COMMITTEE ON THE SAFETY OF NUCLEAR INSTALLATIONS

PRINCIPAL WORKING GROUP No 1

PROCEEDINGS OF THE JOINT SPECIALIST MEETING ON MOTOR-OPERATED VALVE ISSUES IN NUCLEAR POWER PLANTS

Held in Paris, France
25th-27th April, 1994

FOR TECHNICAL REASONS, THIS DOCUMENT IS NOT AVAILABLE ON OLIS.
COMMITTEE ON THE SAFETY OF NUCLEAR INSTALLATIONS

The Committee on the Safety of Nuclear Installations (CSNI) of the OECD Nuclear Energy Agency (NEA), is an international committee made up of senior scientists and engineers. It was set up in 1973 to develop and coordinate the activities of the Nuclear Energy Agency concerning the technical aspects of the design, construction and operation of nuclear installations insofar as they affect the safety of such installations. The Committee's purpose is to foster international cooperation in nuclear safety among the OECD Member countries.

The CSNI constitutes a forum for the exchange of technical information and for collaboration between organizations which can contribute, from their respective backgrounds in research, development, engineering or regulation, to these activities and to the definition of its programme of work. It also reviews the state of knowledge on selected topics of nuclear safety technology and safety assessment, including operating experience. It initiates and conducts programmes identified by these reviews and assessments in order to overcome discrepancies, develop improvements and reach international consensus on technical issues of common interest. It promotes the coordination of work in different Member Countries including the establishment of cooperative research projects and results to participating organizations. Full use is also made of traditional methods of cooperation, such as information exchanges, establishment of working groups, and organization of conferences and specialist meetings.

The greater part of the CSNI's current programme of work is concerned with safety technology of water reactors. The principal areas covered are operating experience and the human factor, reactor coolant system behaviour, various aspects of reactor component integrity, the phenomenology of radioactive releases in reactor accidents and their confinement, containment performance, risk assessment, and severe accidents. The Committee also studies the safety of the nuclear fuel cycle, conducts periodic surveys of the reactor safety research programmes and operates an international mechanism for exchanging reports on safety related nuclear power plant accidents.

In implementing its programme, the CSNI establishes cooperative mechanisms with NEA's Committee of Nuclear Regulatory Activities (CNRA), responsible for the activities of the Agency concerning the regulation, licensing and inspection of nuclear installations with regards to safety. It also cooperates with NEA's Committee on Radiation Protection and Public Health and NEA's Radioactive Waste Management Committee on matters of common interest.
OECD/NEA-IAEA Specialist Meeting on Motor Operated Valve Issues in NPPs
Paris, France, April 25th-27th, 1994

FINAL PROGRAMME

Monday, 25th April, 1994

Morning

9:00 - 9:30  Registration
9:30 - 9:50  Welcome address
9:50 - 10:20 Coffee break

Session #1: Regulatory Activity (Chairman P. JAMET, IPSN)


11:00 - 11:30  Paper #2:SP18  "Reevaluation of Motor-Operated Valves in Spain", A. PEREZ RODRIGUEZ


12:00 - 13:30  Lunch

Session #2: Operating Experience (Chairman, V. TOLSTYKH, IAEA)

Afternoon

13:30 - 14:00  Paper #4: FR5  "Main Findings on MOV Operating Experience in France, R. ZERMIZOGLOU

14:00 - 14:30  Paper #5: HU10  "Valves Maintenance in NPP Paks", G. NEMETH

14:30 - 15:00  Paper #6: US24  "Pressure Locking and Thermal Binding of Gate Valves in the United States", E.J. BROWN (presented by J. ROSENTHAL)

15:00 - 15:20  Coffee break
15:50 - 16:20 Paper #8: US26  "Resolving AOV Problems at LaSalle Station", M. SMITH, M. MURPHY

Tuesday, 26th April, 1994

Morning

Session #3: MOV Improvement Programme (Chairman, K. KOTTHOFF, GRS)

8:30 - 9:00 Paper #11: SW19  "Review of Safety Related Valves for Design Basis Events", G. DENFORS, N. RAUFFMANN, P. KRADEPOHL (presented by G. DENFORS)
9:00 - 9:30 Paper #34:  "Air Operated Valve Preventative Maintenance Program Development/Implementation at Ontario Hydro's Bruce Nuclear Generating Station" B.J. FERGUSON, W. FITZGERALD
9:30 - 10:00 Paper #13: FR3  "Utility and Design Efforts to Improve French MOV Nuclear Power Plants", COPPOLANI, GRENET, RENAUDIER, UHART (presented by Mr. GRENET)
10:00 - 10:30 Coffee break
10:30 - 11:00 Paper #14: SP17  "Valves Reevaluation Program", V. BARBERO
11:30 - 12:00 Paper #16: US34  "Risk Based Approach for Prioritizing Motor-Operated Valves", G.H. WEIDENHAMER
12:00 - 13:30 Lunch

Afternoon

Session #4a): Research and Development (Chairman, J. ROSENTHAL, NRC)

13:30 - 14:00 Paper #17: IAEA37  "IAEA Co-ordinated Research Programme on the Management of Motor Operated Isolating Valves", A. KOSSILOV
<table>
<thead>
<tr>
<th>Time</th>
<th>Paper #</th>
<th>Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>14:00 - 14:30</td>
<td>#18 US21</td>
<td>&quot;U.S. NRC-Sponsored Research Look at Nonlinear Gate Valve Response&quot;, J.C. WATKINS, R. STEELE Jr., K.G. DEWALL</td>
</tr>
<tr>
<td>14:30 - 15:00</td>
<td>#19 RU15</td>
<td>&quot;Assessment of NPP Operational Events due to Valve Failures and Major Areas of the Program to Improve NPP Valves&quot;, B.N. TIUNIN, A.A. ZRELKIN (not presented)</td>
</tr>
<tr>
<td>15:00 - 15:30</td>
<td>#20 US28</td>
<td>&quot;The MOV Disk Factor and Enhancement Methodologies&quot;, C.L. THIBAULT</td>
</tr>
<tr>
<td>15:30 - 15:50</td>
<td></td>
<td>Coffee break</td>
</tr>
<tr>
<td>15:50 - 16:20</td>
<td>#21 US33</td>
<td>&quot;Effects of Internal Corrosion on Motor-Operated Valve Operability&quot;, G.H. WEIDENHAMER</td>
</tr>
<tr>
<td>16:20 - 16:50</td>
<td>#22 FR4</td>
<td>&quot;Electric Spring Return Actuators for Safety Applications&quot;, C. FICHTENBERG</td>
</tr>
<tr>
<td>16:50 - 17:20</td>
<td>#23 US27</td>
<td>&quot;Non-Intrusive Motor Operated Valve Diagnostic Equipment and Testing Experiences&quot;, J.J. BALASCHAK</td>
</tr>
<tr>
<td>17:20 - 17:50</td>
<td>#24 FRG8</td>
<td>&quot;EMG-DREHMO-Actuator with Controlled Friction Clutch&quot;, W. HEMPELMANN, W. HANDEL, P. ZIMMERMANN</td>
</tr>
</tbody>
</table>

**Wednesday, 27th April, 1994**

Morning

**Session #4b: Research and Development (Chairman, A. PEREZ, CSN)**

<table>
<thead>
<tr>
<th>Time</th>
<th>Paper #</th>
<th>Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>8:40 - 9:20</td>
<td>#25 US29</td>
<td>&quot;Improvement in Butterfly Valve Torque Prediction Models Based on Recent Research&quot;, M.S. KALSI, B. ELDIWANY, V. SHARMA</td>
</tr>
<tr>
<td>9:20 - 10:00</td>
<td>#26 FRG6</td>
<td>&quot;Mechanical Design Calculations for Stem Forces in Safety-Related Gate and Globe Valves with Regard to Proper Valve Function and to Load Effects on Valve Components&quot;, R. KUBOSCH, A. SCHEUER</td>
</tr>
<tr>
<td>10:00 - 10:30</td>
<td>#27 US36</td>
<td>&quot;EPRI MOV Performance Prediction Program Overview&quot;, J.F. HOSLER (EPRI)</td>
</tr>
<tr>
<td>10:30 - 11:00</td>
<td></td>
<td>Coffee break</td>
</tr>
</tbody>
</table>
Session #5: Testing and Maintenance (Chairman, T. Scarbrough) U.S. NRC

11:00 - 11:30  Paper #28: B1  "Belgian Experience on MOV's Diagnosis Methods", M. DUBOIS

11:30 - 12:00  Paper #29: JA12  "Adoption of Automatic Diagnostic System for MOVs in Nuclear Power Plants", M. TAKAI, S. TAKEDA, Y. MANABE

12:00 - 13:30  Lunch

Afternoon

13:30 - 14:00  Paper #31: JA11  "The Method for Complete Restoration of Valve Actuator Function After the Overhauling of MOVs", J. SAKAMOTO

14:00 - 14:30  Paper #32: FRG7  "Functioning, Leak Tightness, Maintenance of Gate Valves and Globe Valves in Nuclear Power Stations", W. RIEGER

14:30 - 15:00  Paper #33: US22  "Maintenance of Solenoid Operated Valves", V. VARMA

15:00 - 15:30  Coffee break

15:30 - 16:30  Final Panel Session  Mr. M. DEBES, Electricité de France - General Inspectorate for Safety (Session Leader)
Dr. J. ROSENTHAL, U.S. Nuclear Regulatory Commission, AEOD
Dr. J. HOSLER, EPRI
Mr. COPPOLANI, Electromechanical Department, Framatome

Additional papers not presented:

"Periodic Inspection and Testing for Safety-Related Motor-Operated Valves (MOV) at the NPP Beznau in Switzerland", H. GEISINGER, K. THOMA

"Some Problems of the Ukrainian States Committee on Nuclear and Radiation Safety Regulatory Activity as to NPP Armature Operation", V. GLYGALO, V. KOVYRSHIN, N. ZARITSKY, I. PRIVALKO
Thursday, 28th April, 1994

Technical Visit to EdF Research Centre

Visit of the Mechanics and Technology of Components Branch (MTC)
Les Renardières, 10 AM to 1 PM
OECD Nuclear Energy Agency

Specialist meeting on Motor Operated Valves Issues

April 25-27, 1994 - Paris

Final panel session

Concluding remarks
Introduction:

After a three days meeting, all the major issues about the performance of Motor Operated Valves have been described extensively. So there is only few things to add.

I would just remind some key points about the importance of the problem for safety and reliability of NPPs, about the technical assessment and general approach of the problem and about the remaining work.

1/ The importance of valves for NPPs safety and availability:

There are more than 10 thousand valves in a plant, but only a few hundred of them influence safety. Those valves represent 2 or 3 percent of investments. But they are responsible for a high proportion of maintenance problems, which often exceeds 20% wether in unavailability, maintenance manhours, or personnel radiation doses.

The reasons are multiple. The precise mechanisms involved are very complex. But their rules of behavior are often judged to be extremely simple and are in fact misunderstood by many people. As a result, the design margins may be inadequate. The specific requirements for safety regarding design basis conditions and qualification, high pressure, high flow and temperature, or degraded motor voltage, are difficult to assess and to predict.

It was not until the 1980s that the extent of those problems began apparent, especially after the Davis Besse event during which the MOVs in the auxiliary feedwater system could not be reopened electrically after they had been closed.

On the other hand, the potential safety significance of MOVs failure may be of paramount importance, notably due to common mode aspects if similar pairs of MOVs fail simultaneously.

Two major kinds of problems must be underlined:

- the first kind of problems is related to weaknesses in quality programs:
  They result from poor management of interfaces in the design, inadequate maintenance or operation procedures and training, bad setting of switches, misuse of manual declutch levers, lack of lubrication, lack of requalification.. As an example, I would just mention a common mode failure of the two MOVs at the suction of the 2 containment sumps due to a lack of grease on some 1300MW units in France. Those issues are central in MOVs reliability and they deserve a great attention in the operating process.

- the second kind of problems is related to inadequacies in engineering analysis and design.
  It stems from difficulties to predict the thrust and torque required to open and close valves under design basis conditions. Following incidents, the operability of MOVs under design basis conditions was extensively studied. Problems were identified such as pressure locking and thermal binding, fluid conditions... They can result in undersizing of MOVs actuators. New requirements were defined for the design, testing, inspection and maintenance of safety related MOVs.

This meeting took place at the right moment:

- we have now an extensive experience feedback and many studies, with a better understanding of the problems encountered
- solutions have been devised which are implemented on the plants, or still have to be in some cases. So it is very important to have an overall view of the state of this issue to assess the soundness of the modifications and solutions.

2/ The current technical assessment of MOVs:

As regards the current technical assessments of MOVs problems, I would stress on three points:

- the first is the importance of the reviews, studies and tests which have been launched to understand the factors affecting the operability of MOVs and to validate methodologies to predict MOVs performances. This program involves valve analytical modeling and design, installation, in plant testing, operating and maintenance aspects. Much progress has been done and resulted in a set of guidelines for industry and utilities which are now available to be implemented. Of course there are remaining questions, and studies and tests are still necessary to reduce uncertainties, or to take into account changes in materials such as elimination of satellite for dosimetry reasons.

- the second is the important role which must be played in the future by MOVs automatic diagnostic systems. They have many advantages in analysing significant data, enhancing and streamlining preventive maintenance, detecting faults before any real failure, improving dosimetry and reducing inspection time and human errors. It was stressed that the direct measurement of both stem thrust and motor torque appears to be necessary to improve the accuracy and reliability of the diagnostic. These systems will be used as a predictive tool to sort valves that require maintenance or those that perform as required.

- the third is the importance to develop a method for restoration of valve operating function after overhauling of MOVs. It includes analysis and preparation of work, rigorous checking, requalification and post maintenance testing both at equipment level and at system level.

3/ The overall approach:

As regards the overall approach followed to solve these problems, I would underline also three important aspects which in my opinion illustrates a coherent safety approach:

- the first general aspect is the fact that MOVs problems were identified following performances analysis and root causes analysis. Researchs revealed that the design basis conditions were not fully respected and that corrections were needed. This example illustrates the major importance of safety culture and experience feedback to point out weaknesses and correct them, by taking into account knew knowledge, before any real incidents. This open feedback process is one key to maintain the safety level of NPPs as required by licensing basis.

- the second general aspect is the use of a risk based analysis, developed within interdisciplinary teams including valve experts, system engineers, and plant operation staff. Results of Probabilistic Safety Analysis associated with determinisitic considerations are used to prioritize MOVs and rank them according to their importance for safety. In this approach, the risk of common cause failures must be taken into account to sort out the most important pairs of MOVs for the operability of safety functions, including containment.
- the third general aspect is the grouping of MOVs according to their importance for safety to provide a basis for proportioning the resources in a cost effective manner. It enables to adjust provisional actions, design modifications, maintenance and testing requirements which are necessary to ensure the operability of MOVs under design basis conditions. This illustrates the importance of a sound and wholly consistent process to ensure a good allocation of resources in the treatment of anomalies and non conformities, so as to be sure that the most important issues are adequately dealt with, without unjustified constraints.

4/ The domains of improvement:

Three domains appear still to be developed to reach a satisfactory situation regarding MOVs operability:

- the first is that many reviews have been performed but that in some cases effective measures still remain to be fully implemented on the plants to address the industry guidelines.

- the second is that trend analysis and experience feedback must be actively pursued by engineering teams on site to confirm in the long term the validity of solutions, associated with the use of automatic diagnostic systems.

- the third is the implementation of a rigorous QA program about MOVs operation and maintenance, which is of paramount importance to reach a good level of reliability.

5/ Conclusion:

A large part of MOVs problems are now understood. Solutions exist which must be implemented according to industry guidelines with prioritization according to the importance for safety of the different valves, together with operation and maintenance enhancement. Of course, studies are still necessary to reduce the uncertainties and monitoring of MOVs performance and operability must be actively pursued to maintain the level of safety and to confirm the validity of solutions, through clear priority for safety, vigilance and questioning attitudes.

Multidisciplinary involvement from operations, engineering and maintenance personnel, from utilities, manufacturers and constructors, beginning at the design stage, is essential for a successful program. Knowledge must be integrated from many sources such as MOVs design and materials, radiation protection (reduction of stellite), systems performance and interaction, operating procedures, maintenance practices, preventive and predictive maintenance, setting and testing, diagnostic data evaluation.

This meeting is an excellent example of cooperation between the nuclear industry and regulatory agencies from many countries. It represents a significant step for resolving the MOV issue.

A similar international review of results obtained will certainly be necessary within a few years to assess the progress made and to identify any new possible aging effect about MOVs operation.
Session #1

REGULATORY ACTIVITY

Chairman: Mr. P. Jamet

The session started with a very comprehensive presentation giving an overview of motor operated valve issues. The main aspects of the problem were highlighted: importance for safety, operating experience, regulatory organizations positions, MOV programmes.

Positions and activities of the U.S. NRC, CSN (Spain) and NII (United Kingdom) were then presented. For the U.S., programmes and inspections results related to generic letter 89-10 were described: extensive work has already been performed, problems and weaknesses have been identified, significant progress is being made by U.S. utilities. A detailed description of the CSN position was then given, as well as a precise description of the problems encountered in Spanish plants. Finally, NII highlighted the position of the U.K. safety authority in relation with the Sizewell B plant. Impressive testing programmes were presented.

From a general point of view, Session N° 1 gave the impression that MOV problems are now well identified, that significant efforts have already been made concerning this subject and that such efforts should be maintained to confirm further progress.
MOTOR-OPERATED VALVE PROBLEMS IN THE UNITED STATES
and
ACTIVITIES OF THE U.S. NUCLEAR REGULATORY COMMISSION
TO IMPROVE THE PERFORMANCE OF MOTOR-OPERATED VALVES

Thomas G. Scarbrough
Mechanical Engineering Branch
Division of Engineering
Office of Nuclear Reactor Regulation
U.S. Nuclear Regulatory Commission

Specialist Meeting on Motor-Operated Valve Issues
April 25, 1994

Abstract

The U.S. Nuclear Regulatory Commission (USNRC) requires by regulations that motor-operated valves (MOVs) important to safety be designed, fabricated, erected, and tested to quality standards commensurate with the importance of the safety functions to be performed. Despite these requirements, operating experience and research programs have revealed problems with the performance of MOVs in operating nuclear power plants. Among these problems are inadequate MOV design and incorrect torque, torque bypass, and limit switch settings that have led, or could lead, to failures of MOVs to perform their intended design-basis safety functions. This presentation summarizes MOV problems at U.S. nuclear power plants.

The USNRC staff is conducting inspections of the implementation of programs developed at U.S. nuclear power plants in response to Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance," and its supplements. A significant finding of these inspections is that utilities have found that many MOVs require more thrust to operate under design-basis differential pressure and flow conditions than has been predicted by the standard industry equation with typical valve factors assumed in the past. The USNRC staff has found weaknesses in utility procedures for conducting the differential pressure and flow tests, the acceptance criteria for the tests in evaluating the capability of MOVs to perform their safety functions under design-basis conditions, and feedback of the test results into the methodology used by the utility in predicting the thrust requirements for other MOVs. The USNRC staff also has found that utilities have not progressed sufficiently in resolving concerns about the potential for pressure locking and thermal binding of gate valves. This presentation summarizes activities of the USNRC to ensure that MOVs are capable of performing their design-basis safety functions at U.S. nuclear power plants.

This presentation was prepared by an employee of the United States Nuclear Regulatory Commission. It presents information that does not represent a new staff position. USNRC has neither approved nor disapproved its technical content; however, it is consistent with current staff positions.
OVERVIEW OF MOTOR-OPERATED VALVES

Most fluid systems at nuclear power plants depend to a large extent on the successful operation of motor-operated valves (MOVs) in performing their system functions. MOVs are used in various applications to ensure plant safety and to maintain plant availability. For example, MOVs may be required to open to allow cooling water to be provided to the reactor core, steam generators, or containment building. MOVs may be required to open to allow steam flow for turbine-driven pumps in safety systems providing cooling water to the reactor core, steam generators, or containment building. MOVs may be required to close to prevent loss of coolant from the reactor core or to isolate the reactor containment. Other MOVs may be used to control flow in order to maintain the proper balance of fluids for the production of electric power. To ensure plant safety and to maintain plant availability, MOVs must be capable of performing their functions under design-basis conditions, which may include high fluid differential pressure and flow, high ambient temperature, and degraded motor voltage.

There are several types of valves operated by motor-actuators in nuclear power plants. The most common of these valves in the U.S. are gate, globe, and butterfly valves. There are also various designs of gate, globe, and butterfly valves. For example, gate valves may have flexible wedge, solid wedge, or parallel disks. Globe valves may have flow under or over the disk. Butterfly valves may have disks with symmetric or asymmetric shapes, and offset stems.

A motor can be used to drive the gears in an actuator to open or close the valve. The torque output of the actuator is converted to thrust to operate gate and globe valves. Butterfly valves typically require a second set of actuator gearing to provide the proper torque to rotate the valve disk. After receiving a signal to operate, the motor is typically controlled by the torque applied by the actuator or by the number of rotations of gears in the actuator.

The single assembly of the motor, actuator, and valve is referred to as a motor-operated valve (or MOV). Several firms manufacture various sizes and types of motors, actuators, and valves. Therefore, MOVs can be designed and manufactured for a wide range of applications.

The complex nature of the MOV and the varied conditions under which it must operate demand that careful attention be paid to all applicable activities from design to replacement in order to ensure reliable operation. In the design of the MOV, a suitable analysis must be performed using valid engineering equations and parameters to ensure that the MOV will operate, as intended, under normal plant operations and during design-basis events. Manufacture, installation, preoperational testing, operation, in-service testing, maintenance, and replacement of the MOV must be conducted by trained personnel using proper procedures. Surveillance and testing criteria must be applied on a soundly based frequency in a manner that suitably detects questionable operability or degradation of the MOV. Moreover, these activities must be conducted in accordance with a strong quality assurance
Operating experience at nuclear power plants has revealed weaknesses in many activities associated with MOV performance. For example, some engineering analyses used in the initial design sizing and setting of MOVs were inadequate in predicting the thrust and torque required to open and close valves under design-basis conditions. Shortcomings in maintenance programs, such as inadequate procedures and training, have also resulted in poor MOV performance. Typical inservice testing consisting of stroke time measurement under zero differential pressure and flow conditions has been shown to be insufficient to detect certain deficiencies that could prevent MOVs from performing their safety functions under design-basis conditions. Given these and other weaknesses, increased attention was needed to resolve concerns with respect to the reliability of MOVs in nuclear power plants.

For many years, the nuclear industry and its regulators were aware of problems with the performance of MOVs. However, it was not until the 1980s that the extent of those problems began to become apparent. For example, a complete loss of main and auxiliary feedwater (AFW) occurred at the Davis-Besse Nuclear Power Station on June 9, 1985, during which time the MOVs in the AFW system could not be reopened electrically after they had been inadvertently closed. At Unit 2 of the Catawba Nuclear Station on March 14, 1988, an MOV in the AFW system failed to close completely against high differential pressure and flow, and subsequent testing revealed that other MOVs in the AFW systems of Catawba Units 1 and 2 were unable to close under those conditions. At the Palisades Nuclear Plant on November 21, 1989, an MOV used to isolate the power-operated relief valve failed to close under high differential pressure and flow conditions during a postmodification test. Many more MOV problems have occurred or have been identified over the last few years.

The potential safety significance of MOV failure, the complex phenomena and other factors affecting MOV performance, the wide variety of MOV problems, and the slow progress in resolving those problems led the nuclear industry and regulatory agencies in several countries to establish comprehensive programs to gain assurance that MOVs would perform well in nuclear power plants. For example, in 1989, the U.S. Nuclear Regulatory Commission (USNRC) issued Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance," which requested that U.S. nuclear power utilities establish and implement programs to ensure the capability of MOVs in safety-related systems by reviewing MOV design bases, verifying MOV switch settings initially and periodically, testing MOVs under design-basis conditions where practicable, improving evaluations of MOV failures and necessary corrective action, and determining trends in MOV performance. The USNRC staff requested that nuclear utilities complete the GL 89-10 program within three refueling outages or five years from the issuance of the generic letter, whichever is later.

Nuclear regulatory agencies in a number of countries have placed increased emphasis on ensuring the proper performance of MOVs. The nuclear industry has also recognized the need for increased attention to MOVs and has established programs to improve MOV performance in nuclear power plants. The nuclear industry and regulators throughout the international community have been sharing information on MOV problems and improvements. This NEA/IAEA-sponsored
meeting of MOV specialists is an excellent example of the cooperation between the nuclear industry and regulatory agencies from many countries and represents a significant step toward resolving the MOV issue.
The U.S. Nuclear Regulatory Commission (USNRC) requires by regulations that motor-operated valves (MOV) important to safety be designed, fabricated, erected, and tested to quality standards commensurate with the importance of the safety functions to be performed. For example, Criterion III, "Design Control," of Appendix B, "Quality Assurance Criteria for Nuclear Power Plants and Fuel Reprocessing Plants," to Part 50 of Title 10 of the U.S. Code of Federal Regulations (10 CFR Part 50) requires that utilities operating U.S. nuclear power plants establish control measures to verify the adequacy of design and that vendor requirements are included in the design basis. Criterion V, "Instructions, Procedures, and Drawings," of Appendix B requires the utilities to have procedures and acceptance criteria for the conduct of activities that involve the capability of safety-related equipment to perform its safety function. Criterion XI, "Test Control," of Appendix B requires that procedures for testing components contain provisions for ensuring that adequate test instrumentation is available and that test results be evaluated to assure test requirements are satisfied. Criterion XII, "Control of Measuring and Test Equipment," of Appendix B requires that utilities establish measures to ensure that measuring and test devices used in activities affecting quality are properly calibrated. Criterion XVI, "Corrective Action," of Appendix B requires utilities to establish measures to ensure that conditions adverse to quality, such as deficiencies and defective equipment, are promptly identified and corrected.

Despite the regulatory requirements for MOVs, operating experience and both regulatory and industry research programs have revealed problems with the performance of MOVs in U.S. nuclear power plants. There has been a long history of these problems and regulatory actions for dealing with them. For example, on December 6, 1972, the USNRC issued Bulletin 72-03, which addressed failures of several MOVs to operate. Operating experience and research also identified a wide variety of causes, among which were inadequate design and incorrect torque, torque bypass, and limit switch settings that had led, or could lead, to failures of MOVs to perform their intended functions.

Because of the continuing MOV problems, the USNRC issued Bulletin 85-03, "Motor-Operated Valve Common Mode Failures During Plant Transients Due to Improper Switch Settings," in 1985 requesting U.S. nuclear power plant utilities to develop programs to ensure that MOVs in high-pressure safety-related systems could perform their safety function. In 1988 and 1989, the USNRC staff sponsored tests of MOVs by the Idaho National Engineering Laboratory (INEL) which revealed that certain flexible-wedge gate valves manufactured in the U.S. required more thrust to operate than had been predicted by the valve vendor when sizing and setting the motor actuator. These test results raised concerns that the initial MOV design and qualification, and in-service stroke-time testing required by the American Society of Mechanical Engineers Boiler and Pressure Vessel Code (ASME Code), might be inadequate to ensure the capability of MOVs to perform their design-basis function. On the basis of the implementation of Bulletin 85-03, the staff-sponsored MOV test results, and operating events, the USNRC issued Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and
Surveillance," on June 28, 1989, which requested that U.S. nuclear power plant utilities verify the design-basis capability of safety-related MOVs by dynamic testing in about five years.

**Design and Qualification**

The most significant MOV problems in the U.S. result from the weakness in the initial design and qualification of MOVs before their installation in nuclear power plants. As described below, the weakness in MOV design and qualification contributed to instances where (1) thrust and torque requirements to operate valves were underestimated as a result of the underprediction of friction, or design-basis differential pressure; (2) motor actuator output was overestimated by failing to determine design-basis minimum voltage, ambient temperature effects on motor output, or load sensitive behavior; (3) structural capability of MOV components was insufficient; and (4) the potential for pressure locking and thermal binding of gate valves was inadequately considered.

(1) **Underestimation of Thrust and Torque Requirements**

**Assumption of Valve Friction for Gate Valves**

Gate valve vendors in the U.S. typically have determined the required size and setting of motor actuators for a particular valve based on the operating thrust requirement derived from the sum of (1) the design-basis differential pressure across the valve, multiplied by the area of the valve disk and a valve friction factor, (2) the design-basis system pressure multiplied by the area of the valve stem, and (3) the valve packing load. For the most part, the valve vendors selected a valve friction factor based on an assumption of sliding friction between the valve disk and seat. Tests of MOVs by the Electric Power Research Institute (EPRI), U.S. utilities, and INEL have revealed the typically-used valve friction factors to be inadequate for many gate valves. The resulting underestimation of the thrust required to open or close gate valves has led to some MOVs in safety-related systems in U.S. nuclear power plants being sized or set inadequately and, consequently, failing to operate under differential pressure and flow conditions. U.S. utilities have discovered during analyses or testing in response to GL 89-10 that predicted thrust requirements for numerous safety-related MOVs were inadequate and that modifications to the MOVs were appropriate.

**Assumption of Valve Friction and Flow Area for Globe Valves**

The typical equation used to predict thrust requirements to operate globe valves is similar to the equation for gate valves except that the orientation of the globe valve stem parallel to fluid flow is incorporated. Although apparently not as extensive a problem as with gate valves, testing of globe valves by EPRI and U.S. utilities has revealed thrust requirements greater than predicted by the vendors for some globe valves. EPRI has indicated that the cause of the higher thrust requirements may be related to the flow area that should be
assumed in the thrust equation. Recently, a U.S. globe valve vendor notified the USNRC that, based on testing by EPRI, the thrust predicted for operating its globe valves during design of their motor actuators might be insufficient.

Assumption of Stem Friction Coefficient for Gate and Globe Valves

For a gate or globe valve, the torque required from the motor actuator to open and close the valve is predicted from the thrust requirement multiplied by a stem factor (which is determined from the dimensions of the valve stem and its thread, and an assumed stem friction coefficient). The torque requirement is then used to size the motor actuator. Also, MOVs in U.S. nuclear power plants are not controlled directly by thrust measurement, but by setting torque switches or limit switches. Therefore, an improper assumption for the stem friction coefficient can cause an underestimation of the torque requirement and an inability of the motor actuator to open or close the valve. Stem friction coefficients assumed in the initial design of MOVs appear, for the most part, to have been adequate; however, some U.S. utilities have found limited instances of higher than assumed stem friction coefficients during MOV testing.

Determination of Torque Requirements to Operate Butterfly Valves

Motor actuators for butterfly valves rotate the valve disk to allow fluid flow. A weakness in the design and qualification of butterfly MOVs has led to the underestimation of the torque required to open or close some butterfly valves. From operating events, tests and analyses, U.S. utilities have found some motor actuators to be inadequately sized or set to open or close their butterfly valves.

Assumption of Design-Basis Differential Pressure

As indicated earlier, the differential pressure across the valve is a significant factor in determining the torque and thrust required to operate the valve. Some U.S. utilities have discovered underestimation of torque and thrust requirements as a result of improper assumptions for the differential pressure across the valve under design-basis conditions. The underestimation of torque and thrust requirements might have resulted in some MOVs being sized or set inadequately to perform their safety function.

Overestimation of Motor Actuator Output

Design-Basis Minimum Voltage

The torque delivered by a motor actuator depends on the motor torque-size, motor-actuator gear ratio and efficiency, voltage present at the motor, and other factors. Some U.S. utilities have found the voltage present at particular safety-related MOVs to be less than assumed in the design calculations of the torque and thrust that could be delivered by the MOVs under design-basis conditions. With insufficient voltage, an
MOV could be incapable of performing its safety function under design-basis conditions. Sufficient motor voltage is also important for ensuring the capability of motor-start contactors to start the MOV motor.

**Ambient Temperature Effects on Motor Torque Output**

The ambient temperature surrounding the MOV must be considered in the design and qualification of the MOV. In 1993, an actuator manufacturer notified the USNRC that, under high ambient temperature conditions, the torque delivered by ac motors used to operate its actuators could be less than assumed during design of the MOV. With reduced torque output as a result of high ambient temperature conditions, a motor actuator might be incapable of opening or closing its valve to perform a safety function.

**Load Sensitive Behavior**

In the past few years, research and valve testing have revealed that the thrust delivered by the motor actuator at a specific torque output can be lower when the MOV is operated under differential pressure and flow conditions than when operated without fluid pressure or flow. This load-sensitive behavior (sometimes referred to as rate-of-loading) of motor actuators might be caused by an increase in the stem friction coefficient under loaded conditions. Although not completely understood, load-sensitive behavior of motor actuators has been observed in many MOV tests. The reduced thrust output resulting from load-sensitive behavior might cause a motor actuator to deliver insufficient thrust to close its valve when the torque switch stops the motor, or if the torque-capability of the motor actuator is reached.

(3) **Structural Capability of MOV Components**

The weakness in the design and qualification of MOVs has included inadequate evaluation of the capability of individual components to withstand the thrust and torque exerted upon them. For example, U.S. utilities have discovered cracks in valve yokes that might have prevented motor actuators from operating their valves. In one instance, a manual valve had been converted to a motor-operated valve without appropriate consideration of the additional stress on the valve yoke.

(4) **Potential Pressure Locking and Thermal Binding of Gate Valves**

U.S. utilities have not always adequately addressed the increased thrust requirements for gate valves that might result from pressure locking or thermal binding of gate valves. Pressure locking can occur in a gate valve with two half-disks when the pressure inside the valve bonnet is greater than the pressure upstream and downstream of the valve. Under this condition, thrust is required to overcome differential pressure across both half-disks instead of the typical design assumption of one disk. Thermal binding between the valve body and disk may occur when differences in material properties cause mechanical interference.
following temperature changes. Motor actuators might not have sufficient capability to overcome the increased thrust requirements resulting from pressure locking or thermal binding and can lead to the inability of the associated safety train or system to perform its safety function. Some MOVs have failed to operate because of pressure locking or thermal binding in U.S. nuclear power plants.

Maintenance and Training

Operating experience has revealed weaknesses in procedures for MOV maintenance and training of MOV personnel. These weaknesses have resulted in various problems with the performance of MOVs at U.S. nuclear power plants. Particular significant examples include: (1) stem and stem nut failure of MOVs in both trains of the low-pressure coolant-injection system in a U.S. nuclear power plant could have caused this system to be unable to perform its safety function; (2) loose and cracked motor pinion keys in MOVs at several U.S. nuclear power plants because of inadequate staking or stress that exceeded the material strength prevented the motor from driving the actuator; (3) incorrectly set limit switches causing MOVs not to operate properly; (4) disengaged valve disks from butterfly valve actuators prevented proper valve operation; and (5) improper replacement of manual declutch levers resulting in failure of MOVs to operate.

Root Cause and Trending of MOV Problems

Operating experience has revealed weaknesses in the evaluation of the cause of MOV problems, and the trending of MOV problems at U.S. nuclear power plants. In a few cases, U.S. nuclear plants have remained shutdown for extended periods while the root cause of numerous MOV problems was determined. The extent of MOV problems might have been identified earlier at those nuclear plants with an adequate program to trend MOV problems.

Many MOV problems at U.S. nuclear power plants have been revealed as a result of the comprehensive programs of MOV testing and analyses developed by U.S. nuclear utilities in response to GL 89-10. These MOV problems were corrected by the utilities when identified. The USNRC staff believes that the number of MOV problems, and operating events caused by those problems, are being reduced by the response to GL 89-10, and will significantly decrease as U.S. nuclear utilities complete their GL 89-10 programs.
ACTIVITIES OF THE U.S. NUCLEAR REGULATORY COMMISSION
TO IMPROVE THE PERFORMANCE OF MOTOR-OPERATED VALVES

The USNRC regulations require that components that are important to the safe operation of a nuclear power plant be treated in a manner that provides assurance of their performance. Appendices A, "General Design Criteria for Nuclear Power Plants," and B, "Quality Assurance Criteria for Nuclear Power Plants and Fuel Reprocessing Plants," to 10 CFR Part 50 provide a broad-based framework of requirements for the design, testing, operation and maintenance of components, including motor-operated valves (MOVs), that are important to the safe operation of the plant. With respect to inservice testing of MOVs, 10 CFR 50.55a(g) requires compliance with Section XI, "Rules for Inservice Inspection of Nuclear Power Plant Components," of the ASME Code for MOVs within the scope of that code. The recent revision of the USNRC regulations on maintenance activities in 10 CFR 50.65 also provides requirements that address the performance of certain MOVs in nuclear power plants.

In response to continuing MOV problems, the USNRC staff prepared NUREG-1352 (June 1990), "Action Plans for Motor-Operated Valves and Check Valves," describing actions to organize the activities aimed at resolving the concerns about MOV (and check valve) performance. Among those actions are evaluation of the current regulatory requirements and guidance applicable to MOVs, development of guidance for and coordination of USNRC inspections, completion of USNRC MOV research programs, implementation of the research results, and providing MOV information to the nuclear industry.

A significant task of the MOV action plan is the USNRC staff's review of the implementation of Generic Letter (GL) 89-10 (June 28, 1989), "Safety-Related Motor-Operated Valve Testing and Surveillance," and its supplements at U.S. nuclear power plants. In GL 89-10, the USNRC staff asked U.S. utilities to help ensure the capability of MOVs in safety-related systems by reviewing MOV design bases, verifying MOV switch settings initially and periodically, testing MOVs under design-basis conditions where practicable, improving evaluations of MOV failures and necessary corrective action, and looking for trends in MOV problems. The USNRC staff requested that utilities complete the GL 89-10 program within three refueling outages or five years from the issuance of the generic letter, whichever is later.

The USNRC staff issued Supplement 1 to GL 89-10 on June 13, 1990, to give utilities detailed information on the results of public workshops held in 1989 to discuss the generic letter.

On August 3, 1990, the USNRC staff issued Supplement 2 to GL 89-10 to allow utilities additional time to review and to incorporate the information provided in Supplement 1 into their programs in response to the generic letter.

Tests performed by the Idaho National Engineering Laboratory (INEL) as part of a program by the USNRC Office of Nuclear Regulatory Research reinforced concerns regarding the capability of MOVs to perform their design-basis functions. At a public meeting on April 18, 1990, the INEL researchers
discussed the results of the USNRC-sponsored MOV tests that revealed that more thrust was required to operate the tested valves under high differential-pressure and flow conditions than had been predicted using standard U.S. industry calculations. These test results were directly applicable to the safety function of MOVs used for containment isolation in the high-pressure coolant-injection (HPCI), reactor core isolation cooling (RCIC), and reactor water cleanup (RWCU) systems of boiling-water reactor (BWR) plants. Following a summary review of the capability of MOVs in those systems and discussions with the BWR Owners Group, the USNRC staff issued Supplement 3 to GL 89-10 on October 25, 1990, which asked utilities of BWR plants to perform a plantspecific safety analysis and to evaluate the capability of MOVs used for containment isolation in the HPCI, RCIC, and RWCU systems (and in isolation condenser lines), as applicable. Also, the USNRC staff asked all utilities to consider the results of the MOV tests in their GL 89-10 programs. BWR utilities have completed their evaluations of the MOVs within the scope of Supplement 3 to GL 89-10, and have modified or adjusted many MOVs to provide assurance of their capability to perform their design-basis function.

On February 12, 1992, the USNRC staff issued Supplement 4 to GL 89-10 and stated that BWR utilities need not address inadvertent MOV operation as part of their GL 89-10 programs based on an USNRC-sponsored study of core-melt probability by the Brookhaven National Laboratory (BNL). Nevertheless, the USNRC staff stated its belief that consideration of inadvertent MOV operation benefited safety.

As an integral part of their GL 89-10 programs, most U.S. utilities are relying on MOV diagnostic equipment to obtain information on the thrust required to open or close the valve as well as the thrust delivered by the motor actuator. The various types of MOV diagnostic equipment estimate stem thrust using different parameters, such as spring pack displacement or strain in the stem, mounting bolts, or yoke. Because some utilities make decisions regarding the operability of safety-related MOVs on the basis of thrust readings of diagnostic equipment, the use of MOV diagnostic equipment can have a significant effect on the safe operation of a nuclear power plant.

During the implementation of GL 89-10, the USNRC staff became aware of information that raised a generic concern regarding the reliability of the data provided by MOV diagnostic equipment. For example, the MOV Users Group (MUG) of nuclear utilities on February 3, 1992, released "Final Report – MUG Validation Testing as Performed at Idaho National Engineering Laboratories" (Volume 1), which indicated that MOV diagnostic equipment relying on spring pack displacement to estimate stem thrust was not as accurate as its vendors believed. In addition, on October 2, 1992, a manufacturer of MOV diagnostic equipment that derives thrust from yoke strain calibrated to stem thrust using measured diametral strain of the valve stem and nominal engineering material properties notified the USNRC that two new factors that could affect the thrust values obtained with its equipment involved (1) the possible use of improper stem material constants and (2) the failure to account for a torque effect when the equipment is calibrated to strain in the threaded portion of a valve stem.

The manufacturers of the MOV diagnostic equipment have evaluated the
information revealing the increased inaccuracy of their equipment and have
provided guidance to the utilities for their use in correcting the data
obtained from their equipment. Further, the manufacturers are developing
improved equipment and software to provide more accurate thrust and torque
measurements.

On June 28, 1993, the USNRC issued Supplement 5 to GL 89-10 which asked U.S.
nuclear utilities to reexamine their MOV programs and to identify measures
taken or planned to account for uncertainties in MOV diagnostic equipment.
U.S. nuclear utilities were required to notify the USNRC staff of their
diagnostic equipment and to report their plans to address the information on
the accuracy of MOV diagnostic equipment. The USNRC staff has reviewed the
utility responses to Supplement 5 to GL 89-10 and sent replies to the
individual utilities. USNRC inspections will address specific aspects of
utilities' actions to address MOV diagnostic equipment inaccuracy.

On March 8, 1994, the USNRC issued Supplement 6 to GL 89-10 to transmit
information to U.S. nuclear utilities on the schedule of GL 89-10 programs and
grouping of MOVs to share test data, and to respond to questions raised at the
public workshop held in February 1993 to discuss the generic letter. In
Supplement 6 to GL 89-10, the USNRC staff requests that, if a nuclear utility
intends to extend its schedule for completing the GL 89-10 program, the
utility submit specific information on the capability of those MOVs whose test
schedule will be extended. In Supplement 6, the USNRC staff states that U.S.
utilities are expected to have their safety-related MOVs set up using the
best-available MOV test data by the original completion date accepted by the
USNRC, even if their GL 89-10 test schedule is extended.

The USNRC staff contracted with BNL to perform a core-melt probability study
to address valve mispositioning in pressurized-water reactor (PWR) nuclear
power plants. This study was similar to the BNL study of valve mispositioning
in BWR plants. The USNRC staff is preparing proposed Supplement 7 to GL 89-10
to discuss the USNRC staff's position on the need for considering inadvertent
MOV operation in PWR nuclear power plants as part of the GL 89-10 program.

In March 1993, the USNRC issued NUREG-1275, Volume 9, "Pressure Locking and
Thermal Binding of Gate Valves," which lists the history of pressure-locking
and thermal-binding events, describes the phenomena, discusses the
consequences of locking or binding on valve functionality, summarizes
preventive measures, and assesses the safety significance of the phenomena.
Despite several generic industry communications providing guidance for
identifying susceptible valves and performing appropriate preventive and
corrective measures, pressure-locking and thermal-binding events continue to
occur and, also, might occur when an MOV is needed to perform its safety
function at a U.S. nuclear power plant. Therefore, the USNRC staff is
preparing a proposed generic letter to request that U.S. nuclear utilities
identify power-operated gate valves susceptible to pressure locking and
thermal binding and implement corrective action for those valves within a
specific time schedule.

The USNRC staff issued Temporary Instruction (TI) 2515/109 (January 14, 1991),
"Inspection Requirements for Generic Letter 89-10, Safety-Related Motor-
Operated Valve Testing and Surveillance," to provide guidance for its inspectors to evaluate programs developed at U.S. nuclear power plants in response to GL 89-10. Part 1 of the TI contains guidance for performing inspections to review programs developed in response to GL 89-10. Part 2 of the TI contains guidance for performing inspections to determine the adequacy of the implementation of GL 89-10 programs. The inspections under Part 2 of the TI focus on a sample of specific MOVs to determine the adequate implementation of the overall GL 89-10 program at a nuclear power plant. On June 14, 1993, the USNRC staff issued Revision 1 to the TI to update its guidance based on inspections of GL 89-10 programs conducted to date.

In January 1991, the USNRC staff began inspecting the programs developed by U.S. nuclear utilities in response to GL 89-10. The USNRC staff has performed inspections to review the development of MOV programs in response to GL 89-10 at each U.S. nuclear power plant. In Information Notice 92-17 (February 26, 1992), the USNRC staff summarized the findings of the GL 89-10 inspections conducted up to that time. In 1993, the USNRC staff initiated inspections of the implementation of GL 89-10 programs and has conducted more than thirty inspections to date.

The USNRC inspections of GL 89-10 programs show, for the most part, that U.S. utilities are establishing the scope of their GL 89-10 programs consistent with the recommendations of the generic letter.

With respect to the recommendations of GL 89-10 regarding design-basis reviews of MOVs, U.S. utilities have been reviewing plant documentation (such as the final safety analysis report and technical specifications) to determine the design-basis conditions for safety-related MOVs. Some utilities had focused on differential pressure and had not adequately addressed other design-basis parameters (such as flow, fluid temperature, ambient temperature, and seismic/dynamic effects). Although differential pressure is the primary design-basis parameter used to predict thrust requirements in the present industry equations, other design-basis parameters also need to be considered to ensure that the test results demonstrate that the MOV will operate under design-basis conditions. Many utilities found the need to update their degraded voltage studies to ensure that the design-basis voltage is determined at each MOV. A significant concern from the inspections has been the weaknesses in the evaluation of the potential for pressure locking and thermal binding of gate valves.

With respect to the recommendations of GL 89-10 regarding MOV sizing and switch settings, U.S. utilities use various methods to determine the proper size of MOVs and their appropriate switch settings. Some utilities have increased the valve factors assumed in the industry equation (used to predict the thrust required to operate the valves) to reflect industry and plant-specific experience. However, a few utilities continued to use previous guidance provided by valve vendors in estimating thrust requirements which have been or may be determined to be inadequate during design-basis MOV tests. The USNRC inspectors found that the validation of assumptions for the following parameters in the MOV calculations for sizing and switch settings need improvement: valve friction coefficient (or valve factor), stem friction coefficient, and load-sensitive behavior where the output of the actuator may
be less under dynamic conditions than under zero differential pressure and flow (static) conditions.

With respect to the recommendations of GL 89-10 regarding MOV testing, U.S. utilities have found during differential-pressure and flow testing of MOVs that many gate valves (and some globe and butterfly valves) require more torque or thrust to operate than was predicted by the valve vendors.

Among the most significant inspection concerns regarding MOV testing have been (1) the lack of progress in completing dynamic testing, (2) weaknesses in procedures and acceptance criteria for the tests to evaluate the capability of the MOV to perform its safety function under design-basis conditions, and (3) feedback of the test results into the methodology used by the utility in predicting the thrust and torque requirements for other MOVs. Among other utility activities found to need improvement with respect to testing are (1) justification for grouping of valves to share test information in order to minimize the number of MOVs tested, (2) verification of methods to extrapolate data from test conditions to design-basis conditions, (3) evaluation of anomalies in MOV diagnostic equipment signature traces, and (4) involvement of quality assurance personnel in verifying the accuracy of test data and analyses.

The USNRC regulations and plant-specific technical specifications establish requirements for actions and reporting by U.S. nuclear utilities when safety-related equipment is determined to be, or has been, unable to perform its safety functions. When a problem is found with one of its safety-related MOVs, the U.S. utility must evaluate the impact of that problem on the capability of the MOV to perform its safety function. GL 91-18, "Information to Licensees Regarding Two NRC Inspection Manual Sections on Resolution of Degraded and Nonconforming Conditions and Operability," contains information on guidance provided to USNRC inspectors in the area of operability of safety-related components. This information is also useful to U.S. utilities in evaluating the operability of an MOV found to have a performance problem.

With respect to the recommendations of GL 89-10 regarding periodic verification of MOV capability, many U.S. utilities have stated that they will attempt to use tests of MOVs with diagnostic equipment under static conditions to demonstrate the adequacy of torque switch settings and the continued capability of MOVs to perform their safety functions under design-basis conditions. No utility, as yet, has provided justification for applying the results of tests conducted under static conditions to demonstrate design-basis capability. With respect to postmaintenance testing, many utilities are improving their methods to demonstrate continued capability of MOVs to perform their safety functions under design-basis conditions following maintenance.

With respect to the recommendations of GL 89-10 regarding MOV failures, corrective action, and trending, the USNRC inspectors found weaknesses in some U.S. utilities’ response to MOV failures and deficiencies. Some utilities had not analyzed the root cause of MOV problems thoroughly. Most utilities are attempting to improve the trending of MOV problems, but little progress has been made in implementing those trending programs.
The USNRC staff found that the U.S. utilities have significantly improved their training programs for MOV maintenance and diagnostic testing.

During the GL 89-10 inspections, the USNRC staff found that some U.S. utilities have not made adequate progress toward resolving the MOV issue for their facilities within the recommended schedule of GL 89-10. The USNRC staff has accepted limited extensions of the GL 89-10 schedule for particular utilities where justification has been provided.

The USNRC staff has identified areas requiring further research and analysis to assist the staff in evaluating GL 89-10 programs at U.S. nuclear power plants. For example, NUREG/CR-5720, "Motor-Operated Valve Research Update," provides important information on several areas of MOV behavior under high-load conditions. Additional ongoing research is improving the understanding of MOV performance in support of regulatory activities.

The USNRC staff continues to provide information to the nuclear industry through meetings to assist utilities in resolving MOV issues at their particular facilities. For example, the USNRC staff discusses MOV issues at regulatory information conferences, and presents the status of USNRC activities and current concerns at meetings of the MOV Users Group (MUG) of nuclear utilities. In addition, the USNRC staff and the ASME are sponsoring a symposium to address pumps and valves (including MOVs) to be held in Washington, D.C., in July, 1994.

The USNRC staff issues information notices to alert U.S. nuclear utilities to important aspects of MOV performance. For example, the USNRC issued Information Notice (IN) 92-23, "Results of Validation Testing of Motor-Operated Valve Diagnostic Equipment"; IN 93-74, "High Temperatures Reduce Limitorque AC Motor Operator Torque"; IN 93-88, "Status of Motor-Operated Valve Performance Prediction Program by the Electric Power Research Institute"; IN 93-97, "Failures of Yokes Installed on Walworth Gate and Globe Valves"; IN 93-98, "Motor Brakes on Valve Actuator Motors"; and IN 94-10, "Failure of Motor-Operated Valve Electric Power Train Due to Sheared or Dislodged Motor Pinion Gear Key."

The USNRC staff meets regularly with representatives of the Nuclear Management and Resources Council (NUMARC) and the Electric Power Research Institute (EPRI) to discuss the EPRI MOV Performance Prediction Program. EPRI tested gate, globe, and butterfly valves, and analyzed the results of additional valve tests, as part of its development of a methodology to predict the performance of MOVs. NUMARC has submitted the EPRI MOV Performance Prediction Program as a topical report for USNRC staff review.

The USNRC staff participates on the committees responsible for improving U.S. codes and standards for MOV performance. For example, the USNRC staff participated in the preparation of ASME OM-8, "Startup and Periodic Performance Testing of Electric Motor Operators on Valve Assemblies in Nuclear Power Plants." The USNRC staff is also participating in the revision of ASME Standard QME, "Qualification of Mechanical Equipment Used in Nuclear Power Plants," to improve the functional qualification requirements for valve assemblies.
The USNRC staff recognizes the significant amount of utility resources that has been required to implement MOV programs in response to GL 89-10. However, as discussed earlier in this presentation, the MOV programs established in response to GL 89-10 have led to the identification and resolution of numerous weaknesses in the design, qualification, and maintenance of MOVs, and the corrective action and trending of MOV problems. Through its inspection program, the USNRC staff has found that significant progress has been made by the U.S. nuclear utilities in the design, qualification, and maintenance of MOVs. Therefore, the USNRC staff believes that MOV problems will decrease and that MOV performance will continue to improve in the future.

As the nuclear industry and its regulators work toward resolution of the MOV issue, the USNRC staff plans (1) to continue to inspect MOV programs at U.S. nuclear power plants, (2) to complete the preparation of proposed Supplement 7 to GL 89-10 on the need to consider inadvertent operation of MOVs in PWR nuclear power plants under the GL 89-10 program, (3) to prepare a proposed generic letter on pressure locking and thermal binding of gate valves, (4) to review the EPRI MOV Performance Prediction Program Topical Report, (5) to continue to meet with, and provide information to, U.S. nuclear utilities regarding MOV performance, and (6) to continue international cooperative efforts to improve MOV performance.
REEVALUATION OF MOTOR OPERATED VALVES IN SPAIN.

SUMMARY OF ACTIVITIES

Alfonso J. Pérez
C.S.N. (Spain)

Madrid, April 1994
1. **BACKGROUND**

There are nine operating plants in Spain: six of them (José Cabrera, Almaraz I & II, Ascó I & II, Vandellós II) are Westinghouse designs, two (Garoña, Cofrentes) are G. E. designs, and one (Trillo), of Siemens design. Plants are owned mainly by private companies.

The regulatory body, C.S.N. ("Consejo de Seguridad Nuclear", that in English translates as "Nuclear Safety Council") has his headquarters in Madrid, the nation's capital, with no regional offices. Technical staff is about 195 people.

2. **M.O.V. ISSUE SCOPE**

Plant owners dedicated some effort to follow IE Bulletin 85-03 (mainly in relation with action a. of the Bulletin). Documentation sent to C.S.N. was evaluated, but no generic or specific inspection effort was undertaken.

As a general rule, the regulation of the country of the main supplier is considered applicable in Spain if no other one (national, or international) specifically applies. Utilities must submit, biannually, a document analyzing how recent U.S. regulations have been considered.

In particular, then, Generic Letter 89-10 is applicable in Spain: some indefinites exist (for instance, in schedule aspects) about enforcement: staff engineering judgement, in such cases, is a factor.

3. **INSPECTION ACTIVITIES: SCOPE & SUMMARY**

Inspection activities begun in 1991. Usually, inspections of one or two days have been made (several to each plant).
Up to June 1992, objectives were mainly aimed to:

- enforce the preparation of adequate evaluation program descriptions,

- witness static diagnosis tests.

- transmit the necessity of ΔP tests.

- review MOV CWDs.

From July 1992 up to now, subject items evolved, and inspections are aimed to:

- review the input data for MOV actuator calculations, and examples of such calculations.

- review the calculation process conclusions, corrective actions and implementation schedules.

- witness static diagnosis test,

- gather information about trending on similarity analysis.

4. INSPECTIONS & EVALUATION STATUS PER SPECIFIC PLANT

a) José Cabrera

PWR, Westinghouse design, single loop, 160 Mwe. operating license in 1968.

A total of five inspections were made: their respective covered items were as follows:

- 1st inspection. MOV program scope & static diagnosis results review.
- 2nd. static diagnosis inspection on HCV-908 valve & static diagnosis results review.

- 3rd. operability degree analysis for six valves, on account of actuator calculation & diagnosis results.

- 4th. multipurpose audit to engineering offices (translation of the inspection Agenda is included in the Annex, as an example).

- 5th. activities during refueling outage; dynamic diagnosis on a butterfly valve.

Actuators were calculated by Westinghouse, using Limitorque equations; deficiencies were found in 8 valves, and the most repeated problem in the unconsistency between the supposedly installed capacity and the really measured capability. Corrective actions, of varied range (actuator replacement, spring pack change, motor uprating, obturator design change, ...) have been implemented.

b) S. M. Garoña

BWR, G.E. design, BWR-3, Mark I, 460 Mwe. operating license in 1970.

Five inspections were made, to cover mainly the following:

- 1st inspection. MOV program scope review.

- 2nd. ΔP design basis & static diagnosis results review.

- 3rd. preliminary actuator calculation results review.

- 4th. multipurpose audit to the engineering offices, considering aspects as corrective actions adequacy, high inertia evaluation, actuator calculation
methodology, ΔV, setting changes, ΔP testing & similarity analysis, specific cases of ΔP design bases justification.

- 5th activities during refueling outage: corrective actions implementation degree.

Actuators were calculated by Siemens, that used his usual methodology. Sizing inadequacies were observed in four MOVs, G.L. 89-10, Suppl. 3 related: for those MOVs, one actuator has been recently changed, two are still pending of decision, and the fourth will likely be saved. Additionally, about 8 valves with old SMA actuators, have high inertia problems: new diagnosis will permit a more founded assessment on the acceptability of the actuators.

c) Almaraz I & II

PWR, Westinghouse design, three loops, 930 Mwe, operating license in 1980 and 1983: plants similar to North Anna (U.S.A.).

A total of four inspections were made: covered topics were mainly the following:

- 1st inspection. MOV program scope & static diagnosis results review.

- 2nd diagnosis techniques & results review.

- 3rd actuator calculations status. Calculation completion was delayed to July 1, 1994. The plant accepted the C.S.N. recommendation of giving priority to MOVs with static diagnosis results that do not match enough with the available actuator data.

- 4th available actuator calculation results, diagnosis results review.
Staggering of engineering staff, for the Westinghouse designed plants, seems to have been influential in the unfinished work. Corrective actions on actuators and Δp testing, will be performed mainly during 1995.

d) Ascó I & II

PWR, Westinghouse design, three loops, 930 Mwe, operating license in 1982 and 1985; plants similar to Almaraz I & II.

Five inspections were made to cover mainly the following:

- 1st inspection. MOV program scope, static diagnosis results. CWDs and IE Bulletin 85-03 calculations review.

- 2nd, diagnosis techniques & results, static diagnosis inspection on MOV 3639.

- 3rd, ΔP design bases review, actuator calculation results for Westinghouse supplied MOVs.

- 4th, ΔP testing.

- 5th, actuator calculations for other MOV suppliers. corrective action status. previsions on outage activities.

