NUREG/CR-5406
EGG-2569
Vol. 1

BWR Reactor Water Cleanup System Flexible Wedge Gate Isolation Valve Qualification and High Energy Flow Interruption Test

Analysis and Conclusions

Prepared by K.G. DeWall, R. Steele, Jr.

Idaho National Engineering Laboratory

EG&G Idaho, Inc.

Prepared for
U.S. Nuclear Regulatory Commission

NUREG/CR-5406
EGG-2569
Vol. 1
R1

BWR Reactor Water Cleanup System Flexible Wedge Gate Isolation Valve Qualification and High Energy Flow Interruption Test

Analysis and Conclusions

Manuscript Completed: September 1989
Date Published: October 1989

Prepared by
K.G. DeWall, R. Steele, Jr.

Idaho National Engineering Laboratory
Managed by the U.S. Department of Energy

EG&G Idaho, Inc.
P. O. Box 1625
Idaho Falls, ID 83415

Prepared for
Division of Engineering
Division of Safety Issue Resolution
Office of Nuclear Regulatory Research
U.S. Nuclear Regulatory Commission
Washington, DC 20555
NRC FIN A6857, B5529

ABSTRACT

This report presents the measured data and the analyses performed to date on the full-scale high-energy qualification and flow interruption gate valve testing to develop technical insights for the United States Nuclear Regulatory Commission (USNRC) effort regarding Generic Issue 87 (GI-87). The research was sponsored by the USNRC[a] and conducted by researchers from the Idaho National Engineering Laboratory. We tested two 6-in., 900-lb class valve assemblies, which represent a significant percentage of the reactor water cleanup isolation valves installed in plant applications. These valves were modified before testing by adding a high temperature load cell in the valve stems, which allowed the direct measurement of valve stem thrust during both opening and closing valve cycles. Instrumentation installed in the flow loop and on the valve assemblies measured the important valve and system test responses. Additionally, during the test program, all of the currently popular motor operated valve diagnostic test systems monitored the performance of the valves. Initially the valves were subjected to the hydraulic and leakage qualification tests defined in ANSI B16.41 and then to flow interruption and reopening valve tests at boiling water reactor primary system water temperature and pressure conditions with downstream line break flows. For the two valves tested, results show that (a) the disc factor used in current industry motor operator sizing equations underpredicts actual valve thrust requirements at all high temperature loadings, and for one valve design the equations may require an additional term to account for nonlinear performance, (b) the thrusts required to close the valves were sensitive to the fluid temperature, and (c) the results of testing at lower pressures, temperatures, and flows cannot be extrapolated to design basis pressures, temperatures, and flows for valve designs that have not exhibited linear performance behavior during design basis prototypical testing.

a. Work supported by the U.S. Nuclear Regulatory Commission, Division of Engineering, Office of Nuclear Regulatory Research, under DOE Contract No. DE-AC07-76ID01570.

A6857—Equipment Operability
B5529—Failure of HPCI Steam Line Without Isolation (GI-87)

EXECUTIVE SUMMARY

Recent testing sponsored by the U.S. Nuclear Regulatory Commission (USNRC) showed that for at least some gate valves installed in nuclear applications, the equations used by industry to size the valve operators do not conservatively calculate the thrust needed to close the valves under design basis loadings. The tests also showed that the results of in–situ valve testing at lower loadings cannot be extrapolated to design basis loadings. The testing was conducted by researchers from the Idaho National Engineering Laboratory (INEL) to provide technical data for the USNRC effort regarding Generic Issue 87 (GI–87) "Failure of HPCI Steam Line Without Isolation." The test program also provides information applicable to Generic Issue II.E.6.1, "In–situ Testing of Valves" and a related document, IE Bulletin 85–03, "Motor Operated Valve Common Mode Failures During Plant Transient Due to Improper Switch Settings."

Of the three boiling water reactor (BWR) process lines covered under GI–87, an unisolated break in the reactor water cleanup (RWCU) supply line was selected for the first phase of testing because such a break would have the greatest safety impact. The high pressure coolant injection steam supply line and the reactor core isolation cooling steam supply line will be addressed in subsequent research efforts. All three GI–87 process lines have common features. All communicate with the primary system, pass through containment, and have normally open isolation valves.

IE Bulletin 85–03 required the utilities to develop and implement a program that would ensure that the switch settings on selected safety related motor operated valves (MOVs) are chosen, set, and maintained correctly to accommodate the maximum differential pressures expected on these valves during both normal and abnormal events within the design basis. It is also understood that the USNRC issued Generic Letter No. 89–10 "Safety Related Motor–Operated Valve Testing and Surveillance," which will expand the coverage of IE Bulletin 85–03 to a larger number of safety related valves in the plants. To meet these new valve operating criteria, industry developed new MOV diagnostic test equipment and methods for in–situ valve testing. IE Bulletin 85–03 succeeded in significantly improving the operability of the selected safety related valves because it caused many of the utilities to reanalyze the design basis load for the applicable MOVs and to reset the control switches accordingly. In many cases, these analyses were more complete than the analyses in the original procurement, and the utilities reset the control switches in accordance with the improved analyses.

However, very little design basis testing of valves has been conducted outside the plant to verify the analytic assumptions used to determine valve switch settings. Analytic assumptions are necessary because in many cases the utility cannot test valves at design basis loadings in situ. The GI–87 testing provides some of the first measured valve responses with which industry's valve operator sizing equations can be compared.

In this initial test program, two representative RWCU isolation valves were subjected to the hydraulic qualification tests described in ANSI B16.41, the nuclear valves qualification standard, and then to full flow RWCU pipe break flow interruption tests. In all, fourteen flow interruption tests were performed, ten on Valve A and four on Valve B. In the Valve A tests, the parametric study included varying both the degree of inlet water subcooling and the pressure. Break flows were maintained throughout the 30–second valve closure. The four Valve B tests were all performed at a normal BWR 10°F subcooling, and only the inlet pressure was varied. The test loop and valves were instrumented to determine the valve response to flow, including a load cell installed in the valve stems to measure thrust.

Test results show that for both valve designs tested, the force required to open and close the valves at temperatures above 100°F were significantly higher than the force predicted by the valve manufacturers. Only in the room temperature valve opening tests without flow did the typical industry valve thrust equation predict the valve response. Industry has also assumed that for valve opening thrust requirements, the highest load would be when the disc lifted off the seat. This was also determined for the valves tested not to be true. The highest opening loads with flow occurred at different degrees of opening for both valves, but in both cases they were well off their respective seats when the maximum thrust was measured. Valve closing thrust at full line break flows were higher than anticipated. One of the valves exceeded the pretest calculated closing thrust by one third.

The test results provide evidence for two concerns with MOVs in nuclear power plants. First, proper sizing of motor operators is complicated by the fact that the equation used for calculating the stem force needed to close or open a gate valve does not have terms for temperature, degree of fluid subcooling, internal valve clearances, and the differences in the opening and closing forces not accounted for by the stem rejection term. Second, effective in–situ testing is very difficult because (a) the tests cannot be conducted at design basis conditions and (b) even with the valve loadings properly quantified during the in–situ tests, the results cannot be extrapolated to design basis conditions because the final thrust varies depending on the extent to which disc friction rather than disc seating causes the torque switch to be compressed to torque switch trip and because the stem factor varies with the load imposed during valve operation.

The disc factor of 0.3 typically used in industry to calculate disc friction force is not conservative for either of the valves tested. A disc factor of 0.5 marginally predicts the forces for one valve during both opening and closing. The response of the other valve is enveloped by the 0.5 disc factor during opening but not during closing. Today's tools for analyzing valve response to fluid loadings are not sophisticated enough to detect small design differences that make large response differences.

Temperature also affects the thrust requirements of these gate valves. These facts justify continued qualification testing of prototypical valves at design basis loadings and point out the need for industry to modify the variables in the sizing equation. It may be necessary to add new terms to the equation or to increase the disc factor to a very conservative number to account for the missing terms.

When tests have determined the thrust needed to operate a valve at its design basis loading, utilities can use one of several modern diagnostic systems to conservatively set the motor operator control switches. Industry will have to account for the varying stem factor and for the excessively high thrusts resulting from seat–induced torque switch trips that occur with valve operation with low flow or no flow. However, this method may exceed the allowable thrust on some valve designs. This job will be easier and the result more conservative if both the valve torque and thrust can be measured when the switches are set. If further research proves that there is a proportional relationship between stem load and stem factor, the degree of conservatism can be reduced.

The stem factor is a calculation made to predict the efficiency of the motor operator torque to stem thrust conversion. Until recently industry has always considered the stem factor a constant. Procedures used by two of the more popular in–situ valve diagnostic test systems are based on this premise. Test results show that the stem factor changes with stem load, thus making it very difficult to extrapolate normal in–situ valve testing to design basis conditions.

CONTENTS

FIGURES

TABLES

ACKNOWLEDGMENTS

The authors wish to acknowledge the following companies for their assistance and cooperation in this research effort. Anchor–Darling Valve Company, Velan Incorporated, and Limitorque Corporation contributed to the testing by providing representative test hardware at significantly reduced cost and schedule and information concerning valve motor operator sizing procedures. Liberty Technology, Limitorque Corporation, MOVATS Incorporated, and WYLE Laboratories contributed by providing valve assembly diagnostic support representing typical in–plant test capabilities. Three nuclear utilities, Philadelphia Electric Company, Duke Power Company, and Toledo Edison Company, provided information concerning in–situ equipment configurations and test capabilities. The valve, motor operator, and diagnostic equipment manufacturers and utility representatives all cooperated in the early test planning, critically reviewed test procedures and data analysis and participated in NRC sponsored test results review meetings.

The views expressed in this report are the authors' and may not be the opinion of the valve, motor operator, and/or diagnostic equipment manufacturers who cooperated in this effort.

BWR REACTOR WATER CLEANUP SYSTEM FLEXIBLE WEDGE GATE ISOLATION VALVE QUALIFICATION AND HIGH ENERGY FLOW INTERRUPTION TEST VOLUME I ANALYSIS AND CONCLUSIONS

1. INTRODUCTION

The Idaho National Engineering Laboratory (INEL), under the sponsorship of the United States Nuclear Regulatory Commission (USNRC), is performing research to provide technical input for the resolution of specific generic issues and to provide information to develop and improve industry mechanical equipment qualification and operating and maintenance standards. This overall research effort includes a program that tested the operability (opening and closing) of two full-scale motor-operated gate valves typical of those installed in boiling water reactor (BWR) reactor water cleanup (RWCU) process lines for containment isolation. The valves were parametrically tested at, above, and below the pressure, temperature, and flow conditions of a worse-case downstream pipe break in the RWCU supply line outside containment. The purpose of the test program was to provide technical input for the USNRC effort regarding Generic Issue 87 (GI-87), "Failure of the HPCI Steam Line Without Isolation." The test program also provides information applicable to the motor-operated valve portion of another highly visible generic safety issue, Generic Issue II.E.6.1 (GI-II.E.6.1), "In-situ Testing of Valves," and its related documents, IE Bulletin 85-03, "Motor Operated Valve Common Mode Failures During Plant Transients Due to Improper Switch Settings," and Generic Letter No. 89-10 "Safety-Related Motor Operated Valve Testing and Surveillance," which expands many of the IE Bulletin 85-03 requirements to other safety-related motor-operated valve testing and surveillance.

The analyses performed to date on the measured data obtained during the first phase of the GI-87 valve test program and conclusions are discussed in Volume I of this report. Volume II contains the measured data taken in the more significant test sequences of the test program. The data is also available in IBM PC compatible format, for those who wish to analyze the data, and can be obtained through our DOE Technology Transfer Office, at (208) 526-8318. Volume III is a review of the BWR containment isolation valve designs and piping configurations, qualification methods, and previous research

1.1 Background

GI-87 applies to the BWR process lines that communicate with the primary system, pass through containment, and contain normally open isolation valves. Two steam supply lines, the high-pressure coolant injection (HPCI) and the reactor core isolation cooling (RCIC) lines, and one hot water supply line, the RWCU line, meet these criteria. GI-87 addresses whether the isolation valves in these lines will close in the event of a downstream pipe break outside containment.

a. Mention of specific products and/or manufacturers in this document implies neither endorsement or preference nor disapproval by U.S. government, any of its agencies, or EG&G Idaho, Inc., of the use of a specific product for any purpose.

The project began with a review of the valves installed in these applications (see Appendix B, Volume III), specifically their sizes, manufacturers, previous testing, and risk significance. A survey identified the flexible wedge carbon steel gate valve with a Limitorque[a] motor operator as the predominate valve in the three systems (HPCI, RCIC, and RWCU) addressed by GI–87. The most common valve size for the RCIC system is 4 in., 6 in. for the RWCU system, and 10 in. for the HPCI system. Valve manufacturer Anchor–Darling had the largest share of the installed valves, followed by Velan, Crane, Powell, and Walworth (with about equal shares). It was also determined that a downstream break in the RWCU system would represent the highest risk to the plant, so it was decided that the initial flow isolation testing should provide information on valve operability questions associated with the RWCU environment.

To avoid duplication, we reviewed previous applicable test programs. The reviewed test programs included the Electrical Power Research Institute (EPRI) power–operated relief valve/block valve testing at Duke Power in 1980. This program had three shortcomings: (a) the block valves were stainless steel as opposed to carbon steel; (b) the tests were go/no–go type tests where neither motor–operated valve thrust nor torque were measured; and (c) the EPRI test medium was steam, which would be more applicable to the HPCI and RCIC systems than to the RWCU system. Kraftwerkunion (KWU) of West Germany had tested a 3–in. stainless steel parallel disc gate valve at blowdown flows for the Central Electric Generating Board (CEGB), United Kingdom. Mechanical interference on the downstream disc prevented closure. Kraftwerkunion has also performed full flow interruption testing on a large number of valve types; however, our initial contacts indicated that the information is proprietary. Since that time, Bechtel and KWU have formed an alliance and have indicated that the information may become more available in the future.

The results from the survey of previously completed work determined that adequate and sufficient technical test information was not available for the USNRC effort on GI–87 and that additional testing was required.

The USNRC effort regarding IE Bulletin 85–03 required the utilities to develop and implement a program that would ensure that switch settings on selected safety–related motor–operated valves (MOVs) are chosen, set, and maintained correctly to accommodate the maximum differential pressures expected on these valves during both normal and abnormal events within the design basis. The follow–on generic letter will expand IE Bulletin 85–03 requirements to a large number of safety–related MOVs, including those that may be mispositioned.

New MOV diagnostic test equipment and methods for in–situ valve testing have been developed to meet these new operating criteria. One of the new requirements was that the valve control switches be set correctly for the design basis loading for each valve. However, very little design basis testing of valves has been conducted outside the plants, and in many cases the utilities could not test the valves at in–situ design basis conditions. This situation left the utilities relying on valve motor operator switch settings that were based on analyses of the design basis loadings. Utilities typically verified the torque or thrust levels for each valve through seat or back seat type loadings, with very low hydraulic loadings. To determine if one could extrapolate the results of the testing performed at typical in–situ test conditions to design basis conditions, we invited the manufacturers of the more widely used valve diagnostic test systems to join us in the GI–87 test program. The insights gained from this testing would be applicable to both GI–87 and IE Bulletin 85–03.

1.2 Motor Operator Sizing

The gate valve is a high recovery positive shut–off valve and is used in systems where minimal pressure drop is desired when the valve is open. It is ideally suited to situations where isolation of one part of a system from another is required and control of the dynamic properties of the fluid (throttling) is unnecessary. When the disc (or gate) is in the open position, the run of the valve is free of any obstruction with approximately the same head loss as in the adjacent piping. When the disc is lowered into the seat, the upstream pressure forces it against the seat, creating a seal and isolating the downstream system from the upstream fluid.

Figure 1, a cutaway drawing of a typical motor–operated gate valve, shows the components important to this discussion. The forces needed to close the valve and isolate flow must overcome the resistance imposed by three loads: (a) the disc frictional drag load caused by the differential pressure across the disc as the valve closes, (b) the stem rejection load caused by static pressure on the stem, and (c) the packing drag load. Industry has developed a set of equations for use in sizing motor operators. The first equation in this set predicts the total stem force, as detailed below. Each manufacturer modifies the variables in the equation slightly; however, in the long run the application of the equation is the same.

$$F_t = \mu_d A_d \Delta P \pm A_s P + F_p \tag{1}$$

where

F_t = total stem force

Dynamic component:

μ_d = disc factor

A_d = disc area

ΔP = differential pressure

Static component:

A_s = stem cross–sectional area

P = stem presssure

F_p = packing drag load (a constant).

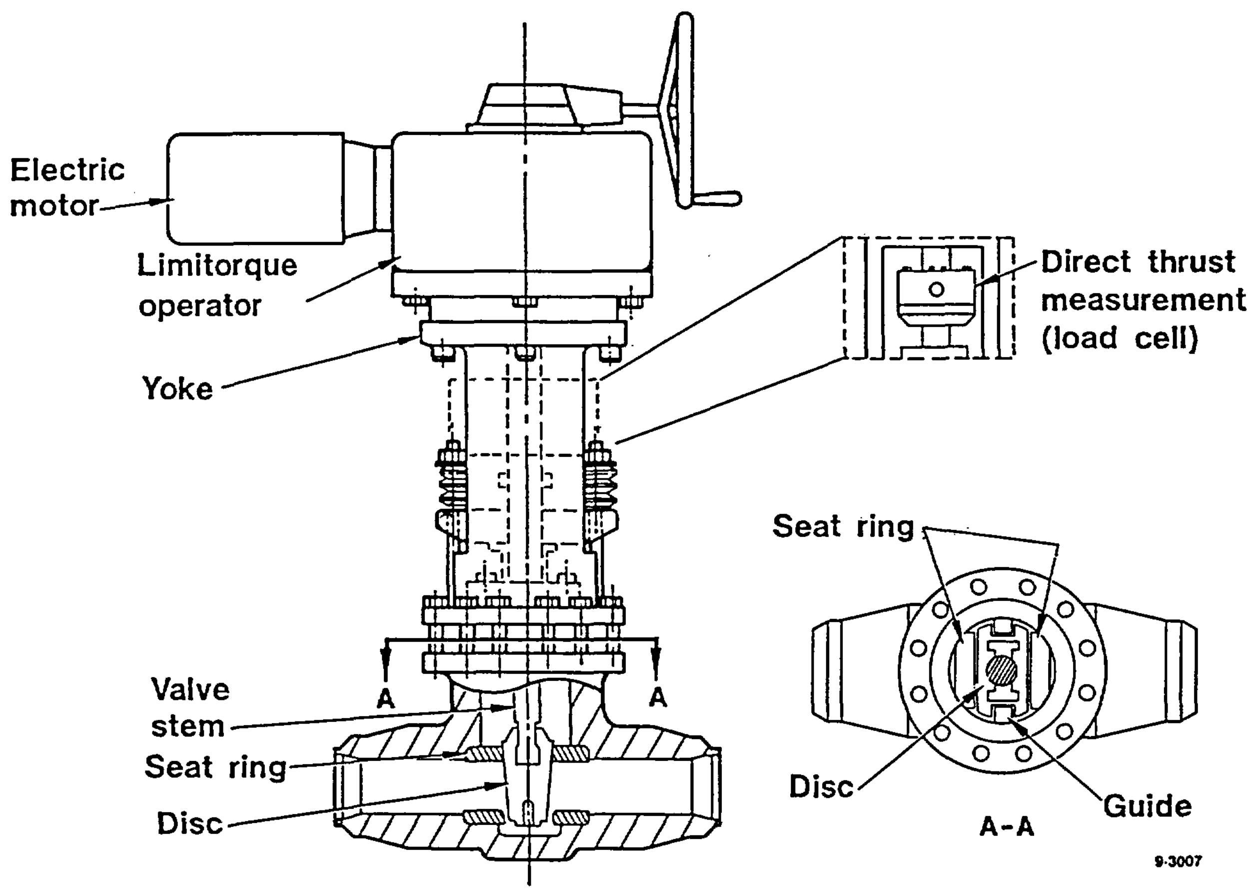

Figure 1. Typical motor–operated valve, similar to the two valves used, which were modified by installing a load cell in each valve stem.

