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Due to Higher Torque Switch Settings
During Operability Testing**

**W. D. Richins
S. D. Snow
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USNRC Technical Monitor: G. H. Weidenhamer

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CONTENTS

1. Abstract	1
2. Introduction	2
3. Industry Experience	4
4. Stem Loads	6
5. Stress Analysis Methods	13
5.1. Material Properties	13
5.2. Stem and Stem Nut Threads	14
5.3. Finite Element Modeling – General	15
5.4. Yokes	16
5.5. Disks, Disk Faces, Stem T–Heads, and Valve Seats	21
5.6. Finite Element Solution Enhancement Routine	29
6. Stress Results	30
6.1. Finite Element Analysis Results	30
6.2. Stress Intensification in the Walworth Yoke	30
6.3. Maximum Stress Range Summary	50
7. Estimates of Increase in Fatigue Usage Factors	55
8. Summary	60
9. Conclusions	62
10. Acknowledgments	63
11. References	64
 APPENDIX A -- Computer Verification and Sample Data	 A–1
APPENDIX B -- Stem and Stem Nut Thread Shear Stress Calculations	B–1

FIGURES

Figure 1 . Key Components of Motor Operated Gate Valves	3
Figure 2 . Details of Typical Gate Valve Components	3
Figure 3 . Typical Stem Thrust Trace for Valve Closing to Opening Cycle	9
Figure 4 . Finite Element Model used for ABAQUS Analysis of Walworth Yoke, Upper View	17
Figure 5 . Finite Element Model used for ABAQUS Analysis of Walworth Yoke, Lower View	18
Figure 6 . Finite Element Model used for ABAQUS Analysis of Anchor/Darling Yoke, Upper View	19
Figure 7 . Finite Element Model used for ABAQUS Analysis of Anchor/Darling Yoke, Lower View	20
Figure 8 . Walworth Stem T-Head Finite Element Model	21
Figure 9 . Walworth Stem T-Head Mating Configurations to Disk ...	22
Figure 10 . Anchor/Darling Stem T-Head Finite Element Model	23
Figure 11 . Walworth Disk Finite Element Model	25
Figure 12 . Anchor/Darling Disk Finite Element Model	27
Figure 13 . Maximum Stress Range for Walworth Yoke, Load Case C	31
Figure 14 . Maximum Stress Range for Anchor/Darling Yoke, Load Case C	33
Figure 15 . Maximum Stress Range for Walworth Stem T-Head, Normal Configuration, Load Case C	35
Figure 16 . Maximum Stress Range for Walworth Stem T-Head, Abnormal Configuration, Load Case C	37
Figure 17 . Maximum Stress Range for Anchor/Darling Stem T-Head, Load Case C	39
Figure 18 . Maximum Stress Range for Walworth Disk, Normal Configuration, Load Case C	41
Figure 19 . Maximum Stress Range for Walworth Disk, Abnormal Configuration, Load Case C	43
Figure 20 . Maximum Stress Range for Anchor/Darling Disk, Load Case C	45
Figure 21 . Details of Maximum Stress Range Concentrations for Walworth Yoke, Load Case B	47
Figure 22 . Maximum Tresca Stress Range Intensification for Walworth Yoke, Load Case B	49

TABLES

Table 1 .	Walworth Valve Closing Stem Loads Predicted using IVA Software	7
Table 2 .	Anchor/Darling Valve Closing Stem Loads Predicted using IVA Software	7
Table 3 .	Comparison of Valve Test Conditions versus Valve Analyses Conditions	8
Table 4 .	Walworth Valve Stem Loads from Testing	10
Table 5 .	Anchor/Darling Valve Stem Loads from Testing	10
Table 6 .	Load Cases – Walworth Valve	12
Table 7 .	Load Cases – Anchor/Darling Valve	12
Table 8 .	Material Properties for Valve Components	13
Table 9 .	Mesh Refinement Comparison for Anchor/Darling, Stem T–Head Model	24
Table 10 .	Maximum Tresca Stress Range for Walworth Valve: Yoke, Stem Threads and Stem Nut Threads	50
Table 11 .	Maximum Tresca Stress Range for Anchor/Darling Valve: Yoke, Stem Threads and Stem Nut Threads	51
Table 12 .	Maximum Tresca Stress Range for Walworth Valve: Stem T–Head, Disk Body, Disk Faces, and Valve Seats	52
Table 13 .	Maximum Tresca Stress Range for Anchor/Darling Valve: Stem T–Head, Disk Body, Disk Faces, and Valve Seats	54
Table 14 .	Estimates of Cumulative Usage Factors for Walworth Valve	58
Table 15 .	Estimates of Cumulative Usage Factors for Anchor/Darling Valve	59

1. ABSTRACT

Some motor operated valves now have higher torque switch settings due to regulatory requirements to ensure valve operability with appropriate margins at design basis conditions. Verifying operability with these settings imposes higher stem loads during periodic inservice testing. These higher test loads increase stresses in the various valve internal parts which may in turn increase the fatigue usage factors. This increased fatigue is judged to be a concern primarily in the valve disks, seats, yokes, stems, and stem nuts. Although the motor operators may also have significantly increased loading, they are being evaluated by the manufacturers and are beyond the scope of this study. Two gate valves representative of both relatively weak and strong valves commonly used in commercial nuclear applications were selected for fatigue analyses. Detailed dimensional and test data were available for both valves from previous studies at the Idaho National Engineering Laboratory. Finite element models were developed to estimate maximum stresses in the internal parts of the valves and to identify the critical areas within the valves where fatigue may be a concern. Loads were estimated using industry standard equations for calculating torque switch settings prior and subsequent to the testing requirements of USNRC Generic Letter 89-10. Test data were used to determine both; 1) the overshoot load between torque switch trip and final seating of the disk during valve closing and 2) the stem thrust required to open the valves. The ranges of peak stresses thus determined were then used to estimate the increase in the fatigue usage factors due to the higher stem thrust loads. The usages that would be accumulated by 100 base cycles plus one or eight test cycles per year over 40 and 60 years of operation were calculated. The fatigue effects for the disk faces and valve seats were not calculated since fatigue curve data were unavailable, however, both are loaded in compression and are not likely to develop fatigue cracks. The disk, yoke, stem T-head, stem threads, and stem nut threads for both valves were shown to have acceptable fatigue life for all load cases.

2. INTRODUCTION

During periodic inservice testing for operability, some motor operated valves (MOVs) within commercial nuclear power plants now have higher torque switch levels due to the requirements of USNRC Generic Letter 89-10 (Ref. 1). USNRC 89-10 requires that all safety related MOVs have a testing program that provides, "...the necessary assurance that they will function when subjected to the design-basis conditions that are to be considered during both normal operation and abnormal events within the design basis of the plant." Inservice testing is done at regular intervals, normally once per quarter. These higher settings increase both the thrust and torque applied to the valve stem and increase stresses in the valve body and various internal parts which may in turn increase the fatigue usage factors.

This study investigates those regions within motor operated gate valves where increased fatigue is judged to be a concern, specifically the disks, seats, yokes, and stems. Details of gate valve components are shown in Figures 1 and 2. Industry experience and known valve fatigue failures relating to this issue are summarized in Section 3 of this report. Although the motor operators may also have significantly increased loading, they are being evaluated by the manufacturers and are beyond the scope of this study.

Two gate valves representative of both relatively weak and strong valves commonly used in commercial nuclear applications were selected for fatigue analyses. Detailed dimensional and test data were available for both valves from previous studies at the Idaho National Engineering Laboratory (Ref. 2). Additional measurements were made directly from the valves to facilitate modeling the necessary details. The general philosophy was to use dimensions and tolerances characteristic of the type of valves being analyzed. Since many of the valve parts are initially cast, welded, and/or ground to produce the final shape, using dimensions taken to very close tolerances is unwarranted. The two valves are:

1. 6" 600 lb Walworth Pressure Seal Gate Valve, with a flexible wedge disk, ASME Sec. III, Class 2, built in 1979.
2. 10" 900 lb Anchor/Darling Pressure Seal Gate Valve, with a flexible wedge disk, ASME Sec. III, Class 1, built in 1982.

The Walworth valve represents a relatively weak valve both in terms of the 600 lb class and the dimensions of the valve internals. In contrast, the Anchor/Darling valve is representative of relatively strong valves of this type. Considerable test data for both valves are available. The test results with conditions most similar to those assumed for inservice testing are summarized in Section 4.

The results of this analysis; 1) identify the critical areas within the valves where fatigue may be a concern, and 2) provide estimates of the increase in fatigue usage factors in these areas caused by the increased loads applied to the stem during inservice testing prior and subsequent to the issuance USNRC 89-10.

Figure 1 . Key Components of Motor Operated Gate Valves

Figure 2 . Details of Typical Gate Valve Components

3. INDUSTRY EXPERIENCE

Two sources of information were used to find out what could be learned from industry experience about fatigue damage to motor operated gate valves. The first source was the Nuclear Plant Reliability Data System (NPRDS). The NPRDS data base was queried to provide Licensing Event Report (LER) narratives for reports that occurred in the last ten years, involved motor operated gate valves, and involved fatigue. 103 records were obtained. The predominant failure mechanism (50%) was packing failure due to aging (fatigue) and wear. Second was seat scoring (16%). Third was gasket failure (10%). Increased thrust would not be likely to cause increased packing wear, but could lead to increases in seat scoring and gasket failure. Note that the failures in these two categories were not catastrophic, but led to unacceptable leakage requiring repair. The remainder of the records indicated causes distributed among categories not likely to be affected by increased thrust loads. These included design flaws (9%), thermal binding (4%), and miscellaneous (11%).

The second source of information on industry experience came from a phone survey of nine members of the MOV Users Group (MUG) from a variety of valve manufacturers/vendors. Each was asked three questions: 1. "Do you know of any fatigue failures that have occurred in the field in motor operated gate valves?"; 2. "Do you know of any fatigue studies done on motor operated gate valves?"; and 3. "Do you have an opinion as to which valve component is most likely to exhibit fatigue failure due to increased thrust settings?" Seven of the nine gave a useful response. Three types of failures were mentioned. The first involved a stem T-head failure resulting from an inappropriate selection of stem material for the application temperature. The second involved cracked Walworth valve yokes identified in Information Notice 93-97 (Ref. 3) as discussed below. The third involved a low cycle fatigue failure in the back seat of a valve set up to trip on torque when back seating (a practice that is not recommended). No fatigue studies were identified. No consensus was found among the opinions offered in response to question 3. Components identified included stem threads, the wedge and seat rings.

Information Notice 93-97 (Ref. 3) identifies fatigue cracks in the cast carbon steel yokes of 9 out of 62 Walworth valves of pressure seal design with semicircular yoke legs at Peach Bottom Atomic Power Station. The valve sizes ranged from 4 to 24 inches and all were safety related components. The Information Notice also stated that, "...the use of high thrust settings can induce fatigue-related failures of MOVs." The Walworth valve analyzed in this report matches the characteristics of those addressed in Information Notice 93-97 and the location of the cracks corresponds to the area of high stress ranges identified for this valve in Section 6 of this report. The Information Notice identified insufficient bolt torque and existing casting defects as factors contributing to the development of the cracks. Insufficient bolt torque can result in high local stresses not typical of most valve installations. Both of these factors are considered anomalous and not directly applicable to this study. The fact that all of the cracked yokes occurred at one plant and all of the same manufacture may indicate problems in installation and inspection rather than valve design.

With the exception of the valve yoke cracking discussed in Information Notice 93-97, no specific candidate for fatigue failure was identified by the industry experience queried in this survey. The most likely problems indicated are the disk scoring and gasket failures identified by the data base search. Neither of these are considered a source of catastrophic failure.

4. STEM LOADS

Fatigue evaluations require precise calculations of the maximum range of peak stresses that occur in the various valve parts experienced during the complete opening and closing cycle of operability testing. The primary loads on the valves during this cycle are stem torque and thrust (axial load) applied to the stem by the motor operator. Note that, for these valves, the stem is stationary (does not rotate) while the stem nut turns around the stem thus driving the stem in and out of the valve body. The stem is in axial compression during the valve closure cycle. For consistency throughout this report, both the stem thrust and torque are taken to be negative during valve closure.

Loading conditions for both valves were selected to best approximate those typical for periodic inservice operability testing. The two valves evaluated here are designed for high pressure safety injection systems normally tested under the conditions of no-flow and no pressure drop, with fluid subcooling of less than 70° F (the fluid is nearly steam). Typical operating temperature and pressure values of 600° F and 1000 PSIG were used.

The program IVA (Isolation Valve Assessment Software, Ref. 4), developed at the INEL, was used to estimate the stem loads at the required torque switch trip settings for these conditions. IVA uses numerous input parameters relating to valve dimensions, packing, stem/stem nut data, operator mechanical and electrical configuration, and the motor power supply. These parameters were set to values judged (Ref. 5) to be typical of those used for determining torque switch settings for operability testing under the conditions described above. Dimensional data were either measured directly from the valve or taken from vendor data. The torque switch setting versus applied torque calibration data used were from those published by Limitorque and included within IVA. Design basis stem factors calculated by IVA were used to estimate the stem torques based on the required stem thrusts. The stem factor is a function of stem/stem nut dimensions with a coefficient of friction of 0.15. A listing of the parameters used in IVA and examples of IVA output for each valve are included in Appendix A.