Some valve suppliers have not finished their calculations on MOV actuators. Several important MOVs (such those located in the HPI lines) require spring pack replacement; some others, actuator change. MOVs with actuators that do not require modifications are being Δp tested first.
e) Cofrentes

BWR. G.E. design, BWR-6, Mark III, 994 Mwe. operating license in 1984; plant similar to Grand Gulf (U.S.A.)

This plant had an early and aggressive approach to the G.L. 89-10 issue: it is the first, in Spain, that has implemented the corrective actions that resulted from the actuator calculations.

A total of five inspections were made: topics covered were mainly the following:

- 1st inspection. ΔP design bases & CWDs review.
- 2nd, to watch static diagnosis tests, and one dinamic test.
- 3rd, preliminary actuator calculation results review, diagnosis results review.
- 4th, status of corrective actions implementation, inspection of one dinamic test.
- 5th, step-by-step review of the actuator calculation (Siemens methodology) in two selected cases: MOV F001 in RWCU suction line, MOV F004 in HPCS injection line.

Thirteen MOV actuators required corrective action: six actuator replacements, four partial upgradings, three cases of increase in cable section. As previously mentioned, these actions were already implemented.

f) Vandellós II

PWR, Westinghouse design, three loops, 982 Mwe. operating license in 1987; plant similar to Asco I & II.
Four inspections were made, to cover mainly the following:

- 1\textsuperscript{st} inspection. MOV program scope. Initially, this plant did not use diagnosis techniques; some time later, the plant accepted a C.S.N. directive on their incorporation.

- 2\textsuperscript{nd} MOV program status (early stage).

- 3\textsuperscript{rd} anticipation of some calculation results: $\Delta P$ test results for one valve.

- 4\textsuperscript{th} results of the incomplete calculation process. Previsions of refueling outage activities. During the very next refueling outage, the plant will perform $\Delta p$ test on valves with calculations already finished.

g) Trillo

PWR, Siemens-KWU design, 1040 Mwe. operating license in 1987.

One inspection was made, to deal mainly with the status of the issue and with the utilization of diagnosis results.

Process is not so direct as with the rest of the plants, because Generic Letter 89-10 is not applicable in this case. A mandatory letter was sent to the plant, requiring the development of a MOV program, on the basis of safety considerations and trends in Germany. Some time later, in March 1993, Siemens made a presentation of the program prepared, that implies to finish the calculations in the middle of 1995, and to complete the program in the last quarter of 1997.
5. GENERAL OVERVIEW. TRENDS

The following remarks intend to complete the picture of the MOV issue in Spain, from the C.S.N. point of view:

a) NPPs in Spain (Trillo is excepted) will have completed the actuators calculation during 1994. At all odds, C.S.N. will proceed with the initiated review, covering also corrective actions and switch resettings, in cases of MOVs selected as representatives.

b) Degraded voltage of 80% nominal has been generally assumed: in very specific cases, it can be unconservative. For selected cases of actuators with slightly insufficient sizing, realistic ΔV calculations have been performed.

c) C.S.N. has not basically received information on the results of the EPRI tests: such information do arrives to the plant owners, through their concerted channels.

d) Diagnosis techniques that measure spring pack displacement as estimation of stem thrust have been progressively abandoned: they are still considered an useful reference, in static diagnosis. Currently, techniques relying on strain measurement in the stem or the yoke are used.

e) Not many ΔP tests have been made, up to now (the number is, obviously, plant dependent). Plants will try to demonstrate the unnecessity of such tests, on the basis of similarity studies, and restrict them to representative MOVs. One plant (S.M. Garoña) has completed the study on his Dikkers valves, and several plants have undertaken a joint study on the Walthon Weir Pacific MOVs (WWP is the main valve supplier in Spain). Some plants prefer to use the two-stage approach performing dynamic tests at reasonable Δp valves, when possible.
ANNEX
INSPECTION AGENDA AT JOSE CABRERA OFFICES

(October 7, 1993)

Related to the MOV reevaluation program, it is proposed to make an audit visit with reference to the following items:

a) Current status of the program, taking into account the Generic Letter 89-10 recommended actions.

b) Input data for MOV actuator calculations. Design bases differential pressures. Review of three or four selected cases.

c) Actuator calculations. Methodology used. Comparison between required and installed thrust & torque. Review of three or four selected cases.

d) Degraded voltage considerations. Calculations performed, if any.

e) Conclusions of the actuator calculation process. Considerations on the adequacy of the proposed corrective actions.

f) Changes on MOV settings. Structural considerations.

g) Previsions on ΔP testing; possibility of performing some of such tests, during the next refueling outage. Utilization of EPRI program data.

h) General assessment, conclusions.
AN UPDATE OF MOTORISED VALVE TESTING AT SIZEWELL B
-REGULATORY VIEWS AND ISSUES

PRESENTATION TO OECD/IAEA SPECIALIST MEETING ON MOTOR OPERATED
VALVE ISSUES

APRIL 25-28 1994

PARIS

DC ANDERSON

HMNII

MERSEYSIDE L20 3LZ

UK

ABSTRACT

This presentation describes the regulatory approach taken, with respect to the licensee's
operability testing programme for motorised valves, at Sizewell B, the UK's first PWR, from
pre-licensing to the present day when the plant is undergoing commissioning.

In particular, it was a requirement at the time of licensing, that the design should reflect
adequate in-depth defence against any safety concerns revealed from international PWR
experience: the operability of MOVs and the associated NRC Generic Letter 89-10 being
such an example.

Prior to fuel-load, assessment by the regulatory authority (NII) of the mechanical plant safety
case has included the particular aspects of valve selection and qualification- in particular flow
interruption testing to ANSI B 16.41-and the qualification of actuators. This presentation
links these aspects to the MOV testing for operability concerns. During this
pre-commissioning stage, some limited instrumented flow interruption testing was conducted
with early diagnostic equipment; this paper briefly describes this work.
Before the commencement of commissioning, a programme of operability testing of safety MOVs was agreed between the licensee and NII. The presentation describes the tasks that are expected of the licensee, Nuclear Electric, in order to demonstrate that an MOV operability programme is in place. In addition, adequate arrangements must be in place to demonstrate the means of retrieving and trending data to establish ageing effects over the life of the plant.

At present, a limited number of static and dynamic MOV tests have been completed. A discussion of the testing and results found during the commissioning stage will form the second part of the presentation.

1 LEGISLATION GOVERNING THE SAFETY OF NUCLEAR INSTALLATIONS

In the United Kingdom, the main legislation governing the safety of nuclear installations is the Health and Safety at Work etc. Act 1974 and the associated relevant statutory provisions of the Nuclear Installations Act 1965. Under the Nuclear Installations Act no site may be used for the purpose of installing or operating any commercial nuclear installation unless a nuclear site license has been granted to a body corporate by the Health and Safety Executive (HSE) and is for the time being in force. HM Nuclear Installations Inspectorate (NII) is that part of HSE responsible for administering this license function.

2 UK REACTOR PROGRAMME AND SIZEWELL B

Historically the UK civil nuclear programme has been based on the Gas Cooled Reactor technology of the Magnox and AGR designs.

The Sizewell B Nuclear Power Station is the UK's first PWR. It comprises the standard Westinghouse four loop PWR, with station layout based on the SNUPPS plants as built at Calloway and Wolf Creek in USA.

The station reference design has been adapted in order to address regulatory requirements in UK and national grid aspects. A Public Inquiry on the Licensing of the station took place from 1983 to 1985.

Subsequent to the Inquiry, the detailed design of the plant has progressed, and discussions between the licensee, Nuclear Electric, and NII on various engineering issues took place; including aspects of valve reliability, operability and testing.

This presentation describes the salient regulatory issues associated with the testing of MOVs for Sizewell B, from the time of the Inquiry to the present, when the mechanical plant is undergoing commissioning. Some of the more important test results are included to illustrate the description.
3 REGULATORY CONCERNS ON THE NEED FOR TESTING OF MOTORISED VALVES FOR SIZEWELL B

It was a requirement at the time of licensing that the design should reflect adequate in-depth defence against any safety concerns revealed from international PWR experience: the operability of MOVs and the associated NRC Generic Letter 89-10 is such an example.

In particular, HMNI was concerned that failure rates to be used in PRA (probabilistic risk analysis) studies might not accurately reflect actual field behaviour.

Furthermore, the advent of PWR technology introduced new environments and demands on valves of which there had been little experience in the UK. As an example the use of wedge gate valves is not prevalent in the UK Civil Reactor Programme and this proved significant in relation to the Motorised Valve diagnostic programme described later. Finally, full flow testing was considered to be necessary for key safety related valves.

PART 1- PRINCIPLES ASSOCIATED WITH THE UK MOV PROGRAMME FOR SIZEWELL B

4 THE USE OF DIAGNOSTIC TECHNIQUES TO DETERMINE OPERABILITY OF SAFETY EQUIPMENT

As the design of Sizewell B was completed, it was apparent that the need for more detailed assessment of in-service testing requirements was needed than had been envisaged at the time of licensing. Discussions with other regulators, notably NRC, and utilities has helped us to formulate our ideas in terms of what is needed for modern plant.

In addition, the quantity of safety equipment on modern plant is now so large as to be impossible to monitor the testing and maintenance of individual components without I.T, as an example there are in excess of 20,000 qualified components on Sizewell B. The use of diagnostic techniques can therefore provide a route for data acquisition, which is often more accurate than the previous means of ensuring operability, with the important additions that data storage and trend analysis can be performed.

This would suggest that the licensee should be able to demonstrate by trending analysis, for a selected component, that the availability is as assumed in the safety case, and that any known ageing effects are being monitored. This suggests that the inspection by the regulator of EQ, IST etc. could be more flexibly applied, rather than the 'tick sheet' methods used in earlier years where less safety equipment needed to be inspected.

The use of the diagnostics used so far on Sizewell B will be discussed later in the presentation. However, it is important to note that NII will require an acceptable MOV programme for operability testing in place prior to fuel load. This means that, where possible, full-flow-design-basis testing should be performed during commissioning to minimise any extrapolation. This is because such flow rates are unlikely to be available once the reactor is operational.

However, it is important to note that NII will require an acceptable MOV programme for operability testing in place prior to fuel load. This means that, where possible,
full-flow-design-basis testing should be performed during commissioning to minimise any extrapolation. This is because such flow rates are unlikely to be achievable once the reactor is operational.

5 POINTS OF DETAIL ON MOTORISED VALVE PERFORMANCE-ASSESSMENT OF COMPONENT PARTS AND IMPLICATIONS

The chart below identifies the main component parts which make up an MOV and which influence the description of operability. In the second part of the presentation, the particular details and some of testing aspects of these are described.

<table>
<thead>
<tr>
<th>Component</th>
<th>parts description</th>
<th>affect on determination of stem thrust</th>
<th>typical method of measurement</th>
<th>method of determination for Sizewell B at this stage in programme</th>
</tr>
</thead>
<tbody>
<tr>
<td>actuator</td>
<td>worm gear assemblies, stem nut/stem</td>
<td>conversion of torque to thrust-stem factor</td>
<td>via stem friction and stem factor</td>
<td>back calculation of stem factor from measured thrust and torque-where pacticable</td>
</tr>
<tr>
<td>actuator control</td>
<td>various limit switches for motor control, valve position, torque control, indication</td>
<td>torque switch and torque control, parallel slide valve closing on limit</td>
<td>indication of limit switch position to be held on trace for use in setting up of valve. Manufacturer's dynamometer tests have provided limiting torque data for each switch setting on every valve actuator.</td>
<td></td>
</tr>
<tr>
<td>valve type</td>
<td>parallel/wedge gate, globe, butterfly</td>
<td>material of stem, thread, packing, disc and guide effects, sealing loads.</td>
<td>valve factor-disc to seat and guide friction effects, effect of dP</td>
<td>Valve factor, thrust measurements during static and dynamic tests. Dynamic at flows as close to design base as possible.</td>
</tr>
<tr>
<td>extras</td>
<td>inertia or load sensitive behaviour, ageing mechanisms,</td>
<td></td>
<td></td>
<td>storage of data from commissioning for trend analysis over life of plant. Maintenance of EQ profile.</td>
</tr>
<tr>
<td>system parameters</td>
<td>mass flow rate, temperature, dp, line pressure, phase and chemistry</td>
<td></td>
<td>Used to establish representative test conditions</td>
<td></td>
</tr>
</tbody>
</table>
6 OUTLINE OF ASSESSMENT ROUTE FOR MOV OPERABILITY

Reference to fig 1, and in particular the part relating to design basis analysis, is the first stage of the assessment process. In this section, the early work on valve qualification, diagnostic testing and the results are discussed.

6.1 INITIAL TESTING AT MARCHWOOD FOR MOV ISOLATION TESTING TO ANSI B 16.41B

The Marchwood Valve Test Facility was commissioned in 1982 to support valve testing and qualification for the proposed UK PWR programme. Facilities were provided for:

- Operability testing of candidate isolation valves using HP or LP Test Loops to test valves for endurance, for a simulated lifetime under PWR primary circuit conditions.
- Isolation valve flow interruption qualification tests in accordance with the requirements of ANSI B 16.41 for valves selected for Sizewell B procurement.

The facility is considered to have fulfilled its requirements and has now been closed. Some of the results of the test work conducted are described below with detailed qualification test work described in the next section.

Parallel slide gate valves suffer from inter-gate effects due to the entrapment of a 'solid' volume of water between the discs. This is alleviated by the provision of an inter-gate relief line system, with a separate manual isolation valve.

Pressure locking of wedge gate valves has been relieved by drilling a hole in the upstream disc of the valve or seal face as appropriate.

Following valve closure, during high energy flow-interruption tests, enhanced wear on parallel slide valves was evident at the disc to guide contact points. In the case of wedge gate valves high unseating torque was required and such effects were consistent with findings in the US during testing and PWR operation.

ANSI B 16.41 permits the generic grouping of valves for qualification. A number of valves were tested in the facility and on completion, the licensee showed that all valves closed against their design basis flows. During this work, a number of valves were instrumented to give stem load and displacement etc. and these will provide some base line data prior to commissioning. For the remainder, NII expects that they will be instrumented prior to flow testing to determine packing load effects, and then tested under normal and as near to design base as possible during commissioning. In the latter conditions, data will be collected which will provide the initial data base for trend analysis over the future operating years of the station.

Fig. 2  Rig result-packed Load Test

Fig. 3 from commissioning
3 in. parallel slide valve
Typical traces for an 80m.m. parallel slide valve are shown in figures 2 and 3 on the previous page. One of the results of the work was the recognition that the use of strain gauges mounted on the valve stem could provide the means of determining stem thrust. Since the valve and actuator manufacturer's test data is available, back calculation to obtain the stem nut friction term should be possible. The data may be used for comparison purposes between test and plant valves; although it is recognised that individual valves could behave differently due to effects such as manufacturer's tolerance variations etc. In addition, limited work on measurement of motor power to the actuator suggested that this could be a means of detecting problems with the actuator, but it is unlikely to be able to determine changes in disc factor—especially for parallel slide valves which close on position, via the limit switch, rather than on torque.

6.2 ACTUATOR PERFORMANCE AND QUALIFICATION-Fig 6
The Cat. 1 safety actuators for Sizewell B have been qualified to IEEE 343 and have therefore been demonstrated to develop their required output under reduced voltage for the loadings defined in that standard. This information plus the dynamometer data, including that at torque-switch trip, provide confidence that the results of the MOV diagnostic programme will establish adequate thrust and torque margin. In addition, the actuator manufacturer has been asked by Nuclear Electric to provide a 'weak-link analysis' to give increased assurance that additional margin is available, should this be needed.

Evidence so far suggests that some actuators appear to be over-designed and over-thrusting could be a potential concern. Defence against this will be reviewed when the programme has been completed.

6.3 COMMISSIONING TESTING

Having established the design base thrust and torque requirements, the flow interruption testing that has been conducted during qualification, and reviewed the actuator qualification, the next part of our assessment process is to review the testing during commissioning and determination of margin.

The licensee's approach has been to apply the 'standard industry equation' for stem thrust. Since the valve factor encompasses several terms including disc/seat, disc/guide, stem/nut friction terms, it is imperative that, where possible, the best means to reduce them, in the determination of thrust and torque, be taken. In addition, aspects such as load sensitive behaviour, and inertia effects, suggest that the best results will be obtained by testing as near to design-base as possible. Nuclear Electric is aware of these concerns and has accepted the need for such testing wherever practicable.

As fig. 7 below indicates, the test sequence for a particular valve involves the established practice of diagnostic testing with zero gauge pressure, system pressure, and finally flow conditions leading to those as near to design-base as possible. This sequence should enable the packing load term and disc friction terms to be examined prior to design base thrust. The conversion of torque from the actuator to the valve disc will also be examined. The stem factor will be determined from the thrust and torque strain gauge output or by working backwards from the disc factor in the valve equation.
6.4 DIFFICULTIES ENCOUNTERED WITH CERTAIN VALVES

The following represent challenges to the strategy described:

Butterfly valves: the design prohibits suitable access in order to install the diagnostic equipment currently being used. The licensee intends to re-examine the vendor's qualification data: certain of these valves are of French manufacture and use diagnostics on the actuators provided for these valves. Further information will be provided in the second part of the presentation. The NII position is that these valves should be addressed for MOV operability. One method of ranking the work involved in predicting operability margin for these valves could be on the basis of the severity of required closure loads; line break valves being such an example.

Parallel slide valves: since they do not close on torque but limit, do not have the problems of insufficient torque and torque by-pass familiar to wedge gate valves. However, some are required to isolate in two directions, suggesting added sophistication of testing. In addition, inter-gate pressurisation, or the difficulty of determining torque-switch trip on a valve that should close smoothly on position are typical difficulties with the use of MOV diagnostics on parallel slide valves.

In addition, in the case of strain gauge installation, there is the problem of availability of smooth stem region—it is very limited in some of the valves to be tested. Many of the valves have anti-rotation devices, which limit access of equipment, the small region of available stem region is further reduced by the thread run-out region and a key-way. Further details are provided in the second part of this presentation.

6.5 DEFINITION OF MARGIN

Margin will be defined on an individual system and valve basis. The flow testing and the associated diagnostics must establish that the calculated closure load in the safety case is produced in practice and that adequate margins for errors in measurement and ageing are included.

PART 2-EXPERIENCES FROM THE TESTING OF MOVs AT SIZEWELL B

Details are provided below of some of the more interesting findings of the programme completed so far. One benefit from the programme so far has been the detection of equipment that has been found installed so as not to conform to the safety case. While it is true that it is part of the job of commissioning to detect such errors, the diagnostics available have made this task much easier. Examples such as incorrectly set torque switches and incorrect clearances on valves are described below.

9

53
6.6 COMMISSIONING TESTS PERFORMED SO FAR

Sizewell B has 165 category 1 MOVs, of which 54 are parallel slide (close/open on limit), 29 are wedge gate (close on torque), 28 are globe valves (close on torque), and 54 butterfly valves (close on limit). The actuators and valves are of UK manufacture with the exception of the Butterfly valves of which some are French. The latter have no exposed stem, hence testing is difficult at present, the licensee hopes to obtain dynamometer data from the manufacturer in order to justify margin.

The actuators have a number of features that appear specific to the UK design, in particular the stem casing is grease packed such that the stem/stem nut is lubricated with each stroke and the actuator has an oil filled rather than grease packed housing. Several different sizes of actuator are used but only a single motor torque rating. Live load packing is used on valves over three inches in diameter.

As discussed in part 1 of this presentation, three tests are performed for each valve where practicable: a no-load test; a static pressure test; and a test under near full flow dp. To perform these tests, 123 were fitted with strain gauges which are intended to provide both torque and thrust. Static tests will be performed on all valves and dynamic tests on 150 valves.

The typical industry valve sizing equation, described in part 1 is used, where the summation of the stem packing load, stem ejection load, and derived disc factor load are combined. About 25% margin is used for uncertainties.

6.7 THRUST AND TORQUE DETERMINATION

One recent problem encountered has been that the torque derived from the strain gauge output appeared, in some cases, to be considerably higher than was expected both from the actuator's dynamometer information and the assumed stem nut friction term. The latter effect was reflected in a lack of linearity between a known standard (see below) and the strain gauge output. In order to determine where the losses might be coming from the anti-rotation device was removed. However, errors were still evident. After, repeated testing it was found that incorrect curing had caused inadequate bonding of the strain gauges to the attachment clip. Subsequent retesting indicated that a linear relationship between torque and thrust was re-established.

However, during the above examination, it was evident that some losses from the actuator were still unexplained. It appears that these losses may be explained by the design of the Sizewell B actuator; a thrust plate is provided at the base of the actuator in order to prevent undue loading of valve components. The plate contains a bearing, work is currently being performed to determine the friction coefficient of this bearing, since it is believed that this region of the actuator is contributing to the approximately 10% errors currently observed.
The work is currently being conducted at Berkley Laboratories and involves the use of a test stand where pure torque and thrust can be imparted to a valve stem and measurements with strain gauges compared to the actuator output.

In addition the above testing enabled NE to examine other devices for accuracy and also to explore the affects of the thread run-out and key-way affects on the strain and torque outputs. Results so far indicate that the devices examined have given good accuracy and that the effects of the strain concentrations appear to be acceptable. NII has not examined these results but it is our intention to do so when the programme of operability testing is complete.

6.8 TORQUE AND LIMIT SWITCH SETTINGS

The use of diagnostics has enabled the licensee to identify the limited number of actuators where the torque output has not matched the dynamometer information supplied with the component. In addition, limit switches have been found incorrectly set.

6.9 OVERTHRUSTING

Generally, the actuators on Sizewell B appear to be over-designed. There is therefore a risk that the actuator if incorrectly set-up could over-thrust resulting in valve damage. While the potential for over-thrusting will be examined during our review, some is due to incorrectly adjusted torque switches. The latter have been determined during site testing and will be rectified.

6.10 UNDER-VOLTAGE EFFECTS

Testing is being done to ensure that actuators, that have been qualified to IEEE 382 perform their rated duty during flow testing. In general the calculation of cable loadings appear conservative, however, testing is still underway and some problems have been found such as those due to cable voltage drop and cable gland disconnection.

6.11 HIGH PACKING LOADS

The equipment has been used to determine high packing loads-valves greater than 3 inches in diameter have live loaded packing, in some cases this had been incorrectly set.

Associated with the above loads was the example of a valve which had been fabricated apparently out-of-tolerance. Once the static testing had been commenced it was apparent that the thrust for valve closure was increasing in a step-wise and jerky manner with excessively high values. On strip-down it was evident that damage in the disc guides had occurred as a result of poor manufacture.
thrust traces showing stick-slip behaviour caused by incorrect tolerances in valve guides

6.12 INTER-GATE RELIEF VALVE CLOSURE

This valve is provided to prevent pressurisation due to fluid ingress into the bonnet of the valve and, during ensuing heat up, high loadings on the valve disc that prevent its operation. In this test the inter-gate relief valve had been left closed and the resulting diagnostic trace showed a considerable increase in required thrust.

Effect of closed inter-gate relief valve on closure thrust
Traces indicating increasing closure force during full stroke static testing

Full trace and enlarged portion showing increased closing thrust during closure for parallel slide valve

6.13 CONCLUDING COMMENTS ON COMMISSIONING SO FAR

The above examples are given relatively early in the MOV programme and are described to show the additional benefits that have so far been accrued. Some of them might have been found by more conventional testing but it is unlikely that as many or in such an organised manner. One reason for the need for sophisticated diagnostic techniques is that the components used as safety equipment on nuclear plant are seldom uniquely manufactured for the nuclear industry; frequently they are supplied by manufacturers who are more accustomed to supplying general engineering industries. Although the procurement procedures and qualification should ensure that the plant will perform in accordance with the safety case, in practice the demands on that equipment from the many fault sequences included for plant designed to modern
standards, place considerable demands on those components and diagnostics are, it is suggested a necessary means to establish operability.

7.0 CONCLUSION

This paper has discussed the principal regulatory issues relating to the testing of safety related motorised valves for the UK's first PWR- Sizewell B. From those issues, it has been shown how the licensee, Nuclear Electric, has produced a programme of qualification testing, and the use of diagnostic techniques, in association with the manufacturers, to give confidence that the issues will be adequately resolved prior to fuel load.

Probably the greatest single advantage over previous programmes has been the extent of flow testing as opposed to analysis and the International experience to draw on.

The results discussed in this paper indicate that, such are the complexities involved in the adjustment of some of the more important safety valves on a PWR, that any future stations would be expected to similarly provide assurance of operability by full flow testing.

8.0 ACKNOWLEDGEMENT

Contacts between ourselves and NRC have enabled us to keep up to date on this topic, which is addressed in USA through Generic Letter 89-10. In addition participation by the licensee at the Motorised Valve Users Group, and other bodies, has ensured that the best International effort has been made available to support this work for Sizewell B.

Many of the views expressed in this paper are personal and therefore do not necessarily reflect those of HMNII. Nevertheless I wish to acknowledge the helpful discussions with NII staff during the formulation of those views and the time made available to produce the paper.

Much of the test information has been made available by Nuclear Electric, largely as a result of our discussions relating to their MOV programme and the Sizewell B Safety Case.

In particular, I should like to acknowledge the help of staff from N.E's PWR Project Group, Knutsford, and those at Sizewell B site.

DC Anderson 22 March 1994
Session #2

OPERATING EXPERIENCE

Chairman: Dr. V. Tolstykh

Seven presentations, made by the representatives from France, Germany, Hungary, Slovak Republic and the United States, provided detailed information on valve-related problems such as inadequate MOV design, incompleteness of testing and diagnostic techniques and difficulties in operation of safety relevant valves.

French specialists put a great deal of effort to better understand MOV problems on failures and uncertainties resulting from their initial sizing. The improvements of French diagnostic technique programme are aimed at ensuring sufficient MOV operability margins for accident conditions and predicting these margins during test conditions.

Results of an in-depth investigation of operating experience, carried out in Germany, were presented in terms of significant failure modes for different types of MOVs. A variety of different causes leading to failure of MOVs has been observed, namely loosening of internal connections, binding due to thermal effects or pressure differences, increased friction between spindle and its nut, wrong actuator type, inadequate/wrong settings, failure of limit or torque switches, failures in actuation logic, etc. Some failure modes clearly indicated a potential for common cause failure of safety related valves.

The U.S. presentations confirmed once more that U.S. organizations are getting more and more knowledge of and experience on MOV issues and they continue to play a leading role in this area.

Pressure locking and thermal binding of gate valves represent potential common-cause failure that could render redundant trains of certain safety related systems or multiple safety systems inoperable. These MOV problems were well addressed by the U.S. NRC representative. One important observation was that the most successful approach identifying valves susceptible to pressure locking and thermal binding had been the use of an interdisciplinary team composed of valve experts, systems engineers and plant operations staff.

Commonwealth Edison's La Salle station embarked on an air-operated valve (AOV) corrective programme because of the increasing trend in a number and severity of problems. The problems involve design installation, operating and maintenance considerations. To date, results of the programme include improved system control, smoother start up, no valve rework to correct inadequate repair, and most notably, a drastic reduction in heat loss into the condenser. Achievements, obtained at the present stage of the programme, provide a sound technical basis for future activities in this area.

59
The representative of the Texas Utilities Company (TU Electric) summarized what his company has learned regarding MOV performance characteristics. The TU Electric MOV programme is oriented to implement the U.S. NRC Generic Letter 89-10 Recommendations.

The paper from the Slovak Republic represented itself an annotated list of events which involved MOVs problems.

The representative from Hungary described the Paks NPPs valve maintenance programme.

All presentations at the session "Operating Experience" were well received by the audience and were intensively discussed.
MAIN FINDINGS
ON MOTOR OPERATED VALVES
OPERATING EXPERIENCE
IN FRANCE
***
JOINT AEN/AIEA SPECIALIST MEETING
ON MOTOR-OPERATED VALVES ISSUE
PARIS, FRANCE
APRIL 25th - 27th 1994

R. ZERMIZOGLOU
INTRODUCTION

Over the years problems relating to various aspects of the design, testing and operation of motor operated valves (M.O.V.) have occurred. Two main events have raised a general concern for the ability of motor operated valves of safety related systems to perform as intended.

The former occurred in February 1986 during the commissioning tests of Unit 1 of Nogent Nuclear Power Plant (NPP) where an apparently minor incident led to discover a generic anomaly affecting over 30 M.O.V. of the safety injection system (RIS) of all 1300 MWe plants. The important fact concerning this anomaly is that its discovery was fortuitous and remained undetected during a long time.

The latter which occurred in May 1991 during the Bugey 5 NPP refueling outage reinforced the concern in valve operability since it made evident that pressure and thermal hydraulic binding effects, already experienced and corrected in some cases during normal operation, could also affect the operability of a large number of safety related valves during accidental conditions.

Both events have been the starting point of extensive and time consuming studies and corrective actions achievements of which are still in progress. This paper recalls the events circumstances of problems identification, the Institute For Nuclear Safety and Protection (IPSN) involvement in their assessment and the corresponding licensee efforts to solve them.

1. UNDERSIZING OF MOTOR OPERATED VALVES

During functional testing of the 1300 MWe Nogent 1 NPP in Novembre 1986, difficulties were encountered in operating the safety injection system motor operated valves (M.O.V.), namely, the isolation valves of the low pressure pumps recirculation lines to the water storage tank (PTR) (Fig.1).

In view of these faults, in February 1987, a preliminary investigation campaign was carried out to measure the torque loads at the different points in the transmissions to determine the positions of the friction points. The actions then planned by the vendor were inspecting, cleaning and if necessary changing the valve stops, the origin of the fault then being attributed to cleanliness problems.

More pronounced investigations, made at the request of IPSN, and confirmed by others 1300 MWe units in Belleville and Cattenom revealed the generic nature of this problem, which affected respectively 33 & 35 M.O.V.s in the safety injection system of each unit of P4 (8) and P4 (12) series.

The origin of the fault was that the equipment had been operated before completion of the safety studies; when these latter resulted in some changes in functional requirements, no verification was made to ensure that the equipment ordered was still compatible. The result was undersizing of the valve drives in over 30 valves.

Other contributing factors to the problem were lack of instructions concerning assembly and post-assembly checking, an inadequate testing programme, lack of documentation of test results, and lack of diagnostic tools.

SAFETY SIGNIFICANCE

This valve drive system undersizing constitutes a common mode failure in both trains of the safety injection system.

Two main types of faults can affect motor-operated valves: leakage and failure to operate. The latter is more serious in terms of safety. The failure of one or more valves of safety injection system and
the spray containment system to operate can compromise cooling of the core and lead to serious damage to the fuel.

Upstream/downstream leakage of isolating valves can result in uncontrolled discharges, particularly in post-accident phases in parts of those circuits that are extensions of the containment.

Calculations of the radiological consequences have shown the significance of the discharges in the event of upstream/downstream leakage in the minimum-flow lines of the safety injection pumps or containment spray pumps. Such a failure was introduced in the preparation of a scenario of a crisis exercise simulating an accident and performed in May 1987 on Paluel 1 NPP and was identical to that which revealed latter the fault during the tests at Nogent 1 NPP : incomplete opening, unsignalled in the control room, of a minimum-flow valve of a RIS pump connected to the safety injection water storage tank (PTR).

NRC issued information Notice n° 91-56 "Potential Radioactive leakage to Tank Vented to Atmosphere " on September 1991, to inform all holder of operating licencees of this potential problem.

FOLLOW UP OF THE ANOMALY

Upon discovery of the faults the licencee was requested :

to establish rapidly the nature and extent of the fault .

to take short-term remedial actions,

- effect a complete M.O.V. design review to enable identification of long-term corrective actions,

- investigate the design quality control, the validity and adequacy of the startup and testing programs, and the implications for future designs.

As soon as difficulties in operating the M.O.V. of the safety injection system were encountered, close attention has been given to them by IPSN including an on line analysis of findings and corrective measures taken by Electricité de France (EDF) which necessitated during a short period, multiple exchanges (meetings, letters, attending tests...) with the different EDF services involved as well in design studies as in implementation and testing of corrective actions on site.

The dossier supplied by EDF in July 1987, gave for each of the governing parameters (system service pressure, differential pressure and flow rate) and for each M.O.V. of the safety injection and containment spray systems, the required value and the value specified in the order.

No valve was found to have been exposed to more than nominal pressure, i.e. all the valve’s bodies and the internal closure and sealing devices were designed for the maximum pressure of the fluid in the circuits.

For the design parameters actuators, no valves of the containment spray system were found to be at fault, on the other hand in the safety injection system, 33 anomalies were found in the P4 units and 35 in those of the P4 series.

Deviations in design parameters concerned flow rate and differential pressure and sometimes both parameters.
Differential pressure deviations were examined on a case by case basis in order to determine the impact on actuators.

As concerns the errors relating to flow rates resulting in dynamic loading, justification was made on a case-by-case basis to determine their acceptability or the need for modification.

In this examination, it was found that, there were grounds to doubt the full operability of the following eight valves:

- RIS 25 to 28 VP (low pressure safety injection zero flow to PTR),
- RIS 31 and 32 VP (partial isolation of the low pressure safety injection flow to the cold legs during direct injection switchover),
- RIS 51 and 52 VP (partial isolation of the low pressure safety injection flow to the cold legs during simultaneous injection switchover).

On the other hand, the other valves, for which the exceeding of limits was observed, were declared servicable.

In November 1987, EDF notified the vendor of valves 45 to 48 VP (medium pressure RIS pump minimum-flow to PTR) of the differences concerning the observed flow rates, informing them that it doubted that they would operate properly and that jamming of the valves could be anticipated.

As a result, corrective actions have been planned for the 12 valves.

**SHORT-TERM MEASURES**

The first measures taken and implemented in early June 1987 on Unit 2 of Cattenom NPP, were the following:

- replacement and requalification of the isolation valves actuator systems of the low pressure RIS pumps minimum-flow lines to the PTR tank (25, 26, 27 and 28 VP),
- increasing the setting of the torque limiters of RIS valves 31 and 32 VP, which are required to close during switching from cold leg injection to simultaneous injection (hot leg and cold leg), and issuing a temporary instruction to provide for valves possible failure to close in the accident phase by using the small primary breach by-pass line, pending a permanent solution.

On June 1987, EDF proposed remedial measures so as to be able to continue with the start up and operating programs of the 4 units, pending replacement of the valve actuators.

Temporary operating procedures involving local confirmation of valve closure were implemented, after checking their applicability by means of on-site tests and after agreement by IPSN.

**LONG-TERM MEASURES**

For the valves considered to be faulty, EDF has implemented in all 1300 MW units the following corrective actions:

- RIS 25 to 28 VP (low pressure safety injection minimum-flow valves to PTR).
In view of the difference between the required flow rate value at the time of the design of these valves and the design flow rate of the valves installed on-site, the modification consisted in replacing the gearmotor by another model, giving a lower output speed with a proportionally higher torque. To check proper operation and to guarantee the operability of these valves with time, EDF agreed to carry out checks after complete disassembly of these valves every ten refueling outages.

- RIS 31 and 32 VP (low pressure safety injection "simultaneous" injection valves to cold legs).

- RIS 51 and 52 VP (medium pressure safety injectitn "simultaneous" injection valves to cold legs).

The difference between the required and as-ordered parameters for valves 31 and 32 VP concern the differential pressure and the flow rate; this led EDF to replace the actuators by equipment twice as powerful, which also entailed redesigning the remote control systems.

The parameter differences for valves 51 and 52 VP only concerned the differential pressure. The actuators were replaced by more powerful models. The remote system was unchanged and the servomotor mounting has been replaced.

Requalification tests have been carried out on unit 2 of Belleville NPP in a specific test configuration (reactor vessel closed and pressurizer manhole open). Additional measurements have been made during execution of these tests (measurement of breakaway torque with a torque wrench, exact measurement of the valve full-stroke time with no flow and at full flow, and recording of motor currents).

- RIS 45 to 48 VP (medium pressure safety injection minimum-flow valves to PTR).

The difference between the flow rate value required at the design stage and the valve design flow rate value led the vendor to call into question the operability of these valves. In particular, the vendor established a maximum speed criterion for the fluid beyond which there is a risk of failure in operation (closure and opening) at stem guide level. As this criterion was exceeded, EDF chose to carry out on-site tests under the worst possible conditions to demonstrate that the valves could nevertheless fulfill their functions. The tests were carried out on Unit 2 of Nogent NPP. The results of the 50 opening cycles showed satisfactory behaviour of the valve and EDF decided to keep these valves in their original condition.

LESSONS LEARNED

One has to keep in mind that reason for the discovery of the anomaly was fortuitous since incomplete closing of the valves could only be detected locally.

The fault remained undetected until the commissioning tests of the eleventh 1300 MW unit, and besides first corrective actions implemented after the fault discovery relied on new settings and cleaning and greasing the couplings. It is probable that if such measures had given good results, the generic anomaly would not have been discovered.

Assessment by IPSN of this anomaly revealed weaknesses from the design level to on site assembly and testing level:
- **Sizing**: design deviations due to a design change in the system without updating the parameters used for actuator selection.

- **Qualification**: conditions could not be extrapolated to design basis conditions.

- **Installation**: lack of instructions concerning installation and post installation checking.

- **Commissioning tests and periodical tests**:
  - dynamic tests conducted only once in the lifetime of the NPP,
  - periodical tests conducted in hydraulic conditions far from maximum loadings practicable in normal conditions,
  - testing methods ineffective in detecting failures, no record of the operating parameters of the different components

In conclusion, main lessons learned from this anomaly concern:

- the necessity to improve the quality control from the design level to on site assembly and testing level,

- inadequacy of current methods used in commissioning and periodicaly testing to assure that M.O.V. will operate when needed and consequently the need to develop and promote the use of diagnostic methods able to predict and identify degradation of valves before failure to operate occurs.

**REQUESTS TO EDF**

Different requests were addressed to EDF following this anomaly concerning:

- identification of design basis loadings,

- sizing methods for valves, valves actuators, valves couplings,

- qualification programs,

- installation quality control,

- research of the worst hydraulic configurations practicable during commissioning tests conditions,

- diagnostic methods development.

Initiatives taken since the discovery have made it possible to correct the deviations observed in the units. EDF has made a certain number of arrangements concerning quality control at design level and on-site installation. Similarly, design changes are from now the subject of modification sheets issued to all departments involved (including those in charge of ordering the equipment) verify the impact of the changes.

Overriding requests were related with:

- sizing method of valves, actuators, couplings and torque limitors fitted to design basis loadings,
developments undertaken or realized to implement M.O.V. diagnostic techniques, like those in practice in foreign countries, capable to correctly assess, trend and evaluate valve failure modes.

EDF RESPONSES

In response to the first request, and within the framework concerning studies of the N4 series (1500 MWe), a joint research and development program between EDF and Framatome has been undertaken in quantifying opening and closing thrusts equations for each type of safety related valve in operation in the different series of units.

New rules have been so established, coherent with the international practice evolution, and applied for actuator sizing checking. These rules however do not take into account thrust values to overcome pressure or thermal binding effects. These effects are considered in another study.

Results of this checking is that most actuators are correctly sized and only a few have to be reviewed.

As concerns the diagnostic method, EDF declared it had experimented with a system analogous to the MOVATS system in Bugey NPP since 1985. Though interesting results were obtained, this system suffered according to EDF some limitations leading the licencie to give priority to preventive maintenance programs improvements while developing at the same time its own diagnostic device (SAMIR).

Such a device has been qualified on Bugey and Tricastin NPP in 1991 and is being extensively implemented on all 900 MWe units and will be implemented on 1300 MWe units after mid 1994.

2. UNIVERSAL JOINT RUPTURE INCIDENTS - (NOGENT 2, OCTOBER 1987)

On 19th October 1987, at Nogent 2 NPP during commissioning tests of RIS motor operated valves, the rupture of an universal joint of valve RIS 30 VP coupling was observed (Fig. 2 et 3).

Previous ruptures had occurred during commissioning tests at St Alban NPP in 1985 and another occurred latter at Penly 1 NPP in 1988.

Investigations showed that, subsequent to crossing the actuators of valves RIS 30 and 34 VP, valve RIS 30 VP was underpowered.

Because this failure did not occur on previous plants on the 900 MWe series, and because of its potential consequences (a rupture may result in a complete unavailability of the valve), an extensive study was made to understand and solve this problem including, inspections, expertise of cracked cross bars, and dynamic torque measurements on a bench that reproduced the installed configuration.
LABORATORY INVESTIGATIONS

These couplings were submitted to breaking torque. In all cases, the cross arm failed at an average value of 530 mdaN, which is about twice the maximum value used by contractors in designing couplings. Maximum torque values of 500 and 580 mdaN were obtained respectively for Nogent and Penly valves in case of torque failure. Inertia effect at closure was found to be very important with torque limit switch failure.

In the Penly and Cattenom cases, inversion of the power supply cables caused the stem to move in the direction opposite to that expected, the open limit switches did not activate and the actuator stalled when the stem stopped on backseat occurred.

This explained both the overtorquing and why breakage occurred during opening.

ACTION TAKEN

Conclusions drawn from these examples are that correct operation of a M.O.V. necessitates a close match of valve and operator. Excessive loading may occur in normal operation from inaccurate torque limit switch setting and inertia effects at closure.

Since an operating experience feedback review indicated satisfactory reliability of torque limiters, EDF considered there was no need to add, as intended before, other devices such as torque limiter failure detector or a second torque limiting device located between the actuator and the valve.

Moreover, EDF precised that means such as stem limit switches and thermal overload protection could detect such faults. Nevertheless the preventive maintenance program had to be improved concerning the greasing of the stem lubrication.

3. NEW INCIDENTS ON SAFETY INJECTION VALVES (BELLEVILLE 1 - DECEMBER 1991 / NOGENT 2 - JUNE 1992)

Two significant incident reports were issued concerning the containment sump recirculation valves RIS 09 and RIS 010 VP on Belleville 1 and Nogent 2 NPP.

BELLEVILLE 1

On December 2nd 1991, with the unit in cold shutdown, while performing a periodical test, valve RIS 09 VP (train A) failed to open. After intervention it was discovered that there were no more threads on the manoeuvre nut.

When replacing the valve actuator, it was observed that the torque limiter was set at 70 mdaN instead of 45 mdaN.

Investigation on the corresponding valve in train B led to the same constatation. According to the supplier, torque limiter set at 70 mdaN was hiding a grease deficiency. The valve was able to operate until thread nut destruction by wear. The cause of the setting modification was not found. (Valves RIS 09 and 010 VP had operated respectively 242 and 160 times)

A fast operating experience feedback form was sent to the plants to inform them of this potential problem.
NOGENT 2

On June 1992, while performing a RIS 09 VP valve operability test, the valve was found locked in an intermediate position. The valve was declared inoperative and according to technical specifications requirements (allowed outage time: 10 hours), the reactor was run to cold shutdown state depressurized.

The licensee assigned the root cause of this anomaly to a grease deficiency which would have induced a premature wear and a sticking of the moving parts.

This valve had operated only about 100 times since the commissioning tests.

LESSON LEARNED

Surveillance testing based on valve’s ability to satisfy test acceptance criteria proved to be insufficient to detect any degradation since the valve operated until complete failure occurred.

This demonstrates again the need of additional measurements to be performed in order to assess the margin between actual valve condition and valve failure.

ACTION TAKEN

Considering the common mode failure risk due to these valves after a quite low number of operations and aware of the efforts carried out by EDF to develop a M.O.V. diagnostic device (SAMIR) IPSN recommended in as a first step, that the M.O.V. preventive maintenance program was completed with the recording of the motor current trace during a close-open cycle.

First results at Nogent 2 NPP led EDF to extend those controls to all safety related M.O.V.s every 3 refueling outages, pending the use of device SAMIR on the 1300 MWe units.

4. FAILURE TO OPEN OF A CONTAINMENT SUMP RECIRCULATION VALVE - (BUGEY 5 - MARCH 1991) - THE PRESSURE LOCKING EFFECT

BUGEY 5 TESTS (MARCH 1991)

Failure to open of valve EAS 14 VB by torque limiter trip occurred in March 1991 during the decennial outage of Bugey 5 NPP while operating the valve with a differential pressure of 0.38 bar after checking the leaktightness of the corresponding containment penetration.

After modification of the torque limiter setting and changing the Belleville spring pack the valve could be opened under a differential pressure of 2.3 bar (afterwards expertise of the Belleville spring pack revealed no aging, stiffness was identical to the initial specification).

Valve EAS 13 VB of the other train was opened without modifying the torque limiter setting.

Valves EAS 13 and 14 VB are containment sump recirculation valves. (Parallel expanding gate valves) They are normally closed and are operated during monthly periodical testing without any differential pressure and with full differential pressure during decennial outages like containment spray valves.
They are required to open after a loss of coolant accident (LOCA) during the recirculation phase with an internal-external differential pressure of 2.3 bar (this value corresponds to the internal containment pressure 30 mn after the incident).

Testing of containment sump recirculation and spray valves under full differential pressure was included in the periodical testing program of the containment spray system at the request of IPSN. It necessitates use of special devices and requires special care.

COMPLEMENTARY ON SITE TESTS

After this failure, EDF informed IPSN of complementary tests to be performed during the next refueling outages in order to:

- better understand the failure of the torque limiter,
- assess the relationship between the torque limiter setting and the maximum ΔP compatible with the valve opening,
- check the behaviour of other types of valves.

New tests conducted at Bugey 5 NPP showed that the actuators could open the valves with a 5 bar differential pressure with the following torque limiter settings:

13,2 mdaN for valve EAS 13 VB
16 mdaN for valve EAS 14 VB

BUGEY 2 TESTS (MAY 1991)

During the Bugey 2 NPP outage, failure to open of valve EAS 14 VB was observed under a 5 bar differential pressure with the torque limiter set at 19 mdaN and with the torque limiter shunt. During this test, a fast depressurization upstream from 5 to 1 bar had maintained a residual pressure of 4.3 bar inside the bonnet valve making evident the pressure locking effect.

FESSENHEIM 2 TESTS (MAY - JUNE 1991)

Tests were performed during Fessenheim 2 NPP refueling outage (Fessenheim NPP has the same type of valve but its actuators are less powerful).

Aware of the pressure locking effect, tests were conducted maintaining a pressure on bonnet valves EAS 13 and 14 VB with no differential pressure upstream and downstream the valves.

The valves failed to open with the torque limiter set at 13 mdaN but succeeded with a setting of 16,7 mdaN, without any torque limiter trip, with a degraded supply voltage UN -15 %, and a bonnet pressure of 2,7 bar.

TRICASTIN 2 TESTS

Tests conducted at Tricastin 2 NPP revealed no problem. Valves EAS 13 VB could be opened with a torque limiter set at 15,3 mdaN under differential pressure of 2,3 and 5 bar.

TRICASTIN 2 valves (Bouvier Darling - Parallel sliding gate valves) are different from Bugey and Fessenheim NPP valves., their actuators are the same as Bugey’s.
FIRST ACTION TAKEN

Upon discovery of the failure to open of valve EAS 14 VB at Bugey 5 NPP and until the failure cause could be enlightened decision was taken to shunt the torque limiter of these valves on the 4 units of Bugey and to maintain it on operation only during periodical testing in order to avoid any valve degradation by mechanical hardening.

During previous decennial outage of Bugey 2 and Bugey 4 NPP, no problem was encountered when testing these valves at full differential pressure.

This temporary instruction has been extended to Fessenheim NPP and was latter included in the post accidental procedures of these plants on the basis of tests performed at Bugey NPP which confirmed that the valves could be manually fully opened within 2 minutes.

EDF LABORATORY TESTS

In order to improve the understanding of the pressure locking effect and to precise the characteristics responses of actuators, tests were carried out in EDF laboratories with valves and valves actuators, including separate tests of these components. Main findings are summarized hereafter:

- pressure locking effect was observed on both types of valves like those of Bugey and Fessenheim NPP,
- besides parallel expanding gate valves were found sensitive to the thermal hydraulic binding effect.

Actuators characterization revealed:

- a good reliability of torque limiter switch settings,
- a weak sensitivity of actuator voltage supply on the torque limiter trip,
- available torque values greater than guaranteed values specified by the manufacturer.

LESSONS LEARNED

Following lessons are drawn from on site tests on containment sump recirculation valves and tests results in EDF laboratories:

- the risk of parallel gate valves failure to open is a real risk because of high pressure water trapped in the bonnet valve; it is a common mode failure that may affect all series of units,
- periodical tests proved to be ineffective in detecting the problem, even when performed under most severe loadings practicable on a site (in this case with the maximum differential pressure expected during an accident),
- insurance of valve operability in severe conditions involves to consider a greater variety of conditions,
- though pressure locking effect had been experienced for a long time in some specific cases, [hydraulic thermal binding effect for heat removal system isolation valves, main steam isolating valves, and pressure binding effect for containment spray valves (CP0 serie)], in normal operating conditions and corrected by modifications, the effect did not lead to further
investigations, this showed a failure to anticipate problems in other conditions or with other valves.

- the valve locking effect was neither identified by the normal qualification process which is supposed to demonstrate valves operability during design basis accidents.

**ACTIONS TAKEN BY EDF**

In order to solve this generic problem, EDF's methodology has been to:

- establish an inventory of gate valves affected by the pressure locking effect,

- associate for each valve a detailed safety analysis in order to select valves and systems deficiencies, which in case of LOCA:
  
  . have a relatively high potential for leaking from the containment to the outside atmosphere,

  . and those that would be required to prevent and mitigate the consequences of such accident,

- implement the appropriate modification:

  Following solutions have been retained:

  . for disc valves required to be tight in one direction: the modification is a hole in the upstream disc for normal flow direction, and a hole in the downstream disc for reverse flow direction,

  . for valves required to be tight in both directions the modification is an external by pass line connected between the valve body and to both upstream and downstream pipes with a passive selector valve turning the over pressure to the lowest pressurised side.

A list of safety related valves inside the containment has been selected which may be subject to the pressure and thermal hydraulic binding effects.

M.O.V. opening of which is not necessary after an accident may be eliminated from this list; nevertheless for containment isolation valves, the decision relies on results of special tests carried out on the different types of gates valves in order to measure their leak rate and check their integrity when submitted to the thermal hydraulic binding effect.

Before an extensive implementation, EDF intends to qualify these modifications on a single plant. In the case of N4 series they are implemented at the design stage.
6 LUBRICATION DEFICIENCY ON VALVE ACTUATORS (TRICASTIN2 - MARCH-MAY 1993)

On March 1993 during a periodical test, actuator of valve RIS 077 VP tripped by actuation of its thermal overload protection. The actuator was replaced and its expertise revealed a heavy degradation of the worm/worm gear couple and a grease deficiency.

This valve is on the boosting line of the high pressure safety injection pump by the low safety injection pump and is opened by the safety injection signal. Similar incident occurred in January 1991 during the same test, worm/worm gear couple of the actuator was changed.

During these 3 years all the actuators had their aging grease replaced by a new grease of better quality.

SAFETY SIGNIFICANCE

On May 1993 during the refueling outage grease deficiency was discovered on six actuators of the same type.

Because half of the safety related valves actuated by a protection signal were concerned, safety significance of this common mode failure was evident and decision was taken to extend the controls to all actuators of this type.

ACTION TAKEN AND RESULTS

Inside containment valve actuators were found correctly greased. Previous current motor current traces were available for all of them (20) and their reexamination led to visit three of them with preventive replacement of two worm/worm gear couples slightly damaged.

All outside containment valve actuators had grease deficiencies more or less important (89), they were replenished; previous motor current traces were available for one third of them and their examination led to visit eighteen of them with preventive replacement of two worm/worm gear couples slightly damaged. A motor current trace was performed after greasing on half of them.

Cause

Two reasons were advanced to explain grease deficiencies:

- greasing was uneasy on the outside valve actuator type,
- greasing instructions did not precise the grease quantity. Besides, aware of problems due to overgreasing of certain internal parts of the actuator, the tendency of intervenants was to reduce the grease quantity.

An assessment by IPSN of actions taken by EDF concluded in:

- a good correlation between grease deficiency and worm/worm gear couple degradation for outside valves,
- requests to perform new motor current traces during the next refueling outage for the two inside valves, degradation of which could not be explained by a grease deficiency (normal trace recording periodicity is 3 years). IPSN considered that controls performed by EDF and reexamination of motor current traces provided good insurance of valves operability during the next cycle.
FURTHER INVESTIGATIONS

Safety Authority requests were:
- to extend the verifications to all other plants,
- to review greasing instructions of different types of actuators.

Following this incident the manufacturer of the actuators has been required by EDF to perform
tests in order to better understand the behaviour of actuators in degraded conditions.

It is worth to underline that the diagnostic device SAMIR has been used on Tricastin NPP for
1991 and has provided reliable informations facilitating the safety assessment of the incident. Though
this benefit may appear insufficient, it is hoped that in the future this device will give the information of
maintenance inadequacy before failure occurs.

CONCLUSION:

The events described or mentionned in this report illustrate the extent of problems related to
M.O.V. operability. The overriding conclusion drawn from this report is undoubtly the false insurance
given by present periodical testing of M.O.V. capability to perform as intented during accidental
conditions.

As result these events have been the starting point of extensive studies and corrective actions
in order to improve this situation.

First efforts have been directed to a better understanding of M.O.V.s problems, in order to
eliminate faults and reduce uncertainties resulting from their initial sizing. Corrective actions involving a
quite large number of M.O.V.s will be implemented to solve problems revealed by operational
experience feedback. Large amount of efforts have been devoted to improve a diagnostic technique
aiming at predicting that during test conditions sufficient margin exists such that M.O.V. can operate
under accidental conditions.

