Enclos:re to WO 96-0107

ENCLOSURE

Calculation of the analyses, thrust and actuator capabilities of EMHV 8802A/B

FORM APF-05D-001-01, REV. 00

CALCULATION COVER SHEET

CALC NO. XX-M-040

DCP/CCP NO. N/A

CALCULATION STATUS DESIGNATION: ☐ PRELIMINARY ☐ COMMITTED ☒ FINAL ☐ SUPERSEDED ☐ VOIDED

CLASSIFICATION: ☒ SAFETY-RELATED ☐ SPECIAL-SCOPE ☐ NONSAFETY-RELATED

COMPUTER CODE/VERSION: N/A

CALCULATION SUBJECT: Pressure Locking Evaluation for Susceptible Valves Per GL 95-07

DESCRIPTION/REVISION SUMMARY:

This calculation determines the maximum bonnet pressure that can be accommodated by valves EJHV8811A/B, EJHV8840, EMHV8802A/B, and ENHV0001/7 with their current motor capability and with consideration for the limiting component.

Revision 1 of this calculation provided additional margin to the pullout forces to account for diagnostic inaccuracies and the repeatability of this load. In addition, valve EJHV8840 was assumed to have 2,300 psi in the bonnet due to check valve testing performed during Refuel VIII per reference V (see also reference Y). With this pressure assumed in the bonnet and the weaklink increased, the total force required was calculated to determine what the load would be for manual operation of the actuator.

⚠ JJH 4/5/9 LDR 4.5.96

REV.	ORIG/DATE	VERF/DATE	APP/DATE	REL
1	Lenny D. Ratzlaff 4.5.96	John Holland 4-5-96	Erin J. Rottler 4-5-96	

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|⚠ STH 4/5/96 LDF 4-5-96

I. PURPOSE / INTENT

The purpose of this calculation is to determine the maximum bonnet pressure due to pressure locking that valves EJHV8811A/B, EJHV8840, EMHV8802A/B, and ENHV0001/7 can overcome based on their current capability and limiting component. This determination supports the evaluations necessary in accordance with Generic Letter (95-07) on pressure locking and thermal binding. Reference Letter WO 96-0023, PIR 95-0313, PIR 95-2170, and PIR 96-1099. |⚠

II. METHODOLOGY

The methodology being used was developed by Commonwealth Edison (ComEd) and is outlined in Attachment A.

III. DESIGN INPUTS / ASSUMPTIONS

The design inputs are referenced next to the appropriate line in the body of the calculation in Section IV. The references can be found in Section VI.

The basic concept of this phenomenon is that pressure builds in the bonnet and is sealed in by the disc. The lesser of the open torque output capability of the motor/actuator and the limiting component (either the valve or actuator) determines the maximum amount of pressure that can be overcome in the bonnet. The current thrust/torque calculation for each MOV is utilized in determining the motor output torque and the limiting component.

IV. CALCULATION

The following pages provide the calculation of the maximum bonnet pressure. In each case, the motor is the limiting thrust component. The resultant bonnet pressure is back-calculated based on this limit from the ComEd methodology.

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CALCULATION SHEET

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ORIGINATOR: Larry D. Ratzlaff DATE: 4.5.96 VERIFIED BY: John Holland DATE: 4-5-96

EJHV8811A

INPUTS

Reference

[N/A]	Bonnet Pressure	Pbonnet	===>	256	psi	(Back-calculated)
[M]	Upstream Pressure	Pup	===>	46	psi	(Would be similar for DP on EJHV8811A)
[N/A]	Downstream Pressure	Pdown	===>	0	psi	(Conservative DP value)
[Att. B]	Disk Thickness	t	===>	2.56	inches	
[Att. B]	Seat Radius	a	===>	6.332	inches	JTH
[Att. B]	Effective Hub Radius	b	===>	2.560	inches	4/5/96
[Att. B]	Hub Length	L	===>	0.890	inches	
[K]	Seat Angle	theta	===>	7	degrees	LDR
[K]	Stem Diameter	Dstem	===>	2.00	inches	4.5.96
[B]	Poisson's Ratio	v	===>	0.3		
[B]	Modulus of Elast.	E	===>	27,600,000		
[L]	Static Pullout Force	Fpo	===>	12,000	lbs	
[N/A]	DP	DP	===>	0	psi	
[N/A]	Line Pressure Closed	LPc	===>	0	psi	
[N/A]	Line Pressure Open	LPo	===>	0	psi	
[N/A]	O10 DP Thrust	O10	===>	0	lbs	
[N/A]	Open DP Run Load	Fpk	===>	0	lbs	
[M]	Motor Capability	OT	===>	22,567	lbs	
[M]	Max Allow Open Thrust	MASTo	===>	45,000	lbs	

CALCULATED VALUES

D	===>	4.240E+07	lbs-in	L11	===>	0.004		Pforce	===>	2.457E+04	lbs
G	===>	1.062E+07		L17	===>	0.118		ystretch	===>	1.924E-05	inches
C2	===>	0.135		DPavg	===>	233.21373	psi	yq	===>	-2.715E-04	inches
C3	===>	0.022		Mrb	===>	-2.499E+03	lbs	ysw	===>	-2.532E-07	in/(lbs/in)
C8	===>	0.707		Qb	===>	1.528E+03	lbs/in	ybw	===>	-5.159E-07	in/(lbs/in)
C9	===>	0.297		ybd	===>	-1.516E-04	inches	ycompr	===>	3.116E-08	in/(lbs/in)
L3	===>	0		Ksa	===>	-0.292		yw	===>	-8.003E-07	in/(lbs/in)
L9	===>	0		ysq	===>	-1.006E-04	inches	Fs	===>	13,496	lbs
				VF	===>	0.000	(See Ref. P)	Mu	===>	0.28	

PRESSURE LOCKING LOADS

Fpiston	===>	805	lbs	Fpreslock	===>	4,212	lbs
Fvert	===>	7,160	lbs	Fpo	===>	12,000	lbs

Ftotal ===> = - Fpiston + Fvert + Fpreslock +F ===> 22,567 lbs

RESULTS

Motor Cap.	===>	22,567	lbs	**Ftotal**	===>	22,567	lbs
MAST	===>	45,000					

MARGIN ===> 0 %

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CALCULATION SHEET

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SHEET 4 OF 33

ORIGINATOR: Lanny D. Ratzlaff DATE: 4.5.96 VERIFIED BY: John Holland DATE: 4/5/96

EJHV8811B

INPUTS

Reference

[N/A]	Bonnet Pressure	Pbonnet	===>	245	psi	(Back-calculated)
[M]	Upstream Pressure	Pup	===>	46	psi	
[N/A]	Downstream Pressure	Pdown	===>	0	psi	(Conservative DP value)
[Att. B]	Disk Thickness	t	===>	2.56	inches	
[Att. B]	Seat Radius	a	===>	6.332	inches	JTH
[Att. B]	Effective Hub Radius	b	===>	2.560	inches	4/5/96
[Att. B]	Hub Length	L	===>	0.890	inches	
[K]	Seat Angle	theta	===>	7	degrees	LDR
[K]	Stem Diameter	Dstem	===>	2.00	inches	4.5.96
[B]	Poisson's Ratio	v	===>	0.3		
[B]	Modulus of Elast.	E	===>	27,600,000		
[N]	Static Pullout Force	Fpo	===>	15,000	lbs	
[N]	DP	DP	===>	207	psi	
[O]	Line Pressure Closed	LPc	===>	203	psi	
[O]	Line Pressure Open	LPo	===>	1	psi	
[N]	O10 DP Thrust	O10	===>	7,114	lbs	
[N]	Open DP Run Load	Fpk	===>	3,058	lbs	
[M]	Motor Capability	OT	===>	22,567	lbs	
[M]	Max Allow Open Thrust	MASTo	===>	45,000	lbs	

CALCULATED VALUES

D	===>	4.240E+07	lbs-in	L11	===>	0.004		Pforce	===>	2.338E+04	lbs
G	===>	1.062E+07		L17	===>	0.118		ystretch	===>	1.831E-05	inches
C2	===>	0.135		DPavg	===>	221.8636	psi	yq	===>	-2.583E-04	inches
C3	===>	0.022		Mrb	===>	-2.377E+03	lbs	ysw	===>	-2.532E-07	in/(lbs/in)
C8	===>	0.707		Qb	===>	1.453E+03	lbs/in	ybw	===>	-5.159E-07	in/(lbs/in)
C9	===>	0.297		ybd	===>	-1.442E-04	inches	ycompr	===>	3.116E-08	in/(lbs/in)
L3	===>	0		Ksa	===>	-0.292		yw	===>	-8.003E-07	in/(lbs/in)
L9	===>	0		ysq	===>	-9.571E-05	inches	Fs	===>	12,839	lbs
				VF	===>	0.180		Mu	===>	0.18	

PRESSURE LOCKING LOADS

Fpiston	===>	769	lbs	Fpreslock	===>	1,525	lbs
Fvert	===>	6,811	lbs	Fpo	===>	15,000	lbs
Ftotal	===>	= - Fpiston + Fvert + Fpreslock +F			===>	22,567	lbs

RESULTS

Motor Cap.	===>	22,567	lbs	**Ftotal**	===>	22,567	lbs
MAST	===>	45,000					
MARGIN	===>	0	%				

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SHEET 5 OF 33

ORIGINATOR: Lanny D. Ratzlaff DATE: 4.5.96 VERIFIED BY: John Hubbard DATE: 4/5/96