Using these input parameters, IVA calculates the minimum required torque switch setting needed to provide enough stem thrust to close the valve based on standard industry equations. Since the torque limit switches can generally only be set to $\pm 1/4$ of a setting as marked on the motor operator, torque switch settings equal to the first 1/4 setting higher than the required minimum values calculated by IVA were used. Three load cases were used to determine the effect of USNRC Generic Letter 89-10 on the calculation of the torque switch settings. Only the disk factor was varied in the IVA calculations using values judged to represent: 1) the nominal torque switch settings used before 89-10 was issued (pre 89-10), with a disk factor of 0.3; 2) the nominal settings used post 89-10, with a disk factor of 0.5; and 3) the maximum settings used post 89-10, with a disk factor of 0.6. These three conditions are referred to as load cases A, B, and C, respectively, in this report. The resulting stem loads calculated by IVA for these three load cases are shown in Tables 1 and 2. The stem/stem nut friction was set at 0.15 for all load cases. A value of 0.2 is sometimes used in the industry when justified but 0.15 is conservative for estimating the maximum stem thrust.

Table 1 .Walworth Valve Closing Stem Loads Predicted using IVA Software

CONDITION	DISK FACTOR	STEM NUT FRICTION	STEM FACTOR, FT	STEM THRUST, LBS	STEM TORQUE, FT-LBS
Pre 89-10 Nominal Torque Setting	0.3	0.15	0.01421	-10,271	-146.0
Post 89-10 Nominal Torque Setting	0.5	0.15	0.01421	-17,431	-247.7
Post 89-10 Maximum Torque Setting	0.6	0.15	0.01421	-21,010	-298.6

Table 2 .Anchor/Darling Valve Closing Stem Loads Predicted using IVA Software

CONDITION	DISK FACTOR	STEM NUT FRICTION	STEM FACTOR, FT	STEM THRUST, LBS	STEM TORQUE, FT-LBS
Pre 89-10 Nominal Torque Setting	0.3	0.15	0.02621	-24,599	-644.8
Post 89-10 Nominal Torque Setting	0.5	0.15	0.02621	-35,435	-928.8
Post 89-10 Maximum Torque Setting	0.6	0.15	0.02621	-42,659	-1118.1

The stem loads calculated by IVA represent the values at torque switch trip during valve closing. Typically, some load overshoot exists in plots of stem load vers time for tests of the closure cycle of MOVs where the stem loads continue to increase while the motor operator and valve internals come to a complete stop at the final seating position. During valve opening, tests generally show a momentary peak in the load trace interpreted to occur while the disk is being released from the seats. Estimates of these additional loads on the valve stem are necessary to accurately evaluate the effects of periodic operability testing on the fatigue life.

Both valves were extensively tested previously at the INEL (Ref. 2) using a variety of conditions including conditions nearly identical to those used in the IVA calculations. Comparison of the actual test conditions with those used for the analyses completed for this report are shown in Table 3. A typical thrust trace recorded during these tests for a single valve operation (closing and opening) is shown in Figure 3. This trace is for closure with no flow, but with internal pressure, a typical in-plant test condition. The closing transient starts at about 4 seconds with a sharp increase in thrust caused by friction in the valve stem packing (packing loads). Since the stem is being compressed, the thrust change is negative. The flat portion of the thrust trace between 4 and 24 seconds represents the valve closing against combined packing and stem rejection loads. The sharp downward trend at about 26 seconds is the seating transient, where thrust loads increase rapidly as the valve disk is wedged. This continues through the torque switch trip point because of a short contactor drop out time prior to interruption of current to the operator motor, and an additional time during which the kinetic energy of motor rotation is dissipated. An elastic relaxation occurs immediately after the maximum closing thrust, and the closing transient is finished at around 28 seconds. The opening transient then begins at about 30 seconds with a sharp decrease in thrust, positive because the stem compression of closing is being relieved. This is followed by a flatter and more erratic period during which the contact surfaces in the stem thread reverse, followed by reversal of contact surfaces between the disc and the stem T-head. Once these clearances are reversed, a sharp thrust transient follows which comes from the static friction between disc and seats, and a small amount of inertial loading in accelerating the disc. This transient reaches the maximum opening thrust at around 32 seconds. This is followed by a reduced thrust demand to continue unseating the disc, and ends with the valve opening against relatively constant packing loads, aided by stem rejection loads, beginning at about 33 seconds.

Table 3 .Comparison of Valve Test Conditions (Ref. 2) versus Valve Analyses Conditions

CONDITIONS USED FOR:	UPSTREAM PRESSURE, PSI	UPSTREAM TEMPERATURE, DEG F	PRESSURE DROP ACROSS VALVE	FLOW
Walworth Test 18	1030	525	none	none
Anchor/Darling Test 18	1130	560	none	none
Analyses – this Report	1000	600	none	none

Figure 3 . Typical Stem Thrust Trace for Valve Closing - Opening Cycle (from Ref. 2)

A summary of the stem loads measured during the tests for the Walworth and Anchor/Darling valves are shown in Tables 4 and 5 , respectively.

Table 4 .Walworth Valve Stem Loads from Testing

LOAD	STEM THRUST, LBS	STEM TORQUE, FT-LBS
Torque Switch Trip During Valve Closing Cycle	-17,261	-197.9
Final Seating at Valve Closing Cycle	-18,983	-225.1
Maximum During Valve Opening Cycle	6003.	91.0

Table 5 .Anchor/Darling Valve Stem Loads from Testing

LOAD	STEM THRUST, LBS	STEM TORQUE, FT-LBS
Torque Switch Trip During Valve Closing Cycle	-42,773	-920.9
Final Seating at Valve Closing Cycle	-44,368	-930.8
Maximum During Valve Opening Cycle	27,191.	615.5

Using the results from both IVA calculations and valve tests, the following steps were taken to arrive at the maximum opening and closing stem loads during one complete cycle of the valves under operability testing conditions. First, adjustments were made to the IVA stem thrusts calculated for the closing cycle by linearly extrapolating the test results to account for the overshoot loads that occur during final seating of the valve disks. The magnitude of the overshoot was assumed to be linearly related to the thrust at torque switch trip. Thrust overshoot was calculated for each stem thrust value at torque switch trip from IVA and then added to the IVA thrust to arrive at the total maximum stem thrust during the valve closing cycle. In the tests, the ratios of the final seating load during closing to the load at torque switch trip are 1.100 for the Walworth valve and 1.037 for the Anchor/Darling valve. These ratios were used to estimate the final seating loads for all load cases.

Next, the maximum stem thrusts required to open the valves were calculated. Proprietary industry reports indicate that the peak stem thrust during valve opening is a linear function of the maximum closing thrust used to seat the disk (this includes the overshoot during closure described above) provided no pressure drop and temperature

changes take place during the cycle. An attempt was made to estimate the stem thrust required to open the valves using both IVA and proprietary industry equations. These estimates exceeded the test results by up to 100% and were judged to be far too conservative for fatigue evaluations. Instead of using these estimates, the opening loads from the valve tests were linearly extrapolated to account for the maximum stem thrusts used during the closing cycle. In the tests, the ratios of the maximum opening load to the final seating load during closing are 0.316 for the Walworth valve and 0.613 for the Anchor/Darling valve. These ratios were used to estimate the maximum opening loads for all load cases.

Finally, the stem torque values for each valve were calculated by multiplying the stem thrusts by the stem factors (listed in Tables 1 and 2) for the respective valves. The resulting stem loads for load cases A, B, and C used in the stress analyses discussed in Section 5 are shown in Tables 6 and 7 . For the Walworth valve, the loads in Table 6 for load case B are very close to those recorded during valve testing listed in Table 4 . Similarly, load case C in Table 7 for the Anchor/Darling valve compares well with those in Table 5 for the test data.

Table 6 .Load Cases – Walworth Valve

LOAD CASE	CONDITION	VALVE CLOSING		VALVE OPENING	
		Stem Thrust, lbs	Stem Torque, ft-lbs	Stem Thrust, lbs	Stem Torque, ft-lbs
A	Pre 89–10 Nominal Torque Setting	–11,296	–160.5	3572	50.8
B	Post 89–10 Nominal Torque Setting	–19,170	–272.4	6062	86.1
C	Post 89–10 Maximum Torque Setting	–23,106	–328.3	7307	103.8

Table 7 .Load Cases – Anchor/Darling Valve

LOAD CASE	CONDITION	VALVE CLOSING		VALVE OPENING	
		Stem Thrust, lbs	Stem Torque, ft-lbs	Stem Thrust, lbs	Stem Torque, ft-lbs
A	Pre 89–10 Nominal Torque Setting	–25,516	–668.8	15,638	409.9
B	Post 89–10 Nominal Torque Setting	–36,756	–963.4	22,526	590.4
C	Post 89–10 Maximum Torque Setting	–44,250	–1159.8	27,119	710.8

5. STRESS ANALYSIS METHODS

5.1. Material Properties

Material properties for the valve components at 70 °F and 600 °F are listed in Table 8. Material specifications for each component were identified by review of catalogs and drawings, and discussions with the manufacturer. Reference details on material properties are listed following the table. Note that the yoke, the stem external threads, and the stem nut internal threads were evaluated at 70 °F, while the stem T-head, the disks, and the valve seats were evaluated at 600 °F.

Table 8 .Material Properties for Valve Components

Component & Material	Material Properties at 70 °F			Material Properties at 600 °F		
	Modulus E (ksi)	Yield S _y (ksi)	Ultimate S _u (ksi)	Modulus E (ksi)	Yield S _y (ksi)	Ultimate S _u (ksi)
6" Walworth Valve Components						
Yoke A216 WCB (casting)	29,500	36.0	70.0	26,700	26.6	70.0
Stem A182 F6b (forging)	29,200	90.0	110.0	26,100	78.5	101.1
Disk A217 WC6 (casting)	29,700	40.0	70.0	26,900	31.4	70.0
Stellite 6 (disk face)	30,000	130.0	130.0	29,000	124.0	124.0
Valve Seats Stellite 6	30,000	130.0	130.0	29,000	124.0	124.0
10" Anchor/Darling Valve Components						
Yoke A216 WCB (casting)	29,500	36.0	70.0	26,700	26.6	70.0
Stem A276-410T (bar)	29,500	80.0	100.0	26,700	65.0	90.0
Disk A216 WCB (casting)	29,500	36.0	70.0	26,700	26.6	70.0
CoCr-A (disk face)	30,000	130.0	130.0	29,000	124.0	124.0
Valve Seats CoCr-A	30,000	130.0	130.0	29,000	124.0	124.0
Stem Nut on Limitorque Valve Operator – Both Valves						
Stem Nut B584 Grade 86300 (Casting)	15,500	60.0	110.0	NA	NA	NA

Table 8 Material Property References:

- A216 WCB: Table U (Ref. 6 , page 424, line 23) for S_u , Table Y-1 (page 460, line 14) for S_y , and Table TM-1 (page 614, $C \leq 0.30$) for E.
- A276-410T (UNS 41000): Material properties at 70 °F are found in Ref. 7, while properties at 600 °F are estimated by comparison to those of similar materials.
- A182 F6b modified (Silicon of 0.5% max. instead of 1.0 – 2.0%): Ref. 8 (page E-11) for S_u and S_y at 70 °F, Table U (Ref. 6, page 428) & Table Y-1 (page 492) for S_u and S_y for similar steels at 600 °F, and Table TM-1 (page 614, material group F) for E.
- A217 WC6 (martensitic SST): Table U (Ref. 6, page 428, line 22) for S_u , Table Y-1 (page 480, line 13) for S_y , and Table TM-1 (page 614, material group C) for E.
- Stellite 6 and CoCr-A: S_u , S_y , and E at 70 °F from Ref. 9 , S_u and S_y at 600 °F from Ref. 10, E at 600 °F is estimated by comparison to similar materials (e.g., HS-21: 62Co-27Cr-5Mo-3Ni, for which elongation remains at 1% from 70 to 1000 °F, and area reduction goes from about 3% at 70 °F up to 4% at 600 °F, indicating that the elastic modulus only slightly reduces in that temperature range.
- B584 Grade 86300 (high strength manganese bronze): (Ref. 7, Volume 2.01, B584) Table 4 for S_u , and S_y , and Ref. 10 (volume 1, page 1047) for E.

5.2. Stem and Stem Nut Threads

Estimates of the maximum shear stresses in the external threads of the stems and the internal threads of the stem nuts were made using the formulas for shear area from ASME/ANSI B1.1 (Ref. 11) modified for use with ACME and Stub ACME threads (Refs. 12 and 13). The calculations are included in Appendix B. Typically on MOVs multi-start threads are used. The Walworth valve has two sets of threads while the Anchor/Darling valve has three. Due to the dimensional tolerances inherent in threads, internal and external threads are generally in contact for only a portion of the thread length. Since the maximum shear stress is needed for fatigue life evaluations, a conservative estimate of the length of threads actually in contact is appropriate. The length of thread engagement was assumed to be equal to the thread pitch for these calculations. This is equivalent to using only the thread area from each thread (two threads for the Walworth valve and three threads for the Anchor/Darling valve) for one complete turn of the stem. The shear stresses calculated in Appendix B were then increased by a factor of 4 per ASME Section III, NB-3232.2 (Ref. 14) to account for the potential for stress concentrations at the root of the threads.