The disc factor (μ_d) normally used for wedge–type gate valves in Equation (1) is 0.3. Note that in this equation the stem rejection load can be either positive or negative depending on whether the valve is closing or opening. This is because the stem rejection load is always in a direction out of the valve body; this load resists valve closure and assists in opening the valve. The packing load is a constant that depends on the packing design, gland nut torque, and direction of operation. The equation is shown divided into two components, which will be referred to in the analysis found later in this report: (a) the dynamic component, which includes the disc load due to differential pressure and (b) the static component, which is the sum of the stem rejection and packing drag loads. The pressure values (P and ΔP) used in the force equation are supplied to the valve manufacturer by each individual plant.

Motor operators control output torque, not valve stem thrust. Thus, in sizing the operator and determining the torque switch setting for motor–operated gate valves, one must consider the conversion of operator output torque to valve stem thrust. This conversion of torque to thrust is one of the equations in the set of motor operator sizing equations. The torque to thrust relationship normally used in sizing motor operators depends on a stem factor calculation, given by

$$T = \mu_s F_t \tag{2}$$

where

$T =$ operator torque

$\mu_s =$ stem factor

$F_t =$ total stem force [from Equation (1)].

The stem factor used in Equation (2) is a function of stem diameter, thread pitch and lead, and the coefficient of friction between the operator stem nut and the valve stem. As in Equation (1), the only variable that cannot be measured in the stem factor equation is the coefficient of friction. Most in industry use a 0.15 or 0.20 coefficient of friction for this parameter. Normally it is assumed that only damage and lubrication of the stem/stem nut threads can significantly alter the stem coefficient of friction. Limitorque personnel, in their diagnostic work, have measured coefficients of friction from 0.10 to 0.20 in actual operation. Losses internal to the motor operator, up to the capacity of the electric motor, will typically be accounted for by the torque spring/switch position. Losses in the stem factor will not be accounted for by the motor operator.

2. TEST OBJECTIVES

As discussed previously, the gate valve qualification and flow interruption testing was performed to provide information to assist in resolving the uncertainties in gate valve operator sizing and torque switch setting. Specific objectives of the testing included the following:

1. Determine the valve stem force required to close a typical RWCU gate valve at typical operating test conditions and under full flow blowdown conditions.

2. Compare valve closing load to opening load at various system operating conditions.

3. Measure valve closure force components such as disc drag, stem rejection, and packing drag loads.

4. Make limited assessments of the effect of temperature, pressure, and valve design on valve closing and opening loads.

5. Evaluate the validity of using present industry standard equations for determining valve stem force.

6. Provide detailed technical information for the above steps to assist in the USNRC effort regarding GI–87.

An additional goal of the INEL testing was to provide information so that specific guidelines might be developed to improve valve qualification and operating and maintenance standards.

3. APPROACH TO TESTING

3.1 Test Design

Two full-scale, representative nuclear valve assemblies were cycled under various design conditions and design basis RWCU pipe break conditions. The valves were manufactured for this test program by Anchor–Darling Valve Company (Valve A) and Velan Incorporated (Valve B), using nuclear design and materials, without third–party inspections. Both valves were modified to incorporate extended yokes (4 in. longer than normal) and the stems were cut in half and threaded to allow installation of a special stem force measurement device. Flanges and safe ends were welded to both sides of each valve for mating with the test system piping.

The first test specimen, Valve A, was a 6–in., 900–lb standard rated, cast steel, flexible–wedge gate valve with a pressure seal bonnet and butt weld ends. The valve seats were hard faced with Stellite and seal–welded to the valve body. The one–piece flexible wedge (disc) was also hard faced with Stellite on the seating faces. The disc guides were carbon steel. The valve was powered by an oversized Limitorque SMB–2–40 electric motor operator. The basic valve design, without the oversized operator, is representative of 40% of the isolation valves installed in BWR RWCU systems.

The second test specimen, Valve B, was a 6–in., 900–lb standard rated, forged steel, flexible–wedge gate valve with a bolted bonnet and butt weld ends. The valve seats were hard faced with Stellite and seal–welded to the valve body. The one–piece flexible wedge (disc) was also hard faced with Stellite on the seating faces. The valve was powered by a Limitorque SMB–0–25 electric motor operator. Representing one of the newer valve assemblies delivered since 1970, the Valve B design incorporated hardfaced disc guide wear surfaces.

Both valves utilized 460–Vac, 3–phase, 60–Hz electric motor operators. To ensure valve closure and data collection at the anticipated greater–than–normal loadings, Valve A utilized a larger, greater–capacity motor operator than would normally be used. The motor operator used with Valve B was sized in accordance with current practices to represent a typical MOV assembly used in nuclear power plants today. Because of their differences in internal design and friction bearing surface design, the two valve assemblies represented a large number of the MOVs used in nuclear power plants today.

The test system used for the subcooled water qualification and flow interruption testing featured a large water tank, heated and pressurized so that various thermal hydraulic conditions could be established and regulated, replicating actual BWR conditions. The water was propelled by high–pressure gaseous nitrogen during the high energy flow interruption testing. The water heating system consisted of a heating section and a high–pressure, high–temperature water pump. The heating section contained an electrical heater, which heated the water as the pump recirculated water from the pressure vessel, through the test section and test valve, and back to the pressure vessel. The test section was a 6–in. pipe with the test specimen mounting flanges and appropriate fittings for obtaining temperature and pressure measurements. The test system also featured a fast–acting (approximately 300–msec opening stroke), hydraulically operated valve, positioned so that when the valve was actuated, the system's fluid was abruptly dumped to the atmosphere, resulting in high–flow (blowdown) conditions through the test specimen. The system is shown schematically in Figure 2.

To accomplish the functional testing, the system contained bleed valves, which provided the means to reduce system pressure on both sides (upstream and downstream) of the test specimen. In this manner, differential pressure conditions could be established across the test valve's disc.

The test system was instrumented to monitor flow, pressure, and temperature at various locations, including the test valve upstream and downstream positions. Motor operator electrical characteristics were also recorded. Valve stem force was monitored using the previously described high–temperature load cell installed between the two halves of the valve stems. The test parameters measured are listed in Table 1.

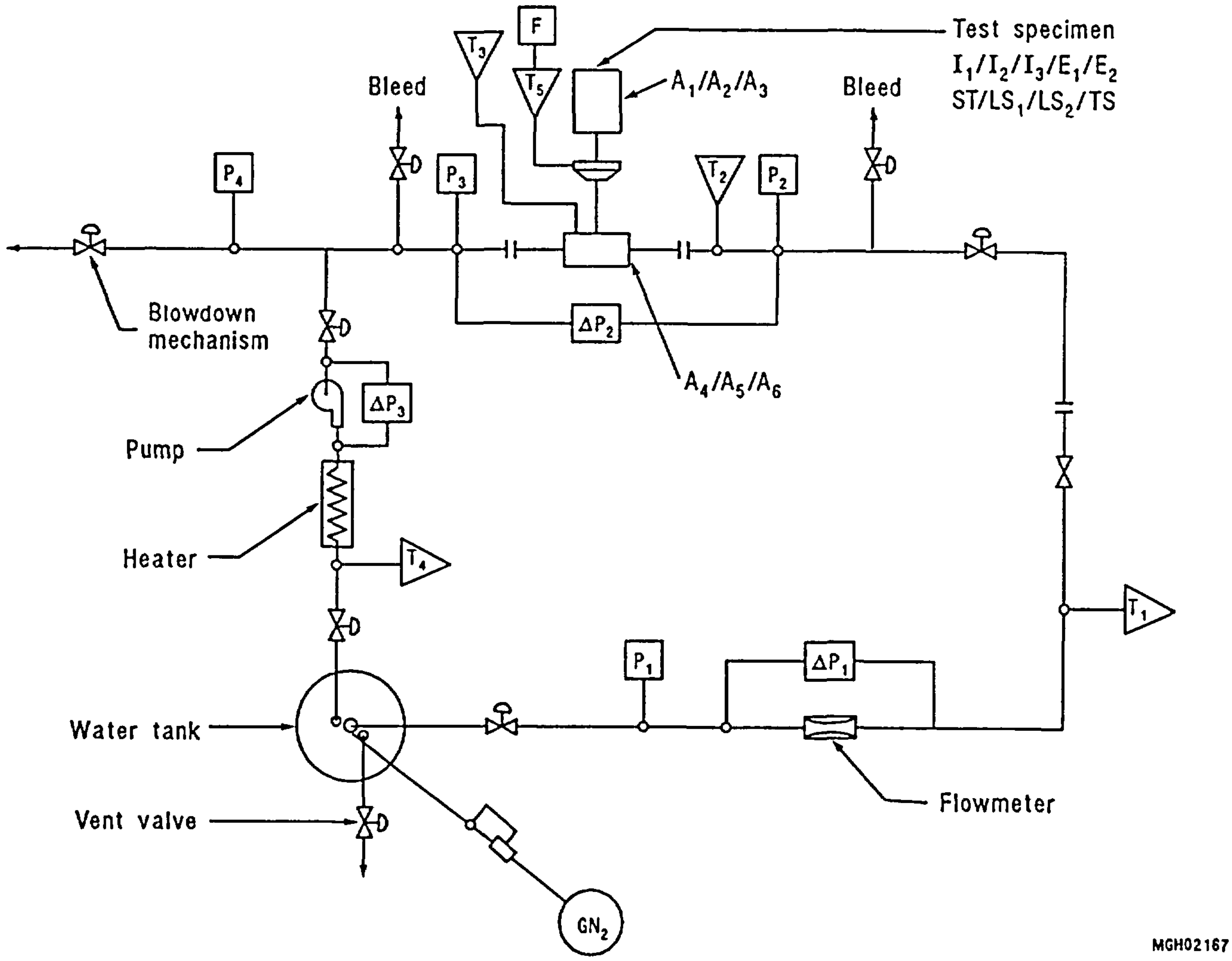

Figure 2. Instrumentation installed in the test loop to monitor temperature (T), pressure (P), pressure differential ΔP, stem force, (F), and flow; motor current and voltage, valve stem position, and other important variables were also measured.

Table 1. Test parameters measured during blowdown tests

Transducer	Measurement	FM Tape	Oscillograph	Data Logger	X–Y Plotter
T1	System water temperature			X	
T2	Test valve inlet water temperature			X	
T3	Test valve body temperature			X	
T4	Heating section water temperature			X	
T5	Load cell temperature			X	
P1	System water pressure	X	X	X	X
P2	Test valve inlet water pressure	X	X	X	
P3	Test valve outlet water pressure	X	X	X	
P4	Discharge section water pressure	X	X	X	
ΔP1	Test valve differential pressure	X	X	X	
ΔP2	Venturi differential pressure	X	X	X	
ΔP3	Pump differential pressure	X		X	
I1	Actuator current	X	X		
I2	Actuator current	X			
I3	Actuator current	X			
E1	Actuator voltage	X	X		
E2	Actuator voltage	X			
ST	Valve stroke–LVDT	X	X		X
LS1	Open limit switch	X	X		
LS2	Close limit switch[a]				
TS	Close torque switch	X	X		
F	Valve stem force	X	X	X	
A1	Acutator acceleration Y	X	X		
A2	Actuator acceleration X	X			
A3	Actuator acceleration Z	X			
A4	Valve body acceleration Y	X	X		
A5	Valve body acceleeration X	X			
A6	Valve body acceleration Z	X			

a. Control room light indicator only.

A secondary objective of the qualification and flow interruption test program was to determine if normal utility in-situ valve testing, using available diagnostic equipment, could be extrapolated to provide assurance of a valve's operability at design basis loadings. Several MOV diagnostic system manufacturers supported this objective by participating in the testing, as listed in Table 2. The manufacturer participation was not a competition but, rather, an attempt to determine what factors need to be considered to provide reasonable assurance of valve operability using each of the diagnostic systems.

3.2 Test Procedure

Upon installation in the test system, each valve assembly was subjected to a typical ANSI B16.41[1] functional qualification test, including the valve leakage test (Annex A), cold cyclic test (Annex B), and hot cyclic test (Annex C). These tests provided a baseline characterization of the valve assembly operation for comparison with the information obtained from the later testing. The valve leakage test established the mainseat valve leakage rate and the packing leakage rate of the test valves, while the cold cyclic test demonstrated the capability of the test valve assembly to open and close under adverse combinations of motive power and system pressure with the assembly at room temperature, not exceeding 100°F. The hot cyclic test sequence was performed to demonstrate the capability of the test valve assemblies to open and close under adverse combinations of motive power and system pressure with the assembly at design temperature, in excess of 100°F. Annex G, flow isolation, was the subject of this test program and thus was not performed as part of the pretest qualification series. Table 3 lists the valve cycles performed during both the qualification tests and the subsequent blowdown test series.

Once baseline qualification testing of each test valve assembly was completed, several test series were performed to address the questions of GI-87. Each test series included leakage tests, cyclic tests without flow, cyclic tests at normal system flow, and cyclic tests at full line break flow conditions. A wide range of design upstream pressures and temperatures were maintained throughout the valve closures, with line break flow limited only by flashing and choked flow in the test loop. Required nonflow data were collected during the preparation period for full-scale flow and postfull-scale flow tests.

Fourteen line break flow tests (see Table 2) were accomplished, ten on Valve A and four on Valve B. The ten tests on Valve A with the oversized operator included a parametric study in both pressure and the degree of water subcooling. Pressures were varied from 600 to 1400 psig valve inlet pressure and the coolant temperatures ranged from 10 to 130°F subcooled. The four tests performed on Valve B with the normal sized operator were performed to demonstrate expected in-service performance with the operator motive power closer to normal. In these tests the pressure was varied from 600 to 1400 psig at a constant 10°F subcooling.

Table 2. Valve diagnostic equipment used for subcooled blowdown tests

Valve	Test Series	Description	Diagnostic Equipment[a]
A	1	Qualification test	MCSA
A	3	Blowdown, 1000 psig, 480°F	MCSA
A	2	Blowdown, 1000 psig, 530°F	None
A	4	Blowdown, 1000 psig, 400°F	V-MODS
A	6	Blowdown, 1400 psig, 530°F	V-MODS
A	5	Blowdown, 1400 psig, 580°F	MOVATS
A	7	Blowdown, 1400 psig, 450°F	MOVATS
A	9	Blowdown, 600 psig, 430°F	None
A	8	Blowdown, 600 psig, 480°F	None
A	10	Blowdown, 600 psig, 350°F	MAC, VOTES
A	11	Blowdown, 1000 psig, 530°F	MCSA
B	1	Qualification test	MCSA
B	2	Blowdown, 1000 psig, 530°F	MCSA, MOVATS
B	3	Blowdown, 1400 psig, 580°F	V-MODS
B	4	Blowdown, 600 psig, 480°F	MAC
B	5	Blowdown, 1000 psig, 530°F	V-MODS

a. MAC — Limitorque Motor Actuator Characterizer
MCSA — ORNL Motor Current Signature Analysis
MOVATS — MOVATS, Inc. (MOV Analysis and Test System)
V-MODS — WYLE Laboratories Valve Motor Operator Diagnostic System
VOTES — Liberty Technology Valve Operator Test & Evaluation System

Table 3. Test step matrix for qualification and blowdown tests

Step Number	Valve Cycle	Description
Qualification tests (Test 1, Valves A and B)		
1	O → C	Close valve at 2220 psig, cold
2		Seat leakage test
3	C → pt O	ΔP opening at 1700 psig, cold
4		Packing leakage test
5	pt O → C → O	Cycle valve at 0 psig, cold
6	O → C	Close valve at 2220 psig, cold
7	C → O	ΔP opening at 1700 psig, cold
8	O → C → O	Cycle valve at 0 psig, cold
9	O → C → O	Cycle valve at 0 psig, cold
10	O → C → O	Cycle valve at 0 psig, cold
11	O → C	Close valve at 2220 psig, cold
12	C → O	ΔP opening at 1700 psig, cold
13	O → C	Close valve at 2220 psig, cold
14	C → O	ΔP opening at 1700 psig, cold
15	O → C	Close valve at 2220 psig, cold
16	C → O	ΔP opening at 1700 psig, cold
17	O → C → O	Cycle valve at 0 psig, cold
18	O → C	Close valve at 1650 psig, 600°F
19	C → O	ΔP opening at 1650 psig, 600°F
20	O → C	Close valve at 1650 psig, 600°F
21	C → O	ΔP opening at 1650 psig, 600°F
22	O → C	Close valve at 1650 psig, 600°F
23	C → O	ΔP opening at 1650 psig, 600°F
24	O → C	Close valve at 1650 psig, 600°F
25	C → O	ΔP opening at 1650 psig, 600°F
26	O → C	close valve at 1650 psig, 600°F

Table 3. (continued)

Step Number	Valve Cycle	Description
Blowdown tests (Valve A, Tests 2–11, and Valve B, Tests 2–5)		
0	O → C → O	Cycle valve at test pressure and temperature
1	O → C	Close valve at test pressure and temperature
2	C → O	ΔP opening at test pressure and temperature
3	O → C	Close valve at 100 gpm, test pressure and temperature
4	C → O	Open valve at 100 gpm, test pressure and temperature
5	O → C	Blowdown at test pressure and temperature
6	C → pt O	Blowdown at test pressure and temperature
7	pt O → C	Blowdown at test pressure and temperature
8	C → O	Open valve at 0 psig and test temperature
9	O → C → O	Cycle valve at 0 psig and test temperature
Final cold testing (Valve A, Test 11, and Valve B, Test 5)		
10	C → O	ΔP opening at 1000 psig, cold
11	O → C	Close valve at 1000 psig, cold
12		Seat leakage test
13	C → pt O	ΔP opening at 1000 psig, cold
14		Packing leakage test
15	pt O → O	Open valve at 1000 psig, cold
16	O → C	Close valve at 1000 psig, cold
17	C → O	ΔP opening at 1000 psig, cold
18	O → C	Close valve at 0 psig, cold
19	C → O	Open valve at 0 psig, cold
20	O → C	Close valve at 0 psig, cold
21	C → O	Open valve at 0 psig, cold
22	O → C	Close valve at 0 psig, cold
23	C → O	Open valve at 0 psig, cold

O open
C close
pt partial valve stroke

4. TEST RESULTS AND INTERPRETATION

A torque switch setting of 2.0 was selected for the Valve A motor operator so that the stem thrust capability was maximized without exceeding the valve and instrumentation capacity. (The torque switch was reset to 2.5 after test 10 to compensate for an observed torque–out anomaly, discussed later.) Valve A closed satisfactorily during all tests; however, the measured stem loads were significantly higher than the stem loads predicted by the valve manufacturer.