EJHV8840

INPUTS

Reference		Parameter		Value	Units	Notes
[N/A]	Bonnet Pressure	Pbonnet	===>	1,199	psi	(Back-calculated)
[A]	Upstream Pressure	Pup	===>	233	psi	
[N/A]	Downstream Pressure	Pdown	===>	0	psi	(Conservative DP value)
[Att. B]	Disk Thickness	t	===>	2.744	inches	
[Att. B]	Seat Radius	a	===>	5.011	inches	JTH 4/5/96
[Att. B]	Effective Hub Radius	b	===>	2.694	inches	
[Att. B]	Hub Length	L	===>	0.820	inches	
[Att. B]	Seat Angle	theta	===>	7	degrees	LDR 4.5.96
[Att. B]	Stem Diameter	Dstem	===>	2.50	inches	
[B]	Poisson's Ratio	v	===>	0.3		
[B]	Modulus of Elast.	E	===>	27,600,000		
[C]	Static Pullout Force	Fpo	===>	36,784	lbs	
[A]	DP	DP	===>	225	psi	
[A]	Line Pressure Closed	LPc	===>	233	psi	
[A]	Line Pressure Open	LPo	===>	8	psi	
[C]	O10 DP Thrust	O10	===>	13,103	lbs	
[C]	Open DP Run Load	Fpk	===>	5,795	lbs	
[C]	Motor Capability	OT	===>	77,764	lbs	
[D]	Max Allow Open Thrust	MASTo	===>	120,000	lbs	

CALCULATED VALUES

D	===>	5.222E+07	lbs-in	L11	===>	0.002		Pforce	===>	6.073E+04	lbs
G	===>	1.062E+07		L17	===>	0.079		ystretch	===>	3.957E-05	inches
C2	===>	0.088		DPavg	===>	1082.8379	psi	yq	===>	-2.702E-04	inches
C3	===>	0.012		Mrb	===>	-3.919E+03	lbs	ysw	===>	-1.281E-07	in/(lbs/in)
C8	===>	0.751		Qb	===>	3.588E+03	lbs/in	ybw	===>	-9.546E-08	in/(lbs/in)
C9	===>	0.284		ybq	===>	-8.214E-05	inches	ycompr	===>	2.051E-08	in/(lbs/in)
L3	===>	0		Ksa	===>	-0.159		yw	===>	-2.441E-07	in/(lbs/in)
L9	===>	0		ysq	===>	-1.485E-04	inches	Fs	===>	34,853	lbs
				VF	===>	0.474		Mu	===>	0.50	

PRESSURE LOCKING LOADS

Fpiston	===>	5,887	lbs	Fpreslock	===>	26,047	lbs
Fvert	===>	20,820	lbs	Fpo	===>	36,784	lbs

Ftotal ===> = - Fpiston + Fvert + Fpreslock +F ===> 77,764 lbs

RESULTS

Motor Cap.	===>	77,764	lbs	**Ftotal**	===>	77,764	lbs
MAST	===>	120,000					

MARGIN ===> 54 %

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	CALCULATION SHEET	CALC NO. XX-M-040 REVISION 01 SHEET 5A OF 33

ORIGINATOR: Larry D. Ratzlaff DATE: 4.5.96 VERIFIED BY: Kent B. Fuller DATE: 4.5.96

EJHV8840

INPUTS

Reference

[N/A]	Bonnet Pressure	Pbonnet	===>	2,300	psi	(Back-calculated)
[A]	Upstream Pressure	Pup	===>	233	psi	
[N/A]	Downstream Pressure	Pdown	===>	0	psi	(Conservative DP value)
[Att. B]	Disk Thickness	t	===>	2.744	inches	
[Att. B]	Seat Radius	a	===>	5.011	inches	* This calculation evaluates the total force required by the handwheel of the actuator when in the manual mode of operation and not the electrical (motor) mode.
[Att. B]	Effective Hub Radius	b	===>	2.694	inches	
[Att. B]	Hub Length	L	===>	0.820	inches	
[Att. B]	Seat Angle	theta	===>	7	degrees	
[Att. B]	Stem Diameter	Dstem	===>	2.50	inches	
[B]	Poisson's Ratio	v	===>	0.3		
[B]	Modulus of Elast.	E	===>	27,600,000		Note: The handwheel torque required to obtain 120,000 lbs of force is:
[C]	Static Pullout Force	Fpo	===>	36,784	lbs	
[A]	DP	DP	===>	225	psi	
[A]	Line Pressure Closed	LPc	===>	233	psi	==> 342 ft-lbs with a stem COF of 0.15
[A]	Line Pressure Open	LPo	===>	8	psi	
[C]	O10 DP Thrust	O10	===>	13,103	lbs	==> 416 ft-lbs with a stem COF of 0.20
[C]	Open DP Run Load	Fpk	===>	5,795	lbs	
[C]	Motor Capability	OT	===>	77,764	lbs	(Maximum handwheel shaft torque is 493 ft-lbs per reference D.)
[D]	Max Allow Open Thrust	MASTo	===>	120,000	lbs	

CALCULATED VALUES

D	===>	5.222E+07	lbs-in	L11	===>	0.002		Pforce	===>	1.225E+05	lbs
G	===>	1.062E+07		L17	===>	0.079		ystretch	===>	7.979E-05	inches
C2	===>	0.088		DPavg	===>	2183.5	psi	yq	===>	-5.448E-04	inches
C3	===>	0.012		Mrb	===>	-7.903E+03	lbs	ysw	===>	-1.281E-07	in/(lbs/in)
C8	===>	0.751		Qb	===>	7.235E+03	lbs/in	ybw	===>	-9.546E-08	in/(lbs/in)
C9	===>	0.284		ybq	===>	-1.656E-04	inches	ycompr	===>	2.051E-08	in/(lbs/in)
L3	===>	0		Ksa	===>	-0.159		yw	===>	-2.441E-07	in/(lbs/in)
L9	===>	0		ysq	===>	-2.994E-04	inches	Fs	===>	70,279	lbs
				VF	===>	0.474		Mu	===>	0.50	

PRESSURE LOCKING LOADS

Fpiston	===>	11,290	lbs	Fpreslock	===>	52,523	lbs
Fvert	===>	41,983	lbs	Fpo	===>	36,784	lbs

Ftotal ===> = - Fpiston + Fvert + Fpreslock +F ===> 120,000 lbs

RESULTS

Motor Cap. ===> 77,764 lbs **Ftotal** ===> 120,000 lbs *

MAST ===> 120,000

MARGIN ===> 0 %

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CALCULATION SHEET

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REVISION 0

SHEET 6 OF 33

ORIGINATOR: Lanny D. Ratzlaff DATE: 4.5.96 VERIFIED BY: John Holland DATE: 4/5/96

EMHV8802A

INPUTS

Reference

Reference	Description	Symbol		Value	Units	Notes
[N/A]	Bonnet Pressure	Pbonnet	===>	3,822	psi	(Back-calculated)
[N/A]	Upstream Pressure	Pup	===>	0	psi	(Conservative DP value)
[N/A]	Downstream Pressure	Pdown	===>	0	psi	(Conservative DP value)
[Att. B]	Disk Thickness	t	===>	1.010	inches	
[Att. B]	Seat Radius	a	===>	2.006	inches	JTH 4/5/96
[Att. B]	Effective Hub Radius	b	===>	1.056	inches	
[Att. B]	Hub Length	L	===>	0.610	inches	
[Att. B]	Seat Angle	theta	===>	7	degrees	LDR 4.5.96
[Att. B]	Stem Diameter	Dstem	===>	1.25	inches	
[B]	Poisson's Ratio	v	===>	0.3		
[B]	Modulus of Elast.	E	===>	27,600,000		
[E]	Static Pullout Force	Fpo	===>	5,000	lbs	
[N/A]	DP	DP	===>	0	psi	
[N/A]	Line Pressure Closed	LPc	===>	0	psi	
[N/A]	Line Pressure Open	LPo	===>	0	psi	
[N/A]	O10 DP Thrust	O10	===>	0	lbs	
[N/A]	Open DP Run Load	Fpk	===>	0	lbs	
[F]	Motor Capability	OT	===>	13,606	lbs	
[F]	Max Allow Open Thrust	MASTo	===>	16,000	lbs	

* Mu was determined based on the worst case open coefficient for similar valves that were DP tested. (Ref. G) (EMHV8801A/B & EMHV8803A/B)

CALCULATED VALUES

D	===>	2.604E+06	lbs-in	L11	===>	0.002		Pforce	===>	3.493E+04	lbs
G	===>	1.062E+07		L17	===>	0.083		ystretch	===>	1.102E-04	inches
C2	===>	0.092		DPavg	===>	3822.1525	psi	yq	===>	-5.157E-04	inches
C3	===>	0.013		Mrb	===>	-2.346E+03	lbs	ysw	===>	-1.441E-07	in/(lbs/in)
C8	===>	0.747		Qb	===>	5.264E+03	lbs/in	ybw	===>	-1.323E-07	in/(lbs/in)
C9	===>	0.286		ybq	===>	-1.643E-04	inches	ycompr	===>	3.976E-08	in/(lbs/in)
L3	===>	0		Ksa	===>	-0.168		yw	===>	-3.161E-07	in/(lbs/in)
L9	===>	0		ysq	===>	-2.412E-04	inches	Fs	===>	20,565	lbs
				VF	===>	0.000		* Mu	===>	0.16	

PRESSURE LOCKING LOADS

Fpiston	===>	4,690	lbs	Fpreslock	===>	1,519	lbs
Fvert	===>	11,777	lbs	Fpo	===>	5,000	lbs

Ftotal ===> = - Fpiston + Fvert + Fpreslock +F ===> 13,606 lbs

RESULTS

Motor Cap.	===>	13,606	lbs	Ftotal	===>	13,606	lbs
MAST	===>	16,000					

MARGIN ===> 0 %

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CALCULATION SHEET

CALC NO. XX-M-040

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ORIGINATOR: Larry D. Ratzlaff DATE: 4.5.96 VERIFIED BY: John Holland DATE: 4/5/96