5.3. Finite Element Modeling – General

The yoke, disks, disk faces, and stem T-heads were modeled using I-DEAS (Ref. 15) to create solid models and finite element meshes. The models were then converted to ABAQUS (Ref. 16) for stress evaluation. Descriptions of each model are presented in Sections 5.4 and 5.5. Computer verification data are included in Appendix A. The development of solid models and finite element meshes of critical parts of both valves was done with the following basic methodology:

1. I-DEAS was used for solid modeling, mesh generation, and applying loads and boundary conditions.
2. ABAQUS was used for finite element analysis and post processing. The capabilities of ABAQUS were needed to treat the interaction within the valve (sliding surfaces, friction, and contact between the various parts). Only elastic material properties were used in order to provide the results required for fatigue life estimates using the methods of the ASME Section III Code (Ref. 14).
3. Dimensions and tolerances were used that are characteristic of the type of valves being analyzed. Since many of the valve parts are initially cast, welded, and/or ground to produce the final shape, using dimensions taken to very close tolerances was unwarranted. Also, only those details judged necessary for analysis of critical parts were modeled.
4. Substructure modeling was used where necessary for reliable solution convergence.
5. Material properties that accurately represent the valve parts were used.
6. Some parts, specifically the disk, are made of two very different materials. The solid models of the disks were partitioned prior to meshing to account for the two material types.
7. The two valves were modeled using techniques that are as similar as possible to allow direct comparison of the results.
8. Loading was due primarily to the thrust and torque applied to the stem. Pressure loading was also applied as appropriate.
9. Accurate estimates of the peak stresses in the various valve components required refining the finite element mesh densities until the peak stresses no longer increased significantly. Due to the geometry and stress distribution of the Walworth yoke, a stress concentration factor was estimated for the location of maximum stress range.

5.4. Yokes

Both yokes were modeled using four-node linear tetrahedron elements (ABAQUS element type C3D4) . A total of 62,652 elements (13,760 nodes) were used to model the Walworth yoke and 76,302 elements (16,250 nodes) were used for the Anchor/Darling yoke. The base of each yoke is attached to the valve body using a ring clamp. This boundary was modeled with translational restraints in all directions along the contact surfaces. The motor operators are attached to the opposite end of the yokes using bolts (the Walworth has 4 and the Anchor/Darling has 8). Four nodes evenly distributed through the yoke flange thickness were used at each bolt location in the models to apply the stem loads. The stem thrust was evenly distributed to the nodes representing the bolts as axial loads (tension for the valve closing cycle). The stem torque was modeled as tangential loads at each node (clockwise about the Y-axis for the valve closing cycle). Details of the models are shown in Figures. 4 – 7 .

Initially, each yoke was modeled using relatively course and even element distributions to identify the locations with highest stress. The models were then remeshed to increase the mesh density in these locations. Next, each model was remeshed with increasing mesh density until the change in the magnitude of the maximum Tresca stresses was less than 5%. This process reduced the possibility that the maximum stress concentrations were underestimated. As discussed in Section 6, the stress distribution in the Walworth yoke required the use of a stress concentration factor. Each yoke model was extensively tested to verify that the loads were correctly applied and that the resulting displacements match those expected with the applied boundary conditions.

Figure 4 . Finite Element Model used for ABAQUS Analysis of Walworth Yoke, Upper View

Figure 5 . Finite Element Model used for ABAQUS Analysis of Walworth Yoke,
Lower View

Figure 6 . Finite Element Model used for ABAQUS Analysis of Anchor/Darling Yoke, Upper View

Figure 7 . Finite Element Model used for ABAQUS Analysis of Anchor/Darling Yoke,
Lower View

5.5. Disks, Disk Faces, Stem T-Heads, and Valve Seats

5.5.1 Walworth Stem T-Head Model

The Walworth stem is made of 1.25" diameter bar with an end section of 1.89" diameter. One round notch is placed in each side to provide a connection between the disk and the T-head. The finite element model of the stem T-head, as shown in Figure 8, is composed of 27,790 linear tetrahedron (ABAQUS element type C3D4) and 536 linear wedge (type C3D6) elements. Loadings were applied as follows: surface

Figure 8 . Walworth Stem T-Head Finite Element Model

pressure of 1,000 psi simulating the valve internal pressure, forces at nodes on the top of the stem to create the opening and closing torque loads, and surface pressure on the top of the stem to simulate the insertion (stem thrust) and removal (stem pullout) loads. Nodal restraints and/or node-to-grounded node gap elements represented the mating disk surfaces. Note that the base of the stem T-head was intended to transmit the stem closing thrust forces to the disk. However, the T-head and disk pair evaluated at the INEL did not meet that design intent since the upper portion of the T-head contacted the disk before the base. This is shown in Figure 9. Therefore, the stem T-head model was evaluated for each defined load case for both the design intended (normal) and abnormal mating configurations.

Normal Closing Configuration

Abnormal Closing Configuration

Figure 9 . Walworth Stem T-Head Mating Configurations to Disk

The Walworth stem T-head mesh of 28,326 elements was verified using the first method described in Section 5.5.2 for the Anchor/Darling stem T-head model. Because the Walworth stem T-head volume is less than half that of the Anchor/Darling, this number of elements gave a much higher element density than the 32,985 element Anchor/Darling model gave. Therefore, accuracy within 0.5% for calculated stresses was expected and comparisons of finer meshes was not considered necessary.

5.5.2 Anchor/Darling Stem T-Head Model

The Anchor/Darling stem is made of 2.0" diameter bar with an end section of 2.25" diameter and a rectangular portion of 2.38" x 2.98". One rectangular notch is placed in each side to provide a connection between the disk and the T-head. The finite element model of the stem T-head, as shown in Figure 10, is composed of 30,168 linear

Isometric View

Figure 10 . Anchor/Darling Stem T-Head Finite Element Model

tetrahedron (type C3D4) and 2,817 linear wedge (type C3D6) elements. Loadings were applied as follows: surface pressure of 1,000 psi simulating the valve internal pressure, forces at nodes on the top of the stem to create the opening and closing torque loads, and surface pressure on the top of the stem to simulate the insertion (stem thrust) and removal (stem pullout) loads. Nodal restraints and/or node-to-grounded node gap elements represented the mating disk surfaces.

The finite element model of the Anchor/Darling stem T-head was verified by two methods. The first method ensured that the basic model response was correct. It entailed applying simple loadings and then comparing the model response to hand calculations. The intent of the second method was to determine the required mesh density (number of elements per unit of volume) to accurately predict model response. Therefore, the model was meshed three times with increasing element densities. (Note that the mesh density was fairly even throughout the volume for each mesh.) A test case loading that included a stem torque and thrust was applied, and the resulting stress levels were compared. Table 9 shows the resulting maximum Tresca stress for each mesh.

Table 9 .Mesh Refinement Comparison for Anchor/Darling Stem T-Head Model

Number of Elements in Model	Maximum Tresca Stress (psi)
26,437	15,504
32,985	16,717
42,576	16,626

The above table shows that the difference in maximum Tresca stress between the two finer-meshed models was only 0.5%. Because the mesh refinement was not limited to a particular portion of the model, but rather was throughout the entire volume, the small change in stresses from one mesh to a finer mesh indicated that the predicted stresses were accurate (i.e., predicted stresses included any peak stress due to geometry or loading). Therefore, the 32,985 element mesh was considered adequate, and was used for the subsequent evaluation of the Anchor/Darling stem T-head.

5.5.3 Walworth Disk Model

The Walworth disk is about 6-1/2" wide and 8-1/4" tall. The outer diameter of the disk faces is about 6.3" and the inner diameter is 4.6". The disk is cast as one solid piece, forming two disk halves connected by a central hub. The disk faces are made by welding the facing material onto the disk and then machining them down to the correct dimensions. The finite element model of the disk, as shown in Figure 11, is composed of 27,156 linear tetrahedron (element C3D4) elements to represent the disk body, and 696 linear wedge (type C3D6) elements to simulate the disk faces. Loadings were applied as follows: surface pressure of 1,000 psi simulating the valve internal pressure (applied to all surfaces except where the disk faces were in contact with the valve seats), forces at nodes on the top of the disk (applied uniformly over the disk areas where the stem T-head contacts) to create the opening and closing stem torque loads, and nodal forces on the disk to simulate the insertion (stem thrust) and removal (stem pullout) loads. Nodal restraints represented the mating valve seats for reacting normal loads (compressive), while nodal forces (based on a coefficient of friction of 0.4) were applied to the faces to represent the friction forces between the disk faces and the

Figure 11 . Walworth Disk Finite Element Model

valve seats. The disk model was evaluated for each defined load case for the design intended (normal) and abnormal stem-to-disk mating configurations.

At this point, a discussion of the method used to apply the friction forces in the models is necessary. The inservice conditions assumed for these valves included constant temperature, constant pressure (upstream, downstream, and within the valve), and no flow through the valve body. The only forces applied to the disk are stem thrust and torque. Neglecting the effects of stem torque, friction forces are generated as the disk faces are moved against the valve seats. The friction force per disk face can be estimated by multiplying a coefficient of friction by the normal (compressive) force on that face. In the Walworth valve, the disk faces are tilted off-vertical (assuming a valve orientation that positions the stem vertically) by 5 degrees. Only a small portion of the disk face normal force would oppose the stem thrust load (the sine of 5 degrees, or 8.7% of the normal force). A significant portion of the disk face normal force, however, creates an opposing friction force (coefficient of friction times the cosine of 5 degrees, or coefficient of friction times 99.6% of the normal force). Note that the friction force is a linear function of the normal force (as well as the stem thrust load). Data for the disk face and valve seat material shows that one sliding on the other exhibit a coefficient of friction of 0.4 at 650 degrees F (static coefficient being equal to the dynamic coefficient of friction). Stem torque loads are reacted by forces on opposite edges of the upstream and downstream disk faces, and act to increase the resulting friction forces. The friction forces due to stem torque loads were considered small and were conservatively neglected in both disk models. The friction forces, simulated as forces at nodes, were applied to the disk faces in the models at the same ramp rate (a linearly increasing function) as the stem thrust loads.

Note that if the valve inservice conditions had specified a net pressure loading (e.g., a valve internal pressure greater than either the upstream or downstream pressures), then the friction forces would no longer be linear functions of the disk normal reaction forces. In this case, the friction forces would only need to be large enough to react the stem thrust loads (and the vertical component of all other pressure loads) while the disk face normal loads could increase with the pressure loads.

5.5.4 Anchor/Darling Disk Model

The Anchor/Darling disk is about 11-3/4" wide and 11-3/4" tall. The outer diameter of the disk faces is about 9.6" and the inner diameter is 7.3". The disk is cast as one solid piece, forming two disk halves connected by a central hub. The disk faces are made by welding the facing material onto the disk and then machining them down to the correct dimensions. The finite element model of the disk, as shown in Figure 12, is composed of 48,165 linear tetrahedron (element C3D4) elements to represent the disk body, and 684 linear wedge (type C3D6) elements to simulate the disk faces. Loadings were applied as follows: surface pressure of 1,000 psi simulating the valve internal pressure (applied to all surfaces except where the disk faces were in contact with the valve seats), forces at nodes on the top of the disk to create the opening and closing stem torque loads, and nodal forces on the disk to simulate the insertion (stem thrust) and removal (stem pullout) loads. Nodal restraints represented the mating valve seats for reacting normal loads (compressive), while nodal forces were applied to the faces to

Figure 12 . Anchor/Darling Disk Finite Element Model

represent the friction forces between the disk faces and the valve seats.

5.5.5 Walworth and Anchor/Darling Disks – Model Verification

The Walworth disk mesh of 27,852 elements, and the Anchor/Darling disk mesh of 48,849 elements were verified using the first method described in Section 5.5.2 for the Anchor/Darling stem T-head model. Because of the relative volumes of the disks, the element densities for the two models were approximately equal. Iterations in mesh refinement (method 2 discussed in Section 5.5.2) indicated that the predicted disk stresses under stem torque and thrust loadings were within about 5% of "exact" values. This was considered acceptably accurate and these models were employed in the subsequent evaluations.

5.5.6 Walworth and Anchor/Darling Valve Seats

The valve seats are mounted into the body of the valve (see Figure. 2). The Walworth valve seats are about 5.84" O.D. by 5.39" I.D., while the Anchor/Darling valve seats are about 8.77" O.D. by 8.39" I.D. The Walworth and Anchor/Darling valve seats were not represented as elements in the disk finite element models, and seat stresses were not directly calculated. However, the valve seats were simulated by using nodal restraints on the disk faces and nodal forces to represent frictional forces. The seat stresses were determined as follows: First, the sum of the seat reaction forces was divided by the seat area to calculate the average seat stresses (compressive and shear). Second, the range of stress from closing to opening were calculated. Third, the principal stress values were determined, and the range of Tresca stress was calculated. Finally, a stress intensification factor of 2 was applied to the seat stresses to account for the fact that the seat normal forces were not uniform (as assumed above), but varied by a factor of 2.