A torque switch setting of 1.75 was selected for the Valve B assembly to provide the needed closure thrust for the given test conditions. However, this setting resulted in delivered stem thrust (as determined by the stem–mounted load cell) below that specified by the valve manufacturer for the highest pressure flow interruption test. Therefore, the torque switch setting was raised to 2.0 before the first flow interruption test. Valve B performed satisfactorily during the lower pressure testing; however, during the 1400 psig test, the operator torqued out before the disc reached the fully closed position (1/4 in. of travel remaining). During this test the valve had closed far enough to produce a seal, with no leakage observed. Higher–than–predicted stem loads during flow isolation and a reduction in delivered stem thrust accounted for the valve not completely closing at its design basis loading.

4.1 Data Analyses

Both valves exhibited higher opening and closing forces at normal operating temperature than would have been predicted using Equation (1) and the 0.3 disc factor typically used in the past by industry. Examples of these higher forces for both valves are shown in Figures 3 through 6. The predicted forces in these figures were calculated using both a 0.3 and a 0.5 disc factor and the actual measured pressures, exposed disc areas, packing drag loads, and stem rejection loads throughout the opening and closing cycles. We used this calculational technique and these plots to look for any deviations from the predictions, and (if they occurred) to determine at what specific point in the opening or closing cycle they occurred. The opening and closing cycles shown in Figures 3 through 6 are all at normal BWR operating temperatures, pressures, and line break flows.

The measured stem force history shown in Figure 3 for the Valve A opening under high flow starts out with the valve closed and the valve stem in compression. We see a decrease in stem compression as the opening cycle begins, and then the stem goes into tension due to the operator hammer blow. The stem force history, for the remainder of the cycle, reflects the combined effects of the disc drag load, the stem rejection (assists opening), and the packing drag load. Figure 5 illustrates this same comparison of actual measurements to calculated values during Valve A closure. The figure shows the measured stem compression (negative values) increasing as the valve closes, until the compression reaches a peak when the flow path is finally blocked. Then the stem compression decreases to a value representing the force required to slide the disc on the full seat ring to the final seating position. Finally, the measured force increases sharply through torque switch trip to the final stem compressive load (at approximately 40,000 lb, not shown in Figure 5). This additional stem force beyond torque switch trip is due to the circuit dropout time and the momentum of the operator motor.

Valve B's measured forces, shown in Figures 4 and 6, follow the general shape of the calculated forces quite well, and we describe this performance as linear. We describe the performance of Valve A, which did not follow the shape of the calculations, as nonlinear. For both valves the measured opening forces are bounded by the 0.5 disc factor calculation; however, they occur much later in the valve cycle than is predicted. The measured closing forces for Valve A are not bounded by either calculation and they do not follow the shape of the curves.

Because of its larger port and stem size and a packing design with greater drag, Valve B would be expected to need about a three percent higher force during closing than Valve A. A comparison of Figures 5 and 6 shows this to be true for the first half of the closing stroke and during the seating period at the end of the valve cycle. However, during the last half of the valve's cycle, down to flow isolation, Valve A required much higher forces than Valve B. Even though both valves were 6–in., 900–lb class, flexible wedge gate valves, they responded to similar thermal hydraulic loadings quite differently.

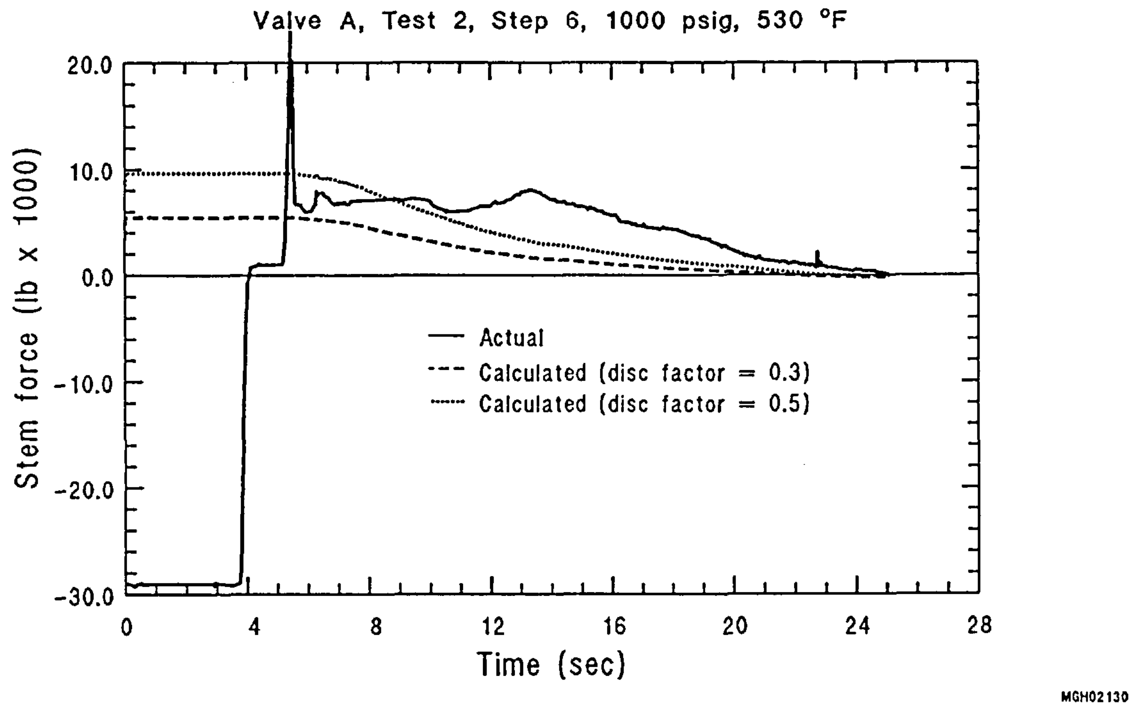

Figure 3. In this Valve A test,. peak thrust encountered during opening was measured not while the disc was being lifted off the seat, but well after flow was established.

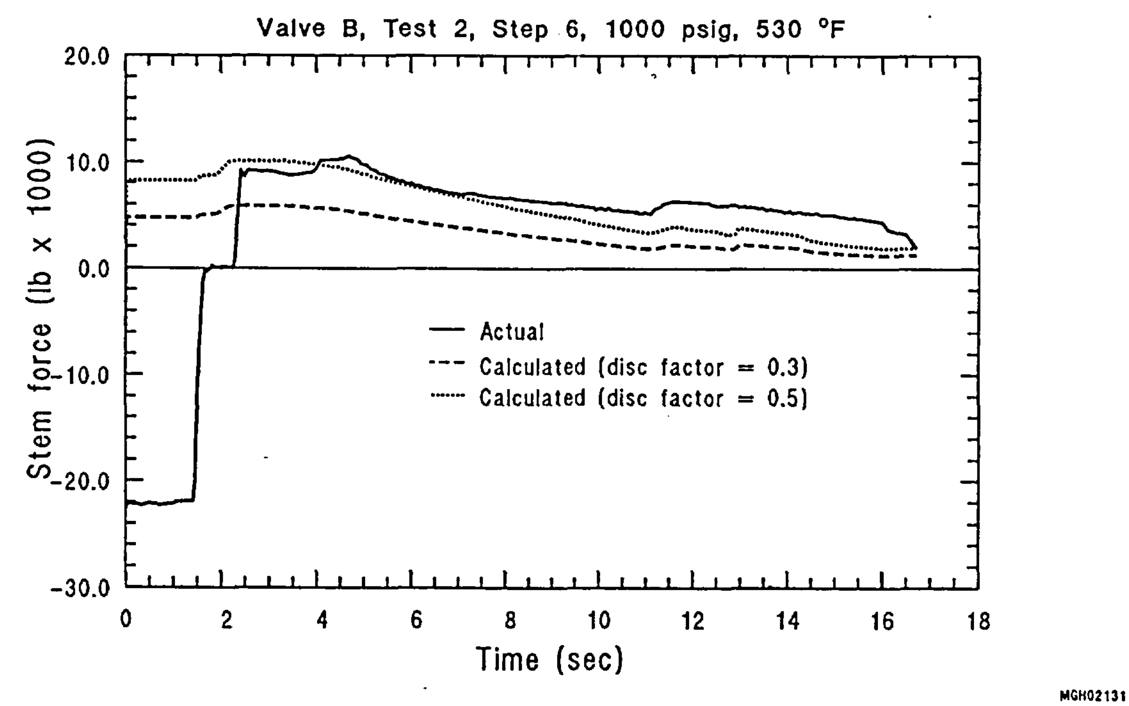

Figure 4. The response of Valve B is similar to that of Valve A (Figure 3); the absence of a spike at the hammer blow is because the valve was not fully seated at the end of the previous closing cycle.

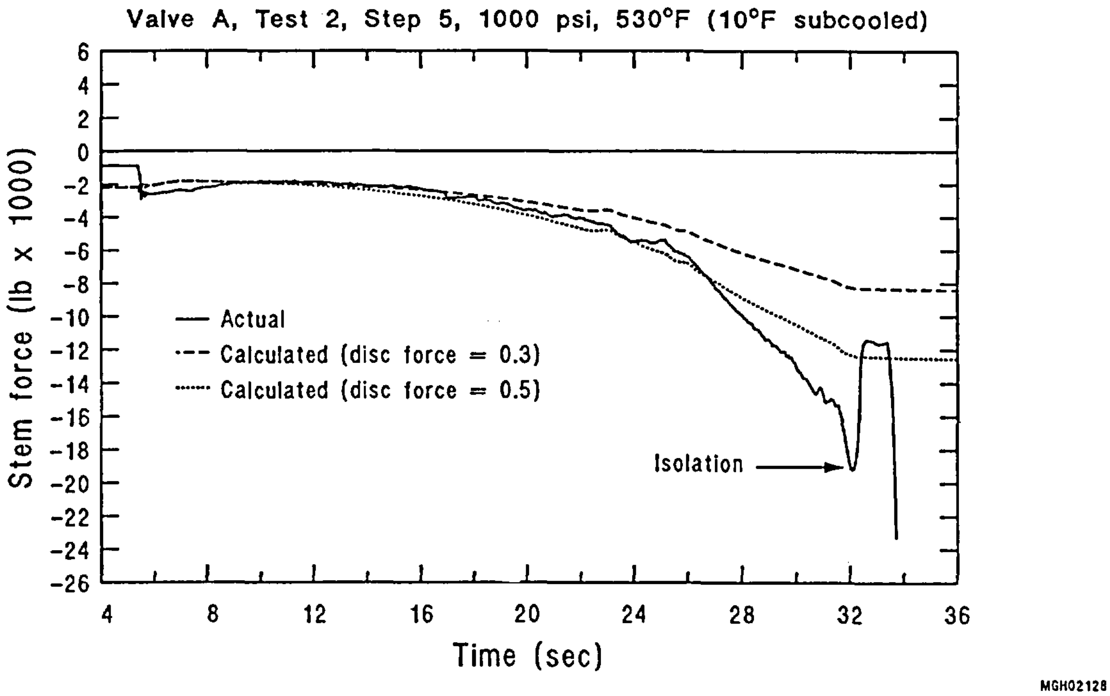

Figure 5. In this Valve A test, the loads measured during closing were greater than the loads calculated using 0.3 and 0.5 disc factors.

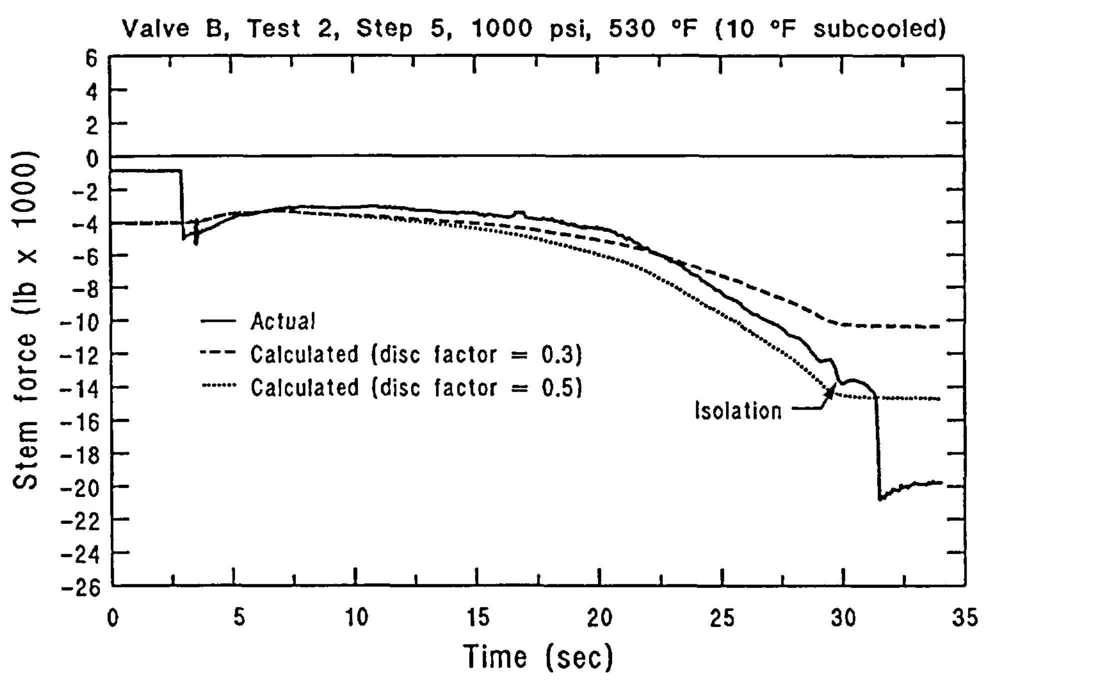

Figure 6. The response of Valve B is more linear than that of Valve A; the calculation using a 0.5 disc factor marginally envelopes the measured load, but the 0.3 calculation is not conservative.

In the initial analysis for the adequacy of Equation (1) it appeared there were two problems: (a) in the case of the linear valve performance, a term may be missing from the equation or the typical 0.3 disc factor may be too low and (b) the equation appeared inappropriate for predicting the performance of a nonlinear valve.

A summary of some of the important design features of the two valves is presented in Table 4. Two design differences may account for the valve's response. The first difference is in the disc guide face and body guide materials. Valve A is typical of most nuclear valves with guide surfaces made of cast carbon steel, type A216–WCB, while Valve B uses similar base materials but hardfaces the disc guide face with Stellite 6. The second difference is in the disc guide to body guide clearances. Valve A had 1/4–in. clearances as opposed to Valve B's 1/8–in. The other differences in valve design, stem diameter and packing drag, would serve to increase the loadings on Valve B.

Disassembly and inspection of the two valves after completion of the test program provided some insight into the nonlinear behavior of Valve A. Inspection of the disc guide surfaces showed a wear pattern, indicating that the disc had tilted downstream as it closed, with a very small bearing area of the disc guide riding on the valve body guides. These small bearing areas show signs of yielding, galling, and plastic deformation. As mentioned the disc to guide clearances on Valve A are twice as large as the clearances on Valve B. We believe that the nonlinear performance of Valve A is the result of the greater disc–to–guide clearance, which allowed the disc to tilt in this valve design. This hypothesis is further confirmed by the fact that at flow isolation, when the disc entered into full contact with the seat ring and the disc sealing surfaces became the primary guiding surface, the closing forces dropped, as shown in Figure 5. The final seating forces are slightly less than the calculation using the 0.5 disc factor.

Figure 7 is a view of the inside of the Valve A body with the two seat sealing surfaces and body guide surfaces. Figure 8 shows the Valve A disc. Sealing surface damage can be seen and the lower right disc guide shows indications of the small bearing area that was engaged during closing. Figure 9 is a view of the disc rotated, looking down the guide surface. The right guide surface shows evidence of yielding, plastic deformation, and significant galling on the lower edge of the disc. Figure 10 provides a close-up of the body guide that mated with the right disc guide shown in Figure 9. This guide also shows signs of galling. Figure 11 shows the left disc guide, showing the same small bearing area and, again evidence of plastic deformation. The damage to Valve A is of course magnified after undergoing ten design basis loadings; however, the wear patterns do provide evidence that the disc tilted in the guides, resulting in the nonlinear performance.

Judging by the wear patterns shown in Figure 8, it is unlikely that Valve A could have produced a tight seal using the downstream face alone. A seal on the downstream face would be necessary to isolate flow if the valve torqued out before full travel but with the disc on the seat. However, the valve maintained its leak integrity throughout testing, indicating proper sealing on the upstream face of the disc—the result of using the oversized operator with a higher–than–necessary seating thrust.

Table 4. A design comparison between Valves A and B

Valve Design	Valve A	Valve B
Disc guide face	A216–WCB	A216–WCB Stellite
Valve body guide	A216–WCB	A–36
Disc sealing surface	Stellite	Stellite
Seat	Stellite	Stellite
Disc guide to body guide clearances	1/4 in.	1/8 in.
Stem diameter	1 1/2 in.	1 3/4 in.
Estimated maximum packing drag	1500 lb	5000 lb

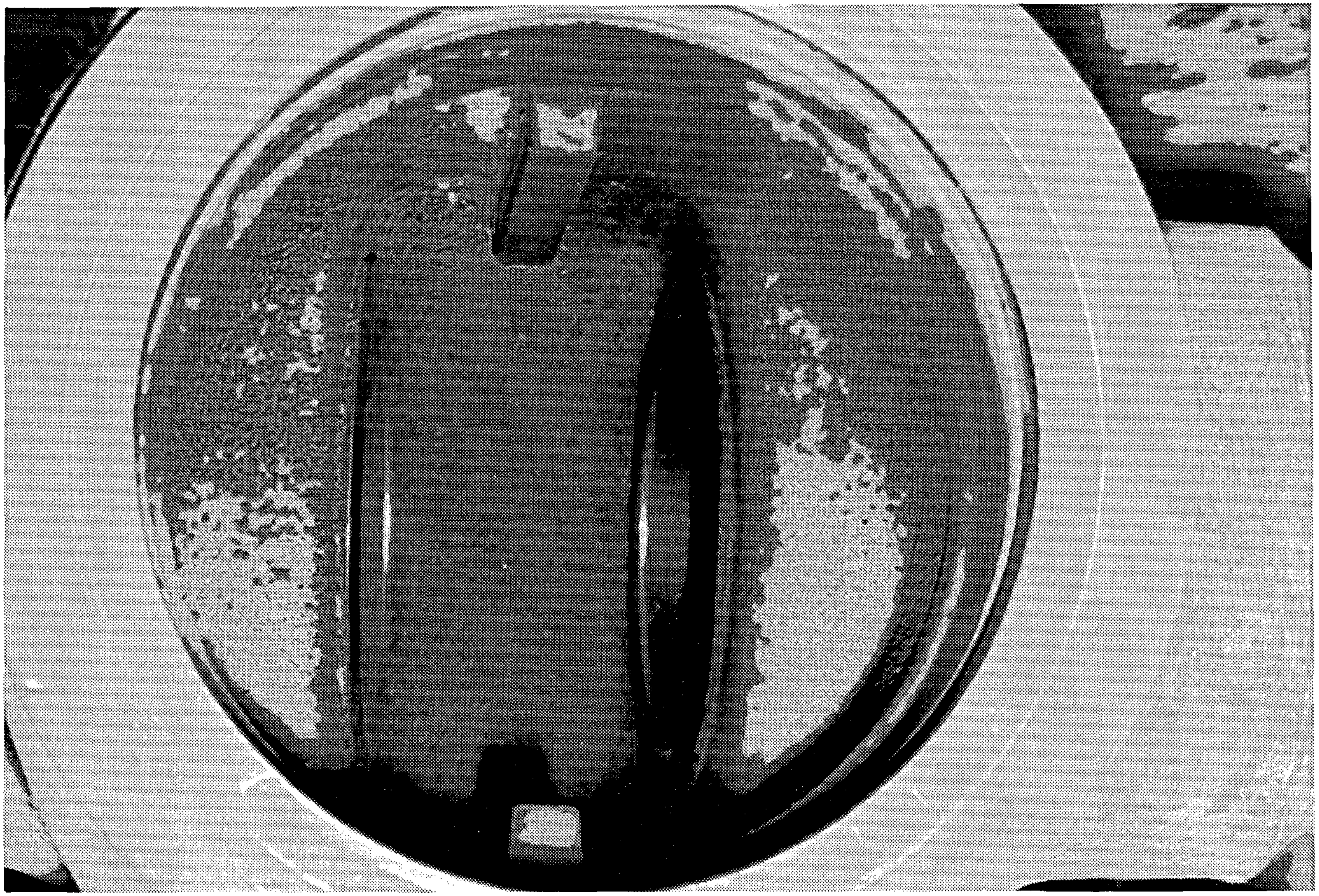

Figure 7. During closure, the Valve A body guides should provide proper disc alignment until the disc contacts the entire seat ring surface.