EMHV8802B

INPUTS

Reference

[N/A]	Bonnet Pressure	Pbonnet	===>	3,378	psi	(Back-calculated)
[N/A]	Upstream Pressure	Pup	===>	0	psi	(Conservative DP value)
[N/A]	Downstream Pressure	Pdown	===>	0	psi	(Conservative DP value)
[Att. B]	Disk Thickness	t	===>	1.010	inches	
[Att. B]	Seat Radius	a	===>	2.006	inches	JTH 4/5/96
[Att. B]	Effective Hub Radius	b	===>	1.056	inches	
[Att. B]	Hub Length	L	===>	0.610	inches	
[Att. B]	Seat Angle	theta	===>	7	degrees	LDR 4.5.96
[Att. B]	Stem Diameter	Dstem	===>	1.25	inches	
[B]	Poisson's Ratio	v	===>	0.3		
[B]	Modulus of Elast.	E	===>	27,600,000		
[E]	Static Pullout Force	Fpo	===>	6,000	lbs	
[N/A]	DP	DP	===>	0	psi	
[N/A]	Line Pressure Closed	LPc	===>	0	psi	
[N/A]	Line Pressure Open	LPo	===>	0	psi	* Mu was determined based on the worst case open coefficient for similar valves that were DP tested. (Ref. G) (EMHV8801A/B & EMHV8803A/B)
[N/A]	O10 DP Thrust	O10	===>	0	lbs	
[N/A]	Open DP Run Load	Fpk	===>	0	lbs	
[F]	Motor Capability	OT	===>	13,606	lbs	
[F]	Max Allow Open Thrust	MASTo	===>	16,000	lbs	

CALCULATED VALUES

D	===>	2.604E+06	lbs-in	L11	===>	0.002		Pforce	===>	3.087E+04	lbs
G	===>	1.062E+07		L17	===>	0.083		ystretch	===>	9.738E-05	inches
C2	===>	0.092		DPavg	===>	3378.026	psi	yq	===>	-4.558E-04	inches
C3	===>	0.013		Mrb	===>	-2.073E+03	lbs	ysw	===>	-1.441E-07	in/(lbs/in)
C8	===>	0.747		Qb	===>	4.653E+03	lbs/in	ybw	===>	-1.323E-07	in/(lbs/in)
C9	===>	0.286		ybq	===>	-1.452E-04	inches	ycompr	===>	3.976E-08	in/(lbs/in)
L3	===>	0		Ksa	===>	-0.168		yw	===>	-3.161E-07	in/(lbs/in)
L9	===>	0		ysq	===>	-2.132E-04	inches	Fs	===>	18,175	lbs
				VF	===>	0.000		* Mu	===>	0.16	

PRESSURE LOCKING LOADS

Fpiston	===>	4,145	lbs	Fpreslock	===>	1,343	lbs
Fvert	===>	10,409	lbs	Fpo	===>	6,000	lbs

Ftotal ===> = - Fpiston + Fvert + Fpreslock +F ===> 13,606 lbs

RESULTS

Motor Cap.	===>	13,606	lbs	**Ftotal**	===>	13,606	lbs
MAST	===>	16,000					

MARGIN ===> 0 %

CALCULATION SHEET

CALC NO. XX-M-040
REVISION 0
SHEET 8 OF 33

ORIGINATOR: Jenny D. Ratzlaff DATE: 4.5.96 VERIFIED BY: John Holland DATE: 4/5/96

ENHV0001

INPUTS

Reference

[N/A]	Bonnet Pressure	Pbonnet	===>	322	psi	(Back-calculated)
[N/A]	Upstream Pressure	Pup	===>	0	psi	(Conservative DP value)
[N/A]	Downstream Pressure	Pdown	===>	0	psi	(Conservative DP value)
[Att. C]	Disk Thickness	t	===>	1.250	inches	
[Att. C]	Seat Radius	a	===>	5.563	inches	JTH 4/5/96
[Att. C]	Effective Hub Radius	b	===>	1.813	inches	
[Att. C]	Hub Length	L	===>	4.063	inches	
[S]	Seat Angle	theta	===>	5	degrees	LDR 4.5.96
[U]	Stem Diameter	Dstem	===>	1.38	inches	
[B]	Poisson's Ratio	v	===>	0.3		
[B]	Modulus of Elast.	E	===>	27,600,000		
[R]	Static Pullout Force	Fpo	===>	7,000	lbs	
[N/A]	DP	DP	===>	0	psi	
[N/A]	Line Pressure Closed	LPc	===>	0	psi	
[N/A]	Line Pressure Open	LPo	===>	0	psi	
[N/A]	O10 DP Thrust	O10	===>	0	lbs	
[N/A]	Open DP Run Load	Fpk	===>	0	lbs	
[S]	Motor Capability	OT	===>	17,766	lbs	
[Q]	Max Allow Open Thrust	MASTo	===>	24,920	lbs	

CALCULATED VALUES

D	===>	4.936E+06 lbs-in	L11	===>	0.006		Pforce	===>	2.797E+04	lbs
G	===>	1.062E+07	L17	===>	0.142		ystretch	===>	1.994E-04	inches
C2	===>	0.164	DPavg	===>	321.87502	psi	yq	===>	-2.227E-03	inches
C3	===>	0.028	Mrb	===>	-3.683E+03	lbs	ysw	===>	-5.641E-07	in/(lbs/in)
C8	===>	0.687	Qb	===>	2.456E+03	lbs/in	ybw	===>	-4.343E-06	in/(lbs/in)
C9	===>	0.288	ybq	===>	-1.724E-03	inches	ycompr	===>	2.492E-07	in/(lbs/in)
L3	===>	0	Ksa	===>	-0.405		yw	===>	-5.156E-06	in/(lbs/in)
L9	===>	0	ysq	===>	-3.037E-04	inches	Fs	===>	15,096	lbs
			VF	===>	0.000	(See Ref. Q)	Mu	===>	0.28	

PRESSURE LOCKING LOADS

Fpiston	===>	478	lbs	Fpreslock	===>	5,790	lbs
Fvert	===>	5,454	lbs	Fpo	===>	7,000	lbs
Ftotal	===>	= - Fpiston + Fvert + Fpreslock +F			===>	17,766	lbs

RESULTS

Motor Cap.	===>	17,766	lbs	**Ftotal**	===>	17,766	lbs
MAST	===>	24,920					

MARGIN ===> 0 %

FORM APF-05D-001-02, REV. 00

CALCULATION SHEET

CALC NO. XX-M-040
REVISION 0
SHEET 9 OF 33

ORIGINATOR: Jenny D. Ratzlaff DATE: 4-5-96 VERIFIED BY: John Hubbard DATE: 4/5/96

ENHV0007

INPUTS

Reference

[N/A]	Bonnet Pressure	Pbonnet	===>	382	psi	(Back-calculated)
[N/A]	Upstream Pressure	Pup	===>	0	psi	(Conservative DP value)
[N/A]	Downstream Pressure	Pdown	===>	0	psi	(Conservative DP value)
[Att. C]	Disk Thickness	t	===>	1.250	inches	
[Att. C]	Seat Radius	a	===>	5.563	inches	JTH 4/5/96
[Att. C]	Effective Hub Radius	b	===>	1.813	inches	
[Att. C]	Hub Length	L	===>	4.063	inches	
[S]	Seat Angle	theta	===>	5	degrees	LDR 4.5.96
[U]	Stem Diameter	Dstem	===>	1.38	inches	
[B]	Poisson's Ratio	v	===>	0.3		
[B]	Modulus of Elast.	E	===>	27,600,000		
[T]	Static Pullout Force	Fpo	===>	5,000	lbs	
[N/A]	DP	DP	===>	0	psi	
[N/A]	Line Pressure Closed	LPc	===>	0	psi	
[N/A]	Line Pressure Open	LPo	===>	0	psi	
[N/A]	O1C DP Thrust	O10	===>	0	lbs	
[N/A]	Open DP Run Load	Fpk	===>	0	lbs	
[F]	Motor Capability	OT	===>	17,766	lbs	
[Q]	Max Allow Open Thrust	MASTo	===>	24,920	lbs	

CALCULATED VALUES

D	===>	4.936E+06 lbs-in	L11	===>	0.006		Pforce	===>	3.316E+04	lbs
G	===>	1.062E+07	L17	===>	0.142		ystretch	===>	2.365E-04	inches
C2	===>	0.164	DPavg	===>	381.66974	psi	yq	===>	-2.641E-03	inches
C3	===>	0.028	Mrb	===>	-4.368E+03	lbs	ysw	===>	-5.641E-07	in/(lbs/in)
C8	===>	0.687	Qb	===>	2.912E+03	lbs/in	ybw	===>	-4.343E-06	in/(lbs/in)
C9	===>	0.288	ybq	===>	-2.044E-03	inches	ycompr	===>	2.492E-07	in/(lbs/in)
L3	===>	0	Ksa	===>	-0.405		yw	===>	-5.156E-06	in/(lbs/in)
L9	===>	0	ysq	===>	-3.601E-04	inches	Fs	===>	17,900	lbs
			VF	===>	0.000	(See Ref. Q)	Mu	===>	0.28	

PRESSURE LOCKING LOADS

Fpiston	===>	567	lbs	Fpreslock	===>	6,866	lbs
Fvert	===>	6,467	lbs	Fpo	===>	5,000	lbs

Ftotal ===> = - Fpiston + Fvert + Fpreslock +F ===> 17,766 lbs

RESULTS

Motor Cap.	===>	17,766	lbs	Ftotal	===>	17,766	lbs
MAST	===>	24,920					

MARGIN ===> 0 %

	CALCULATION SHEET	CALC NO. XX-M-040 REVISION NO. 0 SHEET 10 OF 33

V. RESULTS / CONCLUSIONS

In all cases, sufficient margin exists for each of the valves to accommodate the effects of pressure locking. Further margin also exists if the actual motor capability is determined from the actual field tested stem coefficient of friction.

