
```

CSTT = control switch trip thrust

IMGC = Increased Motor Gearing Capability

4e. Motor Gearing Capability to close valve using White Paper - 125 temperature effects and motor brake applied

```
BMGC - CSTT
----------- * 100  percent
   BMGC
```

CSTT = control switch trip thrust

BMGC = Motor Gearing Capability to close valve using White Paper - 125 temperature effects and motor brake applied

4f. Valve Weak Link Margin to close

```
WLCT - MTC
----------- * 100  percent
   WLCT
```

MTC = maximum thrust in close direction

WLCT = Valve Weak Link Closing Thrust

4l. Motor thrust margin to open valve (only calculated for diagnostically tested valves)

$$\frac{MGC - OPT}{MGC} * 100 \text{ percent}$$

OPT = Open Pull-out Thrust

MGC = Motor Gearing Capability

4m. Decreased Motor Gearing Capability to open valve

$$\frac{DMGC - OPT}{MGC} * 100 \text{ percent}$$

OPT = Open Pull-out Thrust

DMGC = Motor Gearing Capability Decreased not only for voltage but also for ambient temperature

4n. Increased Motor Gearing Capability to open valve using White Paper - 125

$$\frac{IMGC - OPT}{IMGC} * 100 \text{ percent}$$

OPT = Open Pull-out Thrust

IMGC = Increased Motor Gearing Capability

4o. Motor Gearing Capability to open valve using White Paper - 125 temperature effects and motor brake applied

$$\frac{BMGC - OPT}{IMGC} * 100 \text{ percent}$$

OPT = Open Pull-out Thrust

BMGC = Motor Gearing Capability to close valve using White Paper - 125 temperature effects and motor brake applied

4p. **Valve Weak Link Margin to open**

```
WLOT - OPT
--------- * 100  percent
  WLOT
```

OPT = Open Pull-out Thrust

WLOT = Valve Weak Link Opening Thrust

4q. **Actuator Thrust Margin to Open**

```
ATL - OPT
---------- * 100 percent
  ATL
```

OTC = Open Pull-out Thrust

ATL = Actuator Thrust Limit

4r. **Actuator Torque Limit to open**

```
ATL(torque) - OPT(torque)
------------------------- * 100 percent
     ATL(torque)
```

OPT(torque) = calculated maximum torque associated with the measured or calculated open pull-out thrust

ATL(torque) = Actuator Torque Limit

4s. **Kalsi Increased Actuator Thrust Margin to open**

```
ATLK - OPT
----------- * 100 percent
  ATLK
```

OPT = Open Pull-out Thrust

ATLK = Actuator Thrust Limit increased using Kalsi

4t. **Kalsi Increased Actuator Torque Margin to open**

$$\frac{\text{ATLK - OPT(Torque)}}{\text{ATLK(Torque)}} * 100 \text{ percent}$$

OPT(torque) = calculated maximum torque associated with the measured or calculated open pull-out thrust

ATLK(Torque) = Actuator Torque Limit increased using Kalsi

4u. **Dynamic Test Margin (corrected)**

$$\frac{\text{CSTT - FCT(design)}}{\text{FCT(design)}} * 100 \text{ percent}$$

FCT = flow cutoff extrapolated to design pressures in accordance with White Paper 131

CSTT = control switch trip thrust

D. Justification:

The key to calculating the MOV Design Margin, a measure of the MOVs capability to provide a reasonable assurance that the MOV will perform its specified safety function, is establishing the thrust values associated with the current control switch setting, the minimum required thrust, and the maximum allowable thrust.

The certainty with which the thrust values, and therefore the MOV Design Margin, can be determined is a function of the accuracy and conservatism of the MOV design parameters, the repeatability of the MOV control system, and the accuracy of the diagnostic test equipment. The uncertainties associated with these values can be grouped into two types, bias and random. All of the design parameters, such as line pressure, differential pressure, valve disc active area, and motor capability, have been biased in the conservative direction. The calculation of the MOV's thrust values, includes no allowance for random uncertainties such as torque switch repeatability and test equipment accuracies. In contrast, adequate MOV design margin (High margin) as established by the MOV target thrust window (TTW) includes allowances for these random uncertainties as additional conservatism and margin. The exclusion of random uncertainties is justified because conservative bias included in the design parameters provide additional margin which reasonably assure that the valves will perform their safety function.

E. References

1. White Paper 124, "Load Sensitive Behavior (Rate of Loading)"

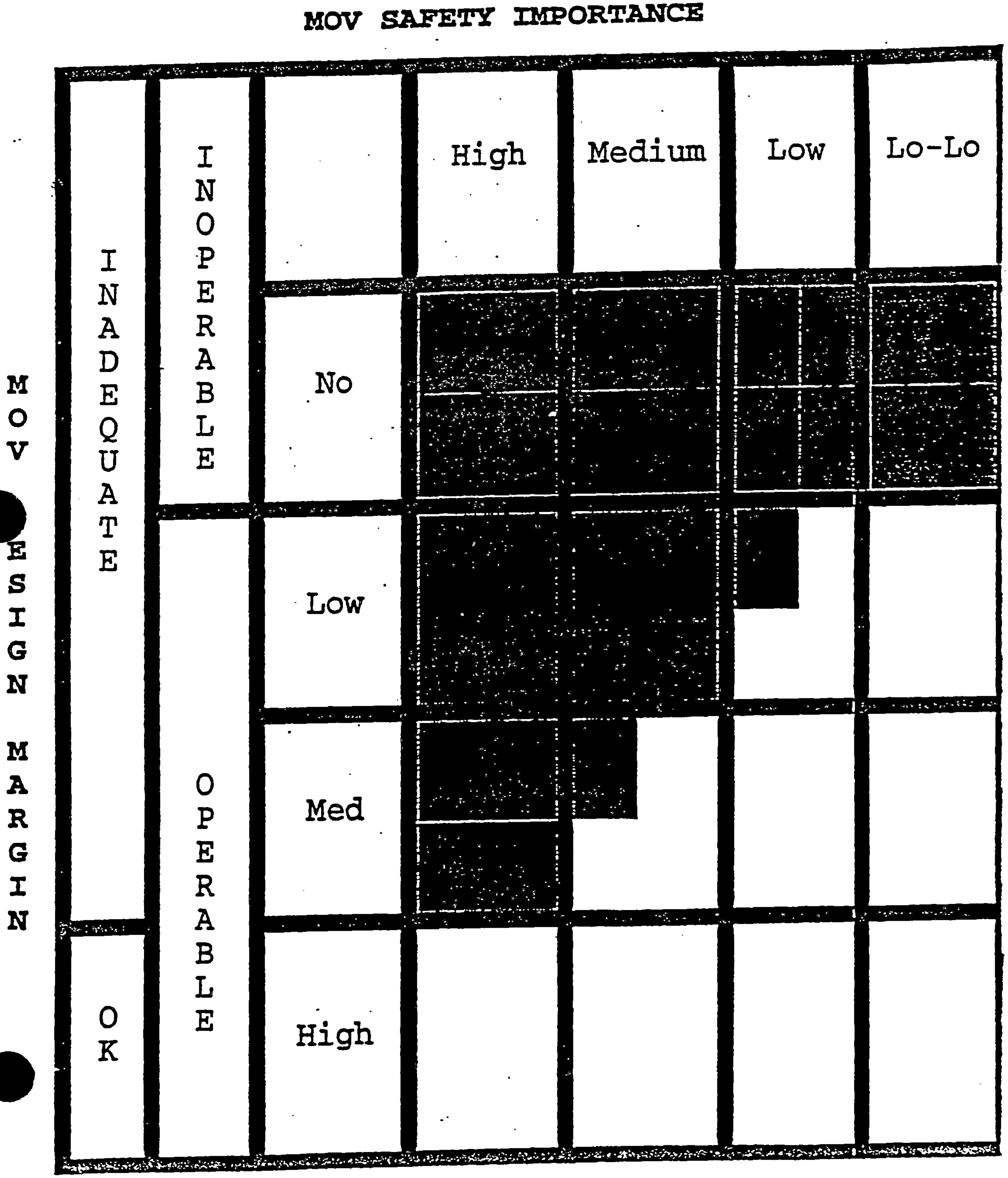
2. White Paper 125, "Increased Motor Capability"

3. White Paper 130, "MOV Problem Resolution"

4. White Paper 107, "Thrust Window Margins, Desired Thrust Windows, Target Thrust Methodology, and Operability Criteria"

5. White Paper 131, "Valve Factor Calculation Methodology"

Figure 1 - MOV Margin and Importance Matrix



Attachment A

MOV Margin Calculation Data Base
Paradox Program
Object PAL (Source Code)

Attachment B

MOV Margin Calculation Data Base

Paradox Program
Sample Reports

ATTACHMENT

6

MPR
ASSOCIATES INC.
ENGINEERS

December 23, 1994

Mr. Paul Dietz
Commonwealth Edison Company
1400 Opus Place
Downers Grove, IL 60515

Subject: Review of Commonwealth Edison Company White Papers: 154 - "Anchor/Darling Valve Factors;" 160 - "Crane Valve Factors;" and 164 - "Anchor/Darling Double Disk Gate Valve Factors."