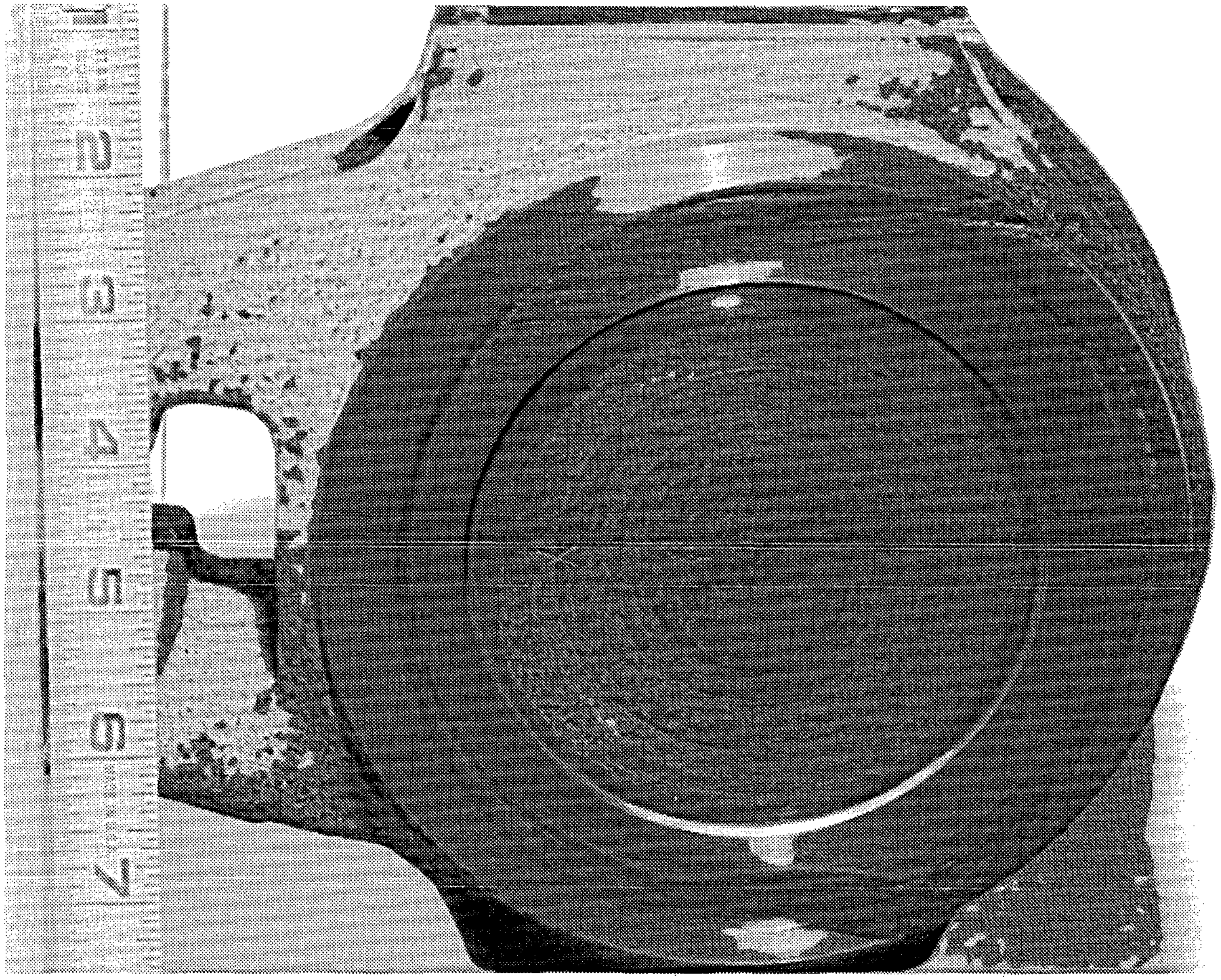

Figure 8. The Valve A disc, which shows sealing surface damage on the downstream face.

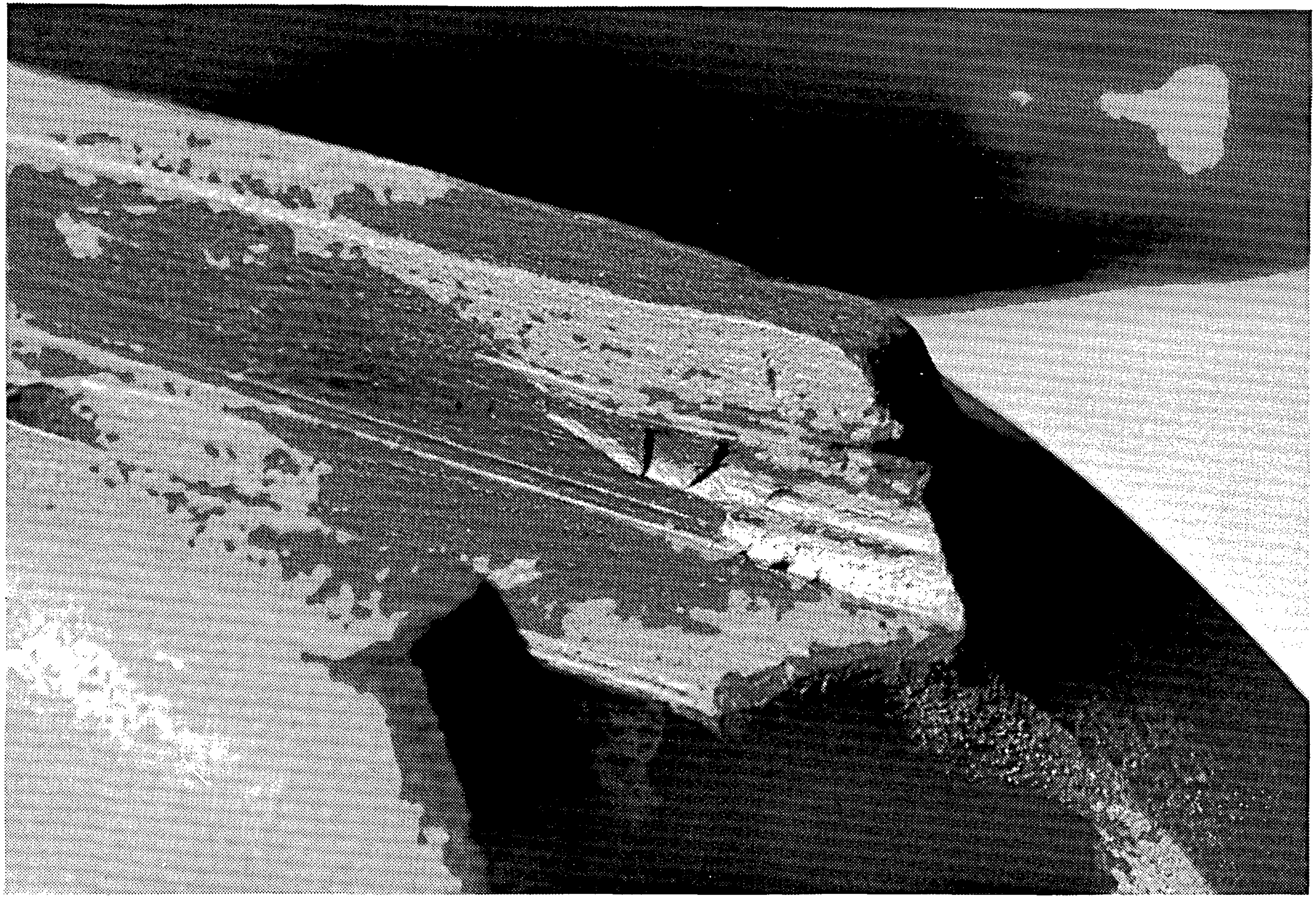

Figure 9. Close–up of the Valve A lower right guide surface, which shows yielding, plastic deformation, and significant galling on the lower edge of the disc.

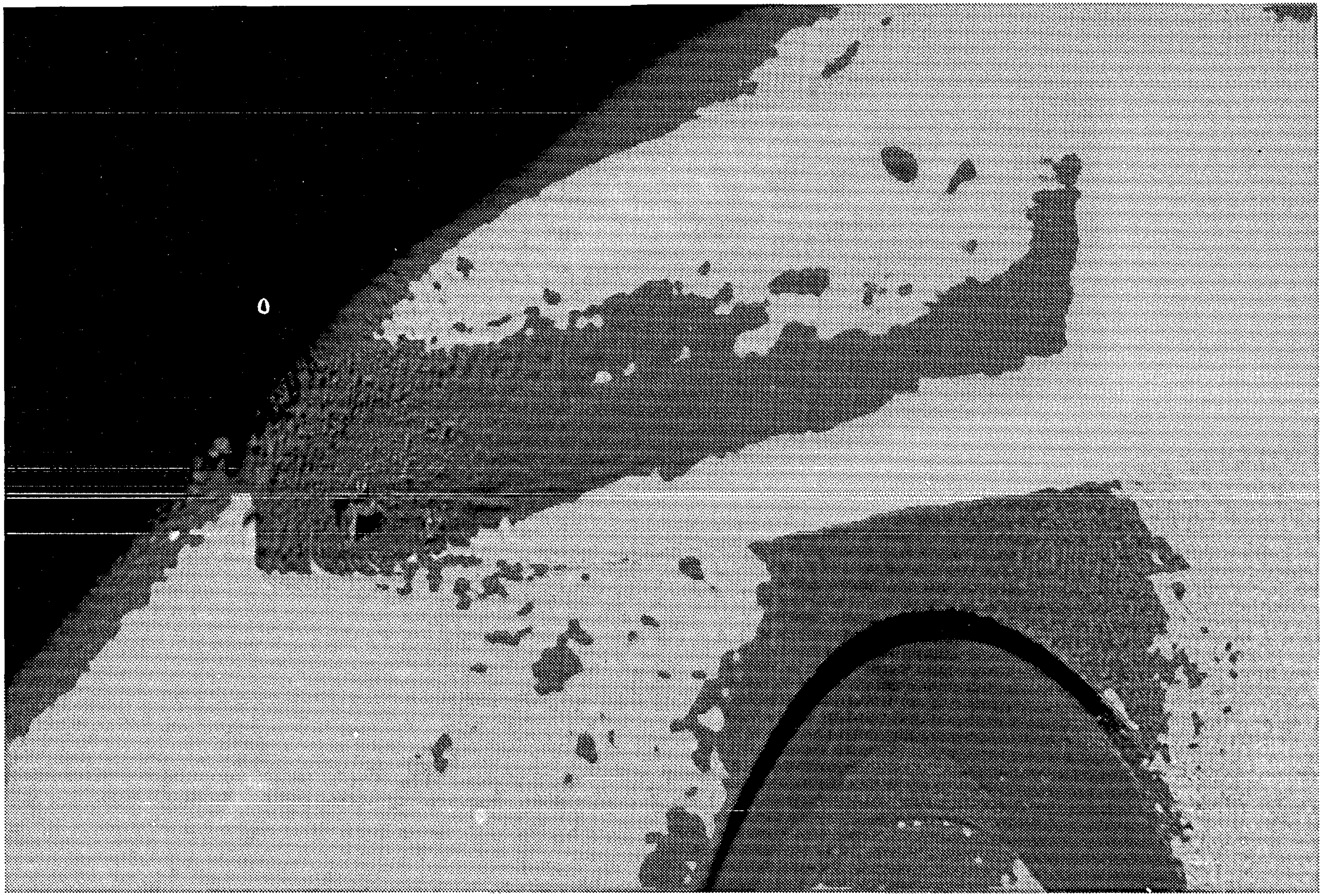

Figure 10. The Valve A body guide, which mates with the right disc guide (shown in Figure 8), shows galling near the fully closed position.

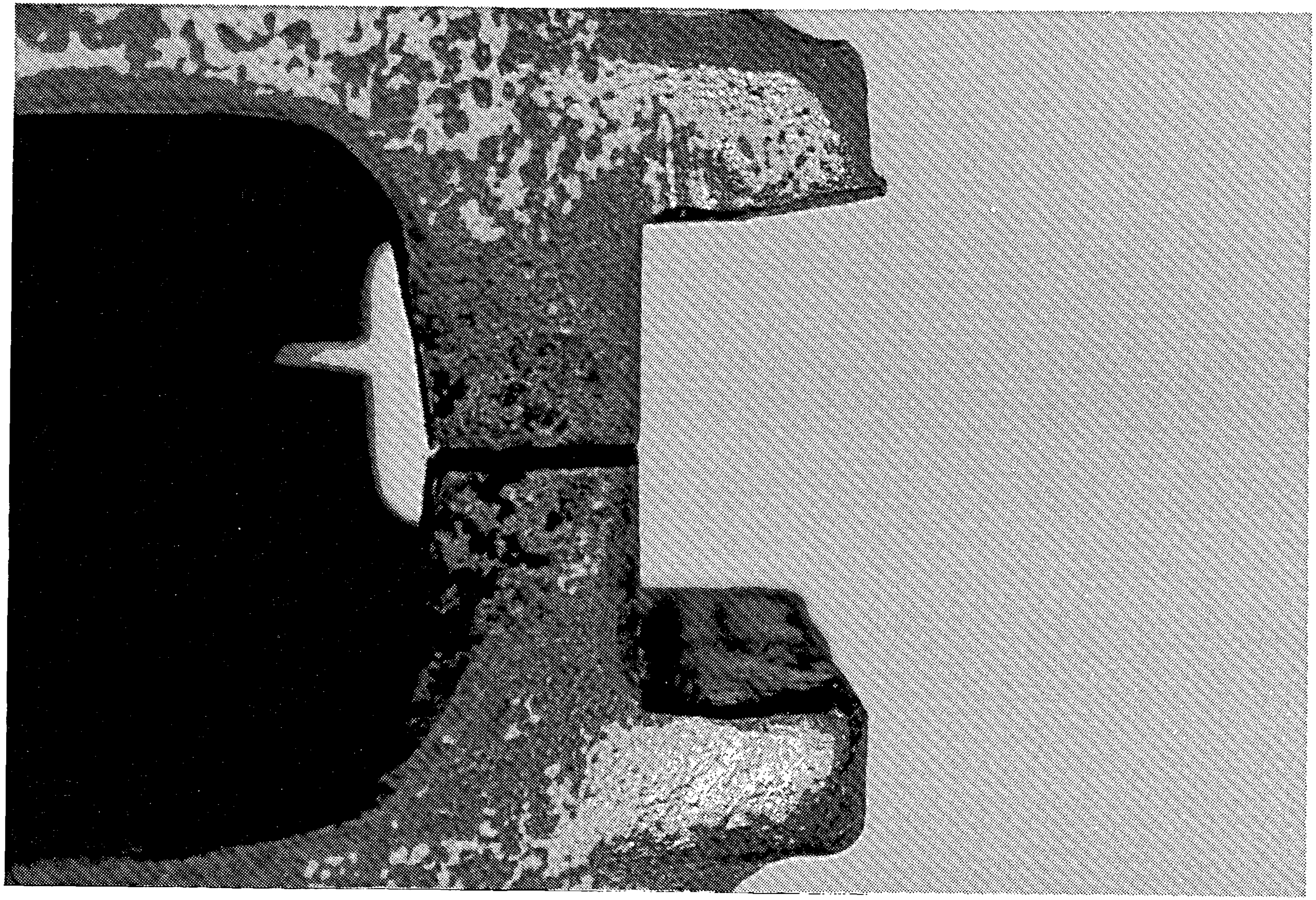

Figure 11. Close–up of the Valve A lower left guide surface, which shows yielding and plastic deformation similar to the right side of the disc.

Disassembly and inspection of Valve B's internal surfaces showed only very minor galling on the body guides. The disc guide surfaces were hardfaced and showed no sign of wear.

It must be noted that a survey of the valves installed in these BWR isolation functions shows that the majority of the valves now in use have nonhardfaced guide surfaces. The manufacturer of Valve B started hardfacing disc guides after 1970, and the other manufacturers only hardface guides on special request. We do not believe the nonlinear performance of Valve A is solely a function of whether the disc guide surfaces are hardfaced, but rather the nonlinear performance is primarily a function of the large disc–to–body guide surface clearances, which allowed the disc to tilt and thus reduce the contact bearing area.

As more and more test data were analyzed, the adequacy of Equation (1) appeared to depend on many fluid parameters. The only tests where Equation (1) with a 0.3 disc factor predicted the response of either valve were the ambient temperature tests with upstream pressure loads. All tests at normal operating temperature required a higher disc factor. In order to assure ourselves that we were not adding conservatism to an inappropriate equation, we performed a detailed analysis, as shown in subsequent areas of this report on each term in the equation to determine if there was a term missing or if the disc factor had been underestimated by industry.

It must be also noted that we were aware of the Westinghouse work performed after the EPRI power–operated relief valve (PORV) and PORV block valve tests conducted at the Duke Power, Marshall facility. Westinghouse found that their disc factors needed to be increased significantly to account for the added friction in the disc–to–body guide surfaces. The initial differences we saw comparing our data with the Westinghouse data were that the Westinghouse data were for stainless steel valves in a steam environment and their problems did not go away at ambient temperature. Our carbon steel valves seemed to be more sensitive to temperature and to the fluid properties as the fluid approached saturation.

4.2 Stem Force—Static Components

We initially divided Equation (1) into static and dynamic components. In order to investigate how well Equation (1) models the actual behavior of the valve, the conservatism needs to be removed from the calculation. If the disc load is eliminated from Equation (1), such as would be the case without flow, what remains is a linear equation in slope–intercept form (y = mx + b), namely

$$F_s = -A_s(P) \pm F_p \; . \qquad (3)$$

Note that this equation has been written so that the stem rejection load is always negative (compression), while the packing load is either negative or positive depending on whether the valve is closing (compression) or opening (tension).