The resultant maximum bonnet pressures for the three valves are as follows:

Valve	Maximum Bonnet Pressure	
EJHV8811A	256 psi	
EJHV8811B	245 psi	
EJHV8840	1,199 psi (electric motor capability)	2,300 psi (manual handwheel capability)
EMHV8802A	3,822 psi	
EMHV8802B	3,378 psi	
ENHV0001	322 psi	
ENHV0007	382 psi	

VI. REFERENCES

A. Procedure TP-TS-112 Rev.0, EJHV8840 MOV DP Test
B. Mark's Standard Handbook for Mechanical Engineers, Ninth Ed.
C. WR 05781-92 and VOTES test #4 (DP test) and test #7 (Static As-Left test)
D. Calculation EJ-M-011 Rev. 6
E. WR 03220-94 and VOTES test #8 (Static As-Left test)
F. Calculation EM-M-016 Rev. 2
G. Calculation EM-M-017 Rev. 7
H. WR 03219-94 and VOTES test #8 (Static As-Left test)
I. PIR 95-0313, Pressure Locking and Thermal Binding
J. Letter WO 96-0023, Submittal to the NRC Concerning Pressure Locking and Thermal Binding of WCGS Valves for Generic Letter 95-07
K. Drawing M-724-00696
L. WR 05770-92 and VOTES test #5 (Static As-Left test)
M. Calculation EJ-M-013 Rev. 4
N. WR 05784-92 and VOTES test #3 & 4 (DP test) and WR 04959-94 test #3 (Static As-Left test)
O. Procedure TP-TS-96 Rev. 0, EJHV8811B MOV DP Test
P. Calculation EJ-M-017 Rev. 2
Q. Calculation EN-M-011 Rev. 2
R. WR 03216-94 and VOTES test #4 (Static As-Left test)
S. Calculation EN-M-005 Rev. 3
T. WR 03215-94 and VOTES test #3 (Static As-Left test)
U. Drawing M-225-00006
V. STS PE-19E and STS PE-40B performed during Refuel VIII
W. Calculation EN-M-013 Rev. 1,
X. Calculation EJ-M-019 Rev. 1
Y. PIR 96-1099 and the supporting operability evaluation per AP 28-001

1 JTH 4/5/96
LDR 4.5.96

PRESSURE LOCKING TESTING AT COMMONWEALTH EDISON

PURPOSE

ComEd has developed a MathCad model for predicting the thrust required to unseat an MOV under pressure locking conditions. This model is based on Roark's Equations for deflection of a plate with a central hub. The MathCad model is being applied to MOVs which a ComEd review determined to be potentially susceptible to pressure locking.

The purpose of the ComEd test program for pressure locking is two-fold. The test data is being used to validate the ComEd MathCad model for pressure locking. In addition, the testing is being performed to determine the whether pressure locking concerns associated with slow heat-up of fluid in the valve bonnet are justified.

The Westinghouse Owners' Group has been supporting the ComEd efforts by soliciting other utility involvement and by providing some funding for the setup and testing of valves at ComEd facilities.

RESULTS TO DATE

ComEd has tested a 10" Crane valve and a 4" Westinghouse valve under pressure locking conditions. The results of this testing are plotted on the attached sheets. As can be seen from these plots, the MathCad model has very accurately predicted the pressure locking unseating force.

In addition, ComEd performed some bonnet fluid heat-up testing on the Westinghouse MOV. This testing showed a bonnet pressurization rate of only 0.4 psi per degree temperature rise. This result is similar to results obtained by Northeast Utilities in pressure locking tests performed earlier this year. Prior to this test, as much air as possible was removed from the valve bonnet by venting through the packing. No seat leakage and very slight packing leakage were observed during the test. Based on this test data, ComEd believes that extraordinary measures (such as those taken by TVA during 1985 tests) must be taken to sufficiently remove air from the valve bonnet so that theoretical pressurization rates of 30 to 100 psi per degree temperature rise can be achieved. TVA had to shake its test valves from side to side prior to submerging them in a heated bath of water to get the measured pressurization rates of 20 to 24 psi per degree temperature rise. Such extraordinary measures are inconsistent with normal plant operations. Consequently, thermally induced pressure locking due to bonnet heat-up appears to represent a design concern rather than an immediate operability concern.

No bonnet fluid heat-up testing of the Crane 10" valve could be performed because slight seat leakage was sufficient to preclude bonnet pressurization by the thermally induced pressure locking mechanism. This information also suggests that the thermally induced pressure locking concern should not be considered a common cause type of failure which has a high probability of occurring.

FUTURE TESTING PLANS

Several utilities have expressed interest in participating in the ComEd test program by supplying test valves or performing testing of their own. This includes Entergy (Grand Gulf), Arizona Public Service, and Carolina Power & Light. Entergy is currently preparing to perform tests on a Velan valve using the ComEd test procedure. APS and CP&L are offering test valves to ComEd. In addition, Anchor/Darling has agreed to provide ComEd with a 6" double-disk gate valve for pressure locking testing. At this time, ComEd is considering testing the valves listed below during October/November of this year:

8" Borg-Warner Flex Wedge Gate Valve from APS
8" Westinghouse Flex Wedge Gate Valve from CP&L
6" Anchor/Darling Flex Wedge Gate Valve from CP&L
6" Anchor/Darling Double-Disk Gate Valve from A/D