Dear Mr. Dietz:

Enclosed is a report of our review of the subject Commonwealth Edison white papers. We consider that they represent an approach that is appropriate and useable for establishing valve factors that should be used to predict stem thrust when a valve cannot be dynamically tested. The approach makes extensive use of available test data and applies that information in a manner that would be expected to be reasonably conservative. Our major comments are as follows:

1. Much of the methodology of these white papers is built upon a "conservative group valve factor" that is defined as the 95% upper confidence band of the <u>average</u> valve factor observed in the tests. Since this quantity is based on predicting the average of the tests, some valves would be expected to have valve factors that are higher than the conservative group valve factor. The argument is advanced that static tests will screen out valves with high valve factors and that conservatism in other aspects of determining the operability of a valve are adequate to assure that there is a positive operating margin. Although we agree that the argument is technically plausible, we consider that quantitative justification should be developed. We understand that a revised approach that accounts for the variation above the average more explicitly, as well as considers other sources of variability (e.g., torque switch repeatability) is under development.

2. The test data and the values in the white papers are based on closure strokes at, or near, ambient temperature; however, we understand that the intent is to apply these white papers to opening strokes and throughout the range of operating temperatures. We agree that it is usually conservative to apply the ambient temperature closure valve factors to elevated temperatures and opening strokes; however, there are a number of special conditions under which that may not be the case. Additional justification of the use of the valve factors for opening strokes and elevated temperatures is needed. In particular, it may be necessary to place some limits on minimum valve factors, or additional conditions on the applicability of the

white papers. Validation of this approach by actual test results for opening strokes and for high temperatures would also be desirable.

3. Although there appears to be some correlation between valve size and differential pressure (DP) and the measured valve factor, the source of that variation is not well established. Consequently, we recommend that the valve factors not be extrapolated to valves larger than those represented by the data nor to DPs greater than those of the data.

4. "Damaged" and "undamaged" valve factors are identified for evaluation of Crane 900 lb valves for blowdown service. We consider that there is substantial uncertainty in attempting to bound the behavior of valves with extensive guide damage; consequently, we recommend that the use of the "damaged" valve factors be limited to operability evaluations. We recommend that valves that appear to be susceptible to damage be reworked or modified (if needed) so that no damage is predicted.

The enclosed report provides more detailed discussion of these major comments as well as other more minor comments. Note that because these white papers are still in the process of preparation and internal revision, our comments are generally confined to basic issues and assumptions. We have discussed our major comments with Mr. Brian Bunte. Please do not hesitate to call if you have any questions or comments regarding the enclosed report.

Sincerely,

Dwight H. Harrison

Enclosure

MPR
ASSOCIATES INC.
ENGINEERS

Enclosure to
MPR letter of
December 23, 1994

REPORT OF REVIEW OF COMMONWEALTH EDISON MOV WHITE PAPERS 154, 160, AND 164

OVERVIEW

This review report documents the approach and conclusions of an independent review of three closely related Commonwealth Edison Company White Papers:

- 154 Anchor/Darling Valve Factors, Revision 0 (Unsigned), September 15, 1994;
- 160 Crane Valve Factors, Revision 0 (Unsigned), September 15, 1994: and
- 164 Anchor/Darling Double Disk Gate Valve Factors, Draft Revision 0, September 15, 1994.

A copy of each of the versions that was used in this review is included as Attachment A to this report.

SCOPE OF WHITE PAPERS 154, 160, AND 164

These white papers describe how valve factors are to be determined for valves that have not been dynamically tested. The white papers use the results of tests of similar valves that have been conducted by Commonwealth Edison, the Electric Power Research Institute (EPRI), and others. The predicted valve factors as functions of valve nominal size and differential pressure (DP) for groups of valves are established based on the test results for comparable valves. The white papers define how valves of the type and manufacturer covered by the particular white paper are grouped by rating, model, or other features. The white papers also indicate how the valve factors are to be used in performing operability/margin reviews and for design calculations and closing out USNRC Generic Letter (GL) 89-10 reviews. The white papers give criteria to be used to identify those valves that may not be typical of the others in the group and may have higher than expected valve factors. These white papers are only applicable to the types of valves that are specifically identified. Although there are a number of differences in the details of the individual papers, they are all based on the same overall approach.

CONCLUSIONS AND RECOMMENDATIONS

The white papers present an approach that appears appropriate and useable for establishing the valve factors that should be used to predict valve stem thrust when a valve cannot be dynamically tested. It makes extensive use of available test information and applies that information in a manner that would be expected to be reasonably conservative. Because these white papers are still in the process of preparation and internal revision, our comments are generally confined to basic issues and assumptions. Our major comments are as follows:

1. A "conservative group valve factor" is defined which is the 95% upper confidence band of the average observed valve factor in tests. Because this is an upper bound estimate of the average valve factor, it will not be conservative for all valves, i.e., some valves will have higher valve factors. The use of the "conservative group valve factor" in MOV operability/margin reviews and design evaluations is defended in the white papers on the basis that:

   - Individual valves with high valve factors are screened out based on static test results (and are then required to meet the more stringent bounding requirements) and

   - The valve-to-valve variations are covered by the minimum required margins in the MOV margin evaluations (as described by white paper 129, Reference 1). The white papers do not provide quantitative justification that this approach will provides a positive net operating margin for all or nearly all valves. We recommend that the white papers be revised to include the needed justification.

   Some of the variation in the observed valve factors is attributable to uncertainties (e.g., instrument errors) in the measurement of the valve factor. This measurement uncertainty is not part of the actual valve factor uncertainty and tends to overestimate the actual variability in valve factor from stroke-to-stroke and from valve-to-valve. Although it should be theoretically possible to account for the measurement variation, that does not appear to be practical at present. The alternative of eliminating all data which are known or suspected of having a large uncertainty could potentially reduce the variation; however, it would be difficult to establish criteria for elimination that do not result in some biasing of the results.

   From our discussion with Mr. Brian Bunte (Commonwealth Edison), we understand that a revised approach is being developed in which the variation in valve factor above the average value is considered. To realistically use the variation, it is evaluated statistically in conjunction with other sources of variation in MOV evaluations (e.g., torque switch repeatability). This appears to be an improved approach which could adequately address our concerns. This approach is similar in some respects to the one used by Black (Texas Utilities Electric) in MOV evaluations (see Reference 2).

2. The test data used to determine valve factors cover only closure strokes at ambient or near-ambient temperature conditions. Although it is not explicitly stated in the white papers, we understand from Mr. Bunte that it is intended to apply the results of the white papers to closing <u>and</u> opening strokes at all temperatures on the following basis:

   - Opening valve factor is determined by making an adjustment to the closing valve factor to account for the effect of the disk wedge angle.

   - Valve factors based on room temperature test results are conservative when applied to higher temperatures.

   Our comments on this approach are as follows:

   a. Opening valve factors can be adequately determined from closing valve factors using a wedge angle correction when opening DP thrust and closing DP thrust are both controlled by the disk sliding on the seat with the flow isolated. Although it is not specifically stated in the white papers, it appears that the closing test data used in the white papers conforms to this criterion. However, valve opening thrust can sometimes be controlled by other phenomena, including guide friction or Bernoulli forces for wedge-type gate valves, and wedge sticking or Bernoulli forces for double-disk valves. Accordingly, determination of opening valve factors from closing factors is not always justified. It appears that this may be addressed by imposing a minimum opening valve factor to account for these other effects. That is, if the valve factor for opening determined by adjusting the closing valve factor is below a minimum, then the minimum value is used instead of the adjusted closing value. This approach is unlikely to be an unreasonable constraint for opening stroke thrust calculations, since most opening stroke thrust evaluations are controlled by unwedging thrust, which is evaluated separately.

   b. The valve factor should decrease as the temperature is increased for sliding of the disk on the seat sealing surfaces when the contact surfaces are both Stellite 6 and the surfaces have be "preconditioned," i.e., have been stroked enough that the maximum friction coefficients are being achieved. Although not stated in the white papers, it appears that the valve factor data are for valves with Stellite 6 hardfacing. Although the data appears to be from typical valves, it does not follow that all the data are from "preconditioned" valves. Accordingly, the conclusion that the valve factors at temperatures above ambient will be bounded by the test results, may not be justified. This could also be addressed by including a minimum valve factor to be used at elevated temperature (e.g., above 200°F).

The justification for the position on opening strokes and elevated temperatures could be significantly strengthened if data from opening strokes and from elevated temperature tests were cited. That data could also assist in identifying minimum valve factors for opening strokes and elevated temperatures.

3. The correlations of valve factor are based on data from a defined range of valve size and DP. Because the source of the variation in measured valve factor with valve size and DP is not well established, we do not consider that the extrapolation of these valve factors to valve sizes and DPs beyond the range of data is justified. We recommend that the use of valve factors from the white papers be limited to nominal valve sizes equal to or less than the maximum size in the data and to DPs equal to or less than the maximum DP in the data.

4. The white paper for the Crane valves (160) identifies the possibility that damage to Crane valves can occur under blowdown conditions, depending on the valve materials and configuration. "Damage valve factors" (which are higher) are required to be used when certain criteria are satisfied. The higher values are based on the results of EPRI and Commonwealth Edison testing. Our concerns with the white paper approach are as follows:

   a. Although the high Crane valve factors for potential damage conditions provide increased confidence that they will bound the actual valve performance, the damage mechanisms observed are inherently not reproducible. Accordingly, maximum thrust required is difficult to predict. We suggest that the use of such valve factors be restricted to operability evaluations and that valves be modified (if needed) so that no valves are predicted to be in the "damage" regime.