5.6. Finite Element Solution Enhancement Routine

The ABAQUS static solution routine used in the evaluation (see Appendix A for ABAQUS verification documentation) calculated the response of a model for a given disk closing (seating or wedging) load condition in Step 1, and followed with a disk opening (unseating or unwedging) condition in Step 2. Separate resulting displacements, stresses, strains, etc., for the closing and opening conditions were available as standard ABAQUS output. However, the fatigue evaluations specifically require that the "stress range" (or difference) in stress states from the closing to opening conditions be calculated. This was not a standard output for ABAQUS. Therefore, a user subroutine written to provide the necessary calculations was included as a supplement to the model input file. A listing of this subroutine is included in Appendix A. The subroutine performed the following operations:

- (1) at each increment in solution Step 1 the stress tensor at all element integration points was written to an array (array #1). Note that a stable solution in ABAQUS generally required that the loads be linearly increased (ramped loading) in several "increments" until the full loads were applied.
- (2) at each increment in solution Step 2 the stress tensor at all element integration points was written to another array (array #2)
- (3) array #2 was subtracted from array #1, resulting in a "difference" array containing stress tensors
- (4) an ABAQUS subroutine was called to calculate the principal stresses for each stress tensor in the difference array
- (5) the Tresca stress, which describes the maximum stress range at a point, was then calculated from the principal stresses. This resulted in an output variable defined as the Tresca Stress Range for the given disk closing/opening cycle.

The above user subroutine was verified using several test models by comparing their results to hand calculations. Additionally, results using the routine on a yoke model and a stem T-head model were spot checked with hand calculations.

Note that the resulting Tresca stress range values from the above routine were not always the largest stress range that a component experienced in the test closing/opening cycle. For example, the valve seats and the disk faces experienced their greatest stress ranges from a "zero stress" condition before closing to the stress state after closing (but before opening). Therefore, the stress range values from the above enhancement routine were compared for all applicable models to the Tresca stress range values for the prior-to-closing (zero stress) to after closing case to determine the maximum range values. In this way, the maximum stress range was always used in the fatigue evaluations.

6. STRESS RESULTS

6.1. Finite Element Analysis Results

Figures 13 to 20 show the maximum Tresca stress ranges identified for load case C defined in Section 4 for the various valve parts using the methods of Section 5. The stress distributions shown are nearly identical for load cases A and B scaled to the maximum values listed in Section 6.3.

6.2. Stress Intensification in the Walworth Yoke

The stress distribution in Figure 13 for the Walworth yoke shows a very concentrated stress location on an edge just below the top flange. Details of this concentration are shown in Figure 21 . The maximum value of the Tresca stress range at this location was estimated by plotting the Tresca stress range at the element integration points from the ABAQUS solution results as a function of distance from the edge. Note that the elements used have only one integration point, located at the element centroid. The elements used are at right angles from the edge. The distance from the edge measured toward the center of the yoke was defined to be negative and the distance measured around the perimeter was positive. Using this definition of distance, the stresses at the integration points are plotted in Figure 22 . Polynomial fits to the data extrapolated from both directions were used to estimate the maximum Tresca stress range at the corner. The Tresca stress range was calculated using the six components of stress at the selected points. The extrapolation could also be performed by independently extrapolating the six components of stress to the point of maximum stress and then computing the Tresca stress range from the extrapolated values. Both methods yield nearly identical results. The increase is a factor of 1.2616 used as a stress intensification factor for all load cases for the Walworth yoke. The maximum stress intensity ranges calculated for the Walworth yoke were multiplied by 1.2616 to obtain the final stress ranges used in Section 7 for the fatigue evaluation.

Figure 13 . Maximum Stress Range for Walworth Yoke, Load Case C

WALWORTH YOKE

LOAD CASE C: POST 89-10 MAXIMUM STEM LOADS

TRESCA STRESS RANGE, PSI

Maximum Stress Range = 73,660 PSI

Figure 14 . Maximum Stress Range for Anchor/Darling Yoke, Load Case C

ANCHOR/DARLING YOKE

LOAD CASE C: POST 89-10 MAXIMUM STEM LOADS

TRESCA STRESS RANGE, PSI

Maximum Stress Range = 14,518 PSI

Figure 15 . Maximum Stress Range for Walworth Stem T-Head, Normal Configuration,
Load Case C

WALWORTH STEM T-HEAD

-NORMAL CLOSING/NORMAL OPENING-
LOAD CASE C: POST 89-10 MAXIMUM STEM LOADS

TRESCA STRESS RANGE, PSI

Maximum Stress Range = 50,068 PSI

Figure 16 . Maximum Stress Range for Walworth Stem T-Head, Abnormal Configuration,
Load Case C

WALWORTH STEM T-HEAD

-ABNORMAL CLOSING/NORMAL OPENING-

LOAD CASE C: POST 89-10 MAXIMUM STEM LOADS

TRESCA STRESS RANGE, PSI

Maximum Stress Range = 214,240 PSI

Figure 17 . Maximum Stress Range for Anchor/Darling Stem T-Head, Load Case C

ANCHOR/DARLING STEM T-HEAD

LOAD CASE C: POST 89-10 MAXIMUM STEM LOADS

TRESCA STRESS RANGE, PSI

Maximum Stress Range = 52,347 PSI

Figure 18 . Maximum Stress Range for Walworth Disk, Normal Configuration,
Load Case C

WALWORTH DISK

-NORMAL CLOSING/NORMAL OPENING-
LOAD CASE C: POST 89-10 MAXIMUM STEM LOADS

TRESCA STRESS RANGE, PSI

Maximum Stress Range = 28,820 PSI

Figure 19 . Maximum Stress Range for Walworth Disk, Abnormal Configuration,
Load Case C

WALWORTH DISK

-ABNORMAL CLOSING/NORMAL OPENING-
LOAD CASE C: POST 89-10 MAXIMUM STEM LOADS

TRESCA STRESS RANGE, PSI

Maximum Stress Range = 41,812 PSI

Figure 20 . Maximum Stress Range for Anchor/Darling Disk, Load Case C

ANCHOR/DARLING DISK

LOAD CASE C: POST 89-10 MAXIMUM STEM LOADS

TRESCA STRESS RANGE, PSI

Maximum Stress Range = 22,468 PSI

Figure 21 . Details of Maximum Stress Range Concentrations for Walworth Yoke,
Load Case B

WALWORTH YOKE - MAX STRESS RANGE

LOAD CASE B: POST 89-10 NOMINAL STEM LOADS

Maximum Stress Range = 61,113 PSI

Figure 22. Maximum Tresca Stress Range Intensification for Walworth Yoke, Load Case B (see text and Fig. 21)

6.3. Maximum Stress Range Summary

Tables 10 and 11 list the calculated maximum Tresca stress range for the Walworth and Anchor/Darling yokes, stem threads, and stem nut threads. These results were taken from the finite element model solutions for the yokes and from Appendix B calculations for the threads. The stresses listed in Table 10 for the Walworth yoke were then multiplied by the stress intensification factor addressed in Section 6.2 for use in the fatigue evaluations of Section 7. For all load cases, the Tresca stress range is calculated using the maximum stress differences experienced in the specific valve part during the complete valve closing/opening cycle (see Sections 4 and 5). The values listed represent the maximum Tresca stress range calculated at any location within the specific valve part.

**Table 10 .Maximum Tresca Stress Range for Walworth Valve:
Yoke, Stem Threads and Stem Nut Threads**

LOAD CASE	CONDITION	MAXIMUM TRESCA STRESS RANGE (PSI)
Yoke:		
A	Pre 89-10 Nominal Torque Setting	36,000*
B	Post 89-10 Nominal Torque Setting	61,113*
C	Post 89-10 Maximum Torque Setting	73,660*
Stem Nut Internal Threads:		
A	Pre 89-10 Nominal Torque Setting	53,348**
B	Post 89-10 Nominal Torque Setting	90,540**
C	Post 89-10 Maximum Torque Setting	109,132**
Stem External Threads:		
A	Pre 89-10 Nominal Torque Setting	63,432**
B	Post 89-10 Nominal Torque Setting	107,648**
C	Post 89-10 Maximum Torque Setting	129,748**

*Does not include the stress intensification factor discussed in Section 6.2

**Maximum range of shear stress using the results of Appendix B increased by a factor of 4 per ASME Section III, NB-3232.2 to account for the potential for stress concentrations at the root of the threads.

Note that the primary plus secondary stress intensity ranges for these components were not calculated because the maximum Tresca stress range was less than $3S_m$.

**Table 11 .Maximum Tresca Stress Range for Anchor/Darling Valve:
Yoke, Stem Threads and Stem Nut Threads**

LOAD CASE	CONDITION	MAXIMUM TRESCA STRESS RANGE (PSI)
Yoke:		
A	Pre 89-10 Nominal Torque Setting	8,372
B	Post 89-10 Nominal Torque Setting	12,064
C	Post 89-10 Maximum Torque Setting	14,518
Stem Nut Internal Threads:		
A	Pre 89-10 Nominal Torque Setting	49,720**
B	Post 89-10 Nominal Torque Setting	71,624**
C	Post 89-10 Maximum Torque Setting	86,224**
Stem External Threads:		
A	Pre 89-10 Nominal Torque Setting	53,948**
B	Post 89-10 Nominal Torque Setting	77,712**
C	Post 89-10 Maximum Torque Setting	93,560**

**Maximum range of shear stress using the results of Appendix B increased by a factor of 4 per ASME Section III, NB-3232.2 to account for the potential for stress concentrations at the root of the threads.

Note that the primary plus secondary stress intensity ranges for these components were not calculated because the maximum Tresca stress range was less than $3S_m$.

Tables 12 and 13 present the calculated maximum Tresca stress range (the primary plus secondary plus peak stress intensity range) and the primary plus secondary stress intensity range (where needed) for the Walworth and Anchor/Darling stem T-heads, disk bodies, disk faces, and valve seats. For some load cases, the maximum Tresca stress ranges for these parts exceeded $3S_m$. The methods of ASME Section III (Ref. 14) require identification of primary plus secondary stress intensity ranges for fatigue evaluation when the stresses exceed $3S_m$. The maximum Tresca stress ranges were calculated by ABAQUS (with the user subroutine discussed in Section 5.6) for the stem T-heads, disk bodies, and disk faces. Primary plus secondary stress intensity ranges basically consist of the calculated stresses without the intensification of a stress concentration. Stress concentrations arise from abrupt changes in geometry (e.g., sharp corners) or application of loads at points. The stress distributions through the finite element models could not be resolved into components of membrane, bending,

and peak stress because of the geometry. Therefore, the methodology for determining the primary plus secondary stress intensity ranges was as follows: First, the element stresses at a concentration, consisting of 1 element < 0.15" thick, were discarded. Then, the highest remaining element Tresca stress range (using element integration point values) was selected to be the primary plus secondary stress intensity range. This was considered to give conservative primary plus secondary stress intensity range values since some peak stress remained.

**Table 12 .Maximum Tresca Stress Range for Walworth Valve:
Stem T-Head, Disk Body, Disk Faces, and Valve Seats**

LOAD CASE	CONDITION	MAXIMUM TRESCA STRESS RANGE (PSI)
Stem T-Head (Normal Configuration):		
A	Pre 89-10 Nominal Torque Setting	24,478
B	Post 89-10 Nominal Torque Setting	41,542
C	Post 89-10 Maximum Torque Setting	50,068
Stem T-Head (Abnormal Configuration):		
A	Pre 89-10 Nominal Torque Setting	98,978*
B	Post 89-10 Nominal Torque Setting	175,770*
C	Post 89-10 Maximum Torque Setting	214,240*
Disk Body (Normal Configuration):		
A	Pre 89-10 Nominal Torque Setting	10,369
B	Post 89-10 Nominal Torque Setting	17,594
C	Post 89-10 Maximum Torque Setting	21,226
Disk Body (Abnormal Configuration):		
A	Pre 89-10 Nominal Torque Setting	20,444
B	Post 89-10 Nominal Torque Setting	34,688
C	Post 89-10 Maximum Torque Setting	41,812

(Table 12. Continued)		
LOAD CASE	CONDITION	MAXIMUM TRESCA STRESS RANGE (PSI)
Disk Faces:		
A	Pre 89-10 Nominal Torque Setting	14,531
B	Post 89-10 Nominal Torque Setting	24,665
C	Post 89-10 Maximum Torque Setting	29,757
Valve Seats:		
A	Pre 89-10 Nominal Torque Setting	15,394**
B	Post 89-10 Nominal Torque Setting	26,124**
C	Post 89-10 Maximum Torque Setting	31,488**

*The primary plus secondary stress intensity range for this component for Load Cases A, B, and C were 35,000 psi, 62,000 psi, and 75,000 psi, respectively. Primary plus secondary stress intensity ranges were not calculated for the remaining components because the maximum Tresca stress range was less than $3S_m$.

**Calculated using the total normal (compressive) reaction load on the restraints that simulated the valve seats and the friction loads divided by the area of the seats (using a seat width of 0.135 in. and a mean seat diameter of 5.55 in.). Also included was a stress intensification factor of 2.