Though valve problems are complex it is hoped that with these efforts combined with increase
attention and deep involvment of operating and maintenance personnel, less and less problems will
remain undetected and unsolved. Similar efforts engaged in foreign countries can only confirm this hope.
Figure 1

SAFETY INJECTION SYSTEM 1300 MWe UNITS

- **ISBP**: Low Pressure Safety Injection
- **RPE**: VENT and DRAIN SYSTEM
- **ISMP**: Medium Pressure Safety Injection
- **RAZ**: NITROGEN DISTRIBUTION
Figure 3

MOTOR-OPERATED VALVE COUPLING
COMITTEE ON THE SAFETY OF NUCLEAR INSTALLATIONS
PRINCIPAL WORKING GROUP N° 1

INTERNATIONAL ATOMIC ENERGY AGENCY
DIVISION OF NUCLEAR SAFETY

VALVE MAINTENANCE IN NPP PAKS

BY
GÁBOR NÉMETH
Head of Department

SPECIALIST MEETING

ON

MOTOR OPERATED VALVE ISSUES
IN NUCLEAR POWER PLANT

Paris, France
25 to 27 April, 1994
NPP PAKS

Dr. Enő PETZ
GENERAL MANAGER

Gábor VÁMOS
PRODUCTION DIRECTOR

Production Chief Engineer
Operation Chief Engineer
Electrical Chief Engineer
I & C Chief Engineer
Mechanical Maintenance Chief Engineering
Chemistry Department

László SzABO
SAFETY DIRECTOR

János NAGY
ECONOMIC DIRECTOR

Dr. Péter TRAMPUSS
GENERAL STAFF

György TóTH
TECHNICAL DIRECTOR
NPP PAKS
Valves Maintenance Department

Head of Department

Primary Valves Maintenance

Secondary Valves Maintenance

Regulatory Valves Maintenance in secondary circuit

Team of Preparation

Relief Safety Valves Maintenance

Actuator Maintenance

Diesel Maintenance
NUCLEAR POWER PLANT OF PAKS
LOAD FACTORS


65.6 73.4 83.1 84.4 83.8 85.7 87.7 86.6 86.4 86.4 85.6

PERCENTAGE

1994.05.03
Valves Maintenance Department
NUCLEAR POWER PLANT OF PAKS
OUTAGETIME

<table>
<thead>
<tr>
<th>YEARS</th>
<th>UNIT 1</th>
<th>UNIT 2</th>
<th>UNIT 3</th>
<th>UNIT 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>1984</td>
<td>73</td>
<td>47</td>
<td>35</td>
<td>37</td>
</tr>
<tr>
<td>1985</td>
<td>45</td>
<td>42</td>
<td>34</td>
<td>35</td>
</tr>
<tr>
<td>1986</td>
<td>46</td>
<td>34</td>
<td>35</td>
<td>30</td>
</tr>
<tr>
<td>1987</td>
<td>68 *</td>
<td>59 *</td>
<td>35</td>
<td>54 *</td>
</tr>
<tr>
<td>1988</td>
<td>33</td>
<td>33</td>
<td>54</td>
<td>30</td>
</tr>
<tr>
<td>1989</td>
<td>35</td>
<td>31</td>
<td>43</td>
<td>63 *</td>
</tr>
<tr>
<td>1990</td>
<td>32</td>
<td>35</td>
<td>45</td>
<td>36</td>
</tr>
<tr>
<td>1991</td>
<td>74 *</td>
<td>35</td>
<td>45</td>
<td>28</td>
</tr>
<tr>
<td>1992</td>
<td>46</td>
<td>76 *</td>
<td>30</td>
<td>41</td>
</tr>
<tr>
<td>1993</td>
<td>42</td>
<td>37</td>
<td>69 *</td>
<td></td>
</tr>
</tbody>
</table>

* Refueling outage

1994.03.20
Valves Maintenance Department
GKFM-AKO ILLESI
Outages
Termeléskiségek részletezése
2461 GWh
Refueling outages 85,47%
Üzemanyag átrakások

Others 1,29%
Egyéb

Forced outages 6,56%
Kényszerkiesés

Short term maintenance 6,68%
Hétvégű karbantartás
CORRECTIVE AND PREVENTIVE MAINTENANCE
NUCLEAR POWER PLANT OF PAKS

Percent

<table>
<thead>
<tr>
<th>Year</th>
<th>Corrective</th>
<th>Preventive</th>
</tr>
</thead>
<tbody>
<tr>
<td>1986</td>
<td>51</td>
<td>49</td>
</tr>
<tr>
<td>1987</td>
<td>37</td>
<td>63</td>
</tr>
<tr>
<td>1988</td>
<td>36</td>
<td>64</td>
</tr>
<tr>
<td>1989</td>
<td>33</td>
<td>67</td>
</tr>
<tr>
<td>1990</td>
<td>30</td>
<td>70</td>
</tr>
<tr>
<td>1991</td>
<td>31</td>
<td>69</td>
</tr>
<tr>
<td>1992</td>
<td>28</td>
<td>72</td>
</tr>
<tr>
<td>1993</td>
<td>25</td>
<td>75</td>
</tr>
</tbody>
</table>
I. Valve Maintenance

1. Valve Maintenance Policy
   - The safety of the NPP
   - Keep the Plant operation
   - Decreasing the costs
   - Upgrading the technical state of the valves

2. Maintenance Strategy
   - Corrective Maintenance
   - Preventive Maintenance
   - Conditions Monitoring

3. Main suppliers of the valves used by Paks NPP Ltd
   - VELAN
   - KSB
   - BABCOCK
   - BOPP & REUTHER
   - Various types of Russia

   Altogether we filed 747 sub-types of valves

4. Main types of valves
   - Safety Relief valves
   - Control valves
   - Gate valves
   - Check valves
   - Globe valves
   - Non-Return valves

5. Scheduling the maintenance
   - Cyclic programmes
   - Feedback of maintenance experiences
   - Feedback of operation history

6. Aspects of valve maintenance
   6.1. Building-up of valve categories based on operations (RCM)
      - maintenance with reactor cool down
      - maintenance with reactor discharge
      - maintenance with decreased reactor performance
      - maintenance with tripping turbine operation
      - maintenance with not affecting operation
6.2. Service pathways
- control of sealings
- control of drive mechanism of torque switches
- greasing of spindles and spindles nuts

6.3. Maintenance cycles
Valve maintenance  - every 8 years
                 - every 4 years
                 - every 3 years
                 - every 2 years
                 - every year (all safety relief valves and controls in general)

7. The graphite program
Preconditions to program execution were as below
- overall control of size of russian-made valves
- manufacturing of new spindles
- control of bolt joints
- setting of covers and gland houses
- grinding of valve seats and nitch discs

8. For each maintained valves the following data are recorded in the computerized data base
- the coordinating foreman
- personal identification codes of those who performed the work
- code of the Work Order
- cost allocation of the Work Order
- time of the work beginning and completion
- description of the work performed on the valve
- materials used from the store

9. Aspects for selecting the valves to be diagnosed
9.1. According to the conditions of repair works
    - maintenance with discharging the coolant from the reactor
    - maintenance with excluding the turbine
    - maintenance causing power loss
9.2. Frequency of actuation of the valves
9.3. Statistic of failure
9.4. Availability for maintenance
9.5. Cost emerging with the failure of the valve
NUCLEAR POWER PLANT OF PAKS
VALVES MAINTENANCE
PROGRAM OF GRAPHITE PACKING

<table>
<thead>
<tr>
<th>DATE</th>
<th>VOLUME</th>
</tr>
</thead>
<tbody>
<tr>
<td>1989</td>
<td>600</td>
</tr>
<tr>
<td>1990</td>
<td>1009</td>
</tr>
<tr>
<td>1991</td>
<td>1129</td>
</tr>
<tr>
<td>1992</td>
<td>433</td>
</tr>
<tr>
<td>1993</td>
<td>599</td>
</tr>
</tbody>
</table>
MOTOR LOAD IN CASE OF DIFFERENT TIGHTENING TORQUE OF GLAND PACKING

[Diagram showing motor load vs. time with various load values indicated.]
II. Maintenance of Valve Actuators

1. Check of the technical state of the actuators
   Mechanical maintenance is grouped into three main categories
   - periodic technical supervision
   - preventive maintenance
   - repair of failures

2. Types of valve actuators installed
   - EMG
   - DREHMO
   - AUMA
   - BERNARD
   - SIEMENS
   - REMOTE
   - Russian types

3. The Russian actuators classifying (according to the torque limitation)
   - Current relay
   - Mechanical one directional (in closing sense)
   - Mechanical bi-directional (both opening and closing sense limitation)
THE MOST CHARACTERISTIC FAILURES
OF VALVE ACTUATORS

1. The manual - motor operation switch - over is inoperable
If the switch-over fails, the pins ensuring the clutch movements bend as the result of operation, and the interlocks seize. In the corrective maintenance the pins are to be replaced, the interlocks - depending on the nature of the wear - are to be renewed or replaced.
Ratio of this type of failure: about 30%

2. The electric motor is inoperable
In the case of short circuitry or burn-out of the windings the spare motor has to be mounted.
Ratio of the failure: about 20%

3. Failing connection between the actuator and the valve
As a result of the insufficient clearance settings or adjustments there is no connection between the actuator and the valve. For the sake of exact adjustment, to ensure the proper clutch connection distances have to be checked, re-measured and set.
Ratio of the failure: about 5%.

4. The actuator is drenched, because of the Plant environment
Having the actuator dismounted and reviewed, it has to be replaced with the same type, the original one is refurbished.
Ratio of this type of failure: About 5%

5. Failure of the limit switch as a result of operational vibrations
By replacing the switch - if the actuator is not damaged - the problem is corrected, the valve functions are restored, the original switch is refurbished.
Ratio of the failure: about 5%
6. No connection between the actuator and the limit switch
   The gap is not set properly, or the lock ensuring the contact is damaged from any cause.
   The clearance in the mechanism has to be set again, the lock has to be replaced.
   Ratio of the failure: about 5%.

7. Loosening of the binding elements in operation
   Binding elements loosen from the operational vibrations. Retightening solves the problem, the valve becomes operable.
   Ratio of the failure: about 10%.

8. Leakage of lubricant from sealing surfaces
   As a result of operational temperature and of ageing the packing sleeves, the lubricant is seeping. The problem could be solved by replacing the sleeves and gaskets.
   Ratio of the failure: 5%

9. Torque setting for tight closing
   If operational environment changes, or if contamination has got into the system, the pre-set torque may prove insufficient, the valve does not provide its function, as its torque is low. By increasing (or decreasing) the torque, we set the proper value within the optimal limits, and having the functional test performed, the valve is operable again.
   Ratio of the failure: 10%

10. Valve troubles during operation
    If any of the protection systems fails (limit switch, torque limiter) or the valve mechanical part is faulty (the gate valve, control valve, valve), mechanical damage of the whole valve system is resulted.
    The damage is corrected by replacing the valve, and the damaged one is refurbished. If the corrective action is impossible during operation, it is performed during outages or the monthly repair stoppages.
    Ratio of the failure: 5%
THE MOST CHARACTERISTIC FAILURES OF VALVE ACTUATORS

- The manual - motor operation switch - over is inoperable
- The electric motor is inoperable
- Failing connection between the actuator and the valve
- The actuator is drenched, because of the Plant environment
- Failure of the limit switch as a result of operational vibrations
- No connection between the actuator and the limit switch
- Loosening of the binding elements in operation
- Leakage of lubricant from sealing surfaces
- Torque setting for tight closing
- Valve troubles during operation
<table>
<thead>
<tr>
<th>FAILURE</th>
<th>CAUSE</th>
<th>ACTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. The motor does not start by pushing the starting key</td>
<td>1. The supply circuit or the magnetic breaker is failed</td>
<td>1. Check the power supply and the magnetic breaker, repair</td>
</tr>
<tr>
<td></td>
<td>2. The switchboard is not under voltage</td>
<td>2. Put the board under voltage</td>
</tr>
<tr>
<td>2. The motor is not switched off as the closing element reaches the</td>
<td>1. The setting of the end positioning cam moved away</td>
<td>1. Stop the motor immediately and:</td>
</tr>
<tr>
<td>&quot;closed&quot; and &quot;open&quot; position</td>
<td>2. The microswitch is failed</td>
<td>1.1 Set the cam position and fix it</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1.2 Replace the microswitch</td>
</tr>
<tr>
<td>3. The gear stops during closing movement and the control lamp lights</td>
<td>1. Moving parts of the valve or that of the gear is jammed</td>
<td>1. Switch the gear into opposite direction and check the starting in</td>
</tr>
<tr>
<td>on the control board</td>
<td></td>
<td>direction where the jamming occurred.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>If stops when repeatedly started, search the wear and repair</td>
</tr>
<tr>
<td>4. In the end position of the valve, the &quot;open&quot; end &quot;closed&quot; control</td>
<td>1. The lamp are out</td>
<td>1. Replace the lamps</td>
</tr>
<tr>
<td>lamp does not light</td>
<td>2. Setting of the end position cams altered</td>
<td>2. Set and fix the cams</td>
</tr>
<tr>
<td></td>
<td>3. No voltage in the circuit</td>
<td>3. Check the control circuit and eliminate the fault.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Restore power supply</td>
</tr>
<tr>
<td>Issue</td>
<td>Solution 1</td>
<td>Solution 2</td>
</tr>
<tr>
<td>----------------------------------------------------------------------</td>
<td>-----------------------------------------------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Both &quot;closed&quot; and &quot;open&quot; lamps light simultaneously on the control board</td>
<td>Short circuity in the switch relay leads</td>
<td>Search the place of the short circuit and repair it.</td>
</tr>
<tr>
<td>6. During the move of the electric gear, the local pointer does not turn.</td>
<td>The fastening nut of the pointer is screwed off</td>
<td>Fasten the nut</td>
</tr>
<tr>
<td>7. Unacceptable leakage on the valve sealing surfaces</td>
<td>Insufficient torque</td>
<td>Set the torque clutch</td>
</tr>
<tr>
<td></td>
<td>Solid particles have got between the sealing surfaces</td>
<td>Clean through-flow cross section of the valve.</td>
</tr>
<tr>
<td>8. By manual closing or opening, the handwheel is rotating very hard or could not rotated at all</td>
<td>Moving parts are jammed in the valve or in the gear</td>
<td>Rotate the handwheel into opposite direction, check the closing or opening. If there is a seizure, investigate its cause and eliminate it.</td>
</tr>
</tbody>
</table>
SUMMARY
of repair works of pneumatic valves, 1993
1-4 Units
PRESSURE LOCKING AND THERMAL BINDING
OF GATE VALVES IN THE UNITED STATES

Dr. Earl J. Brown
Jack E. Rosenthal
Office for Analysis and Evaluation
of Operational Data
U.S. Nuclear Regulatory Commission
Washington, DC 20555

DISCUSSION

The United States Nuclear Regulatory Commission (USNRC) staff has reviewed several operating events involving pressure locking and thermal binding of gate valves. The staff recently issued NUREG-1275, Vol. 9, "Pressure Locking and Thermal Binding of Gate Valves," (March 1993). This report describes the history of motor operated valve (MOV) events, technical aspects of the phenomena, consequences on valve operability if locking or binding occur, preventive measures, and safety significance. The most significant aspects of gate valve binding as a result of these phenomena are that such binding is often synonymous with MOV failure to open and inability of the associated safety train or system to perform their safety functions. Thus, pressure locking and thermal binding represent potential common-cause failure that could render redundant trains of certain safety-related systems or multiple safety systems inoperable.

These disc binding problems have been addressed by the USNRC and the nuclear industry since 1977 (see NUREG-1275, Vol. 9). Throughout the 1980's, the industry issued a number of event reports describing safety-related gate valve failure to operate due to disc binding. These failures were attributed to either pressure locking or thermal binding. There were also multiple generic industry communications with guidance for both identifying susceptible valves and appropriate preventative and corrective measures. However, similar events have continued to occur in the 1990's.

Subsequent to issuing NUREG-1275, Vol. 9, USNRC staff visited several licensees to understand the technical issues related to generic and plant, nuclear steam supply system, system or component specific characteristics, and determine the implementation status of prior industry guidance (both identification of susceptible valves and application of preventative and corrective measures). One important observation was that multiple reviews of pressure locking and thermal binding had been performed; however, there were very few instances where valves had been modified in accordance with industry guidance to alleviate affects of pressure locking.
PHENOMENA

Although the pressure locking and thermal binding phenomena are based on well known concepts, recognizing which valves are susceptible, and when, requires a thorough knowledge of components, systems, and plant operations. The necessary features to develop gate valve pressure locking in fluid systems are fluid in the valve bonnet cavity including the volume between the discs, and a mechanism to cause the bonnet cavity fluid pressure to be greater than considered in the sizing of the valve operator for design basis accidents. The fluid may enter the bonnet cavity via mechanisms as follows: (1) during normal open and close valve cycling at whatever line pressure exists at the time, and (2) a fluid differential pressure across a disc which causes the disc to move slightly away from the seat creating a path to either increase the fluid pressure or fill the bonnet, if it had been empty, with high-pressure fluid. Gate valve pressure locking in a steam system could develop via a steam line pressure which provides a differential pressure across the disc as described in step (2) with a valve configuration that permits condensate to collect and enter the bonnet. In addition, a gate valve bonnet pressurized via steam, but without condensate, could experience pressure locking if the line pressure on the upstream and downstream sides of the discs drops significantly. An example would be a power-operated relief valve (PORV) block valve that was closed because of a leaking PORV with valve opening desired after the reactor coolant system pressure had decreased.

The mechanisms which generally cause higher than anticipated bonnet pressure are: (1) connection to a higher pressure system when isolation is only provided by check valves which easily transmit pressure even when passing leak tightness criteria, and (2) bonnet volume temperature increase which causes thermal expansion of the confined fluid with a corresponding pressure increase. The temperature increase may result from heatup during plant operation, ambient air temperature rise due to leaking components or pipe breaks (accidents), and thermal conduction and convection through connected piping during various modes of plant operation. The valve types affected by pressure locking are flex-wedge, split-wedge, and parallel slide gate valves.

Thermal binding is generally associated with a wedge gate valve that is closed while the system is hot and then allowed to cool before attempting to open the valve. The valve body and disc mechanically interfere because of the different expansion and contraction characteristics of the valve body and disc. The different thermal contraction causes the valve seat to create an interference fit with the disc. Thus, the disc is bound so tightly that reopening is either difficult or impossible until the valve is reheated. Solid wedge type gate valves appear most susceptible to thermal binding. However, there is some evidence that flex-wedge gate valves with a high temperature gradient across the discs and certain manufacturing tolerances can also exhibit binding characteristics.

DESIGN BASIS CONSIDERATIONS

The MOVs under consideration are those used to mitigate an accident. This generally involves valve disc motion, open or close, to either isolate fluid flow or permit fluid flow for injection. The isolation function occurs against either pump head pressure
and flow or water or steam system flow subsequent to a pipe rupture. The open function occurs against pump head pressure and flow. For each function, the loading effects on the disc are similar and valve operator thrust requirements have been determined by testing and a specific load thrust model using only one disc and seat surface. This situation differs from that developed during pressure locking and thermal binding of flexible-wedge, split-wedge, and parallel slide-disc gate valves.

The pressure locking and thermal binding situations of interest are those in which a valve is closed and must open for fluid injection. The loads which the operator must overcome to open the valve are caused by bonnet cavity pressure or physical interference between the discs and valve seats. This may also be in combination with pump head pressure. Thus, the load sources are different from the usually associated loadings, they act on two discs and seats rather than one disc and seat, and the load source is internal to the valve rather than external (except that contribution from the external pump head pressure).

Pressure locking or thermal binding occur as a result of the valve design characteristics (wedge and valve body configuration, flexibility, and material thermal coefficients) when subjected to specific pressures and temperatures during various modes of plant operation. It appears that neither the valve designer nor user were cognizant of the valve response to certain plant design basis conditions which could subject these valves to high pressures or temperature variations. However, irrespective of knowledge about valve behavior, determination of the pressures and temperatures should be part of the MOV design bases. Operating experience indicates this aspect was not considered at many plants. The experience also shows that pressure locking and thermal binding restraining loads can significantly exceed the design basis accident loads currently used to determine valve operator thrust requirements for some MOVs.

Figure 1 depicts a closed flexible-wedge gate valve with equal pressure shown throughout the bonnet and interconnected volume between the discs. The primary issue is the differential pressure effect across both discs on the actuator thrust required to open the valve. Current procedures determine the actuator thrust to overcome differential pressure across one disc only. Thus, in theory, an internal pressure creating a differential pressure across both discs, which was the same as that across one disc, could double the actuator thrust required to open the valve. This illustrates conceptually how pressure locking can significantly increase the thrust required to open an actuator. However, the model described does not bound the differential pressure which could be experienced. The actual bonnet pressure and disc differential pressure depends upon how the bonnet becomes pressurized (such as, a connecting high-pressure system or temperature increase caused by ambient air or connected piping), the differential pressure used to size the actuator, and the actual pressure against each disc. This could represent an actuator thrust multiple significantly greater than two. Thus, valve modification may be needed to assure valve operability.

Based on the differences in which the loads are applied, it appears that testing in conjunction with Generic Letter 89-10 does not replicate the loading conditions for pressure locking and thermal binding. Thus, analytic methods to evaluate loads should be confirmed by appropriate tests.
OPERATING EXPERIENCE AND PREVENTIVE METHODS

Several plants have experienced either pressure locking or thermal binding. These are discussed in NUREG-1275, Vol. 9. The most important message about the events in the NUREG report is that they illustrate how conditions develop or evolve to cause pressure locking and thermal binding rather than identify the most risk significant valves. Examples of safety-related gate valves involved in pressure locking events are:

- low-pressure coolant injection (LPCI) and low-pressure core spray (LPCS) system injection valves,
- core spray system valves,
- residual heat removal (RHR) system, shutdown cooling mode isolation valves,
- RHR hot leg crossover isolation valves,
- RHR containment sumps and suppression pool suction valves,
- high-pressure coolant injection (HPCI) steam admission valves,
- RHR heat exchanger outlet valves,
- emergency feedwater isolation valves.

Examples of safety-related gate valves involved in thermal binding events are:

- reactor depressurization system isolation valves,
- RHR inboard suction isolation valves,
- HPCI steam admission valves,
- PORV block valves,
- reactor coolant system letdown isolation valves,
- RHR suppression pool suction valves,
- containment isolation valves (sample line, letdown heat exchanger inlet header),
- condensate discharge valves,
- reactor feedwater pump discharge valves.

There are several preventive and corrective measures for pressure locking and thermal binding. They are discussed in NUREG-1275, Vol. 9, page 7. Each method has limitations with respect to applicability, safety, effectiveness and cost. Many methods have been used in the past and described in previous generic communications. In general, pressure locking has been alleviated through valve modification to vent the pressure while procedure changes were used to prevent thermal binding from occurring.

IDENTIFYING SUSCEPTIBLE VALVES

USNRC staff discussions with licensees were very helpful in developing an understanding of the conditions, component designs, systems, and operational sequences that impact gate valve susceptibility to pressure locking and thermal binding. This process led to the observation that certain assumptions or perceptions can result in misunderstanding situations that represent potential pressure locking or thermal binding. Three important and incorrect assumptions used to determine susceptibility are:
• Plant records do not show a maintenance or operational history of the problem.

• Leakage past an in-line check valve was assumed to be zero.

• The review for susceptibility was restricted to normal system operation only.

A typical pressurized-water reactor RHR system configuration (simplified) shown in Figure 2 illustrates certain valves that can be susceptible to pressure locking during an accident sequence. The valves include hot leg injection, RHR cross-tie to hot leg injection, RHR cross-tie to safety injection, and the containment sump isolation to the RHR pump suction for recirculation. For some valves, the valve safety function may be needed up to several hours after the accident.

SUMMARY

The USNRC staff investigations indicate that identifying valves susceptible to pressure locking or thermal binding is a complex process involving knowledge of components, systems, and plant operations. Component aspects include understanding valve operation, knowledge of valve design that can affect thrust requirements to lift the discs off the seats, and knowledge of the process used to determine actuator thrust requirements. System considerations involve knowledge of system conditions during all modes of plant operation, pump and valve alignments and operations, and knowledge of pump and valve start signals during all modes of operation including transients, accidents, and recovery actions hours after an accident. Operations aspects relate to knowledge of plant operations from startup to shutdown throughout a full fuel cycle to include systems or trains used and when, system temperature and pressure conditions in various modes, interface conditions (especially temperature and pressure) between systems or trains (or modes of a system), and valve proximity to fluid temperature change (heatup or cooldown) or valve exposure to pressure difference. An observation from discussions with licensees was that those most successful in identifying valves susceptible to pressure locking and thermal binding utilized an interdisciplinary team composed of valve experts, systems engineers, and plant operations staff.

Valves identified susceptible to these binding phenomena need effective valve modifications or appropriate procedures to assure operability. There are several preventive and corrective measures. Each method has advantages or limitations with respect to applicability, safety, effectiveness, and cost. Prioritization of valves for preventive or corrective actions may be established with probabilistic risk assessment (PRA) methods.
Pressure Locking
Flexible-Wedge Gate Valve

Figure 1
OPERATING EXPERIENCES WITH MOTOR OPERATED VALVES
IN THE FRG

K. Kotthoff

H. Liemersdorf

Gesellschaft für Anlagen- und Reaktorsicherheit (GRS) mbH,
Cologne, FRG

Joint NEA / IAEA Specialist Meeting on Motor Operated Valve Issues
Paris, France, April 25th - 27th, 1994
ABSTRACT

The paper deals with findings and lessons learned from the evaluation of safety related events from NPPs regarding MOVs. In the FRG, safety related events are reported to the state authorities since start of the first commercial NPP. Till now more than 3800 safety related events are stored in the data base for reportable events. About 20% of these events are related to valves. The presentation will give a survey of significant findings with respect to different types of MOVs such as valves with motor operators and hydraulic operators, solenoid operated valves and pilot operated valves.

For different types of MOVs the paper presents general findings on failure modes and causes observed. The second part attached to each type of MOVs provides an in-depth discussion of selected failure modes observed, root causes and actions taken. Main emphasis will be given to failure modes which have a potential for dependent failures in safety related equipment.
INTRODUCTION

The paper deals with findings and lessons learned from the evaluation of safety related events from NPPs in the FRG regarding MOVs. In the FRG, safety related events have to be reported to the state authorities. In 1975, the states and the federal government agreed on a set of federal reporting criteria which have to be applied by all utilities. This set of reporting criteria has been changed twice in the past to improve the applicability, to make the criteria more precise and to take account of the technical development in the plants.

From the start of the first commercial NPP in the FRG up to the end of 1993 there are in total 172 years of PWR operation and 92 years of BWR operation. Within this time about 3800 safety related events have been reported to the authorities. Due to the threshold of the reporting criteria, the events reported to the authorities focus on relevant findings. Minor problems, e.g. identified by scheduled maintenance well before failure of a valve are usually not contained in the data file.

On behalf of the Federal Minister of Environment (BMU), the events reported to a state authority are collected and evaluated on the federal level by GRS. This shall assure that safety related events reported to one state and the lessons to be learned are available to all other state authorities and the federal government. Thus, this task complements the efforts of the states.

About 800 of the events reported to the authorities (20%) are related to valve problems. Regarding the scope of the specialist meeting, only those events are of further interest for this presentation which deal with problems related to valves operated by an actuator.

When discussing the results presented in this paper one should keep in mind that the data file may not be homogeneous over time. Due to the technical development of the plants and the insights gained from both the events and the application of the reporting criteria the reporting criteria have been changed twice in the past as mentioned above. Thus, there are some alterations in the points of interest when reporting events. E.g. in the meantime much more emphasis is put on potential and real common cause failures than in the seventies.
2 SCOPE AND DEFINITIONS

The total number of valves installed in a NPP ranges in the order of 20,000. The main types of valves installed are globe valves and check valves followed by butterfly valves, safety valves and gate valves. This figure looks similar for PWRs and BWRs.

Since the reporting criteria are focusing on safety related equipment, only part of these valves are reflected in the reportable events. Taking not into consideration those valves which are not in the scope of the specialist meeting (manually operated valves, check valves, spring loaded valves etc.) between 5 to 10\% of the total number of valves are remaining. These are safety related valves operated by an actuator.

Subdividing the types of actuator for safety related valves, most of the valves are operated by motor actuators. The remaining types of actuators, i.e. solenoid actuators, self-medium operated actuators, hydraulic actuators, pneumatic actuators and spring actuators are only small fractions of all safety related valves. The paper will focus on the following safety related valves:

- globe valves,
- gate valves,
- control valves,
- safety valves.

The types of actuator considered are:

- motor actuators,
- self-medium operated actuators,
- solenoid actuators and
- hydraulic actuators.

When discussing operating experience with a specific component like valves it is important to define the component boundary used for the investigation. For the purpose of this presentation the following equipment has been considered in the evaluation (Fig. 1):
- Valve body and its internals,
- actuator and transmission gear / linkage,
- limit and torque switch devices,
- power supply directly associated with the valve and
- actuation and control logic devices directly associated with the valve.

Furthermore, it is necessary to define the kind of failures which have been considered and those not taken into account. In this paper failures of valves and significant defects are discussed. Minor deviations or malfunctions not related to the valve as defined above have not been taken into account, e.g.
- minor external or internal leakages not effecting proper operation of the valve,
- minor structural defects detected by the routine surveillance program which did not prevent proper operation of the valve,
- wrong valve position due to operator error, e.g. proper position not restored after maintenance or repair work,
- erroneous operation of a valve due to operator error, e.g. by an erroneous actuation signal because of an error during test and maintenance.

Taking these definitions the number of event reports to be considered reduces to about 400, i.e. 10 % of the total number of events reported to the authorities.

The screening of the event reports has been structured according to the types of valve actuators considered. The presentation of the results follows the same structure. For each type of actuator there will be a subchapter on general findings and, if relevant, on differences between different types of valves like globe and gate valves. Selected problems observed for valves with a given type of actuator and lessons learned will be discussed in a second subchapter.
3 MOTOR OPERATED VALVES

The majority of valves with actuators are motor operated valves. Motor operated valves are used in all plant systems, e.g. ECCS and RHRS, reactor auxiliary systems, feedwater and auxiliary feedwater system etc. There is a large variety of valves operated by motor actuators, which may differ by the valve type itself as well as by the technical characteristics of the design and the specific application. The most common valve types are gate valves and globe valves for equipment isolation and control valves for flow rate adjustment.

3.1 General Findings

In summary, about 250 events related to failure of motor operated valves have been identified in the data file. About 40% of these events are related to globe valves, 35% to gate valves and 20% to control valves. The remaining 5% are dealing with multi port valves.

The equipment causing valve failure is illustrated by fig. 2. About 25% of the failures are related to mechanical valve internals, another 25% to problems with limit/torque switches. Power supply, actuation logic, and actuator/transmission are contributing between 17% and 12%. I.e., the figure roughly shows a uniform distribution. Specific weak points cannot be identified.

Fig. 3 compares the equipment causing the valve failure for gate valves and globe valves. In principle, the distribution for both types of valves looks similar. But the figure may indicate that gate valves are more sensitive to failure of mechanical valve internals whereas globe valves are more sensitive to limit/torque switch problems and actuator/transmission failures.

A variety of different causes leading to failure of MOVs has been observed. Relevant causes are:

Mechanical valve internals

- Loosening or failure of internal connections, e.g. stem to disk
- binding due to thermal effects or pressure differences
- increased friction between spindle and spindle nut by e.g. inadequate lubrication or inadequate material pairings
- binding in the packing box
- foreign particles
- deposits on valve internals, e.g. boron

- Actuator/Transmission
  - mechanical problems (bearings, clutch, transmission gear)
  - short circuit of motor
  - wrong actuator type
  - increased friction by inadequate lubrication

- Limit/Torque Switches
  - failure of limit or torque switches
  - inadequate / wrong settings
  - disadjustment

- Power Supply
  - short circuit
  - loose connection or interruption of connection

- Actuation Logic
  - failure of electronic cards
  - wrong memory setting on electronic card by manual valve operation during maintenance
3.2 Significant Failure Modes

3.2.1 Re-evaluation of Gate Valve Design with Respect to Proper Operation under Full Design Pressure Difference

Gate valves are installed in many safety related systems to isolate equipment if required. For some significant gate valves there are difficulties to perform surveillance tests under the pressure difference which may prevail during demand, e.g. during a design basis accident. Thus, the proper operation of the gate valve cannot be completely verified by tests.

In the FRG, a first in-depth investigation on gate valve behaviour under full design pressure difference started more then 12 years ago after failure of a steam isolation gate valve in a conventional power plant. This investigation focused on main steam isolation gate valves in PWRs. It included full scale tests in a test facility. The test confirmed the concern about sufficient displacement forces under full design pressure difference. Two different corrective actions have been taken in German PWRs. Most of the plants replaced the main steam isolation gate valves by globe valves. In the remaining plants disk and seat of the gate valve have been redesigned to prevent excessive friction forces under full flow conditions.

In the last years, a re-evaluation of all safety related motor operated gate valves has been started. This re-evaluation is performed in three steps:

- development of modified more realistic calculation methods for the forces required to operate the valve taking into account the recent state of knowledge

- development of guidance for gate valve design especially for avoidance of excessive friction forces under full flow conditions

- development of guidance for maintenance and surveillance testing.

A more detailed discussion of this program will be given in two German presentations during the meeting.
3.2.2 MOV Failure Because of Unexpected Premature Ageing of Grease

In German BWRs reactor pressure vessels pressure limitation and pressure relief is provided by self-medium operated safety/relief valves. To provide diverse capabilities for this function all BWRs have been backfitted with additional motor operated relief valves.

By surveillance testing failure of such motor operated relief valves has been identified in two BWRs. In both events, valve failure was caused by premature ageing of the grease for the spindle and spindle nut as well as for the radial bearings of the spindle. In-depth investigation revealed the following root cause:

The motor operated relief valves had been equipped with an extensive insulation covering the valve body and part of the actuator. Thus, by heat conduction from the main steam lines via the piping and the valve body the grease could heat up to a temperature in excess of the maximum permissible temperature specified for the grease. It should be mentioned, that proper choice of grease is difficult because of partially contradicting requirements (temperature resistance, radiation resistance, long-term behaviour etc.).

Actions taken are change of grease, shorter surveillance intervals for the grease and modification of the ventilation to reduce the temperature at the valve.

3.2.3 Valve Failure Because of Erroneous Actuation of Torque Switches

To assure a high degree of leak tightness some motor operated valves are firmly moved to the seat. There are several events where such valves did not open during surveillance tests. The valves affected are bypass, suction and shutoff valves in various safety related systems.

The direct cause of the malfunction was actuator trip during unseating of the valve because the torque exceeded the torque switch setpoint. Two root causes for erroneous actuation of the torque switch have been identified: torque switch settings not properly taking into account the torque required to unseat the valve under all conditions and disadjustment of limit switches which caused increased seating torque and thus increased unseating torque.
Corrective actions were: Torque trip setpoints were checked and changed as far as necessary. For part of the valves affected the torque switch was bypassed during unseating by making the torque switch ineffective for some time or some part of the valve stroke. In addition an overall check of torque switch setpoints was performed taking into account all possible loads during valve operation.

3.2.4 Failure of Motor Operated Valves Because of Electronic Card Malfunction

In many German NPPs the ISKAMATIC B remote control system is used for actuation of valves, solenoids and breakers from the control room. Each component in the plant, which can be operated from the control room, is actuated by a specific electronic card. For safety related components the output signal of this card is connected to a second electronic card, which guarantees that signals from the ESFAS actuate the safety related component always in priority to signals from the control room.

There are some events caused by an erroneously set torque switch memory on the second electronic card. It was found out that erroneous setting of the torque switch memory can occur by moving the valve by the handwheel or unplugging the valve cable plug or the electronic card itself when control voltage is not switched off.

To prevent erroneous setting of a torque switch memory on the electronic card the hardware has been changed and the electronic card has been replaced by the modified type.

3.2.5 Failure of Isolation Gate Valve Due to Inadequate Material Pairing of Spindle and Spindle Nut

When operating an isolation gate valve in a ECCS suction line under full flow rate the valve failed. The cause of the failure was increased friction between spindle and spindle nut. Because of wear, the brass spindle nuts of the 4 valves concerned had been replaced by steel nuts during valve maintenance.
Tests under differential pressure conditions showed 50% higher torque for steel nuts to operate the valves than for brass nuts. Therefore, the steel nuts were again replaced by brass ones.

In addition, to reduce wear of brass nuts, for all valves showing a substantial amount of wear at the brass nuts slow running motors are installed.

4 SELF-MEDIUM OPERATED VALVES

The total number of self-medium operated valves installed in safety related and non-safety related plant systems is comparably small. But most of these valves are performing functions of high safety relevance.

In PWRs the pressuriser safety and pressuriser relief valves are self-medium operated. The main steam safety valves and isolation valves are self-medium operated, too. In BWRs the situation is similar. The safety/relief valves as well as the main steam isolation valves are self-medium operated valves. In addition, there are single self-medium operated valves in the non-safety related part of the main steam system in both types of reactors.

At the present, nearly all safety related self-medium operated valves are globe valves. Only one PWR uses gate valves as main steam isolation valves.

4.1 General Findings

The data file contains about 45 events related to self-medium operated valves. Nearly all events deal with globe valves. Fig. 4 illustrates the equipment of self-medium operated valves causing the failures observed. More than 60% of all failures are caused by failures of the pilot valves. Another relevant contributor are failures of mechanical valve internals.

Fig. 5 compares the distribution of failures for isolation and safety valves. For both types of valve application the overall tendency is similar. But the contribution of pilot valve failures is much more dominant for safety valves than for isolation valves.
Based on the evaluation of incidents the following relevant causes which led to failure of self-medium operated valves have been observed:

- Main valve:
  - Binding of the valve piston due to corrosion, foreign particles, deposits (boron), inadequate materials.
  - Problems with bolts, e.g. corrosion, cracks, loss of pre-stress.
  - Deformation of valve internals due to ignition of radiolysis - gas.

- Pilot valve system:
  - Inadequate design of throttles and pilot valves.
  - Binding due to thermal effects and foreign particles.
  - Wrong installation of piston rings.
  - Pressure peaks in pilot valve piping due to dynamic effects or steam condensation.

4.2 Significant Failure Modes

4.2.1 Stress Corrosion Cracking of Main Steam Isolation Valve Components (BWR)

In BWRs, two types of stress corrosion cracking of main steam isolation valve components have been reported:

- Chlorine induced stress corrosion cracking of pilot lines.

- Hydrogens induced stress corrosion cracking of internal valve bolts.

Chlorine induced stress corrosion cracking of main steam isolation valve pilot lines caused defects as well as through wall cracks. Depending on the isolation valve design and the pilot line affected a through wall crack may result in erroneous closure of a main steam isolation valve or may prevent it from closing when demanded.
Originally, the flange connections of the pilot lines had been equipped with seals/gaskets containing traces of chlorine. Later on, these seals/gaskets had been replaced by chlorine free ones. The analysis concluded long-term effects of chlorine from the original seals/gaskets to be the root cause of the events.

The actions taken are:

- Replacement of the piping affected,
- improvement of pipe lay-out to avoid water accumulation,
- corrosion test of pilot lines,
- partial preventive replacement of pilot lines.

Hydrogen induced stress corrosion cracking degraded internal main steam isolation valve bolts which connect the valve shaft to the valve bonnet. The investigation showed that the cracking was initiated by use of a molybdenum disulphide lubricant ($\text{MoS}_2$) and presence of humidity.

Corrective actions taken are:

- Cleaning of all bolt holes,
- use of new bolts manufactured of a less sensitive material and design,
- use of a lubricant free from $\text{MoS}_2$.

4.2.2 Inadvertent Fast Opening of Main Steam Isolation Valves (BWR)

Before opening a closed main steam isolation valve the pressure upstreams and downstreams has to be equalised to ensure smooth opening of the valve. There are two events reporting inadvertent fast opening of main steam isolation valves some time after erroneous closure of the valves. The events happened due to different weakness in the I & C and the particular situation prevailing which resulted in an automatic opening of the valves under a substantial pressure difference. The fast opening caused defects of some internal bolts but did not fail the safety significant function to close the valves. Under more unfavourable pressure conditions the closing function could have been impaired.
The short-term actions taken are:

- Measures to avoid inadvertent opening,
- specification of a minimum opening time combined with the requirement to check the valve internals if the minimum opening time is not met.

In addition, changes in the valve design are discussed to exclude inadvertent fast opening of main steam isolation valves on principle.

4.2.3 Malfunction of Pilot Valves Due to Motion Impairments

Several events report failure of pilot valves of self-medium operated valves in PWR due to motion impairment by internal corrosion in the pilot valves. The corrosion mechanisms observed are:

- growing-together of the corrosion coats of pilot valve seat and disk (contact corrosion)
- sticking of pilot valve seat and disk by organic compounds of lubrication materials (generation of corrosive deposits by ageing under high temperature)
- corrosion of pilot valve internal check piston and respective bushing (chromium corrosion)

Furthermore, thermal binding of pilot valve seat and disk during cool-down due to unfavourable seat design and valve material pairing has been observed.

Corrective actions contain:

- Change of material e. g. choosing different material for disk and seat, hard chromium plating for corrosion prevention,
- improved inspections with shorter intervals,
- reduction of lubricant containing oil or grease for those parts being subjected to media or high temperatures.
4.2.4 Boron Deposits on the Inner Surface of Pressuriser Safety Valves (PWR)

Inadequate pilot valve seat design resulted in small leakage’s of steam into the main valve. Throttling and temperature conditions inside the main valve caused generation of condensate from the steam. Through subsequent evaporation of this condensate the boron content precipitated forming boron deposits on the inner surface of the main valve causing corrosive attacks on piston and piston rings.

Measures to improve pilot valve tightness to reduce leakage into the main valve and to limit boron deposits were implemented.

5 SOLENOID VALVES

Solenoid valves have a widespread application in NPPs. Main areas of application are pilot valves of self-medium operated and pneumatic valves, valves in hydraulic systems, and valves in the actuation of BWR scram systems.

There are major differences in the design of solenoid operated valves depending on the specific application. Furthermore, within one area of application there may be differences, too.

5.1 General Findings

More than 80 events related to failure of solenoid valves have been retrieved from the data file. About 40% of these events were caused by failures of mechanical valve internals or the transmission between valve and solenoid (see Fig. 6). Problems with the solenoid and failures of the actuation logic caused 25% and 18% of the events respectively. Compared to these three areas, other equipment had only small contributions.

Relevant causes observed are corrosion of valve internals, internal leakage, foreign particles, wear or corrosion of the transmission between valve and solenoid, and loose connection between valve and transmission. Failure of the solenoid was mainly due to increased friction or blockage by early ageing of plastics, short circuit or interruption in the coil, and internal corrosion.

14
5.2 Significant Failure Modes

5.2.1 Binding of Solenoid Movement of Pilot Valves

A number of events deal with binding of solenoid movement of pilot valves. Various systems have been affected by this failure mode, e.g. pilot valves of safety/relief valves in BWR, main steam safety and isolation valves in PWR and pilot valves of pneumatic operated butterfly valves in the ventilation system. The event investigation identified a potential for common cause failure.

Two different types of causes have been identified:
- ageing of plastic parts
- deposits on solenoid anchor shaft.

For solenoids operated on closed-circuit current shrinking of plastic bushes of solenoid bearings and hardening of rubber cover plates has been observed. Both effects caused solenoid binding and thus failure of the pilot valve to open when de-energizing the solenoid. To solve the problem modified solenoids with a different type of bushing have been installed. The maintenance strategy for the cover plates has been improved.

Another failure mechanism of solenoids operated on closed-circuit current was caused by long-term settle down of the plastic seal between pilot valve disk and seat. This led to a small displacement of the solenoid anchor which touched the counter anchor. Due to the increased magnetic remanence the anchor did not move when de-energizing the solenoid. Therefore the pilot valve did not open. The plastic seals have been replaced by those of a material more resistant against the long-term settle down effect.

In one NPP coatings adhering to the anchor shafts of pilot valve solenoids have been observed at several pilot valves. Two of the pilot valves did not operate during surveillance testing. The investigation performed concluded high frequency vibration induced erosion of the metallic anchor shaft bushings and deposition of the eroded material on the anchor shaft as the root cause. To prevent recurrence the bushings
will be changed. The surveillance test has been improved to identify beginning of solenoid binding before the solenoid fails.

5.2.2 Incorrect Design of Fusing of the Electrical Supply of the Limit Switches of the Safety Relief Valve (SRV) Pilot Valves Against Short Circuits

The limit switches of the safety / relief valve pilot valves are not qualified for LOCA conditions because the function of these switches is not required in case of LOCA. It was discovered that the fusing of the limit switches was not separated from the fusing of the electronic control module which actuates the solenoids of the pilot valves. Therefore, the actuation of the SRVs pilot valve solenoid would fail during LOCA in case of a short circuit of the limit switches. The spring actuation of the SRVs by high steam pressure was not affected. Non-qualified limit switches are now protected against short circuits by separate fuses.

6 HYDRAULIC OPERATED VALVES

The overall number of hydraulic operated valves in German NPPs is small. Main application are the turbine and turbine bypass stop and control valves. In addition, there are only few hydraulic operated valves in other plant systems. Examples are the isolation valves in the high head safety injection turbine steam supply in some BWRs and the main steam isolation valves in one PWR.

In summary 40 events related to hydraulic operated valves have been reported. Fig. 7 associates these events to the main subequipment of hydraulic valves. More than 40% of the events have been caused by problems in the hydraulic. Mechanical valve internals contributed in the same order of magnitude (about 40 %). The remaining events are mainly related to the actuation logic (15 %).

Relevant causes observed for valve internals are rupture of valve stroke or connection between stroke and transmission, increased friction by pollution or foreign particles, and corrosion problems. Failure of the hydraulic was mainly caused by foreign particles, internal leakage and problems with solenoid operated valves in the hydraulic system. Significant problems with hydraulic operated valves have not been observed.
7 Conclusions

About 10% of the events reported to the authorities are related to failure of safety related valves operated by an actuator. Most of the failure modes observed are strongly depending on

- the particular valve design
- the application of the valve and
- the specific circumstances.

Thus, care has to be shown when trying to apply insights gained to different designs and circumstances.

Some failure modes observed clearly indicated a potential for common cause failure of safety related valves. Those failure modes are of particular interest to draw appropriate lessons.
Figure 2: Motor Operated Valves, Equipment Affected by Failure

Figure 3: Motor Operated Valves, Equipment Affected by Failure for Gate and Globe Valves
Figure 4: Self Medium Operated Valves, Equipment Affected by Failure

Figure 5: Self Medium Operated Valves, Equipment Affected by Failure for Isolation and Safety Valves
Figure 6: Solenoid Operated Valves, Equipment Affected by Failure

Figure 7: Hydraulic Operated Valves, Equipment Affected by Failure
RESOLVING AOV PROBLEMS
AT LASALLE STATION

Presented to:
Joint NEA/IAEA
Specialist Meeting
on
Motor Operated Valve Issues

Paris, France
25-27 April 1994

Prepared By:
Mark Smith of Commonwealth Edison Co.
and
Mike Murphy of Technicon Enterprises, Inc.
RESOLVING RECURRING AIR OPERATED VALVE PROBLEMS AT THE LASALLE NUCLEAR STATION

LaSalle County Nuclear Station

Commonwealth Edison’s LaSalle Station comprises two 1135 MWe boiling water reactor units. The plant is located approximately 40 miles southwest of Chicago. The units were placed into commercial operation in 1982 and 1983 and have enjoyed satisfactory operating performance since that time. Despite its operating record, LaSalle Station has encountered its share of problems with air operated valves (AOVs). The purpose of this paper is to present the current status of the AOV performance improvement program developed for the LaSalle Station by the Commonwealth Edison Company and Technicon Enterprises, Inc.

Background

LaSalle embarked on an AOV corrective program because of the increasing trend in number and severity of problems. The problems being experienced involve design, installation, operating, and maintenance considerations. The problems are due to the number of valves in operation, the important role that AOVs perform in the plant, and the complex nature of these valves. LaSalle contains approximately 2,000 AOVs that have been provided by numerous valve manufacturers and air actuator suppliers. The number and variety of valves further complicate the problem of achieving predictable and reliable valve operation on a continuing basis.

A number of utilities have instituted programs to address AOV problems. These programs are inherently sound, but typically involve a plant wide review of valves. This type of program requires a major monetary commitment, and is study oriented rather than action oriented.

The LaSalle AOV Program

In view of the magnitude and complex nature of the issues, it was concluded that AOV problems could be resolved more quickly and cost-effectively through a focused and action oriented approach. With this approach, actions are rapidly completed for a narrow, carefully selected group of valves. This focused approach also enables the overall program to progress with a controlled level of financial commitment and expenditure. The program uses a proactive, hands-on approach and integrates the investigation, recommendation and implementation activities as a continuous effort. This allows all elements of the program to be developed, demonstrated, and improved on a small-scale basis before they are expanded into use on the larger population of valves in the Station.

To ensure that all problems were addressed, all aspects of AOV maintenance were
assumed to be incorrect and were thoroughly investigated, including engineering parameters, maintenance practices, training, spare parts, valve assembly, calibration, valve data, materials, bench sets, etc. All AOV engineering parameters were researched, validated and placed in a controlled database. Procedures and work instructions were rewritten to ensure tight control over critical engineering parameters.

Correct valve operation and reliability were ensured by upgrading maintenance practices and personnel training. This entailed the evaluation, development, and demonstration of enhanced work practices and procedures and a greatly increased awareness and overall knowledge base for the entire maintenance staff.

Dynamic diagnostic testing equipment was used as a tool for improving root cause analysis, as a proof test of the enhanced procedures, work practices, and training, and to select valves that require maintenance.

The LaSalle AOV Program is designed to methodically evolve toward a state of controlled preventive and predictive maintenance for all AOVs. Prior to implementing the program, maintenance was predominately performed on a corrective basis, only after failure had occurred. The final goal of the LaSalle AOV Program is to effectively manage costs and provide reliable valve operation. To this end, the following steps were taken.

- A small select number of problem valves was chosen to prove the process.
- All critical data for these valves were reassessed, confirmed and controlled in a database.
- Valve and actuator specific maintenance procedures and work instructions were developed to precisely control critical work steps.
- Cross discipline training was conducted within the maintenance groups.
- Maintenance was conducted with engineering oversight to ensure the accuracy of procedures. Critical dimensional data were recorded.
- Post-maintenance diagnostic testing was conducted to confirm valve operation.
- Diagnostic testing was used to evaluate the performance of valves that had already been treated in the program. Test data were tracked and trended. Follow up maintenance was scheduled appropriately.
The program builds on itself in three ways 1) by adding new problem valves, 2) by incorporating valves of similar type as those already in the program, and 3) by maintaining program valves as dictated by tracking and trending data.

Growth in the program can be limited by how much can be handled by available resources. At any point during program evolution, some valves will still be in the corrective phase, some in the preventive phase and some in the predictive phase. The final goal of the program is to achieve a maintenance state in which all AOVs are maintained in a fashion that minimizes the total program cost. It is anticipated that the optimum program will address most valves in either a preventive or predictive manner. Some valves may always be maintained in a corrective fashion simply because it is the most cost effective approach.

Valve Diagnostics

The work performed on the AOV program has demonstrated that dynamic diagnostic testing can provide invaluable information regarding the performance of valves and the identification of corrective actions to improve their performance. A microprocessor based diagnostic system called the FlowScanner supplied by Fisher Controls International, Inc. is being used at LaSalle Station. With the FlowScanner, valve position, input signals, supply pressure, actuator pressure, and instrument air are measured during a single stroke and recorded. Nonstandard performance can be quickly identified without valve disassembly.

Operational diagnostic information has been obtained in this program through the use of valve performance calculations and the FlowScanner. Critical valve characteristics such as actuator spring rate and bench set, seat load, valve stroke, packing and bearing friction, and stroking speed can be presented for close scrutiny. The accuracy of the data provided by the Flowscanner and its ability to obtain reliable response pressures during dynamic valve testing has been used to set positioners and to attain reliable and predictable valve operation.

Diagnostic testing has been useful in the following situations:

- Assisting with root cause analysis,
- Determining what maintenance work needs to be performed,
- During post maintenance recovery to prove the reliability of the valve and provide a check of the accuracy of work performed,
- Performing a baseline test of AOVs for tracking and trending purposes, and
• Testing valves in-service as a predictive inspection tool to determine maintenance frequency. The frequency of diagnostic testing can also be optimized once sufficient data is accumulated.

Using diagnostic testing for these purposes supports the overall maintenance effort. Sound maintenance with good predictive or preventive programs becomes the basis for ensuring AOV performance. This maintenance oriented approach should help to avoid some of the difficulties experienced in implementing IE Bulletin 89-10 such as the over-reliance on diagnostic testing to prove motor operated valve operability.

By way of example, the diagnostic information developed on the Unit 1 Masoneilan Feedwater Heater Drain emergency valves indicated that the current settings did not provide sufficient actuator pressure to properly seat the valve. This supported a recent sizing calculation which showed that the available force was inadequate for this valve. When the actuator pressure was increased, and the improved seat loading eliminated the chronic seat leakage in these valves.

Diagnostics also indicated that the volume booster relay was responsible for erratic operation of the Heater Drain Tank emergency spill valve 1HD059A. When the relay was adjusted, the valve flutter was eliminated. Flowscan results indicated that this flutter had caused accelerated wear of the valve cage. The need for cage inspection and potential replacement at the next outage is being evaluated. In a more generic vein, this same analysis showed that inadequate air line sizing may not allow the volume of air required to assure correct valve operation. Corrective actions to remediate this situation are being developed.

Status of the Program

The results achieved to date have been significant and are being rapidly incorporated into the daily maintenance and operating activities being performed on the affected systems. The evidence obtained demonstrates an increased reliability and more stable operation of the valves selected. This provides a strong justification for continued and augmented AOV program development.

The generic areas in which recurring problems have been found and on which the corrective actions are being concentrated have been limited to the following groups:

• Incorrect Assembly,
• Improper Bench Settings,
• Improper Stem Packing Materials and Procedures,
• Incorrect Materials for Operating Environments,
• Inadequate Documentation of Changes, and
• Incorrect Preventive Maintenance Activities.
The work activities undertaken and the results achieved to date in each of these areas are provided in the following sections.

**Incorrect Assembly**

In most cases, problems with incorrect valve assembly can be attributed to the lack of sufficient detail being provided to the mechanic in the plant. This can range from essentially no specific direction being given in the work package to lengthy but confusing or conflicting instructions. To correct this, work packages were prepared with explicit valve diagrams, detailed work instruction, evaluation, and post-maintenance testing. The work instructions were prepared using a cross-disciplined approach considering mechanical, electrical, and instrument department concerns.

To enhance the effect of improved work instructions, the selected scope of AOV repairs were grouped by type (manufacturer, model) and then prioritized for repair sequence. Consistent work crews were selected to form a repair team for the duration of the AOV repair scope. A given team was assigned to work on valves of similar type so that the team gained in experience and productivity as they progressed through the AOV outage scope.

The lessons learned from a series of stem failures in the Heater Drain Tank level control valves (HD045) provide a premier example of the incorrect assembly problem. The stem failures in these valves were traced to a fatigue failure at the root of the stem threads where they screw into the valve plug. This was corrected through the use of a plug seal ring to stabilize the plug position when the valve is open. In a number of cases it was found that the seal ring was installed upside down on the plug due to the lack of specific instructions in the procedure. This was resolved through the use of detailed steps in the procedure providing specific directions covering the installation of the seal. The procedure revisions also included detailed directions for the critical operation of pinning the stem to the valve plug including the establishment of check points to insure that the pinning is properly performed. A large number of additions were made to the procedure to supplement or clarify the instructions provided in the original version.

The procedure revisions use detailed sketches to highlight areas which have proven to be difficult to assemble or have provided repeated areas of concern related to proper assembly. The procedures also now include a detailed method for coupling actuators to the valve stems which provides for increased coupling accuracy using fewer measurements and removes a large potential for error in this important operation. The increased level of detail provided in the procedures results in a more uniform performance of maintenance operations considering the variations which exist in the experience and skill levels of mechanics assigned to this work. Also, since many of the contract personnel currently being used for balance of plant work have not been exposed to specific AOV training, the added detail provided in the
maintenance procedures has proven to be invaluable.

The value of these simplified procedures was recently demonstrated on Unit 2 where the normal Main Steam Reheat drain valves were lined up and reassembled without problems using these instructions. The excellent results obtained during the subsequent diagnostic tests are attributed to the use of these simplified procedures. By contrast, expanded procedures were not available for the normal Heater Drain valves, and they were reworked only with the use of upgraded packing and diaphragms. Subsequent testing did not demonstrate any corresponding improvement in their performance, but rather indicated the presence of problems internal to the valves which required re-entry to the valves to correct.

Improper Bench Settings / Air Regulators

The correctness of the actuator bench settings used on the Heater Drain system AOVs and the adequacy of the resulting seat loadings was a matter of deep concern. This concern was heightened by the known instances of seat leakage problems on a number of the Heater Drain emergency dump valves. A significant effort was expended to verify that the existing bench settings provided adequate seat loading. A number of valves were found to be deficient in this area. Many of the currently employed bench settings were established at the time of original plant construction, while the ability to account for the forces developed within the valves has advanced considerably since that time. Many factors which were not considered then are routine today and have affected the operation of these valves.

The currently available calculation methods provide a means for identifying bench set problems and for developing corrective measures to assure proper valve operation and seat loading. Bench set and actuator sizing for the Fisher and Masonelian valves used in Heater Drain system were recalculated. These calculations will be performed for other valves entering the program to verify the adequacy of their current bench settings or to identify changes required to achieve proper valve operation.

Valve Stem Packing

The evaluations performed to date have identified serious problems with packing materials being utilized, the arrangement of the packing in the glands, and the procedures used for tightening the packing. These problems have resulted in excessive stem forces which have adversely impacted valve performance as well as inadequate packing compression which has caused packing leaks. A standard packing arrangement has been developed which employs five rings of graphite packing and utilizes lantern rings and/or carbon spacers to axially fill the gland. This results in a uniform, controllable packing load together with an improved degree of support and guidance for the stem. The procedure also defines the method for loading the packing in spring-loaded designs to assure that the packing is properly
compressed at the outset and can be kept that way in the face of packing material shrinkage during extended operating periods. The packing materials to be used are specifically identified and controlled. These changes have led to improved valve performance and reduced maintenance requirements.

Material / Environmental Degradation

Material failures were encountered in the elastomeric materials used in the Heater Drain System valves. The high ambient temperatures in the heater bay and especially in the area adjacent to the condenser wall where the Heater Drain emergency valves are located, resulted in a substantial number of diaphragm and O-ring failures. The research conducted as part of this program has led to the conclusion that the currently used Nitrile/Buna N diaphragms should be replaced with silicone rubber/Viton to resist the aging effects of the high ambient temperatures. The comparable research on the O-rings has not as yet identified a desirable substitute. The materials investigated as potential O-ring substitutes do show an improved temperature capability but do not display an adequate wear resistance to frictional loads. An evaluation of the data provided by the manufacturer indicates that the increased degradation in wear rate offsets the gain in sensitivity to high temperatures. As a result, it has been decided that continued use of the existing O-rings is best where wear resistance is important. Viton O-rings are now being used as replacements for static conditions where high temperature is a problem.

The life-cycling capability of the elastomers being used in the air regulators and the actuator diaphragms represents another area of concern for the Heater Drain system valves. The effect of the high temperatures in the heater bay has led to premature cyclical failures in the past. The lack of an abrasion resistant O-ring material suitable for high temperature use may lead to a requirement to replace positioner O-rings every cycle. The on-going research will be continued to develop the most effective resolution of this problem.

Documentation of Changes

The research work conducted to develop or confirm valve data has led to questionable and conflicting information in many cases. It was found in one case that when valve internals were modified and the required air settings were changed, this data was given to the instrument mechanics, but the follow-up action to document the changes on the instrument data sheets was not completed. Thus when the data sheets were used to develop maintenance information, the information displayed applied to the old internals and was invalid for the existing valve condition.

In another case, changes were made to an actuator spring on one of the two LaSalle Units. On the unit for which this was performed, the change was indicated on the drawing. However, the calculations to verify the bench setting were not performed or
documented. The change was thought to have been made on the other unit but there was no documentation to support this. It was only remembered by some of the old timers and handed down by "word of mouth." The net result was that the bench settings applied to the old springs and would not yield the correct seat loading.

An additional example arose when an increase in air regulator pressure to increase seat loading was made for a number of valves. This information was sent by letter to the concerned maintenance personnel, but was never incorporated into the data sheets. This created significant confusion as to what was correct, the data sheets or the letter. Additionally, as time progressed, familiarity with the letter waned so when it surfaced again many people were surprised and confused by the information. The tendency to have knowledge vested in individuals and have that knowledge transfer with the individual when he moves on or out creates a huge potential for damage.

These represent only a few of the examples that have been uncovered in this program. This tendency to make changes without full documentation is evident, and its effects are quite noticeable. This, of course, is a generic issue, and its resolution is being pursued on that basis.

**Personnel Involvement**

Involvement from operations, engineering and maintenance personnel is essential for a successful program. Knowledge must be integrated from many sources such as systems performance, system interaction, operating procedures, maintenance practices, preventive/predictive maintenance, AOV design, testing, calibration, materials, and diagnostic data evaluation.

The accomplishments achieved in the LaSalle AOV improvement program have been enhanced by the dedicated performance of the involved LaSalle and Technicon personnel. The results can also be attributed to the blend of design, maintenance and operations oriented qualifications of the personnel who have been responsible for the formulation and implementation of the work activities involved in the program.

**Summary of Results**

A number of the most significant and recurring problems in the Feedwater Heater Drain system AOVs have been identified. Root cause evaluations and the development of corrective actions have been completed for a number of these problems. Results of the program to date include improved heater system control, smoother start up, no valve rework to correct inadequate repair, and most notably a drastic reduction in heat loss into the condenser.

The corrective actions are being developed and applied to the valves in an integrated and essentially simultaneous manner. This has resulted in the early achievement of
substantial benefits related to valve performance improvements, potentially extended operating times, and a reduction in corrective maintenance activities and cost. A significant portion of the work completed is associated with problems and resolutions that are common to a large number of AOVs (and other valve types as well). These tools and improvements will be used in support of the AOV performance improvement program as it is expanded to include additional valves in other systems.

Using a somewhat broader perspective, the work completed has demonstrated the following points.