   b. Although the white paper addresses the observed behavior of the Crane valves under blowdown conditions, it does not address the potential for damage to other types of valves under blowdown or for other conditions that result in high DPs with the valve partially open. The implicit assumption is that Crane valves in blowdown service are the only valves covered by USNRC Generic Letter (GL) 89-10 in Commonwealth Edison plants that are susceptible to damage. Because tests have shown a strong effect of sharp Stellite 6 edges on valve operation, this may not be justifiable. As a minimum, we suggest that Commonwealth Edison ensure that all gate valves are free of Stellite 6 sharp edges on:

      - The disk sealing surface OD,

      - The body seat ring sealing surface ID, and

      - The edges of the disk guide slot (especially the bottom edge).

      Minimum chamfers and radii that should be provided to avoid major mechanical damage are given in EPRI TR-103229 (Reference 3).

REVIEW APPROACH

The MPR review approach is as follows for each white paper:

- Review the white paper purpose to ensure that it is clearly and completely stated.

- Review the statement of position to ensure that it:

  - addresses the purpose;
  - is clear and complete; and
  - includes all appropriate restrictions and limitations with regard to its use.

- Review the technical justification to ensure that it:

  - logically presents a case with defends the stated position;
  - makes proper technical use of the theory and data which are referenced;
  - adheres to appropriate requirements of codes, standards, and regulations which are referenced;
  - does not exclude references to key data or requirements;
  - provides a sufficient technical basis for the stated position; and
  - is written in a way to provide a convincing justification.

As part of the review of the justification, comparisons to other data or approaches (e.g., EPRI data or models) that may not have been considered in writing the justification are made.

Note that in the case of review of these three closely related white papers there was also a review of the papers against each other to assure consistency of content and technical justifications.

RESULTS OF REVIEW INCLUDING DETAILED COMMENTS

1. <u>Purpose of White Papers</u>

   The common purpose stated for the three white papers is that they provide a means of determining the valve factor for valves of the type (e.g., manufacturer, rating, etc.) specifically covered by the white paper that have not been dynamically tested. These determinations are based on the results of Commonwealth Edison, EPRI, and other industry testing. The white papers also provide criteria for grouping the valves by type, pressure rating, model, etc.

Comments:

a. Applicability

   A concise statement of the applicability limits of each of the white papers should be provided. The information on applicability is presently found in a number of different sections. Typical topics that may need to be addressed in the applicability limits are as follows:

   - The materials of key components, including: the disk and seat sealing surfaces, the disk and body base metal, the disk guide slots and body guide rails for solid or flexible disk wedge valves, and the upper and lower wedges of double disk valves;

- The fluid medium (e.g., water, steam, air or gas);
- The stroking direction;
- The fluid temperature;
- The valve safety function (e.g., flow isolation, flow initiation, or leaktightness);
- The past maintenance history of a valve (e.g., whether it has been recently refurbished);
- The differential pressure;
- The valve size; and
- The valve orientation (e.g., stem horizontal or vertical, lower wedge upstream or downstream for double-disk valves).

b. Direction of Stroke

The valve factors in the white papers are calculated using data from valve closing strokes and the method of Commonwealth Edison White Paper 131 (Reference 5), which is only applicable to closing strokes. Consequently, the regression lines and confidence bounds on the valve factors are based only on closing strokes. It appears that White Papers 154, 160, and 164 are, therefore, strictly applicable only to closing strokes. It is generally conservative to use a valve factor based on closing strokes for a valve whose safety function is an opening stroke, since the required stem thrust to overcome DP is generally larger for closing strokes than for opening strokes as long as sliding on the seat is the most restrictive condition. An alternative, less conservative approach (which we understand Commonwealth Edison is considering) is to "adjust" the closing valve factors to obtain opening valve factors based on the wedge angle of the disk. This approach is justifiable as long as sliding on the seat is the most restrictive condition in both directions. For solid and flexible wedge gate valves there can be instances where the friction on the guides is more limiting. In particular, this can occur at high temperature or high flow. Further, for double-disk gate valves, there can be an opening condition in which the valve remains wedged as it lifts, which will have a higher valve factor than the closing flow isolation condition (see EPRI TR-103232, Reference 4). Regardless of the approach used for opening (i.e., either applying the closing valve factors to opening or adjusting the closing valve factors for opening) it may be necessary to place some conditions on when this can be done. For example, it may not be appropriate when a valve's opening service involves high flow through the partially open valve or the guide clearances are known to be tight.

c. Logic Diagrams

These white papers specify a detailed logic for use of the valve factors in operability/margin reviews and in closing out GL 89-10. This involves interaction with a number of other Commonwealth Edison White Papers, especially 129 and 131 (References 1 and 5, respectively). All need to be considered together and a master logic diagram would be useful to define their interrelationships and to ensure that the overall process is consistent from application to application in the Commonwealth Edison Plants. It also appears that some material is duplicated between the various white papers. This is not a problem as long as all the white papers are consistent; however, it can substantially complicate changes and revisions.

2. <u>Technical Position of White Papers</u>

In this section the major technical positions of the three white papers are summarized along with comments and recommendations developed in the course of the review. Detailed comments are included in Section 4 of this report.

2.1 Cold Water Valve Factor Derivation (C.1 in 154 and 160, C.2 in 164)

It is the technical position of the white papers that the valve factor is a function of valve nominal size and differential pressure (DP) across the valve. Typically, the valve factor is reduced as the DP and size are increased. In the white papers it is assumed that the relationship of valve factor to the nominal size and the DP can be represented as linear functions. The values of valve factor are provided on figures as a function of valve size. These values have been adjusted to zero DP. To estimate the valve factor for a particular application, the valve factor at zero DP must be reduced by the product of the DP and the pressure adjustment factor. The pressure adjustment factor is unique to a particular group of valves. It is obtained by minimizing the standard error of the linear fit of adjusted valve factor to the valve nominal size.

The lines in the figures that define the valve factors have the following meanings:

- The bold central line is the nominal value of the adjusted (zero DP) valve factor based on the available data. It is a conventional least squares fit to the available applicable data after a DP adjustment is made.

- The dashed lines adjacent to the bold line represent the 95% confidence interval on the nominal (average) valve factor.

- The outermost dashed lines correspond to a 95% confidence interval on the individual data points.

Comments:

a. Spot checks of a few of the calculations have been made and the results were found to be in accordance with the methods of the indicated references. We suggest that the explanation of the lines be expanded as to their meaning in the context of the white papers. A more complete description of the innermost set of lines would be that statistical theory calculates that there is a 95% probability that the actual mean regression line falls within the band indicated. That is, there is a 95% probability that if the entire population of valves were tested, the average of all the valves would fall within the band. Note that this is not the probability that the valve factor for any single valve that is tested will fall within the band—that is the outermost band. For the outermost band, it would be more descriptive to state that there is calculated to be a 95% probability that the observed valve factor for any additional valve that is tested will fall within the band. More detailed comments in this regard are found in Sections 3.1 and 3.2.

b. Since valve operability is generally linked to the upper bound of the bands indicated, the bands selected actually predict a 97.5% probability that the valve factor value will be below the upper curve. This is a greater confidence than is usually considered sufficient for engineering use. That is, normal engineering practice would be to achieve a 95% confidence that the valve factor is bounding. It appears that practice would support a reduction in the bands and still retain a 95% confidence that the value of valve factor will not be exceeded. For example, in the case of the A/D double disk valves, the width of the bands would be reduced to about 83% of the value in the white paper by using a 95% upper bound; for the A/D 900 lb valves, the width of the bands would be reduced to about 82% of the white paper value.

c. Fundamental to all the white papers is the assumption of a linear relationship of valve factor to DP and size. We agree that there appears to be a trend in the data that leads to lower valve factors for large valves with large DPs. We believe that its source is the reduction of Stellite 6 friction coefficient that occurs as contact stress increases. Although both the EPRI tests and tests by other organizations show a general trend of lower friction as contact stress increases, that effect is not particularly linear over the range of stresses. For example, the Battelle friction tests in the EPRI program (Reference 6) did not show a general reduction in friction coefficient with stress when the stress was less than about 10 ksi. At ambient temperature the friction algorithm used in the EPRI gate valve model (Reference 7) has a constant value of friction coefficient of 0.61 until the stress reaches 10 ksi, then the coefficient is reduced linearly to 0.53 at 25 ksi (0.0053 per ksi). It then is reduced linearly to 0.40 at 50 ksi (0.0052 per ksi). The assumption of a linear reduction in valve factor is consistent with the EPRI formulation for the maximum friction, but only when the contact stress is high enough. As discussed below, this indicates that the relationship of valve factor to size could be different depending on the differential pressure.

In this regard, the position does not place any limitations on the DP across the valve that can used to make the adjustment in the valve factor. For example, if an Anchor/Darling 150 lb flexible disk valve were estimated under the rules of White Paper 154 and the DP were 250 psi (an extreme, but potentially possible situation), the conservative group factor would be reduced by 0.31 from the curve valve. For a 10 inch valve this would result in the prediction of a conservative group valve factor of about 0.14 and an average valve factor of 0.09. We consider that it would be unreasonable to expect the average valve factor to be that low. For that reason we believe that some cutoff of the DP effect needs to be made. We recommend that for DPs greater than the range for which there are data, the valve factor at the extreme of the DP range be used. For example, for an A/D 150 lb valve with a DP greater than 114 psi, the valve factor at 114 psi would be used.