ATTACHMENT A
CALC. NO. XX-M-040
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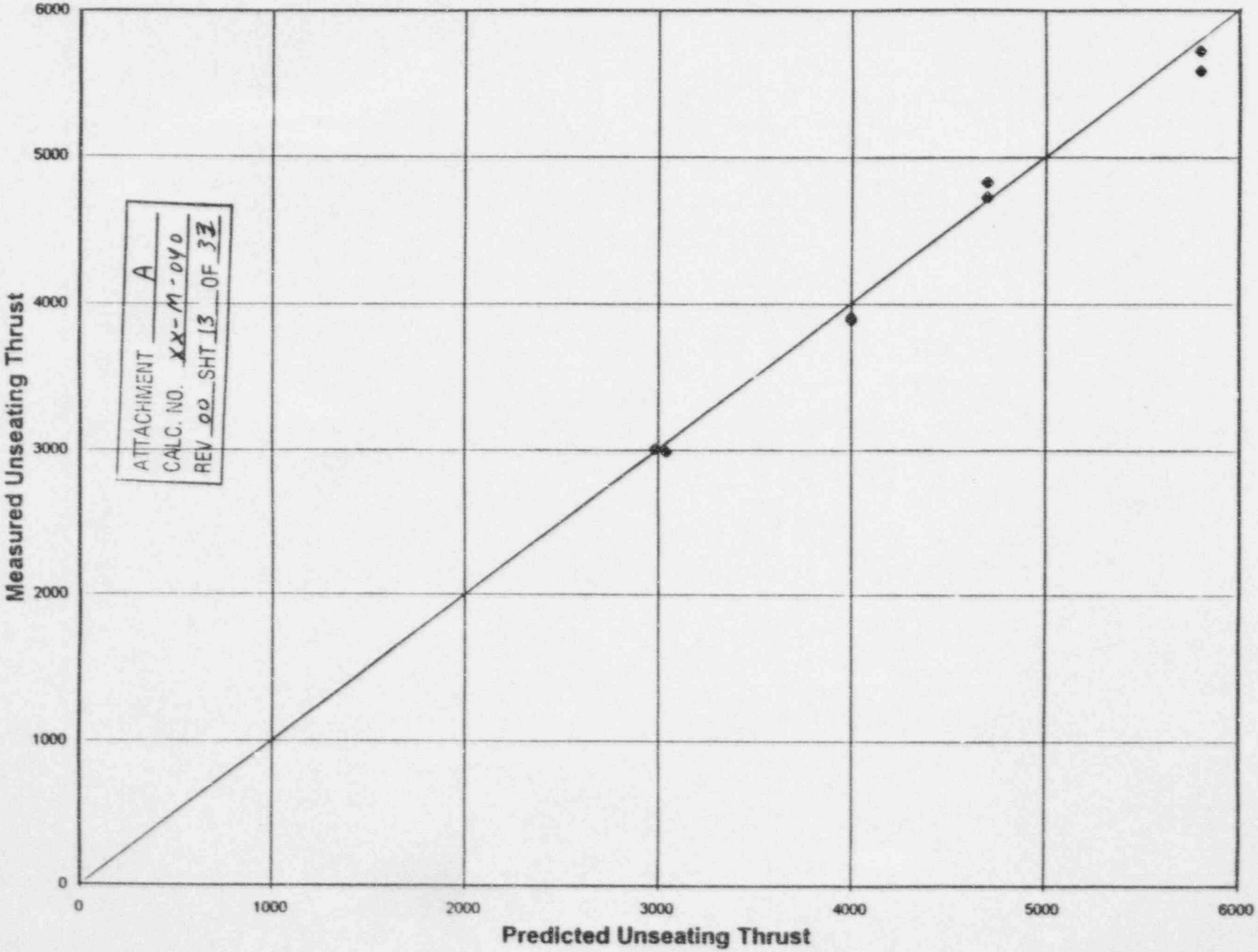
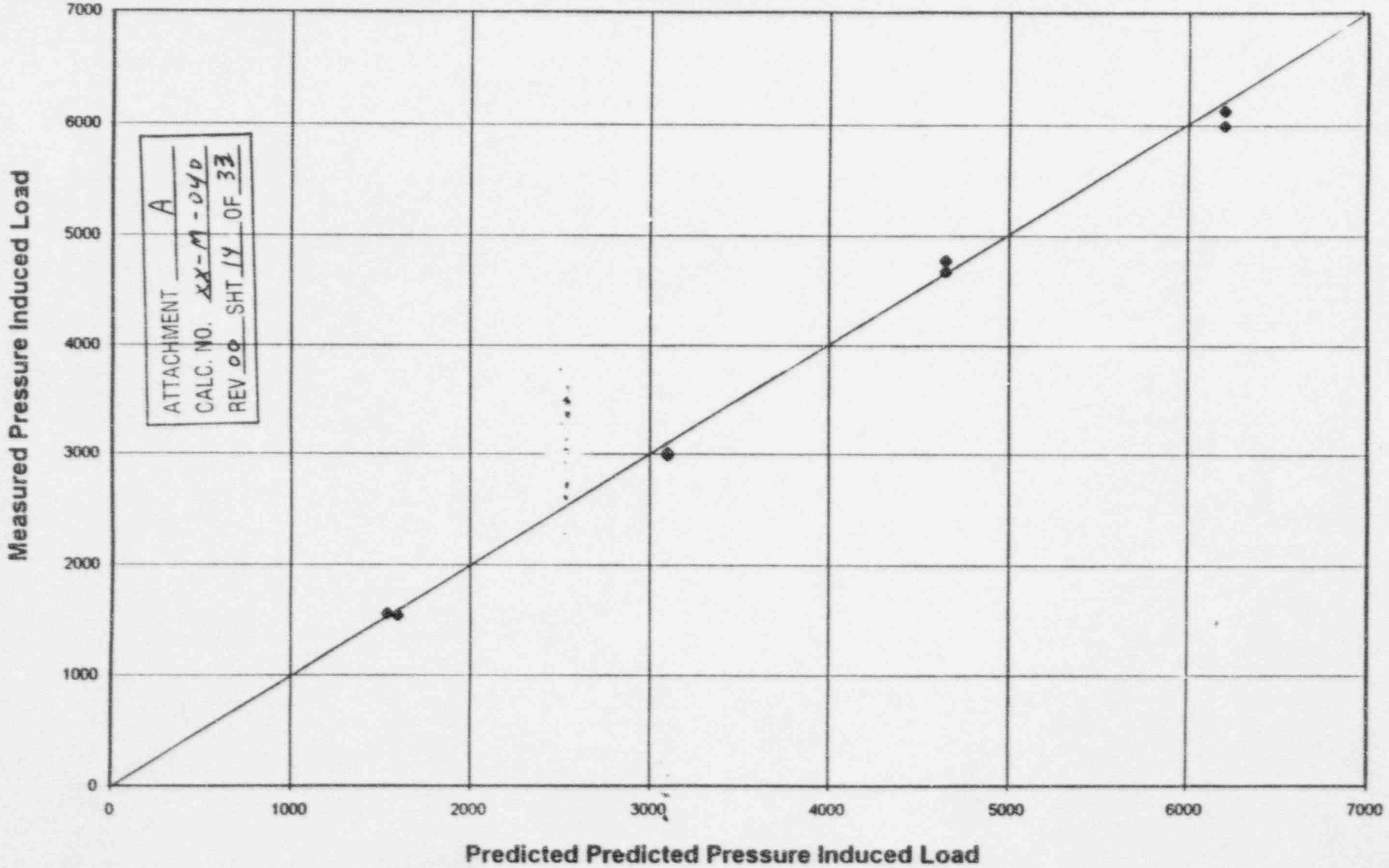
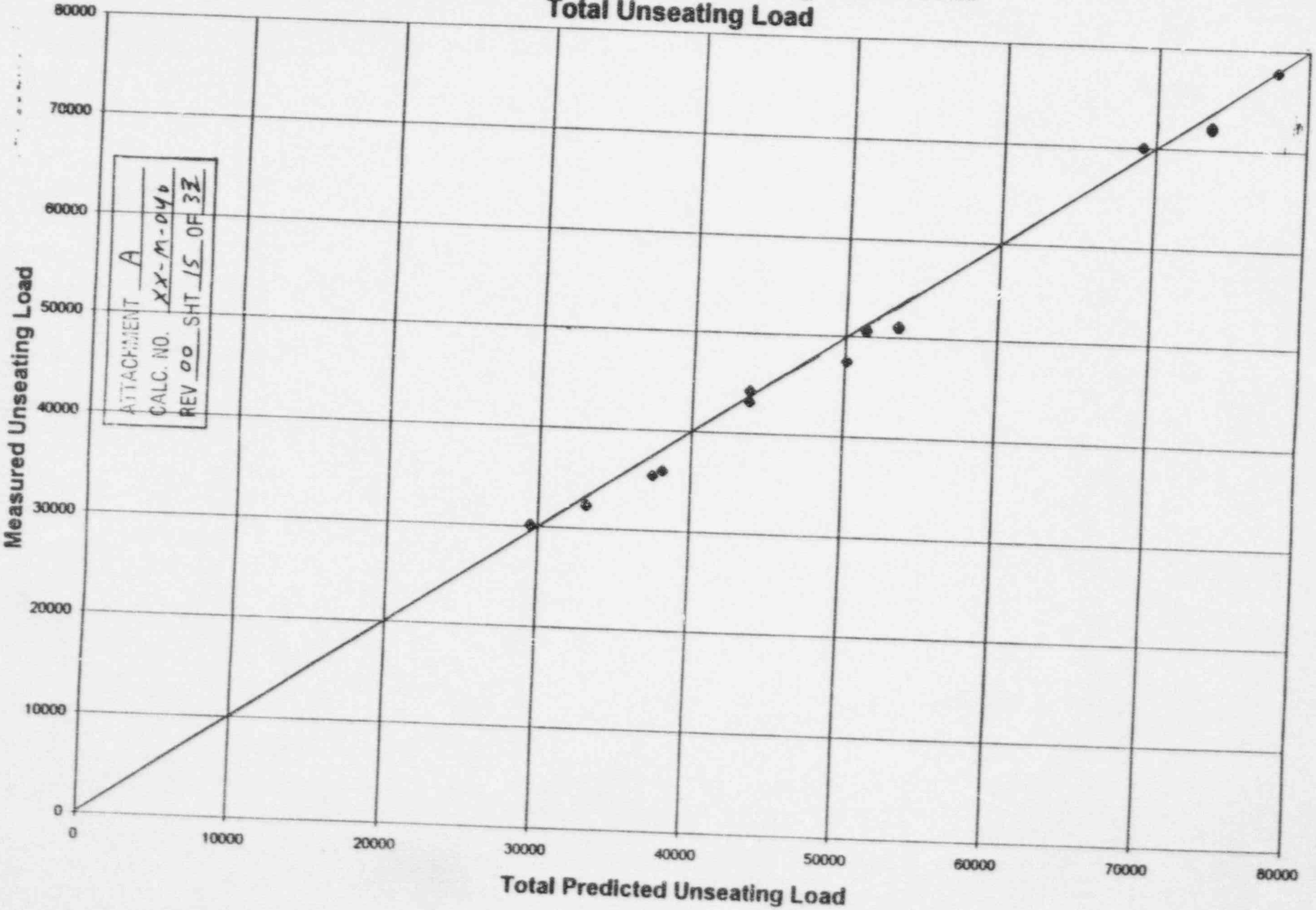
Westinghouse 4" Flex-Wedge Gate Valve
Total Unseating Load



Westinghouse 4" Flex-Wedge Gate Valve
Portion of Unseating Load Due to Pressure
(not including piston effect)



Crane 10" Valve Pressure Locking Test Results
Total Unseating Load



ATTACHMENT A
CALC. NO. XX-M-040
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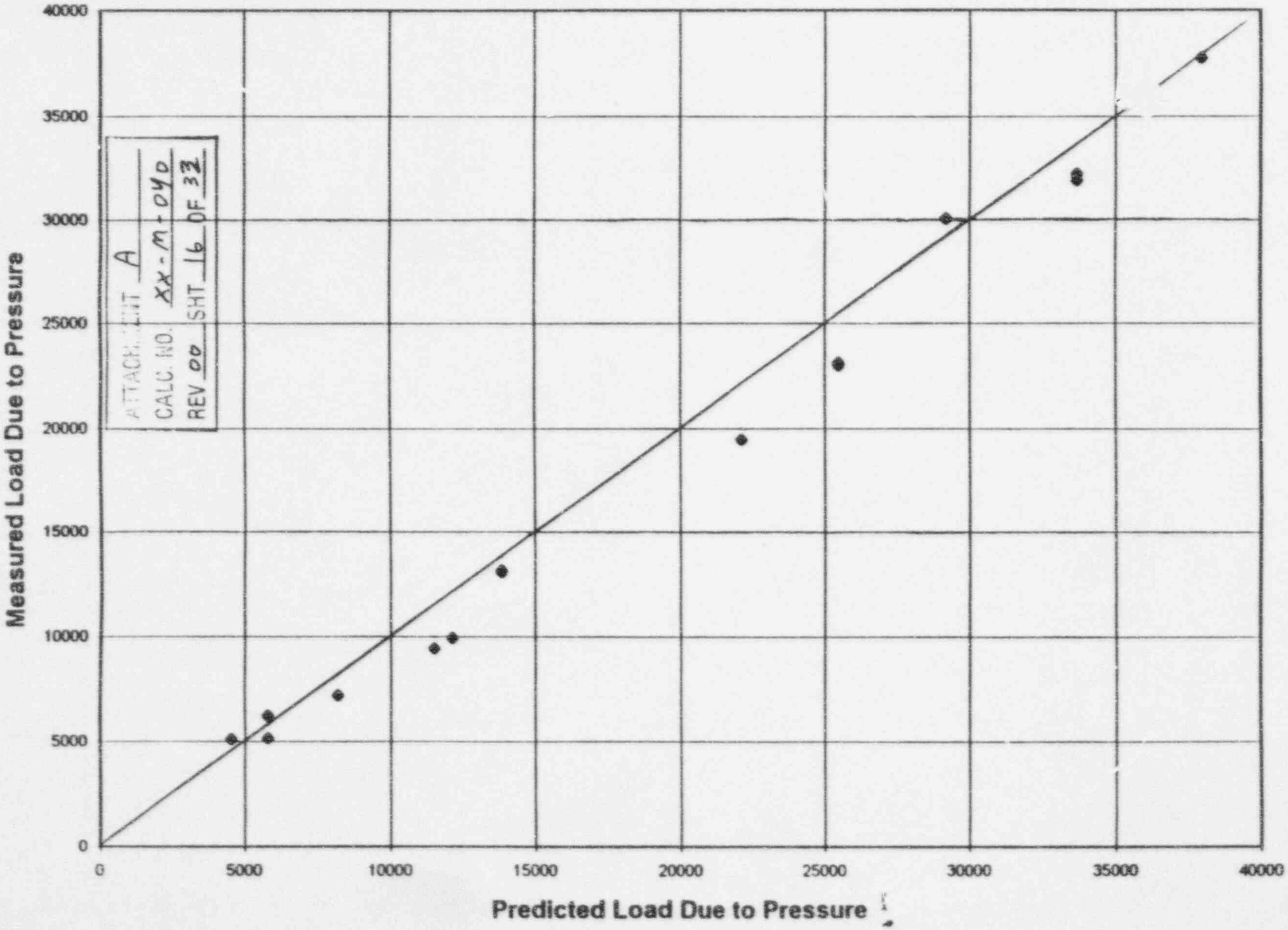
Crane 10" Valve Pressure Locking Test Results
Additional Load (above static unseating)
Due to Bonnet Pressure