Figures 12 and 13 show the test data for Valve A that apply to the above equation. The data plotted are the stem forces measured at mid–stroke (running load) for tests at varying temperatures and pressures but without flow. The line fit through the data points has a slope equal to the stem cross–sectional area and provides an indication of the true packing load for each case. The data show a packing load of 835 lb for opening and 430 lb for closing for Valve A. Both values are well below the 1500–lb maximum packing load used by the manufacturer in the sizing equation. The difference between the two values can be partially accounted for by the weight of the disc and lower half of the stem. This difference also provides evidence that the packing load is affected by direction of travel, possibly caused by water carried with the shaft changing the lubrication of the packing/stem surfaces or by other phenomena associated with stem travel through packing.

Figures 14 and 15 show the Valve B test data that apply to Equation (3). Here again, the data plotted are the stem forces measured at mid–stroke for tests at varying temperatures and pressures but without flow. The line fit through the data points has a slope equal to the stem cross–sectional area and provides an indication of the true packing load for both the opening and closing strokes. The increase in packing forces over those found for Valve A is believed to result from the different packing design and greater stem diameter of this valve.

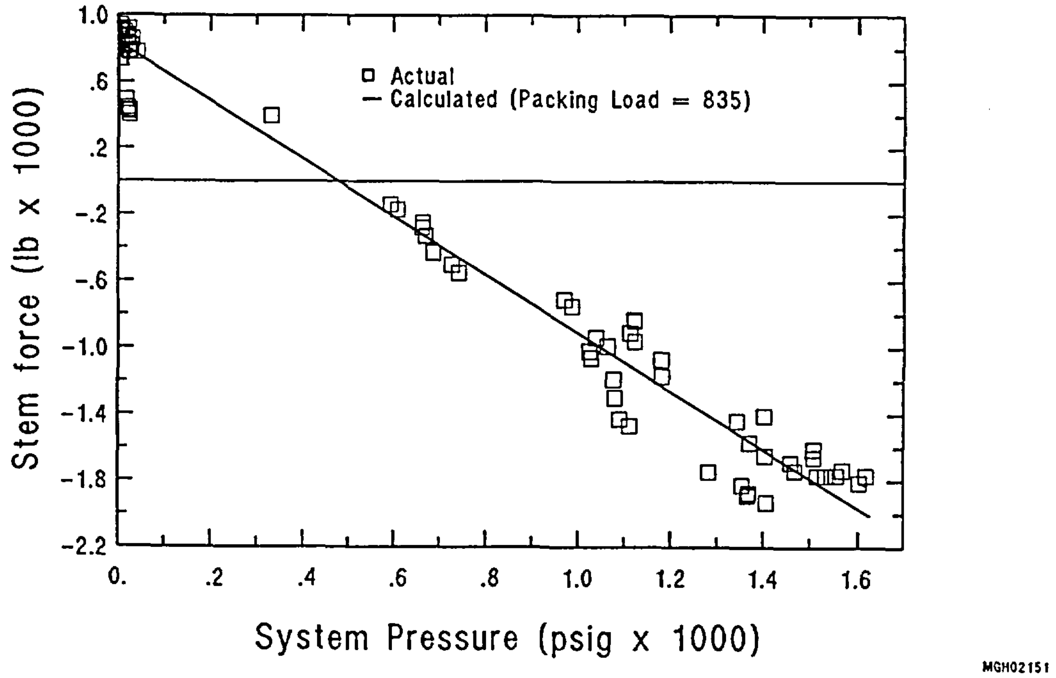

Figure 12. A linear curve fit using the industry equation closely approximates the Valve A running stem forces for opening without flow.

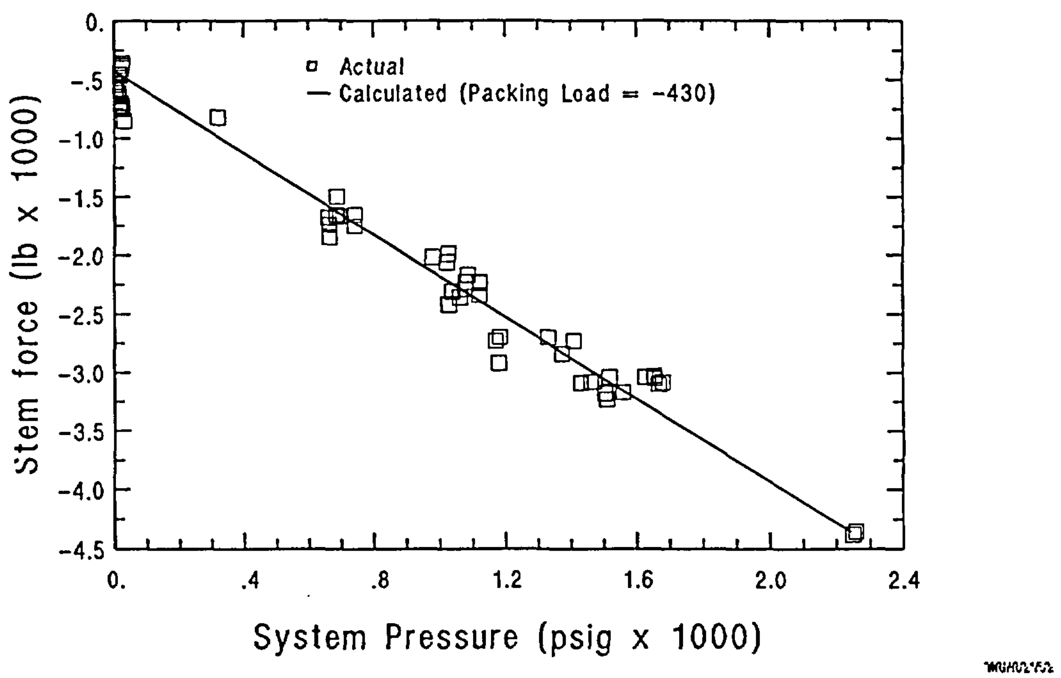

Figure 13. A linear curve fit using the industry equation closely approximates the Valve A running stem forces for closing without flow.

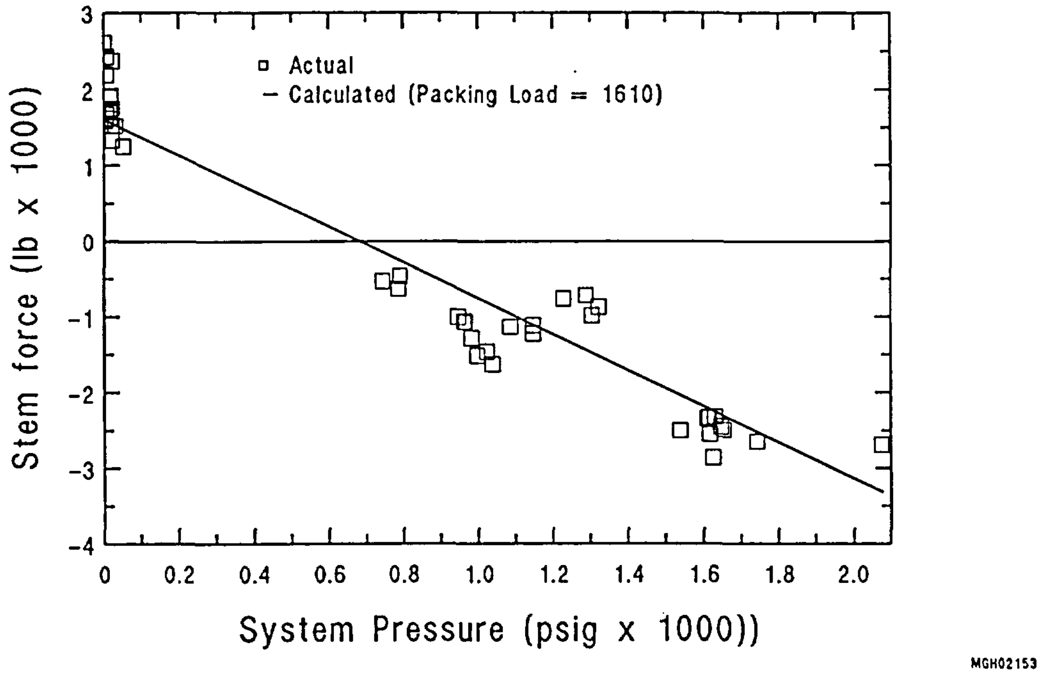

Figure 14. A linear curve fit using the industry equation closely approximates the Valve B running stem forces for opening without flow.

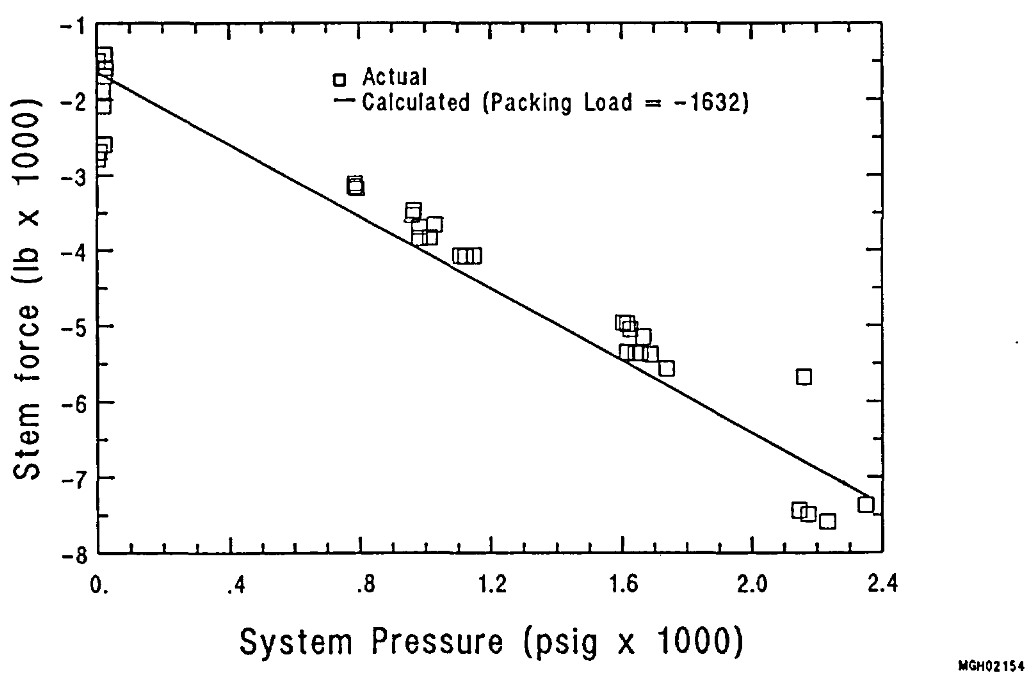

Figure 15. A linear curve fit using the industry equation closely approximates the Valve B running stem forces for closing without flow.

The line fit through the data points for Valve B shows a packing load of 1610–lb opening and 1632–lb closing. The maximum force calculation used by the manufacturer for this case included a 5000–lb packing load. The difference between the opening and closing values is less than expected, given the weight of the disc and lower stem half (approximately 50 lb). This may indicate a directional relationship for packing load, believed to be a characteristic of the packing type used and its orientation.

The measured packing loads for both valves were below the maximums estimated by the manufacturers, and the stem rejection loads were indeed linear with pressure as predicted in Equation (1) for these no–flow tests.

4.3 Stem Force—Dynamic Component

Throughout the testing, both valves repeatedly isolated flow, although at a higher than anticipated thrust. In generating the analytical thrust calculations for comparison to measured valve stem forces, we used actual response histories for all the variables in Equation (1) except the stem factor. This comparison of measured stem forces with values calculated using both the 0.3 and 0.5 disc factors provides a common basis for discussing valve operating characteristics. In the previous section on the static component for Equation (1) we found good agreement with the measured loads versus the calculated loads. This would indicate that if there is a problem with the equation, it is in the dynamic component.

Previously in Figures 3 through 6 we saw that the forces required to close the valves were above those calculated using the 0.3 disc factor; however, using a 0.5 disc factor we came close to bounding the linear behavior of Valve B. We were also able to explain the nonlinear behavior of Valve A, which was apparently due to its design (large guide tolerance). However, as we increase the fluid subcooling, Valve A returns to more linear behavior, as shown in Figure 16. The inlet pressure is the same pressure (1000 psig) as in Figure 5; however, the temperature is 130°F less than in Figure 5. Figures 17 and 18 show this same relationship at the 1400 psig inlet pressure. The actual stem force in Figure 17 shows the same nonlinear response as shown in Figure 5 (both 10°F subcooled cases). Figure 18 shows the same actual versus calculated relationship as shown in Figure 16, which are both 130°F subcooled tests; note that the behavior is linear.

The upstream temperature and pressure measurements (see Figure 2, measurement locations P–2 and T–2) in the 10°F subcooled tests were evaluated. Figure 19 shows that upstream flashing occurred from about the 5% through the 55% closed position, in the 1000 psig, 10°F subcooled test. The nonlinear behavior shown in Figure 5 does not start until about the 26–s time line, which equates to about 66% closed. While flashing and two–phase flow may have occurred upstream of the valve in the 10°F subcooled cases, the upstream liquid is apparently recovered to subcooled before the start of the nonlinear behavior. The actual differential pressures across the valve disc in these four tests are incrementally part of the calculations and are thus accounted for; however, the peak of the nonlinear behavior was at flow isolation and the differential pressure for all four tests at isolation is near 100% or equal to the upstream pressure. The fluid property differences between the 10 and 130°F subcooled cases are not significant. The density is higher in the 130°F subcooled case but the velocity is lower; in the 10°F subcooled case the density is lower and the velocity is higher. The small differences in density and velocity do not account for the differences in valve response.

The single major difference in the tests is fluid temperature. The valves are preheated before the flow interruption test to the initial temperature for that test. As the valve disc closes, lowering the pressure downstream of the disc, the valve is much warmer than the coolant saturation temperature downstream of the valve; this temperature and the lower downstream pressure could dry out the sliding surfaces between the valve disc and the valve body. This would change the interface between the disc guide and valve body guide from a film of water to steam. Steam is much less effective as a lubricant than water, therefore the friction factor on the guide surfaces could be much greater after dryout. This phenomenon (drying out) would occur sooner during the tests starting at 10°F subcooled.

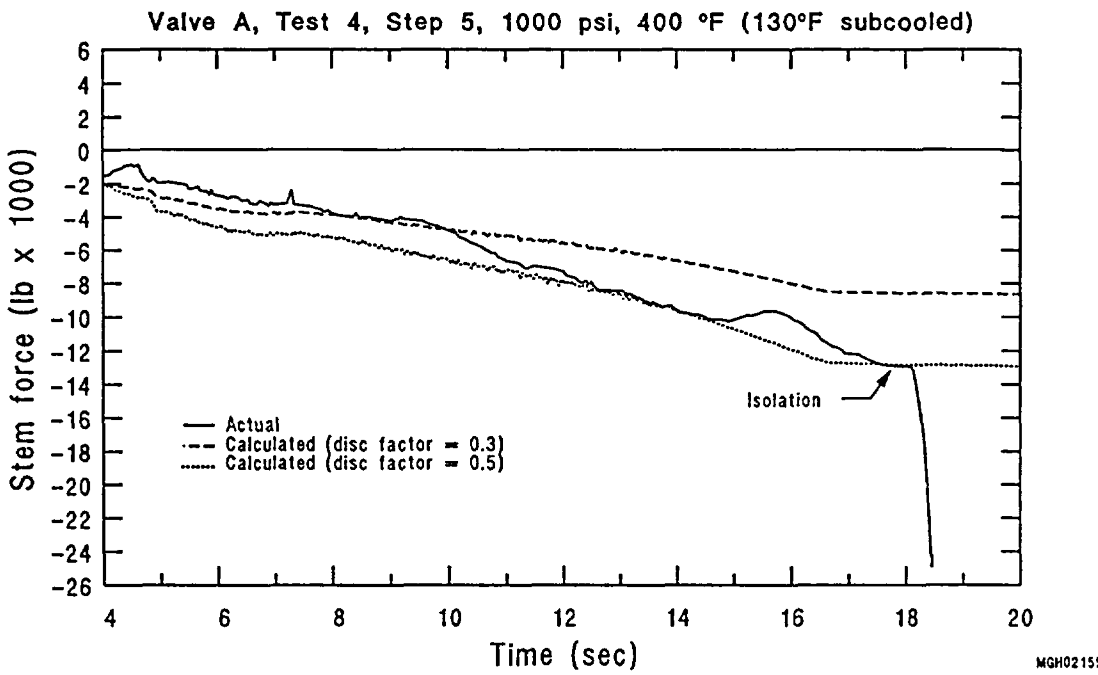

Figure 16. With an increase in the fluid subcooling, Valve A returns to more linear behavior and stem forces are bounded by the 0.5 disc factor calculation.

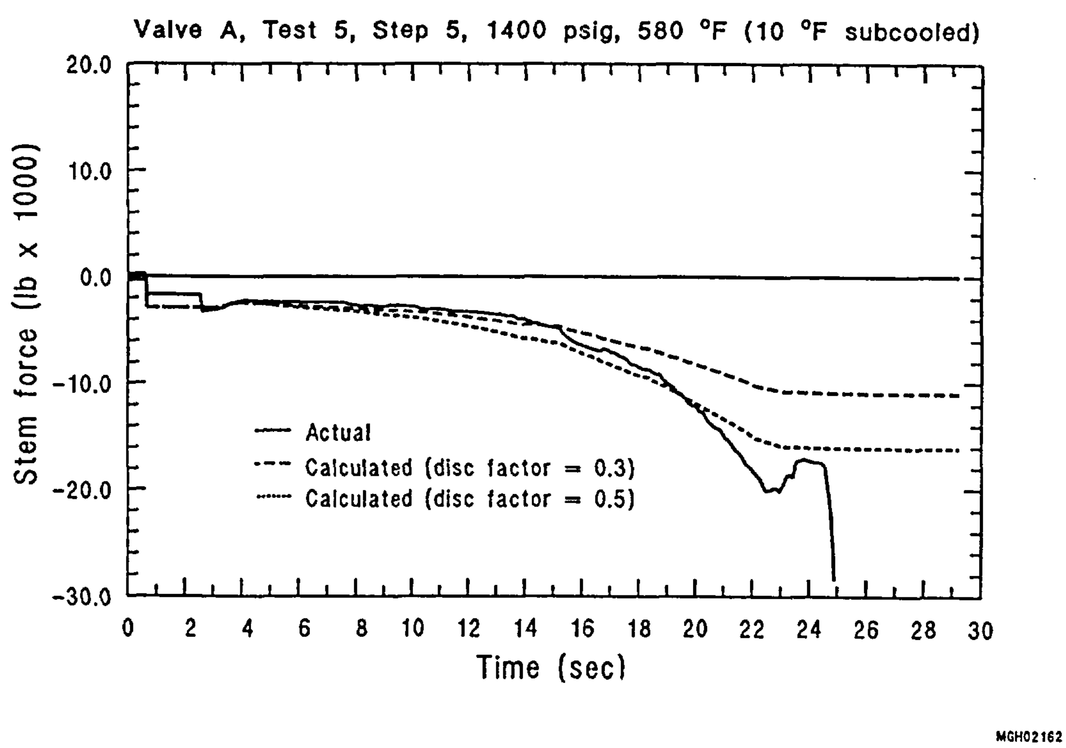

Figure 17. At 1400 psig and 10°F subcooled, Valve A shows the same nonlinear response as in Figure 5.