d. For valves that are proportionally dimensioned throughout the range of size, there should be no effect of size, since the contact area and the DP force should both increase by the square of the valve size. However, it appears that for some valve lines the seat sealing surface width may not be kept in proportion as the valve size is changed. That is, the seats may not become proportionally wider as the valve size increases which, in turn, tends to increase the contact stress on larger valves for the same DP. This feature would tend to yield lower friction as the valve size increases. It is not evident that linearity of this effect would be expected. However, we would expect the overall size effect to be relatively modest, since the friction is insensitive to stress in the lower stress ranges. The large effect of valve size for some of the valves, for example the valve factor for 900 lb Anchor/Darling valves changes from 0.7 to 0.4 between 3 and 12 inches nominal size, raises questions as to the validity of the assumption and makes any extrapolation difficult to justify.

   In this regard, the application of a statistical test to one of the sets of data (150 lb A/D) using the method of NBS Handbook 91 (Reference 8), Section 5-4.1.6 does not support the assumption of linearity with size at the 95 or even 90 percent confidence level. It is not evident that the added complexity of the regression analysis is warranted on strictly statistical grounds; however, it does not appear the introduce any problems as long as it is not applied to very large or very small valves (i.e., outside the range of the data). In this regard, we consider that some size limits should be placed on the use of the valve factor predictions relative to the available data base. We understand from discussions with Mr. Brian Bunte of Commonwealth Edison that later revisions of these white papers have instituted such limits for some valves.

e. The white papers do not explicitly address the temperatures for which the valve factors are applicable. We understand that the data used in the white papers are from tests at ambient temperature and that it is proposed to evaluate all temperatures using the approach in the white papers, i.e., the values are not changed for service temperatures above ambient. This is not fully justified, as explained below:

   - Tests of Stellite 6 generally show that the friction between the disk and the body seat decreases with temperature. The amount of the decrease depends on whether the valves are fully "preconditioned." For valves that are not fully preconditioned, increasing the temperature may not always result in a decrease of friction coefficient (See, for example, Figures 4-1-4 and 4-1-8 in

Reference 6). Although the data used in the white papers is from typical valves, they are not necessarily fully preconditioned valves.

- In contrast to the sliding of Stellite 6 on Stellite 6, the sliding of carbon steel on carbon steel results in generally higher friction as the temperature is increased. Consequently, guide friction often increases with temperature. Guide friction can be of importance in some special situations such as blowdown or high flow at partially open conditions or when the guide clearances are small. Guide friction has usually been found to be more important in opening strokes, than in closing strokes.

Although these potential shortcomings in the method may only affect a limited number of actual applications, we consider it would be prudent to assure that some minimum valve factor is applied at temperatures above ambient. In this regard, the application of the white papers' valve factor predictions to any available higher temperature valve test data is recommended to provide additional confirmation of its applicability and provide some guidance as to what minimum valve factor should be adopted.

2.2 Use of Valve Factors in Performing Operability/Margin Reviews (C.2 in 154 and 160, C.3 in 164)

The basic position on operability/margin reviews is that operability can be determined using the upper bound of the average valve factor.

Comments:

a. The basic position is justified on the basis that there should be enough margin in the other parameters involved (e.g., DP and line pressure) and in the margin criteria (e.g., 35% for high margin valves) of White Paper 129 (Reference 1) to account for the valve-to-valve variations in valve factor and other sources of random variation such as diagnostic instrumentation uncertainty. Although this is certainly a reasonable position based on engineering judgement, there is little quantitative justification. The position would be significantly stronger if some quantitative description of these other margins and their expected variability could be developed. This information (in conjunction with the already available information on the valve factor) could potentially be used to show that there is some quantitative confidence that the valve is operable. We understand from Mr. Brian Bunte (Commonwealth Edison) that an approach is being developed which considers all of the random uncertainty contributors.

b. These sections are essentially the description of a logic tree or sequence that needs to be followed to conduct the review. It is relatively difficult to follow the logic that is described. For our own review we have found that putting the logic in a diagram form was helpful in understanding the order and relationships of the various steps. We suggest that presentation of some of this material in logic diagrams be considered.

c. These sections (as well as some subsequent sections) use a "DP load" of 2000 lbs (presumably the stem thrust load from DP) as a criterion for various actions. As used, this criterion is independent of valve size; however, stem thrust from DP is a strong function of valve size. In particular, while 2000 lbs may be considered a "small" stem thrust for large valves, the same thrust may be "large" for small valves. The justification for using the value of 2000 lbs for all valves sizes needs to be expanded or the criteria for "low" DP needs to be dependent on valve size.

2.3 Use of Valve Factors For Design Calculations and Closing Out MOVs Under GL 89-10 (C.3 is 154 and 160, C.4 in 164)

The basic position on design adequacy and close-out of GL 89-10 is that the valve is adequate if all of the following are satisfied:

- The valve will meet the high margin criteria for the operator if the conservative group valve factor (highest prediction of the average valve factor for the group) is used.

- The valve will meet the low margin criteria for the operator if the bounding group factor (highest prediction of valve factor for an individual valve) is used.

- The valve has no indication of potentially high valve factors.

Comments:

a. As commented previously in conjunction with the operability/margin reviews, this position is reasonable based on engineering judgment, but would be strengthened considerably if it could be justified quantitatively.

b. As above, these sections are essentially the description of a logic tree or sequence that needs to be followed to conduct the review. Presentation as a logic diagram could significantly clarify the order and relationships of the various steps.

2.4 Steam Blowdown Valve Factor Determination (C.4 in 160 and C.5 in 164, not in 154)

It is the position in White Paper 160 that the valve factor of Crane 900 lb class valves under steam blowdown conditions can be obtained by considering the materials used in the guides and the clearances in the guides. These considerations as well as valve size determine when "damaged" or "undamaged" valve factors are used. In the version of White Paper 160 that was available for review, this section was not complete.

Comments:

a. The position in White Paper 160 is based on valve test results that show (for the particular valves tested) that the edges of the disk slot and the material of the guide rail are not as important in the determining the valve factor as is clearance in the guides. Although that is the case for the particular valves tested, laboratory tests by Battelle (Reference 6) show that sharp Stellite edges in contact with a soft material such as carbon or stainless steels would be expected to result in serious mechanical damage and would be unpredictable. In order to get that situation (a sharp edge-on-flat contact) the disk must be tipped on the guides. Disk tipping depends on the location of the guides and the effective location of the hydraulic forces. If the disk were to tip, greater guide clearance would lead to a greater tip angle and would not necessarily improve the performance. We recommend that, as a minimum, the "damaged" valve factors be applied to those valves that have sharp Stellite 6 edges on the bottom of the disk guide slots. However, even those values may not be conservative based on the Battelle tests and it would be prudent to rework those edges at the first opportunity.

b. In the case of White Paper 164 there is no derivation/justification for the position on steam blowdown factors. The figures showing data are also not included. Consequently, MPR cannot comment on whether the position is justifiable.

3. <u>Technical Justification of White Papers</u>

3.1 Basis for DP Correction (D.2 in 154, D.3 in 160 and 164)

It is assumed in the white papers that the valve factor is a linear function of the DP. The factor used to adjust to valve factor to zero pressure differential is obtained by varying the factor until the standard error of the linear regression as a function of valve size is minimized. In general there is a negative correlation between the valve factor and the DP, i.e., the valve factor decreases as the DP increases. The specific factors that were applied are as follows:

- A/D 150 lb - 0.00125 per psi
- A/D 300 lb - 0.0002 per psi
- A/D 900 lb - 0.00005 per psi
- Crane 300 lb - 0.0004 per psi
- Crane 900 lb - zero
- A/D Double Disk - zero

Comment:

a. An alternative that should be equivalent to the method in the white papers would be to fit the data with a regression plane rather than a line. The plane would be based on two variables: the valve size and the DP. Where data were included in the white paper, the coefficient on the DP term in the regression plane equation should be essentially the same as the DP factor obtained in the white papers. The procedure to construct a regression plane is found in most standard statistical texts (see Reference 9, page 429, for example). It was applied to the available data with the following results:

- A/D 150 lb - 0.00127 per psi
- A/D 300 lb - 0.000173 per psi
- A/D 900 lb - 0.0000430 per psi
- Crane 300 lb - No data in white paper
- Crane 900 lb - No data in white paper
- A/D Double Disk - 0.00000752 per psi

These are in satisfactory agreement with the values in the white paper. Since the statistical program that was used to handle the data probably has the capability to make a regression in two variables, Commonwealth Edison should consider using the regression plane to determine the DP factor. Although it would be laborious to do by hand, it is very simple using a computer and avoids the need for repeated trials to obtain a result.

Note that, except for the A/D 150 lb valves, the adjustment in the valve factor as a result of the DP factor will generally be small—less that 0.1. In the case of the A/D 150 lb valves, that is not the case. For a 200 psi pressure differential the valve factor will be adjusted by 0.25. Because of the magnitude of the adjustment, the valve factor at high DPs may be quite low. As previously commented, we consider that it would be prudent to avoid any extrapolation to a higher pressure than that for which there is data. Accordingly, we recommend that for the A/D 150 lb valves that the adjustment for DP be limited to about 114x0.00125 or 0.15.