**Table 13 .Maximum Tresca Stress Range for Anchor/Darling Valve:
Stem T-Head, Disk Body, Disk Faces, and Valve Seats**

LOAD CASE	CONDITION	MAXIMUM TRESCA STRESS RANGE (PSI)*
Stem T-Head:		
A	Pre 89-10 Nominal Torque Setting	30,674
B	Post 89-10 Nominal Torque Setting	43,678
C	Post 89-10 Maximum Torque Setting	52,347
Disk Body:		
A	Pre 89-10 Nominal Torque Setting	12,955
B	Post 89-10 Nominal Torque Setting	18,660
C	Post 89-10 Maximum Torque Setting	22,468
Disk Faces:		
A	Pre 89-10 Nominal Torque Setting	12,067
B	Post 89-10 Nominal Torque Setting	17,389
C	Post 89-10 Maximum Torque Setting	20,968
Valve Seats:		
A	Pre 89-10 Nominal Torque Setting	19,354**
B	Post 89-10 Nominal Torque Setting	27,876**
C	Post 89-10 Maximum Torque Setting	33,558**

*Primary plus secondary stress intensity ranges were not calculated for these components because the maximum Tresca stress range was less than $3S_m$.

**Calculated using the total normal (compressive) reaction load on the restraints that simulated the valve seats and the friction loads divided by the area of the seats (using a seat width of 0.188 in. and a mean seat diameter of 8.64 in.). Also included was a stress intensification factor of 2.

7. ESTIMATES OF INCREASE IN FATIGUE USAGE FACTORS

The method chosen for the fatigue evaluation was to use the ASME Code (Ref. 14) Class 1 fatigue approach as a screening method to eliminate areas with low cumulative usage factors (CUFs) from further evaluation. Valve components that could be shown to meet the ASME Code CUF limit of 1.0 for 40 or 60 years for all load cases are judged to have an acceptable increase in fatigue usage due to the increase in torque switch settings. Components for which the CUF exceeds 1.0 require further assessment. The ASME Code design curves were developed by reducing the best-fit curves to the applicable test data by factors of 2 on stress or 20 on cycles, whichever was more conservative. The data were developed using small specimens that were tested at room temperature in an air environment. It is assumed that the design curves give a high confidence that fatigue failures would not occur. The factors of 2 on stress and 20 on cycles are intended to account for parameters such as size, surface finish, data scatter, and environment. Use of the design curves have been estimated to give a factor of safety of about 3 on cycles to failure for nuclear components under actual operating conditions (Ref. 17).

The yoke and disk body materials are listed in Section III of the ASME Code. However, the stem nut, stem, disk faces and valve seats are not ASME Code approved materials. Consequently, approved stress limits and fatigue curves for these materials were not available. Representative values for these materials obtained using the ASME Code methodology were used.

Tables 14 and 15 summarize the calculations for estimates of the increases in CUFs for the 6-in. Walworth and the 10-in. Anchor Darling valves, respectively. The fatigue evaluation considered seven locations for each valve for three inservice testing conditions identified as load cases A, B, and C (see Section 4).

The range in column two of the tables is the primary plus secondary plus peak stress intensity range, calculated in the stress analysis and reported in Section 6. The finite element mesh for the yoke of the Walworth valve was not quite fine enough to fully capture the peak stresses, so polynomial curve fits to adjacent calculation points were used to estimate the peak stress (Section 6.2). The primary plus secondary stress intensity range limit of $3S_m$ was used for Code approved materials. The other materials were high-strength, so an S_m value equal to one-third the tensile strength (S_u) at temperature was used. Thus $3S_m$ was equal to the tensile strength for those materials. The primary plus secondary stress intensity range (S_n) was determined by the stress analysis and reported in Section 6. Where the primary plus secondary plus peak stress intensity range is less than $3S_m$, there is no need to calculate the S_n value, since it was also obviously less than $3S_m$. The S_n 's for all locations were less than $3S_m$.

The alternating stress intensity S_{alt} is one-half the product of K_e and the primary plus secondary plus peak stress intensity range. To select the allowable number of cycles from the fatigue curve, S_{alt} is corrected by multiplying by the ratio of the modulus of elasticity (E) on the appropriate fatigue curve to the modulus of elasticity used in the analysis. $E_{analysis}$ is the modulus of elasticity value used in the stress analysis reported in Section 6, and E_{curve} is the modulus of elasticity associated with the appropriate fatigue curve. For the yoke and disk body, the modulus of elasticity on the fatigue curve

on Figure I-9.1 of Section III of the ASME Code was used. The materials for the stem, stem nut, disk faces and valve seats are not ASME Code materials. All are high strength materials (S_u greater than 90 ksi). The stem nut material appears to be similar to a high-strength bolting material, and the stem material appears to be similar to a high-strength, high-alloy steel. For the stem and stem nut S_{alt} values in Tables 14 and 15, the most conservative of the possible fatigue curves in Section III of the ASME Code that could be used (that is, the curve that would result in the fewest number of allowable cycles for a given S_{alt}), is curve I-9.4, the high-strength bolting curve. Using this curve gave an acceptable fatigue life (last column of Tables 14 and 15) for these materials, so no further evaluation was required.

The disk faces and valve seats are a very hard wear-resistant, cobalt-based alloy (Stellite 6) that is used for surfaces with high compressive contact stresses such as ball bearings and valve seats. There is no appropriate fatigue curve in the ASME Code. The manufacturer was contacted but had no fatigue data. A few high-cycle fatigue curves for other stellites (Stellites 25 and 188) were found in Brown et al. (Ref. 18), but no modulus of elasticity values were given on the curves. Therefore, it is assumed that the modulus of elasticity used for the curves in Brown et al. is the same as that used in the analysis.

Using room temperature data for 1-in. bar of L-605 (Stellite 25) (20 Cr, 15 W, 10 Ni, balance Co versus 28 Cr, 5 W, 1 C, balance Co for Stellite 6), and a factor of 2 on stress, the allowable cycles would be $>10^7$ for the Walworth and Anchor Darling disk faces and valve seats. (The factor of 2 on stress governed rather than the factor of 20 on cycles.) The CUFs over 40 years would be minimal. A similar result was found using data at room temperature for Haynes Alloy 188 (22 Cr, 22 Ni, 14 W, low C, balance Co). The ASME Code data were also for room temperature and the factors of 2 on stress and 20 on cycles are intended to account for environmental effects such as temperature. Since the disk faces and valve seats are loaded in compression, but never in tension, it is expected that their fatigue lives would exceed that predicted for small tensile fatigue specimens. Field data indicates that the predominant failure mode has been scratching of the faces caused by seating, followed by erosion of the scratches by high-pressure, high-temperature steam.

The allowable numbers of cycles (N) were computed by interpolation between the values listed in the appropriate fatigue curve, for example, in Tables I-9.1 and I-9.4 of the ASME Code. The usages that would be accumulated by 100 base cycles plus one or eight test cycles per year over 40 and 60 years of operation were calculated. Even using load case C, the CUFs based on eight cycles per year for 60 years are less than 1 for all subcomponents, so no further evaluation was required.

Reaching a CUF of 1 does not represent a step increase in the probability of fatigue failures. It has been estimated that a CUF of 1 calculated using ASME Code design curves represents low probability of crack initiation for laboratory specimens for which the stress level is well known. A CUF of 20 (accounting for the factor of 20 on cycles in the ASME Code design curve) would represent a 50% probability of crack initiation for test specimens where the stress level and number of cycles is well known. The test specimens are small polished specimens that have been tested in air at room temperature. As stated above, a CUF of 1 has been estimated to represent

approximately a factor of safety of 3 on components in service in nuclear power plants (Ref. 17). A CUF of 3, therefore, might represent a 50% failure probability for nuclear components. For cases where the CUF only slightly exceeds 1, the probability of a fatigue failure is expected to remain relatively low. For a CUF of 3, however, there may be a 50% probability of failure for components in the field, presuming that the calculated stress levels and assumed load cycles are representative of the service conditions actually being experienced by the component.

Table 14. Estimates of Cumulative Usage Factors for Walworth Valve

Case ^b	Stress Range (psi) ^{a,c}	3S _m ^d (ksi)	S _n (psi) ^c	K _e	S _{alt} (psi)	E _{analysis} (ksi)	E _{curve} (ksi)	adj. S _{alt} (psi) ^e	S _u (ksi)	N ^f (cycles)	usage ^g				
											40 yr		60 yr		
											8/yr	1/yr	8/yr	1/yr	
Yoke (A216 WCB) (Figure I-9.1)															
A	45418	69.9	-	1	22709	29.5	30	23094	70	49002	0.009	0.003	0.012	0.003	
B	77100	69.9	-	1	38550	29.5	30	39203	70	9116	0.046	0.015	0.064	0.018	
C	92929	69.9	-	1	46465	29.5	30	47252	70	5238	0.080	0.027	0.111	0.031	
Internal threads - stem nut (B584 Grade 86300) (Figure I-9.4) ^h															
A	53348	110	-	1	26674	15.5	30	51627	110	3794	0.111	0.037	0.153	0.042	
B	90540	110	-	1	45270	15.5	30	87619	110	1307	0.321	0.107	0.444	0.122	
C	109132	110	-	1	54566	15.5	30	105612	110	900	0.467	0.156	0.645	0.178	
External threads - stem (A182 F6b) (Figure I-9.4) ^h															
A	63432	110	-	1	31716	29.2	30	32585	110	11364	0.037	0.012	0.051	0.014	
B	107648	110	-	1	53824	29.2	30	55299	110	3305	0.127	0.042	0.176	0.048	
C	129748	110	-	1	64874	29.2	30	66651	110	2271	0.185	0.062	0.255	0.070	
Stem T-head (normal) (A182 F6b) (Figure I-9.4) ^h															
A	24478	101	-	1	12239	26.1	30	14068	101	762581	0.001	0.000	0.001	0.000	
B	41542	101	-	1	20771	26.1	30	23875	101	34679	0.012	0.004	0.017	0.005	
C	50068	101	-	1	25034	26.1	30	28775	101	15043	0.028	0.009	0.039	0.011	
Stem T-head (abnormal) (A182 F6b) (Figure I-9.4) ^h															
A	98978	101	35000	1	49489	26.1	30	56884	101	3203	0.131	0.044	0.181	0.050	
B	175770	101	62000	1	87885	26.1	30	101017	101	981	0.428	0.143	0.591	0.163	
C	214240	101	75000	1	107120	26.1	30	123126	101	668	0.629	0.210	0.868	0.239	
Disk body (A217 WC6) (Figure I-9.1)															
A	20444	62.7	-	1	10222	26.9	30	11400	70	> 10 ⁶	0.000	0.000	0.000	0.000	
B	34688	62.7	-	1	17344	26.9	30	19343	70	112795	0.004	0.001	0.005	0.001	
C	41812	62.7	-	1	20906	26.9	30	23315	70	47954	0.009	0.003	0.012	0.003	
Disk faces [CoCr-A (Stellite 6)] ⁱ (Brown et al., 1995)															
A	14531	124	-	1	7266	29	29	7266	124	> 10 ⁷	0.000	0.000	0.000	0.000	
B	24665	124	-	1	12333	29	29	12333	124	> 10 ⁷	0.000	0.000	0.000	0.000	
C	29757	124	-	1	14879	29	29	14879	124	> 10 ⁷	0.000	0.000	0.000	0.000	
Valve seats [CoCr-A (Stellite 6)] ⁱ (Brown et al., 1995)															
A	15394	124	-	1	7697	29	29	7697	124	> 10 ⁷	0.000	0.000	0.000	0.000	
B	26124	124	-	1	13062	29	29	13062	124	> 10 ⁷	0.000	0.000	0.000	0.000	
C	31488	124	-	1	15744	29	29	15744	124	> 10 ⁷	0.000	0.000	0.000	0.000	

Table 15. Estimates of Cumulative Usage Factors for Anchor Darling Valve

Case ^b	Stress Range (psi) ^c	3S _m ^d (ksi)	S _n (psi) ^c	K _e	S _{alt} (psi)	E _{analysis} (ksi)	E _{curve} (ksi)	adj. S _{alt} (psi) ^e	S _u (ksi)	N ^f (cycles)	usage ^g				
											40 yr		60 yr		
											8/yr	1/yr	8/yr	1/yr	
Yoke (A216 WCB) (Figure I-9.1)															
A	8372	69.9	-	1	4186	29.5	30	4257	70	> 10 ⁶	0.000	0.000	0.000	0.000	
B	12064	69.9	-	1	6032	29.5	30	6134	70	> 10 ⁶	0.000	0.000	0.000	0.000	
C	14518	69.9	-	1	7259	29.5	30	7382	70	> 10 ⁶	0.000	0.000	0.000	0.000	
Internal threads - stem nut (B584 Grade 86300) (Figure I-9.4) ^h															
A	49720	110	-	1	24860	15.5	30	48116	110	4371	0.096	0.032	0.133	0.037	
B	71624	110	-	1	35812	15.5	30	69314	110	2099	0.200	0.067	0.276	0.076	
C	86224	110	-	1	43112	15.5	30	83443	110	1442	0.291	0.097	0.402	0.111	
External threads - stem (A276-410T) (Figure I-9.4) ^h															
A	53948	100	-	1	26974	29.2	30	27713	100	18492	0.023	0.008	0.031	0.009	
B	77712	100	-	1	38856	29.2	30	39921	100	6724	0.062	0.021	0.086	0.024	
C	93560	100	-	1	46780	29.2	30	48062	100	4381	0.096	0.032	0.132	0.037	
Stem T-head (A276-410T) (Figure I-9.4) ^h															
A	30674	90	-	1	15337	26.7	30	17233	90	183761	0.002	0.001	0.003	0.001	
B	43678	90	-	1	21839	26.7	30	24538	90	30676	0.014	0.005	0.019	0.005	
C	52347	90	-	1	26174	26.7	30	29408	90	15469	0.027	0.009	0.037	0.010	
Disk body (A216 WCB) (Figure I-9.1)															
A	12955	53.4	-	1	6478	26.7	30	7278	70	> 10 ⁶	0.000	0.000	0.000	0.000	
B	18660	53.4	-	1	9330	26.7	30	10483	70	> 10 ⁶	0.000	0.000	0.000	0.000	
C	22468	53.4	-	1	11234	26.7	30	12622	70	915932	0.000	0.000	0.001	0.000	
Disk faces [CoCr-A (Stellite 6)] ⁱ (Brown et al., 1995)															
A	12067	124	-	1	6034	29	29	6034	124	> 10 ⁷	0.000	0.000	0.000	0.000	
B	17389	124	-	1	8695	29	29	8695	124	> 10 ⁷	0.000	0.000	0.000	0.000	
C	20968	124	-	1	10484	29	29	10484	124	> 10 ⁷	0.000	0.000	0.000	0.000	
Valve seats [CoCr-A (Stellite 6)] ⁱ (Brown et al., 1995)															
A	19354	124	-	1	9677	29	29	9677	124	> 10 ⁷	0.000	0.000	0.000	0.000	
B	27876	124	-	1	13938	29	29	13938	124	> 10 ⁷	0.000	0.000	0.000	0.000	
C	33558	124	-	1	16779	29	29	16779	124	> 10 ⁷	0.000	0.000	0.000	0.000	