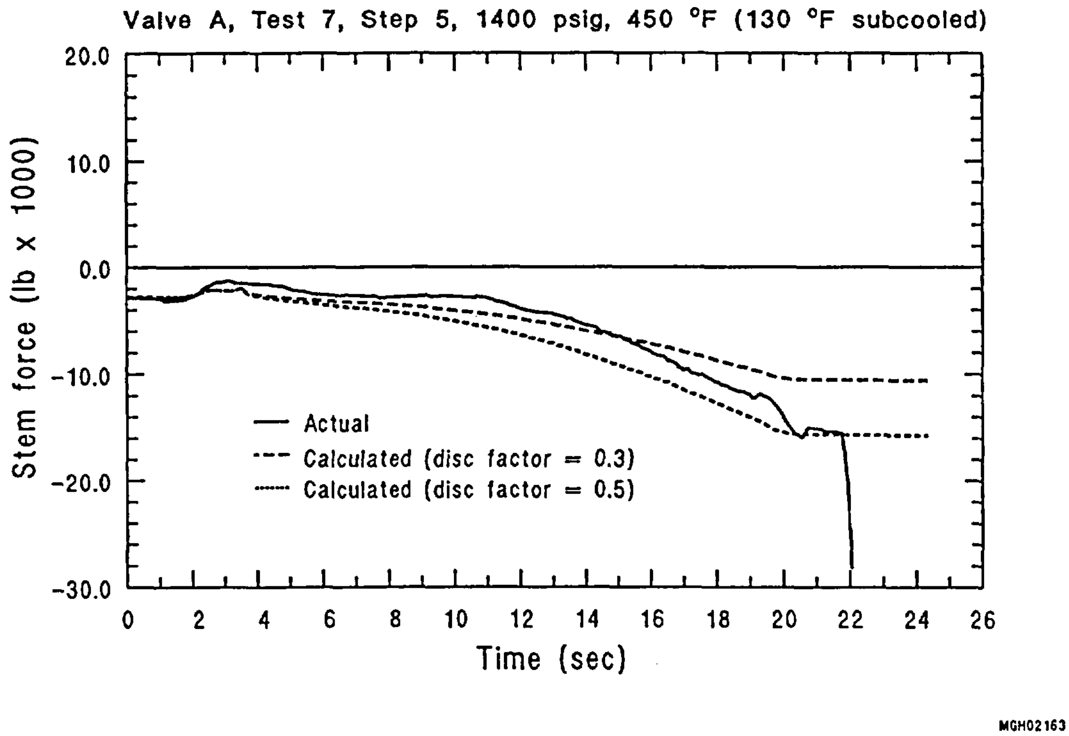

Figure 18. At 1400 psig and 130°F subcooled, Valve A returns to more linear behavior and stem forces are again bounded by the 0.5 disc factor calculation.

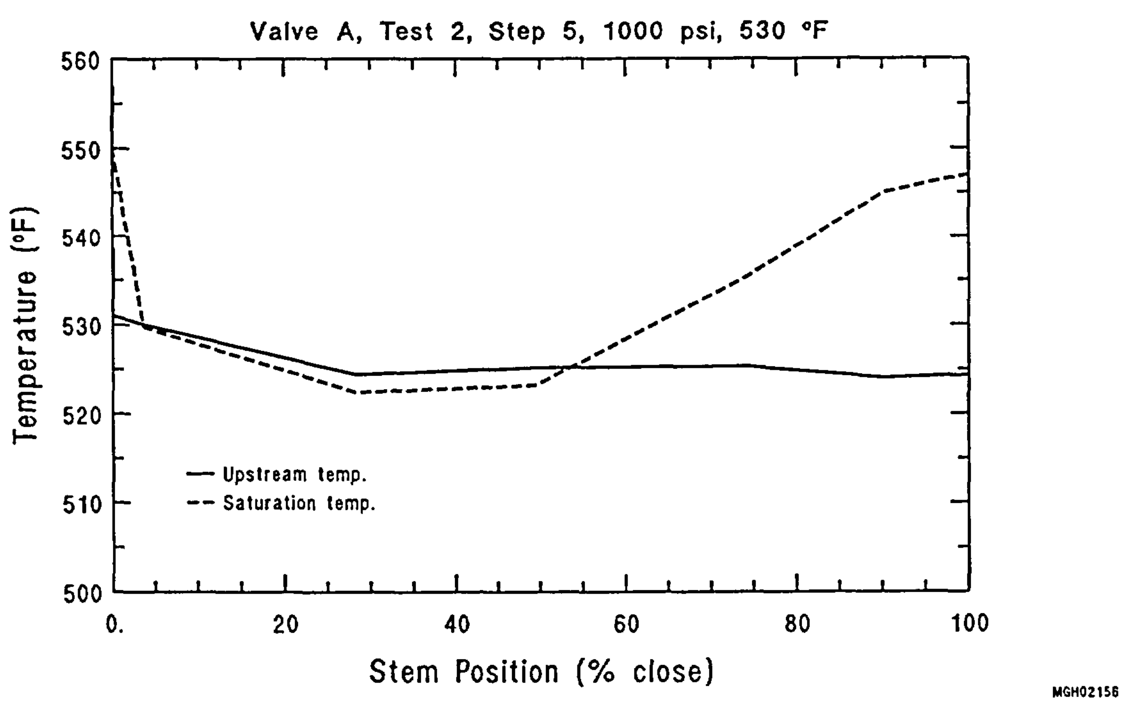

Figure 19. Flashing occurred upstream of Valve A during the early part of the closure stroke, but recovered to subcooled prior to the start of the nonlinear behavior (compare Figure 5).

A second example of temperature affecting stem thrust requirements is shown in Figures 20 through 23. Figures 20 and 21 show the stem force requirements to open Valves A and B respectively, against upstream pressure with the downstream side vented (opening ΔP test) at ambient temperature. The calculated forces are again shown with a 0.3 and 0.5 disc factor. As can be seen, both valve opening histories are enveloped by the 0.3 calculation. Figures 22 and 23 show the same opening tests with a slightly lower starting pressure, but at 580°F. Note the actual stem force requirements have increased and are no longer enveloped by the 0.3 disc factor calculation. These tests were performed during the initial valve qualification tests. The slightly lower inlet pressure is due to temperature considerations on the 900-lb class valves. The primary difference in test conditions is temperature.

Review of the dynamic component of Equation (1) shows that the total force was measured, the disc area at anytime during the opening or closing cycle was known from measured disc positions, and the differential pressure was measured. The unknown value is the disc factor. A disc factor of 0.3 has been used to size most gate valve motor operators in the past. That number appears to be unconservative for the valves tested in this program at conditions above ambient temperature. The 0.5 disc factor appears to marginally bound most linear valve responses at temperature. Nonlinear valve responses may be caused by design problems, thus Equation (1) is inappropriate for predicting those responses.

4.4 Valve Opening Versus Closing Tests

Valve opening stem forces and the comparison of opening loads versus closing loads were evaluated for the following reasons: (a) the safety function of some valves is to open against the full system ΔP and (b) opening tests have been used by industry in the past to predict closing loads. If this practice can be substantiated, it might in some cases provide utilities the ability to determine disc factors in situ.

As discussed in the previous section under temperature effects, we performed a cold opening ΔP test at 1700 psig and less than 100°F during the qualification tests required by ANSI B16.41, Annex B. The valve was closed under static pressure in this test and the downstream piping was vented. The valve was then opened with the system differential pressure across the disc. Figure 20 shows the data from this test with Valve A. The stem force history starts out with the valve closed and the valve stem in compression. As the opening cycle begins we see a decrease in stem compression and then the stem goes into tension with the operator hammer blow. Next, the stem force shows the drag load as the disc slides on the seat and then a decrease in tension as the valve unseats and system pressure is equalized. Once pressure on both sides of the valve is equal, the stem force shows the sum of the packing drag load (tension) and the stem rejection load (compression). Figure 21 shows the same test for Valve B. The figures also show a comparison between the measured stem forces and calculations using the 0.3 and 0.5 disc factors. As previously stated, the 0.3 disc factor calculation envelopes the forces measured for both valves during the cold opening ΔP test.

Reviewing Figures 22 and 23, which show the results of the Annex C hot opening ΔP tests against a slightly lower pressure differential and with a fluid temperature of 580°F, we again see the forces needed to open the valves are significantly higher than those measured during the cold tests, and the calculation with a 0.5 disc factor only marginally envelopes the responses. The temperature effects on the disc factors show that a correct disc factor cannot be determined from cold testing.

The hot opening ΔP test may identify the correct disc factor for valve closing requirements, which have shown linear disc friction characteristics through complete qualification, including flow interruption testing. Figure 24 shows the results from the Valve B opening ΔP test performed at BWR normal operating conditions just before the full flow isolation test. The relationship between actuals and calculated is very similar to the Valve B Annex C test (the results of which are plotted in Figure 23) and also to the pipe break flow isolation test for the same fluid conditions shown in Figure 25. The data from all three tests appear to support the use of a disc factor very near 0.5.

A similar analysis for Valve A was not successful as none of the opening or lightly loaded closing tests provided insights into the nonlinear disc friction behavior of Valve A seen in the 10°F subcooled flow isolation tests, shown in Figures 5 and 17.

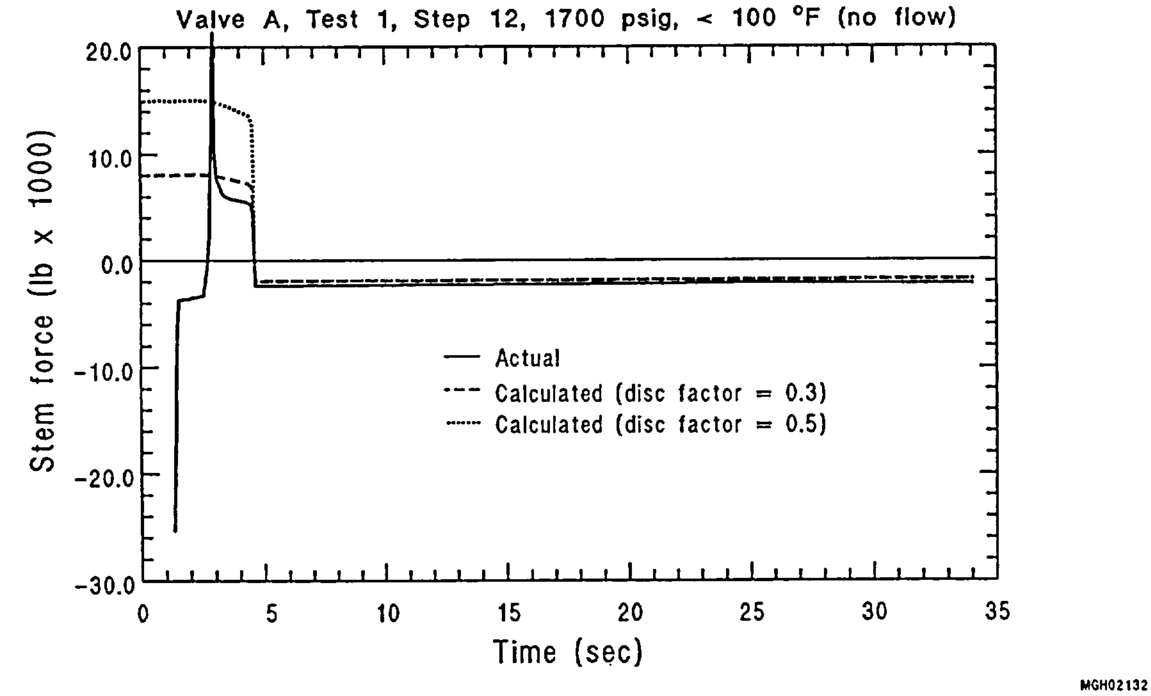

Figure 20. With Valve A opening against differential pressure only (no flow, cold fluid), the measured thrust is enveloped by the 0.3 disc factor calculation.

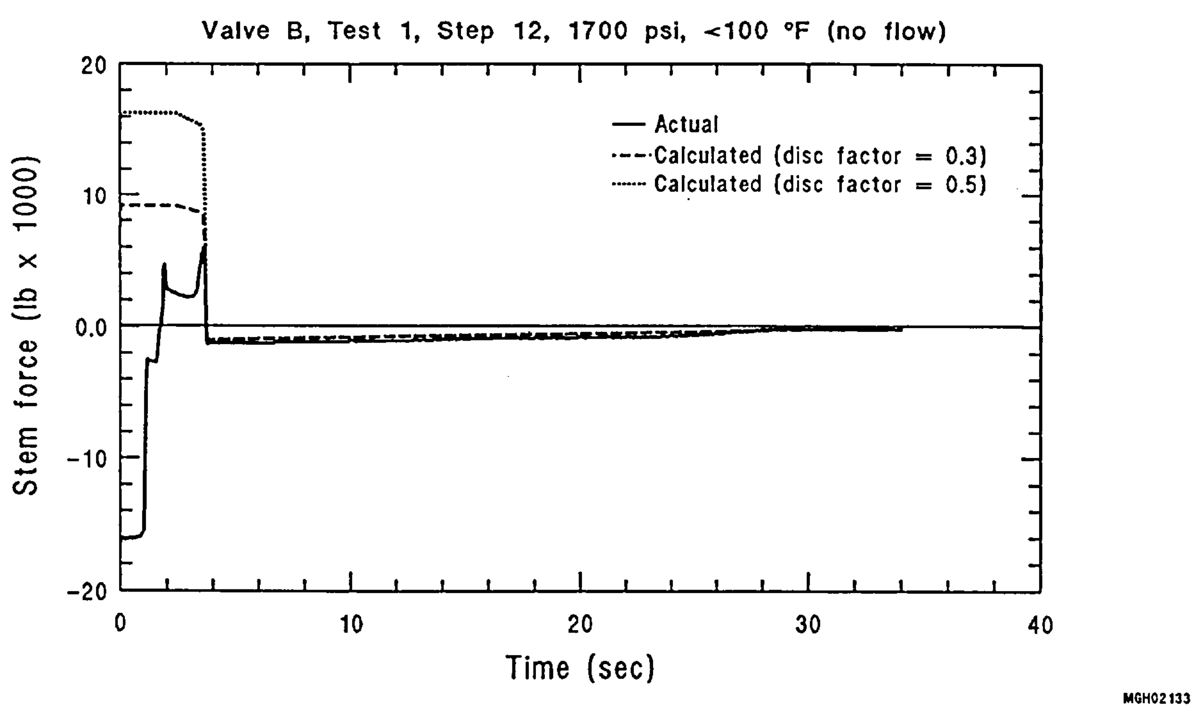

Figure 21. With Valve B opening against differential pressure only (no flow, cold fluid), the measured thrust is enveloped by the 0.3 disc factor calculation.

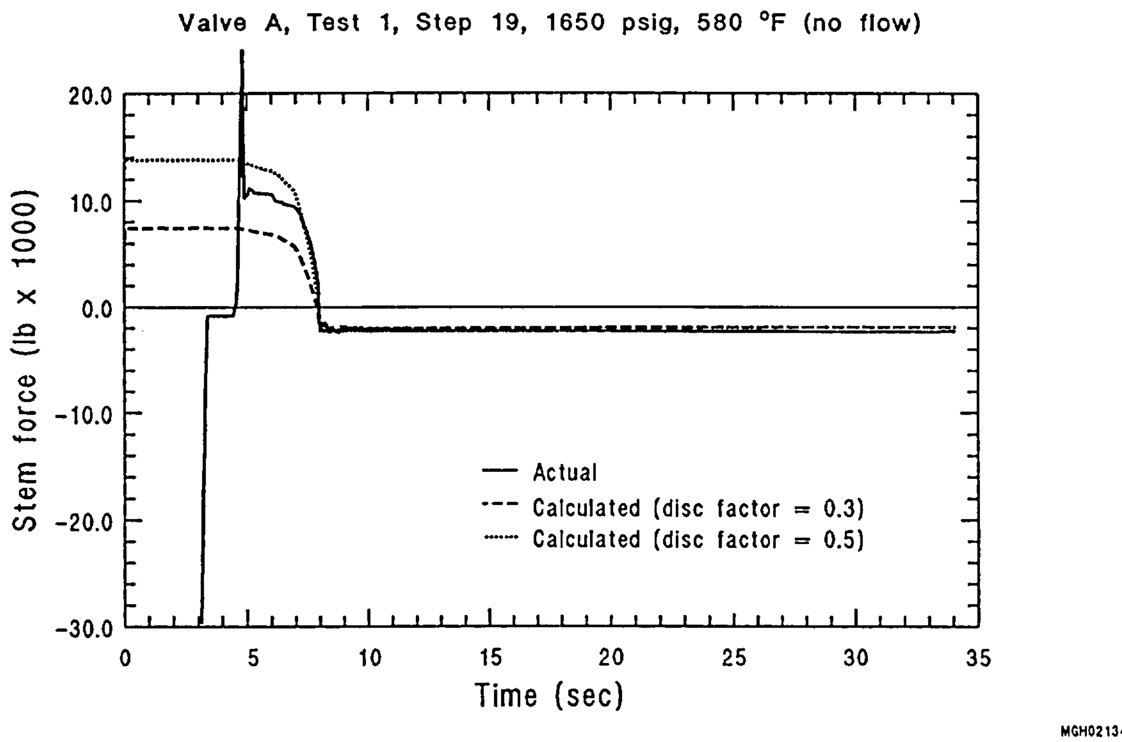

Figure 22. With an increase in fluid temperature, the Valve A measured thrust is not enveloped by the 0.3 disc factor (compare Figure 20).

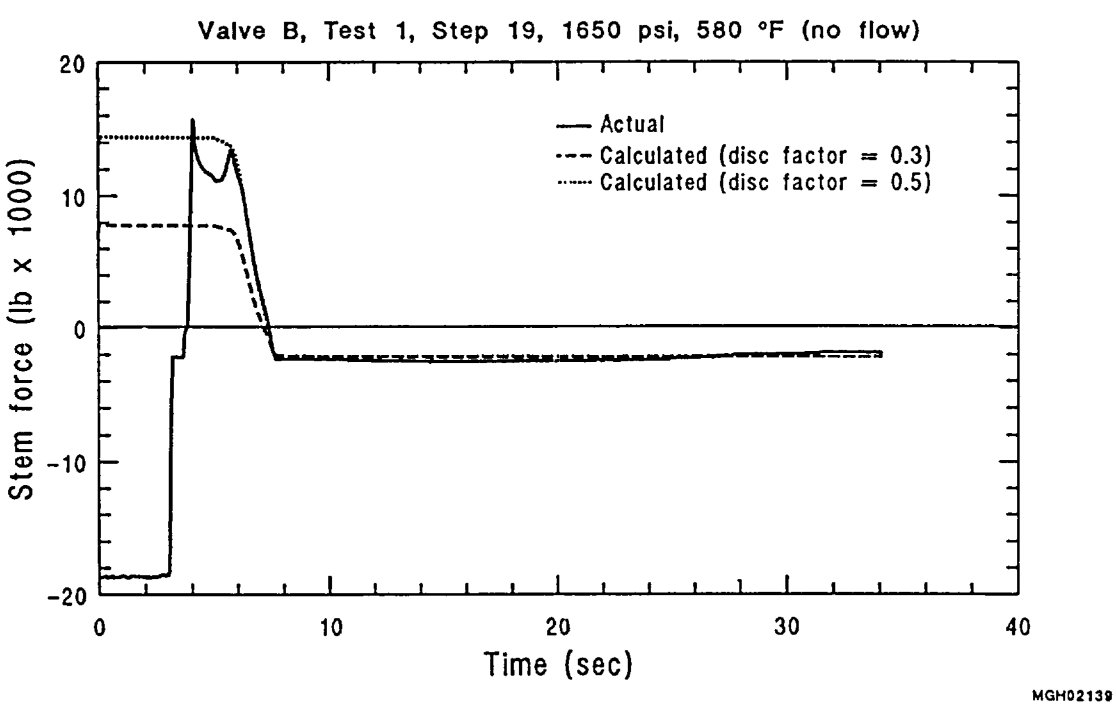

Figure 23. With an increase in fluid temperature, the Valve B measured thrust is not enveloped by the 0.3 disc factor (compare Figure 21).