3.2 Determination of Best Fit Straight Line and Confidence Bounds (D.4 in 154, D.5 in 160, and D.6 in 164)

In this section NBS Handbook 91 (Reference 8) is cited as the basis for the statistical treatment of the data. Since NBS Handbook 91 contains many sections and procedures, the specific portions that are applied should be identified. Our review indicates the following specific sections are involved:

- The first or smaller confidence band is obtained by the procedure in Section 5-4.1.2.2 (page 5-18) of NBS Handbook 91. It is formally defined as: confidence interval estimate for a single point on a regression line (i.e., the mean value of Y corresponding to a chosen value of X). This is, in effect, the confidence that can be assigned to the average valve factor at a particular valve size. It implies that if the entire population of valves could be tested and their valve factors determined, there is a 95% probability that the average of the total population would fall within the band. As stated in the white papers, as the number of valve tests increases, the width of this band decreases.

- The second or larger confidence band is obtained by the procedure in Section 5-4.1.2.3 (page 5-19) of NBS Handbook 91. It is formally defined as: confidence interval estimate for a single (future) value of Y corresponding to a chosen value of X. This is an estimate of the probability that when another valve test is run it will lie within the band. That is, it is estimated that if another valve were tested, there is a 95% probability that its observed valve factor would fall within the indicated band. The width of this band depends on the valve-to-valve variation in the observed valve factor and to some extent on the amount of data. The observed variations in valve factor reflect real differences between individual valves, difference between individual strokes of the same valve, and variations because of inaccuracies in the test conditions or instrumentation. The width of this band will probably be reduced only slightly by the collection of more data.

Comment:

a. The method used in the white papers only addresses the confidence bands for variation in valve size. There is a comparable effect for DP as a variable. Commonwealth Edison has chosen not to include this variable in determining the confidence bands. Although the reasons are not stated in the white papers, it is evident from the data that the variation in valve factor with DP is usually much less important than the variation with changes in size. Accordingly, the effect on the confidence bands because of the DP should be relatively minor and is probably not worth the added complexity. This is not so evident for the A/D 150 lb valves, however. For those valves, there are two concerns:

- The magnitude of the DP effect is high and
- The DP data are concentrated in a relatively small band (between about 50 and 120 psi).

Although it is possible to calculate the confidence bands on the regression plane that are comparable to those obtained for the line, it is probably simpler to limit the DP adjustment to the value corresponding to the highest DP data point. (As discussed in comment 3.1.a., this would be an adjustment of about 0.15.)

3.3 Description of Valve Design and Basis for Three Relevant Factors (D.4 in 164 only)

Comments:

a. It is stated in White Paper 164 that the spread disk valve factor is calculated using the White Paper 131 (Reference 5) method. However, White Paper 131 appears to be applicable to flexible or solid wedge gate valves only. Appropriate justification for using White Paper 131 for double disk gate valves should be provided.

b. White Papers 154 and 160 use the terminology "spread disk" valve factor. That terminology is unique to the double disk gate valves and should not be used in these two white papers, since they apply only to solid and flexible disk gate valves. The condition for flexible or solid wedge gate valves that would be somewhat analogous to the spread disk condition for double disk gate valves is at initial wedging. Although there is potentially a small difference between the valve factor at flow isolation and initial wedging, it is not usually a practical limit and is not normally used or quoted.

4 Detailed Comments on White Papers

4.1 Generally Applicable Comments

a. Definition of Terms

There are a number of terms used in these white papers that are defined in very specific ways in this application. These definitions tend to be scattered throughout the documents. We suggest grouping all of these terms and their definitions in a single section in each white paper devoted only to such definitions. Some specific definitions that we consider should be in the section of definitions are listed below.

Nominal Valve Factor - Best-fit straight regression line of valve factor test data.

Average Group Valve Factor - use existing definition.

Conservative Group Valve Factor - use existing definition.

Bounding Group Valve Factor - use existing definition.

Measured Valve Factor - use existing definition, with the following addition - "The measured valve factor is calculated according to the method in Reference __" (WP-131).

Apparent Valve Factor - same as measured valve factor.

Fully Degraded Valve Factor - The valve factor for a specific valve which is expected to be the maximum valve factor under normal operating conditions.

Design DP Load - Stem load calculated as $[DP_{design} \bullet Area_{seat}] \bullet VF_{CG}$ where DP_{design} is the design basis DP, $Area_{seat}$ is the flow area of the valve, and VF_{CG} is the Conservative Group Valve Factor.

Test DP Load - Stem load calculated as $[DP_{test} \bullet Area_{seat}] \bullet VF_{CG}$ where DP_{test} is the test DP, $Area_{seat}$ is the flow area of the valve, and VF_{CG} is the Conservative Group Valve Factor.

Low Margin, High Margin - Categories of stem thrust margin as defined in WP 129.

DP Correction Factor - A factor subtracted from the plotted valve factor at zero DP conditions to adjust the factor to higher DP conditions. The DP correction factor is used as follows:

$$VF_{DP} = VF_0 - CF_{DP} \bullet DP$$

where VF_0 is the valve factor at zero DP, VF_{DP} is the valve factor predicted at a pressure differential of DP, and CF_{DP} is the DP Correction Factor. The correction factor is a function of the valve class. The calculated valve factor and zero DP valve factor are dependent on valve size and class.

Flow Isolation Valve Factor - Valve factor calculated using the stem thrust at flow isolation (WP-164).

Spread Disk Valve Factor - Valve factor calculated using the stem thrust during disk spreading (WP-164).

4.2 White Paper 154

a. Section 3.2 - The data from two EPRI tests of 150 lb 6 inch valves are excluded because they have "inconsistently low valve factors." The one EPRI valve had a factor of 0.166 at a DP of 90 psi and the other 0.142 at 275 psi. For the test with the lower DP and using the DP adjustment factor of 0.00125, this leads to a zero DP valve factor value of 0.166+0.00125x90 or 0.279. Although outside the inner band, the value from the EPRI low pressure test appears to be no lower than the four data points from 10 inch valves at DPs of 77-78 psi that resulted in measured valve factors in the range of 0.13 to 0.17. In the case of the higher pressure test, the differential pressure is very much higher (more than a factor of 2) than any other data point. Using the DP adjustment of 0.00125, this would lead to a zero DP valve factor of 0.142+0.00125x275 or 0.486. That value is actually slightly higher than the linear best fit regression line. It is not evident that either one of these data points can be excluded on the basis indicated in the white paper.

b. For the 300 lb class valves, 8 of the 25 data points are for the single 10 inch valve tested by EPRI. This would appear to weigh the results of that one valve very heavily in the resulting linear regression. We recommend that only two data points be included for this valve: one at 240 psi and the other at 630-640 psi.

c. For the 300 lb class valves, the results for the 18 inch valve (from EPRI testing) showed inexplicably low friction coefficients (<0.3) during loop testing after showing much higher (>0.5) friction coefficient at the end of preconditioning. We suggest that data from the 18 inch valve low friction data be excluded from the white paper.

4.3. White Paper 160

a. In Section 2.0 of Part D a safety factor of 1.3 on the observed "damaged" valve factors and a safety factor of 1.2 on the "undamaged" valve factors is applied. These factors are stated to be based on "engineering judgement." Although there are not enough data points to perform meaningful statistical analyses, it would be desirable to develop further justification for these values. We are particularly concerned that the white paper is, in effect, making predictions for damaged valves that are probably not truly "predictable." That is, we are concerned that valves which can sustain major damage could become totally inoperable—could not be positioned to meet their required function irrespective of the capability of the operator. We consider that it would be preferable to rework those valves that might be "damaged" at the earliest opportunity.

b. No data on size, DP, and measured valve factor are included in the version provided for review.

4.4. White Paper 164

a. No data on isolation valve factors are included in the version provided for review

REFERENCES

1. MOV Design Margin Evaluation and Diagnostic Feedback Evaluation. Downers Grove, Ill.: Commonwealth Edison Company, December 15, 1993. White Paper 129, Revision 0.

2. Black, Bill R., (Texas Utilities Electric Company, Comanche Peak Steam Electric Station), "Results of the Motor-Operated Valve Engineering and Testing Program." Proceedings of the Third NRC/ASME Symposium on Valve and Pump Testing. Hyatt Regency Hotel, Washington, DC, July 18-21, 1994. 101-141. Washington, D.C.: U. S. Government Printing Office. U. S. Nuclear Regulatory Commission, NUREG/CP-0137, EGG-2742, Volume 1.

3. Damerell, P. S., D. H. Harrison, R. O. Vollmer, T. A. Walker, M. S. Kalsi, and J. K. Wang. Gate Valve Model Report. Alexandria, Va.: MPR Associates, Inc. and Sugar Land, Tex.: Kalsi Engineering, Inc., October 1994. Electric Power Institute TR-103229, Research Project 3433-16.

4. Damerell, P. S., and D. R. McGowan. Stem Thrust Prediction Method for Anchor/Darling Double Disc Gate Valves. Alexandria, Va.: MPR Associates, Inc., November 1994. Electric Power Research Institute TR-103232, Research Project 3343-16.