I. PURPOSE/OBJECTIVE

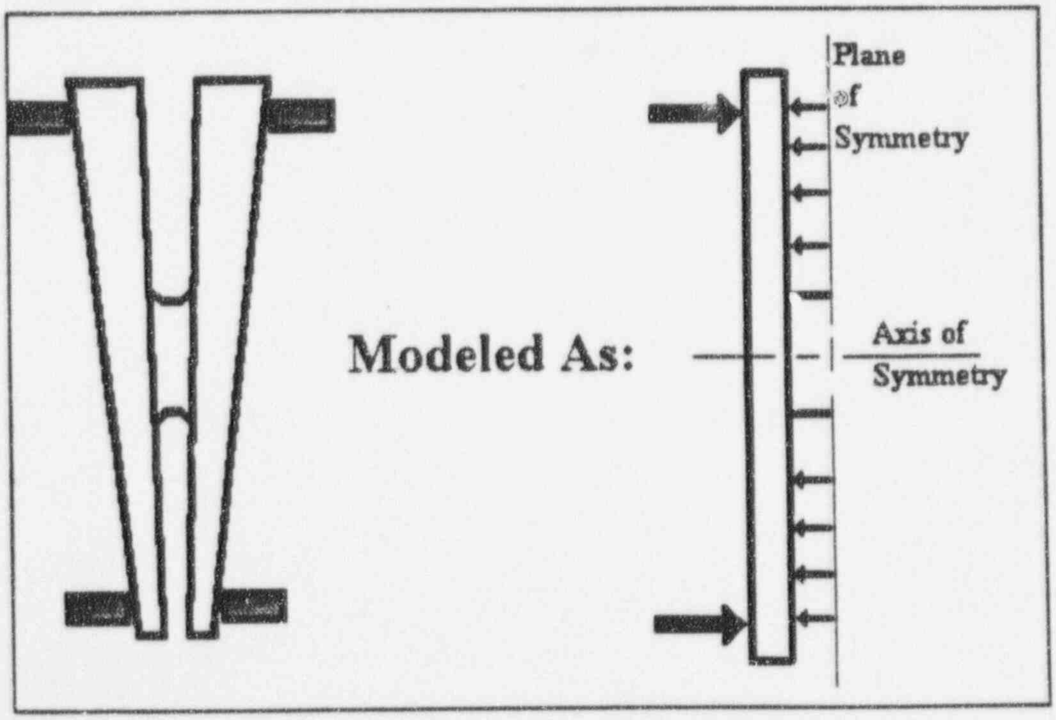
The purpose of this calculation is to available marging (MGC to Required Thrust to Open) for MOV 3-1501-22A which has been determined to be susceptible to the pressure locking phenomena. The MOV is installed in the Low Pressure Injection (LPCI) System at Dresden.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

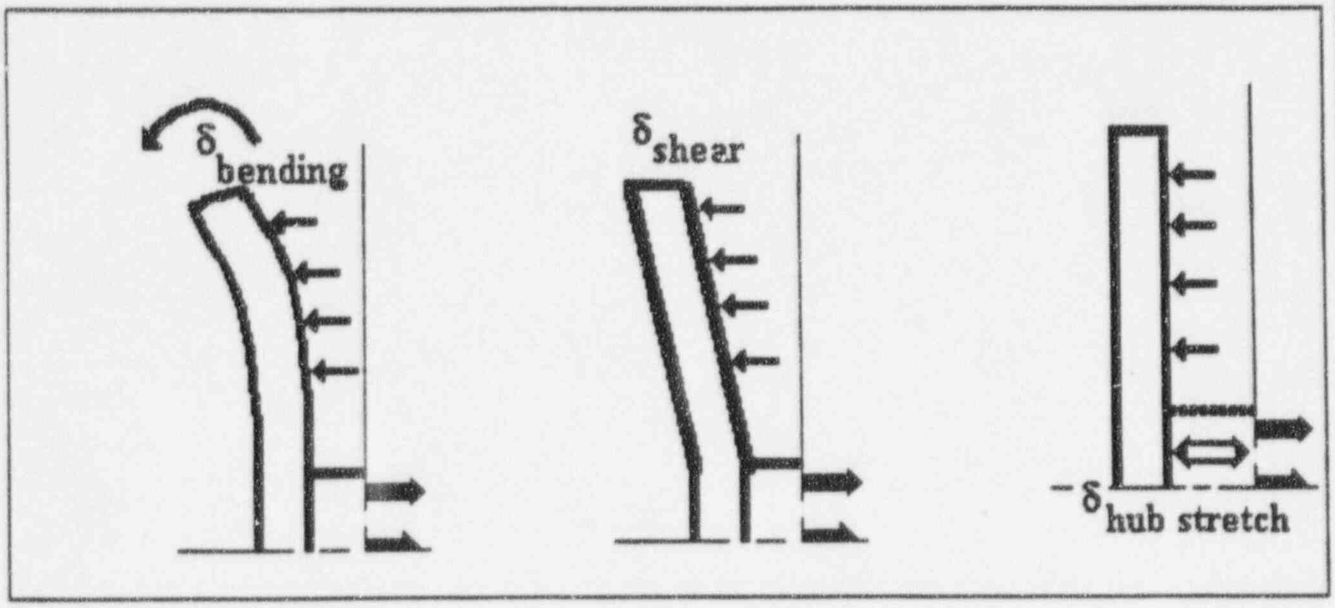
The methodology for calculating the thrust required to open the MOVs under the pressure locking scenario is based on the Reference 1 (Roark's) engineering handbook. This methodology has been previously applied by ComEd in the References 2 and 10 calculations. The methodology determines the total force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

Pressure Locking Component of Force Required to Open the Valve

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



The pressure force is assumed to act uniformly upon the inner surface of the disk between the hub diameter and the outer disk diameter. The outer edge of the disk is assumed to be unimpeded and allowed to deflect away from the pressure force. In addition, the disk hub is allowed to stretch. The total displacement at the outer edge of the valve disk due to shear and bending and due to hub stretch are calculated using the reference 1 equations.



An evenly distributed force is assumed to act between the valve seat and the outer edge of the valve disk. This force acts to deflect the outer diameter of the valve disk inward and to compress the disk hub. The pressure force is reacted to by an increase in this contact force between the valve disk and seats. The valve body seats are conservatively assumed to be fixed. Therefore, the deflection due to the known pressure load must be balanced by the deflection due to the unknown seat load. The deflection due to the pressure force is first calculated. Then, the reference 1 equations are used to determine the contact force between the seat and disk which results in a deflection which is equal and opposite to the deflection due to the pressure force.

ATTACHMENT A
CALC. NO. XX-M-040
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The coefficient of friction between the seat and disk is determined based on the open valve factor from a DP test. The stem force required to overcome the contact load between the seat and disk which opposes the pressure force is equal to:

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

Static Unseating Force

The static unseating force represent the open packing load and pullout force due to wedging of the valve disk during closure. These loads are superimposed on the loads due to the pressure forces which occur during pressure locking. The value for this load is based on static test data for the MOVs.

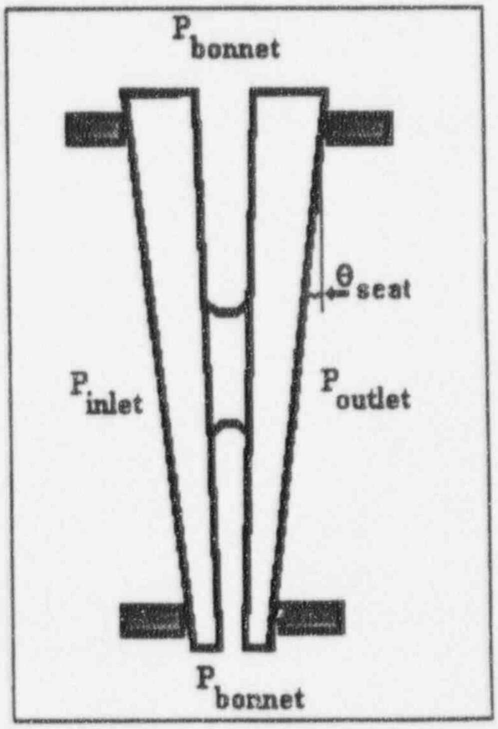
Piston Effect

The piston effect due to valve internal pressure exceeding outside pressure is calculated using the standard industry equation. This force assists movement of the valve stem in the open direction.

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

"Reverse Piston Effect"(F_{vert})

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



ATTACHMENT A
CALC. NO. XX-M-040
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Total Force Required to Overcome Pressure Locking

As mentioned previously, the total stem force (tension) required to overcome pressure locking is the sum of the four components discussed above. All of the terms are positive with the exception of the piston effect component.

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

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

ATTACHMENT A
CALC. NO. XX-M-040
REV 00 SHT 20 OF 33

Determination of Open Valve Factor

Review of the VOTES traces for the subject MOV indicate a ~4000 lbf inconsistency between the thrust zero at O4 (valve closed) and the thrust zero at C3 (valve opened). This is explained by the valve's horizontal orientation. The weight of the valve stem when the valve is open is putting a side load on the valve yoke causing the zero mismatch.