- The use of the focused approach adopted for this program has been shown to be very effective. It has allowed specific work practices to be developed, demonstrated and improved on a small scale basis before they are expanded to use on a larger population of AOVs. The work to date has been performed in a very cost-effective and cost-manageable fashion.

- Maintenance activities are at the core of a good valve improvement program. The development of enhanced procedures and detailed work packages is essential to properly control maintenance activities. The increased awareness and improved understanding of valve problems and their resolution creates an environment for success.

- Diagnostics have been demonstrated to be very useful in selectively identifying problem valves, in supporting root cause assessments and in demonstrating the effectiveness of improved maintenance activities. However, without effective improvements, the diagnostics will do nothing more than confirm the inadequacy of current maintenance practices.

- Valve data must be compiled into a single controlled and dependable database that can be accessed by any department but controlled by only one. All changes must be documented in the database for future reference.

A significant benefit was achieved with this program, namely a marked improvement in the performance of the overall Heater Drain system. This resulted from the correct operation of the heaters' normal and emergency (dump) valves. A significant reduction in required condensate booster flow at full power has been documented. Required flow has been reduced to 20,000 gallons per minute (gpm), down from 21,500 gpm. This reduced flow will yield significant saving in the condensate polishing system.

Prior to the AOV improvement program, investigations into the erratic operation of the Heater Drain system were focused on system design problems. Given correct AOV
operation, system design problems, if they truly exist, will no longer be masked by poorly performing valves. This result also emphasizes the global nature of the AOV improvement program and the fact that root cause analysis of system problems can be more effectively evaluated once all individual system components are performing as designed.

Future Activities

An assessment will be made of the results achieved in each phase of the AOV performance improvement program to assess the costs expended and the benefits gained. If the results warrant, the program will be extended to additional valves. The performance of subsequent phases will be greatly enhanced and eased by the application of the tools developed and the lessons learned from the first phase of work. The performance of the valves involved in the program will be monitored on a continuing basis to insure that the corrective actions taken continue to be effective or indicate that additional remediating actions are required. Diagnostic testing will be used as a predictive tool to distinguish between valves that require maintenance from those that continue to perform as required.
NUCLEAR REGULATORY AUTHORITY
OF SLOVAK REPUBLIC

MOTOR OPERATED VALVES
main findings in Slovak NPPs

A specialist meeting

Motor operated valve issues

Paris, France
April 25th-27th, 1994

Prepared by: Miroslav LIPÁR
Head, department of
Nuclear Safety Evaluation
1. Slovak Nuclear Power Plants

1.1. In operation

1.1.1. Jaslovské Bohunice V-1 WWER 440 V-230
   two units, installed capacity 2 x 440 MW
   Commissioning 1978, 1980 respectively

1.1.2. Jaslovské Bohunice V-2 WWER 440 V-213
   two units, installed capacity 2 x 440 MW
   Commissioning 1984, 1985 respectively

1.2. Under construction

1.2.1. Mochovce WWER 440 V-213
   four units, installed capacity 4 x 440 MW

1.3. Under decommissioning
   Jaslovské Bohunice A-1 KS-150 HWGCR
   one unit, in operation since 1972 till 1977

Share of NPPs production in Slovakia ~ 55%
2. Number of Operational Events Reported and number of Event involved valve problems

2.1.

V - 1  
NPP

<table>
<thead>
<tr>
<th>Year</th>
<th>V - 1</th>
<th>NPP</th>
</tr>
</thead>
<tbody>
<tr>
<td>1992</td>
<td>78</td>
<td>93</td>
</tr>
<tr>
<td>1993</td>
<td>9</td>
<td>11</td>
</tr>
</tbody>
</table>

Number of Operational Events Reported

Number of Events involved valve problems

2.2.

V - 2  
NPP

<table>
<thead>
<tr>
<th>Year</th>
<th>V - 2</th>
<th>NPP</th>
</tr>
</thead>
<tbody>
<tr>
<td>1992</td>
<td>58</td>
<td>66</td>
</tr>
<tr>
<td>1993</td>
<td>9</td>
<td>6</td>
</tr>
</tbody>
</table>

Number of Operational Events Reported

Number of Events involved valve problems

143
3. List of Events Involved Valve Problems

3.1. V-1 NPP 1992

3.1.1. Inleak of pressurizer safety valve

3.1.2. Inleak of pressurizer injection valve

*Event describe:*
At full power, the operation of two extra electrical heaters were necessary to maintain pressurizer parameters. During a visual inspection in the confinement a broken thread was found at an injection valve into pressurizer which resulted in the inleak of the valve into pressurizer. The root cause of the leak was an inadequate lubrication of the valve shaft due to its inaccessibility during the operation in severe working environment (high temperature).

*Correction action:*
Replaced of control valves and new lubrication program.

3.1.3. Leakage from pressurizer safety valve flange

3.1.4. Failure of gate valves to close
(interconnection of main steam headers between unit 1 and unit 2, limit switch and brake failure)

3.1.5. Failure of control valve to close against high differential pressure
(control valve of steam generator level)

3.1.6. Failure of control valve to operate
(control valve of steam generator level, fall out of stop pin from operating device at control desk)

3.1.7. Inleak of pressurizer safety valve

3.1.8. Failure of gate valve to fully open
(outlet of spray cooler)

3.1.9. Turbine bypass valves opening
3.2. V-1 NPP 1993

3.2.1. Failure of gate valve to operate
(incorrectly connected bus to valve breaker during reconstruction)

3.2.2. Failure of gate valve to open (auxiliary feedwater system to SG, limit switch failure)

3.2.3. Failure of turbine control valve, jump of turbine load from 87 to 170 MW
(damage of mechanical part of hydraulic actuator, by electrical erosion)

3.2.4. Failure of control valves to control
(control valves of steam generators level)
3 times

Event describe:
There were installed new control valves and actuators of SGs level, during the 93 outage. After startup of the unit there were unstabilled control of SGs level. Control valves were in self-closing conditions (without electrical motor) because of high pressure to the valve spindle and the valve plug. The master controller should keep nominal SG level with correction of steam and feedwater flow, so after flow reduction, there was order to open the control valve again.
If operator changed mode of operation to manual, the control valve started to close. This situation was unsuitable for safety operation, operators and also from reliability and life time of power, control relay, electrical motor point of view.

Correction action:
After team inspection of Nuclear Regulatory Authority, decision was issued to replaced electrical actuators with magnetic brake. All actuators were replaced (for unit 1 and 2) and control of SGs level is stable now.
3.2.5. Failure of gate valves of spray system to fully close
(real spray of confinement)
2 times

Event describe:
There was a surveillance test of confinement spray system at unit 2. After
startup of pump to recirculation line some parameters were changed:
- moisture inside confinement
- water inside confinement
(leak before break diagnostic systems)
After that operator stopped the pump.
During a visual inspection, two parallel valves were found not fully
closed.
(70 round of the manual wheel, or 15 mm by spindle).
The root cause of the event was change of the limit switches of the
discharge valves from torque switch setting to position limit switch
setting.

Correction action:
- to change limit switches to torque switch setting
- visual inspection of confinement
- new flowmeter of spray water.

3.2.6. Oil leakage of quick acting valves - main steam line
(failure of O-ring during warranty)
2 times

3.2.7. Failure of gate valve to open
(common discharge valve of primary circuit charging pumps -
- mechanical problem of actuator, fall out of shaft feather)
3.3. V-2 NPP 1992

3.3.1. Failure of turbine control valve
  (damage of mechanical part of hydraulic actuator, by electrical; erosion)

3.3.2. Failure of control valve to close against high differential pressure

3.3.3. Inleak of gate valve, quick acting valve and control valve of feedwater to SG (40 t/h)

3.3.4. Failure of gate valves - extract versus auxiliary steam from low pressure heater (limit switch failure)

3.3.5. Inleak of pressurizer safety valve
  3 times

3.3.6. Inleak of turbine control valve (failure of O-ring)

3.3.7. Failure of gate valves to close (interconnection of main steam headers between V-1 and V-2, limit switch and brake failure)
3.4. V-2 NPP 1993

3.4.1. Leakage of control valve body by erosion
        (condensate from high pressure heater to feedwater tank)

3.4.2. Inleak of pressurizer safety valve
        2 times

3.4.3. Failure of Solenoid valve of pressurizer valve to operate
        (limit switch failure)

3.4.4. Failure of gate valve of spray system for cable compartment
        (real spray of cable compartment, limit switch failure)

3.4.5. Leakage of control valve body by erosion
        (minimum flow of condensate pumps)
4. Conclusions

4.1. Distribution of MOV failures:
- electrical parts, limit switches
- design, construction
- mechanical parts
- working conditions.

4.2. Correction actions of Nuclear Regulatory Authority:
- to make selection of safety related MOVs
- to judge capability and design
- QA of new MOVs
- to improve program of regular inspections, tests, including limit switches and control systems
- operational experience feedback.
MOTOR OPERATED VALVE PROGRAM:
LESSONS LEARNED

Program to comply with
USNRC GENERIC LETTER 89-10 RECOMMENDATIONS

BILL R. BLACK, P.E.
Senior Engineer
MOV Project Engineer

TEXAS UTILITIES ELECTRIC
COMANCHE PEAK STEAM ELECTRIC STATION
Mail Zone E15, P.O. Box 1002, Glen Rose, Texas 76043
817-897-6477 (FAX: 817-897-6777)
ABSTRACT

The Texas Utilities Electric Company (TU Electric) motor operated valve (MOV) program for implementing the recommendations of Generic Letter 89-10 has typically included the following: refurbishing each actuator, verifying each actuator's as-built configuration, testing each actuator's motor on a dynamometer, testing each actuator's torque spring pack (which is used to control the torque developed), testing each fully refurbished and reassembled actuator on a torque test stand, and testing as many MOVs as practicable both without fluid flow through the valve and with the maximum test conditions reasonably achievable ("Static" and "DP" conditions, respectively). Test data is acquired at 1000 samples per second for stem thrust, stem torque, stem position, actuator compensator spring pack deflection, actuator torque spring pack deflection, motor current, motor voltage, motor three phase power, valve upstream pressure, and valve downstream pressure, wherever practicable. With this and other information, the following has been accomplished:

* Equations used to predict stem thrust and stem torque requirements to close and open rising stem valves under both Static and DP conditions have been verified for Comanche Peak gate and globe valves.

* Motor and gear train selection methods specified by the actuator manufacturer generally (but not always) underestimate motor capability; test data analysis has quantified "Actuator Performance Factors" which allow credit to be taken for the greater capability the motor typically has for delivering torque to the worm gear.

* Actuator output torque at motor stall or at torque switch trip, if developed on a test stand which does not apply a thrust load to the actuator, is typically greater than the torque applied to a threaded valve stem (which does apply a thrust load to the actuator); this "Stem Thrust Effect" has been accounted for in the methods used to determine appropriate torque switch settings, actuator gear ratios, and motor size.

* Differences between Static and DP condition stem loads at close torque switch or close limit switch trip due to the "Rate of Loading" effect have been justified for each group of similar MOVs; this permits the use of Static testing to verify motor capability sufficiency and switch setting adequacy to ensure the MOV will perform its design basis functions under maximum design basis conditions.
NOMENCLATURE

Disk Position

Effect a gate valve’s disk may stop a small distance further out of the valve seat under DP conditions than it does under Static conditions. For limit switch controlled closure MOVs this results in greater stem thrusts and greater stem torques being developed at and after close limit switch trip under DP conditions than under Static conditions. For torque switch controlled MOVs, this difference in final disk position does not affect the torques or thrusts developed.

DP condition with differential pressure across and fluid flow through the valve body.

DP Test stroke test of the MOV under DP conditions.

Performance

Factor the average value by which AC motor capability to deliver torque to the actuator worm gear (with power supplied at 80% of the motor nameplate voltage) exceeds the value predicted using the standard industry method: see equation (2).

Rate of Loading

Effect a given torque spring pack deflection value corresponds to greater stem thrust under Static conditions than it does under DP conditions. Sometimes referred to as “Load Sensitive Behavior.” Thus, for a torque switch controlled MOV, the stem thrust at torque switch trip is greater under Static conditions than under DP conditions. For a limit switch controlled MOV where the same thrust is developed under both Static and DP conditions, the stem torque at limit switch trip is greater under DP conditions than under Static conditions.

Static condition without differential pressure across and fluid flow through the valve body.

Static Test stroke test of the MOV under Static conditions.
Stem Thrust

Effect: the reduction of actuator output torque capability of the motor, the reduction of actuator output torque at torque switch trip, and (more generally) the reduction of actuator output torque for a given magnitude of SPF, resulting from a thrust load imparted by the valve stem to the actuator drive sleeve via the threaded stem nut.

Torque Spring

Pack: the actuator spring pack which deflects in direct proportion to the torque output of the actuator.

Acronyms

Appendix B provides a list of acronyms used in this paper.
BACKGROUND

In June 1989 the NRC issued Generic Letter (GL) 89-10 with a recommended date of around June 1994 for completion of design basis reviews, testing where practicable at or near design basis conditions, analysis of test data, incorporation of analysis results into the calculations and programs for appropriately setting control switches, implementation of appropriate switch settings, and establishing a program to maintain correct switch settings for the remainder of the plant operating life. This paper summarizes some of what TU Electric has learned regarding MOV performance characteristics, and the context in which TU Electric data analysis has been performed.

To verify actuator configuration and to ensure test data was obtained for actuators in "as good as new" condition, all MOV actuators were refurbished prior to baseline testing. By means of spring pack, motor, actuator, and MOV tests, the following relationships were typically determined to assist verifying initial engineering assumptions:

<table>
<thead>
<tr>
<th>Torque switch setting versus:</th>
<th>torque spring pack deflection (SPD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque spring pack force (SPF) versus:</td>
<td>torque spring pack deflection (SPD)</td>
</tr>
<tr>
<td></td>
<td>actuator output torque (AOTQw) on torque test stand</td>
</tr>
<tr>
<td></td>
<td>actuator output torque (AOTQ) in situ</td>
</tr>
<tr>
<td>Motor three phase power &amp; voltage verses:</td>
<td>motor shaft torque &amp; speed</td>
</tr>
<tr>
<td></td>
<td>actuator output torque (AOTQw) on torque test stand</td>
</tr>
<tr>
<td></td>
<td>actuator output torque (AOTQ) in situ</td>
</tr>
<tr>
<td>Stem thrust versus:</td>
<td>compensator spring pack deflection</td>
</tr>
<tr>
<td></td>
<td>stem torque (AOTQ) under Static conditions</td>
</tr>
<tr>
<td></td>
<td>stem torque (AOTQ) under DP conditions</td>
</tr>
<tr>
<td></td>
<td>differential &amp; upstream pressure</td>
</tr>
<tr>
<td></td>
<td>stem travel under Static &amp; DP conditions</td>
</tr>
</tbody>
</table>
In the Summer of 1993, test data was analyzed for groups of similar actuators and similar valves to verify and adjust the initial engineering calculation assumptions. An early assessment of the impact of the test data analysis indicated that if for a group of similar MOVs the worst case values of running thrusts, valve factors, "rate of loading" effects, stem factors, and motor capability to deliver torque to the stem nut were all assumed to occur simultaneously for each MOV in the group, many groups of MOVs which had during testing demonstrated adequate capability to fulfill their design basis functions could not be shown by engineering calculation to be capable of fulfilling their design basis functions.

Thus, further analysis was performed to reduce the amount of conservatism retained in the calculation. The most significant amount of effort was expended in (1) performing a statistical analysis by which average and associated uncertainty values for each of these MOV performance parameters were determined for use in the engineering calculation, (2) revising the engineering calculation to use these results, and (3) revising the test procedures to use the revised calculation's results.

Laboratory tests have shown that the magnitude of the "rate of loading" effect for a given valve can be affected by adjusting the "running thrusts". It is therefore important to note that TU Electric Static and DP tests were conducted with normal packing loads, and with DP test conditions typically at or near the design basis DP conditions. For groups of similar MOVs, TU Electric "rate of loading" effects may therefore be considered statistically "independent" from running thrusts. This means that those MOVs which were not DP tested are most likely to have a "rate of loading" effect equal in magnitude to the average value determined for those MOVs which were tested under both Static and DP conditions.

Similarly, valve factors and stem factors were determined from test data obtained at conditions at (or near) which the MOVs are expected to operate in the future. The analysis results obtained for groups of similar MOVs for running thrusts, valve factors, "rate of loading" effects, stem factors, and motor capabilities may therefore be considered independent from each other. A change in one parameter for one MOV is unlikely to appreciably change the average or the range of that parameter or any other parameter determined for the group of similar MOVs.

Statistical methods have been applied to the test data for each MOV performance parameter to determine an uncertainty by which the average value of each parameter may be increased or decreased to obtain upper and lower bounds which envelop the test data. Because the average
and associated uncertainty of each of these MOV performance parameters are independent from the average and associated uncertainty of each other parameter (in the context discussed above), the various uncertainties may be combined by the "Square Root of the Sum of the Squares" (SRSS) technique in engineering calculations and in the test procedures. TU Electric typically determined uncertainties as the greater of (a) two sample standard deviations of the test data, or (b) the difference between the average and the greatest data point. The uncertainty was then typically divided by the average value to obtain the uncertainty as a percentage of the average value.

It is very important to keep in mind that MOV performance parameter uncertainties calculated in this manner reflect both data scatter and measurement error. Averages tend to "average out" the uncertainties. By not "correcting" the test data prior to determining the averages and uncertainties, the engineering calculation reflects the true magnitudes of the uncertainties in the baseline test program’s measurements. For example, high measurement uncertainty due to low load magnitudes contribute to the high uncertainty of the valve factor determined for valve group GT12 in Table 8c. The average valve factor for this group is within typical ranges, but the high uncertainty results in an unusually high upper bound value for the "worst case" valve factor.

Comparison of actual test data with the results of SRSS combinations of uncertainties has confirmed the adequacy of this technique to provide a high degree of confidence that TU Electric's Comanche Peak MOVs will operate properly under their maximum design basis condition. The worst case values of all of the parameters need not be assumed to occur simultaneously.

SUMMARY OF TEST RESULTS

Due to TU Electric MOV maintenance practices, and test data collection and analysis methods at the Comanche Peak nuclear power plant, the observations presented herein may be unique to Comanche Peak. These observations need not be assumed by others without confirmation of these effects for their MOVs, and their data collection and analysis methods. While the author has attempted to ensure the data and discussions herein are correct and of sufficient detail, it is inevitable that some aspects will not have been fully explained to the readers satisfaction. The author may be contacted for further information.

Summary discussions of test data and analysis results follow. Appendix A provides a more detailed discussion of most of these summaries.
Actuator Motor Capabilities

TU Electric has defined actuator AC motor "design capability" as being 64% of the motor torque rating when the motor is supplied with power at 80% of the motor's nameplate voltage. Dynamometer testing in which a braking torque is gradually applied to the rotating motor shaft over a period between 10 to 40 seconds is an easy and effective way to verify that a motor is not below this design capability. It is the author's opinion that MOV test programs should include some means to verify that the installed motors will produce the torques assumed in engineering calculations, and to justify the use of motors which may not.

In general, the actuator motors are able to produce their design capability. TU Electric has found a few random instances in which motors did not achieve this amount of torque prior to stalling. One set of nine identical motors removed from spare actuators in the warehouse all failed to produce their design capability torque.

Actuator Performance Factors

IF (a) the actuator motor is supplied with power at 80% of the motor nameplate voltage, AND (b) the actuator output torque is gradually increased over several seconds, AND (c) the actuator drive sleeve is not subjected to a thrust load by the stem, THEN the average actuator output torque at motor stall is well above the value predicted by the "standard" industry methodology. Test data shows that the "standard" industry methodology is conservative (underestimates the motor capability to produce actuator output torque) for all actuator configurations tested by TU Electric except the Size 00 actuator with the 15 ft.lb 3400 rpm motor and a worm set gear ratio of 45:1. (See Tables 1 and 2, and "Appendix A")

Actuator Effective Moment Arms

TU Electric has determined from test data on torque test stands the average "effective actuator moment arm" length by which a torque spring pack force value can be multiplied to estimate the actuator output torque, and an uncertainty within which the actual output torque is likely to be. TU Electric has done the same using test data from in situ Static and DP testing of MOVs, where the torque is measured by strain gages installed on the valve stem. The average effective moment arm length determined using in situ test data
indicates a thrust load on the actuator drive sleeve causes a loss of torque within the actuator that is not observed during torque stand testing. (See Table 3, and "Appendix A")

*Actuator Stem Thrust Effects
TU Electric has determined for several actuator configurations the average reduction of actuator output torque caused by stem thrust loads of magnitudes required for properly operating Comanche Peak MOVs. Uncertainties associated with these reduction factors have also been determined. (See Tables 4 and 5, and "Appendix A")

*Net Effect of Actuator Performance Factors and Stem Thrust Effects
In general, the use of the actuator manufacturer's specified "Pullout" efficiency instead of the "Running" efficiency is sufficient to bound the worst anticipated combined effect of the Performance Factor and the Stem Thrust Effect. TU Electric test data analysis indicates that the use of the "Pullout" efficiency is not in all cases sufficient. (See Table 6, and "Appendix A")

*Running Thrusts
Running thrusts for Comanche Peak MOVs with stem diameters less than 1.25 inches are generally bounded by a load of 1200 lb/inch of stem diameter. For larger stem diameters, the load per inch of stem diameter generally increases with increasing stem diameter up to 2600 lb/inch for a three inch diameter stem. (See Table 7, and "Appendix A")

*Valve Factors for Westinghouse and Borg-Warner Gate Valves
TU Electric data analysis has determined average closing stroke valve factors as low as 0.23 and as high as 0.56 for Westinghouse-manufactured gate valves under pumped flow conditions. The Westinghouse-specified valve factors for these MOVs are greater than the average test results. However, the data scatter indicates there is a slight possibility for some valves in some groups of similar valves to have valve factors greater than specified by Westinghouse. The PORV block valves tested under "blowdown" conditions had valve factors of 0.67, which exceeds the 0.56 value specified by Westinghouse. The valve factors determined for Westinghouse valves are based on the vendor-specified inside diameter of the valve body seat ring plus 1/16 inch.

TU Electric data analysis has determined average closing stroke valve factors as low as 0.24 and as high as 0.50 for Borg-Warner gate valves under pumped flow conditions. The
data scatter indicates there is a possibility for some valves in some groups of similar valves to have valve factors as high as 0.63. The valve factors determined for Borg-Warner valves are based on the average of the inside and outside valve seating surface design diameters specified by the manufacturer.

TU Electric test data analysis has determined both closing and opening stroke valve factors which are demonstrated to be sufficient for predicting the stem thrusts required to close and to open Comanche Peak MOVs under their most severe design basis conditions. (See Table 8, and "Appendix A")

*Repeatability of Stem Thrust at Close Limit Switch Trip

For MOVs with compensator spring packs which compress as they react stem thrust, the close limit switch may be set to continue rotating the stem nut until the disk is fully seated under both Static and DP conditions, and until any additional wedging thrust necessary to produce a seal is developed. If the limit switch is set in this manner, the disk and the stem will stop traveling at essentially the same position under both Static and DP conditions. Since the same number of stem nut rotations ensures the same relative travel between the stem nut and the stem, the stem nut must compress the compensator spring pack essentially the same amount each stroke. Since the compensator spring pack preload and stiffness do not change from stroke to stroke, essentially the same stem thrust is developed each stroke at close limit switch trip.

TU Electric analysis of 145 data points for 35 limit switch controlled closure MOVs has demonstrated that the stem thrust at close limit switch trip is generally well within +/- 3% for those MOVs which have close limit switches set to terminate the closing stroke after the valve disk has reached the valve seat under both Static and DP conditions:

Average variance from perfect repeatability = 0.01%
Maximum variance from perfect repeatability = 2.69%
Average plus 3 standard deviations of the variances from perfect repeatability = 2.1%

For reasons not yet determined, one group of Borg-Warner gate valves and the Westinghouse PORV block valves produce greater thrust at close limit switch trip under DP conditions than under Static conditions. The disk appears to terminate the closing stroke in
a position slightly further out of the valve seat under DP conditions than under Static conditions. This "Disk Position Effect" has been included in the "Rate of Loading" analysis performed for limit switch controlled MOVs.

**"Rate of Loading" Effects for Torque Switch Controlled Closure MOVs**

The average "Rate of Loading" effect for torque switch controlled closure gate MOVs is a 4% greater stem thrust under Static conditions than under DP conditions. A worst case magnitude for this effect is about 30% for gate MOVs. These values are based on test results for 45 gate MOVs which were tested under both Static and DP pumped flow conditions. For the PORV block valves under "blowdown" conditions, the average "Rate of Loading" effect was 30%. The postulated worst case "Rate of Loading" effect for these MOVs under torque switch controlled closure is 58% (the greater of the two test data points is 40%). For globe valves, the average effect is 11%, with a maximum test result of 51%, for 15 test data points. (See Tables 9 and 10, and "Appendix A")

**"Rate of Loading" Effects for Limit Switch Controlled Closure MOVs**

The "Rate of Loading" effect is primarily one of greater stem thread friction coefficients during valve strokes under DP conditions than during valve strokes under Static conditions. For limit switch controlled closure MOVs which produce the same stem thrust at close limit switch trip under both Static and DP conditions, the "Rate of Loading" effect causes the stem torque at closer limit switch trip to be greater by an average of 2% under DP conditions than under Static conditions. The worst case increase of stem torque under DP conditions as compared to Static conditions is postulated to be 15% based on analysis of Comanche Peak test data.

One group of Borg-Warner gate valves demonstrated higher stem thrusts at close limit switch trip under DP pumped flow conditions than under Static conditions (the "Disk Position Effect"). With a thrust increase under DP conditions postulated to be as much as 9%, the resulting stem torque would also be 9% greater for the same stem factor. Coupled with the "Rate of Loading" effect (greater stem factors under DP conditions), TU Electric data analysis projects a worst case combined effect of 29% greater torque under DP conditions than under Static conditions at close limit switch trip.

The PORV block valves, wired for limit switch controlled closure, demonstrated an average increase in stem thrust at close limit switch trip of 17% under DP "blowdown" conditions.
versus "Static" conditions. The postulated maximum increase is 31%. Coupled with the "Rate of Loading" effect (greater stem factors under DP conditions than under Static conditions), TU Electric data analysis determined an average increase in stem torque of 47%, a maximum test result of a 66% increase, and a projected a worst case increase of 88%. (See Table 11, and "Appendix A")

*Stem Thread Friction Coefficient Range
TU Electric has observed tremendous variation in stem thread friction coefficients for groups of nominally identical MOVs in nominally identical condition with the same type grease of nominally identical quality and quantity on the stem threads. Analysis of the test data suggests assuming a range of 0.05 to 0.20, with an average of 0.12; for the stem thread friction coefficient of any MOV for which there is no test data to justify otherwise. At initial unwedging of the disk from the valve seat, a range of 0.03 to 0.19, with an average of 0.10, appears to be appropriate. (See Table 12, and "Appendix A")

*Margin for Stem Factor Degradation
Analysis of presently available data indicates that on the average there is no degradation (increase) of the stem friction coefficient over an outage cycle and that if any degradation does occur, it is unlikely to result in a stem factor increase greater than 10%. Thus, an uncertainty of 10% is used to account for potential degradation of the stem factor.

TU Electric presently cleans and relubricates all MOV stems each refueling outage. Over a period of N refueling cycles, the initial assumption may be that the degradation of the stem factor over these N cycles will not be more than determined by equation (1):

\[
\text{Stem Factor Degradation over N Refueling Cycles} = (N)^{\frac{1}{2}} (0.10)
\]  

(1)

MOVs which have calibrated stem-mounted strain gages for measuring both thrust and torque can continue to provide test data throughout the remaining plant life. By analyzing this data, the TU Electric MOV program can refine as necessary the magnitude of the uncertainty used to account for stem factor degradation.
STATIC VERIFICATION TESTING FOR THE LIFE OF THE PLANT

All MOVs which could be tested under DP conditions were DP tested if the achievable test conditions were considered by engineering to be sufficiently close to the maximum design basis conditions so that the test data collected could be reliably used to verify the initial assumptions made by engineering. Approximately 60% of the MOVs in Unit 1 and 90% in Unit 2 were tested under DP conditions.

TU Electric intends to use Static testing as much as possible throughout the remainder of the plant operating life to verify the readiness of MOVs to perform their design basis functions. Individual MOVs may experience increases in their valve factors from the values observed during baseline testing. But, based on TU Electric’s test results and the results of industry test data, the valve factor values used by TU Electric are reasonable upper bound values.

TU Electric’s test data analysis identifies groups of similar MOVs and applies the results of each group’s test data analysis to all MOVs in the group. This accommodates the potential for varying MOV performance of each MOV over its remaining service life. Approximately one half of the MOVs in the group performed worse than the average, while the other half performed better than the average. Yet, each MOV in the group is treated as if its performance was the worst of the group.

The magnitudes used by TU Electric for the “Rate of Loading” factor, “Stem Thrust Effect” factor, and the range of potential stem thread friction coefficients, contribute margin which, if not needed to compensate for the actual magnitudes of these effects, is also available to compensate for reasonably anticipated increases in valve factor values. A high degree of confidence is being provided that the MOVs which are important to the safe operation and shut down of Comanche Peak are capable of performing their design basis functions. The information and processes utilized by TU Electric are believed to be commensurate with the best available information and processes in the nuclear power industry.
ACKNOWLEDGEMENTS

Texas Utilities Electric Company management is to be commended for their excellent support of the Comanche Peak motor operated valve program. Engineering, testing, and refurbishment personnel, most of whom have not been TU Electric employees, are to be commended for their willingness to adopt TU Electric's MOV program as their own and make fine intellectual contributions toward its success, along with their many months of laborious implementation of the program. Without these many people and their endurance of frequent changes to the engineering calculation, test procedures, and test equipment, the TU Electric MOV program would not have realized its successes. We have worked as a team, and to each individual: thank you.

Sid Chiu first proposed that TU Electric analyze the test data to determine averages and associated uncertainties, and to combine the uncertainties by the SRSS methodology. Charlie Catino and Rickey Page worked with the author for several months analyzing and reanalyzing data, while Dave Manning, Johnnie Verricchie, Jim Lee, Brian Robinson and others assisted by compiling and checking the vast amounts of data analyzed. Sid Chiu and Scott Rosenberger reviewed the resulting analysis. Scott Rosenberger, Tom August, Tom Brown and others worked with the author to prepare a complete revision of the engineering calculation which utilizes the test data analysis results. Charlie Catino and Rickey Page worked with the author to issue major revisions of in situ MOV test procedures in order to use the results of the new engineering calculation.

Those who reviewed and commented on drafts of this paper or the test data discussed herein include: Kevin G. DeWall and Robert Steele, Jr. of the Idaho National Engineering Laboratory (INEL); Dr. Gerald H. Weidenhamer of the United States Nuclear Regulatory Commission; Norman Dingman of Nebraska Public Power District; Dr. Kalsi of Kalsi Engineering Corporation; Tim Cline of Duke Power; and Paul Damerell of MPR Associates.

Opinions expressed in this paper are those of the author and do not necessarily reflect the opinions of TU Electric, TU Electric's employees, TU Electric's contractors, or those who reviewed this paper.
TABLE 1: Motor Capability: Actual versus Initial Design Calculation

24 different actuators tested to stall at 80% voltage in close, open, or both directions.
Actuator = Size 00; Motor = 15 ft.lb 3400 rpm; worm/worm gear ratio = 45:1

<table>
<thead>
<tr>
<th>OAR</th>
<th>AOTQ80w (ft.lb)</th>
<th>AOTQ80stall close (ft.lb)</th>
<th>AOTQ80stall open (ft.lb)</th>
<th>close ratio</th>
<th>open ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>119.0</td>
<td>108.0</td>
<td>0.9983</td>
<td>0.9060</td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>123.1</td>
<td>112.8</td>
<td>1.0327</td>
<td>0.9463</td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>131.7</td>
<td>118.4</td>
<td>1.1049</td>
<td>0.9933</td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>129.9</td>
<td>119.7</td>
<td>1.0898</td>
<td>1.0042</td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>135.2</td>
<td>124.0</td>
<td>1.1342</td>
<td>1.0403</td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>140.6</td>
<td>126.8</td>
<td>1.1795</td>
<td>1.0638</td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>131.9</td>
<td>131.9</td>
<td>1.1065</td>
<td></td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>134.5</td>
<td>135.0</td>
<td>1.1284</td>
<td>1.1326</td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>138.5</td>
<td>138.5</td>
<td>1.1619</td>
<td></td>
</tr>
<tr>
<td>23.0:1</td>
<td>119.2</td>
<td>147.9</td>
<td>147.9</td>
<td>1.2408</td>
<td></td>
</tr>
<tr>
<td>30.0:1</td>
<td>155.5</td>
<td>172.9</td>
<td>155.6</td>
<td>1.1119</td>
<td>1.0006</td>
</tr>
<tr>
<td>30.0:1</td>
<td>155.5</td>
<td>160.6</td>
<td>160.6</td>
<td>1.0328</td>
<td></td>
</tr>
<tr>
<td>31.9:1</td>
<td>165.4</td>
<td>173.1</td>
<td>173.1</td>
<td>1.0466</td>
<td></td>
</tr>
<tr>
<td>31.9:1</td>
<td>165.4</td>
<td>171.6</td>
<td>161.4</td>
<td>1.0375</td>
<td>0.9758</td>
</tr>
<tr>
<td>31.9:1</td>
<td>165.4</td>
<td>184.4</td>
<td>184.4</td>
<td>1.1149</td>
<td></td>
</tr>
<tr>
<td>31.9:1</td>
<td>165.4</td>
<td>213.6</td>
<td>195.7</td>
<td>1.2914</td>
<td>1.1832</td>
</tr>
<tr>
<td>34.1:1</td>
<td>176.8</td>
<td>185.0</td>
<td>185.0</td>
<td>1.0464</td>
<td></td>
</tr>
</tbody>
</table>

(Table 1 is continued on the next page.)
<table>
<thead>
<tr>
<th>OAR</th>
<th>AOTQ80w (ft.lb)</th>
<th>AOTQ80stall close (ft.lb)</th>
<th>AOTQ80stall open (ft.lb)</th>
<th>close ratio</th>
<th>open ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>34.1:1</td>
<td>176.8</td>
<td></td>
<td>202.0</td>
<td></td>
<td>1.1425</td>
</tr>
<tr>
<td>36.3:1</td>
<td>188.2</td>
<td>229.5</td>
<td></td>
<td>1.2194</td>
<td></td>
</tr>
<tr>
<td>36.3:1</td>
<td>188.2</td>
<td>198.0</td>
<td></td>
<td>1.0521</td>
<td></td>
</tr>
<tr>
<td>36.3:1</td>
<td>188.2</td>
<td>193.2</td>
<td>187.3</td>
<td>1.0266</td>
<td>0.9952</td>
</tr>
<tr>
<td>36.3:1</td>
<td>188.2</td>
<td></td>
<td>193.0</td>
<td></td>
<td>1.0255</td>
</tr>
<tr>
<td>36.3:1</td>
<td>188.2</td>
<td>200.0</td>
<td>199.0</td>
<td>1.0627</td>
<td>1.0574</td>
</tr>
<tr>
<td>36.3:1</td>
<td>188.2</td>
<td>233.9</td>
<td>216.6</td>
<td>1.2428</td>
<td>1.1509</td>
</tr>
</tbody>
</table>

The "Performance Factor" is the average of all close and open direction ratios: \( PF = 1.083 \)

One sample standard deviation of all the ratios: \( s = 0.087 \)

The average minus two sample standard deviations: \( PF - 2s = 0.909 \)

The minimum ratio of all the close and open direction data: \( \text{MIN} = 0.906 \)

The uncertainty associated with PF is the percentage by which PF must be reduced to obtain the lesser of (a) the value of the difference \( PF - 2s \), or (b) the value of MIN, whichever is less:

\[
ePF = \{ 1 - \left( \frac{PF - 2s}{PF} \right) \} \text{ or } \{ 1 - \left( \frac{\text{MIN}}{PF} \right) \} = \left( 1 - \left( \frac{0.906}{1.083} \right) \right) = 0.16
\]
<table>
<thead>
<tr>
<th>Act.</th>
<th>MTQ (ft.lb)</th>
<th>RPM</th>
<th>OAR Range</th>
<th>W/WG ratio</th>
<th>RE</th>
<th>PF</th>
<th>ePF</th>
<th>(PF)(1 - ePF)</th>
<th>(PF)(1 - ePF)/(RE)</th>
<th>&gt; RE? (yes/no)</th>
</tr>
</thead>
<tbody>
<tr>
<td>00</td>
<td>2</td>
<td>1700</td>
<td>23.0-63.3</td>
<td>50:1</td>
<td>0.50</td>
<td>1.21</td>
<td>0.16</td>
<td>1.02</td>
<td>0.51</td>
<td>yes</td>
</tr>
<tr>
<td>000</td>
<td>5</td>
<td>1700</td>
<td>33.3</td>
<td>50:1</td>
<td>0.50</td>
<td>1.21</td>
<td>0.16</td>
<td>1.02</td>
<td>0.51</td>
<td>yes</td>
</tr>
<tr>
<td>00</td>
<td>10</td>
<td>1700</td>
<td>23.0-63.3</td>
<td>45:1</td>
<td>0.50</td>
<td>1.41</td>
<td>0.16</td>
<td>1.18</td>
<td>0.59</td>
<td>yes</td>
</tr>
<tr>
<td>00</td>
<td>10</td>
<td>3400</td>
<td>31.6-34.1</td>
<td>45:1</td>
<td>0.60</td>
<td>1.27</td>
<td>0.21</td>
<td>1.00</td>
<td>0.60</td>
<td>yes</td>
</tr>
<tr>
<td>00</td>
<td>15</td>
<td>3400</td>
<td>23.0-36.3</td>
<td>45:1</td>
<td>0.60</td>
<td>1.08</td>
<td>0.16</td>
<td>0.91</td>
<td>0.55</td>
<td>NO</td>
</tr>
<tr>
<td>0</td>
<td>25</td>
<td>1700</td>
<td>29.6</td>
<td>37:1</td>
<td>0.55</td>
<td>1.25</td>
<td>0.08</td>
<td>1.15</td>
<td>0.63</td>
<td>yes</td>
</tr>
<tr>
<td>0</td>
<td>40</td>
<td>1700</td>
<td>29.6</td>
<td>37:1</td>
<td>0.55</td>
<td>1.12</td>
<td>0.06</td>
<td>1.05</td>
<td>0.58</td>
<td>yes</td>
</tr>
<tr>
<td>1</td>
<td>60</td>
<td>3400</td>
<td>27.2-32.1</td>
<td>34:1</td>
<td>0.60</td>
<td>1.10</td>
<td>0.08</td>
<td>1.01</td>
<td>0.61</td>
<td>yes</td>
</tr>
<tr>
<td>2</td>
<td>80</td>
<td>3400</td>
<td>27.8</td>
<td>33:1</td>
<td>0.60</td>
<td>1.10</td>
<td>0.09</td>
<td>1.00</td>
<td>0.60</td>
<td>yes</td>
</tr>
</tbody>
</table>
TABLE 3: Values of "MARM", "MARMw", and "eTQ"

<table>
<thead>
<tr>
<th>Actuator Type-Size</th>
<th>MARM (ft)</th>
<th>MARMw (ft)</th>
<th>Approx. eTQ (ft.lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMB-000</td>
<td>0.11</td>
<td></td>
<td>6</td>
</tr>
<tr>
<td>SMB-00</td>
<td>0.14</td>
<td>0.17</td>
<td>20</td>
</tr>
<tr>
<td>SB-00</td>
<td>0.16</td>
<td>0.17</td>
<td>20</td>
</tr>
<tr>
<td>SMB-0</td>
<td>0.20</td>
<td>0.24</td>
<td>40</td>
</tr>
<tr>
<td>SB-1</td>
<td>0.23</td>
<td>0.26</td>
<td>60</td>
</tr>
<tr>
<td>SB-2</td>
<td>0.28</td>
<td>0.31</td>
<td>90</td>
</tr>
</tbody>
</table>

The above values apply throughout the closing stroke and in the opening stroke after unwedging of the disk from the valve seat.
<table>
<thead>
<tr>
<th>SPF</th>
<th>TQsg</th>
<th>TQtts</th>
<th>(TQtts/TQsg) / (MARMw/MARM)</th>
<th>(TQsg)(ST)(1 + eST)</th>
<th>&gt; TQtts? (yes/no)</th>
</tr>
</thead>
<tbody>
<tr>
<td>915</td>
<td>199</td>
<td>263</td>
<td>1.171*</td>
<td>263</td>
<td>yes</td>
</tr>
<tr>
<td>1890</td>
<td>425</td>
<td>464</td>
<td>0.968</td>
<td>562</td>
<td>yes</td>
</tr>
<tr>
<td>1124</td>
<td>261</td>
<td>263</td>
<td>0.893*</td>
<td>345</td>
<td>yes</td>
</tr>
<tr>
<td>2462</td>
<td>561</td>
<td>599</td>
<td>0.947</td>
<td>741</td>
<td>yes</td>
</tr>
<tr>
<td>1789</td>
<td>370</td>
<td>414</td>
<td>0.992</td>
<td>489</td>
<td>yes</td>
</tr>
<tr>
<td>834</td>
<td>136</td>
<td>178</td>
<td>1.160*</td>
<td>180</td>
<td>yes</td>
</tr>
<tr>
<td>1485</td>
<td>386</td>
<td>436</td>
<td>1.001</td>
<td>510</td>
<td>yes</td>
</tr>
<tr>
<td>592</td>
<td>159</td>
<td>183</td>
<td>1.020*</td>
<td>210</td>
<td>yes</td>
</tr>
<tr>
<td>1505</td>
<td>388</td>
<td>449</td>
<td>1.026</td>
<td>513</td>
<td>yes</td>
</tr>
<tr>
<td>2270</td>
<td>511</td>
<td>600</td>
<td>1.041</td>
<td>675</td>
<td>yes</td>
</tr>
<tr>
<td>1219</td>
<td>287</td>
<td>321</td>
<td>0.991</td>
<td>379</td>
<td>yes</td>
</tr>
<tr>
<td>2114</td>
<td>485</td>
<td>553</td>
<td>1.011</td>
<td>641</td>
<td>yes</td>
</tr>
<tr>
<td>2482</td>
<td>577</td>
<td>636</td>
<td>0.977</td>
<td>763</td>
<td>yes</td>
</tr>
<tr>
<td>1415</td>
<td>311</td>
<td>362</td>
<td>1.032</td>
<td>411</td>
<td>yes</td>
</tr>
<tr>
<td>2482</td>
<td>550</td>
<td>636</td>
<td>1.025</td>
<td>727</td>
<td>yes</td>
</tr>
</tbody>
</table>

*Torque spring pack deflection magnitudes were typical for settings of "1": 10% repeatability.

MARMw = 0.2572 ft., and MARM = 0.2279 ft.

ST = (MARMw / MARM)(AVE) = 1.15

eST = the greater of [2s/AVE] or [(MAX/AVE) - 1] = 0.15

where,

for the column labeled "(TQtts/TQsg)/(MARMw/MARM)":

the average of the data, AVE = 1.017
one sample standard deviation, s = 0.0713

the maximum data point, MAX = 1.171
(Note: MAX > AVE + 2s = 1.159)
TABLE 5: "Stem Thrust Effect" Magnitudes and Uncertainties

<table>
<thead>
<tr>
<th>Actuator Type-Size</th>
<th>ST</th>
<th>eST</th>
<th>Maximum TQttts/TQsg</th>
<th>&lt; ST(1 + eST) ? (yes/no)</th>
<th>Minimum TQttts/TQsg</th>
<th>&gt; ST(1-eST) ? (yes/no)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMB-00</td>
<td>1.20</td>
<td>0.18</td>
<td>1.32</td>
<td>1.42 yes</td>
<td>1.05</td>
<td>0.98 yes</td>
</tr>
<tr>
<td>SMB-0</td>
<td>1.23</td>
<td>0.18</td>
<td>1.44</td>
<td>1.45 yes</td>
<td>1.02</td>
<td>1.00 yes</td>
</tr>
<tr>
<td>SB-1</td>
<td>1.15</td>
<td>0.15</td>
<td>1.32</td>
<td>1.32 yes</td>
<td>1.01</td>
<td>0.98 yes</td>
</tr>
<tr>
<td>SB-2</td>
<td>1.10</td>
<td>0.13</td>
<td>1.19</td>
<td>1.24 yes</td>
<td>0.99</td>
<td>0.96 yes</td>
</tr>
</tbody>
</table>

Note that for each actuator type-size the maximum test data point is less than (or equal to) the maximum predicted "Stem Thrust Effect" and the least test data point is greater than the minimum predicted "Stem Thrust Effect". This provides verification that the predictions are appropriate (slightly conservative).
### TABLE 6a: Net Impact of "Performance Factors" and "Stem Thrust Effects"

<table>
<thead>
<tr>
<th>Actuator Type-Size</th>
<th>MTQ (ft.lb)</th>
<th>RPM</th>
<th>worm-to-worm gear ratio</th>
<th>PF</th>
<th>ePF</th>
<th>ST</th>
<th>eST</th>
<th>PF/ST</th>
<th>NET</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMB-00</td>
<td>10</td>
<td>1700</td>
<td>45:1</td>
<td>1.40</td>
<td>0.16</td>
<td>1.20</td>
<td>0.18</td>
<td>1.17</td>
<td>0.89</td>
</tr>
<tr>
<td>SMB-00</td>
<td>10</td>
<td>3400</td>
<td>45:1</td>
<td>1.27</td>
<td>0.21</td>
<td>1.20</td>
<td>0.18</td>
<td>1.06</td>
<td>0.77</td>
</tr>
<tr>
<td>SMB-00</td>
<td>15</td>
<td>3400</td>
<td>45:1</td>
<td>1.00</td>
<td>0.16</td>
<td>1.20</td>
<td>0.18</td>
<td>0.83</td>
<td>0.68</td>
</tr>
<tr>
<td>SMB-0</td>
<td>25</td>
<td>1700</td>
<td>37:1</td>
<td>1.25</td>
<td>0.08</td>
<td>1.23</td>
<td>0.18</td>
<td>1.02</td>
<td>0.82</td>
</tr>
<tr>
<td>SMB-0</td>
<td>40</td>
<td>1700</td>
<td>37:1</td>
<td>1.12</td>
<td>0.06</td>
<td>1.23</td>
<td>0.18</td>
<td>0.91</td>
<td>0.74</td>
</tr>
<tr>
<td>SB-1</td>
<td>60</td>
<td>3400</td>
<td>34:1</td>
<td>1.10</td>
<td>0.08</td>
<td>1.15</td>
<td>0.15</td>
<td>0.96</td>
<td>0.79</td>
</tr>
<tr>
<td>SB-2</td>
<td>80</td>
<td>3400</td>
<td>33:1</td>
<td>1.10</td>
<td>0.09</td>
<td>1.10</td>
<td>0.13</td>
<td>1.00</td>
<td>0.84</td>
</tr>
</tbody>
</table>

*The Table 6a column labeled "PF/ST" shows the average combined effect of PF and ST.  
*The Table 6a column labeled "NET" shows the lower bound combined effect of PF and ST.  
*The value of "NET" includes the effects of actuator output torque repeatability, and in each case exceeds the magnitude of the actuator repeatability uncertainty specified by the actuator manufacturer:

\[
NET = (PF) / \left(1 - \left[ ePF^2 + eST^2 \right]^{1/2} \right) / (ST)
\]
TABLE 6b: Using a Reduced "Running Efficiency" versus using the "Pullout Efficiency"

<table>
<thead>
<tr>
<th>Actuator Type-Size</th>
<th>MTQ (ft.lb)</th>
<th>RPM</th>
<th>worm-to-worm gear ratio</th>
<th>RE</th>
<th>(RE)(NET)</th>
<th>&gt; POE ? (yes/no)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMB-00</td>
<td>10</td>
<td>1700</td>
<td>45:1</td>
<td>0.50</td>
<td>0.45</td>
<td>0.40 yes</td>
</tr>
<tr>
<td>SMB-00</td>
<td>10</td>
<td>3400</td>
<td>45:1</td>
<td>0.60</td>
<td>0.46</td>
<td>0.45 yes</td>
</tr>
<tr>
<td>SMB-00</td>
<td>15</td>
<td>3400</td>
<td>45:1</td>
<td>0.60</td>
<td>0.41</td>
<td>0.45 NO</td>
</tr>
<tr>
<td>SMB-0</td>
<td>25</td>
<td>1700</td>
<td>37:1</td>
<td>0.55</td>
<td>0.45</td>
<td>0.45 yes</td>
</tr>
<tr>
<td>SMB-0</td>
<td>40</td>
<td>1700</td>
<td>37:1</td>
<td>0.55</td>
<td>0.41</td>
<td>0.45 NO</td>
</tr>
<tr>
<td>SB-1</td>
<td>60</td>
<td>3400</td>
<td>34:1</td>
<td>0.60</td>
<td>0.47</td>
<td>0.45 yes</td>
</tr>
<tr>
<td>SB-2</td>
<td>80</td>
<td>3400</td>
<td>33:1</td>
<td>0.60</td>
<td>0.50</td>
<td>0.45 yes</td>
</tr>
</tbody>
</table>

*Table 6b multiplies "NET" times the "Running Efficiency" (RE) published by the actuator manufacturer and compares the result to the "Pullout Efficiency" (POE) published by the actuator manufacturer. In most cases, the use of POE is observed to be conservative: it is less than the product (RE)(NET).*
TABLE 7: Results of Running Thrust Data Analysis

<table>
<thead>
<tr>
<th>Dp (inch)</th>
<th>RT (lb)</th>
<th>eRT (%/100)</th>
<th>RT/Dp (lb/in)</th>
<th>RT(1 + eRT)/Dp</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.750</td>
<td>620</td>
<td>0.50</td>
<td>827</td>
<td>1240</td>
</tr>
<tr>
<td>1.000</td>
<td>310</td>
<td>1.00</td>
<td>310</td>
<td>620</td>
</tr>
<tr>
<td>1.000</td>
<td>730</td>
<td>0.65</td>
<td>730</td>
<td>1205</td>
</tr>
<tr>
<td>1.125</td>
<td>500</td>
<td>0.75</td>
<td>444</td>
<td>778</td>
</tr>
<tr>
<td>1.250</td>
<td>720</td>
<td>0.75</td>
<td>576</td>
<td>1008</td>
</tr>
<tr>
<td>1.375</td>
<td>1370</td>
<td>0.70</td>
<td>996</td>
<td>1694</td>
</tr>
<tr>
<td>1.875</td>
<td>2100</td>
<td>0.70</td>
<td>1120</td>
<td>1904</td>
</tr>
<tr>
<td>2.000</td>
<td>2100</td>
<td>0.60</td>
<td>1050</td>
<td>1680</td>
</tr>
<tr>
<td>2.500</td>
<td>3200</td>
<td>0.30</td>
<td>1280</td>
<td>1664</td>
</tr>
<tr>
<td>3.000</td>
<td>3650</td>
<td>1.10</td>
<td>1217</td>
<td>2555</td>
</tr>
<tr>
<td>Group ID</td>
<td>Manufacturer</td>
<td>Size (inch)</td>
<td>Do (inch)</td>
<td>Dp (inch)</td>
</tr>
<tr>
<td>----------</td>
<td>--------------</td>
<td>-------------</td>
<td>-----------</td>
<td>-----------</td>
</tr>
<tr>
<td>GT1</td>
<td>W</td>
<td>3</td>
<td>2.688</td>
<td>1.250</td>
</tr>
<tr>
<td>GT2</td>
<td>W</td>
<td>3</td>
<td>2.688</td>
<td>1.250</td>
</tr>
<tr>
<td>GT3</td>
<td>W</td>
<td>4</td>
<td>3.508</td>
<td>1.250</td>
</tr>
<tr>
<td>GT4</td>
<td>W</td>
<td>4</td>
<td>3.623</td>
<td>1.250</td>
</tr>
<tr>
<td>GT5</td>
<td>BW</td>
<td>4</td>
<td>4.020</td>
<td>1.375</td>
</tr>
<tr>
<td>GT6</td>
<td>BW</td>
<td>4</td>
<td>4.020</td>
<td>1.375</td>
</tr>
<tr>
<td>GT7</td>
<td>BW</td>
<td>4</td>
<td>4.070</td>
<td>1.000</td>
</tr>
<tr>
<td>GT8</td>
<td>W</td>
<td>6</td>
<td>6.128</td>
<td>1.250</td>
</tr>
<tr>
<td>GT9</td>
<td>BW</td>
<td>6</td>
<td>6.310</td>
<td>1.250</td>
</tr>
<tr>
<td>GT10</td>
<td>W</td>
<td>8</td>
<td>6.586</td>
<td>1.250</td>
</tr>
<tr>
<td>GT11</td>
<td>W</td>
<td>10</td>
<td>8.818</td>
<td>2.500</td>
</tr>
<tr>
<td>GT12</td>
<td>W</td>
<td>10</td>
<td>10.09</td>
<td>2.000</td>
</tr>
<tr>
<td>GT13</td>
<td>W</td>
<td>12</td>
<td>10.57</td>
<td>3.000</td>
</tr>
<tr>
<td>GT14</td>
<td>W</td>
<td>14</td>
<td>12.07</td>
<td>2.000</td>
</tr>
<tr>
<td>GT15</td>
<td>BW</td>
<td>16</td>
<td>10.57</td>
<td>3.000</td>
</tr>
<tr>
<td>GT16</td>
<td>BW</td>
<td>8</td>
<td>8.260</td>
<td>1.375</td>
</tr>
</tbody>
</table>

(1) Manufacturers: "W" = Westinghouse; "BW" = Borg-Warner.
<table>
<thead>
<tr>
<th>Group ID</th>
<th>Manufacturer(1)</th>
<th># MOVs in the group (2)</th>
<th># DP tested Unit 1; Unit 2</th>
<th># DP tested with Stem Strain gages</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT1</td>
<td>W</td>
<td>4</td>
<td>0;2</td>
<td>2</td>
</tr>
<tr>
<td>GT2</td>
<td>W</td>
<td>4</td>
<td>2;2</td>
<td>4</td>
</tr>
<tr>
<td>GT3</td>
<td>W</td>
<td>14</td>
<td>7;7</td>
<td>14</td>
</tr>
<tr>
<td>GT4</td>
<td>W</td>
<td>4</td>
<td>0;2</td>
<td>2</td>
</tr>
<tr>
<td>GT5</td>
<td>BW</td>
<td>16</td>
<td>4;8</td>
<td>12</td>
</tr>
<tr>
<td>GT6</td>
<td>BW</td>
<td>4</td>
<td>0;2</td>
<td>2</td>
</tr>
<tr>
<td>GT7</td>
<td>BW</td>
<td>4</td>
<td>0;2</td>
<td>2</td>
</tr>
<tr>
<td>GT8</td>
<td>W</td>
<td>10</td>
<td>5;5</td>
<td>7</td>
</tr>
<tr>
<td>GT9</td>
<td>BW</td>
<td>10</td>
<td>3;5</td>
<td>3</td>
</tr>
<tr>
<td>GT10</td>
<td>W</td>
<td>10</td>
<td>2;5</td>
<td>6</td>
</tr>
<tr>
<td>GT11</td>
<td>W</td>
<td>14</td>
<td>3;7</td>
<td>7</td>
</tr>
<tr>
<td>GT12</td>
<td>W</td>
<td>4</td>
<td>2;2</td>
<td>4</td>
</tr>
<tr>
<td>GT13</td>
<td>W</td>
<td>8</td>
<td>0;4</td>
<td>4</td>
</tr>
<tr>
<td>GT14</td>
<td>W</td>
<td>8</td>
<td>0;4(open only)</td>
<td>4</td>
</tr>
<tr>
<td>GT15</td>
<td>BW</td>
<td>12</td>
<td>0;2</td>
<td>2</td>
</tr>
<tr>
<td>GT16</td>
<td>BW</td>
<td>10</td>
<td>4;5</td>
<td>4</td>
</tr>
</tbody>
</table>

(1) Manufacturers: "W" = Westinghouse; "BW" = Borg-Warner.
(2) All of the valves in each group are tested under Static conditions.
TABLE 8c: Borg-Warner and Westinghouse Gate Valve Factors, Closing

<table>
<thead>
<tr>
<th>Group ID</th>
<th>DPTc range (psid)</th>
<th>VFc</th>
<th>eVFc</th>
<th>(VFc)(1 + eVFc)</th>
<th>&gt; Max VFct? (yes/no)</th>
<th>Manufacturer's VFct</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT1</td>
<td>2010-2058</td>
<td>0.67</td>
<td>0.02</td>
<td>0.68</td>
<td>0.67 yes</td>
<td>0.56(3)</td>
</tr>
<tr>
<td>GT2</td>
<td>1657-2846</td>
<td>0.29</td>
<td>0.63</td>
<td>0.47</td>
<td>0.41 yes</td>
<td>0.56</td>
</tr>
<tr>
<td>GT3</td>
<td>811-2862</td>
<td>0.40</td>
<td>0.63</td>
<td>0.65</td>
<td>0.54 yes</td>
<td>0.61 ...</td>
</tr>
<tr>
<td>GT4</td>
<td>255-255</td>
<td>0.23</td>
<td>0.70</td>
<td>0.39</td>
<td>0.28 yes</td>
<td>0.63</td>
</tr>
<tr>
<td>GT5</td>
<td>1528-1749</td>
<td>0.48</td>
<td>0.31</td>
<td>0.63</td>
<td>0.62 yes</td>
<td>0.30(3)</td>
</tr>
<tr>
<td>GT6</td>
<td>2578-2878</td>
<td>0.50</td>
<td>0.06</td>
<td>0.53</td>
<td>0.51 yes</td>
<td>0.30(3)</td>
</tr>
<tr>
<td>GT7</td>
<td>95-102</td>
<td>0.43</td>
<td>0.16</td>
<td>0.50</td>
<td>0.45 yes</td>
<td>0.30(3)</td>
</tr>
<tr>
<td>GT8</td>
<td>131-251</td>
<td>0.36</td>
<td>0.37</td>
<td>0.49</td>
<td>0.42 yes</td>
<td>0.63</td>
</tr>
<tr>
<td>GT9</td>
<td>290-399</td>
<td>0.24</td>
<td>0.36</td>
<td>0.33</td>
<td>0.29 yes</td>
<td>0.30</td>
</tr>
<tr>
<td>GT10</td>
<td>126-254</td>
<td>0.38</td>
<td>0.51</td>
<td>0.57</td>
<td>0.50 yes</td>
<td>0.59</td>
</tr>
<tr>
<td>GT11</td>
<td>51-267</td>
<td>0.56</td>
<td>0.70</td>
<td>0.95</td>
<td>0.89 yes</td>
<td>0.64(3)</td>
</tr>
<tr>
<td>GT12</td>
<td>195-261</td>
<td>0.47</td>
<td>0.87</td>
<td>0.88</td>
<td>0.68 yes</td>
<td>0.59(3)</td>
</tr>
<tr>
<td>GT13</td>
<td>NA(4)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>GT14</td>
<td>NA(4)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>GT15</td>
<td>298-304</td>
<td>0.39</td>
<td>0.11</td>
<td>0.43</td>
<td>0.40 yes</td>
<td>0.30(3)</td>
</tr>
<tr>
<td>GT16</td>
<td>107-171</td>
<td>0.47</td>
<td>0.31</td>
<td>0.62</td>
<td>0.57 yes</td>
<td>0.30(3)</td>
</tr>
</tbody>
</table>