5. Minimum Required Thrust and Valve Factor Calculation Methodology. Downers Grove, Ill.: Commonwealth Edison Company, March 28, 1994. White Paper 131, Revision 0.

6. Shaffer, S. J., R. D. Stockwell, E. Patterson, and J. Kannel. Friction Separate Effects Testing - Final Report. Columbus, Ohio: Battelle, November 1993. Electric Power Research Institute TR-103119, Research Project 3433-16.

7. Harrison, D. H., Algorithms For Estimating Friction Coefficients At Sliding Contacts In A Gate Valve. Alexandria, Va.: MPR Associates, Inc., September 1994. MPR-1409, Revision 2. (This is also Appendix E of Reference 5.)

8. M. G. Natrella. Experimental Statistics. Washington, D.C.: U. S. Department of Commerce, 1963. National Bureau of Standards Handbook 91

9. Van Matre, J.G., and G. H. Gilbreath. Statistics for Business and Economics. Plano, Tex.: Business Publications, Inc., 1983. Revised Edition.

MPR
ASSOCIATES INC.
ENGINEERS

Attachment A to
MPR Enclosure of
December 23, 1994

White Papers

154 "Anchor/Darling Valve Factors"

Revision 0 (Unsigned), September 15, 1994

160 "Crane Valve Factors"

Revision 0 (Unsigned), September 15, 1994

164 "Anchor/Darling Double Disk Gate Valve Factors"

Draft Revision 0, September 15, 1994

Commonwealth Edison Company

Corporate MOV Program Support

ATTACHMENT

7

JAN -9 1995

MPR
ASSOCIATES INC.
ENGINEERS

January 4, 1995

Mr. Paul Dietz
Commonwealth Edison Company
1400 Opus Place, Suite 400
Downers Grove, IL 60515

Subject: Overall Review of Commonwealth Edison White Papers

References:
1. MPR letter from R. Vollmer to P. Dietz (Commonwealth Edison) dated November 4, 1994; Review of White Paper 156
2. MPR letter from E. Wenzinger to P. Dietz (Commonwealth Edison) dated November 7, 1994; Review of White Paper 125
3. MPR letter from D. Harrison to P. Dietz (Commonwealth Edison) dated November 15, 1994; Review of White Paper 122
4. MPR letter from M. Albers to P. Dietz (Commonwealth Edison) dated December 6, 1994; Review of White Paper 129
5. MPR letter from D. Harrison to P. Dietz (Commonwealth Edison) dated December 23, 1994; Review of White Papers 154, 160 and 164

Dear Mr. Dietz:

In our work under Purchase Order 810975, we reviewed seven Commonwealth Edison white papers which document and justify key technical positions in your MOV program. The results of these reviews are documented in References 1 through 5. In addition to these reviews of individual white papers, you requested that we perform an overall review of the Commonwealth Edison program to determine if there are any key technical issues not adequately covered by the white papers. We have performed this overall review and our results are documented in this letter.

To perform this review, we first tabulated the current white papers and briefly summarized the content of each one. For this table, we grouped the white papers into 12 categories.

320 KING STREET ALEXANDRIA, VA 22314-3238 703-519-0200 FAX: 703-519-0224

- Stem Friction Coefficient
- Rate-of-Loading
- Limitorque Actuators
- Actuator Output Capability
- Gate and Globe Valve Required Thrust
- Butterfly Valve Required Torque
- Margin Evaluation
- MOV Testing
- Electrical
- Weak-Link
- 89-10 Scope
- Miscellaneous

The list of current white papers is attached to this letter. The next step in this process was to compare the key MOV technical concerns, uncertainties and issues in each of the above categories to the content of the white papers. Our identification of key issues and concerns is based on our experience with utility MOV programs and the EPRI MOV program. We reviewed the white papers to determine if each of the key issues was addressed in the white papers and if the positions appeared to be appropriate. Because of the limited scope of this task, we did not technically review the adequacy of the justifications for the positions. Note that the seven white papers discussed in References 1 through 5 were reviewed in more detail.

Our comments based on the overall review are summarized below.

Margin Evaluation

Several detailed comments in this area were provided in our review (Reference 4) of WP-129, "MOV Design Margin Evaluations and Diagnostic Test Feedback Evaluation." As you know, demonstrating the adequacy of MOVs involves showing that there is appropriate margin between the MOV actual and required performance. The key white papers related to margin evaluation are WP-107, WP-129 and WP-142; however, a considerable number of other white papers provide technical support for these three white papers. It appears from our detailed review of some of the white papers and our general review of others, that there are some overlaps and gaps in the coverage and connectivity of the white papers. As an example, in WP-154, "Anchor/Darling Valve Factors," the use of valve factors in margin evaluations per WP-129 is extensively discussed. Some of that discussion overlaps existing discussions in WP-129. However, the use of such valve factors in the context of WP-107, "Guideline for Determining Target Thrust Windows," is not covered in either WP-154 or WP-107. It appears the overall process could be clarified by a logic diagram showing how MOV margin is evaluated per guidance in all of the white papers. WP-000 (still in preparation) may be a suitable place for this logic chart.

Stem Friction Coefficient

The Commonwealth Edison approach for addressing stem friction coefficient degradation with time is to provide margin for an assumed increase of 0.05 in 36 months. This approach appears to be reasonable as an engineering judgment, but data are not presented in the white papers to justify the value and to demonstrate quantitatively that it bounds the expected performance. Further, there is some evidence from the EPRI program that stem friction coefficient under static conditions can decrease with time and cumulative strokes. This phenomenon has the effect of increasing the output thrust for torque-switch controlled valves, thereby reducing the margin against maximum permissible thrust limits. The Commonwealth Edison white papers do not identify means (e.g. trending) to address the potential for stem friction decrease.

Actuator Output Capability

Several detailed comments in this area were provided in our review (Reference 2) of WP-125, "Installed Motor Capability Evaluation." As you know, actuator gearing efficiency is an important parameter in this evaluation, and we understand that Commonwealth Edison is considering efficiency testing. Although it is not within the scope of WP-125, the adequacy of actuator regreasing criteria and intervals needs to be justified to ensure that the desired actuator performance is obtained in service.

Rate-of-Loading (ROL)

The approach used for rate-of-loading (ROL) does not account for those cases where the ROL effect is greater than 5%. The 5% rate-of-loading effect used in margin evaluations per WP-107 and WP-129 is based on 5% being the observed mean effect in Commonwealth Edison data. In WP-124, "Load Sensitive Behavior/Rate-of-Loading," instances of higher ROL in data are dismissed as being instrument uncertainty. Although a mean ROL effect of 5% is consistent with data from other plants and from EPRI testing, instances of ROL higher than 5% do occur as discussed in EPRI TR-103226, "Methods to Address Rate-of-Loading in Torque Switch Controlled MOVs." However, these variations from a mean value of 5% appear to be a "random" variation. ROL values higher than 5% need to be considered, but the random variation above 5% should be combined in a statistically meaningful manner with other sources of uncertainty. We understand through Mr. Bunte that Commonwealth Edison is evaluating this type of approach.

Gate and Globe Valve Required Thrust

The methods for determining gate valve required opening thrust are not explained in the white papers. Specifically, two situations need to be separately evaluated and the maximum value used: (1) unwedging thrust, which is mainly dependent on how hard the valve is closed and (2) DP thrust, which is dependent on the conditions during opening.

WP-154, 160 and 164 describe required valve factors for several groups (type, manufacturer, and pressure class) of gate valves, which are to be used when the valves are not DP tested. For valve groups not covered by these white papers, the valve factors to be used are not defined specifically by white papers. In WP-129 a prioritized logic is presented for determining valve factor. In the absence of appropriate DP test data, a valve factor of 0.5 ends up being used. Based on the results in WP-154, 160 and 164, 0.5 may not be a conservative value.

The effects of gate valve pressure locking and thermal binding are not addressed in the white papers. The NRC expectations on these issues in the context of Generic Letter 89-10 are not completely defined. Pressure locking and thermal binding may be addressed in a separate generic letter. Accordingly, the treatment of pressure locking and thermal binding in your GL 89-10 program documentation may not have to be immediately resolved. However, pressure locking and thermal binding are genuine technical concerns and eventually a technical approach and justification will be required in this area. We suggest a documented approach in a "white paper" form be developed by Commonwealth Edison.

The potential for increased globe valve thrust requirements under blowdown conditions (such as observed in one valve test in the EPRI program) does not appear to be addressed in the white papers. If Commonwealth Edison has no GL 89-10 globe valves in blowdown service, then this concern does not need to be addressed.

Butterfly Valve Required Torque

Although we did not review these white papers in detail, there does not appear to be a justification of the butterfly valve torque methods documented in WP-147 and WP-157 against butterfly valve data. The methods appear to be based on a mixture of the approach in EPRI NP-7501 and vendor-supplied coefficients. Based on the experience in developing the final EPRI butterfly valve method in EPRI TR-103224, "Butterfly Valve Model Description Report," the Commonwealth Edison approach may not be justified. For example, some of the coefficients in NP-7501 were modified based on results from testing. Also, in some instances vendor-supplied coefficients were found to be non-conservative.