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

$$\boxed{VF_{open} = \frac{C10_{thrust} - Running_{thrust} + \frac{\pi}{4} D^2_{stem} (C10_{line\ pressure} - Running_{line\ pressure})}{DP \times \frac{\pi}{4} D^2_{seat}}}$$

The apparent margin between MGC and required thrust is calculated. No acceptance criteria is provided in this calculation. The Station will establish the acceptability of the available margin as part of the operability analysis for this valve.

III. ASSUMPTIONS

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

ATTACHMENT A
CALC. NO. XX-M-04D
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2. The coefficient of friction between the valve seat and disk is assumed to be the same under pressure locking conditions as it is under DP conditions. This assumption (in combination with assumption 2) is considered to be justified based on bench marking of the calculation against ComEd and EPRI pressure locking test data for similar flex-wedge gate valves.

3. The upstream, downstream, and bonnet pressure values are based on a scenario in which the valve bonnet is pressurized to reactor pressure by leakage past adjacent check valves. A LOCA occurs which causes the reactor pressure drop off a rate defined in the applicable fuel analysis. The LPCI and LPCS pumps come up to speed, and the subject valves receive a signal to open. The pressure values are based on a review of the fuel analysis and FSAR for each station. These values should be reviewed by the affected stations for accuracy prior to final acceptance of this calculation. (See LIMITATIONS at end of this calculation.)

IV. DESIGN INPUTS

1. Valve Disk Geometry information is based on the Reference 4 Fax from Crane Valve Company (Attachment 1)

2. Motor Data is taken from the Reference 5 report.

V. REFERENCES

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

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

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

ATTACHMENT A
CALC. NO. XX-M-040
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4. Crane Telecopies from Dave Dwyer and Bruce Harry to Brian Bunte (ComEd) dated 5/3/95 and 6/16/95, Attachment 1

5. ComEd MOV AC Motor Test Program Report Part 1, Revision 1, CHRON 213741

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

7. ComEd AC Induction Motor Test Report, Part 1, Chron 213741 dated April 27, 1995

8. Thrust values are taken from the following static and DP VOTES tests:

	Station	Valve	Test Number(s)	Test Date
A.	Dresden	3-1501-22A	40	7/6/94
B.	Dresden	3-1501-22A	42	7/6/94

9. Margin Review Databases for Dresden Station (dated 11/11/94)

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

VI. CALCULATIONS

MathCad calculation of:

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

is provided on the next 4 pages.

ATTACHMENT A
CALC. NO. XX-M-040
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EXAMPLE

INPUTS:

Bonnet Pressure	$P_{bonnet} = 1005 \cdot psi$	**Assumption 5**
Upstream Pressure	$P_{up} = 380 \cdot psi$	**Assumption 5**
Downstream Pressure	$P_{down} = 350 \cdot psi$	**Assumption 5**
Disk Thickness	$t = 3 \cdot in$	**Attachment 1**
Seat Radius	$a = 6.385 \cdot in$	**Reference 9**
Hub Radius	$b = 2.125 \cdot in$	**Attachment 1**
Hub Length	$L = 2.4375 \cdot in$	**Attachment 1**
Seat Angle	$theta = 5 \cdot deg$	**Reference 9**
Poisson's Ratio (disk)	$v = .3$	**Typical of Carbon Steel**
Mod. of Elast. (disk)	$E = 27.6 \cdot 10^6 \cdot psi$	**Typical of Carbon Steel**
Static Pullout Force (Test 42)	$F_{po} = 36921 \cdot lbf$	**Reference 8B**
O10 Thrust (DP test)	$O10 = 10048 \cdot lbf$	**Reference 8A**
Open Run Thrust (DP)	$Run = 2268 \cdot lbf$	**Reference 8A**
DP	$DPtest = 278.5 \cdot psi$	**Reference 8A**
LP (valve closed)	$LPclose = 302.6 \cdot psi$	**Reference 8A**
LP (valve open)	$LPopen = 45.6 \cdot psi$	**Reference 8A**
Stem Diameter	$D_{stem} = 3 \cdot in$	**Reference 9**

VALVE FACTOR CALCULATION

Valve Factor:

$$VF = \frac{(O10 - Run) + \frac{\pi}{4} \cdot D_{stem}{}^2 \cdot (LPclose - LPopen)}{\pi \cdot (a)^2 \cdot DPtest} \qquad VF = 0.269$$

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

$$mu = VF \cdot \frac{\cos(theta)}{1 - VF \cdot \sin(theta)} \qquad mu = 0.274$$

PRESSURE FORCE CALCULATIONS

Average DP across disks:

$$DPavg = P_{bonnet} - \frac{P_{up} + P_{down}}{2} \qquad DPavg = 640 \cdot psi$$

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Disk Stiffness Constants(Reference 1, Table 24)

$$D := \frac{E \cdot (t)^3}{12 \cdot (1 - \nu^2)} \qquad D = 6.824 \cdot 10^7 \cdot \text{lbf} \cdot \text{in}$$

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

Geometry Factors: (Reference 1, Table 24)

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

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

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

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

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

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

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

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

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

$$M_{rb} := \frac{-DPavg \cdot a^2}{C_8} \cdot \left[\frac{C_9}{2 \cdot a \cdot b} \cdot (a^2 - b^2) - L_{17} \right] \qquad M_{rb} = -9.381 \cdot 10^3 \cdot \text{lbf}$$

$$Q_b := \frac{DPavg}{2 \cdot b} \cdot (a^2 - b^2) \qquad Q_b = 5.459 \cdot 10^3 \cdot \frac{\text{lbf}}{\text{in}}$$

ATTACHMENT A
CALC. NO. XX-M-040
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Deflection due to pressure and bending(Reference 1, Table 24, Case 2L)

$y_{bq} = M_{rb} \cdot \frac{a^2}{D} \cdot C_2 + Q_b \cdot \frac{a^3}{D} \cdot C_3 - \frac{DPavg \cdot a^4}{D} \cdot L_{11}$ $\quad y_{bq} = -4.138 \cdot 10^{-4} \cdot in$

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

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

$y_{sq} = \frac{K_{sa} \cdot DPavg \cdot a^2}{t \cdot G}$ $\quad y_{sq} = -3.223 \cdot 10^{-4} \cdot in$

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

$P_{force} = 3.1416 \cdot (a^2 - b^2) \cdot DPavg$ $\quad P_{force} = 7.289 \cdot 10^4 \cdot lbf$

$y_{stretch} = \frac{P_{force}}{3.1416 \cdot b^2} \cdot \frac{L}{(2 \cdot E)}$ $\quad y_{stretch} = 2.269 \cdot 10^{-4} \cdot in$

Total Deflection due to pressure forces:

$y_q = y_{bq} + y_{sq} - y_{stretch}$ $\quad y_q = -9.629 \cdot 10^{-4} \cdot in$

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

$y_{sw} := \frac{-1.2 \cdot \left(\frac{a}{a}\right) \cdot \ln\left(\frac{a}{b}\right) \cdot a}{t \cdot G}$ $\quad y_{sw} = -2.647 \cdot 10^{-7} \cdot \frac{in}{\left(\frac{lbf}{in}\right)}$

(per lbf/in)

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

$y_{bw} = -\left(\frac{a^3}{D}\right) \cdot \left[\left(\frac{C_2}{C_8}\right) \cdot \left[\left(\frac{a \cdot C_9}{b}\right) - L_9 \right] - \left[\left(\frac{a}{b}\right) \cdot C_3 \right] + L_3 \right]$ $\quad y_{bw} = -4.608 \cdot 10^{-7} \cdot \frac{in}{\left(\frac{lbf}{in}\right)}$

(per lbf/in)

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

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

(per lbf/in)

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

$y_w = y_{bw} + y_{sw} - y_{compr}$ $\quad y_w = -8.504 \cdot 10^{-7} \cdot \frac{in}{\left(\frac{lbf}{in}\right)}$

(per lbf/in)

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in

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

$F_s = 2 \cdot \pi \cdot a \cdot \dfrac{y_q}{y_w}$ $\qquad$ $F_s = 4.543 \cdot 10^4 \cdot lbf$

UNSEATING FORCES

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

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

$F_{vert} = \pi \cdot a^2 \cdot \sin(theta) \cdot (2 \cdot P_{bonnet} - P_{up} - P_{down})$ $\qquad$ $F_{vert} = 1.429 \cdot 10^4 \cdot lbf$

$F_{preslock} = 2 \cdot F_s \cdot (mu \cdot \cos(theta) - \sin(theta))$ $\qquad$ $F_{preslock} = 1.692 \cdot 10^4 \cdot lbf$

$F_{total} = -F_{piston} + F_{vert} + F_{preslock} + F_{po}$ $\qquad$ $F_{po} = 3.692 \cdot 10^4 \cdot lbf$

$F_{total} = 6.103 \cdot 10^4 \cdot lbf$

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	$MR = 88.2 \cdot ft \cdot lbf$	**Reference 9**
Temperature Factor:	$Tf = 0.98$	**Reference 9**
Degraded Voltage:	$DV \quad 402 \cdot volt$	**Reference 9**
Under Voltage Factor:	$n \quad 2.187$	**Reference 5 (Typical)**
Stem Factor:	$SF = .0275 \cdot ft$	**Reference 9**
Overall Ratio:	$OAR = 48.45$	**Reference 9**
Pullout Efficiency:	$Eff_{po} = 0.65$	**Reference 9**

CALCULATIONS:

$MGC_{avail.} = MR \cdot Tf \cdot OAR \cdot Eff_{po} \cdot \dfrac{\left(\dfrac{DV}{460 \cdot volt}\right)^n}{SF}$ $\qquad$ $MGC_{avail.} = 7.372 \cdot 10^4 \cdot lbf$

$Margin = \dfrac{MGC_{avail.} - F_{total}}{F_{total}}$ $\qquad$ $Margin = 0.208$

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Westinghouse
Electric Corporation

8021 Flint
Lenexa, Kansas 66214

SAP96-100

February 5, 1996

96-00220

Mr. Lanny Ratzlaff
Wolf Creek Nuclear Operating Corp
P. O. Box 411
Burlington, KS 66839

Subject: ***Geometric Parameters for Valve Pressure Locking Analysis***

Dear Mr. Ratzlaff:

In response to your recent request, this letter transmits geometric parameters for several Westinghouse valves. These geometric parameters are for use in the pressure locking analysis of these valves, using the Commonwealth Edison methodology, being done by Wolf Creek to support their response.