Notes

- The ranges for the yoke have been multiplied by a stress concentration factor of 1.2616.
- Case A is the old method, Case B is the new method, Case C is a worst-case estimate of the new method.
- The stress intensity ranges were determined in the stress analysis (Section 6).
- 3S_m is based on ASME Code values, or if not a Code material, S_m is the lesser of 2S_y/3 or S_u/3.
- adj. S_{alt} is S_{alt} multiplied by E_{curve} divided by E_{analysis}.
- N is the allowable number of cycles from the ASME Code fatigue curve.
- usage is in the cumulative usage factor for 100 base cycles and 8 cycles/yr or 1 cycle/yr.
- The stem material is assumed to be a high-strength bolting material (ASME Code Figure I-9.4).
- Includes factor of 2 on stress and 20 on cycles using data from Brown et al., 1995.

8. SUMMARY

Some motor operated valves within commercial nuclear power plants now have higher torque switch settings due to the requirements for operability testing addressed by USNRC Generic Letter 89-10. These higher settings increase both the thrust and torque applied to the valve stem and increase stresses in the valve body and various internal parts which may in turn increase the fatigue usage factors. Although the motor operators may also have significantly increased loading, they were not included in the scope of this study. In addition, only motor operated gate valves were considered. The regions within motor operated gate valves where increased fatigue is judged to be a concern include the valve seats, disks, yokes, stems, and stems nuts. The results of this analysis; 1) identify the critical areas within the valves where fatigue may be a concern, and 2) provide estimates of the increase in fatigue usage factors in these areas caused by the increased loads applied to the stem during inservice testing prior and subsequent to the issuance of USNRC 89-10.

A survey of valve fatigue failures was conducted as part of this study. A USNRC Information Notice identifies fatigue cracks in the yoke of some Walworth valves of design similar to the valve evaluated here. The location of the cracks corresponds to the area of high stress ranges identified for this valve in Section 6 of this report but the CUFs are much less than one indicating that fatigue cracking is not predicted by the analyses (see discussion below). With the exception of the valve yoke cracking, no specific candidate for fatigue failure was identified by the industry experience queried in this survey. The most likely problems indicated are the disk scoring and gasket failures. Neither of these are considered a source of catastrophic failure.

Two valves available at INEL were used for this study. Both are pressure seal gate valves of flexible wedge disk design. A 6 in., 600 lb. Walworth valve was selected to represent a class of relatively weak valves in terms of internal dimensions. In contrast, a 10 in., 900 lb. Anchor/Darling valve was used to represent relatively strong valves of this type. Considerable test data for both valves were available.

Three potential inservice testing load conditions were considered in this report to evaluate the effects of increased torque switch settings on the fatigue life of the various valve components. The following torque switch settings were considered:

Load Case A), the nominal settings used before 89-10 was issued.

Load Case B), the nominal settings used after 89-10 was issued.

Load Case C), the maximum settings used after 89-10 was issued.

Stem thrust and torque during valve closing for the three load cases were estimated with equations commonly used and accepted within the commercial nuclear industry. Adjustments were made, based on valve test data for the two valves evaluated, to account for overshoot after torque switch trip and to estimate the valve opening loads. These loads were then applied to detailed finite element models of the various valve

parts or used in hand calculations to estimate the maximum Tresca stress ranges that occur during the inservice testing.

The fatigue evaluation considered seven locations for each valve for the three inservice testing load cases. The ASME Code Class 1 fatigue approach was used as a screening method to eliminate areas with low CUFs from further evaluation. Assuming 100 base cycles plus eight test cycles per year over 40 and 60 years of operation, the disk, valve yoke, stem threads, stem nut threads, and stem T-head for both valves were shown to have acceptable fatigue life for all load cases. The ASME Code cumulative usage factors (CUFs) for these components do not exceed 1.0 for 40 and 60 years of operability testing for all torque switch settings considered.

The stress intensity ranges in disk faces and valve seats for both valves are relatively low. The faces and seats are made of a very hard wear-resistant, cobalt-based alloy (Stellite 6) that is used for surfaces with high compressive contact stresses such as ball bearings and valve seats. These faces and seats are loaded in compression and would not likely develop fatigue cracks. Field experience has indicated that scratches eroded by high-pressure, high-temperature steam is the most likely cause of degradation.

Although no fatigue failures due to inservice testing were predicted for the Walworth yoke, the highest stress concentrations were shown to occur at the location where fatigue failures have occurred in one operating plant. The CUFs for the Walworth valve yoke are much less than 1.0 indicating that fatigue cracking is not predicted by the analysis. The Information Notice identified insufficient bolt torque and existing casting defects as factors contributing to the fatigue cracking. Insufficient bolt torque can result in high local stresses not typical of most valve installations. Both of these factors are considered anomalous and not directly applicable to this study. The fact that all of the cracked yokes occurred at one plant and all of the same manufacture may indicate problems in installation and inspection rather than valve design.

The analysis performed here only considers the effects of inservice testing for operability at a rate of one or eight tests per year plus 100 base cycles on the fatigue life of the various valve components. The effects of other cycles of the valves are, of course, cumulative and could significantly reduce the fatigue lives predicted in this study. For most valves of this type, however, the assumption of 100 base cycles is conservative.

9. CONCLUSIONS

The two valves analyzed in this study are judged to be typical of a large number of gate valves currently installed in commercial nuclear plants and subject to periodic inservice testing. The dimensions, tolerances, and material properties used are characteristic of this type of valve. The loads were based on industry equations, Limitorque data, and test data from each of the two valves analyzed. The finite element meshes were refined until the maximum Tresca stress ranges no longer increased significantly. All of these factors provide for accurate estimates of both the location and magnitude of the maximum stress ranges. Considerable variability exists between individual values in the dimensions of the castings, the material properties, the design differences between valve manufacturers, and the actual plant conditions during testing which will cause variations in the stress range magnitudes and distributions. The results presented here are, however, judged to be representative of the fatigue effects of inservice testing for this type of valve.

Considering the inherent conservatism in the material properties and the fatigue data, the results of the analysis show that, for this type of valve, increasing the torque switch settings for inservice testing have the following effects on fatigue of the valve components:

- (1) Fatigue failures are unlikely to occur in the valve disks, yokes, stem T-heads, stem nut threads, and stem threads. Fatigue cracks in Walworth valve yokes have been identified in one commercial plant but are not predicted by this analysis. The probable causes of these cracks were insufficient bolt torque and existing casting defects.
- (2) Although the maximum stress ranges are relatively low, the effects on fatigue life for the disk faces and valve seats could not be estimated since fatigue curves are not available. The faces and seats are, however, loaded in compression and would not likely develop fatigue cracks. Field experience has indicated that scratches eroded by high-pressure, high-temperature steam is the most likely cause of degradation.

10. ACKNOWLEDGMENTS

The authors would like to acknowledge the assistance of H. L. Magleby of Lockheed Martin Idaho Technologies for coordinating this effort with the USNRC and for providing considerable technical direction. R. Steele, K. G. DeWall, and J. C. Watkins were an invaluable resource providing help with details of the valves, valves test data, plant operations, and the use of the IVA software. The report was reviewed for technical content by G. K. Miller and H. L. Magleby, both of Lockheed Martin Idaho Technologies.

11. REFERENCES

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Appendix A Computer Verification and Sample Data

Appendix A includes:

1. Documentation of computer configuration and verification
2. Listing of ABAQUS finite element solution enhancement subroutine (see description in Section 5.7).
3. Sample listing of IVA data – Walworth Valve.
4. Sample listing of IVA data – Anchor/Darling Valve.

1. COMPUTER CONFIGURATION AND VERIFICATION

The following documentation presents the traceability for computer programs used in the analysis reported here.

Report Title: Effects on Fatigue Life of Gate Valves Due to Higher Torque Switch Settings During Operability Testing

Author: W. D. Richins, S. D. Snow, G. K. Miller, M. J. Russell, and A. G. Ware

Date: December 1995

Program Used: I-DEAS **Version:** Master Series 1.3

Computer Used: DEC ALPHA Station ASHTON

Verification Manual/Test Problem Manual/Example Manual:

♦SDRC I-DEAS Verification Manual, Structural Dynamics Research Corporation, Milford, OH

♦C. C. O'Brien letter to Applied Mechanics, Jan. 24, 1995, "Verification of SDRC I-DEAS MS 1.3C Software for the DEC Platform," CCO-03-95.

Program Used: ABAQUS **Version:** 5.4

Computer Used: DEC ALPHA Stations ASHTON and CASPER

Verification Manual/Test Problem Manual/Example Manual:

♦ABAQUS Example Problems Manual, Hibbitt, Karlsson, and Sorenson, Inc., Providence, Rhode Island, 1994.

♦C. C. O'Brien letter to Applied Mechanics, February 7, 1995, "Verification of HKS ABAQUS Version 5.4 for the DEC Alpha Platform," CCO-04-95.

Program Used: MathCad **Version:** 4.0 **Module:** NA

Computer Used: DEC5000 **Model:** 5000/200

Verification Manual/Test Problem Manual/Example Manual: None

Author: NA

Date: NA

Comments: Application validated by hand calculation during check.

2. ABAQUS FINITE ELEMENT SOLUTION ENHANCEMENT ROUTINE

*USER OUTPUT VARIABLES

25,

*USER SUBROUTINE

SUBROUTINE UVARM(UVAR,DIRECT,T,TIME,DTIME,CMNAME,ORNAME,

1 NUVARM,NOEL,NPT,NLAYER,NSPT,KSTEP,KINC,

2 NDI,NSHR)

C

INCLUDE 'ABA_PARAM.INC'

C

CHARACTER*8 CMNAME,ORNAME

DIMENSION UVAR(*),TIME(2),DIRECT(3,3),T(3,3)

DIMENSION S(6),PS(3),ANPS(3,3)

C USER DEFINED DIMENSION STATEMENTS

CHARACTER*3 FLGRAY(15)

DIMENSION ARRAY(15),JARRAY(15)

C READ IN STEP 1 STRESS TENSOR INTO USER-DEFINED VARIABLE UVAR

IF (KSTEP.EQ. 1) THEN

CALL GETVRM('S',ARRAY,JARRAY,FLGRAY,JRCD)

UVAR(1)=ARRAY(1) !S11

UVAR(2)=ARRAY(2) !S22

UVAR(3)=ARRAY(3) !S33

UVAR(4)=ARRAY(4) !S12

UVAR(5)=ARRAY(5) !S13

UVAR(6)=ARRAY(6) !S23

if (jrzd .ne. 0) then

write(0,*) "jrzd=",jrzd

endif

C

C READ IN STEP 2 STRESS TENSOR INTO USER-DEFINED VARIABLE UVAR

ELSE IF (KSTEP.EQ. 2) THEN

CALL GETVRM('S',ARRAY,JARRAY,FLGRAY,JRCD)