(3) Manufacturer's equivalent VFct value is less than maximum test result VFct.
(4) There is no closing requirement within the plant design basis for this group of MOVs.
TABLE 8d: Borg-Warner and Westinghouse Gate Valve Factors, Opening

<table>
<thead>
<tr>
<th>Group ID</th>
<th>DPTo range (psid)</th>
<th>VFo</th>
<th>eVFo</th>
<th>(VFo)(1 + eVFo)</th>
<th>&gt; Max VFot?</th>
<th>Manufacturer’s VFo</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT1</td>
<td>2081-2128</td>
<td>0.53</td>
<td>0.13</td>
<td>0.60</td>
<td>0.55</td>
<td>yes</td>
</tr>
<tr>
<td>GT2</td>
<td>1815-2851</td>
<td>0.32</td>
<td>0.63</td>
<td>0.52</td>
<td>0.43</td>
<td>yes</td>
</tr>
<tr>
<td>GT3</td>
<td>1651-2740</td>
<td>0.37</td>
<td>0.44</td>
<td>0.53</td>
<td>0.48</td>
<td>yes</td>
</tr>
<tr>
<td>GT4</td>
<td>200-200</td>
<td>0.44</td>
<td>0.49</td>
<td>0.66</td>
<td>0.52</td>
<td>yes</td>
</tr>
<tr>
<td>GT5</td>
<td>1521-1637</td>
<td>0.41</td>
<td>0.33</td>
<td>0.55</td>
<td>0.54</td>
<td>yes</td>
</tr>
<tr>
<td>GT6</td>
<td>2579-2628</td>
<td>0.39</td>
<td>0.09</td>
<td>0.43</td>
<td>0.40</td>
<td>yes</td>
</tr>
<tr>
<td>GT7</td>
<td>139-158</td>
<td>0.36</td>
<td>0.78</td>
<td>0.64</td>
<td>0.45</td>
<td>yes</td>
</tr>
<tr>
<td>GT8</td>
<td>121-211</td>
<td>0.39</td>
<td>0.40</td>
<td>0.55</td>
<td>0.51</td>
<td>yes</td>
</tr>
<tr>
<td>GT9</td>
<td>97-215</td>
<td>0.38</td>
<td>0.62</td>
<td>0.62</td>
<td>0.48</td>
<td>yes</td>
</tr>
<tr>
<td>GT10</td>
<td>117-198</td>
<td>0.35</td>
<td>0.46</td>
<td>0.51</td>
<td>0.45</td>
<td>yes</td>
</tr>
<tr>
<td>GT11</td>
<td>523-1288</td>
<td>0.54</td>
<td>0.48</td>
<td>0.80</td>
<td>0.67</td>
<td>yes</td>
</tr>
<tr>
<td>GT12</td>
<td>187-252</td>
<td>0.53</td>
<td>0.85</td>
<td>0.98</td>
<td>0.84</td>
<td>yes</td>
</tr>
<tr>
<td>GT13</td>
<td>441-459</td>
<td>0.53</td>
<td>0.33</td>
<td>0.70</td>
<td>0.65</td>
<td>yes</td>
</tr>
<tr>
<td>GT14</td>
<td>454-465</td>
<td>0.33</td>
<td>0.34</td>
<td>0.44</td>
<td>0.37</td>
<td>yes</td>
</tr>
<tr>
<td>GT15</td>
<td>212-305</td>
<td>0.35</td>
<td>0.12</td>
<td>0.39</td>
<td>0.36</td>
<td>yes</td>
</tr>
<tr>
<td>GT16</td>
<td>NA(6)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(5) Manufacturer’s equivalent VFo value is less than maximum test result VFot.
(6) There is no opening requirement within the plant design basis for this group of MOVs.
<table>
<thead>
<tr>
<th>Group ID</th>
<th>max RSFt</th>
<th>maximum VFcts data</th>
<th>AVE VFcts</th>
<th>AVE VFcts + 2s</th>
<th><em>MAX VFcts</em></th>
<th>VFc</th>
<th>eVFc</th>
<th>(VFc)(1 + eVFc)</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT1</td>
<td>1.396</td>
<td>1.029</td>
<td>0.941</td>
<td>1.189</td>
<td>1.189</td>
<td>0.67</td>
<td>0.02</td>
<td>0.68</td>
</tr>
<tr>
<td>GT6</td>
<td>1.176</td>
<td>0.600</td>
<td>0.547</td>
<td>0.698</td>
<td>0.698</td>
<td>0.50</td>
<td>0.06</td>
<td>0.53</td>
</tr>
<tr>
<td>GT7</td>
<td>1.111</td>
<td>0.493</td>
<td>0.492</td>
<td>0.495</td>
<td>0.495</td>
<td>0.43</td>
<td>0.16</td>
<td>0.50</td>
</tr>
<tr>
<td>GT9</td>
<td>1.028</td>
<td>0.305</td>
<td>0.262</td>
<td>0.383</td>
<td>0.383</td>
<td>0.26</td>
<td>0.47</td>
<td>0.38</td>
</tr>
<tr>
<td>GT11</td>
<td>1.024</td>
<td>0.897</td>
<td>0.523</td>
<td>0.916</td>
<td>0.916</td>
<td>0.56</td>
<td>0.70</td>
<td>0.95</td>
</tr>
<tr>
<td>GT12</td>
<td>1.150</td>
<td>0.765</td>
<td>0.724</td>
<td>0.841</td>
<td>0.841</td>
<td>0.47</td>
<td>0.87</td>
<td>0.88</td>
</tr>
<tr>
<td>GT15</td>
<td>1.082</td>
<td>0.439</td>
<td>0.405</td>
<td>0.501</td>
<td>0.501</td>
<td>0.39</td>
<td>0.11</td>
<td>0.43</td>
</tr>
<tr>
<td>GT16</td>
<td>1.174</td>
<td>0.567</td>
<td>0.508</td>
<td>0.594</td>
<td>0.594</td>
<td>0.47</td>
<td>0.31</td>
<td>0.62</td>
</tr>
</tbody>
</table>
### TABLE 9b: Rate of Loading Effect for Torque Switch Controlled Closure MOVs

<table>
<thead>
<tr>
<th>Group ID</th>
<th>ROL</th>
<th>((VFc)(1 + eVFc)/(ROL))</th>
<th>eROL</th>
<th>VFrol</th>
<th>&gt; &quot;MAX VFcts&quot;? (yes/no)</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT1</td>
<td>1.40</td>
<td>0.96</td>
<td>0.26</td>
<td>1.19</td>
<td>&gt; 1.03 yes</td>
</tr>
<tr>
<td>GT6</td>
<td>1.09</td>
<td>0.58</td>
<td>0.27</td>
<td>0.70</td>
<td>&gt; 0.60 yes</td>
</tr>
<tr>
<td>GT7</td>
<td>1.00</td>
<td>0.50</td>
<td>0.00</td>
<td>0.50</td>
<td>&gt; 0.49 yes</td>
</tr>
<tr>
<td>GT9</td>
<td>1.01</td>
<td>0.39</td>
<td>0.00</td>
<td>0.39</td>
<td>&gt; 0.31 yes</td>
</tr>
<tr>
<td>GT11</td>
<td>1.00</td>
<td>0.95</td>
<td>0.00</td>
<td>0.95</td>
<td>&gt; 0.90 yes</td>
</tr>
<tr>
<td>GT12</td>
<td>1.00</td>
<td>0.88</td>
<td>0.00</td>
<td>0.88</td>
<td>&gt; 0.77 yes</td>
</tr>
<tr>
<td>GT15</td>
<td>1.04</td>
<td>0.45</td>
<td>0.21</td>
<td>0.50</td>
<td>&gt; 0.44 yes</td>
</tr>
<tr>
<td>GT16</td>
<td>1.00</td>
<td>0.62</td>
<td>0.00</td>
<td>0.62</td>
<td>&gt; 0.57 yes</td>
</tr>
</tbody>
</table>
TABLE 10: "Rate of Loading" as a Ratio of the DP to the Static Condition Stem Factor

<table>
<thead>
<tr>
<th>MOV Type</th>
<th>GLOBE</th>
<th>GATE, Group GT1</th>
<th>GATE, Groups GT2 to GT16</th>
</tr>
</thead>
<tbody>
<tr>
<td>average ROL = AVE RSFt =</td>
<td>1.11</td>
<td>1.30</td>
<td>1.04</td>
</tr>
<tr>
<td>sample standard deviation, s =</td>
<td>0.15</td>
<td>0.14</td>
<td>0.10</td>
</tr>
<tr>
<td>( AVE RSFt + 2s ) =</td>
<td>1.41</td>
<td>1.58</td>
<td>1.24</td>
</tr>
<tr>
<td>( AVE RSFt + 3s ) =</td>
<td>1.56</td>
<td>1.72</td>
<td>1.34</td>
</tr>
<tr>
<td>maximum ROL = Max RSFt =</td>
<td>1.51</td>
<td>1.40</td>
<td>1.31</td>
</tr>
<tr>
<td>number of data points, n =</td>
<td>15</td>
<td>2</td>
<td>45</td>
</tr>
<tr>
<td>Group ID</td>
<td>DTQ</td>
<td>eDTQ</td>
<td>DT</td>
</tr>
<tr>
<td>---------</td>
<td>------</td>
<td>------</td>
<td>-----</td>
</tr>
<tr>
<td>GT1</td>
<td>0.68</td>
<td>0.22</td>
<td>0.85</td>
</tr>
<tr>
<td>GT2</td>
<td>0.98</td>
<td>0.11</td>
<td>1.00</td>
</tr>
<tr>
<td>GT3</td>
<td>0.98</td>
<td>0.11</td>
<td>1.00</td>
</tr>
<tr>
<td>GT8</td>
<td>0.98</td>
<td>0.11</td>
<td>1.00</td>
</tr>
<tr>
<td>GT9</td>
<td>0.98</td>
<td>0.11</td>
<td>1.00</td>
</tr>
<tr>
<td>GT10</td>
<td>0.98</td>
<td>0.11</td>
<td>1.00</td>
</tr>
<tr>
<td>GT12</td>
<td>0.98</td>
<td>0.11</td>
<td>1.00</td>
</tr>
<tr>
<td>GT16</td>
<td>0.92</td>
<td>0.15</td>
<td>0.98</td>
</tr>
</tbody>
</table>
### TABLE 12: Stem Thread Friction Coefficient Range

<table>
<thead>
<tr>
<th></th>
<th>During Initial Unseating (Unwedging of the Disk)</th>
<th>At Other points of Interest: Peak Load to Overcome DP, Control Switch Trip, and Total Load in the Closing and Opening Strokes</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Based on Analysis of over 700 Test Data Points from Calibrated Strain Gages Mounted on the Stem</strong></td>
<td><strong>Uun, u = 0.19</strong></td>
<td><strong>Uu = 0.20</strong></td>
</tr>
<tr>
<td><strong>Maximum Friction Coefficient</strong></td>
<td><strong>Uun = 0.10</strong></td>
<td><strong>U = 0.12</strong></td>
</tr>
<tr>
<td><strong>Average Friction Coefficient</strong></td>
<td><strong>Uun = 0.03</strong></td>
<td><strong>UI = 0.05</strong></td>
</tr>
<tr>
<td><strong>Minimum Friction Coefficient</strong></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
APPENDIX A : DETAILED DISCUSSION OF TEST DATA AND ANALYSIS RESULTS

This appendix discusses in further detail most of the test results summarized in the main body of this paper. Equation (1) of this paper was presented in the main body of this paper. Thus, the first equation presented in this appendix is equation (2).

Actuator Performance Factors

TU Electric has tested numerous actuators on torque test stands by gradually applying braking torques over several seconds until their motors stalled with power supplied at 80% of the motors' nameplate voltage. The torque test stands use splined stem adapters instead of threaded stem nuts. Thus, during torque stand testing no thrust load is applied to the actuator drive sleeve. In each of these cases, with the actuator having just been fully refurbished, it is assumed that the measured actuator output torque is essentially equal to the torque applied by the actuator worm to the actuator worm gear.

TU Electric test data indicates that when the measured motor stall torque is not less than its design capability as defined by TU Electric, the capability of the motor to deliver torque to the worm gear (when the motor is powered by 80% of the motor nameplate voltage) during torque stand testing is generally greater than the value of AOTQ80w predicted by the common design calculation shown in equation (2):

$$AOTQ80w = (0.8)^2 (MTQ)(OAR)(AF)(RE)$$  (2)

where

- 0.8 = factor for power supply at 80% of nameplate voltage
- MTQ = motor torque rating (ft.lb)
- OAR = overall gear ratio of the actuator from motor to stem nut (or HBC input shaft)
- AF = application factor = 0.9
- RE = "running efficiency" of actuator gear train, from actuator vendor design data

Actuator configurations have been considered to be similar if the actuators (a) are the same size, (b) have the same motor torque rating and speed, and (c) have the same vendor-specified value for RE. For each such group of similar actuators, TU Electric has defined a "Performance Factor" (PF) which has been applied to each actuator in the similarity group. As shown in equation (3), the
value of PF is determined as the average of the ratios of the measured stall torque value (AOTQ80stall) divided by design value predicted by equation (2):

\[ PF = \text{average of all values of the ratio } \left( \frac{\text{AOTQ80stall}}{\text{AOTQ80w}} \right) \]  (3)

The value of PF is greater than 1.00 for all groups of similar actuators tested by TU Electric. Also determined is an uncertainty (ePF) associated with the value of PF. The engineering calculation now predicts the motor's nominal and minimum capabilities at 80% of motor nameplate voltage to deliver torque to the worm gear as follows:

nominal (average) value;

\[ NTQ_{\text{max,w}} = (0.8)^2 \left( \frac{\text{MTQ}}{\text{OAAR}} \right) \left( \frac{\text{AF}}{\text{RE}} \right) \left( \frac{\text{PF}}{} \right) \]  (4)

lower bound value;

\[ TQ_{\text{max,w}} = (0.8)^2 \left( \frac{\text{MTQ}}{\text{OAAR}} \right) \left( \frac{\text{AF}}{\text{RE}} \right) \left( \frac{\text{PF}}{} \right) (1 - \text{ePF}) \]  (5)

Of course, if elevated room temperature causes the motor's capability to be decreased, an appropriately decreased motor torque rating is used instead of the nameplate torque rating.

Only one group of similar actuators has been observed to frequently produce measured actuator output stall torques (at 80% voltage) less than the value predicted by equation (2). For this group, the value of PF is about 1.10. The least value of the ratio of measured to calculated capability is 0.91. As shown in Table 1, an uncertainty ePF of 16% is determined based on the test data.

Table 2 summarizes the results for several groups of similar actuators as determined by TU Electric engineering. The column heading "PF(1 - ePF)" is the ratio of (a) the minimum expected motor torque capability based on testing divided by (b) the motor torque capability predicted equation (2). In general there is very good agreement with equation (2). However, for the 15 ft.lb 3400 rpm motor on a Size 00 actuator, equation (2) appears to overestimate the motor capability.

The Table 2 column heading "PF(1 - ePF(RE)" is a modified running efficiency value based on TU Electric test data analysis. Table 2 compares this product to the design running efficiency specified by the actuator manufacturer. It may be that gear train efficiencies are not the correct explanation for the differences observed. Differences in motor characteristics may be more of a factor. This supposition is based on the data for the Size 00 and Size 0 actuators.
Actuator Effective Moment Arm Lengths

TU Electric also extracted for analysis the following torque test stand data: torque spring pack deflections and corresponding actuator output torques. The forces applied to the torque spring pack to produce the deflections were then obtained by reviewing the deflection versus force data collected during compression testing of the torque spring packs. Correspondencies were thereby established between torque spring pack forces and actuator output torques produced on the torque test stand.

The torque values were divided by their corresponding torque spring pack force values to obtain the unique "effective moment arm" length for each set of data. The average effective moment arm length (MARMw) was then determined. Friction loads within the actuator cause the actual relationships (the effective moment arm lengths) between actuator output torques and torque spring pack forces to differ slightly from the magnitude of the worm gear pitch radius.

The products of the individual torque spring pack force (SPF) values and the average effective moment arm MARMw were next calculated. Differences between these calculated torques and the measured torques were then used to determine an uncertainty (eTQw). All measured torques are within the range provided by equation (6):

\[ AOTQw = (SPF)(MARMw) +/- eTQw \]  

(6)

It is assumed that torques measured on a torque test stand are appropriate for comparison to the actuator manufacturer's specified actuator gear train torque ratings (TQrt). TU Electric uses the uncertainty eTQw and the value of MARMw to convert the actuator torque rating to a corresponding maximum allowable torque spring pack force calculated by equation (7):

\[ SPFrt = (TQrt / MARMw)(1 - (eTQw / TQrt)) \]  

(7)

Motor capability is similarly converted to a maximum allowable torque spring pack force (SPFmax). The result is reduced by the SRSS combination of the uncertainties, as shown by equation (8):

\[ SPFmax = (NTQmax,w / MARMw)(1 - [(ePF)^2 + (eTQw / NTQmax,w)^2]^{1/2}) \]  

(8)
Effects of Stem Thrust on Actuator Output Torque

Following actuator refurbishment and torque stand testing, the actuators were reinstalled on the valves in the power plant. TU Electric engineering had made the assumption early in the TU Electric MOV program that thrust loads on the drive sleeve of the actuator negligibly affect the relationship between torque spring pack force and actuator output torque determined from torque stand testing. TU Electric analyzed test data collected during in situ Static and DP condition testing to verify this assumption.

TU Electric was surprised that some test data indicated a significant loss of actuator output torque when the actuator drive sleeve was subjected to thrust loading. The actuator manufacturer indicated they did not expect this result either. This has been observed by TU Electric for actuators sizes 00, 0, 1, and 2. The author believes if test data were available for other actuator sizes, this "Stem Thrust Effect" would be noticed in that data as well.

For a small sample of MOVs, TU Electric also examined the relationships between motor three phase power measurements and actuator output torques for both torque stand testing and in situ testing. The conclusion was reached that both the spring pack data and the power data indicated the same thing regarding actuator output torques: thrust loads on the drive sleeve tend to reduce the actuator output torque for a given value of torque spring pack deflection (such as a torque switch setting, or at the motor stall condition). The data indicated that this loss did not occur until a substantial amount of thrust was developed, but well within the normal operating load range of the actuator.

The effective moment arm lengths by which torque spring pack force values may be multiplied to obtain the torques delivered by a threaded stem nut to a threaded valve stem were then determined using data at stem thrust magnitudes appropriate for the successful operation of Comanche Peak MOVs. The method of analysis was the same as described earlier for the analysis of torque test stand data, except that the torque measurements were obtained using stems with strain gages installed in a full bridge and calibrated. Since the in situ actuator output torque is less than the torque stand actuator output torque for a given value of torque spring pack force, the length of the effective moment arm (MARM) is less than the value of MARMw. In the same manner as described earlier for torque test stand data, an uncertainty (eTQ) was determined for in situ actuator output torques calculated using MARM. All measured stem torques are within the range provided by equation (9):
AOTQ = (SPF)(MARM) +/- eTQ

Just as actuator torque limits were converted to maximum allowable torque spring pack force values, the stem torque limits (TQSEQ) obtained from the seismic qualification documents are converted by TU Electric to equivalent maximum allowable torque spring pack forces as shown in equation (10):

SPFSEQ = (TQSEQ/MARM)(1 - (eTQ/TQSEQ))

Typical of factors affected by friction coefficients, manufacturing tolerances, assembly practices, component wear, and measurement error, there is substantial scatter in the test data collected to determine the average "effective worm gear moment arm length" (MARMw) which relates torque spring pack force to the approximate torque (AOTQw) on the worm gear, and the average "effective drive sleeve moment arm length" (MARM) which relates torque spring pack force to the torque (AOTQ) applied by the stem nut to the stem. The degree of dispersion of the test data about these average effective moment arm lengths is accounted for by the uncertainties eTQw and eTQ.

Table 3 provides values for both MARM and MARMw as determined by TU Electric for actuator sizes 000, 00, 0, 1, and 2. The approximate uncertainty of a torque calculated as the product of torque spring pack force and one of these average "effective moment arm lengths" is in the column labeled "approx. eTQ".

Based on analysis of unseating test data, during unwedging of the valve disk from the valve seat the average values of the effective moment arm lengths appear to be about 10% less than the average values at other points of interest in the closing and opening strokes. Thus, TU Electric has determined different values for the average effective moment arm lengths and the associated uncertainties for use in evaluating loads during unwedging of the disk from the valve seat.

**Stem Thrust Effect**

At the February 1994 MOV Users Group meeting the actuator manufacturer's representative stated that the manufacturer has not performed testing to verify that thrust loads do not reduce the available torque. TU Electric has introduced a "Stem Thrust Effect" factor into its engineering
calculation to account for the average magnitude of this effect, as well as an associated uncertainty value based on statistical analysis of the test data.

The author has postulated that when thrust loads on the drive sleeve are great enough, deflections of components within the actuator drive sleeve become great enough so that parts rub against each other, although these parts do not rub against each other at lesser thrust loads. The deflections involved may occur within one part such as the gasket at the housing-to-housing cover joint, or within a subassembly of parts, such as a drive sleeve bearing.

The industry has been made aware that excessive torque on the actuator cover bolts may compress the actuator cover gasket too much and place the upper and lower drive sleeve bearings under excessive axial load, with a resultant reduction in torque delivered to the stem nut at torque switch trip (or at motor stall). The application of a thrust load to the actuator drive sleeve by the stem (through the stem nut) is conceptually very similar to the application of load by the cover bolts. The difference is that the load imparted by the stem will increase the compressive load on one bearing while decreasing the load on the other bearing.

Another possibility suggested to the author by Tim Cline of Duke Power is that movement of the actuator drive sleeve along the stem axis may result in a less efficient interface between the worm and the worm gear. Tim Cline postulated that a lesser efficiency may result from the mid-plane of the worm gear moving out of the plane in which the worm axis resides.

Much testing of actuators has been conducted using torque test stands: by the actuator manufacturer, by contractors to the actuator manufacturer, by test equipment vendors, and by utilities. If this torque stand test data has been used to determine the relationship between torque spring pack force and actuator output torque, the users of this data must recognize that the relationship may be applicable only if the actuator is used in a load condition similar to that on the torque test stand. The relationship may be valid for determining the torque at the worm gear for verification that the motor capability and the actuator torque rating are not exceeded. However, the relationship may overestimate the torque which is delivered to the threaded stem nut for producing stem thrust.

Using test data collected for SB-1 actuators during torque stand and in situ testing, Table 4 demonstrates the method used to determine the average magnitude of the "Stem Thrust Effect" (ST) and the associated uncertainty (eST). Table 4 shows the values of TQsg and TQtt. Each
measured stem torque is converted into a value comparable to torque test stand data by multiplying TQsg by the ratio (MARMw/MARM). The measured actuator output torque (TQts) on the torque test stand is divided by the calculated torque. Since the average of the ratios is not equal to 1.00, it is clear that a better correlation between actuator output torques with and without thrust loading on the actuator drive sleeve may be obtained by multiplying TQsg by the product (AVE)(MARMw/MARM). It is this product which is called the average "Stem Thrust Effect" (ST). The associated uncertainty (eST) is calculated in this case to bound the maximum data point, which is further from the average value than two sample standard deviations. (For other actuator sizes, two sample standard deviations bound all test data.)

Since two different tests were conducted to obtain the torque stand and the in situ data for each actuator, the influence of actuator output torque repeatability (REPts) at a given value of torque spring pack deflection must be considered. The average magnitude of the "Stem Thrust Effect" (ST) is minimally affected by REPts because averaging the data tends to "average out" the effect of REPts. The spread of data about the average is affected by REPts. The values determined by TU Electric for uncertainties eST therefore include the influence of REPts. TU Electric has not performed testing to quantify actuator output torque repeatability versus torque spring pack deflection, and therefore has not reduced the magnitude of each eST value by the effect of REPts. This measure of conservatism could be removed.

Table 5 presents for several actuator types and sizes the average magnitude of the "Stem Thrust Effect" (ST) and the associated uncertainty (eST) as determined by TU Electric.

Net Impact of the Performance Factor and the Stem Thrust Effect

The net effect of (a) the extra (for most actuators) capability of the motor to deliver torque to the worm gear that the "Performance Factors" justify, and (b) the reduced capability of the motor to deliver torque to the stem nut that the "Stem Thrust Effects" demonstrate, are now discussed. The ratio (PF/ST) shown in Table 6a is the average combined effect. This average is then further reduced by the SRSS combination of uncertainties eST and ePF to obtain "NET", the lower bound of the combined effect determined by equation (11):

\[
NET = \left( \frac{PF}{ST} \right)(1 - \left[ \left( \frac{ePF}{ST} \right)^2 + (eST)^2 \right]^{1/2})
\]

(11)
The value of NET is the coefficient by which the motor capability to deliver torque to the stem of a rising (non-rotating) stem valve as determined by equation (2) must be multiplied to obtain a stem torque (TQmax) value which accounts for the Stem Thrust Effect's reduction of torque delivered to the threaded stem and the Performance Factor's increase (for most actuator configurations) of predicted motor capability to deliver torque when there is no thrust load on the drive sleeve. The value of TQmax is determined as shown in equation (12):

\[
TQ_{\text{max}} = (0.8)^2 \left( \frac{MTQ}{OAR} \right) \left( \frac{AF}{(RE)(NET)} \right) \tag{12a}
\]

\[
TQ_{\text{max}} = (0.8)^2 (MTQ)(OAR)(AF)(RE)(PF/ST)(1 - \left[ (ePF)^2 + (eST)^2 \right]^{1/2}) \tag{12b}
\]

As Table 6b shows, the net effect would be bounded if the actuator manufacturer's "Pullout Efficiency" (POE) were used instead of the "Running Efficiency" (RE) for certain actuator configurations. For other configurations, a value greater than the "Pullout Efficiency" is justifiable. In other cases, even the use of the actuator manufacturer's "Pullout Efficiency" does not appear to provide a lower bound prediction of the motor's capability to deliver torque to the valve stem when the motor is powered by 80% of the nameplate voltage.

While the actuator manufacturer is continuing to use the "Pullout Efficiency" for sizing actuators for new orders, the manufacturer has stated that utilities may use the "Running Efficiency" when evaluating the capabilities of actuators already installed in power plants. It is the author's opinion that TU Electric test data indicates this allowance for using "Running Efficiencies" may need to be further reviewed as it may not in all cases provide appropriate levels of assurance that safety related MOVs would fulfill their design basis functions. In fact, since two configurations in Table 6b show that even the use of the "Pullout Efficiency" may not always suffice, it may be that other reviews are also needed.

Further Investigation of the Stem Thrust Effect

It would probably be worthwhile for others to perform tests which would either confirm or refine the results TU Electric has obtained. An improved technique for performing reliable tests would be to test the actuator on a test stand which has the capability of not only providing a resisting torque to cause the motor to stall, but to also simultaneously apply thrust loads which can be held.
constant or varied as the torque load increases. Testing on such a fixture may justify reducing the conservatism in the magnitudes of the "Stem Thrust Effects" determined by TU Electric, especially if the effects of actuator output repeatability are removed from the analysis results.

Tests of this sort could be used in conjunction with inspections and analyses of actuator drive sleeve parts to determine what, if any, parts begin to adversely interact with each other when high thrust loads are imposed, and the magnitudes of the thrust loads which initiate this interaction. Investigation could also include identifying actuator modifications that might eliminate or reduce the magnitude of the "Stem Thrust Effect" and whether such a modification would be economically feasible.

Running Thrusts

Differences between the running thrusts of nominally identical MOVs with nominally identical packing configurations and packing follower bolt torques are typically significantly greater than the differences between open and close running thrust values for these same MOVs. Thus, Static condition closing and opening running thrust test data have been combined to determine an average running thrust (RT) value assumed for both directions of stem travel in the engineering calculation.

An uncertainty (eRT) associated with the average running thrust value is based on the greater of (a) the maximum test result, and (b) the average value plus 2 standard deviations of the data. Because running thrust magnitudes are typically small, the test instrument measurement uncertainty is typically high. This may contribute to the degree of scatter of the test data. The large uncertainty is tolerable since running thrusts are typically a small part of the MOV thrust requirements.

Running thrusts are primarily due to the packing friction force, which may be calculated by a packing manufacturer's equation, but are also affected by the friction loads developed due to the torque load on the stem being reacted throughout the stem travel. For MOVs at TU Electric's Comanche Peak plant, there were several MOVs with measured running thrusts greater than the packing manufacturer's calculated packing friction loads.
The results of analyzing the test data are presented in Table 7. There are two sets of results for stems of 1.00 inch diameter because the difference in the packing configuration and preload, and perhaps the valves too, made a very definite difference in the actual running thrusts.

**Valve Factors for Westinghouse and Borg-Warner Gate Valves**

The "standard" equation for the maximum stem thrust required to move a gate valve disk (gate) to the valve seat under DP conditions may be solved for the valve factor. Using test data, the actual closing stroke valve factor (VFct) may be determined as shown in equation (13):

\[
VFct = \frac{[T_d - RT_{s,c} - (Ap)(Puct)]}{[(Ao)(DPT_c)]}
\]  

(13)

where

- \(Ao\) = nominal seat orifice area based on a selected diameter, Do
- \(DPT_c\) = maximum differential pressure measured during the closing stroke of the DP test
- \(Ap\) = area of the stem cross-section at the packing
- \(Puct\) = maximum upstream pressure measured during the closing stroke of the DP test
- \(RT_{s,c}\) = average Static test running thrust over the last 10% or so of the closing stroke
- \(T_d\) = maximum thrust required at any point during the closing stroke of the DP test until the disk begins to wedge into the valve body seat
- \(Do\) = For Westinghouse valves, the seat ring inside design diameter specified by the manufacturer, plus 1/16 inch. For Borg-Warner valves, the average of the valve body seating surface's inside and outside design diameters specified by the manufacturer.

The "standard" equation for the maximum stem thrust required to move a gate out of the valve seat (after initial unwedging of the disk) under DP conditions may be solved for the valve factor. Using test data, the actual opening stroke valve factor (VFot) may be determined as shown in equation (14):

\[
VFot = \frac{[T_{D,O} - RT_{c,o} + (Ap)(Poot)]}{[(Ao)(DPT_o)]}
\]  

(14)

where \(Ao\) and \(Ap\) are as defined earlier, and

- \(DPT_o\) = maximum differential pressure measured during the opening stroke of the DP test
- \(Poot\) = maximum upstream pressure measured during the opening stroke of the DP test
RTs,o = average Static test running thrust over the first 10% or so of the opening stroke
TDPo = maximum thrust required at any point during the opening stroke of the DP test after
initial unwedging of the disk (overcoming the static friction coefficient in the valve seat)

Table 8a justifies the grouping of similar valves. Table 8b identifies the number of MOVs in each
group of similar valves, the number of MOVs in each group that were DP tested in each of
Comanche Peak Units 1 and 2, and the number of MOVs in each group that were tested with stem
strain gage measurements. Table 8c provides a summary of closing stroke valve factor test
results: the average value (VFc), the maximum value (Max VFct), and an uncertainty (eVFc)
associated with the average value which is based on the greater of (a) the average value plus 2
sample standard deviations, and (b) the maximum VFct value. Table 8d provides a summary of
opening stroke valve factor test results: the average value (VFo), the maximum value (Max VFot),
and an uncertainty (eVFo) associated with the average value which is based on the greater of (a)
the average value plus 2 sample standard deviations, and (b) the maximum VFot value. In all cases
for both closing and opening, the sum of the average value plus 2 standard deviations was greater
than the maximum valve factor derived from test data. Tables 8c and 8d compare the valve
vendor’s original valve factor values to the maximum test results.

For each valve in the similarity group, TU Electric now calculates by equation (15) the maximum
thrust (Tseat,d) which may be required to close the valve under DP conditions and initiate wedging
of the disk into the valve body seat:

\[ T_{\text{seat},d} = (RT)(1 + eRT) + (Ap)(Pupc) + (Ao)(DPRc)(VFc)(1 + eVFc) \] (15)

where RT, eRT, Ap, and Ao are as defined earlier, and

\[ eVFc \] = uncertainty associated with the value of VFc
\[ DPRc \] = closing stroke maximum design basis (required) differential pressure
\[ Pupc \] = closing stroke maximum design basis upstream pressure accompanying DPRc
\[ VFc \] = the average valve factor in the closing stroke for a group of nominally identical
valves

For each valve in the similarity group, TU Electric now calculates by equation (16) the maximum
thrust (Tod) which may be required to open the valve under DP conditions after initial unwedging of
the disk (overcoming the static friction coefficient in the valve seat):
\[ T_{od} = (RT)(1 + eRT) \cdot (Ap)(Pup0) + (Ao)(DPr0)(VFo)(1 + eVFo) \]  

(16)

where RT, eRT, Ap, and Ao are as defined earlier, and

\[ eVFo \quad = \text{uncertainty associated with the value of VFo} \]
\[ DPr0 \quad = \text{opening stroke maximum design basis (required) differential pressure} \]
\[ Pup0 \quad = \text{opening stroke maximum design basis upstream pressure accompanying DPr0} \]
\[ VFo \quad = \text{the average valve factor in the opening stroke for a group of nominally identical valves after initial unwedging of the disk (overcoming the static friction coefficient of the valve seat)} \]

"Rate of Loading Effect" Magnitudes for Torque Switch Controlled Closure MOVs

Actuator output torque at torque switch trip (AOTQtst) is the same within actuator repeatability (REPts), under both Static and DP conditions. However, under Static conditions the stem thrust at torque switch trip is sometimes noticeably greater than under DP conditions. Initially called the "rate-of-loading effect" (ROL), it is more correctly an effect primarily of varying stem friction coefficients from stroke to stroke. A motor operated valve that is susceptible to ROL effects may exhibit significantly different ROL magnitudes during different strokes.

TU Electric assumes that even if only one valve in a group manifests susceptibility to the ROL effect during testing, then any other MOV in the group may manifest a ROL effect during the design basis condition which may occur at any time in the remaining plant operating life. Also, TU Electric does not necessarily assume that the maximum possible ROL effect was demonstrated during testing. TU Electric uses statistical analysis to determine the average rate of loading effect (ROL) plus 2 sample standard deviations, and compares this sum to the maximum ROL effect observed during testing. TU Electric assumes the greater of these two values is the maximum possible ROL effect. An uncertainty (eROL) is determined based on the greater of the two values.

The SRSS combination of eROL with other uncertainties diminishes the impact of this effect on the minimum required stem thrust at close torque switch trip. TU Electric also determines whether the assumed valve factors and their uncertainties have enough conservatism in their values to account for the ROL effect. If so, then no additional ROL "penalty" is imposed. The following decision process is used to determine the values of ROL and eROL:
STEP 1:
Recognize that if the magnitude of the stem factor at close torque switch trip is greater under DP conditions than under Static conditions, then the ratio of the DP condition stem factor (SFct,d) divided by the Static condition stem factor (SFct,s) is greater than 1.00. Define the ratio (RSFt) of these stem factors, equation (17):

$$\text{RSFt} = \left( \frac{\text{SFct,d}}{\text{SFct,s}} \right) = \left( \frac{Tcst,s}{TQcst,s} \right) / \left( \frac{Tcst,d}{TQcst,d} \right) > 1.00 \quad (17)$$

STEP 2:
TU Electric has reviewed ROL effects both at the torque switch trip event and at the maximum thrust load to overcome DP conditions, and has found that the scatter of data is such that using data at the torque switch event is sufficient to approximate the effects at the maximum load to overcome DP conditions. Thus, by using equation (18) the maximum thrust (Tdpt,d) observed during testing to overcome the test DP conditions can be converted into a corresponding minimum required thrust (Tcst,s) at close torque switch trip under Static conditions that will ensure the torque switch will not trip (excluding the effects of actuator output torque repeatability) prior to the valve disk reaching the close valve seat and initiating wedging of the disk into the seat:

$$Tcst,s = (\text{RSFt})(\text{Tdpt,d}) \quad (18)$$

STEP 3:
Along with other test data, use Tcst,s in place of Tdpt,d in the "standard" industry equation for calculating thrust requirements, and solve for the "Static Equivalent Valve Factor" (VFcts). Instead of solving for VFct with equation (13), equation (19) below is solved for VFcts as shown in equation (20) [Note: equations (13) and (19) are applicable to gate valves; the equations for globe valves are slightly different]:

$$Tcst,s = (\text{RSFt})(\text{Tdpt,d}) = (\text{VFcts})(\text{DPTc})(A_o) + (P_{upc})(A_p) + RTs,c \quad (19)$$

$$VFcts = \left[ \frac{Tdpt,s - RTs,c - (A_p)(P_{upc})}{(A_o)(DPTc)} \right] \quad (20)$$

where

VFcts = the "Static Equivalent Valve Factor" which accounts for the Rate of Loading effect observed for a particular MOV during Static and DP testing
As is clear from equation (19), the use of VFcts in the "standard" industry equation for predicting thrust requirements ensures the calculated minimum required Static condition thrust at close torque switch trip is great enough so that the close torque switch will not trip at a thrust of magnitude equal to Tdpt,d under the tested DP conditions (excluding the influence of actuator output torque repeatability at torque switch trip).

STEP 4:
For each group of similar MOVs, determine the average VFcts value (AVE VFcts) and the sample standard deviation. Determine the maximum potential VFcts value (MAX VFcts) as the greater of (a) the maximum test data value for VFcts, and (b) the sum of AVE VFcts plus 2 standard deviations.

STEP 5:
Compare the value of MAX VFcts to the maximum postulated valve factor ((VFc)(1 + eVFc)) for the group of similar MOVs. If the uncertainty by which VFc is increased is sufficiently great, then there is need to further increase the minimum required thrust prediction for "Rate of Loading" effects. The determination of ROL and eROL is as follows:

A. IF \(( VFc)(1 + eVFc) > \) MAX VFcts \( \) THEN \( ROL = 1.000 \) and \( eROL = 0.000 \) \( (21) \)

B. OTHERWISE \( ROL = \) the greater of 1.00 and the ratio (AVE VFcts / VFc) \( (22) \)

where

\( ROL \) = a factor by which the average valve factor (VFc) determined for the group of MOVs tested under DP conditions is multiplied to obtain the average "Static Equivalent Valve Factor". In no case is the value of ROL less than 1.00.

This takes care of the average Rate of Loading effect for the group of valves, but the maximum potential Rate of Loading effect must still be considered. This is addressed by the uncertainty \( eROL \), the value of which is determined as follows:

C. IF \(( VFc)(1 + eVFc)(ROL) > \) MAX VFcts \( \) THEN \( eROL = 0.000 \) \( (23) \)
D. OTHERWISE
\[
e\text{eROL} = \left( \left\{ \frac{\text{MAX VFcts}}{(\text{VFc})(\text{ROL})} \right\} - 1 \right)^2 - (\text{eVFc})^2 \right)^{1/2} \tag{24}
\]

Thus, the Rate of Loading effect is accounted for by adjustment of the valve factor which affects the predicted minimum required thrust. The adjusted valve factor (VFrol) is not less than MAX VFcts:
\[
\text{VFrol} = (\text{VFc})(\text{ROL})(1 + \left\{ \text{eVFc}^2 + \text{eROL}^2 \right\}^{1/2}) \tag{25}
\]

Tables 9a and 9b provide the results of this analysis.

Since the author is not aware of any other organization which has addressed the Rate of Loading effect in this manner, Table 10 provides Rate of Loading results in the terms with which most are familiar (assuming the "Rate of Loading" factor, ROLt, for a Static and DP tested MOV equals RSFl). Table 10 includes results for globe valves as well, whereas Tables 9a and 9b include only gate valve results.

Equation (26) below shows the TU Electric calculation for the minimum stem thrust (Treq,s) that is required under Static test conditions at close torque switch trip (excluding the effects of torque switch repeatability, margin for stem factor degradation, and test instrumentation measurement uncertainty) to ensure that sufficient thrust will be developed at close torque switch trip under maximum design basis DP conditions:
\[
T_{\text{seat},s} = (RT)(1 + eRT) + (Ap)(Ppc) + (Ao)(DPrco)(VFc)(ROL)(1 + [(\text{eVFc})^2 + (\text{eROL})^2]^{1/2}) \tag{26}
\]

To this minimum required stem thrust an additional load is added for selected MOVs to ensure that sufficient thrust is developed prior to close torque switch trip to not only reach the valve seat under maximum design basis DP conditions, but to also press the disk seat surface into the body seat surface with sufficient force to effect a pressure seal.
"Rate of Loading Effect" Magnitudes for Limit Switch Controlled Closure MOVs

Limit switch controlled closure is used only with some of the actuators having compensator spring packs. In these cases, the thrust at control switch trip occurs at a specific number of turns of the limit switch rotor. The number of turns at which the close limit switch is set ensures the disk is wedged into the valve seat, even under DP conditions, with the compensator spring pack compressed a specific amount. Since the compensator spring pack deflects the same amount under both Static and DP conditions, the stem thrust at close limit switch is the same under both conditions:

\[ T_{cst,d} = T_{cst,s} \]  

where

\( T_{cst,d} \) = stem thrust at close limit switch trip under DP test conditions

\( T_{cst,s} \) = stem thrust at close limit switch trip under Static test conditions

The Stem Factor (SF) may be different under the two conditions, just as for torque switch controlled closure MOVs, with the DP condition having the greater friction coefficient. Consequently, the stem torque at close limit switch trip may be greater under the DP condition:

\[ T_{Qcst,d} > T_{Qcst,s} \]  

where

\( T_{Qcst,d} \) = stem torque at close limit switch trip under DP test conditions

\( T_{Qcst,s} \) = stem torque at close limit switch trip under Static test conditions

Thus, from this point of view, the "Rate of Loading" effect for limit switch controlled closure MOVs is the same as for torque switch controlled. For limit switch controlled closure MOVs TU Electric uses the familiar ratio of stem factors to determine from Static and DP test data the "Rate of Loading" effect factor (ROLt):

\[ ROLt = RSF_t > 1.00 \]  

\[ ROLt = \left( \frac{SF_{cst,d}}{SF_{cst,s}} \right) = \left( \frac{T_{cst,s}}{T_{Qcst,s}} \right) / \left( \frac{T_{cst,d}}{T_{Qcst,d}} \right) \]  

Since \( T_{cst,s} = T_{cst,d} \) the above can be simplified to:
\[ ROL_t = \frac{TQ_{cst,d}}{TQ_{cst,s}} > 1.00 \]  

Also, since \( T_{cst,d} = T_{cst,s} \) the Static test minimum required stem thrust at control switch trip given by equation (26) is equal to the value calculated for the maximum design basis DP condition by equation (15). This is accomplished for limit switch controlled closure MOVs by using the following values for ROL and eROL in equation (26):

\[
\begin{align*}
ROL &= 1.00 \\
eROL &= 0.00
\end{align*}
\]

The Rate of Loading effect for limit switch controlled closure MOVs must be accounted for when verifying with a Static test that (1) the motor's minimum capability is not exceeded by the torque at close limit switch trip \( (TQ_{cst,s}) \), and (2) the structural torque rating of the actuator and the qualified operating torque of the valve structure are not exceeded by the total torque \( (TQ_{TOT}) \) developed after limit switch trip due to (a) the time it takes the electrical control system to remove power from the motor, and (b) the kinetic energy of the moving parts, primarily the motor rotor, which continue to drive the disk into the valve seat until all parts stop moving. This verification could be accomplished by increasing the measured values of \( TQ_{cst,s} \) and \( TQ_{TOT} \) by the applicable Rate of Loading factor, and verifying that the increased torques do not exceed their limits. This approach requires manipulation of the test data following the test in order to know if the test result is acceptable, and this lengthens the time required to complete a test and verify the results are acceptable. The approach TU Electric uses is to multiply the limiting torque values by the inverse of the typical Rate of Loading factor prior to performing the test, and to then directly compare the test results against the reduced limits.

The inverse of \( ROL_t \) is the factor \( DTQt \):

\[ DTQt = \frac{1}{ROL_t} = \frac{TQ_{cst,s}}{TQ_{cst,d}} < 1.00 \]  

TU Electric has statistically analyzed the \( DTQt \) test data for 24 MOVs to determine the average value of the "Rate of Loading" effect \( (DTQ) \) for limit switch controlled MOVs, and an uncertainty \( (eDTQ) \) associated with the value of \( DTQ \). The uncertainty \( eDTQ \) is combined by the SRSS technique with other uncertainties in the test procedure, including the repeatability of thrust at close limit switch trip (REPs). The total uncertainty is used along with the value of \( DTQ \) to reduce
the maximum allowable torque values for comparison with the measured values of TQcst,s and TQcst,Ts.

Table 11 presents test data analysis results for DTQ and eDTQ. Since both Static and DP tests are required to obtain test data, when determining the value of eDTQ, the effects of actuator output thrust repeatability at close limit switch trip (+/- 3%) have been analytically extracted.

**Disk Position Effect**

Some limit switch closed MOVs exhibit another phenomenon under DP conditions that is unrelated to the change in friction coefficients. Under DP conditions the disk's final position in the valve seat may be with a different orientation than under Static conditions. While the author has not determined precisely what is occurring, the author has postulated from the observed effects that the stem's final position in the valve under DP conditions is slightly further out of the valve seat (a distance "dl" on the order of magnitude of 0.025 to 0.055 inches) than it is under Static conditions. It may be that a change of dimensions of the valve internals could prevent this from occurring.

With the close limit switch setting unchanged, the stem nut's final position relative to the stem is the same. This means that the stem nut must compress the compensator spring pack of the actuator an additional distance "dl". The additional compression of the compensator spring pack under DP conditions results in a greater stem thrust at close limit switch trip under DP conditions (Tcst,d) than under Static conditions (Tcst,s). In turn, the greater Tcst,d value requires a greater torque delivered to the stem nut at close limit switch trip under DP conditions (TQcst,d) than under Static conditions (TQcst,s).

Thus, for some limit switch controlled closure MOVs, not only is there a "Rate of Loading" effect to consider when setting (or verifying the acceptability of the setting) the close limit switch under Static conditions, there is a "Disk Position Effect." While caused by a completely different phenomenon than the "Rate of Loading" effect, the result is similar and is treated similarly. The ratio (DTd) of Tcst,s divided by Tcst,d is the factor by which the thrust limits must be multiplied to obtain reduced thrust limits used during a Static test to verify that the thrust limits will not be exceeded during a stroke under DP conditions. The ratio (DTQd) of TQcst,s divided by TQcst,d is the factor by which the torque limits must be multiplied to obtain reduced torque limits used during
a Static test to verify that the torque limit will not be exceeded during a stroke under DP conditions.

Although TU Electric data analysis has not separated the individual components of the two phenomenon, the following example which uses data from a group GT1 MOV shows the two effects are additive:

\[ TQ_{cst,d} = \left( \frac{T_{cst,d}}{T_{cst,s}} \right) \left( \frac{S_{Fcst,d}}{S_{Fcst,s}} \right) \left( TQ_{cst,s} \right) \]

167 ft.lb = \{(1.1105)\(1.1717\)\(128.0\)

167 ft.lb = 167 ft.lb

Based on analysis of DTt and DTQt test data for individual MOVs, the average magnitudes of the "Disk Position" and "Rate of Loading" combined effects are "DT" for the thrust limit reduction factor and "DTQ" for the torque limit reduction factor. Associated with DT and DTQ are uncertainties "eDT" and "eDTQ", respectively. The values of eDT and eDTQ are determined with recognition that TU Electric is conservatively assuming REPs = 3% and that independent uncertainties are combined by the SRSS method. As shown in Table 11, there were two Comanche Peak valve groups, GT1 and GT16, which showed sensitivity to the "Disk Position" effect.

Equation (33) shows how the Rate of Loading effect is accounted for in the engineering calculation's adjustment of the motor capability at 80% voltage in the closing stroke of a limit switch controlled closure MOV. The uncertainty shown in equation (12b) is further combined with the uncertainty eDTQ, while the average Rate of Loading effect, DTQ, is directly applied to the calculated motor capability:

\[ TQ_{\text{max},c} = (0.8)^2 \left( \frac{MTQ}{OAR} \right) \left( \frac{AF}{RE} \right) \left( \frac{PF}{ST} \right) \left( DTQ \right) \left( 1 - [ePF^2 + eST^2 + eDTQ^2]^{1/2} \right) \]  \hspace{1cm} (33)

Equation (34) shows how the Rate of Loading effect for limit switch controlled closure MOVs is accounted for in the engineering calculation's adjustment of the thrust load limit which is compared in the test procedure to the total thrust developed after close limit switch trip:

\[ TT_{TOT \text{max},c} = ( T \text{limit }) ( DT ) ( 1 - eDT ) \]  \hspace{1cm} (34)
For torque switch controlled closure MOVs, the values of DTQ and eDTQ are such that there is no reduction of the total thrust limit:

\[
DTQ = 1.00, \text{ and } eDTQ = 0.00
\]

**Stem Friction Coefficient Range**

Stem friction coefficients have been calculated using measured stem thrust and stem torque data at the following points of interest in the closing and opening strokes for a total of over 700 data points which were then statistically analyzed to determine average values and standard deviations:

- Peak thrust to overcome DP condition prior to wedging
- Thrust at close control switch trip
- Peak thrust following close control switch trip
- Peak thrust to initiate unwedging of the disk
- Peak thrust to overcome DP condition after unwedging the disk

There appears to be a trend of lesser stem thread friction coefficients during initial unwedging of the disk from the valve seat than at other times in the closing and opening strokes. There was no indication that closing stroke friction coefficients were in general any different from opening stroke friction coefficients at other points of interest. Some MOV groups appear to have less variation in stem friction coefficients than others. The reason for this has not been determined.

Table 12 provides the results of the analysis for all data points taken together. TU Electric uses similar stem friction coefficient analysis results to calculate six corresponding stem factors. For the average unseating stem factor, associated uncertainties are determined by which the average stem factor value must be increased and decreased to obtain the maximum and the minimum stem factors. For the average closing and opening stroke stem factor, associated uncertainties are determined by which the average stem factor value must be increased and decreased to obtain the maximum and the minimum stem factors. Any conversion between stem thrust and stem torque uses the average stem factor value and combines the stem factor uncertainty with other uncertainties by the SRSS technique.

TU Electric has assumed that any one MOV out of a group of similar MOVs can have stem friction coefficients anywhere within the range determined for that group at some point in the operating life.
of the plant. This risk is accounted for in the engineering calculation by combining the stem factor uncertainties with other uncertainties by the SRSS technique.

Tim Cline of Duke Power has suggested that for some MOVs the outer edge of the stem thread may rub on the stem nut thread in a way which develops radial loads at the corner of the thread and the major diameter. Radial loads would be accompanied by tangential friction loads that consume torque. Less torque would therefor be available for producing thrust. The standard ACME thread equation for relating torque to thrust does not account for this potential source of inefficiency. If the outer edge of the stem thread were more rounded, it may be that the stem thread efficiency could be increased, with the result that stem factor values would be reduced.

Equation (35) shows the calculation of the minimum stem torque required at close torque switch or limit switch trip (excluding any extra torque required for producing a pressure seal, margin for stem factor degradation, control switch repeatability effects, and test instrument measurement uncertainty):

\[
T_{Q_{\text{seat}}} = RTQs + T_{Q_{\text{req,c}}} + T_{Q_{\text{dpr,c}}}
\]  

(35)

where

\[
RTQs = ( SF )( RT )( 1 + [ ( eRT )^2 + ( eSFu )^2 ]^{1/2}
\]

(36)

\[
T_{Q_{\text{req,c}}} = ( SF )( Ap ) ( Pucp )( 1 + eSFu )
\]

(37)

\[
T_{Q_{\text{dpr,c}}} = ( SF )( Ap )( DPRc )( VFc )( 1 + [ ( eVFc )^2 + ( eSFu )^2 ]^{1/2}
\]

(38)

SF = the stem factor based on the average stem thread friction coefficient

\[
eSFu = ( SFu - SF ) / SF
\]

(39)

SFu = the stem factor for the maximum postulated stem thread friction coefficient
APPENDIX B : ACRONYMS

AF  application factor, specified by the actuator manufacturer, equal to 0.9.

Ao  valve orifice area based on the mean diameter of the valve seat ring as specified by the valve manufacturer.

AOT  Actuator Output Thrust, produced at the interface of the actuator’s threaded stem nut and the threaded stem.

AOTQ  Actuator Output TorQue applied by the actuator’s threaded stem nut to the valve stem.

AOTQ80stall  measured output torque of the actuator when the motor stalls during torque stand testing with power to the motor at 80% of the motor nameplate voltage.

AOTQ80w  the "standard" predicted capability of the motor to deliver torque to the worm gear when the motor is powered by 80% of the motor nameplate voltage and the resisting drive sleeve torque is gradually applied over several seconds:

\[ \text{MTQ80} \times \text{OAR} \times \text{AF} \times \text{RE} \]

AOTQtst  actuator output torque at torque switch trip.

AOTQw  Actuator Output TorQue with no thrust load applied to the actuator drive sleeve, such as occurs during testing on a torque test stand. Also, the input torque to an "HBC" gearbox. Approximately the torque applied by the worm to the worm gear.

Ao  nominal seat orifice area based on the mean seat diameter specified by the valve vendor.

Ap  stem cross-section area at the packing.

AVE  average of a set of values.

AVE VFcts  the average VFcts value.

54

206
CTQun,ds a coefficient (not less than 1.00) based on Static and DP test data which when multiplied by the measured Static test unseating torque (TQun,s) provides a maximum predicted unseating torque under the more severe of the Static and the DP conditions.

CTun,ds a coefficient (not less than 1.00) based on Static and DP test data which when multiplied by the measured Static test unseating thrust (Tun,s) provides a maximum predicted unseating thrust under the more severe of the Static and the DP conditions.

Cwdg the "wedging coefficient", based on analysis of test data under both Static and DP conditions, by which the total closing stroke thrust can be multiplied to obtain an estimate of the thrust required to unseat the valve under the more severe of the Static and the DP conditions.

DPRc closing stroke maximum design basis (required) differential pressure.

DPRO opening stroke maximum design basis (required) differential pressure.

DPTc maximum differential pressure measured during the closing stroke of the DP test.

DPTo maximum differential pressure measured during the opening stroke of the DP test.

DT the average value of the "Disk Position Effect" (the average of the DTt values) on stem thrust magnitudes at and after close limit switch trip.

DTt as observed from Static and DP testing of a given MOV, the factor by which the thrust limits must be multiplied to obtain reduced thrust limits which may be used during evaluation of Static test results to verify that the thrust limits will not be exceeded during a stroke under DP conditions:  

\[ \left( \frac{T_{cst,s}}{T_{cst,d}} \right) \]

DTQ the average value of the combination effects of the "Rate of Loading" and "Disk Position" phenomena for limit switch controlled MOVs (the average of the DTQt values).
the net effect of the "Rate of Loading" and "Disk Position" phenomena observed from a set of Static and DP tests of a limit switch controlled MOV, being the factor by which the torque limits must be multiplied to obtain decreased torque limits for use during Static testing to verify the torque limits will not be exceeded at or after close limit switch trip under DP conditions, equal to the inverse of ROLt:

\[
\left( \frac{1}{ROLt} \right) = \left( \frac{TQcst_s}{TQcst_d} \right) < 1.00
\]

eDSF uncertainty associated with potential stem factor degradation (10%).

eDT uncertainty associated with the value of DT.

eDTQ uncertainty associated with the value of DTQ.

eMARM uncertainty associated with the value of MARM.

eMARMw uncertainty associated with the value of MARMw.

ePF uncertainty associated with the value of PF.

eROL uncertainty associated with the value of ROL.

eSFu uncertainty associated with the value of SF relative to the value of SFu:

\[
\left( \frac{SFu}{SF} - 1 \right)
\]

eST uncertainty associated with the value of ST.

eVFc uncertainty associated with the value of VFc.

eVFo uncertainty associated with the value of VFo.

GL Generic Letter issued by the NRC.

MARM the "effective drive sleeve moment arm length" which relates SPF to the torque applied by the stem nut to the stem (AOTQ).
MARMw  the average "effective worm gear moment arm length" which relates SPF to the torque on the worm gear (AOTQw).

MAX VFcts  the maximum potential VFcts value, being the greater of (a) the maximum test data value for VFcts, and (b) the sum of AVE VFcts plus 2 standard deviations of the VFcts data.

MOV  Motor Operated Valve.

MTQ  motor torque rating (ft.lb), the motor nameplate "start torque".