As discussed in EPRI TR-103224, the effects of upstream flow disturbances such as elbows on the required butterfly valve torque are important. It appears that this issue is not addressed in the required torque calculation methods.

Weak-Link

Based on our experience with GL 89-10 programs at other plants, it is helpful and useful to document the criteria used for performing weak-link evaluations. The benefits come not only from clarifying and justifying the criteria, but also from the fact that use of this document simplifies procurement and review of externally supplied analyses. Two of the white papers (WP-117 and WP-162) address specific criteria, but there appears to be no overall document covering criteria for weak-link evaluations. We suggest such a document be prepared.

Miscellaneous

None of the MOV evaluations described in the white papers address the MOV capability to survive motor stall. Survival of motor stall is not necessarily required within the context of GL 89-10 (since stall events typically do not occur within the design basis envelope). Inadvertent motor stall is not a common event, but it can occur during MOV testing. It is good practice, if practical, to have an MOV configuration which can withstand motor stall without damage. Note that evaluation of a motor stall scenario requires defining a realistic lower-bound value for stem friction coefficient. Based on our experience with in-plant testing and EPRI flow loop testing, the stem friction coefficient can be as low as 0.05 under static loading conditions.

Criteria should be defined for inspection and maintenance of gate valve internals, to ensure that they are in a condition to promote reliable operation following instances in which a valve is disassembled. Key criteria include the following.

- All sharp edges on disk guide slots, disk OD and seat ring ID should be rounded to achieve the minimum edge contours defined in EPRI TR-103229 "Gate Valve Model Report."

- The body guide rail width and disk guide slot width should be within the manufacturers specified range. Discrepancies should be corrected.

- There should be no evidence of material damage on disk guide slots, body guide rails, disk sealing face, seat ring sealing face, stem or stem-to-disk connection. Damage indications should be corrected and the root cause should be resolved.

Please call if you have any questions or comments on this letter.

Sincerely,

Paul S. Damerell

Paul S. Damerell

Enclosure

Summary of Commonwealth Edison White Papers by Category

CATEGORY: STEM FRICTION COEFFICIENT

WP - #	Title	Scope/Objectives
101	Justification for CECo Stem Coefficient of Friction Assumptions for Motor Operated Valves (MOVs) with Rising Stems	For rising stem MOVs, this white paper indicates that margin for lubricant degradation is to be included based on a change in stem-to-stem nut friction coefficient from 0.15 to 0.20 (0.10 to 0.15 in Westinghouse valves) in 36 months. For torque closed valves, this degradation is included as margin on the bottom of the target thrust window. For limit closed valves, this degradation is included as a margin on actuator torque capability and torque rating.
103	Classification of Stem-to-Stem Nut Cleaning/Relubrication as Minor Maintenance	This white paper justifies the classification of stem-to-stem nut cleaning and relubrication as minor maintenance, meaning that post-maintenance diagnostic testing is not required. Simply stated, the justification is based on the fact that the design calculations cover a range of conditions ranging from freshly lubricated to degraded lubrication.
139	Basis for using Static, As-Left Stem Friction Coefficient in Design Basis Calculations	For rising stem, torque-switch controlled MOVs, this white paper justifies use of diagnostic results from static testing with a freshly relubricated stem on the basis that the differences in performance in going to design basis conditions are adequately accounted for. The principles are given but no quantitative values are provided.
143	Stem Factor Determination using Test Data and Activities to Improve Stem Factors that are Larger than Expected	For rising and rising rotating stem valves, this white paper indicates stem factor at TST is calculated as the ratio of torque to thrust. Prioritized guidance for torque measurement sources are provided. A checklist of items is provided to consider when stem factor is higher than expected. Quantitative criteria for "higher than expected" are not provided.

CATEGORY: RATE-OF-LOADING		
WP - #	**Title**	**Scope/Objectives**
124	Load Sensitive Behavior/Rate-of-Loading	For gate valves, this white paper indicates that: (1) If a valve is tested at ≥80% DP, use lesser of dynamic or static thrust at TST, and (2) If a valve is tested at <80% DP, use thrust at TST from static test without correction. The position is justified because average observed ROL in 1991 data is low (5%).
153	Evaluation of Load Sensitive Behavior (LSB) and Valve Factor Outliers	This white paper provides criteria for identifying LSB outliers (amount of LSB exceeds measurement uncertainties), and valve factor outliers (>1.2 for globes and >0.8 for gates). The white paper also provides checklists for determining potential causes of outliers in terms of performing better data evaluations or identifying extenuating circumstances.
CATEGORY: LIMITORQUE ACTUATORS		
WP - #	**Title**	**Scope/Objectives**
102	MOV Torque Switch Limiter Plates	For Limitorque actuators, this white paper provides justification for treating limiter plates as a non-safety related component. Accordingly, it is concluded to be acceptable, without further justification, to remove limiter plates.
122	Limitorque Operator Thrust and Torque Rating Extension	For Limitorque SMB-000 through SMB-1 actuators, this white paper provides a curve for thrust limit extension based on number of loading cycles. Torque limits are not extended but overtorquing can be evaluated in terms of number of allowable cycles.
148	Increase in Gear Box (HBC) Output Torque Ratings	This white paper documents increased output torque ratings for Limitorque HBC gear boxes used with butterfly valves. Rating increases are given for H0BC, H1BC, H2BC, H3BC, H4BC, H5BC, H6BC and H7BC gear boxes, based on information given by Limitorque from their own testing.
156	AISI 1018 Motor Pinion Key Torque Limits	This white paper provides allowable torque loadings for different actuator configurations based on 1018 key strength limits.

CATEGORY: ACTUATOR OUTPUT CAPABILITY		
WP - #	**Title**	**Scope/Objectives**
125	Installed Motor Capability Evaluation	This white paper provides equations and values of necessary constants for determining Limitorque actuator output torque capability under design basis conditions. (Analogous to Limitorque "SEL" document).
138	Use of Generic Spring Pack Curves	This white paper describes the use of generic spring pack curves for actuator performance determination.
159	Capability Requirements for MOVs with Potential for Stem/Stem Nut or Worm/Worm Gear Locking	This white paper provides guidance for evaluations which need to be performed to determine required actuator capability for self-actuating MOVs.
163	Evaluating Thrust on SMB, SB and SBD Type Limitorque Actuators using Limit Switch Control	This white paper describes methods for using limit switch control for closure strokes of rising stem valves using Limitorque SMB, SB and SBD actuators. Four methods are described covering: no torque switch, two variations of torque switch in series, and torque switch in parallel.

CATEGORY: GATE AND GLOBE VALVE REQUIRED THRUST		
WP - #	**Title**	**Scope/Objectives**
107	Guideline for Determining Target Thrust Windows	This white paper describes how target thrust window is determined including effects of diagnostic uncertainty, TS repeatability, ROL and SF degradation. The paper does not give explicit equations, but gives some default factors: (1) unwedging factor = 0.8, (2) stem friction (undegraded) = 0.15, and (3) stem friction (degraded) = 0.20 (or 0.05 + undegraded).
128	Rotating Rising Stem Valves	This white paper develops the required thrust and torque equations for rising, rotating stem valves, using the "conventional" terms developed for rising stem valves.
131	Minimum Required Thrust and Valve Factor Calculation Methodology	For closing strokes of rising stem gate and globe valves, this white paper provides formulas for calculating required thrust and also provides methods to calculate valve factor from data. Guidelines for extrapolating measured thrust are also included.

CATEGORY: GATE AND GLOBE VALVE REQUIRED THRUST (Continued)		
WP - #	**Title**	**Scope/Objectives**
134	EPRI's MOV Testing Program Measured Valve Factors	This white paper presents the results from EPRI MOV testing of gate valves in terms of "valve factors." (Note: EPRI presents their results in terms of apparent friction coefficient.) The formulas for determining open and close valve factor from apparent friction coefficient are included in the white paper. Several guidelines are included in the position regarding the effects of DP, flow, temperature and blowdown.
146	Disc Unwedging Factor	For gate valve opening strokes, this white paper justifies use of an unwedging factor of 0.8 when no data are available.
153	Evaluation of Load Sensitive Behavior & Valve Factor Outliers	This white paper provides criteria for identifying LSB outliers (amount of LSB exceeds measurement uncertainties), and valve factor outliers (>1.2 for globes; >0.8 for gates). The white paper also provides checklists for determining potential causes of outliers in terms of performing better data evaluations or identifying extenuating circumstances.
154	Anchor/Darling Valve Factors	This white paper provides valve factor values for A/D carbon steel, flexible wedge gate valves (for closure strokes) which are a function of DP and size. "Conservative" and "bounding" valve factors are provided which are used in different manners in the context of WP-129. (Use in WP-107 is not spelled out.)
159	Capability Requirements for MOVs with Potential for Stem/Stem Nut or Worm/Worm Gear Locking	This white paper provides guidance for evaluations which need to be performed to determine required actuator capability for self-actuating MOVs.
160	Crane Valve Factors	This white paper provides valve factor values for Crane carbon steel and stainless steel flexible wedge gate valves (for closure strokes), which are a function of DP and size, and which cover blowdown situations, including potential damage. "Conservative" and "bounding" valve factors are provided which are used in different manners in the context of WP-129. (Use in WP-107 is not spelled out.)
164	Anchor/Darling Double Disc Gate Valve Factors	This white paper provides valve factor values for A/D double disc gate valves (for closure strokes), which are a function of DP and size. "Conservative" and "bounding" valve factors are provided which are used in different manners in the context of WP-129. (Use in WP-107 is not spelled out.)