UVAR(7)=ARRAY(1) !S11

UVAR(8)=ARRAY(2) !S22

UVAR(9)=ARRAY(3) !S33

UVAR(10)=ARRAY(4) !S12

UVAR(11)=ARRAY(5) !S13

UVAR(12)=ARRAY(6) !S23

if (jrzd .ne. 0) then

write(0,*) "jrzd=",jrzd

endif

C

C CALCULATE THE DIFFERENCE BETWEEN STRESS TENSORS OF STEPS 1 & 2

C

UVAR(13)=UVAR(1)-UVAR(7) !delta S11

UVAR(14)=UVAR(2)-UVAR(8) !delta S22

UVAR(15)=UVAR(3)-UVAR(9) !delta S33

```

      UVAR(16)=UVAR(4)-UVAR(10)      !delta S12
      UVAR(17)=UVAR(5)-UVAR(11)      !delta S13
      UVAR(18)=UVAR(6)-UVAR(12)      !delta S23
ENDIF
C
C      CALCULATE THE PRINCIPAL STRESSES FOR THE "STRESS DIFFERENCE" TENSOR
C
C
      S(1)=UVAR(13)
      S(2)=UVAR(14)
      S(3)=UVAR(15)
      S(4)=UVAR(16)
      S(5)=UVAR(17)
      S(6)=UVAR(18)
      NDI=3
      NSHR=3
      LSTR=1
      CALL SPRIND(S,PS,ANPS,LSTR,NDI,NSHR)
C
C      UVAR19 THRU 21 ARE THE PRINCIPAL STRESSES
C
      UVAR(19)=PS(1)
      UVAR(20)=PS(2)
      UVAR(21)=PS(3)
C
C      CALCULATE STRESS INTENSITIES
C
      UVAR(22)=ABS(UVAR(19)-UVAR(20))
      UVAR(23)=ABS(UVAR(20)-UVAR(21))
      UVAR(24)=ABS(UVAR(21)-UVAR(19))
C
C      DETERMINE WHICH STRESS INTENSITY IS THE LARGEST (i.e.,UVAR(25)=TRESCA)
C
      IF ((UVAR(22) .GE. UVAR(23)) .AND. (UVAR(22) .GE. UVAR(24))) THEN
        UVAR(25)=UVAR(22)
      ENDIF
C
      IF ((UVAR(23) .GE. UVAR(24)) .AND. (UVAR(23) .GE. UVAR(22))) THEN
        UVAR(25)=UVAR(23)
      ENDIF
C
      IF ((UVAR(24) .GE. UVAR(22)) .AND. (UVAR(24) .GE. UVAR(23))) THEN
        UVAR(25)=UVAR(24)
      ENDIF
C
      RETURN
      END