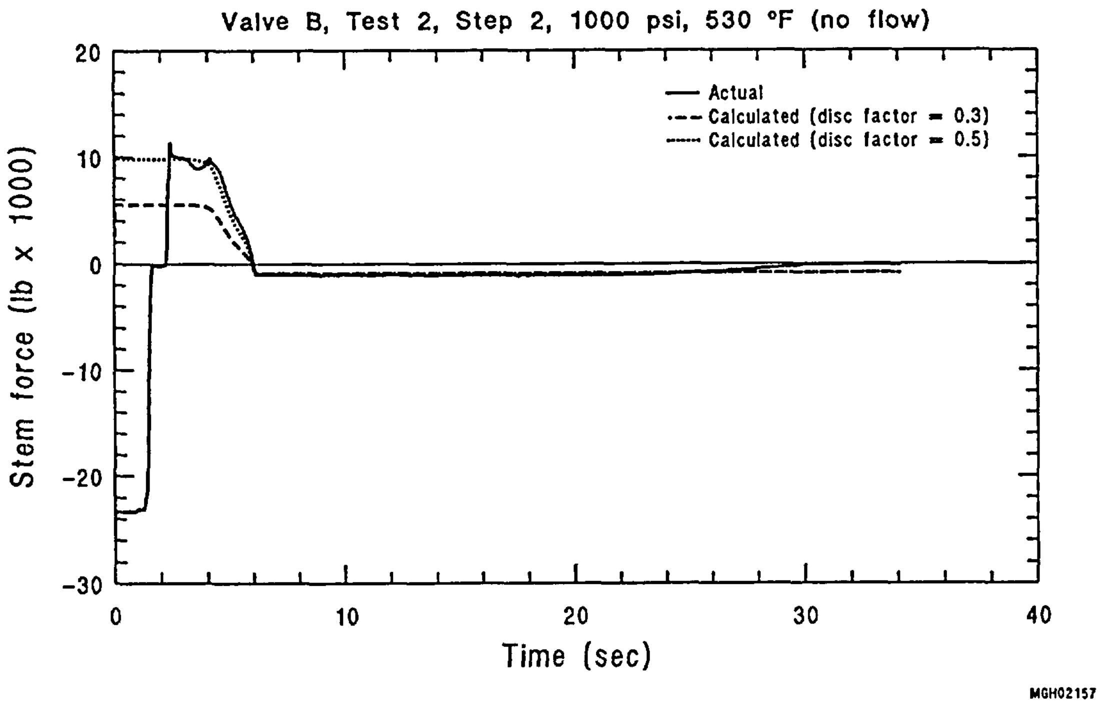

Figure 24. The Valve B ΔP opening test at normal BWR operating conditions, like the high flow interruption test (see following figure), appears to isolate the disc factor at or near 0.5.

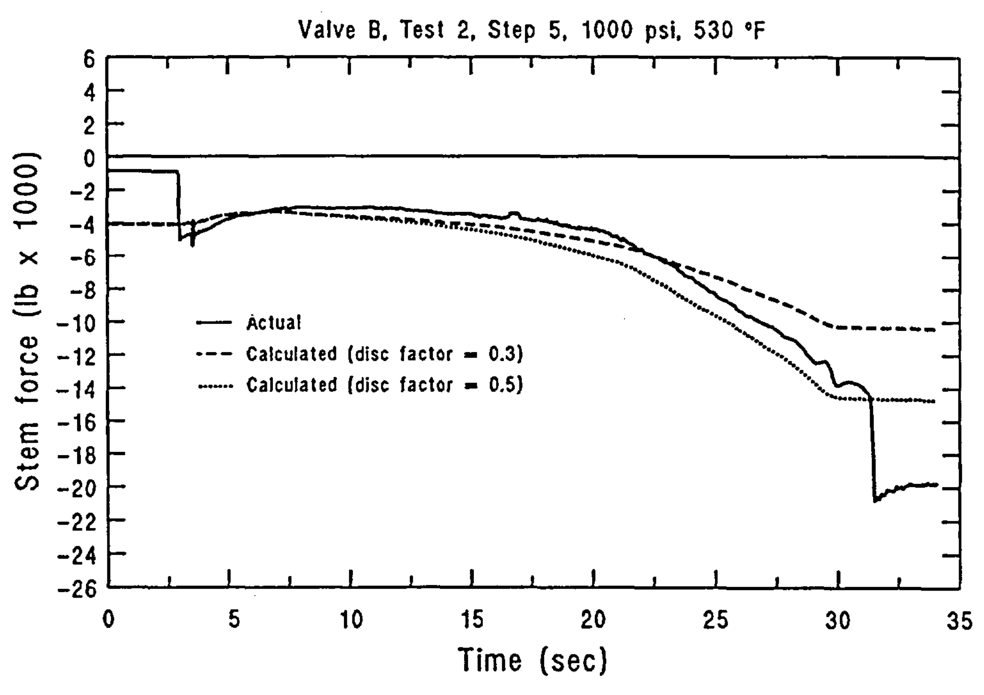

Figure 25. For the Valve B pipe break flow interruption test at normal BWR operating conditions, the relationship between measured and calculated stem thrust is very similar to that seen in the ΔP opening test (previous figure).

The test results indicate that valve opening and opening versus closing comparisons are dependent on the following factors, which unfortunately may make the test infeasible for some valve locations.

1. The valve and fluid have to be at operating temperature and pressure since cold water tests do not produce the same results.

2. Type testing must have been performed to verify that the valve design exhibits linear disc friction characteristics.

3. The inlet pressure source must be large enough for in–situ testing so that the increasing leakage as the valve opens does not drop the inlet pressure significantly, before maximum disc load is determined.

4.5 Stem Factor

The test results were also analyzed to determine the adequacy of the torque to thrust conversion [Equation (2)], specifically the stem factor term. As previously stated, valve and motor operator manufacturers use one of two basic coefficients of friction in these calculations, 0.15 or 0.20. The 0.15 constant coefficient of friction was found to be conservative. The 0.20 value would be considered very conservative unless the thrust, which might be obtained from a motor operator sized for a 0.20 coefficient and operating with a 0.10 coefficient, overstressed the valve. While many people believe the stem factor is a constant, we found that the stem factor actually varies with valve loading.

We believe stem factor will surface as a problem as maintenance and motor operator diagnostic testing is performed. It is well known that as a motor operator ages and as maintenance is performed, the correspondence between the torque switch setting and the delivered output thrust becomes less reliable. Modern diagnostic test equipment for valve motor operators has allowed the utilities to recalibrate the motor operator torque switch in situ. However, the variability of the stem factor and the deceptively high thrusts of valve seat–induced torqueouts could result in improperly set motor operator control switches. Test results and the following analysis point out some of the ways in which the stem factor can be a problem in recalibrating motor operator output torque. Valve B was selected for this analysis because of its linear performance during both the opening and closing tests and because the Valve B motor operator was normally sized for the loadings. This allowed adequate torque spring deflection and good operator output torque to stem thrust comparisons.

Analysis of our test results showed that the final thrust in the valve stem varied depending on how the motor operator was loaded before and at torque switch trip. Initially we believed this variability to be a function of motor operator momentum; however, the measured motor operator parameters did not bear this out. The measured parameters did show that when the valve was lightly loaded before the moment the disc wedged in the seat, the stem factor was low. When the valve was highly loaded before seating, the stem factor was higher, resulting in a poorer conversion of the torque to thrust. With high loads prior to valve sealing, the torque spring was deflected by the disc load almost to the point of torque switch trip before the disc first contacted the seat. The initial contact with the seat, combined with the disc loads, was enough to trip the torque switch. From this point to the time the motor controller drops out and the motor operator momentum is spent, the worm acts like the input to a planetary gear where the remaining revolutions of the motor are split between the worm turning the stem nut and the worm climbing the worm gear and compressing the torque spring past the torque switch trip point (see Figure 26). With light loads, however, the disc is already wedged very tightly at torque switch trip, and the remaining revolutions of the motor are not split but all go into overcompression of the torque spring, and thus the resulting final stem forces are higher. The test results discussed below are consistent with this explanation.

Before the start of the qualification test, we set the torque switch to deliver 18,000 lb of thrust, as specified by the valve manufacturer for a full flow closure at 1,400 psig. In setting the torque switch, we used the load cell installed in the valve stem to measure the thrust, and we manually turned the handwheel to close and seat the valve, so there was no motor momentum involved with the determination of the torque switch position versus output thrust relationship.

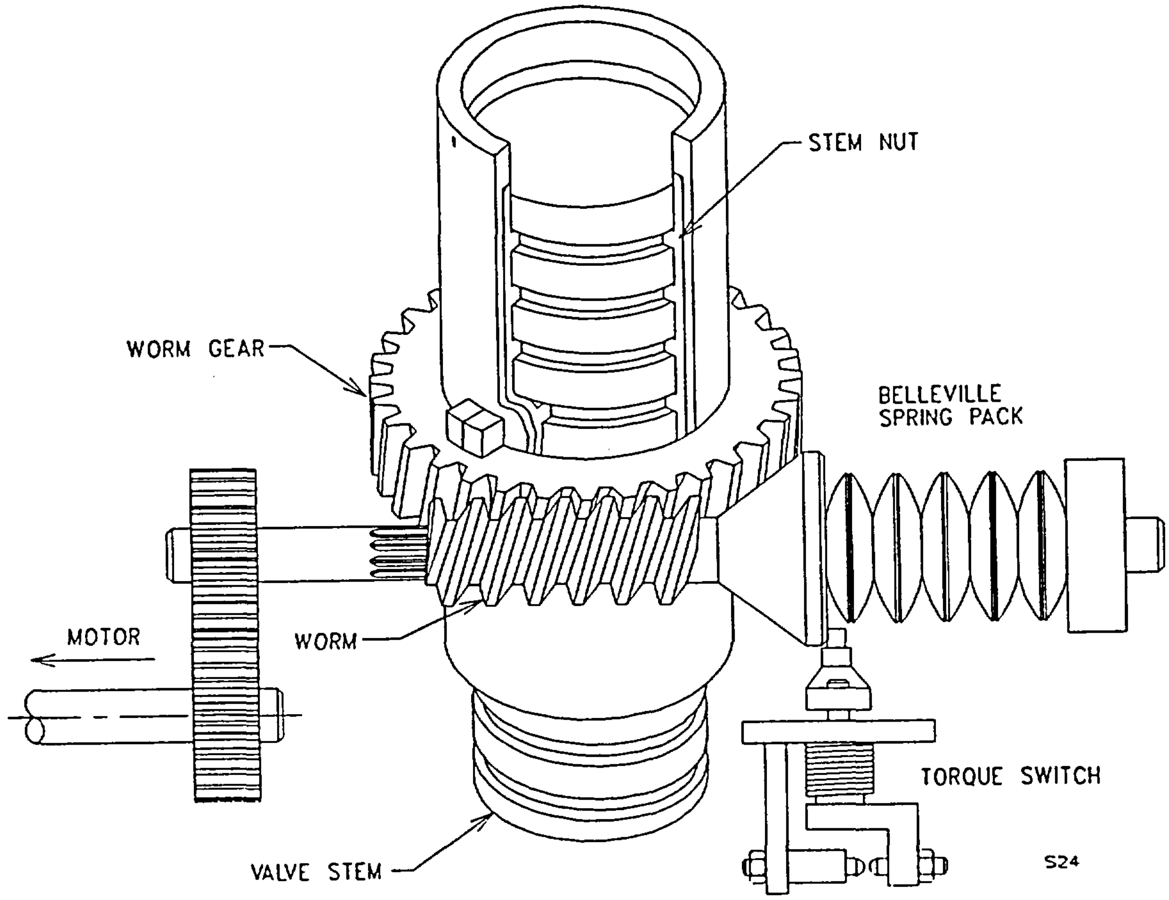

Figure 26. At the torque switch trip, the worm may either turn the worm gear and drive the disc deeper into the seat, or (if the disc will move no further) climb the worm gear, overcompressing the torque spring and producing additional thrust in the valve stem.

Figure 27 shows the forces measured as the valve closed against pressure only. This test is typical of what a utility might be able to do. Note the final thrust (22,000 lb) with the valve lightly loaded and with torque switch trip induced by the disc wedging in the valve seat. When the disc contacts the seat and the torque switch is tripped, power continues to be supplied to the motor until the motor controller drops out (typically a time lag of 15 to 60 ms). At this time, the valve disc is wedged deeply in the seat. After the motor electrical power is broken, there is a period of deceleration of the motor operator components. This deceleration is proportional to the speed and mass of the motor operator, primarily the motor. (With fast acting valves, there can also be significant momentum in the valve internals.)

Both the dropout time of the motor controller and the motor operator momentum show up as additional force after torque switch trip; these additional revolutions of the motor produce little if any additional movement of the disc and instead result in overcompression of the torque spring. In seismically qualified valves with very stiff yokes, the motion is divided between overcompression of the torque spring and compression of the stem. In addition, the low stem factor that accompanies these relatively low valve loadings allows a better conversion of torque to thrust, producing a higher measured force in the valve stem at torque switch trip.

In Figure 28, we see the forces measured as the same valve closed against three different pressures at high flows. Note that with the same torque switch setting, the force when the torque switch tripped in the 600 psig test with high flow is less, at 18,100 lb, than the force when the torque switch tripped in the no-flow static pressure test, at 19,900 lb (see Figure 27).

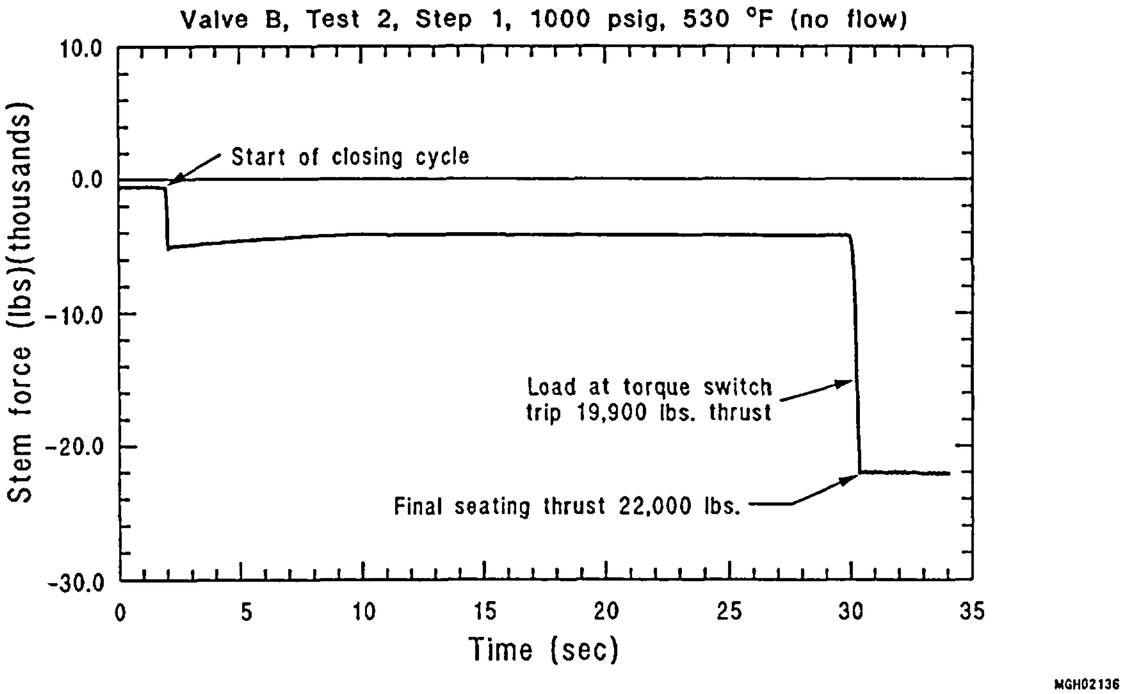

Figure 27. Though the torque switch was set to trip at 18,000-lb thrust in the absence of operator momentum, this lightly loaded valve achieved a significantly higher final seating thrust.

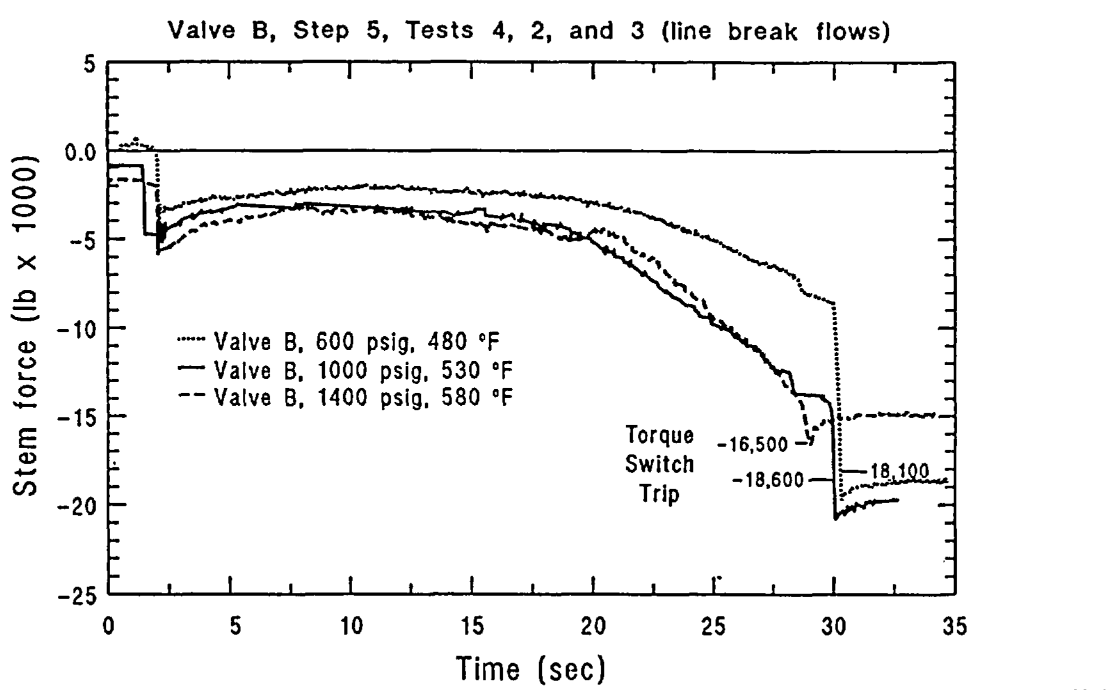

Figure 28. As stem factor and operator momentum are affected by increased loadings, the final thrust is less even though the torque switch setting is the same.