CATEGORY: GATE AND GLOBE VALVE REQUIRED THRUST (Continued)		
WP - #	**Title**	
165	Valve Factor Predictability for Globe Valves	This white paper indicates that a valve factor of 1.1 should be used for globe valves and provides guidance for selecting the proper area for use in calculating DP thrust.
168	Gate Valve Factor Outlier Analysis	This white paper may be deleted or may be restricted to addressing variations in the "valve condition" load. Discussion of valve factor outliers will probably be incorporated in WP-154, 160 and 164.

CATEGORY: BUTTERFLY VALVE REQUIRED TORQUE		
WP - #	**Title**	**Scope/Objectives**
144	GL 89-10 Motor Operated Butterfly Valve Evaluation—Minimum Required Torque (Seating/Unseating) for the Initial Operability Review	For butterfly valves, this white paper documents an approach to calculate seating/unseating torque which is based on EPRI NP-7501. The approach does not consider fluid dynamic torque and is only considered applicable for initial operability review. Manufacturers information for determining seating/unseating torque is included in the white paper.
147	Butterfly Valve Fluid Dynamic Torque Calculations Based on EPRI NP-7501	This white paper documents the baseline method for determining torque requirements (not just fluid dynamic torque) for GL 89-10 butterfly valves in liquid (water) service. It is based on the method in EPRI NP-7501. Deviations from the EPRI methodology are required to be documented and justified in individual valve calculations.
157	Butterfly Valve Case Specific Assumptions used in Fluid Dynamic Torque Calculations	This white paper documents specific cases where deviations from the butterfly valve torque calculation method in WP-147 were taken. The principal deviations relate to use of vendor information for torque coefficients and bearing friction coefficients.

Summary of Commonwealth Edison White Papers by Category

CATEGORY: MARGIN EVALUATION

WP - #	Title	Scope/Objectives
107	Guideline for Determining Target Thrust Windows	This white paper describes how target thrust window is determined including effects of diagnostic uncertainty, TS repeatability, ROL and SF degradation. The paper does not give explicit equations, but gives some default factors: (1) unwedging factor = 0.8, (2) stem friction (undegraded) = 0.15, and (3) stem friction (degraded) = 0.20 (or 0.05 + undegraded).
129	MOV Design Margin Evaluation and Diagnostic Test Feedback Evaluation	This white paper provides margin criteria for evaluating an MOV setup. Margin is determined to be either high, medium, low or none.
130	MOV Problem Resolution	This white paper provides guidance for resolving MOV situations which have no, low, or medium margin.
142	GL 89-10 Motor Operated Butterfly Valve Margin Evaluation	For limit-closed and torque-closed butterfly valves, this white paper defines the margins which have to be satisfied for the butterfly valve to be considered operable. Margins are defined for both the open and close stroke directions.

CATEGORY: MOV TESTING

WP - #	Title	Scope/Objectives
108	DP Testing of Motor-Operated Valves	For gate and globe valves, this white paper provides guidance for determining which valves are practicable to DP test and for methods of DP testing.
123	Testing of Torque Sealed Motor Operated Butterfly Valves (ZION)	For butterfly valves, this white paper defines an approach for obtaining stem strain data in addition to VOTES torque cartridge (VTC) data in pre-installation static testing of selected butterfly valves at Zion. Also, use of VTC and strain gage data to determine HBC unit efficiency is discussed.
126	Test Close-Out Criteria for DP Testing	For gate and globe valves, this white paper describes actions required in the field to determine MOV operability and to close out a DP test package.
135	Post Maintenance Testing Recommendations	This white paper documents the needed post maintenance testing for several different MOV maintenance activities.
137	Independent Review of MOV test Results	This white paper indicates that an independent review of MOV diagnostic test results should be provided. The Station MOV Coordinator is suggested as the independent reviewer. A series of suggested review checklist items are included in the white paper.

CATEGORY: MOV TESTING (Continued)

WP - #	Title	Scope/Objectives
145	Evaluating Auxiliary Sensor Tests	For rising stem valves, this white paper provides guidance for applying torque correction factors to VOTES thrust measurements.
151	Determination of Equipment Inaccuracy of Measured Thrust Values for Test Evaluation	This white paper provides a methodology and worksheet for determining uncertainty associated with VOTES thrust measurements. Torque switch repeatability is included the uncertainty determination.
152	Butterfly Valve Static and Dynamic Test Acceptance Criteria and Extrapolation Methods	For limit closed butterfly valves, this white paper provides methods for evaluating diagnostic test results including interpreting traces, extrapolating test results, and establishing test acceptance criteria. It covers both open and closure strokes under both static and DP conditions.
166	Low DP Load Testing	For rising stem valves, this white paper provides a criterion for determining whether DP test conditions qualify as "low load". The criterion is a DP thrust of 2000 lbs or less. When this criterion is satisfied, DP tests are neither useful nor necessary. Minimum margins to be implemented in such cases are also provided.
167	Inertia Factor Test Data Results	For rising stem valves operating in the closure direction, this white paper summarizes data for inertia factor (ratio of maximum developed thrust to thrust at TST). No methodology for determining inertia factor is provided; instead, users are instructed to find data for a similar application.

CATEGORY: ELECTRICAL

WP - #	Title	Scope/Objectives
105	Sensitivity of MOV Terminal Voltages to Changes in Thermal Overload Heater Elements	This white paper determines factors for several actuator motors which determine the effect on motor terminal voltage of changing the thermal overload heater element. Only changes to lower current rating heaters are covered, since these are the only changes that can result in a reduced terminal voltage.

Summary of Commonwealth Edison White Papers by Category

WP - #	Title	Scope/Objectives
CATEGORY: ELECTRICAL (Continued)		
106	Design Basis for AC Voltage Calculations to Determine MOV Operability during Degraded AC Voltage	For AC motor actuators, this white paper provides top level guidance on determining degraded voltage conditions. The minimum voltage at the MCC is defined based on the minimum expected voltage from offsite transmission sources, reduced by 1%. Voltage drop to the motor terminal is accounted for using locked rotor current.
119	MOV Voltage Measurement	This white paper justifies the use of voltage measurements made at the MCC rather than at the motor terminals.
127	Transient In-Rush Current	This white paper justifies neglecting measured transient inrush current and using locked rotor current for cable voltage drop calculations.
169	Valve Actuator Motor Power Factor	For AC motors in valve actuators, this white paper provides a set of power factors to be used in motor terminal voltage calculations. The paper is expected to be updated based on the results of Commonwealth Edison motor testing.

WP - #	Title	Scope/Objectives
CATEGORY: WEAK-LINK		
117	Use of Yield Strength as a Component Failure Criteria	This white paper justifies the use of minimum yield strength as a valve component stress criteria in load capacity calculations.
162	MOV Accelerations for Non-Seismically Designed Piping	This white paper provides guidance on how to determine seismic accelerations for MOVs in non-seismically analyzed piping. The guidance basically directs limited scope computer analyses to be performed.

Summary of Commonwealth Edison White Papers by Category

CATEGORY: 89-10 SCOPE		
WP - #	**Title**	**Scope/Objectives**
132	Removal of Standby Gas Treatment Butterfly Valves from GL 89-10 Program	This white paper provides justification for removing standby gas treatment butterfly valves from the GL 89-10 Program, based on guidance provided by the NRC in GL 89-10 and its supplements.
133	Ongoing Site Specific MOV Program Living Schedule	
155	Deletion of Valves from GL 89-10 Program	This white paper provides guidance based on GL 89-10 and its supplements covering the needed justification for removing an MOV from the GL 89-10 program.
150	CECo's Position on PWR MOV Misposition	For MOVs installed in PWRs, this white paper provides criteria and a methodology to determine whether mispositioning needs to be considered as part of defining the MOV design basis conditions.

CATEGORY: MISCELLANEOUS		
WP - #	**Title**	**Scope/Objectives**
000	MOV Program Document	This white paper provides a roadmap to all of the white papers in the context of the overall Commonwealth Edison GL 89-10 program.
104	MOV Failure Definition for Tracking and Trending	This white paper defines the set of conditions which constitute an MOV "failure" for the purposes of tracking and trending. The conditions are based on criteria which fail to be satisfied during operation of the MOV.
115	Manual Seating and Backseating of MOVs	This white paper provides guidance on the approach to manually seat and backseat valves, so that valve damage is avoided. The methods require measurement of handwheel torque, handwheel rim pull, or VOTES strain gage information.

Summary of Commonwealth Edison White Papers by Category

CATEGORY: MISCELLANEOUS (Continued)		
WP - #	Title	
136	MOV Training Requirements	This white paper documents the training requirements for personnel involved in the GL 89-10 Programs. Specific elements of training are mentioned, including: general understanding of valves, generic training on MOVs, mechanical and electric maintenance of MOVs, site specific procedures, and engineering design training.
141	Reverse Pressurization of Jamesbury Butterfly Valves Installed at Byron and Braidwood	For seating/unseating of Jamesbury butterfly valves, the required torque determined using the "shaft downstream" assumption can also be used for the "shaft upstream" configuration, without any multiplier.
161	Modification/Maintenance Considerations	This white paper documents lessons learned and key concerns that need to be addressed for performing MOV equipment modifications (e.g, motor change, gear change).

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