```

3. SAMPLE LISTING OF IVA DATA – WALWORTH VALVE.

```

#####
IVA-4.10          MOTOR OPERATED GATE VALVE CLOSING          Date:11-01-1995
CAPABILITY ASSESSMENT          Time: 09:36:02
Page: 1 of 3

File:C:\IVA\WAL6POST.IVA
Plant/Unit:NRC VALVE FATIGUE STUDY
System:post 89-10 nominal case
Number:4SDP-MOV422B          P&ID Coord:
Valve Manuf:Walworth          Type:Gate          Size:6
Oper. Manuf:Limitorque          Model:SMB/SB/SBD-0          Number:
Comments:

#####
GATE VALVE CLOSING DATA
Design Basis Stem Thrust Response:Predictable
INEL Friction Factor:_.400          Disc Factor:_.500
INEL Friction Factor Offset:50.0          Hooking Factor:
Fluid Subcooling:Less than 70°F          Seat Angle (deg):5
Pressure (psi):1000.          DP (psid):1000.          Packing (lbf):1250.
Seat MD (in):5.515          Industry Equation Disc:          Stem Through Packing:
Seat ID (in):5.380          Diameter (in):5.515          Diameter (in):1.250
Seat OD (in):5.650          Area (in²):23.888          Area (in²):1.227
#####
STEM/STEM NUT CLOSING DATA
Type of Threads:29½ General ACME          Stem Lubricant:
Stem Diameter at stem-nut:1.250          Lead Description:
Threads/inch:          Pitch:_.250000          Lead:_.500000
Design Basis Stem/Stem-Nut Friction:_.150          Stem Factor:_.01421
Stem/Stem-Nut Friction During Stall:_.150          Stem Factor:_.01421
Stroke Length (in):5.940          Time (sec):29.165          Speed (in/min):12.220
#####
OPERATOR MECHANICAL CLOSING DATA
Overall Ratio:69.560          Worm Gear Ratio:37:1          Moment Arm (in):
Operator efficiency:_.400          Limits From:Limitorque          Thrust (lbf):24000.
Stall efficiency:_.500          Spring Pack:0501-184          Torque (ft-lbf):500.0
Operator Setup On:Torque Switch Only          Torque Switch Setting:1.75
Thrust (lbf):          Stem Friction:
Torque (ft-lbf):          Stem Factor:
#####
OPERATOR ELECTRICAL CLOSING DATA
Motor Assessment Method:Full Voltage Range
APPLICATION FACTOR: Nom Voltage:_.900          Min Voltage:_.900          Max Voltage:_.900
RATED: Voltage (volts):460.0          STALL: Current (amps):6.2
Torque (ft-lbf):25.0          Torque (ft-lbf):31.0
Speed (RPM):1700.          Power Factor:
MOV Temperature (°F):          % Current Loss:          % Torque Loss:
Motor Frame:
#####
POWER SUPPLY CLOSING DATA
Voltage Loss Calculation Method:Voltage Divider
Voltage Type:AC-3Ø          % Min Voltage:80.0          Min Voltage:368.0
Nominal Voltage:460.0          % Max Voltage:110.0          Max Voltage:506.0
Thermal Overload Resistance:          Starting Resistance:
#####

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IVA-4.10 MOTOR OPERATED GATE VALVE CLOSING Date:11-01-1995
CAPABILITY ASSESSMENT Time: 09:36:02
Page: 3 of 3

File:C:\IVA\WAL6POST.IVA
Plant/Unit:NRC VALVE FATIGUE STUDY
System:post 89-10 nominal case
Number:4SDP-MOV422B P&ID Coord:
Valve Manuf:Walworth Type:Gate Size:6
Oper. Manuf:Limitorque Model:SMB/SB/SBD-0 Number:

=====
CAPABILITIES REQUIREMENTS Current Operator Settings
Torque Switch: 1.75
Available Available Available INEL Industry Output Torque: 247.7
at Max. at Nom. at Min. Estimate Estimate Output Thrust: 17431.
Voltage Voltage Voltage f=.448 i=.500
82628. 44056. 28196. 13697. 14421. Stem Thrust
Margin
27.3 20.9 Percent Margin
1174.1 626.0 400.7 194.6 204.9 Operator Torque
Margin
27.3 20.9 Percent Margin
33.8 22.5 14.4 7.0 7.4 Motor Torque
506.0 460.0 368.0 Voltage @ MOV
506.0 460.0 368.0 Source Voltage
Reference Maximum Minimum Minimum
3.61 2.50 1.49 1.54 Torque Switch Setting
=====

4. SAMPLE LISTING OF IVA DATA – ANCHOR/DARLING VALVE.

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#####f
IVA-4.10          MOTOR OPERATED GATE VALVE CLOSING          Date:11-01-1995
CAPABILITY ASSESSMENT          Time: 09:38:08
Page: 1 of 3

File:C:\IVA\ANC10PST.IVA
Plant/Unit:NRC VALVE FATIGUE STUDY
System:post 89-10 nominal case
Number:E-6162-8-2          P&ID Coord:
Valve Manuf:Anchor/Darling          Type:Gate          Size:10
Oper. Manuf:Limitorque          Model:SMB/SB/SBD-1          Number:
Comments:

#####;
GATE VALVE CLOSING DATA
Design Basis Stem Thrust Response:Predictable
INEL Friction Factor:.400          Disc Factor:.500
INEL Friction Factor Offset:50.0          Hooking Factor:
Fluid Subcooling:Less than 70°F          Seat Angle (deg):3
Pressure (psi):1000.          DP (psid):1000.          Packing (lbf):2000.
Seat MD (in):8.640          Industry Equation Disc:          Stem Through Packing:
Seat ID (in):8.515          Diameter (in):8.640          Diameter (in):2.000
Seat OD (in):8.766          Area (in²):58.630          Area (in²):3.142
#####N
STEM/STEM NUT CLOSING DATA
Type of Threads:29½ Stub ACME          Stem Lubricant:
Stem Diameter at stem-nut:2.000          Lead Description:
Threads/inch:          Pitch:.333333          Lead:1.000000
Design Basis Stem/Stem-Nut Friction:.150          Stem Factor:.02621
Stem/Stem-Nut Friction During Stall:.150          Stem Factor:.02621
Stroke Length (in):9.690          Time (sec):28.232          Speed (in/min):20.594
#####N
OPERATOR MECHANICAL CLOSING DATA
Overall Ratio:82.550          Worm Gear Ratio:34:1          Moment Arm (in):2.710
Operator efficiency:.400          Limits From:Limitorque          Thrust (lbf):45000.
Stall efficiency:.500          Spring Pack:0701-212          Torque (ft-lbf):850.0
Operator Setup On:Torque Switch Only          Torque Switch Setting:3.00
Thrust (lbf):          Stem Friction:
Torque (ft-lbf):          Stem Factor:
#####N
OPERATOR ELECTRICAL CLOSING DATA
Motor Assessment Method:Full Voltage Range
APPLICATION FACTOR: Nom Voltage:.900          Min Voltage:.900          Max Voltage:.900
RATED: Voltage (volts):460.0          STALL: Current (amps):
Torque (ft-lbf):60.0          Torque (ft-lbf):60.0
Speed (RPM):1700.          Power Factor:
MOV Temperature (°F):          % Current Loss:          % Torque Loss:
Motor Frame:
#####N
POWER SUPPLY CLOSING DATA
Voltage Loss Calculation Method:Voltage Divider
Voltage Type:AC-3Ø          % Min Voltage:80.0          Min Voltage:368.0
Nominal Voltage:460.0          % Max Voltage:110.0          Max Voltage:506.0
Thermal Overload Resistance:          Starting Resistance:
#####$

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File: C:\IVA\ANC10PST.IVA
 Plant/Unit: NRC VALVE FATIGUE STUDY
 System: post 89-10 nominal case
 Number: E-6162-8-2
 P&ID Coord:
 Valve Manuf: Anchor/Darling
 Type: Gate
 Size: 10
 Oper. Manuf: Limitorque
 Model: SMB/SB/SBD-1
 Number:

Torque Switch Calibration Data: Torque Switch vs Torque
 Source: Limitorque

Table	Row	Torque Switch Setting	Operator Torque	Table	Row	Torque Switch Setting	Operator Torque
	1	1.00	175.0		11	3.50	
	2	1.25	283.0		12	3.75	
	3	1.50	350.0		13	4.00	
	4	1.75	438.0		14	4.25	
	5	2.00	550.0		15		
	6	2.25	638.0		16		
	7	2.50	750.0		17		
	8	2.75	838.0		18		
	9	3.00			19		
	10	3.25			20		

Table	Row	Cable Description	Length (ft)	Cable Size	Max. Temp
	1				
	2				
	3				
	4				
	5				
	6				
	7				
	8				
	9				
	10				
	11				
	12				
	13				
	14				
	15				
	16				
	17				
	18				
	19				
	20				

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=====f
IVA-4.10          MOTOR OPERATED GATE VALVE CLOSING          Date:11-01-1995
CAPABILITY ASSESSMENT          Time: 09:38:08
                                Page: 3 of 3

File:C:\IVA\ANC10PST.IVA
Plant/Unit:NRC VALVE FATIGUE STUDY
System:post 89-10 nominal case
Number:E-6162-8-2          P&ID Coord:
Valve Manuf:Anchor/Darling          Type:Gate          Size:10
Oper. Manuf:Limitorque          Model:SMB/SB/SBD-1          Number:
=====j
CAPABILITIES          REQUIREMENTS          Current Operator Settings
Available Available Available INEL Industry Torque Switch: 3.00
at Max. at Nom. at Min. Estimate Estimate Output Torque: 928.8
Voltage Voltage Voltage f= .449 i= .500 Output Thrust: 35435.
102896. 68031. 43540. 32160. 34457. Stem Thrust
. . 3275. 978. Margin
. . 10.2 2.8 Percent Margin
2696.9 1783.1 1141.2 842.9 903.1 Operator Torque
. . 85.8 25.6 Margin
. . 10.2 2.8 Percent Margin
65.3 54.0 34.6 25.5 27.4 Motor Torque
506.0 460.0 368.0 Voltage @ MOV
506.0 460.0 368.0 Source Voltage
Reference Maximum Minimum Minimum
. 5.26 3.56 2.77 2.93 Torque Switch Setting
=====¥

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Appendix B STEM AND STEM NUT THREAD SHEAR STRESS CALCULATIONS

Appendix B includes MATHCAD (Ref. 23) calculations of shear stress in the stem and stem nut threads for the Walworth and Anchor/Darling valves.

CALCULATION WORK SHEET

Page B-2 of B-9 PagesSubject: Stem and Stem Nut Thread Shear Stress Calculations Date: Oct. 1995Prepared by: W. D. Richins Checked by: M. J. Russell

1) MAXIMUM STRESS IN THE WALWORTH VALVE STEM AND STEM NUT

The following estimates the maximum shear stress in the external threads of the valve stem and the internal threads of the stem nut for the 6" Walworth valve. The formulas from ASME/ANSI B1.1-1989 modified for use with ACME threads are used here. A formula which gives results identical to those listed below for the shear area of the external thread is given in ASME/ANSI B1.5-1988. Note that ANSI B1.1 applies to UN and UNR thread forms with the angle of thread equal to 60 degrees and single start threads. The valve stem and stem nut threads are ACME threads with the angle of thread equal to 29 degrees and multi-start threads. Thus, $\tan 30^\circ$ is replaced by $\tan 14.5^\circ$ and the calculated shear area is multiplied by the number of threads.

For the shear stresses in the threads estimated below, only one complete turn of each of the two threads is assumed to be in contact. Thus, the length of thread engagement is equal to the thread pitch.

Thread type = 29 deg General ACME, Class 2G Assumed

Pitch = 0.25

Lead = 0.5

Stem Diameter = 1.250

$$p := 0.25 \cdot \text{in}$$

$$p = \text{pitch.}$$

$$n := \frac{1}{p}$$

$$n = 4 \cdot \frac{1}{\text{in}}$$

 n = threads per inch for a single start thread.

$$NT := 2$$

NT = number of threads

$$\text{Lead} := NT \cdot p$$

$$\text{Lead} = 0.5 \cdot \text{in}$$

Lead = number of threads advanced for each revolution of the stem

$$D_{\min} := 1.240 \cdot \text{in}$$

 $D_{\min}$ = minimum major diameter of the external thread

$$D_{2\max} := 1.1701 \cdot \text{in}$$

 $D_{2\max}$ = maximum pitch diameter of the internal thread.

$$D_{1\max} := 1.0600 \cdot \text{in}$$

 $D_{1\max}$ = maximum minor diameter of the internal thread

$$D_{2\min} := 1.1210 \cdot \text{in}$$

 $D_{2\min}$ = minimum pitch diameter of the external thread.

$$LE := 0.25 \cdot \text{in}$$

LE = length of thread engagement

CALCULATION WORK SHEET

Page B-3 of B-9 Pages

Subject: Stem and Stem Nut Thread Shear Stress Calculations Date: Oct. 1995

Prepared by: W. D. Richins Checked by: M. J. Russell

AS_s = External Thread Shear Area

$$AS_s := NT \cdot \left[\pi \cdot n \cdot LE \cdot D_{1max} \cdot \left[\frac{1}{2 \cdot n} + \tan(14.5 \cdot \text{deg}) \cdot (D_{2min} - D_{1max}) \right] \right]$$

$$AS_s = 0.938 \cdot \text{in}^2$$

AS_n = Internal Thread Shear Area

$$AS_n := NT \cdot \left[\pi \cdot n \cdot LE \cdot D_{min} \cdot \left[\frac{1}{2 \cdot n} + \tan(14.5 \cdot \text{deg}) \cdot (D_{min} - D_{2max}) \right] \right]$$

$$AS_n = 1.115 \cdot \text{in}^2$$

The following loads for the Walworth valve are from Section 4 of the report.

$$F_{st1} := -11296 \cdot \text{lbf}$$

F_{st1} = Stem thrust load for load case A, valve closing

$$F_{st2} := -19170 \cdot \text{lbf}$$

F_{st2} = Stem thrust load for load case B, valve closing

$$F_{st3} := -23106 \cdot \text{lbf}$$

F_{st1} = Stem thrust load for load case C, valve closing

$$F_{st4} := 3572 \cdot \text{lbf}$$

F_{st4} = Stem thrust load for load case A, valve opening

$$F_{st5} := 6062 \cdot \text{lbf}$$

F_{st5} = Stem thrust load for load case B, valve opening

$$F_{st6} := 7307 \cdot \text{lbf}$$

F_{st6} = Stem thrust load for load case C, valve opening

CALCULATION WORK SHEET

Page B-4 of B-9 Pages

Subject: Stem and Stem Nut Thread Shear Stress Calculations Date: Oct. 1995

Prepared by: W. D. Richins Checked by: M. J. Russell

SHEAR STRESSES IN THE INTERNAL THREADS FOR WALWORTH VALVE:

Shear Stress for Valve Closing Cycle

$$S_{i1} := \frac{F_{st1}}{AS_n} \quad S_{i1} = -10133 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{i1} = \text{Shear Stress for internal threads - load case A}$$

$$S_{i2} := \frac{F_{st2}}{AS_n} \quad S_{i2} = -17197 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{i2} = \text{Shear Stress for internal threads - load case B}$$

$$S_{i3} := \frac{F_{st3}}{AS_n} \quad S_{i3} = -20728 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{i3} = \text{Shear Stress for internal threads - load case C}$$

Shear Stress for Valve Opening Cycle

$$S_{i4} := \frac{F_{st4}}{AS_n} \quad S_{i4} = 3204 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{i4} = \text{Shear Stress for internal threads - load case A}$$

$$S_{i5} := \frac{F_{st5}}{AS_n} \quad S_{i5} = 5438 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{i5} = \text{Shear Stress for internal threads - load case B}$$

$$S_{i6} := \frac{F_{st6}}{AS_n} \quad S_{i6} = 6555 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{i6} = \text{Shear Stress for internal threads - load case C}$$

CALCULATION WORK SHEET

Page B-5 of B-9 Pages

Subject: Stem and Stem Nut Thread Shear Stress Calculations Date: Oct. 1995

Prepared by: W. D. Richins Checked by: M. J. Russell

SHEAR STRESSES IN THE EXTERNAL THREADS FOR WALWORTH VALVE:

Shear Stress for Valve Closing Cycle

$$S_{e1} := \frac{F_{st1}}{AS_s} \quad S_{e1} = -12048 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e1} = \text{Shear Stress for external threads - load case A}$$

$$S_{e2} := \frac{F_{st2}}{AS_s} \quad S_{e2} = -20446 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e2} = \text{Shear Stress for external threads - load case B}$$

$$S_{e3} := \frac{F_{st3}}{AS_s} \quad S_{e3} = -24644 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e3} = \text{Shear Stress for external threads - load case C}$$

Shear Stress for Valve Opening Cycle

$$S_{e4} := \frac{F_{st4}}{AS_s} \quad S_{e4} = 3810 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e4} = \text{Shear Stress for external threads - load case A}$$

$$S_{e5} := \frac{F_{st5}}{AS_s} \quad S_{e5} = 6466 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e5} = \text{Shear Stress for external threads - load case B}$$

$$S_{e6} := \frac{F_{st6}}{AS_s} \quad S_{e6} = 7793 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e6} = \text{Shear Stress for external threads - load case C}$$

CALCULATION WORK SHEET

Page B-6 of B-9 Pages

Subject: Stem and Stem Nut Thread Shear Stress Calculations Date: Oct. 1995

Prepared by: W. D. Richins Checked by: M. J. Russell

2) MAXIMUM STRESS IN THE 10" ANCHOR/DARLING VALVE STEM AND STEM NUT

The following estimates the maximum shear stress in the external threads of the valve stem and the internal threads of the stem nut for the 10" Anchor/Darling valve. The formulas from ASME/ANSI B1.1-1989 modified for use with Stub ACME threads are used here. A formula which gives results identical to those listed below for the shear area of the external thread is given in ASME/ANSI B1.5-1988. Note that ANSI B1.1 applies to UN and UNR thread forms with the angle of thread equal to 60 degrees and single start threads. The valve stem and stem nut threads are Stub ACME threads with the angle of thread equal to 29 degrees and multi-start threads. Thus, $\tan 30^\circ$ is replaced by $\tan 14.5^\circ$ and the calculated shear area is multiplied by the number of threads.

For the shear stresses in the threads estimated below, only one complete turn of each of the three threads is assumed to be in contact. Thus, the length of thread engagement is equal to the thread pitch.

Thread type = 29 deg Stub ACME, Class 2G Assumed

Pitch = 0.3333

Lead = 1.0

Stem Diameter = 2.0

$$p := \frac{1}{3} \cdot \text{in}$$

p = pitch.

$$n := \frac{1}{p}$$

$$n = 3 \cdot \frac{1}{\text{in}}$$

n = threads per inch for a single start thread.

$$NT := 3$$

NT = number of threads

$$\text{Lead} := NT \cdot p$$

$$\text{Lead} = 1 \cdot \text{in}$$

Lead = number of threads advanced for each revolution of the stem

$$D_{\min} := 1.9875 \cdot \text{in}$$

$D_{\min}$ = minimum major diameter of the external thread

$$D_{2\max} := 1.9485 \cdot \text{in}$$

$D_{2\max}$ = maximum pitch diameter of the internal thread.

$$D_{1\max} := 1.8625 \cdot \text{in}$$

$D_{1\max}$ = maximum minor diameter of the internal thread

$$D_{2\min} := 1.8902 \cdot \text{in}$$

$D_{2\min}$ = minimum pitch diameter of the external thread.

$$LE := \frac{1}{3} \cdot \text{in}$$

LE = length of thread engagement

CALCULATION WORK SHEET

Page B-7 of B-9 Pages

Subject: Stem and Stem Nut Thread Shear Stress Calculations Date: Oct. 1995

Prepared by: W. D. Richins Checked by: M. J. Russell

AS_s = External Thread Shear Area

$$AS_s := NT \cdot \left[\pi \cdot n \cdot LE \cdot D_{1max} \cdot \left[\frac{1}{2 \cdot n} + \tan(14.5 \cdot \text{deg}) \cdot (D_{2min} - D_{1max}) \right] \right]$$

$$AS_s = 3.051 \cdot \text{in}^2$$

AS_n = Internal Thread Shear Area

$$AS_n := NT \cdot \left[\pi \cdot n \cdot LE \cdot D_{min} \cdot \left[\frac{1}{2 \cdot n} + \tan(14.5 \cdot \text{deg}) \cdot (D_{min} - D_{2max}) \right] \right]$$

$$AS_n = 3.311 \cdot \text{in}^2$$

The following loads for the Anchor/Darling valve are from Section 4 of the report.

$$F_{st1} := -25516 \cdot \text{lbf}$$

F_{st1} = Stem thrust load for load case A, valve closing

$$F_{st2} := -36756 \cdot \text{lbf}$$

F_{st2} = Stem thrust load for load case B, valve closing

$$F_{st3} := -44250 \cdot \text{lbf}$$

F_{st1} = Stem thrust load for load case C, valve closing

$$F_{st4} := 15638 \cdot \text{lbf}$$

F_{st4} = Stem thrust load for load case A, valve opening

$$F_{st5} := 22526 \cdot \text{lbf}$$

F_{st5} = Stem thrust load for load case B, valve opening

$$F_{st6} := 27119 \cdot \text{lbf}$$

F_{st6} = Stem thrust load for load case C, valve opening

CALCULATION WORK SHEET

Page B-8 of B-9 Pages

Subject: Stem and Stem Nut Thread Shear Stress Calculations Date: Oct. 1995

Prepared by: W. D. Richins Checked by: M. J. Russell

SHEAR STRESSES IN THE INTERNAL THREADS FOR ANCHOR/DARLING VALVE:

Shear Stress for Valve Closing Cycle

$$S_{i1} := \frac{F_{st1}}{AS_n}$$

$$S_{i1} = -7707 \cdot \frac{\text{lbf}}{\text{in}^2}$$

S_{i1} = Shear Stress for internal threads - load case A

$$S_{i2} := \frac{F_{st2}}{AS_n}$$

$$S_{i2} = -11102 \cdot \frac{\text{lbf}}{\text{in}^2}$$

S_{i2} = Shear Stress for internal threads - load case B

$$S_{i3} := \frac{F_{st3}}{AS_n}$$

$$S_{i3} = -13365 \cdot \frac{\text{lbf}}{\text{in}^2}$$

S_{i3} = Shear Stress for internal threads - load case C

Shear Stress for Valve Opening Cycle

$$S_{i4} := \frac{F_{st4}}{AS_n}$$

$$S_{i4} = 4723 \cdot \frac{\text{lbf}}{\text{in}^2}$$

S_{i4} = Shear Stress for internal threads - load case A

$$S_{i5} := \frac{F_{st5}}{AS_n}$$

$$S_{i5} = 6804 \cdot \frac{\text{lbf}}{\text{in}^2}$$

S_{i5} = Shear Stress for internal threads - load case B

$$S_{i6} := \frac{F_{st6}}{AS_n}$$

$$S_{i6} = 8191 \cdot \frac{\text{lbf}}{\text{in}^2}$$

S_{i6} = Shear Stress for internal threads - load case C

CALCULATION WORK SHEET

Page B-9 of B-9 Pages

Subject: Stem and Stem Nut Thread Shear Stress Calculations Date: Oct. 1995

Prepared by: W. D. Richins Checked by: M. J. Russell

SHEAR STRESSES IN THE EXTERNAL THREADS FOR ANCHOR/DARLING VALVE:

Shear Stress for Valve Closing Cycle

$$S_{e1} := \frac{F_{st1}}{AS_s} \quad S_{e1} = -8362 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e1} = \text{Shear Stress for external threads - load case A}$$

$$S_{e2} := \frac{F_{st2}}{AS_s} \quad S_{e2} = -12046 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e2} = \text{Shear Stress for external threads - load case B}$$

$$S_{e3} := \frac{F_{st3}}{AS_s} \quad S_{e3} = -14502 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e3} = \text{Shear Stress for external threads - load case C}$$

Shear Stress for Valve Opening Cycle

$$S_{e4} := \frac{F_{st4}}{AS_s} \quad S_{e4} = 5125 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e4} = \text{Shear Stress for external threads - load case A}$$

$$S_{e5} := \frac{F_{st5}}{AS_s} \quad S_{e5} = 7382 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e5} = \text{Shear Stress for external threads - load case B}$$

$$S_{e6} := \frac{F_{st6}}{AS_s} \quad S_{e6} = 8888 \cdot \frac{\text{lbf}}{\text{in}^2} \quad S_{e6} = \text{Shear Stress for external threads - load case C}$$