The geometric parameters which we are providing are as follows:

Seat Angle: The value provided is one half of the angle between the two (2) faces of the disc, from the gate disc drawing. For all of these valves, the seat angle is 7 degrees.

Stem Diameter: This is the stem diameter where the stem passes through the packing.

Hub Radius: When the hub is circular, the hub radius is the hub radius from the gate disc drawing. When the hub is non-circular, the hub radius is the radius of a circle which has the same area as the actual hub.

Hub Length: The hub length is the distance between the two (2) disc flanges at the bottom of the wedge. It is recognized that this may give a larger hub length than is actually present in the hub itself, but this is the dimension used in the Commonwealth Edison analysis of the Westinghouse 4" gate valve which was used to verify their technique against test results.

Disc Thickness: The value reported is equal to the wedge thickness at mid-height of the wedge as given on the gate disc drawing, minus the hub length defined above, quantity divided by 2. This disc thickness is actually the thickness of the disk at its outside edge at mid-height of the wedge, and as with the hub length dimension, this is the value used in the Commonwealth Edison calculation for the Westinghouse 4" gate valve which was used to validate the model.

Mr. Lanny Ratzlaff
Wolf Creek Nuclear Operating Corp

February 5, 1996
Page Two (2)

Seat Radius: The values given are the mean seat radius as given by the following formula:

seat radius = square root of [(OD squared plus ID squared)/8]

The outside seat diameter used in this formula is the outside diameter of the seat area given on the seat ring drawing. The inside seat diameter is taken as the average of the inside seat diameter from the seat ring drawing and the inside seat diameter from the seat ring drawing divided by the cosine of 7 degrees. The outside seat diameter is given in the plane of the seat, while the inside seat diameter is given in the vertical direction. The inside of the seat ring is actually thus an ellipse, and we are using the average diameter of the ellipse.

The table of values is attached.

Very truly yours,

WESTINGHOUSE ELECTRIC CORPORATION

M. C. Bollenbach

M. C. Bollenbach
Special Sales Representative
North American Field Sales

MCB:sls
ATTACHMENT

cc: Mr. Carl Dumsday (W) WCNOC Site

TABLE OF VALUES

VALVE ID	VALVE TAG No.	GPO ASSY SERIAL NUMBER	DRAWING
4GM78FNA	8802A	04000GM88FNB0D000W750008	9749D75
4GM78FNA	8802B	04000GM88FNB0D000W750011	9749D75
10GM78FNC	8840	10000GM88FNB0D005W750006	9747D09
12GM88SEF	8702A	12002GM88SEH0F000W750002	3D21943
12GM88SEB	8702B	12002GM88SEH0D000W750004	3D21499
14GM84FED	8811A	14000GM84FEH0E005W750001	1D99954
14GM84FED	8811B	14000GM84FEH0E005W750002	1D99954

DIAMETER TAG No.	DISK THICKNESS t, inches	SEAT RADIUS inches	HUB RADIUS inches	HUB LENGTH inches	SEAT ANGLE degrees	STEM inches
8802A	1.01	2.006	1.056	0.61	7	1.25
8802B	1.01	2.006	1.056	0.61	7	1.25
8840	2.744	5.011	2.694	0.82	7	2.50
8702A	3.225	5.987	3.225	1.31	7	3.00
8702B	3.225	5.987	3.225	1.31	7	3.00
8811A	2.560	6.332	2.56	0.89	7	2.00
8811B	2.560	6.332	2.56	0.89	7	2.00

Attachment C to Calc. XX-M-040
Page 31 of 32

ANCHOR/DARLING VALVE COMPANY

TELECOPIER NUMBER 316-364-4095 DATE 2/8/96

TO: LANNY D. RATZLAFF

NUMBER OF PAGES 1 PLUS INSTRUCTION SHEET = 2

MESSAGE FROM F. BENSINGER

ANCHOR/DARLING VALVE COMPANY
WILLIAMSPORT, PENNSYLVANIA

VALVE #	DISK THICKNESS	SEAT DIAMETER	HUB DIAMETER	HUB LENGTH
ENHV0001	1 1/2 AT HUB DIA.			4 3/8 AT 12 O'CLOCK
ENHV0007	1 1/4 AT SEAT DIA.	11.125	3.625	3 3/4 AT 6 O'CLOCK

All dimensions are in inches.

2/8/96

(ASSEMBLY DRAWING # 93-14050)

Independent verification of the inputs and outputs for EMHV8802B found on sheet 5 of this calc.

Inputs:

Bonnet Pressure	Pbonnet =	4,269	psi
Upstream Pressure	Pup =	0	psi
Downstream Pressure	Pdown =	0	psi
Disk Thickness	t =	1.010	in
Seat Radius	a =	2.006	in
Effective Hub Radius	b =	1.056	in
Hub Length	L =	0.610	in
Seat Angle	theta =	7	deg
Stem Diameter	Dstem =	1.25	in
Poisson's Ratio	v =	0.3	
Modulus of Elast.	E =	27,600,000	psi
Static Pullout Force	Fpo =	3,993	lbs
DP	DP =	0	psi
Line Pressure Closed	LPc =	0	psi
Line Pressure Open	LPo =	0	psi
O10 DP Thrust	O10 =	0	lbs
Open DR Run Load	Fpk =	0	lbs
Motor Capability	OT =	13,606	lbs
Max Allow Open Thrust	MASTo =	16,000	lbs

STEVE LANE PERFORMED THIS ~~INDEPENT~~ INDEPENDENT VERIFICATION TO CONFIRM THE WORK SHEET USED IN THE BODY OF THIS CALCULATION.
JTH 2/14/96

Calculated Values:

D =	2604057 lbs-in	L11 =	0.002	Pforce =	39013 lbs
G =	10615385	L17 =	0.083	ystretch =	1.231E-04 in
C2 =	0.092	DPavg =	4269 psi	yq =	-0.000576 in
C3 =	0.013	Mrb =	-2620 lbs	ysw =	-1.44E-07 in^2/lb
C8 =	0.747	Qb =	5880 lbs/in	ybw =	-1.32E-07 in^2/lb
C9 =	0.286	ybd =	-0.000184 in	ycompr =	3.976E-08 in^2/lb
L3 =	0	Ksa =	-0.168	yw =	-3.16E-07 in^2/lb
L9 =	0	ysq =	-0.000269	Fs =	22969 lbs
		VF =	0.000	Mu =	0.16

Pressure Locking Loads:

Fpiston =	5239 lbs	Fpreslock =	1697 lbs
Fvert =	13154 lbs	Fpo =	3993 lbs
Ftotal =	13605 lbs		

Results:

Motor Cap =	13606 lbs	Ftotal =	13605 lbs
MAST =	16000 lbs		

Margin = 7E-05 %

The results of this verification agrees with those on sheet 5. Therefore, the results on sheet 5 are correct.

ATTACHMENT 'E' TO
CALCULATION XX-M-040 REV. 1
Pg. 1 of 2

EC-577 East, Mail Stop 5-32

WESTINGHOUSE ELECTRIC CORPORATION
SYSTEMS & MAJOR PROJECTS DIVISION / MSE

COVER SHEET FOR TELECOPY NO. (412) 374-6639
WIN: 284-6639

Attention:	LANNY RATZLAFF
Location:	
Telecopy No.:	316-364-4095
Confirmation No.:	

Notes:

attached is a response on the 10GM78FNC, Dwg 9747D09 ~~&~~ for opening loads of 120,000#

From:	TJ Matty	Phone:	6451
Date:	4/5/91	Telecopy No.: (412) 374-6639 or WIN 284-6639 or WIN 284-6647	

NUMBER OF PAGES (including cover sheet) 2

ATTACHMENT 'E' To
CALCULATION XX-M-040 Rev. 1
Pg. 2 of 2

VALVE - 10GM78FNC
DWG - 9747D09

QUESTION: WHAT IS THE BASIS OF THE MAXIMUM OPERATING LOAD OF 112084 LB FOR THE VALVE AND CAN THE VALVE WITHSTAND AN OPENING LOAD OF 120,000 LB IF REQUIRED.

BASIS FOR CONCLUSION:

THE ALLOWABLE OPENING LOAD OF 112084 LB WAS BASED ON THE VALVE LINK AT 650F. THIS VALUE WAS BASED ON AN ALLOWABLE OF .65 Sy WHICH WESTINGHOUSE GENERALLY USES FOR AS-LEFT CONDITIONS. AT LOWER TEMPERATURES THE LOADS WOULD BE:

600F - 113676 LB
500F - 116616 LB
400F - 120414 LB

ALL THESE VALUES ARE BASED ON THE SAME ALLOWABLE AS THE 112084 LB.

PLEASE NOTE THAT THE OPERATING LOAD APPLIED TO THE STEM IS NOT THE LOAD ON THE INTERNAL COMPONENTS SINCE SOME OF THE LOAD IS USED TO PULL THE STEM THROUGH THE PACKING.

ALL THE OTHER COMPONENTS AS LINK, PIN, STEM AND DISC CAN TAKE OPENING LOADS GREATER THAN 150,000 LB.

CONCLUSION:

THE VALVE WOULD BE ACCEPTABLE WITH A 120,000 LB OPENING LOAD BASED ON THE ALLOWABLE STRESS (.65 Sy), THE FACT THE TEMPERATURE WOULD NOT BE AT 650F AND THE FACT THAT THE PACKING LOADS WOULD REDUCE THE LOADS ON THE LINK.

JJM 4/5/90