The valve closing at 1,000 psig shows a significantly higher load before isolation of flow. Just before this test, the valve stem was lubricated, and a slightly higher thrust (18,600 lb) was obtained when the torque switch tripped. However, the valve stem position and the subsequent reopening of the valve indicated that the valve was lightly seated and the measured force was a reflection more of closing load than of seating load. During the closing at 1,400 psig inlet pressure (the design basis for operator sizing and torque switch setting), the valve marginally isolated flow but did not seat; the operator tripped on disc friction. The thrust when the torque switch tripped was lower, at 16,500 lb, a 17% reduction in the thrust at torque switch trip and a 25% reduction in final thrust, as compared to the lightly loaded case, shown in Figure 27. The diagnostic equipment monitoring the operator performance showed that in contrast to the varying thrust, the operator output torque varied less than 3% for all valve loadings.

The variability of the stem factor under changing valve stem loads is shown in Figure 29. This stem factor history is derived from measurements of stem force and measurements of torque spring deflection mathematically converted to operator torque. This figure shows that the stem factor increased with load, resulting in a less efficient conversion of operator torque to valve stem thrust. In this and other stem factor versus load comparisons, it appears there may be a proportional relationship between the increase in load and the increase in stem factor.

4.6 Operator Torque Switch Trip Anomaly

During the flow interruption testing of Valve A, there were three incidences of anomalous operator torque switch trip behavior. It is believed these incidences occurred in conjunction with installation and removal of the MOV diagnostic test equipment. The valve stem forces associated with the torque switch trip were normal in the numerous tests performed with diagnostic devices installed. The anomaly appeared in the form of abnormally low values of the torque–out stem forces during the tests immediately after removal of two types of diagnostic equipment (see Table 2 for installation and removal sequence of diagnostic equipment). The

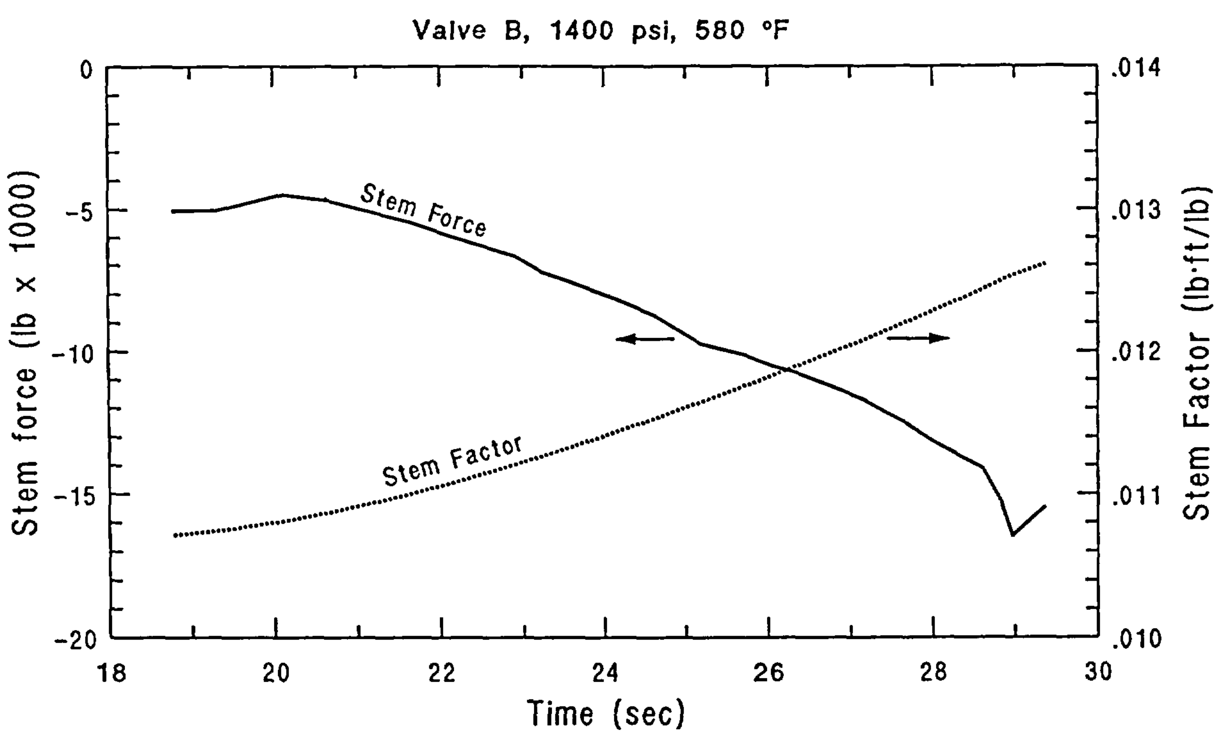

Figure 29. The stem factor increases with load, reducing the available thrust at a given torque.

investigation that followed the discovery of the low stem forces showed that incorrect installation of the motor operator spring pack lock ring was the problem. The removal of the diagnostic test equipment and the subsequent incorrect installation of the lock ring invalidated the findings of the diagnostic test. A recent problem and investigation at Brunswick (LER 87–023–01)[2] identified a similar lock ring installation problem and illustrates the potential for invalidated diagnostic testing and out–of–calibration torque switch positions.

The point at which the torque switch contacts open depends only on the setting of the torque switch, spring constant of the torque spring, and spring pack preload and/or gap. No matter what causes the stem force to increase, whether flow loads, valve reaching full stroke, or even an obstacle in the disc path, the switch will always open when the torque spring compresses to the predetermined point. The force at torque switch trip was used to trace the function of the operator from one test to another through various diagnostic equipment installations and removals. Figure 30 shows the average stem compression at torque switch trip for each of the eleven tests performed on Valve A, arranged in chronological order. The force measurements were made using the INEL load cell installed as an integral part of the valve stem.

During tests 1 through 7 the valve operator functioned consistently, with a stem compression at torque switch trip of approximately 33,000 lb. Tests 9 and 8, accomplished without operator diagnostic monitoring, showed consistent torque–out forces, but at a significantly reduced level. Here a drop of approximately 10,000 lb appeared in the torque–out stem compression.

Two different sets of valve operator diagnostic equipment were installed to monitor test 10, and the valve stem torque–out compression returned to about the same level as tests 1 through 7 (relubrication of the valve stem threads increased loads slightly). The diagnostic equipment was removed after test 10, and the results of test 11 show a similar reduction in force, even after the torque switch setting was increased from 2.0 to 2.5.

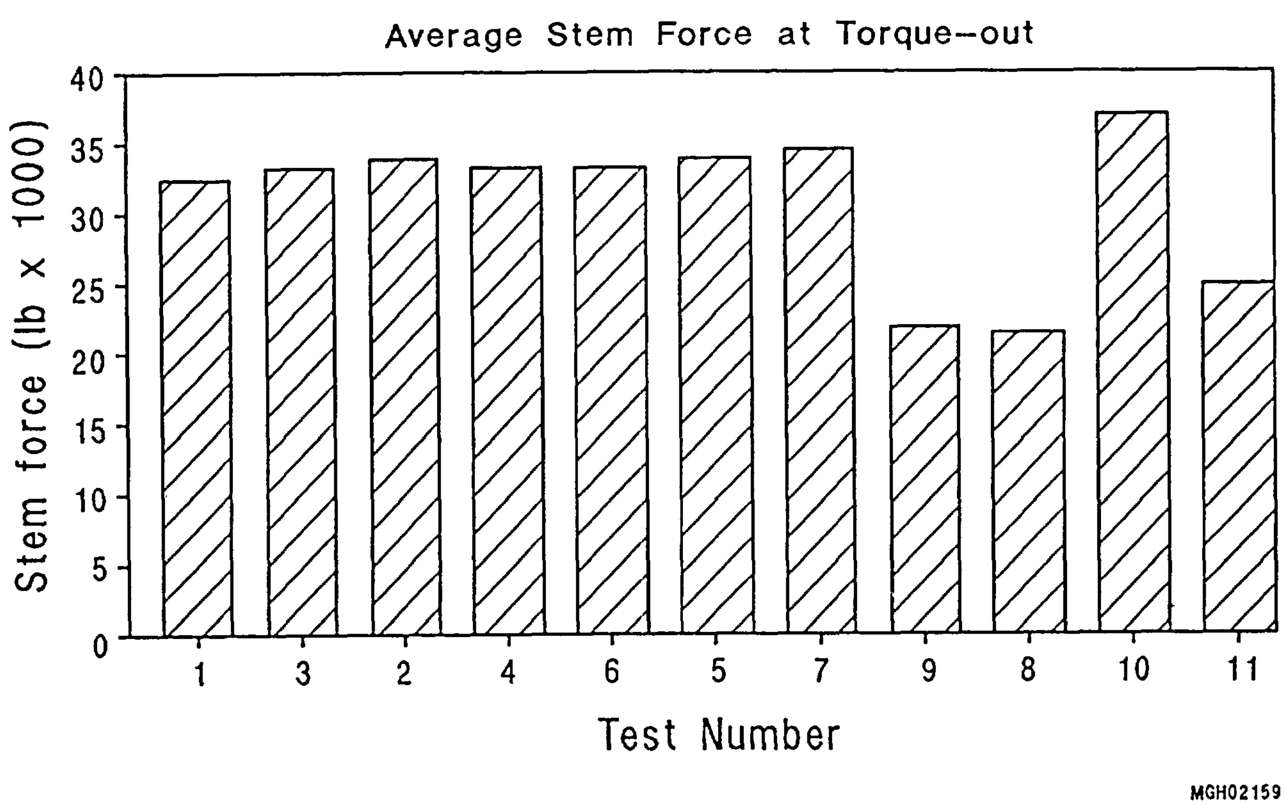

Figure 30. Below–normal stem forces at torque switch trip were observed for three test series with Valve A.

After completing testing for Valve A, the Limitorque motor operator was removed and partially disassembled by Limitorque representatives, with INEL personnel attending. The spring pack cover was removed and the internal configuration inspected. The lock ring that retains the torque spring and its locking set screw appeared to be properly installed. The set screw was removed and a special tool was used to attempt to further tighten the lock ring. The ring was tightened almost one full turn before it reached its proper position.

Limitorque design documents were used to correlate the loosening of the lock ring to torque switch setting and torque–out thrust. One full turn of the lock ring is equivalent to 19 degrees rotation of the torque switch; one full torque switch setting is about 21 degrees. This loosening of the lock ring had the effect of backing off the torque switch from 2.0 (the actual setting) to 1.1 (the equivalent setting). From the torque spring curve the loss of thrust was estimated at 10,600 lb, very close to the discrepancy in the measured data.

Improper positioning of the lock ring sometime after test 7 but before the next test would explain the reduction in stem force after test 7. How it happened is not completely understood. None of the diagnostic devices installed before test 9 required the removal or adjustment of the torque spring lock ring; in fact, several of the devices are designed to detect spring pack gap, the result of improper lock ring installation. Review of the data taken by the various diagnostic devices shows no indication of spring pack gap. Also, none of the devices are designed such that their installation would correct this problem, with the exception of the Limitorque motor actuator characterizer (MAC) device, which was installed for test 10.

Installation of the MAC device requires the removal of the torque spring lock ring to facilitate the installation of its spring pack load cell device. According to the Limitorque technician, the position of the lock ring was marked before removal and the number of turns during removal was noted. The load cell device was installed and tightened to the proper position to provide the design spring preload. After testing, the load cell was removed and the lock ring was installed the appropriate number of turns to the previously marked position. We believe this explains the similar reduction in stem force before and after test 10.

A similar problem at Brunswick was evaluated using the information found in LER 87–023–01. In this case, the HPCI steam line isolation valve (a GI–87 valve) had successfully undergone several diagnostic tests using the MAC system. Later, the valve motor failed on opening for an unrelated reason. During the subsequent motor operator check, greatly reduced torque–out forces were measured. Investigating personnel discovered that a burr on the threads of the spring pack housing cover had prevented the lock ring from being fully installed after diagnostic testing and had caused the lower–than–expected torque readings.

Both the GI–87 testing and Brunswick instances of improper lock ring positioning could have been easily diagnosed. A simple measurement of the lock ring position can be compared with both the position of the torque spring transducer during testing and the manufacturer design position in order to validate post–test valve operation. Apparently this procedure was not completed for the tests described above.

5. CONCLUSIONS

The typical industry sizing equation using the standard variables did not conservatively estimate the total thrust needed to close the tested valves; disc factors higher than the normal 0.3 disc factor (μ_d) were encountered. The valve thrust equation [Equation (1)] needs to better model the behavior of valves exposed to slightly subcooled fluid conditions in BWRs. The disc factor needs to be increased for both the opening and closing direction to account for the higher loads associated with high temperature operation. The thrust sizing equation is not applicable to valves that sustain damage (such as galling and plastic deformation of the sliding surfaces) at design basis loadings. Flashing and two-phase flow appear to add a yet unquantified factor to the closing load.

The design basis hot water blowdown testing has shown that, given enough thrust, typical gate valves will close against the high flow resulting from a line break. Proper operator sizing depends on correct identification of the values for the sizing equation. Evidence exists that values used in the past may not be conservative for all valve applications, especially at design basis loadings. The following items need to be considered during sizing of gate valve operators:

1. Gate valve guide design and clearances can have a significant effect on the operator stem thrust requirements at design basis fluid loadings.

2. The degree of subcooling at the valve inlet can greatly influence valve closure forces. Valve operator force requirements increase as inlet fluid conditions approach saturation temperatures.

3. Industry trends toward using 100% system pressure for all pressure terms in the sizing calculation are justified for high-flow applications.

Tests have shown that some form of valve type testing outside the plant might be necessary to establish specific valve design thrust requirements and verify that a given valve design exhibits linear characteristics when subjected to design loads. For the valves that have a linear thrust response, valve opening tests (with a full pressure drop and no flow) at normal operating temperatures performed with valve diagnostic test equipment can provide insights for the valve disc factor and therefore degradation in valve performance for both opening and closing. Diagnostic test equipment that measures both thrust and torque will provide a better set of measured values for analysis.

Contrary to common belief, the ratio of operator torque to stem thrust [stem factor (μ_s)] is not a constant but changes with valve loading. The effect of a changing stem factor on in-plant testing is significant. In-situ testing loads (the thrust developed when the process of valve seating causes the torque switch to trip) typically result in low stem factors and high stem thrusts. A utility might set the torque switch to a lower value to protect the valve from excessive thrust at normal loadings due to both the high stem thrust at torque switch trip and large increase in force due to momentum loading after torque switch trip. Then, if the valve is subjected to design basis loadings, the stem factor will increase, and the thrust may be too low. Diagnostic systems that measure operator torque and thrust would provide stem factor information for extrapolation, thus assuring that, if the valve is subjected to design basis loadings and the stem factor increases, the valve will have sufficient thrust to close.

Although not conclusive, these test results suggest that the change in stem factor may be predictable. If the relationship between stem factor and load can be established, the change in stem factor could be quantified for design calculations and in-situ testing.

Improper operator lock ring installation following test or maintenance can invalidate in-situ test results and render the valve unable to perform its design function. This is important in light of the present trend by utilities to perform diagnostic testing of safety-related valve assemblies to answer regulatory concerns such as those expressed by IE Bulletin 85-03. A final quality check following diagnostic testing and maintenance must be made to ensure correct lock ring installation.

6. REFERENCES

1. ANSI B16.41, *Functional Qualification Requirements for Power Operated Active Valve Assemblies for Nuclear Power Plants*, 1983.

2. Brunswick Steam Electric Plant Unit 1, *Inoperability of High Pressure Injection (HPCI) System (E41) Due to Failure of HPCI Turbine Steam Inlet Isolation Valve E41–F001*, LER 87–023–01, June 27, 1988.

NRC FORM 335 (2-89) NRCM 1102, 3201, 3202

U.S. NUCLEAR REGULATORY COMMISSION

BIBLIOGRAPHIC DATA SHEET

(See instructions on the reverse)

1. REPORT NUMBER (Assigned by NRC. Add Vol., Supp., Rev., and Addendum Numbers, if any.)

NUREG/CR-5406
EGG-2569
Vol. 1

2. TITLE AND SUBTITLE

BWR Reactor Water Cleanup System Flexible Wedge Gate Isolation Valve Qualification and High Energy Flow Interruption Test

Analysis and Conclusions

3. DATE REPORT PUBLISHED

MONTH	YEAR
October	1989

4. FIN OR GRANT NUMBER

A6857 and B5529

5. AUTHOR(S)

Kevin G. DeWall

Robert Steele, Jr.

6. TYPE OF REPORT

Technical

7. PERIOD COVERED (Inclusive Dates)

8. PERFORMING ORGANIZATION – NAME AND ADDRESS (If NRC, provide Division, Office or Region, U.S. Nuclear Regulatory Commission, and mailing address; if contractor, provide name and mailing address.)

Idaho National Engineering Laboratory
EG&G Idaho, Inc.
P.O. Box 1625
Idaho Falls, ID 83415

9. SPONSORING ORGANIZATION – NAME AND ADDRESS (If NRC, type "Same as above"; if contractor, provide NRC Division, Office or Region, U.S. Nuclear Regulatory Commission, and mailing address.)

Division of Engineering and Division of Safety Issue Resolution
Office of Nuclear Regulatory Research
U.S. Nuclear Regulatory Commission
Washington, DC 20555

10. SUPPLEMENTARY NOTES

11. ABSTRACT (200 words or less)

This report presents the measured data and the analyses performed to date on the full-scale high-energy qualification and flow interruption gate valve testing to develop technical insights for the United States Nuclear Regulatory Commission (USNRC) effort regarding Generic Issue 87 (GI-87). The research was sponsored by the USNRC[a] and conducted by researchers from the Idaho National Engineering Laboratory. We tested two 6-in., 900-lb class valve assemblies, which represent a significant percentage of the reactor water cleanup isolation valves installed in plant applications. These valves were modified before testing by adding a high temperature load cell in the valve stems, which allowed the direct measurement of valve stem thrust during both opening and closing valve cycles. Instrumentation installed in the flow loop and on the valve assemblies measured the important valve and system test responses. Additionally, during the test program, all of the currently popular motor operated valve diagnostic test systems monitored the performance of the valves. Initially the valves were subjected to the hydraulic and leakage qualification tests defined in ANSI B16.41 and then to flow interruption and reopening valve tests at boiling water reactor primary system water temperature and pressure conditions with downstream line break flows. For the two valves tested, results show that (a) the disc factor used in current industry motor operator sizing equations underpredicts actual valve thrust requirements at all high temperature loadings, and for one valve design the equations may require an additional term to account for nonlinear performance, (b) the thrusts required to close the valves were sensitive to the fluid temperature, and (c) the results of testing at lower pressures, temperatures, and flows cannot be extrapolated to design basis pressures, temperatures, and flows for valve designs that have not exhibited linear performance behavior during design basis prototypical testing.

12. KEY WORDS/DESCRIPTORS (List words or phrases that will assist researchers in locating the report.)

valve testing
BWR reactor water cleanup
Generic Issue 87 (GI-87)

13. AVAILABILITY STATEMENT

Unlimited

14. SECURITY CLASSIFICATION

(This Page) Unclassified

(This Report) Unclassified

15. NUMBER OF PAGES

16. PRICE

UNITED STATES
NUCLEAR REGULATORY COMMISSION
WASHINGTON, D.C. 20555

OFFICIAL BUSINESS
PENALTY FOR PRIVATE USE, $300

SPECIAL FOURTH-CLASS RATE
POSTAGE & FEES PAID
USNRC
PERMIT No. G-67

NUREG/CR-5406, Vol. 1

BWR REACTOR WATER CLEANUP SYSTEM FLEXIBLE WEDGE GATE ISOLATION VALVE QUALIFICATION AND HIGH ENERGY FLOW INTERRUPTION TEST

OCTOBER 1989