MTQ80  AC motor design capability defined by as 64% of the motor torque rating (MTQ) when the motor is supplied with power at 80% of the motor's nameplate voltage: 
\[
\left(0.80\right)^2 \left(\text{MTQ}\right)
\]

MTQstall  measured motor shaft stall torque during dynamometer testing with power supplied at 80% of the motor's nameplate voltage, and where the test was performed on a dynamometer by gradually applying a braking torque to the motor shaft over a period typically somewhere between 10 to 40 seconds.

NA  Not Applicable.

NET  the coefficient by which the motor capability to deliver torque to the stem of a rising non-rotating stem valve, commonly assumed to be the value calculated for AOTQ80w, must be multiplied to obtain a value which accounts for the "Stem Thrust Effect" and the "Performance Factor": 
\[
\left( PF \right) \left(1 - \left[\left(ePF\right)^2 + \left(eST\right)^2\right]^{1/2}\right) / \text{ST}
\]

NRC  United States Nuclear Regulatory Commission.

OAR  overall gear ratio of the actuator from motor to stem nut (or HBC input shaft).
actuator "Performance Factor" defined by TU Electric for a group of similar actuators, equal to the average of the ratios of the measured stall torque value (AOTQ80stall) divided by design value (AOTQ80w) for each actuator in the similarity group.

"Pullout Efficiency" of the actuator, specified by the actuator manufacturer.

closing stroke maximum design basis upstream pressure accompanying DPRc.

opening stroke maximum design basis upstream pressure accompanying DPRo.

maximum upstream pressure measured during the closing stroke of the DP test.

maximum upstream pressure measured during the opening stroke of the DP test.

"Running Efficiency" of the actuator, specified by the actuator manufacturer for each combination of motor rpm, actuator size, and OAR.

repeatability of stem thrust at close limit switch trip where the close limit switch trips only when the valve disk is being wedged into the valve seat and the actuator compensator spring pack is deflecting in response to the stem thrust load (+/- 3% based on testing).

actuator vendor's specified torque switch repeatability (+/- 5%, 10%, or 20%).

Or, a factor equal to the average magnitude of the "Rate of Loading" effect (average of ROLt values) by which the average valve factor (VFc) of an MOV is multiplied to obtain the average "Static Equivalent Valve Factor". In no case is the value of ROL less than 1.00.

the "Rate of Loading" effect observed for a given MOV from a set of DP and Static tests. Often defined as being equal to RSFlt.
RSFt  
the ratio of the DP test stem factor divided by the Static test stem factor at close torque switch trip. It is this factor that is often considered by the industry to be the “Rate of Loading”:
\[
(SF_{cst,d} / SF_{cst,s}) = (T_{cst,s} / T_{Qcst,s}) / (T_{cst,d} / T_{Qcst,d}) > 1.00
\]

RT  
running thrust.

RT_{s,c}  
average Static test running thrust over the last 10% or so of the closing stroke.

RT_{s,o}  
average Static test running thrust over the first 10% or so of the opening stroke.

s  
sample standard deviation of a set of values.

SF  
“stem factor”. The average stem factor at points of interest in the closing and opening strokes, except at initial unwedging of the disk (see SF_{u}).

SF_{u}  
the maximum postulated stem factor at points of interest in the closing and the opening strokes, except at initial unwedging of the disk (see SF_{u,u}).

SF_{10}  
stem factor for a 0.10 friction coefficient.

SF_{15}  
stem factor for a 0.15 friction coefficient.

SF_{20}  
stem factor for a 0.20 friction coefficient.

SPD  
Spring Pack Deflection of the torque spring pack.

SPF  
Spring Pack Force, the force which causes the torque spring pack to deflect.

SRSS  
the square root of the sum of the squares of two or more numbers.

ST  
the average value of the ratio of AOTQw divided by AOTQ, a factor to account for the “Stem Thrust Effect”.

T_{cst,d}  
stem thrust at close limit switch trip under DP test conditions.
Tcst,s stem thrust at close limit switch trip under Static test conditions. Or, the estimated minimum required thrust at close torque switch trip under Static conditions that will ensure the thrust Tdpt,d can be produced in the closing stroke under the DP test conditions without tripping the close torque switch (excluding the effects of actuator output torque repeatability):
( RSft )/ ( Tdpt,d )

TDPo maximum thrust required at any point during the opening stroke of the DP test after initial unwedging of the disk (overcoming the static friction coefficient in the valve seat)

Tdpt,d maximum thrust required at any point during the closing stroke of the DP test until the disk begins to wedge into the valve body seat.

TQcst,d stem torque at close limit switch trip under DP test conditions.

TQcst,s stem torque at close limit switch trip under Static test conditions.

TQreq maximum stem torque potentially required to produce thrusts Treq,d and Treq,s.

TQsg stem torque value measured using calibrated stem-mounted strain gages.

TQstall actual actuator output torque at the motor stall condition with no thrust load on the drive sleeve.

TQTOTs the total torque developed after control (torque or limit) switch trip due to (a) the time it takes the electrical control system to remove power from the motor, and (b) the kinetic energy of the moving parts, primarily the motor rotor, which continue to drive the stem until all parts stop moving.

TQtrp,s Static test torque at close limit switch or torque switch trip.

TQtts actuator output torque measured during torque stand testing which applies a resisting torque to the actuator drive sleeve (with no thrust load on the drive sleeve).
\( T_{req,d} \) maximum thrust to reach seat under DP conditions and initiate wedging:
\[
( V_{Fc} )( A_0 ) ( D_{PRc} ) + ( A_p )( P_{upc} ) + RT
\]

\( T_{req,s} \) Thrust produced under Static conditions by the stem torque magnitude which produced \( T_{req,d} \) under DP conditions.

\( TTOTS \) the total thrust developed after control (torque or limit) switch trip due to (a) the time it takes the electrical control system to remove power from the motor, and (b) the kinetic energy of the moving parts, primarily the motor rotor, which continue to drive the stem until all parts stop moving.

\( T_{trp,s} \) Static test thrust at close limit switch or torque switch trip.

\( TSS \) Torque Switch Setting.

\( U \) the average stem thread friction coefficient at points of interest in the closing and the opening strokes, except at initial unwedging of the disk (see \( U_{un} \)).

\( U_l \) the minimum postulated stem thread friction coefficient at points of interest in the closing and the opening strokes, except at initial unwedging of the disk (see \( U_{un,l} \)).

\( U_u \) the maximum postulated stem thread friction coefficient at points of interest in the closing and the opening strokes, except at initial unwedging of the disk (see \( U_{un,u} \)).

\( U_{un} \) the average stem thread friction coefficient during initial unwedging of the disk from the valve seat.

\( U_{un,l} \) the minimum postulated stem thread friction coefficient during initial unwedging of the disk from the valve seat.

\( U_{un,u} \) the maximum postulated stem thread friction coefficient during initial unwedging of the disk from the valve seat.

\( V_{Fc} \) the average valve factor in the closing stroke for a group of nominally identical valves.
VFct the actual closing stroke valve factor determined from test data.

VFcts the "Static Equivalent Valve Factor" which accounts for the Rate of Loading effect observed for a particular MOV during Static and DP testing. The use of VFcts in the "standard" industry equation for predicting thrust requirements ensures the calculated minimum required Static condition thrust at close torque switch trip is great enough so that the close torque switch will not trip at a thrust of magnitude equal to Tdpt,d under the tested DP conditions (excluding the influence of actuator output torque repeatability at torque switch trip).

VFo the average valve factor in the opening stroke for a group of nominally identical valves after initial unwedging of the disk (overcoming the static friction coefficient of the valve seat).

VFot the actual opening stroke valve factor determined from test data after initial unwedging of the disk (overcoming the static friction coefficient in the valve seat).

VFrol a valve factor not less than "MAX VFcts" which has been adjusted to account for the average valve factor (VFc) and its uncertainty (eVFc), and the average "Rate of Loading" effect (ROL) and its uncertainty (eROL) calculated as follows:

\[
VFc \times ROL \left(1 + \sqrt{eVFc^2 + eROL^2}\right)^{1/2}
\]

W/WG ratio the gear ratio provided by the worm and the worm gear.

Other nomenclature is defined in the text of this paper where it is used.
MOTOR OPERATED VALVE PROGRAM: LESSONS LEARNED

Program to comply with
USNRC GENERIC LETTER 89-10
RECOMMENDATIONS

BILL R. BLACK, P.E.
Senior Engineer
MOV Project Engineer

TEXAS UTILITIES ELECTRIC
COMANCHE PEAK STEAM ELECTRIC STATION
Mail Zone E15, P.O. Box 1002
Glen Rose, Texas 76043
817-897-6477 (FAX: 817-897-6777)
ACTUATOR MOTOR CAPABILITIES

- Generally produce 64% of rated start torque at 80% voltage.

- Random failures to do so found during dynamometer testing of several sizes.

- One set of nine identical spare 10 ft.lb 3400 rpm motors failed to do so.
ACTUATOR PERFORMANCE FACTORS (PG 1)

- PREDICTED ACTUATOR OUTPUT TORQUE CAPABILITY AT 80% AC VOLTAGE WITH NO THRUST LOAD ON THE ACTUATOR DRIVE SLEEVE:

\[ \text{AOTQ}_{80\text{w}} = (0.64)(\text{MTQ})(\text{OAR})(\text{AF})(\text{RE}) \]

- ACTUAL TORQUE CAPABILITY, GRADUALLY LOADED: AOTQ_{80\text{stall}}

- TEST RESULTS FOR 9 ACTUATOR CONFIGURATIONS:

  SIZE 000 2 FT.LB 1700 RPM MOTOR TO
  SIZE 2 80 FT.LB 3400 RPM MOTOR
ACTUATOR PERFORMANCE FACTORS (PG 2)

AVERAGE PERFORMANCE:

108% TO 141% GREATER THAN THE PREDICTED CAPABILITY

MINIMUM PERFORMANCE:

91% TO 118% OF THE PREDICTED CAPABILITY

91% : SIZE 00 15 FT.LB MOTOR 45:1 WORM SET RATIO

100% OR MORE : ALL OTHERS
**ACTUATOR EFFECTIVE MOMENT ARMS**

- RELATE TORQUE SPRING PACK FORCE TO ACTUATOR OUTPUT TORQUE

<table>
<thead>
<tr>
<th>ACTUATOR TYPE-SIZE</th>
<th>NO THRUST</th>
<th>WITH THRUST</th>
<th>UNCERTAINTY (FT.LB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMB-000</td>
<td>0.11</td>
<td></td>
<td>6</td>
</tr>
<tr>
<td>SMB-00</td>
<td>0.17</td>
<td>0.14</td>
<td>20</td>
</tr>
<tr>
<td>SB-00</td>
<td>0.17</td>
<td>0.16</td>
<td>20</td>
</tr>
<tr>
<td>SMB-0</td>
<td>0.24</td>
<td>0.20</td>
<td>40</td>
</tr>
<tr>
<td>SB-1</td>
<td>0.26</td>
<td>0.23</td>
<td>60</td>
</tr>
<tr>
<td>SB-2</td>
<td>0.31</td>
<td>0.28</td>
<td>90</td>
</tr>
</tbody>
</table>
STEM THRUST EFFECT

\[ ST = \frac{\text{ACTUATOR OUTPUT TORQUE WITHOUT TRUST LOAD}}{\text{ACTUATOR OUTPUT TORQUE WITH TRUST LOAD}} \]

<table>
<thead>
<tr>
<th>ACTUATOR TYPE-SIZE</th>
<th>LEAST TEST RESULT</th>
<th>AVERAGE TEST RESULT</th>
<th>MAXIMUM TEST RESULT</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMB-000</td>
<td>1.05</td>
<td>1.20</td>
<td>1.32</td>
</tr>
<tr>
<td>SMB-0</td>
<td>1.02</td>
<td>1.23</td>
<td>1.44</td>
</tr>
<tr>
<td>SB-1</td>
<td>1.01</td>
<td>1.15</td>
<td>1.32</td>
</tr>
<tr>
<td>SB-2</td>
<td>0.99</td>
<td>1.10</td>
<td>1.19</td>
</tr>
</tbody>
</table>
NET EFFECT OF
PERFORMANCE FACTOR
AND
STEM THRUST EFFECT

\[ \text{NET} = \left( \frac{PF}{ST} \right) \left( 1 - \left[ \left( ePF \right)^2 + \left( eST \right)^2 \right]^{1/2} \right) \]

<table>
<thead>
<tr>
<th>ACTUATOR TYPE-SIZE</th>
<th>MTQ (FT.LB)</th>
<th>RPM</th>
<th>RE</th>
<th>(RE)(NET)</th>
<th>&gt; POE</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMB-00</td>
<td>10</td>
<td>1700</td>
<td>0.50</td>
<td>0.45</td>
<td>0.40 yes</td>
</tr>
<tr>
<td>SMB-00</td>
<td>10</td>
<td>3400</td>
<td>0.60</td>
<td>0.46</td>
<td>0.45 yes</td>
</tr>
<tr>
<td>SMB-00</td>
<td>15</td>
<td>3400</td>
<td>0.60</td>
<td>0.41</td>
<td>0.45 NO</td>
</tr>
<tr>
<td>SMB-0</td>
<td>25</td>
<td>1700</td>
<td>0.55</td>
<td>0.45</td>
<td>0.45 yes</td>
</tr>
<tr>
<td>SMB-0</td>
<td>40</td>
<td>1700</td>
<td>0.55</td>
<td>0.41</td>
<td>0.45 NO</td>
</tr>
<tr>
<td>SB-1</td>
<td>60</td>
<td>3400</td>
<td>0.60</td>
<td>0.47</td>
<td>0.45 yes</td>
</tr>
<tr>
<td>SB-2</td>
<td>80</td>
<td>3400</td>
<td>0.60</td>
<td>0.50</td>
<td>0.45 yes</td>
</tr>
</tbody>
</table>
RUNNING THRUSTS

RUNNING THRUSTS = PACKING FRICTION + OTHER

- LARGE VARIATION DESPITE WELL CONTROLLED PACKING PROGRAM

- HIGHER THAN ANTICIPATED FOR THE LARGER DIAMETER STEMS (PARTIALLY DUE TO GREATER MEASUREMENT UNCERTAINTY AT LOW LOADS)

<table>
<thead>
<tr>
<th>STEM DIA. (INCH)</th>
<th>AVERAGE (LB/INCH)</th>
<th>UPPER BOUND (LB/INCH)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/4</td>
<td>827</td>
<td>1240</td>
</tr>
<tr>
<td>1</td>
<td>310</td>
<td>620</td>
</tr>
<tr>
<td>1</td>
<td>730</td>
<td>1205</td>
</tr>
<tr>
<td>1 1/8</td>
<td>444</td>
<td>778</td>
</tr>
<tr>
<td>1 1/4</td>
<td>576</td>
<td>1008</td>
</tr>
<tr>
<td>1 3/8</td>
<td>996</td>
<td>1694</td>
</tr>
<tr>
<td>1 7/8</td>
<td>1120</td>
<td>1904</td>
</tr>
<tr>
<td>2</td>
<td>1050</td>
<td>1680</td>
</tr>
<tr>
<td>2 1/2</td>
<td>1280</td>
<td>1664</td>
</tr>
<tr>
<td>3</td>
<td>1217</td>
<td>2555</td>
</tr>
</tbody>
</table>
WESTINGHOUSE VALVE FACTORS (PG 1)

WESTINGHOUSE REQUIRED THRUST EQUATIONS WERE GENERALLY CONSERVATIVE IN THE CLOSING AND OPENING STROKES.

PORV BLOCK VALVES:
3 INCH GATE, "BLOWDOWN CONDITIONS"
2300 PSI  210000 LB/HR

CLOSING VALVE FACTOR:
TEST DATA, 0.67 > DESIGN, 0.56

OPENING VALVE FACTOR:
TEST DATA, 0.55 < DESIGN, 0.56

223
DATA FOR LARGE GATE VALVES:

- HIGH MEASUREMENT UNCERTAINTY DUE TO LOW LOADS

- CLOSE AND OPEN VALVE FACTORS:
  
  TEST DATA > DESIGN VALVE FACTOR

- DUE TO UNCERTAINTY OF THE THRUST AND DP MEASUREMENT ERRORS ?

- DESIGN MARGINS ACCOMODATE THE HIGH TEST DATA VALVE FACTORS
BORG-WARNER VALVE FACTORS

CLOSING DATA:

- NON-CONSERVATIVE DESIGN
- VALVE FACTOR = 0.3

<table>
<thead>
<tr>
<th>VALVE SIZE</th>
<th>TEST DP (psi)</th>
<th>AVERAGE VALVE FACTOR</th>
<th>MAXIMUM VALVE FACTOR TEST DATA</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>1600</td>
<td>0.48</td>
<td>0.62</td>
</tr>
<tr>
<td>4</td>
<td>2700</td>
<td>0.50</td>
<td>0.51</td>
</tr>
<tr>
<td>4</td>
<td>100</td>
<td>0.43</td>
<td>0.45</td>
</tr>
<tr>
<td>6</td>
<td>350</td>
<td>0.24</td>
<td>0.29</td>
</tr>
<tr>
<td>8</td>
<td>150</td>
<td>0.47</td>
<td>0.57</td>
</tr>
<tr>
<td>16</td>
<td>300</td>
<td>0.39</td>
<td>0.40</td>
</tr>
</tbody>
</table>

OPENING DATA:

- GENERALLY CONSERVATIVE
"RATE OF LOADING"
TORQUE SWITCH CLOSE

- HIGHER STEM THREAD FRICTION
  COEFFICIENT DURING DESIGN BASIS DP
  CONDITION THAN DURING STATIC TEST
  CONDITION

- CLOSE TORQUE SWITCH SETTING
  PRODUCES GREATER THRUST UNDER
  STATIC CONDITIONS BY THE RATIO OF THE
  STEM FACTORS

- VALVE FACTOR UNCERTAINTY:

  SOMETIMES SUFFICIENT TO ENVELOP
  DEMONSTRATED "RATE OF LOADING" EFFECTS.

  IN THESE CASES, RATE OF LOADING
  EFFECT CAN BE IGNORED.
"RATE OF LOADING"
TORQUE SWITCH CLOSE

<table>
<thead>
<tr>
<th>STEM FACTOR RATIO</th>
<th>GLOBE VALVE</th>
<th>GATE: BLOWDOWN SERVICE</th>
<th>GATE: PUMPED FLOW</th>
</tr>
</thead>
<tbody>
<tr>
<td>AVERAGE OF TEST DATA</td>
<td>1.11</td>
<td>1.30</td>
<td>1.04</td>
</tr>
<tr>
<td>MAXIMUM TEST DATA</td>
<td>1.51</td>
<td>1.40</td>
<td>1.31</td>
</tr>
</tbody>
</table>
RATE OF LOADING
LIMIT SWITCH CLOSE

(HG 1)

- HIGHER STEM THREAD FRICTION COEFFICIENT DURING DESIGN BASIS DP CONDITION THAN DURING STATIC TEST CONDITION

- CLOSE LIMIT SWITCH SETTING PRODUCES GREATER TORQUE UNDER DP CONDITIONS BY THE RATIO OF THE STEM FACTORS
"Disk Position Effect"

Disk stops traveling slightly further out of the valve seat under DP conditions.

Equal stem nut rotations. Therefor, get additional compression of compensator spring pack and greater stem thrust at limit switch trip.

Greater thrust results in greater torque at limit switch trip.
RATE OF LOADING
LIMIT SWITCH CLOSE

(PG 3)

WITHOUT "DISK POSITION EFFECT":

- AVERAGE EFFECT = 2% GREATER TORQUE
- MAXIMUM TEST DATA = 12% GREATER TORQUE

WITH "DISK POSITION EFFECT":

<table>
<thead>
<tr>
<th></th>
<th>THRUST INCREASE</th>
<th>TORQUE INCREASE</th>
</tr>
</thead>
<tbody>
<tr>
<td>WESTING HOUSE</td>
<td>AVERAGE</td>
<td>17%</td>
</tr>
<tr>
<td></td>
<td>MAXIMUM</td>
<td>26%</td>
</tr>
<tr>
<td>4 INCH GATE</td>
<td>AVERAGE</td>
<td>2%</td>
</tr>
<tr>
<td></td>
<td>MAXIMUM</td>
<td>7%</td>
</tr>
<tr>
<td>BORG-WARNER</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8 INCH GATE</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
STEM THREAD FRICTION COEFFICIENT RANGE

- ANALYSIS OF > 700 DATA POINTS
- 97 MOVS STATIC TESTED, OF WHICH 80 WERE ALSO DP TESTED
- CALIBRATED STEM-MOUNTED STRAIN GAGES

POINTS OF INTEREST:

- MAXIMUM THRUST IN CLOSING DP TEST PRIOR TO WEDGING
- CLOSE TORQUE OR LIMIT SWITCH TRIP
- TOTAL LOAD
- UNWEDGING OF THE DISK ("CRACKING")
- PEAK THRUST AFTER INITIAL UNWEDGING

FRICTION COEFFICIENT RANGE OBSERVED TO BE COMMON:

  UNWEDGING : 0.03 TO 0.19
  ELSEWHERE : 0.05 TO 0.20
SQUARE ROOT OF THE SUM OF THE SQUARES

METHODOLOGY

- WORST CASE OF EVERY PERFORMANCE CHARACTERISTIC IS NOT LIKELY TO OCCUR SIMULTANEously IN ONE MOV.

- IF ASSUME SIMULTANEous OCCURENCE OF THE AVERAGE OF EACH PERFORMANCE CHARACTERISTIC, THEN:

  50% OF THE MOVs PERFORM BETTER
  50% OF THE MOVs PERFORM WORSE

- SRSS TECHNIQUE

  PROVIDES RESULT WHICH IS BETWEEN THE RESULTS OF THE TWO ALTERNATIVES ABOVE.

  IS COMMON TECHNIQUE FOR COMBINING THE UNCERTAINTIES OF STATISTICALLY INDEPENDENT FACTORS
SQUARE ROOT OF THE SUM OF THE SQUARES METHODOLOGY

(PG 2)

SRSS TECHNIQUE

(1) DETERMINE AVERAGE PERFORMANCE CHARACTERISTIC.
(2) DETERMINE 2 SAMPLE STANDARD DEVIATIONS OF TEST DATA.
(3) DETERMINE MAXIMUM TEST DATA DEVIATION FROM THE AVERAGE.
(4) SELECT LARGER OF (2) AND (3).
(5) DIVIDE RESULT OF (4) BY THE AVERAGE TO OBTAIN THE UNCERTAINTY AS A PERCENTAGE OF THE AVERAGE.
(6) COMBINE UNCERTAINTIES BY THE SRSS TECHNIQUE AS APPLICABLE.
(7) INCREASE OR DECREASE, AS APPROPRIATE, THE AVERAGE VALUE OBTAINED IN STEP (1) BY THE RESULT OF (6).

RANGE = ( AVERAGE ) ( 1 +/- SRSS RESULT )

SRSS TECHNIQUE BOUNDS TEST DATA & PROVIDES CONFIDENCE IN MOV OPERABILITY
Session #3

MOV IMPROVEMENT PROGRAMME

Chairman: Dr. K. Kothhoff (GRS)

The papers presented in this session addressed three different subjects:
- review and improvement of safety related MOVs
- development of PRA based methods to prioritise MOVs
- preventive maintenance program for AOVs.

Three papers presented by representatives of utilities of three different countries discussed the efforts undertaken to review and improve safety related MOVs. These three utilities are performing a comprehensive study on safety related MOVs based on analytical investigations supported by tests. The overall approach applied is similar and has not been questioned during the discussion. Regarding the details and assumptions of the investigations, the discussion revealed some differences, e.g. in the friction coefficients and safety factors assumed which may need further discussion.

Two papers presented by representatives of a utility organisation and a regulator organisation respectively addressed most recent development of PRA based methodologies to prioritise MOVs with respect to their safety significance.

This tool is intended to provide guidance for decision-making on amount and schedule of the review to be performed for the different safety related valves in NPPs. Furthermore, schedules for test and maintenance of MOVs could be established based on their risk contribution. Though such a tool was felt beneficial, the discussion raised some concern regarding failure modes to be taken into account, failure data to be used and completeness of the approach.

The last paper described the development and implementation of an air operated valve preventive maintenance program based on an AOV diagnostic system. The paper emphasized the importance of the integrated preventive maintenance program implemented at the plant to manage and control the AOV program as well as all other preventive maintenance programs.
Paper No 11:SW19 for the presentation on the Specialist Meeting on MOV Issues, April 25th - 27th, 1994, Paris, France

Review of Safety Related Valves for Design Basis Events by NPP-Oskarshamn, Sweden

OKG, a Sydkraft Company, is located on the Swedish east coast about 30 km north of Oskarshamn. The three nuclear power plants (BWR) are named after the town of Oskarshamn and its founder, King Oskar 1. They are usually called 01 (442 MW), 02 (605 MW) and 03 (1160 MW).

Introduction

Ability of function of safety related valves today is in discussion in all countries with NPPs with special concentration on air and motor operated gate valves. Ever since the NRC Generic Letter 89-10 pointed out that valves and actuators in US NPP's possibly might not be able to meet all safety requirements valve problems have gained more and more attention.

The same reason made the NPP Oskarshamn start an investigation and improvement program. First the air and motor operated isolation valves of OKG 1-3 were studied in the VMAN project. The aim was to verify the ability of function of the installed valves and actuators by the use of well founded methods. OKG decided to use the experience of Siemens/KWU on the field of valves and actuators which they as turn key deliverer of NPP's gathered during design, verification, commissioning and operating of valves and actuators especially in safety systems.

Figure 1 illustrates the procedure for the valve and actuator review which OKG assisted by S-KWU intends to state for OKG in the VMAN project. It distinguishes between the ability of function, which is one off qualification of valve, respective actuator, type or design for a specified task, and the readiness for function, which is achieved by suitable maintenance and controlled by surveillance and recurrent testing. I am going to give you some information about the essential aspects of our VMAN project and, according to the topic of the today's meeting, I am concentrating on motor operated valves.
System Conditions

Basis for the investigation was a detailed definition of the system requirements for the valves and their actuators. Valves with identical size and type were grouped. The system conditions (pressure, differential pressure and temperature) for the enveloping loadcase for the opening and closing direction were associated.

For the isolation valves of the VMAN project closing after pipe rupture under blow down conditions proved to be the enveloping loadcase. Inside containment ambient conditions with temperatures up to 180°C were assumed. To assure adequate dimensions for external loads an acceleration of 6 g horizontal caused by induced vibration were superposed on the upper valve parts.

Influencing Factors

Considering valve/actuator assemblies as part of an overall system, it is necessary to determine the essential influences at interfaces to other parts of the system (Fig. 2). The three essential interfaces to be taken into consideration for valve/actuator assemblies are those to the connecting piping system, the power supply system and the control system. System loads on the valve include pressure and temperature, as well as forces and moments imposed by the connecting piping and the resulting accelerations. The differential pressure across the obturator plays a part in determining not only the required actuating forces and torques but also the loads to which a valve is subjected.

The most important factor for the power supply system is the input voltage at the actuator motor. It must be ensured that the use of fuses and bimetallic elements as protective devices does not lead to a reduction of the actuator availability. Interaction between actuator control and plant I & C equipment and its effect on valve operation must be defined and limited. The types of actuator control, i.e. stroke-limiting or torque-limiting, and the dead time which refers to the delay between torque switch trip and power dropout are important factors.

In order to provide a more practicable method of analysis it makes sense to define an interface between the valve and the actuator in terms of torque. This allows valve and actuator to be considered separately. In addition to the quantitative determination of influencing factors at the interfaces the verification of the ability of function also requires a consideration of their tolerances scattering in a large extend and a definition of allowable boundaries with a well founded method. The essential influencing factors and the used tolerances are given in figure 3. These values are qualified by experimental investigations and the effecting sum of simultaneously occurring tolerances is determined by the Gaussian approximation with the propagation of random errors.

Based on this method the influencing factors are regarded by their nominal values for the analytical investigation. Their tolerances are simplified taken into account by the use of safety factors. They cover conservatively the sums of effects determined by means of error propagation.
Actuator Qualification

Part of the analytical investigation is the check of actuator size and torque switch setting. It is not part of the VMAN project but of course the general qualification of the motor actuators has to be verified by performance of type tests according to IEEF or equivalent standards.

Hereto the mechanics of functional behavior of valve and actuator is analytically determined using physical factors which are experimentally confirmed.

Analytical Investigation

The postulated accidents as for instance pipe rupture can't be simulated in the power plant. Therefore the analytical verification shall demonstrate the ability of function of the valves, as far as possible.

Concerning the safety related function in a first step the actuating forces and torques are determined by means of an analytical model. This model serves as a tool to define the torques required for valve actuation and to size the actuator. It allows determination of the switch off and protection torque-settings, taking into consideration the type of actuator control. Obturators which are forced into the seat against differential pressure require torque switch off when motor operated to ensure seat leaktightness (e.g. globe valves). Valves where the leaktightness is caused by the differential pressure itself are controlled by limit switch off (e.g. gate valves). The same limit control is performed for all valve types stroking open.

For the ability of function not only a sufficiently sized actuator is required. But it must also be verified that the opening and closing of the valves is not hindered by unacceptable deformations.

Therefore in a second step all external loads acting on a valve assembly are taken into consideration in the stress analysis. All pressure retaining parts had been subject of investigation during the erection of the power plants according to the relevant codes and standards. The stress analysis now done in the VMAN project verifies the acceptability of loads arising from actuating forces, using an analytical loading model. Additionally loads caused by induced vibrations are superposed. The analysis takes into consideration all component parts located on the loadtransmission path (Fig. 4). A distinction is made between actuating forces which occur under design conditions, and incident related actuating forces which can be caused by either component failure or human error. Actuation forces under design conditions are ensured by a component-related margin to yield strength. For motor operated valves actuating forces are considered to comply with design conditions until excessive torque is not signalled by motor switch off through the protection torque or activation of the bimetallic element. This means that during stroking and for valves with stroke - limit control for the fully-closed or - open position, analysis takes into consideration the torque setting plus the switch trip tolerance of the actuator torque switch equipment.
In the case of torque-limit control, the excessive torque acting on the valve in its fully-closed position due for the most part to the switch off delay time is taken into consideration as a function of valve seat stiffness and actuator type.

Incident-related actuating forces are treated as a single failure. Functional capability of the system is maintained through adequate redundancy. For valves in which consequential damage must be completely ruled out, the acceptability of loads arising is shown in an integrity verification program for all possible incidents using the highest actuator torque output in the event of failure of the actuator switch off.

The applicability of the above mentioned functional and loading model depends on the valve design. Therefore in a third step an evaluation of the valve design shall assure that the valve's actuation under worst case conditions follows the assumed mechanical rules. Essentially sliding materials, gaps and tolerances, seating and backseating situations, contact pressure of the sliding areas and tilting tendency of discs/disc holder or wedge of gate valves are evaluated. Valve restraining after seating influenced by temperature changes and possibly by pipe forces for wedge gate valves is analyzed.

With the results of the investigation weak points of the actual situation and improvement possibilities are discovered. Essential hints were gained about actuator sizing, switch-off-policy, weak links in the upper valve parts, design details for disc and wedge guiding of gate valves and maintenance requirements. They are now carefully evaluated and realized not only by administrative alterations concerning actuation and handling of valves and actuators but also by possible hard ware improvements, which will be realized together with the valve manufacturer and S-KWU.

In the following the main aspects of the analytical investigation of motor operated valves shall be discussed in detail.

**Functional model**

The analytical model for a gate valve considers stroke-limit switch off for both opening and closing. The analytical model for a globe valve considers torque switch off for seating combined with a dynamic overloading factor (Fig. 5).

Both of these models include the physical and mechanical factors. However, only compliance with specified design conditions ensures that these relationships are valid.

The use of the simplified analytical models is allowable under the following conditions; the antirotation device must have a relatively long torque arm, which is generally provided in standard designs, and the valve stem nut must be bearing-mounted in both vertical directions. Both of these design features cause small frictional forces and moments. The analytical model remains consistent if these influences are included in the friction value for the stem screw thread or in the frictional forces taken into consideration for experimental verification.
Friction in the seat and guides of gate valves is dependept on the combination of materials, the contact pressure of surfaces in sliding contact and deformation and tilting influences. Friction between the valve stem and the stem nut is dependent on the combination of materials, the contact pressure of surfaces in sliding contact, the condition of the lubrication and the ambient conditions. The frictional force in the packing is dependent on the geometry, material, system pressure and to a very considerable degree - on the preload. The in/out torque ratio at stem nut release in torque controlled valves is determined solely by the stem nut friction value and the stem pitch. The breakage torque for unseating is therefore dependent on the effective torque in the fully-closed position.

The trapezoidal screw threads for valve stems as per DIN 103 exhibit for the preferred series screw thread friction values of approximately 0.15. Experimental investigations have verified this value when using chromium steels together with bronze or brass in a lubricated condition.

Given compliance with the design conditions, integral tests of parallel-seat and wedge-disc gate valves resulted in friction values of $0.4 \pm 0.1$ for stellite cladded seats.

As already mentioned the tolerances of the influencing parameters are considered by conservative safety factors. They are determined by the method of error propagation. For gate valves a safety factors $S = 1.5$ is used and for globe valves $S = 1.35$.

Operational experience in OKG shows that generally a torque ratio of $M_{\text{r}}/M_{\text{c}} = 1.5$ for unseating of globe and wedge gate valves is sufficient.

**Loading Model**

The analytical model for the loads and stresses of the valve parts due to actuating torques and forces is given in figure 6. For gate valves with limit switch off in both directions the adjusted protective torque plus tolerance is the base for the stress calculation. For globe valves with torque switch off for seating the excessive torque and an additional load factor have to be considered. The load factor conservatively covers the tolerances of the load influencing parameters and based on the method of error propagation proved to be sufficient with $f_{\text{c}} = 1.35$. The aim is to verify that the stresses in the parts of the load path don't exceed elastical boundaries even if thrusts and moments caused by external influences such as induced vibrations are superposed.

In the case of switch off failure the stall torque of the actuator at maximum voltage is used to determine the loads on the pressure retaining valve parts again concerning possible tolerances by means of the load factor. The aim is here to assure leak tightness of the pressure retaining valve parts in the case of incident related stem forces and torques.
Design Evaluation

Figure 7 illustrates the derivation of the contact stresses on seats and guides across the stroke depending on the pressure loaded area of the disc assembly and the differential pressure related to the stroke. Acceptable values for the mean contact stress should not exceed 100 N/mm².

The tilting tendency of discs on the seat and disc holder in the guides mainly depends on the geometrical conditions. The tilting tendency is a function of the moments acting on the disc assembly caused by stem force and the pressure force. Figure 8 shows the tilting effect of the disc sliding on the seat. Figure 9 illustrates the tilting effect in the guides here for the particular disc holder design with guide groves running on guide ribs at the valve body. The tilting tendency amplifies the local increase of contact stress and can possibly lead to line- or even point-contact. Depending on the geometrical conditions in gate valves tilting can't be avoided completely but at least tilting and high mean contact stress shall not coincide. Figure 10 shows by way of a 200 mm double disc parallel gate valve the amounts of contact stress and tilting moment when the valve is closed under differential pressure according to pipe rupture conditions. An increase of sliding friction on the seat would initiate a self amplifying effect when excessive local contact stresses due to disc tilting cause higher friction values. The aim is to make the valves actuating behavior under accident conditions predictable by comparing it with under blow down conditions successfully tested valve designs.

Main Results of the Investigation

The investigation revealed very different capabilities of valves and actuators. Fortunately a lot of valves proved to meet the actual requirements. On the other hand a lot of improvements possibilities were identified. The main are:

- Raising of torque switch setting, and if necessary choice of a bigger actuator to meet the increased required torque to actuate the valves under design base conditions.
- Change from torque switch off to limit switch off for wedge disc gate valves without leakage problems to avoid problems with valve restraining in the fully closed position. Exact adjustment instructions have to be obeyed and controlled.
- Alteration of stem and upper valve parts to those of stronger dimensions to meet actuating loads arising under design base conditions.
- Redesign of the disc holder guidance to improve sliding situation by reducing the contract stresses and the tilting tendency.
- Reinforcement of guide ribs in the valve body to increase the load capacity for closing under full flow conditions.
Further Steps

At present time we are realizing the alterations and hardware improvements together with the valve manufacturers and intend to finish the project VMAN with the plant revisions in 1995.

In the meantime we have started to improve the communication between different electric and mechanical departments of our company which are responsible for design, operation and maintenance concerning valve and actuator problems. We are going to continue to train the understanding of the staff about the interactions between valve, actuator and the connected equipment, so that they get used to deal the problems as a whole.

The investigation shows that the analytical procedure can't cover all influences during the whole lifetime.

There are still deficiencies where the knowledge about valve and actuator behavior shall be completed by information based on test results. Therefore adequate surveillance - and in-service-tests shall be performed, which reliably detect anomalies or unallowable drifts. Periodic tests with measurement of active power are supplemented by regular system checks, internal visual examinations and bench tests of actuator torque. These procedures enable detection of any changes in valve and actuator performance. Measurement of active power at the motor control center requires relatively little effort. Evaluation criterias and guidelines for necessary measures can be derived from the calculation model and the design evaluation.

Combined with preventive maintenance procedures which are oriented on function affecting influences we intend to optimize the long-term readiness for function of all safety related motor operated valves.

Enclosures

Figure 1: Integral Valve Performance Conset
Figure 2: Interfaces of Motor Operated Valves
Figure 3: Interfaces, Influences and Tolerances
Figure 4: Component Parts Located on Load-Transmission Path
Figure 5: Functional Model
Figure 6: Loading Model
Figure 7: Pressure Loaded and Sliding Areas
Figure 8: Disc Tilting on the Seat
Figure 9: Disc Holder Tilting in the Guides
Figure 10: Disc/Seat Sliding of a Double Disc Parallel Gate Valve DN 150
SIEMENS

Review of Safety Related Valves for Design Basis Events
by NPP-Oskarshamn, Sweden

ABILITY OF FUNCTION
ANALYTICAL CONCEPT + READINESS FOR FUNCTION
TRENDS DURING LIFETIME

CALCULATION
ACTUATOR,
VALVE
COMPONENTS + DESIGN
EVALUATION
FUNCTION RELEVANT
FEATURES,
MECHANICS

BASELINE
MEASUREMENT
IN PLANT + RECURRENT
TESTS
IN PLANT
DURING LIFETIME

FULL FLOW TESTING
IN TESTRIG OR PLANT,
COVERING OPERATION AND
ACCIDENT CONDITIONS

MAINTENANCE
MEASURES DURING
LIFETIME

Figure 1

Integral Valve Performance Concept
SIEMENS

Review of Safety Related Valves for Design Basis Events
by NPP-Oskarshamn, Sweden

Plant I & C

Power Supply

Control and Monitoring

Motor switching and protective equipment

Switches and signal equipment

Ambient conditions

Accelerations

Piping end loads

System conditions

Differential pressure

Interfaces of Motor Operated Valves

Figure 2

245
# Review of Safety Related Valves for Design Basis Events
by NPP-Oskarshamn, Sweden

<table>
<thead>
<tr>
<th>Influences Nominal Value</th>
<th>Tolerances</th>
<th>Application</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction Seat/Guides $\mu = 0.4$</td>
<td>25 %</td>
<td>25 %</td>
</tr>
<tr>
<td>Packing Force $F_{St} = f$ (Preload)</td>
<td>50 %</td>
<td>50 %</td>
</tr>
<tr>
<td>Stem Thread Factor $\mu_G = 0.15$</td>
<td>33.3 %</td>
<td>33.3 %</td>
</tr>
<tr>
<td>Actuator Torque Setting $M_E$</td>
<td>10 %</td>
<td>10 %</td>
</tr>
<tr>
<td>Motor Voltage 380 V</td>
<td>15 %</td>
<td>10 %</td>
</tr>
</tbody>
</table>

**Interfaces, Influences and Tolerances**

**Figure 3**
SIEMENS

Review of Safety Related Valves for Design Basis Events
by NPP-Oskarshamn, Sweden

Stop nut
Key
Bearing
Stem nut
Stem
Antirotation device
Yoke assembly
(incl. flange / bolt connections)
Bonnet
Body flange connection
Stem/disc connection
Disc assembly

Component Parts Located on Load-Transmission Path Figure 4

NDM 3/03 94/Kp-G8

247
Review of Safety Related Valves for Design Basis Events
by NPP-Oskarshamn, Sweden

Gate Valves (limit switch off):

\[ F_T = \mu \times A_p \times \Delta p \pm A_{Sp} \times p + F_{St} \]

\[ M_T = F_T \times C_G \]

\[ M_E = M_T \times S \]

\[ M_{W} = M_{E} \times C_{U} \]

Globe Valves (torque switch off):

\[ F_T = A_W \times \Delta p \pm A_{Sp} \times p + F_{St} \]

\[ M_T = F_T \times C_G \]

\[ M_E = M_T \times S \]

\[ M_{W} = M_{E} \times C_{U} \]

Unseating:

\[ M_E = M_{W} \times C_{R/R} \times S \]

- \( M_T \) = Total required torque
- \( M_E \) = Actuator torque setting
- \( M_{W} \) = Torque effective in fully-closed position
- \( F_T \) = Total required stem thrust
- \( F_{St} \) = Frictional force in packing assembly
- \( A_W \) = Effective area
- \( A_{Sp} \) = Stem cross section area
- \( A_p \) = Disc surface area
- \( \mu \) = Frictional coefficient in seat/guides
- \( \Delta p \) = Differential pressure
- \( p \) = System pressure
- \( C_G \) = Stem thread factor
- \( C_{U} \) = Excess torque factor
- \( C_{R/R} \) = Torque ratio, stroking closed/open
- \( S \) = Safety margin
Review of Safety Related Valves for Design Basis Events
by NPP-Oskarshamn, Sweden

limit switch off:

\[ F_L = M_E \times \frac{C}{C_G} \]
\[ \sigma_i = f_{i1} (P, F_L, M_E) + f_{i2} (F_{EX}, M_{EX}) \]
\[ \leq R_{p0,2T}/S_B \]

torque switch off:

\[ F_L = M_E \times f_L \times \frac{C_0}{C_G} \]
\[ \sigma_i = f_{i1} (P, F_L, M_E) + f_{i2} (F_{EX}, M_{EX}) \]
\[ \leq R_{p0,2T}/S_B \]

switch off failure:

\[ F_{stall} = M_{stall} \times f_L/C_G \]
\[ \sigma_i = f_i (F_{stall}, M_{stall}) \]
\[ \leq R_{mT}/S_D \]

\[ M_E = \text{Actuator torque setting} \]
\[ M_{stall} = \text{Stall torque at switch off failure} \]
\[ M_{EX} = \text{Moment caused by external loads} \]
\[ F_L = \text{Total resulting stem thrust} \]
\[ F_{stall} = \text{Stem thrust at switch off failure} \]
\[ F_{EX} = \text{Forces caused by external loads} \]
\[ f_L = \text{Load factor} \]
\[ f_i() = \text{Function of () for parts in the load path} \]
\[ C_0 = \text{Excess torque factor} \]
\[ C_G = \text{Stem thread factor} \]
\[ \sigma_i = \text{Stresses on parts in the load path} \]
\[ R_{p0,2T} = \text{Yield strength at relevant temperature} \]
\[ R_{mT} = \text{Ultimate strength at relevant temperature} \]
\[ S_B = \text{Safety margin for normal operation} \]
\[ S_D = \text{Safety margin for integrity verification} \]
SIEMENS

Review of Safety Related Valves for Design Basis Events
by NPP-Oskarshamn, Sweden

\[ P_{Fm} = \frac{F_p}{A_S} \]

\[ F_p = \int_{A_{p1}} \Delta p \cdot dA_{p1} + \int_{A_{p2}} \Delta p \cdot dA_{p2} \]

\[ A_{p1}, A_{p2} \quad \text{flow induced area} \]

\[ 2 \cdot A_S \quad \text{sliding area} \]

Pressure Loaded and Sliding Areas

Figure 7
\[ \Sigma M = M_R - M_P \]

if \( \Sigma M > 0 \Rightarrow \) tilting!

with

\[ M_R = F_S \cdot B \]
\[ M_P = F_P \cdot z_D \]

\( \Sigma M \) - sum of moments around the tilting axis

\( F_P \) - pressure force in the origin of force of the pressure loaded disc area

\( F_S \) - stem force

\( z_D \) - lever arm of pressure force

\( B \) - lever arm of stem force

**Disc Tilting on the Seat**

*Figure 8*
\[ \Sigma M = M_p - M_R \]

if \( \Sigma M > 0 \Rightarrow \) tilting!

with

\[ M_R = F_S \cdot \frac{t}{2} \]

\[ M_p = F_p \cdot e \]

\( \Sigma M \) - sum of moments around the tilting axis
\( F_p \) - pressure force in the origin of force of the pressure loaded disc area
\( F_S \) - stem force
\( e \) - lever arm of pressure force
\( \frac{t}{2} \) - lever arm of stem force

**Disc Holder Tilting in the Guides**

Figure 9
SIEMENS

Review of Safety Related Valves for Design Basis Events
by NPP-Oskarshamn, Sweden

Disc/Seat Sliding of a Double Disc Parallel Gate Valve DN 150

Figure 10
ONCE/IAEA Joint Specialist Meeting on Motor Operated Valve Issues

April 25 - 27, 1994
Paris, France

AIR OPERATED VALVE PREVENTIVE MAINTENANCE PROGRAM
DEVELOPMENT/IMPLEMENTATION
AT
ONTARIO HYDRO'S BRUCE NUCLEAR GENERATING STATION "A"

BRIAN J. FERGUSON
ONTARIO HYDRO
WILLIAM FITZGERALD
FISHER CONTROLS
ABSTRACT

This paper describes the development and implementation of the Air Operated Valve (AOV) Preventive Maintenance (PM) Program at Ontario Hydro's Bruce Nuclear Generating Station "A" (Bruce NGS "A"). This is one of a number of new PM programs at Bruce NGS "A" that are being managed by the Integrated Preventive Maintenance Program (IPMP). The IPMP has allowed us to take firm control over all PM activities at Bruce NGS "A" by forcing us to critically assess our PM capabilities and to establish "single point" program management responsibility. In addition, AOV diagnostic system fundamentals, AOV PM Program development/implementation, and valve test results are discussed in detail.
1.0 INTRODUCTION

Air Operated Valve diagnostics is one of a number of new preventive/predictive maintenance technologies that have become available to the nuclear industry recently. The full benefit of these new techniques can only be realized when Utilities incorporate them into their PM programs. In time, this will allow them to make the transition from performing only "diagnostics" to "predictive maintenance" where component deterioration is monitored and function is maintained by taking corrective action before failure occurs.

Bruce Nuclear Generating Station "A" has taken the lead, for Ontario Hydro, in developing and implementing a Preventive Maintenance Optimization Program. The drive for this activity was provided by Peer and Regulatory concern with the quality of our PM program, the desire to contain the growth of maintenance costs, and to improve plant system reliability.

The program has two parts:

1. Reliability Centred Maintenance (RCM) analysis, and
2. The Integrated Preventive Maintenance Program (IPMP), the umbrella under which all Bruce NGS "A" PM Programs are managed.

This paper will describe the Integrated Preventive Maintenance Program and the Air Operated Valve (AOV) Preventive Maintenance Program that is now managed and controlled by the IPMP.
2.0 THE INTEGRATED PREVENTIVE MAINTENANCE PROGRAM AND PM CAPABILITY ASSESSMENT

The IPMP has two component parts:
1. A procedural framework as shown in Figure 1, and
2. The PM Engineering Organization as illustrated in Figure 2.

The four procedures, listed in Figure 1, completely describe how the IPMP will operate throughout the Station's lifecycle by defining program strategy, staff roles and responsibilities, RCM methodology, current plant predictive maintenance capabilities, and feedback and trending of PM program activities.

The PM Engineering Organization establishes single point responsibility for the entire PM program. This group has responsibility for RCM analysis, PM task administration, and feedback, trending and analysis of equipment performance. Performance trending is the key to maintaining a "living" PM program.

A crucial step in establishment of the IPMP was performing the predictive maintenance capability assessment (see Figure 3).

This process assessed and inventoried our current preventive/predictive maintenance capabilities and provided a "pick list" for our RCM analysts as they began the optimization of the Station PM Program.

AOV diagnostics was one of the key technologies that we used to build a better PM program. Incorporation of this new technology into the Bruce NGS "A" PM program will be the focus of the remainder of this paper.
3.0 AIR OPERATED VALVE PREVENTIVE MAINTENANCE PROGRAM

3.1 Introduction

Traditionally, Air Operated Valves (AOV) have received little attention by our Maintenance staff. Valve maintenance has been predominantly corrective and calibration/setup of AOVs was treated as an art, not a science.

The use of diagnostic analysis for assessing AOV performance at Bruce NGS "A" began as a pilot study in 1991. This study demonstrated the benefit of using the technology as a predictive maintenance and calibration tool for the Air Operated Valves in our Station. As a result, an AOV PM program based on the use of AOV diagnostics has been incorporated into the Bruce NGS "A" Integrated Preventive Maintenance Program.

3.2 Diagnostic System

The diagnostic system in use at Ontario Hydro is the Fisher Controls "Flowscanner". This system was developed, by Fisher Controls, to aid in the evaluation and diagnosis of problems with pneumatically-operated control valves. It is a portable, self-contained, computer-based data acquisition system that collects data from the control valve assembly, and it's accessories. The system software operates on this data and presents it as a characteristic trace or "signature" for the valve. Assessment of valve condition requires interpretation of this signature and other software generated data.
The system can be used on many valve types and is able to collect up to 13 different channels of data at one time. The standard "Flowscanner" set-up for a typical control valve is shown in Figure 4. In this case, the valve has a double-acting piston operator, volume boosters at both inputs to the piston, a double-acting positioner, and a current to pressure (I/P) transducer. The system measures valve stem position using a linear encoder attached to the valve stem. It also measures/records pneumatic pressures and, in this example, would measure/record the following pressures:

- Top piston pressure
- Bottom piston pressure
- Positioner output #1
- Positioner output #2
- Positioner input and I/P output
- Supply Pressure

The system can also generate or monitor the control signal coming to the valve at the input to the I/P. Using this channel to generate a control signal, the valve can be sent through a number of tests, including step response, ramped input, or a standard deviation cycle test.

In the standard diagnostic test, we first collect and record information about the valve construction and service conditions. This is summarized on what is called the "Nametag" screen. (see Figure 5 for a standard spring-and-diaphragm actuated control valve). This information is used to help determine how the valve should react during the test, which can then be compared to actual operation for diagnosis of problems. Once this information has been collected, it is stored in the system database and can be used each time a diagnostic test is run.
Secondly, we install the diagnostic system on a valve in the field. A travel transducer is temporarily connected to the stem and the built-in pressure transducers in the diagnostic system are connected to the pneumatic lines. This can normally be done at the positioner gauge block, which eliminates the need to break into the tubing. NB: Many Utilities have permanently mounted quick-disconnects in the pneumatic lines to facilitate hook-up, and testing.

Finally, we connect to the I/P input so that the valve can be controlled with the system. As mentioned earlier, the system can also monitor a control signal from another source, like the control room, if Operations prefers to maintain control of the valve. At this point, the valve is ready to test. The set-up normally takes about 30 minutes, however, this time can reduced with some advance planning.

Before discussing what can be learned from the test sequence, we need to review the conditions under which the tests should be carried out.

In order to gain meaningful data the valve must be operated. Ideally, it is preferable to move it over the complete stroke so that the operation can be checked at all positions. This normally means testing with the valve on bypass or during an outage, since fully stroking the valve can disturb the process. However, if there are problems that might surface due to temperature or flow, it's better to test under actual conditions so that the chances of detecting the root cause of the problem are maximized. This may mean stroking the valve over a limited range of travel so that the process stays under control. This still gives valuable information over the range of travel tested, and in many cases, it is all that is needed if the valve spends most of it's time in this range. Having said this, most tests are run under static conditions and results have shown that the majority of operational problems
encountered can be detected without flow or temperature.

The standard test utilizes what is called a "dynamic scan" where the input signal is ramped from one value to a second value and back again. The two endpoints, along with the speed of the ramping, can be selected in the software. The standard test is run from 0 to 24 mA over 50 seconds (each direction). Some selected results, for a valve tested using this procedure, are shown in Figures 6 through 14. Each of these figures is described below.

FIGURE 6: This figure shows the overall valve response in inches of travel, plotted against the input signal in mA. As such, it includes the performance of all the components that make up the valve assembly: the I/P, the Positioner, the Actuator, and the Valve body assembly. The valve position is shown as the valve closes and then reverses and opens again in response to the ramped input signal. The arrows show the stroking direction. In looking at this signature, the spacing between the two parallel lines tells us how much dynamic error is present, the endpoints help us to define the zero and span for the assembly, the straightness of the lines shows linearity, and the travel scale defines the valve travel.

FIGURE 7: This figure shows typical data analysis performed by the system. "Dynamic error band" is a new term, by the developer of the system, to reflect that the input signal is constantly changing and therefore adds some dynamic error to the normal "hysteresis-plus-deadband" that would be measured in a static deviation cycle. Each valve type has a range of acceptable values for the performance data summarized by the system.

FIGURE 8: Because of the way the data is collected, the overall valve performance can be checked, as shown in the two previous graphs, and then the function of the individual components can be evaluated to help determine how they might be affecting overall valve performance. In this way, problems can be isolated to a
specific component and corrective action taken on only those components that are malfunctioning. This figure shows the performance of the I/P. Note: These results were generated from the same test data as Figures 6 and 7. No additional tests were required. The data is just being looked at in another way.

FIGURE 9: This plot shows positioner performance.

FIGURE 10: This plot is important, in that it shows the net pressure in the actuator plotted vs valve travel. The net pressure is transformed into a force using the actuator effective area. By examining this force vs. travel curve, we can learn a great deal about what is going on inside the valve without requiring valve disassembly. This plot shows the pressure vs travel in each stroking direction and also permits us see how the valve is seating as the plug hits the seat. By looking at the spacing between the lines in each stroking direction, one can determine the friction in the assembly, since it is the only force that reverses itself with stroking direction. Based on these values, one can determine if the packing is properly adjusted or if there is sticking or jumping inside the valve due to bearing problems.

This particular actuator has a spring, the slope of the lines indicates the stiffness of the spring and the relative position of the lines on the graph indicates how much tension has been initially wound into the spring. Both of these characteristics determine how well the valve will perform in service.

The seat load and contact are very important in insuring that the valve will properly shut off. By looking at the upper right hand portion of the curve one can determine if the plug is contacting the seat and how much seat load is being developed. The system corrects this reading for the unbalance forces in the valve due to pressure, and provides a value for the seat load in service.
FIGURE 11: This plot shows the results once the curve from Figure 10 has been analyzed.

FIGURE 12: This is a "zoomed" view of the valve travel near the seat that enables one to determine if the plug has hit on the seat and, if so, how much load is being developed. The plug initially contacts the seat at about 19 psi and loads up as the pressure increases to 28 psi. The shape of the curve can tell us if the plug is making proper contact and if the seating surfaces are in good condition.

FIGURE 13: This plot shows the supply pressure plotted against travel, and permits verification that the supply pressure and air volume are consistent with the needs of the valve assembly. Test results have shown that many valves have inadequate air volume due to the size of the connecting lines or the size of the airset on the assembly. As a result, the valve strokes much more slowly than it should, degrading control or safety performance. When this situation exists, this plot will show a significant drop in pressure as the valve strokes. As can be seen, there was no problem with supply pressure for this valve.

FIGURE 14: This is a summary report that extracts data from the "Nametag" for the specified column, and the data from the test curves for the measured column. Comments are added by the system user.

To this point, signatures have shown typical results for a standard test and for a valve in good condition. It is also beneficial to show some examples of other types of tests, as well as some curves for valves that are not performing to specification. These are shown in Figures 15 to 21.

FIGURE 15: This illustrates what happens to I/P performance when there is a leak in the pneumatic relay.
FIGURE 16: This illustrates how the signature changes if the friction increases either due to a malfunction or a change in packing type. Note that increased friction does not change the benchset but it can negatively affect seat load since it changes the initial seating contact pressure.

FIGURE 17: This shows how the signature changes if we introduce flow through the valve. The shift in the curve is caused by the pressure unbalance on the plug and stem. In fact, if we compare the curves with and without flow, the unbalance forces on the internal parts can be determined, which can aid in identifying instability problems due to flow.

FIGURE 18: This plot shows an overlay comparison of the net pressure curves with and without pressure, and flow through the valve. Note that the effect of flow becomes more pronounced as the plug approaches the seat, as we would expect since this valve is "flow up" and is unbalanced.

FIGURE 19: This shows an alternate type of test that follows the SAMA standard for a deviation cycle test. The valve is sent through a 5 step test where valve position is recorded vs. the input signal, permitting the static "hysteresis-plus-deadband" and linearity to be calculated.

FIGURE 20: This figure shows how the valve responds to a 100% step change in both directions, and permits things like stroking time, deadtime, rise time etc. to be measured for a valve. This information can be important from both a control and safety standpoint.

FIGURE 21: This shows what happens to the friction levels when the valve stem is worn due to high cycling over a given range of travel. The low levels of friction over this range would probably also mean that the packing is leaking.
All the possible variations in performance "signatures" are too numerous to cover here, however these examples should illustrate system fundamentals, and how AOV diagnostics can be used for valve condition assessment.

3.3 The AOV Program

3.3.1 Program Procedure

A procedure (BGA-OPDP-2.08-0, currently in draft form) has been written to document the Bruce NGS "A" AOV PM program.

The three key elements of this program are as follows:

1. Organization,
2. Valve priority ranking, and
3. Preventive and corrective maintenance work flow.

3.3.2 Organization

A full time "Valve Diagnostic Team" has been dedicated to this program. It's structure is shown in Figure 22. This team is responsible for both the AOV and MOV PM programs, and is a specialist "Days Only" crew composed of both control (electrical/I&C) and mechanical maintainers. Their duties include:

1. Collection, interpretation and storage of valve signature data.
2. Repair, calibration and setup of valves in the program.

Program management and engineering support is provided by an engineer from the Maintenance Engineering Unit.
3.3.3  **PM Priorities**

Currently there are approximately 100 valves (per unit) in the PM program. They have been selected by the following criteria:

1. Valves inside containment.
2. Problem valves identified by Operators or Engineering, and
3. Valves identified as critical by RCM analysis.

Diagnostic frequency has been set at once every 3 years. (including complete elastomer replacement on first test). However, as RCM analysis proceeds and trend data accumulates, frequencies will be refined to reflect failure history and service conditions.

3.3.4  **Preventive and Corrective Maintenance Work Flow**

The flow of AOV diagnostics and repair work within the maintenance department is described by a flowchart in Figure 23. It addresses two paths:

1. Corrective, and
2. Preventive Maintenance (or Call-ups).

In the case of corrective maintenance it is important to note that:
- AOV deficiencies can be initiated by maintenance or operations staff.
- AOV work is usually assigned to the Valve Diagnostic Crew but in the case of emergency, Duty Shift maintenance personnel can perform troubleshooting and repairs. However, an "as left" signature of the valve must be taken by the Valve Diagnostic Team.
All preventive maintenance is performed by the Valve Diagnostic Team. This includes performing "as found" and "as left" diagnostics, repair, calibration and Post Maintenance Testing (PMT) for each valve.

3.4 Results

Since the program began, a total of 451 scans have been performed, 413 (91%) as preventative (including valve set-up and Post Maintenance Testing) and 38 (9%) as corrective maintenance. 351 of these were initial and 100 follow-up or subsequent scans.

Figure A is a summary of faults found for all preventive and corrective analyses.

It is interesting to note that a significant reduction in valve faults has occurred when comparing the first to second scan. i.e. - valves requiring no maintenance increased from 12% to 22%. - all fault frequencies decreased.
The data also indicated that more work must be done to improve the ability of I/Ps and positioners to maintain proper calibration settings.

It is also worthwhile to review plots of preventive, and corrective data individually.

Preventative Maintenance (PM): (328 valves - 413 scans)

Figure B is a summary of PM scans only.

- preventative maintenance tasks focus on rebuilding the positioners and I/Ps, a check on the valve operating characteristics (stroke, stroke time, bench set, packing friction) and the condition of the valve internals and air supply (including PRVs, boosters). The high incidence of I/Ps and
positioners requiring calibration suggests that our PM focus of rebuilding these items is correct.

- for valves scanned for the first time (322 scans - 78%), over 65% of the I/Ps and positioners required calibration. The next highest failure/adjustment was the bench set at 15%. The remainder were all under 10%. 12% required no maintenance at all.

- for valves previously scanned (88 scans - 22%), approximately 45% of the I/Ps and positioners required calibration. The remainder of the failures were under 5%. 22% required no maintenance at all.

Corrective Maintenance (CM): (37 valves - 38 scans)

Figure C is a summary of faults for CM scans only.

- corrective maintenance incorporates two actions: 1) troubleshooting and 2) repair of obvious defects (ie. air leaks). After each incidence of corrective maintenance the valve was completely set-up according to specification and in some cases the I/P and positioner were rebuilt preventively.

- for valves scanned for the first time (29 scans - 76%), over 48% and 55% of the I/P's and positioners respectively required calibration. The next highest failure/adjustment were positioner and I/P rebuilds at 37% and 31% respectively. The remainder were 10% or less.
3.5 COST BENEFIT

The AOV Program costs to date can be summarized as follows:

Hardware:

3 Flowscanners - $200K

Labour/Material (per year)

Net additional PM costs - labour - $55K
- materials - $44K

Engineering - $50K

$149K

The major benefit of this program is a reduction of forced outage rate due to AOV failures. A review of our forced outage history shows that AOV failures cost Bruce NGS "A" approximately $1M per year.

Therefore first year benefit can be calculated to be $650K, and subsequent years to be $850K.

4.0 CONCLUSIONS

The Integrated Preventive Maintenance Program has enabled us to take control of our Preventive Maintenance Program at Bruce N.G.S. "A". In addition it has provided us with a means for integrating new predictive maintenance technologies into our preventive maintenance program.

We have also developed and implemented an AOV preventive maintenance program at Bruce N.G.S."A". This program is expected to save $650K in its first year of implementation and $850K per year in each following year.
Integrated Preventive Maintenance Program Procedure Hierarchy

Figure 1
PM Engineering Organization

Figure 2
### Figure-3

**Condition Monitoring and Diagnostic Technologies**

**Capability/Responsibility Matrix**

<table>
<thead>
<tr>
<th>Technology</th>
<th>Responsibility</th>
<th>Equipment/System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vibration Analysis Rotating Equipment</td>
<td>Engineering Services,</td>
<td>VITEC Data Collection and Analysis System Wave Tek FFT Analyzer</td>
</tr>
<tr>
<td></td>
<td>Equipment Monitoring Crew</td>
<td></td>
</tr>
<tr>
<td>Lubricant Analysis Rotating Equipment, Transformers</td>
<td>Maintenance Support,</td>
<td>Samples sent off-site</td>
</tr>
<tr>
<td></td>
<td>Mech Maintenance, Operators</td>
<td></td>
</tr>
<tr>
<td>Wear Particle Analysis Rotating Equipment</td>
<td>Maintenance Support,</td>
<td>Samples sent off-site</td>
</tr>
<tr>
<td></td>
<td>Mech Maintenance</td>
<td></td>
</tr>
<tr>
<td>Fluid Flow Monitoring Pumps, Heat Exchangers</td>
<td>Engineering Services,</td>
<td>Controlatron 990</td>
</tr>
<tr>
<td></td>
<td>Control Maintenance</td>
<td></td>
</tr>
<tr>
<td>Motor Winding Testing Megger, Polarization Index, DC Hi Pot, Surge Comparison</td>
<td>Control maintenance</td>
<td>ELECTROM TIG Winding Analyzer (1 unit)</td>
</tr>
<tr>
<td>Generator Testing Megger, AC Hi Pot, DC Ramp, Partial Discharge</td>
<td>Nuclear Technology Services</td>
<td></td>
</tr>
<tr>
<td>Motor Operated Valve Testing</td>
<td>Maintenance Support,</td>
<td>Liberty VOTES System (2 units)</td>
</tr>
<tr>
<td></td>
<td>Control Maintenance</td>
<td></td>
</tr>
<tr>
<td>Air Operated Valve Testing</td>
<td>Maintenance Support,</td>
<td>Fisher Flow Scanner System (3 units)</td>
</tr>
<tr>
<td></td>
<td>Control Maintenance</td>
<td></td>
</tr>
<tr>
<td>Check Valve Monitoring</td>
<td>Maintenance Support</td>
<td>System To Be Purchased (Late 1994)</td>
</tr>
<tr>
<td>Steam Trap Monitoring</td>
<td>Maintenance Support,</td>
<td>TLV Trapman System (1 unit)</td>
</tr>
<tr>
<td></td>
<td>Equip Monitoring Crew</td>
<td></td>
</tr>
<tr>
<td>Thermography on Electrical Equipments</td>
<td>BNPD Central Services</td>
<td>AGEMA Themovision Model 470 and 870</td>
</tr>
<tr>
<td>Cable Testing Megger and Hi Pot Test on Power Cables, TDR on Coax Cables</td>
<td>Control Maintenance</td>
<td>Time Domain Reflectometry (TDR) Equipment</td>
</tr>
</tbody>
</table>

273
<table>
<thead>
<tr>
<th>Tag#: mc123</th>
<th>Serial#: 1234</th>
<th>03-28-1994</th>
</tr>
</thead>
<tbody>
<tr>
<td>Description: Test Valve</td>
<td>16:40:33</td>
<td></td>
</tr>
<tr>
<td>Plant Site: Mckinney Plant TX</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Body Style: ES</th>
<th>Body Size: 1 IN</th>
<th>Class: 150</th>
</tr>
</thead>
</table>

|-------------|--------------------|----------------------|

<table>
<thead>
<tr>
<th>Port Diameter: 1-5/16 IN</th>
<th>UnBalanced Area: 1.35 Square IN</th>
<th>The valve is UNBALANCED</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Stem Diameter: 3/8 IN</th>
<th>Spec. Packing Friction: 38</th>
<th>Packing Type: TFE / Single</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Total Stem Friction: 38 LBS</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Leak Class: IV</th>
<th>Spec. Inlet Pressure: 150</th>
<th>Seat Type: Metal</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Outlet Pressure: 20 PSIG</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Acquired Seat Load: 164 LBS</th>
<th>Valve Travel: .75 IN</th>
<th>Stroking Time:</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Open: SEC</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Close: SEC</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Actuator Type: 657 Style 34</th>
<th>Bench Set: 3-15 PSIG</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Pressure Closes the Valve</td>
<td>Effective Area 69 Sq IN</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ACCESSORIES NOTED</th>
<th>I/P Type: 546</th>
<th>Resistance: 176 Ohms; Output: 3-15 PSIG</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Positioner Type: 3582</th>
<th>Zero Control signal = OPEN</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Other:</th>
<th>Other:</th>
<th>Other:</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Noted Comments:</th>
<th>Valve used for testing on flow.</th>
</tr>
</thead>
</table>

Figure 5
Figure 6
I/P out: Pos. In Pressure in PSIG

Input Current (mA) to I/P Converter

Figure 8
<table>
<thead>
<tr>
<th>Test Parameter</th>
<th>Spec'd</th>
<th>Measured</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>OVERALL VALVE CONTROL</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Travel (IN)</td>
<td>.75</td>
<td>.86</td>
<td>Slight overtravel; adjust</td>
</tr>
<tr>
<td>Dyn. Zero Travel (ma)</td>
<td>4</td>
<td>4.56</td>
<td>High - adjust I/P (see below)</td>
</tr>
<tr>
<td>Dyn. Max. Travel</td>
<td>20</td>
<td>20.93</td>
<td>High - adjust positioner (see below)</td>
</tr>
<tr>
<td>Dyn. Err. Band(%)</td>
<td>N/A</td>
<td>2.138</td>
<td>OK</td>
</tr>
<tr>
<td>Dyn. Lyn. (±/-)</td>
<td>N/A</td>
<td>.641</td>
<td>OK</td>
</tr>
<tr>
<td>VALVE AND ACTUATOR DATA</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Friction Av. (LBS)</td>
<td>38</td>
<td>21</td>
<td>Low but OK; Teflon is often lower than specs</td>
</tr>
<tr>
<td>Maximum</td>
<td>38</td>
<td>46</td>
<td>specs</td>
</tr>
<tr>
<td>Minimum</td>
<td>N/A</td>
<td>13</td>
<td></td>
</tr>
<tr>
<td>Seat Load - Test (LBS)</td>
<td>N/A</td>
<td>866</td>
<td>OK</td>
</tr>
<tr>
<td>- in Service</td>
<td>164</td>
<td>689</td>
<td>OK</td>
</tr>
<tr>
<td>Spring Rate</td>
<td></td>
<td>1054</td>
<td></td>
</tr>
<tr>
<td>BenchSet (PSIG)</td>
<td>3-15</td>
<td>2.32-13.77</td>
<td>Low but OK</td>
</tr>
<tr>
<td>POSITIONER DATA</td>
<td>Type: 3582</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dyn. Set (PSIG)</td>
<td>3-15</td>
<td>3.17-15.97</td>
<td>Calibrate so valve is fully stroked at</td>
</tr>
<tr>
<td>Dyn. Err. Band(%)</td>
<td>N/A</td>
<td>.817</td>
<td>15 PSI</td>
</tr>
<tr>
<td>Dyn. Lyn. (±/-)</td>
<td>N/A</td>
<td>.535</td>
<td>OK</td>
</tr>
<tr>
<td>I/P TRANSDUCER DATA</td>
<td>Type: 546</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dyn. Set (PSIG)</td>
<td>3-15</td>
<td>2.67-15.16</td>
<td>Calibrate so valve is full stroked at</td>
</tr>
<tr>
<td>Dyn. Err. Band(%)</td>
<td>N/A</td>
<td>1.364</td>
<td>4 mA (3 psi)</td>
</tr>
<tr>
<td>Dyn. Lyn. (±/-)</td>
<td>N/A</td>
<td>.457</td>
<td>OK</td>
</tr>
<tr>
<td>SEAT EVALUATION</td>
<td></td>
<td></td>
<td>OK</td>
</tr>
<tr>
<td>RECOMMENDED REPAIR:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VALVE Actuator</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>POSITIONER Actuator</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>I/P Transducer</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 14

284
Figure 17

Valve Actuator Travel - IN

Actuator Net Pressure in psig

92 LBS
Seal Load @ Test: 690 LBS
Bench Set: 2.59 - 16.65 psig
Total Travel: 0.847 IN
Spring Rate: 128 Lb/in
Minimum Friction: 96 LBS
Maximum Friction: 151 LBS
Average Friction: 105 LBS

Dynamic Scan 60 Seconds
Flowmeter Diagnostic Service
FISHER DIAGNOSTIC SERVICES
03/27/1993 10:59:34
mcl33 1234
Figure 18
Figure 21
Figure 22

Valve Diagnostic Crew

MAINTENANCE SUPERVISOR

MOV CREW

AOV CREW

FOREMAN

FOREMAN

6 I/C TECHS

7 I/C TECHS

MAINTENANCE ENGINEERING

HX CREW

FOREMAN

FOREMAN

9 MECHANICS
Preventative Maintenance 328 Valves - 413 Scans
Bruce A - AOY Maintenance Program Failure Rates
BRUCE A
OPERATIONS DEPARTMENT PROCEDURE

TITLE: AIR OPERATED VALVE (AOV) MAINTENANCE PROGRAM

0.0 REVISION ABSTRACT
This is a new procedure (DRAFT).

1.0 SCOPE
This procedure defines the Air Operated Valve (AOV) Program and is applicable to all air operated valves at Bruce A.

2.0 PURPOSE
The purpose of this program is to provide a system for monitoring the condition and performance of station AOV's and to perform required AOV predictive/preventative/corrective maintenance in order to achieve the following goals:

a) a reduction in maintenance costs associated with AOV repair through early fault detection and accurate diagnosis of AOV problems.

b) accurate, repeatable and traceable calibration of AOV's.

c) improve worker safety and reduce radiation dose received by maintenance staff by focusing on predictive maintenance.

d) to maintain a trending database of AOV performance for the purpose of predictive maintenance.

e) improved control loop performance.

f) optimization of AOV spare parts.
Section 3.1.7 of the QA Manual specifies that a maintenance program will be implemented to ensure the reliability and effectiveness of equipment and systems. Air Operated Valve maintenance is part of this maintenance program.

4.0 RESPONSIBILITIES

The AOV program utilizes a dedicated maintenance crew with an engineer from Maintenance Support acting as the program coordinator. A Control Maintenance SMS will perform the program supervisor duties since AOV maintenance is of a predominately control nature, however, both control and mechanical maintainers will be required to work together to achieve the goals of the program.

In order to optimize manpower and consolidate valve maintenance expertise, the AOV crew will act as part of a larger AOV/MOV/HX Crew (see below for crew structure). Within this crew there will be control maintainers assigned full time to AOV diagnostics/repair and mechanical maintainers available to provide mechanical support for valve repairs. Operator assistance will be required to provide equipment isolation during AOV diagnostics and maintenance.

Valve Diagnostic Crew

```
       SMS Control
          -------- Mtce Support
          
AOV Crew       MOV Crew       HX Crew
          
1 Sr C/M  1 Sr C/M  1 UTS II

           6 C/M          7 C/M          9 M/M
```

A comprehensive list of duties is as follows;
4.1 Production - Maintenance Support

The role of the maintenance engineer is to;
- implement the AOV program.
- initiate the purchase, repair and fabrication of equipment necessary for the development of the AOV program.
- organize maintenance personnel training on AOV related issues (ie. diagnostic testing, repair methods)
- create an AOV test database for storing and trending test results.
- monitor and trend AOV performance.
- collect technical information related to AOV design, application, testing, maintenance, and operating experience and establish valve operability requirements.
- develop maintenance practices and diagnostic techniques by maintaining contacts with external organizations (ie. other utilities, equipment manufacturers, EPRI, INPO, AOV Users Group).
- develop and implement field testing/inspection methodology.
- provide program information reports to management as required.
- monitor and assess program effectiveness.

4.2 Production - Control Maintenance

- provide and supervise a group of qualified technicians for AOV set-up, diagnostic testing, inspections and maintenance.
- perform initial evaluation of diagnostic test results.
- route diagnostic results to the Maintenance Support program coordinator.
- document all work performed on AOV's using the Fisher Flows scanner Data Disks, the BIMS Work Management System, the Q&A AOV database and AOV calibration sheets.
- prepare/revise maintenance procedures for diagnostic testing and maintenance activities.
- ensure that sufficient parts are ordered from stores to complete AOV maintenance activities. Notify the AOV coordinator if stock levels require adjustment.

4.3 Production - Mechanical Maintenance

- provide and supervise a group of qualified maintainers for AOV internal inspections, repairs and maintenance.
document all work performed on AOV's using the BIMS Work Management System.
- prepare/revise maintenance procedures for maintenance activities.
- ensure that sufficient parts are ordered from stores to complete maintenance activities. Notify the AOV coordinator if stock levels require adjustment.

4.4 Production - Operators

- provide control panel and field assistance to maintenance crews for AOV testing and maintenance.

4.5 Technical - Responsible System Supervisors

- inform the AOV coordinator of upcoming changes to systems which will affect AOV operation and diagnostics.
- assist with AOV root cause and performance analysis.
- assist the Maintenance and Planning sections in the execution of testing, inspection, and maintenance.

5.0 PROGRAM DESCRIPTION

The AOV Program is a predictive/preventative maintenance approach intended to ensure the optimum performance of station AOV's is achieved in the most cost efficient manner. To accomplish this goal, the program has been built around the use of non-intrusive testing systems that facilitate AOV calibration and diagnostics. These systems allow non-intrusive determination of the mechanical and control condition of air operated valves. As a result, maintenance effort can be focused on the valves with real problems. At this time, the Fisher Flowscanner diagnostic system is used.

5.1 Diagnostic Equipment

The Flowscanner is a portable, battery powered diagnostic system that tests and analyses the performance and condition of air operated valves and their accessories. Valves can be tested without being removed from the system and at any system pressure. The Flowscanner measures or generates the control signal to a valve and then records all associated pressures (I/P, positioner, supply pressure) and the valve stem travel. An on-board 386 CPU records all
measured data points (8000 per scan), which can be plotted and analyzed on a built in LCD screen. The graphical displays include plots of travel vs time, pressures vs time, and pressures vs input signals. Built in software also allows manipulation of the data to determine valve characteristics such as bench set, spring rate, seat load and linearity.

Overall, the Flowscanner will be used to troubleshoot problem valves, calibrate and post maintenance test valves after maintenance, and test critical valves on a routine basis as part of predictive/preventative maintenance.

5.2 Equipment Listing

As indicated in the scope, this program applies to all AOV's within the operating control of Bruce A Operations. The existing Q&A AOV databases will be used as a reference for the AOV's covered by the program.

5.3 Equipment Performance

The Flowscanner will be used to evaluate the following characteristics of AOV performance:

a) seat load
b) packing friction
c) bench set
d) spring constant
e) valve travel
f) I/P functionality
g) positioner functionality
h) air supply pressure

5.4 Equipment Performance Standards and Targets

Each AOV will be evaluated against design and manufacturer documentation. This includes Ontario Hydro Control Valve specifications, AECL Design Specifications, valve manufacturers information, design drawings and operating documentation. The measured valve parameters in the previous section will be compared with this information and where deficiencies exist, the appropriate maintenance action will be initiated. Before each valve test, the Flowscanner will be loaded with the data (ie. spring rate, bench set, packing friction, operating conditions, supply pressure) from the appropriate information source (see Appendix 1 for sample valve nametag information).
5.5 Inspection, Testing and Maintenance Requirements

The AOV program will cover three main categories AOV maintenance. This includes troubleshooting of problem AOV's, routine testing/inspections of AOV's under the call-up system and calibration of AOV's following maintenance and installation.

5.5.1 Calibration

The Flowscanner will be used to calibrate AOV's as per the specifications set out in design information. The results will be recorded on the Calibration records in the Control Maintenance shop and in a Q&A database. This will include calibration of AOV's following maintenance as part of post maintenance testing requirements.

5.5.2 Troubleshooting

The Flowscanner will be used to troubleshoot problem valves as per the work flow shown in section 5.6.1. Troubleshooting can occur with the system operating or shutdown and may include dynamic, static or step tests as necessary to correctly diagnose the valve problems.

5.5.3 Valve Inspections

With the Flowscanner it is possible to assess the internal condition of the valve without opening it for inspection. For this reason, internal inspections will only occur if the Flowscanner indicates internal degradation of the valve. When an AOV is opened, the following components should be inspected for wear, corrosion and erosion:

a) valve seat
b) valve disc, plug, trim.
c) stem
d) valve body (erosion must less than 10 % of wall thickness).

The condition of elastomers should also be inspected (O-rings, diaphragm), however, changeout of these items will be included in regular preventative maintenance tasks to maintain them in a "like new" condition.

In addition to internal inspections, whenever an AOV is visited, the external conditions of the valve and actuator should be inspected. This includes signs of mechanical damage, tightness of joints, valve stem cleanliness, gland/bonnet leaks, condition of air

2_08_0
supply lines and the condition of associated actuator accessories (positioner, regulators, solenoids, I/P).

### 5.6 Inspection and Testing

The call up system will be used to schedule and specify the required testing frequency for AOV's.

#### 5.6.1 Call-Up System

The Reliability Centered Maintenance (RCM) group is currently in the process of identifying the AOV's critical to safe station operation. As each AOV is identified, a call-up is prepared to have the valve "flowscanned" at periodic intervals (usually 1 to 3 years). The results of the Flowscan will then be used to identify required maintenance for the valve. In addition to the Flowscan requirement, the call-ups also specify minor PM tasks such as elastomer and relay replacement.

Feed back from the AOV program will be used to revise time based maintenance activities for AOV's (i.e. elastomer replacement, relay replacements, packing changes). Those AOV's deemed not to be critical to station operation will not have call-ups generated (i.e. run to failure, Flowscanned and repaired).

### 5.7 Maintenance

#### 5.7.1 Work Flow

This section describes the flow of Maintenance work (Control and/or Mechanical) for AOV diagnostics and repairs. Maintenance activities may include valve repair/replacement, replacement of consumables (gaskets, elastomers), repair/replacement of accessories (I/P, positioner) and re-packing. After maintenance, the AOV should be returned to service in an "as new" condition, which meets or exceeds the original valve specifications.

**Corrective Maintenance**

(refer to flowchart #1)

a) a work package is initiated which identifies a deficiency with an AOV.

b) the DR will be identified as a Shutdown or Non-Shutdown job, or as an Emergency Work Package (formerly called a Crossed DR).
c) in the case of any Emergency Work Package, "Planning" will not be involved. This will be the exception, not the rule.

d) if a shutdown is not required, any Emergency Work Package will be assigned to Maintenance (Control and/or Mechanical). Whether it is assigned to the duty shift or Valve Diagnostic Crew will be determined on a case-by-case basis.

e) if the job is not Emergency work, Planning (Maintenance Assessor) will assess and assign the work package to the Valve Diagnostic Crew.

f) the Duty Shift Maintenance personnel will still perform trouble-shooting and maintenance on valves as required.

g) if the Duty Shift performs the work, it is still essential that the Valve Diagnostic Crew take an As Left signature with the Flowscanner in order to meet post maintenance testing requirements and to keep performance trending data current and accurate.

h) upon completion of repairs, set-up and testing, Maintenance personnel will complete a WMS work report and file a calibration record stating As Found and As Left results. This documentation will be evaluated on a continuing basis by the PM Engineer for trending and analysis.

Preventative Maintenance
(refer to flowchart #1)

a) One component of the Station Preventative Maintenance (PM) program is AoV diagnostic testing.

b) Callups created for valve PM will specify diagnostic testing, using the Flowscanner.

c) The maintenance assessor will assign these callups to the Valve Diagnostic Crew.

d) Based on displayed results, the Valve Diagnostic Crew will determine (based on training and experience) what Maintenance, if any, is required on the valve.
AOV Deficiency
- CORRECTIVE MAINTENANCE

Work Package initiated

Emergency Work Package

No → Planning

Yes → Valve Diagnostic Team Available

No → Duty Shift Crew Troubleshooting

Yes → Valve Diagnostic Crew Tests
Valve using Flowscanner
- obtain As Found signature

Valve repaired and tested

As left valve signature, if possible

Maintenance Work Report and Calibration Sheets

END

FLOWCHART #1
If no faults are identified, the performance "signature" generated by the Flowscanner will be used as a reference against subsequent diagnostic tests to determine valve wear or performance degradation trends.

Upon completion of repairs, set-up and testing, Maintenance personnel will complete a WMS work report and file a calibration record stating As Found and As Left results. This documentation will be evaluated on a continuing basis by the PM Engineer for the Feedback Trending and Analysis program.

5.7.2 Valve Packing

The current station initiative to eliminate Asbestos and Teflon Packing (for nuclear systems) and replace it with Graphite packing has the potential to impact on the operation of AOV's due to the substantially higher packing friction forces developed with graphite packing. In order to ensure valve operability is not compromised, a list of valves which require packing changes will be developed. It will include valves in nuclear systems which currently have teflon or asbestos packing and chronic leakers in other systems. Each valve on the list will be analyzed to determine if a packing change is acceptable given the current operating parameters of the valve (i.e. system pressure, spring constant, air supply pressure). For those valves which may require re-design, the RSS and Bruce design will be contacted to determine what, if any, changes may be required or permitted.

5.8 Predictive Analysis Methods

As the quantity of performance data accumulates for each valve, it will become possible to utilize predictive maintenance analysis techniques to predict valve degradation and to develop time directed preventative maintenance activities. A Q&A database will be developed to record data for this purpose. Potential methods of predictive maintenance analysis include trend analysis, pattern recognition, correlation, tests against limits or ranges, relative comparison data and statistical process analysis. These methods will be utilized on a periodic basis to evaluate the critical AOV's identified by RCM analysis.

5.9 Quality Control

In order to be confident in the results obtained from
the Flowscanner, it must be made a traceable piece of test equipment. The Bruce A Cal lab will calibrate the Flowscanner's pressure transducers, voltage and current source to the following minimum accuracies:

Pressure Transducers - 0.015% or 0.45 psi
Current Source - 0.01% or 0.6 milliamps
Voltage Source - 0.001% or 0.001 volt

In addition to the physical equipment, quality control is also required for the software used within the Flowscanner. The AOV program coordinator will ensure that the correct version of the Flowscanner is installed on the machine and will implement and record updates. To ensure software quality, Fisher Controls performs hand calculations to verify the output of each version of Flowscanner software. Fisher Controls is in the process of developing Flowscanner software which meets industry standards on software quality control. When it is made available, this software will be evaluated for use at Bruce A.

Data used by the Flowscanner must also be quality controlled. All data input into the Flowscanner valve nametag will be verified by the AOV program coordinator to ensure that the valve is correctly evaluated. The data on the valve nameplate will also be verified against the Flowscanner data during the field test. Any discrepancies between the field and the Flowscanner will be resolved by the AOV program coordinator.

Quality control of output data is required to ensure that historical data is not lost. All test results will be backed-up to ensure data integrity.

Calibration records stored on the LAN in the Q&A database and kept in the control maintenance shop will be verified by the Senior Control Tech responsible for the AOV control technicians. A sample Q&A calibration record is shown in Appendix 2.

5.10 Documentation

The results of all Flowscans will be documented on floppy disks, BIMS work reports, Q&A calibration records and control maintenance shop calibration records. The floppy disks will be stored in a secure location in the AOV shop. Copies of recent Flowscans will be sent to the AOV program coordinator on a periodic basis to create back-up copies on a tape drive. In order to ensure that the most efficient and secure means are used for storing Flowscanner data, the
AOV Program Coordinator will work with the Information Systems group to develop suitable storage mediums for Flowscanner data (i.e. LAN based, CD ROM).

In addition to the data collected from the Flowscanner, documentation standards are also required for valve packing recommendations, information reports to management, recommendations to the RSS on potential valve upgrades, and databases. Q&A databases will be developed to compile AOV failures data for use in predictive maintenance analysis.

6.0 PROGRAM IMPLEMENTATION

Implementation of the AOV program began with a pilot study initiated in 1991. Since that time, the AOV crew has been established and trained, two Flowscanners have been purchased, a substantial volume of test data has been collected, and partial databases have been set-up. At this stage in the program implementation, roles and responsibilities will be formalized to coordinate the activities already underway.

Program targets for 1994/95 include:

- AOV packing evaluation completed
- all critical AOV's identified
- Q&A failure trending database set-up
- spare parts review completed
- 400 valves scanned per year
- yearly program reports
- all AOV Crew Control Maintainers fully trained on the Flowscanner (Fisher Control's Basic and Advanced courses).

6.1 Training

One of the most important factors to ensure the success of the AOV program is the need for trained and experienced personnel. Required training in the use of the Flowscanner will be provided through current the AOV crew personnel and Fisher Controls. Valve maintenance, inspection and functional training will be provided on an as required basis through WNTD and valve manufacturers.

6.2 Program Development/Improvements

The effectiveness of the AOV program will be reviewed two years after implementation to assess the program strengths and weaknesses and to set new goals. Potential program improvements include the addition of
quick disconnects to AOV's to allow quick Flowscanner set-up, additional valve inspection requirements (ie. non-intrusive flow measurement, ultrasonics), process loop tuning and dynamic testing of valves.

7.0 SPARE PARTS

To ensure the effectiveness of the AOV program, a spare parts review will be initiated to ensure optimum quantities of spares are available for required maintenance. This review will include an update of the ESP system to identify required spares and to set optimum stocking levels. The review will be a joint effort of the AOV suppliers, the Materials Management Unit and the AOV Program Coordinator. The initial review will then be followed up with periodic reviews as program experience dictates.

8.0 REFERENCES

A New Predictive Maintenance Technique For Preventative Maintenance Programs, B. Ferguson, D. Cooper, 1992.

Control Valve Diagnostics and Testing at Ontario Hydro BHGS 'A', B. Ferguson, D. Cooper, 1991


Air Operated Valve Maintenance and Test Program, Point Beach Nuclear Plant, Draft Copy, 1993.

BGA-EPM-01-0 Bruce A Check Valve Program

BGA-EPM-02-0 Bruce A Motor Operated Valve Program

BGA-INF-09130-2 Bruce A Information Report, The Use of the Fisher 'Flowscanner': An Air Operated Valve Diagnostic Tool

BGA-PRSP-0.0.09-2 Control Valves and Associated Instrumentation

BGA-STPR-1.5.23 Equipment Programs


<table>
<thead>
<tr>
<th>Plant Site: Bruce A</th>
<th>Description</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>*Tag#</th>
<th>Serial#</th>
</tr>
</thead>
<tbody>
<tr>
<td>Body</td>
<td>IN</td>
</tr>
<tr>
<td>Body Class</td>
<td>LB</td>
</tr>
<tr>
<td>Trim</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>*Flow</th>
<th>*Flow Pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Port</td>
<td>Dia. IN</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>*Unbalanced Area</th>
<th>SqIn</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Actuator (PgDn,PgUp)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spec. Benchset: PSIG</td>
</tr>
<tr>
<td>*Effective Area SqIn</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Accessories (PgDn,PgUp)</th>
</tr>
</thead>
<tbody>
<tr>
<td>I/P Range PSIG</td>
</tr>
<tr>
<td>Input *Resistance</td>
</tr>
<tr>
<td>Positioner Type</td>
</tr>
<tr>
<td>*Zero Control Signal</td>
</tr>
<tr>
<td>Other</td>
</tr>
</tbody>
</table>

| F1 For Help - F10 to Save | ALT+Letter for Menus | Changes Fields |

**Appendix 1:**

**Valve NameTag**
## Valve Name Tag Information

<table>
<thead>
<tr>
<th>USI</th>
<th>LOCATION EL</th>
<th>COLUMN</th>
<th>DATE TESTED</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>W.P.#</th>
<th>FLOW SHEET</th>
<th>FS</th>
<th>ELEC.DWG #</th>
<th>SHT</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SCN:</th>
<th>VALVE SER #</th>
<th>AIR SUP</th>
<th>PSI SYS PRESS</th>
<th>MPA</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>VALVE STYLE:</th>
<th>VALVE TYPE:</th>
<th>BODY SIZE:</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>TRIM:</th>
<th>PORT DIA:</th>
<th>UNBALANCED AREA:</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>STEM DIAMETER:</th>
<th>PACKING:</th>
<th>AIR OPEN/CLOSE:</th>
<th>EFFECTIVE AREA:</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ACTUATOR:</th>
<th>INSTRUMENTS:</th>
<th>Additional Information</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

## Overall Valve Response

<table>
<thead>
<tr>
<th>USI</th>
<th>W.P.#</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Total Travel</th>
<th>Expected</th>
<th>As Found</th>
<th>As Left</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>in/deg</td>
<td>in/deg</td>
<td>in/deg</td>
</tr>
<tr>
<td>(DN) Fully at</td>
<td>ma</td>
<td>ma</td>
<td>ma</td>
</tr>
<tr>
<td>(UP) Starts to at</td>
<td>ma</td>
<td>ma</td>
<td>ma</td>
</tr>
<tr>
<td>(DN) Starts to at</td>
<td>ma</td>
<td>ma</td>
<td>ma</td>
</tr>
<tr>
<td>(UP) Fully at Dynamic Error Band(Avg)</td>
<td>ma</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Linearity</td>
<td>+/-</td>
<td>+/-</td>
<td>%</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>TRACEABLE TEST EQUIPMENT</th>
<th>TECHNICIAN:</th>
<th>VERIFIED:</th>
</tr>
</thead>
<tbody>
<tr>
<td>-C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-C</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Appendix 2: Calibration Record
### I/P PERFORMANCE

<table>
<thead>
<tr>
<th>USI</th>
<th>W.P.#</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Expected</td>
</tr>
<tr>
<td>Pressure at</td>
<td>mA</td>
</tr>
<tr>
<td>Pressure at</td>
<td>mA</td>
</tr>
<tr>
<td>Pressure at</td>
<td>mA</td>
</tr>
<tr>
<td>Dynamic Error Band (Avg)</td>
<td>%</td>
</tr>
<tr>
<td>Linearity</td>
<td>+/-</td>
</tr>
</tbody>
</table>

**Additional Information**

**TRACEABLE TEST EQUIPMENT**

- C
- C
- C

TECHNICIAN:

VERIFIED:

### POSITIONER PERFORMANCE

<table>
<thead>
<tr>
<th>USI</th>
<th>POSITIONER</th>
<th>W.P.#</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Expected</td>
<td>As Found</td>
</tr>
<tr>
<td>Total Travel</td>
<td>in/deg</td>
<td>in/deg</td>
</tr>
<tr>
<td>&gt; (DN) Fully at</td>
<td>psig</td>
<td>psig</td>
</tr>
<tr>
<td>&gt; (UP) Starts to at</td>
<td>psig</td>
<td>psig</td>
</tr>
<tr>
<td>&gt; (DN) Starts to at</td>
<td>psig</td>
<td>psig</td>
</tr>
<tr>
<td>&gt; (UP) Fully at</td>
<td>psig</td>
<td>%</td>
</tr>
<tr>
<td>Dynamic Error Band (Avg)</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Linearity</td>
<td>+/-</td>
<td>%</td>
</tr>
</tbody>
</table>

**Additional Information**

**TRACEABLE TEST EQUIPMENT**

- C
- C
- C

TECHNICIAN:

VERIFIED:

---

**Appendix 2:**

Calibration Record (cont)
### VALVE & ACTUATOR FORCES/TORQUES

<table>
<thead>
<tr>
<th></th>
<th>Expected</th>
<th>As Found</th>
<th>As Left</th>
</tr>
</thead>
<tbody>
<tr>
<td>SL</td>
<td>Avg. Friction</td>
<td>lbs</td>
<td>lbs</td>
</tr>
<tr>
<td>ID</td>
<td>Max. Friction</td>
<td>lbs</td>
<td>lbs</td>
</tr>
<tr>
<td>ES</td>
<td>Min. Friction</td>
<td>lbs</td>
<td>lbs</td>
</tr>
<tr>
<td>TE</td>
<td>Spring Rate</td>
<td>lbs/in</td>
<td>lbs/in</td>
</tr>
<tr>
<td></td>
<td>Bench Set</td>
<td>-</td>
<td>psi</td>
</tr>
<tr>
<td>EM</td>
<td>Seat Load</td>
<td>lbs</td>
<td>lbs</td>
</tr>
<tr>
<td></td>
<td>Serv. Seat Load</td>
<td>lbs</td>
<td>lbs</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>Expected</th>
<th>As Found</th>
<th>As Left</th>
</tr>
</thead>
<tbody>
<tr>
<td>RT</td>
<td>Avg. Torque</td>
<td>in-lbs</td>
<td>in-lbs</td>
</tr>
<tr>
<td>TA</td>
<td>Max. Torque</td>
<td>in-lbs</td>
<td>in-lbs</td>
</tr>
<tr>
<td>AR</td>
<td>Min. Torque</td>
<td>in-lbs</td>
<td>in-lbs</td>
</tr>
<tr>
<td>RY</td>
<td>Spring Rate</td>
<td>lbs/in</td>
<td>lbs/in</td>
</tr>
<tr>
<td></td>
<td>Bench Set</td>
<td>-</td>
<td>psi</td>
</tr>
<tr>
<td></td>
<td>Additional Information</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

**Appendix 2:**
**Calibration Record (cont)**
UTILITY AND DESIGNER EFFORTS TO IMPROVE FRENCH MOV NUCLEAR POWER PLANTS

MM. COPPOLANI*, GRENET**, RENAUDIER**, UHART**

* FRAMATOME
** ELECTRICITE DE FRANCE
1 - INTRODUCTION

Nuclear plants use a large number of valves (roughly 15,000 valves in a French 1300 MWe plant). Fortunately, only a few hundred are important for safety. But these last valves are usually operated by actuators which have been regarded up until recently as components of minor interest. Main efforts regarded the valve itself. Recent unexpected events in Movs in French and foreign NPPs require utilities to look in depth at actuators related to the valves and to change our way of thinking.

This paper, after giving some examples of Movs failures, describes French EDF and FRAMATOME actions carried out, first, to analyse phenomenons and, secondly, to improve safety on plants in this area.

2 - SOME UNEXPECTED EVENTS ON MOV'S

- In April, 1991, during a periodic test on CSS system, at Bugey plant, a 300 mm diameter gate valve operated by an electric actuator failed to open although there was no pressure in the pipe where it was fitted.

- Some years ago, test loops were performed to simulate a pipe rupture in a 250 mm gate valve. This valve failed to close completely during the first tests. To achieve the closure, the torque had to be increased from 46 daN x m to 68 daN x m.

In fact, these two incidents have completely different sources and solutions to solve them are different too. The first incident is related to the introduction of pressure in the valve body whereas the second one can be explained by actuator undersizing.

3 - PRESSURE LOCKING EFFECTS

We have identified three causes to explain pressure locking effects.

3.1 - THERMAL HYDRAULIC BINDING

This phenomenon is very well known: when a gate valve is closed while cold water is flowing in the pipe, water is locked inside the body and between the two discs. Then, if this valve is heated by the upstream fluid, this makes water inside the body expand; consequently, the pressure raises dramatically and can reach very high level. Some tests performed at EDF laboratories show that the rate of increasing pressure can be 1 bar per degree Celsius. Moreover, the pressure is applied on both upstream and downstream discs. As a result, the actuator fails to open the valve. This effect is called "Thermal hydraulic binding".

It is important to notice that heating can be provided in three ways: with the upstream fluid (even if it is a dead leg because turbulences); with the opening of the bypass valve to heat up the downstream piping; also with the environment, after a LOCA for example.

This effect, as the other ones, is directly linked to the tightness of the valve. The tighter it is, the higher the pressure.
3.2 - Differential Pressure Locking

The second cause of unexpected enclosing pressure in the valve body can be explained as follows: a closed gate valve is not completely tight in the upstream side; so, the pressure can enter the body. Later, if the upstream pressure is decreasing, (because a pump stops for instance) the body pressure can be trapped in the valve. In the same way as before, the actuator can fail if this pressure is higher than this taken into account in the design. This effect is called "Differential pressure locking".

3.3 - Piston Effect

The extent of the last effect depends on the technology of the valve, the fluid and the speed of the closure. When a valve is going to be closed, the flow path is cut and the tightness reached while there is small distance still to be covered before achieving complete closure. This small distance is related to the introduction of a small volume of the valve stem. This volume is sufficient to increase the pressure body.

This effect is heavily dependent on the local geometry of the discs and the seats. It also depends on the speed of the closure: if the speed is very high, the overpressure can occur before the complete cut of the path flow (because the volume introduced by the stem can be higher than the water volume which can be released during the same time through the remaining opening between the discs and the seats).

Besides, this effect cannot occur if the valve is closed while the fluid is steam. This effect is called "Piston effect".

4 - Analysis and Solutions for Pressure Locking Effects

It can be easily seen that both first and second pressure locking effects can lead to a high differential pressure between the body and the pipe. Subsequently, they cannot be taken into account in the actuator design. On the other hand, the third effect can be evaluated by testing; the tests have shown that, for some technologies, the rise in the pressure is not very high (i.e. a few bars). Then, this last effect can be introduced in the actuator sizing.

Therefore, depending on the effect, the solutions set up to solve the problems of overpressure have to be different and are described in the next section.

4.1 - Solution against Thermal Hydraulic Binding and Differential Pressure Locking Effects:

If possible, the most simple solution is to make a small hole in the seat or in the disc in order to set up a connection between the valve body and the pipe.

For valves required to be tight in one direction only, this is a good solution because the second disc/seat insures tightness.

Another solution to prevent these harmful effects consists in an external bypass line between the valve body and the pipe. This bypass is equipped with a manual isolation valve to be able to cut the flow completely, if necessary.

This solution was preferred to a spring-loaded safety valve fitted on the housing because this last component leads to an overpressure when it operates and one have to take it into account in the actuator sizing.

For valves required to be tight in both directions, the bypass is connected to both upstream and downstream pipes and a passive selector valve turns the overpressure to the lowest pressurised side.
In order to reduce cost modifications, functional studies have been carried out to target the valves which must be modified. Only gate technology is concerned and some valves don't require modifications if one be sure that the overpressure cannot occur (a valve which remains open while and after a LOCA for instance).

4.2 - SOLUTION AGAINST PISTON EFFECT

Tests performed to know the overpressure piston effect have shown that it's relatively low, especially for the wedge gate valves. For these last valves, the tightness on the upstream side is achieved almost at the end of the closure; so, the stem travel left to carry out before finishing the closure is small and, consequently, the piston effect too. Then, it can be ignored or introduced in the actuator sizing.

5 - ACTUATOR SIZING

For sizing an actuator, one has to know resistant forces but also actuator torque. Both have been reviewed by EDF and FRAMATOME.

5.1 - RESISTANT FORCES

To achieve the calculus of the total force required during the closure of a gate valve, several basic forces have to be taken into account:

- **stem packing force**: some years ago, EDF performed important tests in this area; these tests were carried out with real packing, stem steels and tight forces packing. Therefore, these forces are well known; we use the following formula to compute the resistance force packing:

  \[\begin{align*}
  &\text{if } 1.5 \, P \leq 100 \text{ bar} & F_1 &= 10 \times S \\
  &\text{if } 1.5 \, P > 100 \text{ bar} & F_1 &= \frac{P}{100} \times 1.5 \times 10 \times S \\
  \end{align*}\]

where \( F_1 \) is in daN,

\( P \) is the maximal operating pressure

and \( S \) is the area of the stem packing sliding on the stem (cm²).

- **piston effect force**: it is easy to know and there is no doubt about the result:

  \[ F_2 = P \times S_2 \]

\( S_2 \) being the stem cross section area;
- discs/seats force:
  This is the key point in the calculus; usually the most simple formula used is
  the following:

  \[ F_3 = P \times S_3 \times f \]

  where \( S_3 \) is the area of the path flow and \( f \) the friction coefficient between the
disc and the seat.

  The problem is with the value of \( f \) because it isn't really a constant. Besides, it
  is very difficult to evaluate it by computation because disc and seat during
closure are not parallel and the angle between both always changes (due to
functional clearances). Therefore, it has to be evaluated by testing.

  Tests performed by EDF on site and on test loops have shown that, except in
the case of high pressure pipe break a suitable value for \( f \) is 0.4.

- safety margins:

  Besides, to be sure that the computations are an envelope value for the
resistant force, one must take into account some other small forces and
uncertainties (setting torque limiter switches, effort to achieve wedging...). That
is why EDF and FRAMATOME add a percentage from 10 to 30 % to the
previous calculations to obtain the total real resistance force; so:

  \[ F = A (F_1 + F_2 + F_3) \]

  Where \( A = 1.15 \) to 1.3

5.2 - Actuator Torque

Actuators on French NPPs are provided only by one supplier. Each actuator has a
guaranteed torque. In this area too, some tests have been carried out by EDF and
they have shown that the real torque delivered by the actuator is higher than this
which is guaranteed. Thus, the actuator torque taken into consideration is now the
real torque measured by testing.

On the other hand, tests are under way to see if this torque is still provided when the
environment temperature increases (i.e. after a LOCA).

5.3 - New Rules for Actuator Sizing

Adding improved better resistant forces and actuator torque knowledge, the new
rules for actuator sizing include other parameters like:

- bolt/nut friction coefficient; it has been shown, by testing, that 0.15 is a suitable
  value;
- mechanical efficiency: it is taken equal to 0.9 except when a remote
  mechanical operation is fitted; then the value is 0.8;
- the voltage taken into account is the minimum one which can occur i.e. 0.85 of
  the nominal voltage.

The new rules of sizing have been applied to all safety valves; some of them have
their setting torque switches to be reviewed.
6 - OTHER ACTIONS

Among the other actions carried out to improve level of safety, one can say:

- the checking of setting torque limiter switches on all safety valves on site;
- the complete review of the test procedure for normal operating conditions qualification, especially to improve actuator performance knowledge, including mechanical ageing;
- the rules to shunt the torque switches have been reviewed too; generally speaking, the torque limiter is shunt to open the valve and operates to close it;
- the development of a diagnosis system: SAMIR.

7 - CONCLUSION

Most of the actions planned have been achieved.

Among the actions under way, the most important is probably to find a convenient and reliable test to be sure that the real valve with the real actuator which are fitted on the plant are able to fulfil their function for the all conditions for which they were designed.

A great effort has been made by EDF, FRAMATOME valves and actuator suppliers to improve safety level on MOVs and one can say that the informations gathered especially through the tests has changed our way of thinking about valve actuator importance: it is no longer a secondary component but an essential piece of the valve.
MOTOR-OPERATED
VALVE REVALUATION PROGRAMME

PRESENTATION AT THE
SPECIALIST MEETING ON MOTOR
OPERATED VALVE ISSUES
OEDC NUCLEAR AGENCY

PARIS 25-28 APRIL 1994

VICTOR BARBERO  IBERDROLA
JAVIER MARTIN  NUCLENOR
EDUARDO CUETO  ALMARAZ N.P.P.
1. **INTRODUCTION**

Spanish nuclear power plants are obliged to comply with the regulations of their country of origin. The publishing in the United States of Generic Letter 89-10 has therefore resulted in the implementation of actions to demonstrate the operability of safety-related motor-operated valves.

In Spain, six of the seven nuclear power plants in operation were built with American Technology.

The plants represented here are Almaraz, Sta. Mª Garoña and Cofrentes.

Almaraz is a Westinghouse-supplied PWR and has two 930 MW units. Sta. Mª Garoña and Cofrentes are BWR’s that were supplied by General Electric. The former has one 460 MW unit and the latter, one 990 MW unit.

We shall divide the programme into three major blocks, as shown in Figure 1.

1. Calculations.
2. Grouping and Testing.
3. Checks and Adjustments.

2. **CALCULATIONS**

Figure 2 shows a flow-diagram of the activities.

The principal activities are: definition of scope, review of valve operating conditions, calculation of necessary stresses, actuator capacity, valve structural capacity, setting of margins and determination of setpoints.

2.1. **SCOPE**

The first step is to define the valves to be reviewed.

The scope includes all active safety-related valves. In PWR plants this includes any valves whose change of position might affect operation of the safety-related system and which lack any blocking system to prevent accidental operation from the control room.
Almaraz, Garoña and Cofrentes have a scope of 123x2, 772 and 162 valves respectively, distributed as shown below.

<table>
<thead>
<tr>
<th>TABLE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supplier</td>
</tr>
<tr>
<td>ALMARAZ 1</td>
</tr>
<tr>
<td>Babcock W.</td>
</tr>
<tr>
<td>Kerotest</td>
</tr>
<tr>
<td>Gimpel</td>
</tr>
<tr>
<td>TOTAL</td>
</tr>
<tr>
<td>ALMARAZ 2</td>
</tr>
<tr>
<td>Babcock W.</td>
</tr>
<tr>
<td>Kerotest</td>
</tr>
<tr>
<td>Gimpel</td>
</tr>
<tr>
<td>TOTAL</td>
</tr>
<tr>
<td>S.M. GAROÑA</td>
</tr>
<tr>
<td>Walthon WP</td>
</tr>
<tr>
<td>Crane</td>
</tr>
<tr>
<td>Sulzer</td>
</tr>
<tr>
<td>TOTAL</td>
</tr>
<tr>
<td>COFRENTES</td>
</tr>
<tr>
<td>Babcock W</td>
</tr>
<tr>
<td>Masonian</td>
</tr>
<tr>
<td>Dewrance</td>
</tr>
<tr>
<td>Amvi</td>
</tr>
<tr>
<td>TOTAL</td>
</tr>
</tbody>
</table>
2.2 REVIEW OF DESIGN BASES

The next step is to review the valve operating conditions, which makes it necessary to establish the circumstances in which it is required to operate.

The situation that maximises differential pressure is selected and the flow under these conditions is calculated.

These calculations are made both for the opening and closing movements and it is also necessary to determine whether the safety function is related to opening or closure, or both.

2.3 CALCULATION OF REQUIRED STRESS

Next it is necessary to calculate the stresses required to activate the valves. At the PWR plants, Westinghouse made the calculations for its valves and Siemens performed the calculations for the remaining valves, using its own methodology.

At the BWR plants all calculations were carried out by Siemens.

The calculations gave the torques and thrusts required for the valve to open and close.

2.4 ACTUATOR CAPACITY

The next phase is to verify that the structural capacity of the actuator is in excess of the capacity required to activate the valve. This information may also be obtained directly from the manufacturer. It is also necessary to check that the actuator is sufficiently dimensioned to give the necessary torques and that the operating time is less than that required for the valve.

Compliance with these conditions will result in the actuator being valid to actuate the valve in question.

If the actuator is not valid, it will be necessary to analyze the possibility of downgrading the process conditions. Should this not be possible, the actuator will be replaced or modified.

Typical modifications include modification of gears, change of spring pack and change of motor.
2.5 VALVE CAPACITY

The next step is to check the valve's capacity to bear the actuation stress by analyzing the weakest points of the stress transmission chain. If any point does not bear the stress, the process conditions are reanalysed to see if they can be downgraded. If this is not possible, modifications are made to the incorrectly dimensioned components such as bolts, stem or yoke, or a new valve is installed.

2.6 MARGINS AND ADJUSTMENTS

To set the valve torque switches the lower and upper limits of the valve must be set. The lower limit is the limit required for the valve to open or close and is obtained from the stress calculation.

The upper limit will be the lowest of the three following values: actuator structural limit, maximum torque given by the actuator and valve structural capacity.

The torque switch setpoint must be between these margins.

3. GROUPING AND TESTING

Very often it is not possible to demonstrate the operability of the valves by performing tests in conditions of maximum flow and differential pressure, and pipe break simulations are never possible. Furthermore, valves will probably be damaged if they are tested under off-normal operating conditions. As a result, although a valve might fulfill its mission by operating once, after the test it could suffer permanent damage that would prevent it from operating again in conditions of maximum differential pressure and flow.

To reduce the number of tests, valve operability should therefore be justified by a combination of analytical methods and tests.

Figure 3 is a flow-diagram of activities. The main activities are: compilation of data, grouping of valves, identification of valves to be tested, localization of information or performance of on-site or laboratory tests, interpretation of tests and application of results to the group, review of calculations and valve validation.

3.1 DATA

The first step is to compile the data necessary. This fundamentally comprises the process conditions, internal measurements of valve components, and material information.
3.2 GROUPING

The methodology used by several Spanish power plants, which has been developed by SIEMENS, consists of forming groups of valves having common characteristics: same manufacturer, same design, same pressure class, and same materials. The operation of the valve throughout its stroke is analyzed, considering both the stability of the disk and its possible interaction with the seat in intermediate positions, and stresses and deflections in guides and seat.

Figure 4 is a typical graph in which disk stability and turning torque, increase in differential pressure and disk-seat friction stress, are seen in relation to the stroke.

Figure 5 is a comparison of stresses for a family of valves, and shows that the 8 and 14 inch valves bear the greatest levels of stress and are candidates for testing. The information obtained from the tests will be applicable to the other valves of the group.

Specifically, the following items will be analyzed:

A. Gate valves.

- Flow contact and cutoff.
- Axial and lateral disk tilt angles.
- Disk stability point in seat and body guides.
- Point of contact between disk and seat.
- Increase in differential pressure during stroke.
- Disk tilting moment.
- Body guide deflection.
- Disk deflection.
- Average and local bearing stress between disk-seat and disk-body guides.
- Bearing stress between disk and seat.
- Bending stress in disk upstream and downstream of flow.
- Bending stress in disk and body guides.
- Stress in body guide welds.
- Interaction between body guides and disk.

B. Globe Valves.

- Tensile/compressive stress in the stem.
- Bearing stress between stem/key or yoke/pin.
- Additional friction force of antirotation device.
- Disk guiding mechanism.
3.3 TESTING

After selecting the test candidates it is necessary to determine whether any test information is applicable to these valves.

The tests may be obtained from the programme developed by EPRI or from certain manufacturers, test laboratories or other plants.

The information is analyzed to confirm whether the materials and dimensions match our designs and whether the fluid flow and differential pressure conditions are applicable.

This situation would be the easiest to resolve because with the existing information it would be possible to confirm the validity of the valve calculations or to revise them.

Should it not be possible to locate valves of the same design tested under the applicable conditions, the possibility of performing tests at the plants is analyzed. The tests will be conducted under the conditions most similar to the accident conditions postulated for operation of the valve.

The test procedures indicate the conditions under which the tests must be performed, such as:

- Alignment of other valves.
- Equipment necessary.
- Installation of instrumentation.
- Permanent instrumentation to be used.
- Systems for recording process variables.
- Diagnosis equipment.
- Valve actuation mode.

When it is not possible to simulate the process conditions at the plant, it will be necessary to compare the characteristics of our test candidate valves with other tested valves. By reviewing aspects such as:

- Materials
- Stem-disk connection
- Guide configuration
- Shape of disk

and comparing the valves on the basis of the aforementioned operability analyses, it will be possible to validate application of the results of the tested valves to our valves.

Should the design and testing conditions of the valves tested by others not match the conditions of our valves, laboratory tests will be necessary.
3.4 EVALUATION OF RESULTS

Analysis of the test data or results is facilitated by preparation of evaluation procedures. These include standardised collection of necessary data, and the calculation procedure.

The test results are extrapolated to all the valves of the group and are compared with the theoretical results that would be obtained by using the calculation methodology.

Valves are considered valid if the theoretical results cover the test results with a margin in excess of 10%. Otherwise engineering justifications must be implemented to validate the valves and, if this is not possible, the valves must be recalculated with more conservative coefficients.

4. CHECKS AND ADJUSTMENTS

A fundamental task of valve maintenance is checking the state of the valve and the setpoints (Fig.6).

Firstly, it permits identification of any problems in the valve and the need for some type of repair.

Secondly, the stress observed will give an indication of some its components, such as: piston effect, packing load or inertia. It will also confirm some of the values used in the calculations and whether or not they should be reviewed.

4.1 DIAGNOSIS

Valve diagnosis consists of determining the state of the valve and actuator without the need to disassemble them, measuring:

Opening and closing torque and/or thrust.
Motor power or intensity and voltage.
Switch actuation.
Movement or turning of torque switch.

Present diagnosis equipment measures stress with strain gauges with a level of accuracy in excess of ±10%.

4.2 ACTUATOR ADJUSTMENT BENCH

This device is an electric brake that provides a resisting torque that increases linearly with time. When it is coupled to an actuator the device makes it possible to see the values at which the opening and closing torque switches trigger and to adjust them the desired value.
It must be used to perform adjustments to valves where it is not possible to use a thrust or torque meter.

The precision of the equipment is far in excess of that of diagnosis equipment and the adjustments are not affected by the ROL.

4.3 ROL

ROL is a change in the ratio between the force transmitted to the stem and the movement of the spring pack, depending on whether the load is applied to the actuator gradually or suddenly.

The phenomenon is the result of a reduction of the stem nut-stem friction coefficient when rapid load increases occur, such as upon closure of a gate valve without differential pressure.

In all the American-built plants in Spain, valves close with torque switches and open with limit switches, so this phenomenon will occur during testing and adjustments made without differential pressure.

4.4 SWITCH ADJUSTMENT

Switches may be adjusted in three ways, as follows:

1. Actuator coupled to the valve and test without flow and ΔP.
2. Actuator coupled to the valve and test with flow and ΔP.
3. Actuator coupled to a test bench.

Ideally, the adjustments should be made in conditions as similar as possible to those in which the valve is required to operate, because the adjustment will avoid the ROL and at the same time, valve functionality will be verified. This is not always possible as a result of test implementation problems.

The adjustment without ΔP permits the state of the valve to be determined in a single operation but does not eliminate the ROL, and this must be taken into account in the adjustments.

The third alternative is to adjust the torque switches on a test bench. This solution avoids the ROL and also provides much higher levels of precision than the diagnosis equipment.

Consequently the best combination for performing the initial valve adjustments and reviews is adjustment of torque switches on a calibration bench and diagnosis of the valve without differential pressure.
4.5 DIAGNOSIS PROGRAMME

The initial review and adjustments will be implemented during three outages, and the state of the valves will be reviewed in 5-year periods or three outages.

The current tendency of the plants is to reduce the duration of the outages, so the implementation of valve diagnosis may form part of the critical path in these outages.

An efficient diagnosis is obtained by using the electrical magnitudes as a reference of the stresses and adjustments. This may be done from the MCC, thus avoiding the problems associated with having to take the diagnosis equipment to the valve and radiological dose problems.

Hence any valve that can be actuated during operation can be diagnosed without having to wait for an outage.

To use diagnosis based on measurement of electrical parameters, it is necessary to correlate the intensity or power consumed by the motor with the stem thrust. This entails a prior strain-gauge diagnosis, which will give the thus and power graphs in the testing conditions.

The correlation between stem thrust and power is good in tests performed under similar conditions, so integration of the information obtained from bench adjustments and strain-gauge diagnosis, and the measurements of electrical parameters will make it possible to relate parameter evolution with stress. Any abnormal variation will indicate a need to perform a more accurate diagnosis and to readjust the switches.

To facilitate valve diagnosis, it is very useful to install fast-fitting connectors containing not only outside power and control cables but also any other control logic points that identify opening and closure of contacts and which should be monitored. The use of permanent stress-measuring instrumentation is better still, of course, but would have a greater economic impact.
5. CONCLUSION

The two major problems for the three plants are justification of valve operability and reduction of diagnosis time.

As regards the first issue, the most feasible solution for justification involves functional analysis of the valves, determining which have the greatest internal loads and validation of the latter by the use of in-house or external tests.

As regards the second issue, the first diagnosis should consist of strain-gauge measurement of stress and switch adjustment. Following this first diagnosis, the most efficient method for determining the state of the valve is measurement of electrical magnitudes. A more accurate diagnosis should only be conducted if some type of anomaly is detected in the interpretation of the electrical data.
VALVE REVALUATION PROGRAM

FIGURE 1
CALCULATIONS

DEFINE SCOPE

DESIGN BASES REVIEW

REQUIRED STRESS CALCULATION

CHECK ACTUATOR CAPACITY

IS THE ACTUATOR CORRECTLY DIMENSIONED?

YES

CAN THE DESIGN BASES BE MODIFIED?

YES

CHECK WEAK POINTS OF VALVE

NO

MODIFY ACTUATOR COMPONENTS OR REPLACE IT

NO

MODIFY VALVE

IS THE VALVE CORRECTLY DIMENSIONED?

YES

SET STRESS MARGINS FOR VALVE

MAKE ADJUSTMENTS

FIGURE 2
COMPARISON OF MAXIMUM AVERAGE STRESSES FOR
6", 8", 12", 14", AND 16" VALVES

C.S. = CARBON STEEL
S.S. = STAINLESS STEEL

LOCAL DISK-GUIDE BEARING STRESS
DISK-SEAT BEARING STRESS
DWNSTRM. DISK BENDING STRESS
BODY GUIDE BENDING STRESS
DISK GUIDE BENDING STRESS
CHECKS AND ADJUSTMENTS

DIAGNOSIS AND ADJUSTMENTS

ARE THERE ANY ANOMALIES IN THE VALVE?

REPAIR

EVALUATE RESULTS

CALCULATIONS IN DIAGNOSIS CONDITIONS

IS THERE A MARGIN?

VALIDATE

JUSTIFY OR RECALCULATE

FIGURE 6
OPTIMIZING SAFETY BENEFITS IN ASSURING THE PERFORMANCE OF MOTOR-OPERATED VALVES

COMMITTEE ON THE SAFETY OF NUCLEAR INSTALLATIONS
PRINCIPAL WORKING GROUP No 1

JOINT SPECIALIST MEETING
ON
MOTOR-OPERATED VALVE ISSUES

PARIS, FRANCE
APRIL 25th-27th, 1994

R. Clive Callaway
Nuclear Energy Institute, (NEI) Inc.
1776 Eye Street, N.W., Suite 300
Washington, DC 20006-3706
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td></td>
</tr>
<tr>
<td>Introduction</td>
<td>1</td>
</tr>
<tr>
<td>2.0</td>
<td></td>
</tr>
<tr>
<td>Purpose</td>
<td>1</td>
</tr>
<tr>
<td>3.0</td>
<td></td>
</tr>
<tr>
<td>Approach</td>
<td>2</td>
</tr>
<tr>
<td>4.0</td>
<td></td>
</tr>
<tr>
<td>Safety Enhancements</td>
<td>2</td>
</tr>
<tr>
<td>4.1</td>
<td></td>
</tr>
<tr>
<td>Direct Safety Enhancements</td>
<td>2</td>
</tr>
<tr>
<td>4.2</td>
<td></td>
</tr>
<tr>
<td>Indirect Safety Enhancement</td>
<td>3</td>
</tr>
<tr>
<td>5.0</td>
<td></td>
</tr>
<tr>
<td>Risk Ranking of MOVs</td>
<td>3</td>
</tr>
<tr>
<td>5.1</td>
<td></td>
</tr>
<tr>
<td>Choice of Ranking Method</td>
<td>3</td>
</tr>
<tr>
<td>5.2</td>
<td></td>
</tr>
<tr>
<td>Completeness Issues</td>
<td>4</td>
</tr>
<tr>
<td>5.2.1</td>
<td></td>
</tr>
<tr>
<td>PSA Truncation Limits</td>
<td>4</td>
</tr>
<tr>
<td>5.2.2</td>
<td></td>
</tr>
<tr>
<td>Initiating Events, &quot;Supercomponents,&quot; Operator Errors, and Multiple Failure Modes</td>
<td>5</td>
</tr>
<tr>
<td>5.2.3</td>
<td></td>
</tr>
<tr>
<td>MOVs Not Modeled in PSA</td>
<td>6</td>
</tr>
<tr>
<td>5.2.4</td>
<td></td>
</tr>
<tr>
<td>Level 2 PSA MOVs</td>
<td>6</td>
</tr>
<tr>
<td>5.2.5</td>
<td></td>
</tr>
<tr>
<td>Common Cause Failures</td>
<td>7</td>
</tr>
<tr>
<td>6.0</td>
<td></td>
</tr>
<tr>
<td>Importance Categories</td>
<td>8</td>
</tr>
<tr>
<td>6.1</td>
<td></td>
</tr>
<tr>
<td>Number of Categories</td>
<td>8</td>
</tr>
<tr>
<td>6.2</td>
<td></td>
</tr>
<tr>
<td>Boundaries Between Categories</td>
<td>8</td>
</tr>
<tr>
<td>6.3</td>
<td></td>
</tr>
<tr>
<td>Other Considerations</td>
<td>9</td>
</tr>
<tr>
<td>7.0</td>
<td></td>
</tr>
<tr>
<td>Applying a Graded Approach</td>
<td>10</td>
</tr>
<tr>
<td>7.1</td>
<td></td>
</tr>
<tr>
<td>High Priority MOVs</td>
<td>11</td>
</tr>
<tr>
<td>7.2</td>
<td></td>
</tr>
<tr>
<td>Medium Priority MOVs</td>
<td>12</td>
</tr>
<tr>
<td>7.3</td>
<td></td>
</tr>
<tr>
<td>Low Priority MOVs</td>
<td>13</td>
</tr>
<tr>
<td>7.4</td>
<td></td>
</tr>
<tr>
<td>Post Maintenance and Periodic Testing</td>
<td>13</td>
</tr>
</tbody>
</table>
1.0 INTRODUCTION

For the past several years, both the United States nuclear industry and the U.S. Nuclear Regulatory Commission (USNRC) have devoted significant attention and resources aimed at improving the performance of motor-operated valves (MOVs). Generic Letter (GL) 89-10 recommends that safety-related MOVs be analyzed and tested where practicable to demonstrate their functionality under design basis conditions. The recommended MOV testing includes static baseline testing to set up the MOV actuators and dynamic flow testing to verify functionality. The GL also recommends that MOVs be periodically tested to verify that control switch settings are being properly maintained and that tracking and trending MOV performance be conducted. The GL further recommends that the analysis and initial testing be completed by June 1994 and that the periodic testing be performed on a five-year or three refueling outage frequency for all MOVs, unless a longer interval can be justified.

Clearly, the level of attention and resources given to MOVs has resulted in an improved understanding of the design, operation, and maintenance of these components. The enhanced knowledge of these types of valves provides the engineering basis for maintaining reliable performance over their service lives. The accumulation of this knowledge, however, has come at a tremendous expense that continues to absorb industry and regulatory attention and resources. Given that the contribution to safety of individual MOVs varies widely, a number of questions have been raised concerning whether this expenditure is resulting in commensurate safety benefits and to what extent this expenditure should continue.

The underlying principle of this document is to apply resources in a manner that is commensurate with the safety significance of individual MOVs. This approach should not only lead to increased levels of confidence in plant safety, but should also result in a more efficient use of industry and regulatory resources.

2.0 PURPOSE

The purpose of this document is to outline a graded approach that responds to the GL 89-10 recommendations while optimizing the safety benefits in assuring MOV performance. It is not aimed at reducing the number of MOVs within the scope of GL 89-10.

This document provides policy-level guidance for structuring a graded approach that is founded on a blend of probabilistic and deterministic methods. It is intended for the voluntary use of nuclear utilities for consideration in managing their efforts to close out GL 89-10 and beyond. This guidance does not preclude closure of GL 89-10 by other means.
3.0 APPROACH

The approach consists of three main steps:

1. Rank MOVs according to their importance to safety using probabilistic safety assessment (PSA) techniques (Section 5.0);

2. Prioritize MOVs into importance categories using a blend of (PSA) ranking information and deterministic insights (Section 6.0); and

3. Apply specific requirements to each group of MOVs (Section 7.0).

This approach has similarities to the approach adopted by the industry in responding to the Maintenance Rule (see NUMARC 93-01, Industry Guidelines for Monitoring the Effectiveness of Maintenance at Nuclear Power Plants). Once the initial valve testing phase of GL 89-10 has been completed, it is intended that MOV periodic testing efforts be folded into the plant’s existing programs used to implement the Maintenance Rule.

The three-step approach embodies the underlying principle noted earlier in that greater attention and resources would be devoted to those MOVs of higher importance. Correspondingly, the groups comprise MOVs that make smaller contributions to safety would command a lesser degree of attention and resources. The benefits of a graded approach are numerous and are discussed in Section 4.0.

4.0 SAFETY ENHANCEMENTS

Several observations have been made to support the conclusion that safety would be enhanced, directly and indirectly, by applying a graded approach to assuring MOV performance that is founded on both probabilistic and deterministic insights.

4.1 Direct Safety Enhancements

Greater attention and resources devoted to the high priority MOVs could translate into many direct safety enhancements. First, schedules could be structured to test this group of valves earlier than the other groups of the lower priority MOVs. Thus, the timeliness of any modifications or adjustments identified by analysis of the test results would be improved. Second, this group of valves could be subjected to, where practicable and meaningful, more frequent periodic tests than the lower priority groups. Again, the timeliness of any problem identification and resolution would be improved. Third, requirements associated with the high priority group of MOVs are expected to be more
rigorous and demanding in nature than for the other groups. This provides added assurance that any problems that may impact the functionality of the valves will be identified and resolved.

4.2 **Indirect Safety Enhancements**

There are important indirect safety benefits to this three-step approach as well. Safety analyses identify the safety significant valves and the impact of their potential failure on plant safety. In addition, these analyses identify important scenarios that provide information with respect to the operational demand that may be placed on a given valve. Such information is valuable because it relates the performance of the important MOVs to the broader context of plant safety. This allows more rational decision-making, more efficient use of resources, and is central to optimizing safety benefits.

The use of safety analyses may identify safety significant MOVs that are not within the scope of GL 89-10. It is recommended that as part of a plant’s overall safety mission, such significant MOVs should be evaluated for possible addition to a plant’s MOV program. It can be legitimately argued that, conversely, when the analyses identify MOVs with very low safety significance, they should be removed from the scope of GL 89-10. PSA perspectives, however, are a relatively new factor in implementing this program. Thus, it is conservatively recommended that such low safety significant MOVs remain within the scope of the GL 89-10 effort. Their lesser contribution to safety would be reflected in their initial test schedules, type of tests, frequency of periodic testing and other application parameters.

5.0 **PSA RANKING OF MOVs**

PSA ranking is the first step in determining the relative importance of MOVs. Considerations for ranking should include the choice of PSA techniques, truncation levels, and methods to accommodate common cause failures.

5.1 **Choice of Ranking Method**

**RECOMMENDATION:** For determining the relative safety significance of MOVs, any well known ranking method may be used, as long as it is done in a consistent and technically correct manner.

There are a number of different risk indices or importance measures by which MOVs may be ranked. These include Risk Achievement Worth (RAW), Fussel-Vesely (F-V), Risk Reduction Worth (RRW) and others.
A number of comparisons have been made to determine whether ranking methods result in markedly different placements of the relative importance of individual MOVs. It has been concluded that many ranking methods yield comparable results. In fact, if the failure probabilities of all the components in a given ranking are the same, then the RAW and RRW techniques would result in the same ranking. If vastly different failure probabilities among MOVs are used (e.g., from plant-specific data), then greater care should be used in the choice of ranking techniques. A utility’s choice of a ranking method might also be determined by the ease in which its software accomplishes a particular ranking task. One sensitivity analysis, which used MOV failure rate as a variable to determine its influence on core damage frequency (CDF), resulted in identifying the same safety significant MOVs as did a standard ranking process. In this case, and perhaps more generally as well, proper identification of the safety significant MOVs did not rely on the use of a standard ranking method.

Small differences between the relative contribution of core damage of MOVs in one ranking method versus another are less of a concern because the valves are subsequently grouped into similar importance categories. Therefore, it is likely that if two different ranking methods resulted in somewhat different rankings of MOVs, the same valves would occupy the same importance categories.

5.2 Completeness Issues

Several issues should be considered in order to comprehensively identify and appropriately rank MOVs using (PSA) techniques and other methods.

5.2.1 PSA Truncation Limits

RECOMMENDATION: Truncation limits should be considered when constructing the list of safety significant MOVs.

In order to be confident that certain important MOVs are not truncated out of a PSA, the relationship between truncation level and the number of MOVs above that level should be investigated. Truncation issues vary in significance from one ranking method to another, e.g., they are less important for Fussel-Vesely unless the truncation limits are such that a significant percentage of the CDF is not accounted for in the cutsets.

Rather than performing sensitivity studies on the relationship between truncation level and the total number of MOVs to be considered, it may be possible to cause some otherwise excluded MOVs to be identified by assuming a high MOV failure rate and then proceeding with ranking and categorizing.
From an overall safety perspective, truncation may not be an important issue. Those MOVs that might be identified at very low truncation levels or through high assumed failure rates are likely to be of low safety significance. The information currently available indicates that the bulk of the safety significant MOVs is determined by the small population of MOVs that is in the higher safety category. Truncation effects are most likely to be of concern when the effects of multiple valve failures are considered such as in common cause evaluation or in a bounding analysis where a number of valves of low individual importance is assumed to fail.

5.2.2 Initiating Events, "Supercomponents" (Independent Sub-Fault Trees), Operator Errors, and Multiple Failure Modes

RECOMMENDATION: The utility should identify MOVs that may be “masked” within initiating events or within supercomponents and account for the valve importance appropriately.

There are several mechanisms by which the true importance of an MOV may be affected. These should be accounted for when ranking MOVs.

Initiating events may include certain MOV failures. The linking of the initiating event importance to CDF may not be automatic depending on the PSA software used. One may need to extract the role of an MOV from the initiating event.

Sometimes independent sub-fault trees are modularized, i.e., used as “supercomponents,” and treated as basic events in the analyses. The PSA should be reviewed to identify any MOVs included in such modules so that their importance may be evaluated. In some situations, operator error may dominate a particular event and MOV failure may not be explicitly modeled. Operator error rates consistent with the PSA may be used to evaluate MOV importance. Where operator recovery actions have been modeled in a PSA and this enhances the importance of an MOV or introduces an MOV not explicitly modeled, this greater importance should be reflected in the MOV’s ranking.

Some MOVs may have more than one important failure mode (e.g., in one scenario it might be failure to open and in another it might be failure to close). The overall importance of an MOV should be determined by considering all pertinent failure modes.
5.2.3 MOVs Not Modeled In PSA

RECOMMENDATION: Review the scope of GL 89-10 for MOV’s not modeled in the PSA. Based on the MOV’s importance, determine if it should be added to the PSA. If an MOV is unimportant, place it in the lowest priority category.

Just as there may be MOVs in a plant’s PSA that do not appear within the GL 89-10 scope, there can be valves within the scope that do not appear within a plant’s PSA. Such valves should be reviewed to determine if they belong in the PSA. If so, then they should be added to the PSA and their importances determined using the ranking methods discussed previously.

For all other MOVs within the scope of GL 89-10, but not modeled in the PSA, deterministic methods should be used to establish their relative importance. Section 6.3 provides further guidance on using deterministic methods.

5.2.4 Level 2 PSA MOVs

RECOMMENDATION: The methods used and the results of analyzing MOV’s identified by Level 2 PSAs should be described when ranking and categorizing such valves. Those MOVs in a given safety category identified by a Level 2 PSA analysis should be added to the MOV’s from Level 1 PSA analyses that are also in that category. Valves in the GL 89-10 scope that are not important to large releases should be placed in the category of lowest importance. When an MOV has been categorized by both Level 1 and Level 2 PSA analyses, the requirements of the higher safety category should be applied.

All discussion up to this point has been on Level 1 PSA analysis, i.e., core damage frequency. Some MOVs are important because of the roles they play in Level 2 PSA analysis (containment failure, source term).

At this time ranking technology is much more developed for Level 1 PSA analysis than for Level 2 analysis. Nonetheless, those PSAs with Level 2 analyses can establish analogous rankings. Containment event trees can be developed and quantified concurrently with the Level 1 PSA event trees. The quantification of the Level 2 trees with the Level 1 trees allows all systems to be included in the generation of basic event importances for the containment failure frequency and for the large release frequency.
Various analyses have shown that large releases, though infrequent and of low probability, tend to dominate offsite consequences. Therefore, those MOVs identified by Level 2 analysis may be ranked according to their importance to large release frequency only.

Safety or priority categories, discussed later, should be populated by MOVs that derive safety significance from both Level 1 and Level 2 analyses. One approach is to set the MOVs that are important to the top "x" percent of the large release frequency into the high priority category. Here "x" is the same percentage of the core damage frequency that the high safety category covers for Level 1.

Various other methods, including engineering judgment, can be used to identify and place Level 2 MOVs into appropriate safety categories.

5.2.5 Common Cause Failures

**RECOMMENDATION:** The impact of potential common cause failures should be considered in establishing the safety significance of MOVs. If the potential for common cause failure is credible based on the plant's physical design and response, an assessment of the scenarios should be performed to adjust the importance ranking of MOVs, as appropriate.

A number of ranking studies sponsored by the NRC and the industry have examined the effects of the failure rate or unavailability of individual MOVs, i.e., perturbations taken one valve at a time. Using the standard ranking methods identified above, it has been observed that the most highly ranked MOV, when unavailable, does not usually have a large impact on raising the baseline CDF. This can be attributed to the redundancy and diversity designed into all nuclear plants.

More recent studies sponsored by the NRC and industry have begun to investigate the impact of multiple MOV failures due to common cause events. Common cause failures are examined when developing a PSA. However, typical PSA common cause analyses are limited to intra-system analyses, i.e., failures that occur within a single system. The failure of a group of MOVs can result in higher CDF values than the sum of the individual contributions to CDF. These same studies also indicate that the particular combinations or configurations of MOVs that could have high contributions to CDF are but a small portion of the total MOV population. The particular MOVs and configurations of importance are likely to be plant specific and would require some common cause event to make them simultaneously unavailable. Further, when factoring in differences between MOV design, size, function, operating conditions, actuation times
and location within the plant, the likelihood of a common cause event rendering several
valves inoperable simultaneously is reduced. Thus, the results of these studies may
converge to identify the same safety significant MOVs derived from standard ranking
processes.

Inter-system failures, i.e., failures that involve more than one system, are not usually
modeled for the physical differences cited above. One approach to examining inter-
system common cause failures of MOVs would assign a higher failure rate/demand, such
as 0.087, to those MOVs that would have to operate in a given accident situation under
“high dp” conditions. The failure rate of 8.7 percent is consistent with the assumed
failure rate in NUREG/CR 5140. The resultant impact on CDF could then be used to
compare the importance of these groups of MOVs with their importance based on the
individual MOV ranking. Adjustments to which priority category the MOV is assigned
could then be made, as appropriate.

6.0 IMPORTANCE CATEGORIES

The purpose of ranking MOVs according to their importance to safety is to assign specific
requirements according to safety significance. Since there are many MOVs within the
scope of GL 89-10, it is impractical to develop a different set of requirements for each
MOV. Therefore, once MOVs have been ranked according to safety significance, it is
useful to group them into priority or importance categories. Each category would then
have a number of distinct requirements, such as initial test schedule, type of test, periodic
testing frequency, etc., associated with it.

6.1 Number of Categories

RECOMMENDATION: Divide the ranked list of MOVs into three or four categories.

There are no rigorous rules that establish the number of categories. Too few categories
can result in a loss of the benefits of gradation of requirements. Too many categories can
make implementation burdensome and the distinction between requirements from
category to category somewhat arbitrary and unimportant. After several trials, experience
indicates that three or four categories are optimum for MOV categorization.

6.2 Boundaries Between Categories

RECOMMENDATION: Choose category boundaries so that completeness issues are
addressed, and so that each category can have distinct requirements assigned to it.
The boundaries between categories is a matter of engineering judgment. Those individual MOVs whose failure would significantly increase the potential for core damage should be placed in the high priority category. Additionally, if the assessment of common cause events resulted in a group of MOVs having a significant impact on CDF, then those MOVs should be added to the high priority category as well. It is suggested that the high priority category have a non-zero population. However, if sufficient justification exists for a particular plant, based on operational or test data, it is possible to have no MOVs that are of high safety significance.

As an example to illustrate the above, one group of utilities who are using a three-category structure and Fussel-Vesely (F-V) as the risk importance measure uses the following categorization criteria:

<table>
<thead>
<tr>
<th>Category</th>
<th>Criterion</th>
</tr>
</thead>
<tbody>
<tr>
<td>High:</td>
<td>F-V &gt; 0.01</td>
</tr>
<tr>
<td>Medium:</td>
<td>0.01 &gt; F-V &gt; 0.001</td>
</tr>
<tr>
<td>Low:</td>
<td>F-V &lt; 0.001</td>
</tr>
</tbody>
</table>

6.3 Other Considerations

**RECOMMENDATION:** Review of the PSA-derived ranking and categorization of MOVs by an independent group of knowledgeable people should be considered. Adjustments to the program based on deterministic insights and engineering judgments to increase scope and/or move MOVs to higher or lower categories should be considered.

The identification of MOVs and their eventual assignment to a safety category has so far been largely based on PSA techniques. However, one should also use non-PSA techniques, e.g., deterministic insights and engineering judgment, to supplement the PSA methodologies.

The PSA-derived ranking and categorization of MOVs should be submitted to an in-house panel knowledgeable of the plant, design basis analyses, accident analyses, and MOV operation for final review and adjustment. The ranking should be accompanied by an explanation of the results in terms of the plant design features and responses to postulated accidents and transients.
Considerations as to why certain MOVs may or may not be important from a deterministic perspective could include:

- **Diversity** - the MOV is located in a system that is functionally redundant to another system having no MOVs.

- **Margin** - the adequacy of the design margin of the MOV may warrant increased or decreased requirements.

- **Configuration** - the MOV is normally in position to fulfill its intended function.

- **Performance history** - the past reliability of the MOV may warrant increased or decreased requirements.

- **PSA scope** - the MOV is important in scenarios that are not explicitly modeled, e.g., external events.

- **Shutdown operation** - the MOVs that could result in loss of decay heat removal, inadvertent loss of inventory, or loss of containment function.

7.0 **APPLYING A GRADED APPROACH**

The implementation of a graded approach is founded on the blend of the probabilistic and deterministic methods used to rank and categorize the MOVs. Many of the deterministic insights result from the valve-specific information derived from the design review and baseline testing of all MOVs within the scope of the program. Specific requirements can then be established for each priority category that provide reasonable assurance of MOV performance commensurate to plant safety for each classification.

Once the design review is performed using best available methods and data, switch settings are established and confirmed through static testing. Available static margin is then determined. Determining design-basis requirements, available margin, and safety significance of the valve, provide a valuable input into the decision making process for determining the desirability for dynamically testing a given MOV. If operability is in question, the licensee may be required by regulation to evaluate the operability of the MOV and take appropriate actions in accordance with plant's Technical Specifications.

In order to verify the performance of those MOVs not dynamically tested and as such close out GL 89-10, sufficient and relevant direct or indirect dynamic test data are needed. Absent test data, appropriate engineering methods can be used to justify switch
settings. Using a graded approach to close GL 89-10, it would be appropriate, if not ideal, to gather in-house test data by dynamically testing the most safety significant valves. For those valves not dynamically tested, engineering assumptions would be applied with conservatism that is proportional to the safety significance of the MOV.

7.1 High Priority MOVs

**RECOMMENDATION:** High priority valves should be dynamically tested if testing is both practicable and meaningful. This testing should be performed at the risk-significant accident conditions, where appropriate.

Since the consequence of failure of a highly ranked MOV could be significant under certain accident scenarios, preference should be given for dynamically testing this group of valves. Where appropriate, valves should be tested at risk-significant accident conditions. This approach to testing is distinctly different from testing valves to extreme conditions that are hypothesized to occur during highly improbable events, such as the instantaneous double-ended guillotine break of the largest primary system pipe. These severe tests of low probability events are of concern because of their potential to degrade MOV reliability during the more likely accident sequences.

The decision to validate MOV performance capability through testing under differential pressure and flow conditions should be considered where:

- The test is practicable, i.e.,
  - The resultant test configuration does not place the plant in a condition outside the limits defined in the plant's technical specifications.
  - The test process will not endanger personnel nor unacceptably affect equipment reliability.
  - The test can be completed using permanently installed plant equipment or temporarily installed instrumentation not requiring system modifications.
  - The process flow rate is representative of the existing system conditions and capability.
The resultant test data will be meaningful, i.e.,

- Test data obtained when the equipment is reasonably challenged and represents the maximum achievable dp which gives extrapolatable data at or near design or risk-significant accident conditions.

- Will provide meaningful validation data of sufficient quality and accuracy to support performance assumptions such as grouping methods or other in-plant modeling efforts.

If the tested valve fails the plant's acceptance criteria, appropriate modifications must be made to meet plant's acceptance criteria. The valve should then be re-tested at similar conditions to verify acceptable performance.

For high priority valves that are not tested at differential pressures, best available analytical methods and data should be used to assess performance capability. A utility should consider relevant data from other plants and the industry sponsored EPRI Performance Prediction MOV Research Program when in house data is not available. If analytical alternatives are used to assess the performance of high priority valves, conservative, if not, bounding values should be used.

### 7.2 Medium Priority MOVs

**RECOMMENDATION:** *Engineering methods or dynamic testing should be considered when validating performance capability.*

The criteria used for deciding whether to dynamically test or modify a medium priority valve need not be as limiting and bounding as those used for valves considered as high safety contributors. However, since validation of analytical methods is desirable, testing medium-priority valves may provide needed in-house data to verify design and performance assumptions.

Considerations and criteria that can be used for determining adequate margin or performance confidence to meet plant acceptance criteria include:

- Valve behavior predictable with reasonable assurance using validated methods; and

- Past performance indicates reliability.
If the margin assessment for medium priority valves meets the plant acceptance criteria, the initial baseline recommendations for GL 89-10 are satisfied.

If reasonable assurance of performance through analytical assessment cannot be made, the decision to demonstrate performance capability through testing under differential pressure and flow conditions should be considered where:

- The test is practicable (defined in Section 7.1).
- The resultant test data will be meaningful (defined in Section 7.1).
- Resources are appropriately applied relative to the safety significance and priority of the MOV, i.e., could resources be better spent on higher priority MOVs or other components.
- Adequate grouping criteria cannot be met.

7.3 **Low Priority MOVs**

**RECOMMENDATION:** Engineering methods can be used to obtain reasonable assurance of performance capability.

For this group of valves, analytical methods are sufficient to assess performance capability. Engineering assumptions should be applied with conservatism that is proportional to the safety significance of the MOV. The intent of this guidance is not to preclude testing low priority valves or to exclude them from the overall scope of GL 89-10. Dynamically testing these valves may serve as a surrogate for testing higher priority valves, validating data to establish grouping methods, or validating analytical assumptions.

7.4 **Post Maintenance And Periodic Testing**

**RECOMMENDATION:** Once the initial design specifications are verified for each MOV within the program, the subsequent surveillance testing frequency can be based on relative safety contribution, performance history and available design margin. Relative safety contribution should also be reflected in the type of post maintenance testing as well as the particular maintenance activity.

Action item "d" in Generic Letter 89-10 recommends that correct switch settings are determined and maintained throughout the life of the plant. Also, action item "j" recommends that after maintenance or adjustments and periodically thereafter MOVs may
need re-testing in order to identify potential degradation or misadjustments. As stated in the generic letter, "The surveillance interval should be based on the licensee's evaluation of the safety importance of each MOV as well as its maintenance and performance history."

The generic letter further recommends that, "The surveillance interval should not exceed five years or three refueling outages, whichever is longer, unless a longer interval can be justified (see item h) for any particular MOV."

Accordingly, the generic letter does not prescribe the manner in which a plant verifies continued correct switch settings, whether after maintenance or periodically. Nor is the frequency a prescribed time interval.

As insights are gained from the ongoing research and in-situ testing, the type of re-testing and the periodicity may need to be adjusted. In addition, diagnostic techniques are evolving to less intrusive and less expensive testing. Plants need not commit to prescribed schedules for periodic testing. The periodic reverifications schedules should be flexible and reflect the priority assigned to the valve groups. High priority valves should be reverified more frequently than the lower priority valves.

Testing considerations relative to post maintenance and periodic testing are different from the initial verification of acceptable design specifications. In general, re-test requirements should be based on the type of degradation over time (e.g., erosion, corrosion, wear, grease hardening, etc.) and relative safety significance.

Once the initial valve testing phase of GL 89-10 has been completed, it is suggested that the MOV periodic testing efforts be folded into the plant's program used to implement the Maintenance Rule. Further guidance for periodic verification of MOV design-basis capabilities can be found in OM -SG, 1990, Part 8, "Startup and Periodic Testing of Electric Motor Operators on Valve Assemblies In Nuclear Power Plants."
Risk Based Approach for Prioritizing Motor-Operated Valves

Gerald H. Weidenhamer *
William E. Vesely **

ABSTRACT

The United States Nuclear Regulatory Commission (USNRC) issued Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance" in June 1989. This regulatory document was issued because of the concern for the performance of motor-operated valves (MOVs) under accident loads. The GL 89-10 requests licensees to develop a plan to assure that safety-related MOVs will accomplish their functions under design basis loads. It also requests licensees to perform design-basis flow tests on the MOVs to establish actual thrust requirements where practicable. In addition, it is recommended that the MOVs be periodically verified to ensure that they will perform their functions throughout the life of the plant.

This paper identifies an approach by which MOVs can be prioritized based on risk importances to fulfill the periodic verification part of GL 89-10. Also, schedules for tests and maintenance of MOVs can be established based on their risk contributions. Those MOVs having the largest impact on plant safety can be grouped and would be reverified (tested) and maintained at more frequent periodic intervals. Those MOVs with lower impacts on plant safety can be placed in a second group and reverified and maintained at longer periodic intervals. A third group of safety-related MOVs would contain those that have the lowest impact on plant safety. This last group would be reverified and maintained at even longer periodic intervals. This approach for prioritizing MOVs and establishing groups for scheduling reverification tests and maintenance intervals can be cost effective and can aid in optimizing resources for implementing GL 89-10 for the remaining life of the plant.

Background

On June 28, 1989, the NRC issued Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance." The reason for issuing GL 89-10 was that assessments of the reliability of safety-related MOVs, based on extrapolations of the available information from valve surveillances at that time, indicated that more than expected MOV malfunctions were occurring and that additional measures should be taken to ensure operation under design basis conditions for the life of the plant. One of the main items identified in GL 89-10 is that licensees need to periodically verify torque switch settings to ensure that MOVs will accomplish their intended functions over the life of the nuclear power plants in the USA.

---

* United States Nuclear Regulatory Commission, Washington, D.C. 20555
** Science Applications International Corporation, 655 Metro Place South, Columbus, Ohio 43017
The intent of periodic verification is to assure that the licensees consider the effects of aging degradation that can cause reduced performance of motors, operators, and/or the valves after long periods. These aging degradations can cause malfunctions and result in premature trips of the MOVs such that discs can stop at partial stroke positions.

The licensees have expended significant resources to implement GL 89-10 and they are investigating ways to make the periodic verification efforts more cost effective. Therefore, in December 1992, some licensees inquired whether the NRC licensing office would be receptive to risk-based approaches for identifying MOVs most important to plant safety. The plan is that most of the resources could be devoted to these most important MOVs. The NRC licensing office subsequently stated that NRC is receptive to risk-based approaches for identifying the most important MOVs; however, NRC also stated that this approach should not be the basis for eliminating any safety-related MOVs from GL 89-10 programs (see Supplement 5 to GL 89-10).

At that time, the NRC Office of Nuclear Regulatory Research was requested to assist the licensing office in evaluating this approach and identify problems that might typically be encountered during the performance of a risk-based analysis.

This paper describes an approach that was used to identify and prioritize the most risk important MOVs in a typical PWR plant as an aid for meeting the periodic verification part of GL 89-10. During the performance of this analysis, problem areas were identified and are listed in the "Conclusions" section of this paper. Other observations were made and these are also reported under "Conclusions."

**Discussion**

In 1990, a methodology* was developed for the NRC Office of Nuclear Regulatory Research that permits time dependent aging rates to be incorporated into typical PRA failure rate models. Current PRA models use constant failure rates for the same components in the entire plant. Since many of the same components (such as MOVs) are scattered throughout the entire plant, it is apparent that not all the same components have the same failure rates. This new methodology permits the inclusion of different failure rates for any of the components. In addition, one of the strong points of this methodology is that the effects of single component aging as well as multiple component aging (due to common cause) and their interactions can also be evaluated by this methodology. Appropriate failure rates can be identified and the effects on changes to core damage frequency (CDF) can be determined. The component(s)

---

having the largest effect on changes in CDF can then be conveniently identified and prioritized. Reductions in core damage due to component replacements or maintenance can also be evaluated using the same methodology. A very important feature of this method is that existing PRA models can be utilized to determine the effects of aging on plant risk with some reservations. (See items 1 through 7 under "Conclusions.")

Analysis

The application of this methodology, including the determination of time dependent failures, can be complicated; however, assumptions can be made to simplify the problem of identifying which MOVs are most important for safety.

Assumptions

As stated above, for a time dependent analysis, the actual failure rates would need to be determined. Instead of determining the actual failure rates, for this analysis it will be assumed that each of the MOVs will fail (failure probability of unity) and the impact that each MOV failure has on the change in core damage is to be determined. Redundant line MOVs can also be assumed to fail.

By assuming a failure probability of unity for a particular MOV, the question being addressed is "How important is this MOV to the safety of the plant?" Since the idea of risk-based prioritization for the application described in this paper is to devote resources to those MOVs that are most important, an assumption such as this, i.e., failure probability of unity, is acceptable and conducting a PRA is less complicated.

Results

An analysis of a typical PWR plant in the USA was performed using the methodology and assumptions identified above. Table 1 lists the MOV identification number and MOV system locations, respectively.

Single MOV Failures:

For this part of the analysis, each MOV is assumed to fail, while all the other MOVs retain their original assumed failure rates equivalent to 0.003 failures per demand. The constant failure rate equivalent to 0.003 failures per demand has been utilized in typical PRAs by the PWR licensees. The effect of a single MOV failure on the change in CDF is then determined. Each successive safety-related MOV is allowed to fail and the impact on change to CDF determined for each MOV failure. After all single MOVs have been evaluated, the same analysis is repeated for multiple MOVs due to common cause.

Table 2 lists those MOVs (based on single failures) having the largest impact on plant safety. Specifically, the table shows that failure of either MOV LPR-1862A or LPI-1890C would increase the baseline CDF by a factor of 5. The baseline CDF is the value calculated when all
components in a typical PRA model are assigned their respective constant failure rates. Failure of the MOVs identified in the remaining list shows that the CDF would increase only by factors of 3 or 4.

Failures of MOV Pairs:

If the analysis were to end at this point, one might conclude that very little "periodic verification" would be required to ensure safety of the plant through its remaining life. Also, the analyst would likely conclude that although an increase of 5 in the CDF is of concern, immediate steps would not be required for periodic verification of the two MOVs to fulfill GL 89-10. The analyst would be correct in interpreting the results; however, the analyst would need to consider the possibility that multiple MOVs are allowed to fail simultaneously. The most simple example of this condition occurs in redundant pipes where safety-related MOVs are installed. The justification for allowing two MOVs in redundant piping to fail simultaneously is that these two valves are likely to be identical - same manufacturer, size, and type (including motor-operator). If the torque switches for one of these MOVs are not set properly to fulfill GL 89-10, it is also highly probable that the torque switches in the other redundant MOV will not be set properly. Also, both MOVs are located in the same environment and both would be affected equally.

Table 3 shows the results of the PRA for the MOV pairs evaluation. The failures of MOVs PPS-1535 and PPS-1536 cause the CDF to increase by a factor of 700. Failures of some of the other MOV pairs on the list also impact the CDF significantly. Other failures of MOV pairs increase CDF by factors less than 5 and are not included in Table 3. Note that the highest ranking pair of MOVs show up in the middle of the list in Table 2. In addition, some MOVs in Table 2 are not identified in Table 3. This case illustrates NRC's major concern with MOV prioritization, i.e., the prioritized list of important MOVs (for a particular nuclear power plant) should be reasonably the same regardless of the method(s) used to develop the list. This means that important MOVs should not be inadvertently excluded from the prioritized list nor should the level of risk importance of a particular MOV be underestimated as would be the case if only the effects of single MOV failures are analyzed.

It is apparent that simultaneous, multiple MOV failures can be serious. Therefore, it would be advantageous for licensees to periodically verify that those MOVs contained in Table 3 are set up properly. The 16 MOVs listed in Table 3 represent about 10% of the safety-related MOVs in a plant. Therefore, a utility might proposed that verifying these 16 MOVs on a more frequent periodic basis would be a reasonable approach to satisfying this part of GL 89-10. A test schedule developed using only Table 2 data may put a plant at high risk if periodic verification is extended to long intervals for the high risk important MOVs.

The failure of MOV pairs was investigated to address common cause effects. If there are redundancies of more than two MOVs in a plant, then higher combinations of MOVs simultaneously failing would also need
to be considered. The methodology identified earlier can be utilized to analyze this latter case.

Modifications to Prioritized List of MOVs

The analysis completed thus far considers failures of single MOVs and pairs of redundant MOVs. For the list shown in Figure 3 to be complete, it would now be necessary to consider cascading effects. Cascading describes the effect that a failed MOV would have on the performance of other components. Therefore, this second component would also be allowed to fail. Since a failed MOV can cause another important component from fulfilling its design basis function, this MOV's impact on CDF should also include the impact of failing the second component. The total importance of this MOV would then be determined from summing the changes in CDF due to failure of both components. To consider these effects, the analyst would assume specific MOVs to fail followed by the influence of each MOV failure on other important components, such as pumps, electrical components, etc. To accomplish this, it would be necessary to consider the design basis conditions for each MOV and the location of the other component (in system or proximity). The analysis would be completed in the same manner as the MOV pairs.

It is conceivable that the resulting changes in CDF due to cascading effects would be as large as or even greater than the values shown in Table 3. If this is the case, it would be concluded that these additional MOVs should be part of the prioritized list.

This latter scenario would be followed until all potential MOV interactions with other components have been evaluated.

Some existing PRA models may not have all MOVs covered by GL 89-10 in their fault/event trees. Therefore, these MOVs will not be included in the PRA model and their effects on CDF cannot be determined. For these cases, the MOVs should be included in the prioritized list. For example, containment isolation (MOVs) may be included in level 2 PRA models which calculate radiological release frequencies, but not in level 1 PRA models which calculate core damage frequency but not releases. For these cases, the MOVs should be included in the prioritized list unless it can be shown otherwise.

The final list of all safety related MOVs can be divided into groups based on their risk importance. The highest priority group would require more frequent periodic verification tests. A second group of MOVs, those having minor impacts on plant risk, could be tested after longer operation intervals. A third group of MOVs, those having low impacts on plant risk, could be tested after still longer operation periods. The grouping described above would provide the basis for proportioning the resources in a cost effective manner. Most of the resources would be devoted to the highest priority MOVs while smaller amounts of resources would be devoted to the lower priority MOVs.
Conclusions

This effort was undertaken to determine whether ranking of MOVs can be accomplished using risk-based technology. To accomplish this, simplifying assumptions were made. For this approach, it became clear that to obtain an accurate list of MOVs, certain guidelines are important. The following guidance identifies those areas that should be considered when MOVs are to be ranked in accordance with this method:

1. The MOVs included in the PRA are only a subset of those addressed by GL 89-10.
2. The particular failure modes addressed by GL 89-10 need to be those covered by the PRA.
3. Common cause failures among MOVs as treated in the PRA may be inadequate for GL 89-10 response.
4. PRA importance prioritizations of MOVs needs to consider joint (multiple) MOV importances.
5. MOVs may be unimportant for core damage frequency prevention but may be important for consequence mitigation, shutdown risk control, or when external events are considered.
6. The cascading effects of MOV failures may not be adequately addressed in the PRA.
7. Truncations in the PRA may cause MOV importances to be underestimated.
8. The MOV prioritization criteria and test scheduling criteria need to account for the associated risk impacts.
9. The prioritized groupings of MOV dynamic tests and MOV test schedules need to be validated for their risk control.
10. To develop a complete resource response to GL 89-10, the PRA evaluations need to be integrated with deterministic evaluations.

Note: The NRC will publish the report of this work in September 1994.

### Table 1: System, Component and Event Identifiers (PWR)

#### System Identifiers

<table>
<thead>
<tr>
<th>System Identifier</th>
<th>System Name</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACC</td>
<td>Accumulators</td>
</tr>
<tr>
<td>HPI</td>
<td>High Pressure Safety Injection System</td>
</tr>
<tr>
<td>LPI</td>
<td>Low Pressure Safety Injection System</td>
</tr>
<tr>
<td>LPR</td>
<td>Low Pressure Recirculation System</td>
</tr>
<tr>
<td>PPS</td>
<td>Primary Pressure Relief System</td>
</tr>
</tbody>
</table>

#### Component Identifiers

<table>
<thead>
<tr>
<th>Component Identifier</th>
<th>Component Name</th>
</tr>
</thead>
<tbody>
<tr>
<td>MOV</td>
<td>Motor Operated Valve</td>
</tr>
</tbody>
</table>

#### Failure Mode Identifiers

<table>
<thead>
<tr>
<th>Failure Code</th>
<th>Failure Mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>FC</td>
<td>Loss of Function</td>
</tr>
<tr>
<td>FT</td>
<td>Failure to Transfer</td>
</tr>
<tr>
<td>PG</td>
<td>Plugged</td>
</tr>
</tbody>
</table>

### Table 2: CDF Importances of Individual MOVs (PWR)

<table>
<thead>
<tr>
<th>Valve Identifier</th>
<th>CDF Relative Importance</th>
</tr>
</thead>
<tbody>
<tr>
<td>LPR-MOV-FT-1862A</td>
<td>5</td>
</tr>
<tr>
<td>LPR-MOV-PG-1890C</td>
<td>5</td>
</tr>
<tr>
<td>ACC-MOV-PG-1865C</td>
<td>4</td>
</tr>
<tr>
<td>ACC-MOV-PG-1865B</td>
<td>4</td>
</tr>
<tr>
<td>LPR-MOV-FT-1860A</td>
<td>3</td>
</tr>
<tr>
<td>LPI-MOV-PG-1890A</td>
<td>3</td>
</tr>
<tr>
<td>PPS-MOV-FC-1535</td>
<td>3</td>
</tr>
<tr>
<td>PPS-MOV-FC-1536</td>
<td>3</td>
</tr>
<tr>
<td>HPI-MOV-FT-1350</td>
<td>3</td>
</tr>
<tr>
<td>LPR-MOV-FT-1862B</td>
<td>1</td>
</tr>
<tr>
<td>LPR-MOV-FT-1860B</td>
<td>1</td>
</tr>
<tr>
<td>HPI-MOV-FT-1115C</td>
<td>1</td>
</tr>
<tr>
<td>HPI-MOV-FT-1115E</td>
<td>1</td>
</tr>
<tr>
<td>LPR-MOV-FT-1890B</td>
<td>1</td>
</tr>
<tr>
<td>HPI-MOV-FT-1115B</td>
<td>1</td>
</tr>
<tr>
<td>HPI-MOV-FT-1115D</td>
<td>1</td>
</tr>
</tbody>
</table>

### Table 3: CDF Importances of Pairs of MOVs (PWR)

<table>
<thead>
<tr>
<th>Valve Pair</th>
<th>CDF Relative Importance</th>
</tr>
</thead>
<tbody>
<tr>
<td>PPS-MOV-FC-1535</td>
<td>700</td>
</tr>
<tr>
<td>LPR-MOV-FT-1862A</td>
<td>80</td>
</tr>
<tr>
<td>LPR-MOV-FT-1860A</td>
<td>50</td>
</tr>
<tr>
<td>LPR-MOV-FT-1860B</td>
<td>50</td>
</tr>
<tr>
<td>HPI-MOV-FT-1115C</td>
<td>30</td>
</tr>
<tr>
<td>HPI-MOV-FT-1115E</td>
<td>30</td>
</tr>
<tr>
<td>LPR-MOV-FT-1890A</td>
<td>30</td>
</tr>
<tr>
<td>LPR-MOV-FT-1860A</td>
<td>30</td>
</tr>
<tr>
<td>HPI-MOV-FT-1115B</td>
<td>30</td>
</tr>
<tr>
<td>HPI-MOV-FT-1115D</td>
<td>30</td>
</tr>
</tbody>
</table>

365
A decade ago, motor-operated valve operating experience pointed out problems with limit and torque switch settings. Since then, it has been recognized that problems existed pertaining to the valve, operator, controls and system applications. The international knowledge of MOV design, operation and maintenance has also significantly improved. The meeting session on research development included 7 papers addressing technical issues and new equipment designed to address MOV issues. The papers, taken as a whole, are illustrative of the growing knowledge and sophistication of the international nuclear community.

An IAEA representative described an overall integrated research program illustrates the perceived need for an integrated holistic approach to MOV issues.

A report on non-linear gate valve response experiments and analysis addressed the issue of valve testing at less than full flow. Results of this work identify gate tilt and mechanical interference due to excessive clearances result in high torques needed to stop flow or fully seat gate valves. Another speaker stressed the need for improved disk factors including consideration of valve improvements and modifications including material changes to minimize tilt and mechanical interference and better control frictional forces.

Another paper described corrosion experiments and the relationship to valve operability issues. Two speakers described advanced motor operator designs.

Finally, a session speaker addressed valve diagnostic equipment based on measurement of torque and thrust as the valve stem. Over time, there has been an evolution of diagnostic equipment to progressively take measurements closer and closer to the actuated component and to develop more non-intrusive equipment.

Taken together, the evolution of knowledge and means of addressing issues is evident.
IAEA CO-ORDINATED RESEARCH PROGRAMME ON THE MANAGEMENT OF AGEING OF MOTOR OPERATED ISOLATING VALVE

Paper for a presentation at the OECD/IAEA Specialists Meeting on MOTOR OPERATED VALVE ISSUES April 25-27, 1994, Paris, France

A. KOSSILOV
International Atomic Energy Agency Division of Nuclear Power Wagramerstrasse 5, P.O.Box 100 A-1400 Vienna, Austria
IAEA CO-ORDINATED RESEARCH PROGRAMME (CRP) ON
THE MANAGEMENT OF AGEING OF MOTOR OPERATED
ISOLATING VALVE

A. Kossilov
International Atomic Energy Agency,
Vienna, Austria

Abstract

Ageing in NPPs must be effectively managed to ensure that required safety margins are maintained throughout plant service life, including any extended life. Many organizations in Member States are developing the necessary technical basis for managing nuclear power plant ageing. Motor Operated Valves (MOV) are extensively used in almost all plant fluid-mechanical systems. Operating experience indicates that the operational readiness of nuclear power plant safety related systems have been affected by MOV degradation and failures. The aim of the IAEA co-ordinated research programme (CRP) on "Management of Ageing of Motor Operated Isolating Valve" is to improve the understanding and management of MOV ageing and thus to help assure required MOV performance. The paper describes a scientific background of the CRP, working plan and results achieved.

1. INTRODUCTION

In December 1992 the Co-ordinated Research Programme on "Management of Ageing of Motor Operated Isolating Valve" was approved as a part of CRPs on "Pilot Studies on Management of Ageing of Nuclear Power Plant Components" [1]. Organizations from Canada, Finland, Germany, India, and Russian Federation have submitted proposals for research agreements in the subject area. Organizations from Czech Republic, France, Sweden, and the United States of America have indicated their interest in the operation and development. Actual participants are organized in a CRP network to facilitate co-operative work.

The objective of the meeting held from 7 to 10 December 1993, Helsinki/Espoo, Finland was to prepare a background and implementation plan for the CRP. National reports on research topics were also presented.

The following objectives for the CRP were agreed:

1. To improve the understanding of MOV ageing mechanisms and effects, and thus help assure the functionality of MOVs under both normal and accident conditions.

2. To identify effective and practical methods for monitoring of MOV ageing capable of timely detection of MOV anomalies attributable to age related degradation.

3. To develop guidelines for risk and reliability assessment of MOV ageing.

4. To improve MOV qualification methods and formulate MOV qualification guidelines.

5. To establish guidelines for effective MOV maintenance to alleviate ageing effects and
The following tasks were suggested for implementation:

- Understanding of MOV ageing
- Monitoring of MOV ageing
- Risk and reliability assessment of MOV ageing
- MOV qualification methods and guidelines
- Guidelines for MOV maintenance.

During the meeting the IAEA document TECDOC-670, "Pilot Studies on Management of Ageing of Nuclear Power Plant Components - Results of Phase I," October 1992 was considered [1]. Chapter 3 in that document contains the results of the initial pilot studies on MOVs. Specifically, information on understanding ageing and methods for mitigating and monitoring ageing of these components is highlighted. The chapter also identifies existing knowledge and technology gaps, and it defines follow-up work that will lead to either eliminating or reducing these gaps. Therefore, chapter 3 provides the technical basis for the implementation of the Phase II of the MOV programme and shall be considered by the CRP members when completing the respective tasks.

2. OPERATING EXPERIENCE

Current knowledge about MOV failures and problem areas has been obtained from plant operating histories and test programmers. In addition to plant failure reports, event and maintenance reporting systems and manufacturers' operating and maintenance manuals and notifications offer information on degradation mechanisms. These degradation mechanisms could lead to failures without adequate in-service test and maintenance programmers. The simulated full scale test programmers conducted by several Member States have shown valve problem areas that could only be identified in a real plant environment if a design basis event were to occur.

The failure data consist mostly of failures that have occurred in test situations (periodic, motion testing of in-service valves) but little knowledge is available from real demand abnormal or accident situations with more severe operational and environmental conditions (e.g. differential pressure, high temperature).

The major problems leading to critical valve failures have generally occurred in the valve actuator, e.g. torque switch malfunction. Among the mechanical valve parts, the stem is the main contributor to critical failures.

Identification of the root causes of failures is in many cases difficult and thus the age-dependency of failures cannot always be defined. As an example, torque switch malfunction may be due to wrong adjustment, drifting, grease hardening, spring loosening, etc. It should be noted that the detection of defects (degraded condition) prior to MOV failures allows better identification of root cause(s) related to ageing.

The list in Table I describes a variety of degradation mechanisms which can result in MOV failures.
<table>
<thead>
<tr>
<th>SITE</th>
<th>DEGRADATION DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gate valve assembly</td>
<td>Gate wear, corrosion&lt;br&gt;Guide wear, corrosion&lt;br&gt;Yoke bushing wear&lt;br&gt;Valve stem wear, corrosion, distortion&lt;br&gt;Fastener loosening&lt;br&gt;Valve seat wear, corrosion&lt;br&gt;Bonnet seal deterioration&lt;br&gt;Stem packing wear, deterioration&lt;br&gt;Valve body erosion</td>
</tr>
<tr>
<td>Gearbox assembly</td>
<td>Gear wear&lt;br&gt;Shaft wear, distortion&lt;br&gt;Fastener loosening&lt;br&gt;Stem nut wear&lt;br&gt;Stem lock nut loosening&lt;br&gt;Spring pack response change&lt;br&gt;Drive sleeve wear&lt;br&gt;Clutch mechanism wear&lt;br&gt;Seal wear, deterioration&lt;br&gt;Bearing wear, corrosion&lt;br&gt;Lubricant deterioration, hardening&lt;br&gt;Stem nut lubrication degradation</td>
</tr>
<tr>
<td>Electric motor</td>
<td>Bearing wear, corrosion&lt;br&gt;Insulation breakdown</td>
</tr>
<tr>
<td>Switches</td>
<td>Contact pitting, corrosion&lt;br&gt;Gear/cam wear&lt;br&gt;Insulation breakdown&lt;br&gt;Parts breaking&lt;br&gt;Fastener loosening&lt;br&gt;Grease hardening</td>
</tr>
</tbody>
</table>
3. CURRENT UNDERSTANDING OF MOV AGEING

The current knowledge on MOV ageing has been obtained primarily from operating experience and qualification programmes. Operating experience with motor operated valves shows that failure mechanisms can be divided into two basic categories:

(i) those failure mechanisms that result in excessive friction such that valve actuator/motor cannot drive the valve open or closed in the required time period

(ii) those failure mechanisms that prevent the valve from being operated over its full stroke.

In category (i) the failure is normally due to:
- lack of adequate lubrication causing wear and "jamming" of valve stem or wear of the bearing
- overtightening of valve packing
- corrosion of bearings, stem guides or valve stems
- deposition of "crud" on valve seat
- motor trip on overload.

In category (ii) the failure is normally due to an electrical fault or malfunction of electrical switch, i.e.:
- failure of switchgear
- failure of limit switch in open or closed position
- faulty torque switch operation.

Another failure in this category might be the uncoupling of motor drive shaft from valve stem (before or after gearbox).

In a failure analysis performed in Finland, of forty-six critical failures occurring over the period considered in the study, about one third were category (i) failures and the remainder were category (ii). A main contributor to the electrical failures was the oxidization of contractors. In a French study, 25% of failures of electric valves were due to the actuator.

Other age related failure causes identified from operating experience in other Member States are listed below:
- DC motor failure due to insulation breakdown caused by electromagnetic spikes
- non-disengagement of hand control device
- total blockage of kinetic transmission
- misadjustment or drift of switches (for example if the torque spring is loaded over a long period of time, the spring takes a permanent set and delivers less torque for a given torque switch setting)
- incorrect sizing of certain elements
- stem nut lubrication degradation.

374
The identification of age related failures from operating experience is often difficult because, due to the complexity of the-valve-actuator system, the root causes attributable to ageing cannot always be identified with certainty. To achieve better understanding of MOV ageing, ageing mechanisms listed in Table I should be further evaluated. The investigation should focus on those parts of the MOV which form the pressure boundary and those parts which are important for valve performance and operability.

4. MONITORING OF MOV AGEING

In-service inspection and testing of MOV actuating systems are undertaken during refuelling and at prescribed intervals in accordance with national standards.

Present inspection and testing methods for verifying pressure boundary integrity are generally accepted as sufficient. However, valves in older plants need to be inspected more extensively and frequently for the existence of minimum wall thickness and for the existence of cracks and abrasion inside the casing than valves in newer plants. Also, provisions for inspecting aged valve bodies in other safety related fluid-mechanical systems must be made to avoid surprises.

Current methods to verify valve capability frequently use valve stroke time as a measure of valve performance but these are of limited value as it takes a large load change in motor output to produce a detectable change in the speed of a typical MOV high torque motor. There is now an effort to develop better in-service test procedures by the use of monitoring systems; current thinking is that the measurement of motor power would be a more useful test than stroke time test. The CEA and EDF are developing an electrical power monitoring system. MOV diagnostics using electrical current signature analysis has been developed in research sponsored by the USNRC at ORNL.

Electrical power and current signature analysis are both qualitative actuator measurements. None of these measurement systems can provide information on the conversion of torque to thrust that takes place in the stem nut, which is outside of the control of the torque switch. The conversion of torque to thrust is dependent on stem nut lubrication and on stem loading. Industry has referred to this phenomenon as a rate of loading effect. This phenomenon can invalidate any in-service test method that cannot load the valve to its design basis load during the in-service test.

The typical in-service test is performed with an unloaded valve (no pressure or flow). Therefore, the load on the actuator is minimal. For limit switch valve control there is no load on the actuating system other than the load due to valve internal friction. For torque switch controlled valves the disc hits the seat and the torque and thrust build up very rapidly (milliseconds). This transient is very difficult for most power measurement systems to follow because of the complex internal voltage, amperage, and wattage calculations. Systems that measure raw peak to peak current and voltage and then process the data through the software after the test may be the best in following the transient. The systems would require at least 1000 Hz sample rate to be of any value.
Developments in monitoring MOV ageing should be aimed at measuring torque and stem thrust along with motor power to perform a complete diagnosis of motor operated valve performance. To make this effort more meaningful, the test should be performed with the highest design pressure and flow load that can be practically obtained on the valve. This is particularly important for limit controlled valves where there is not a seat load. For these valves the analyst may wish to consider motor actuator dynamometer testing. For torque-controlled valves, the seat loads make the test more meaningful.

In-service tests are performed to verify limit switch and torque switch settings which assure valve closure and sufficient thrust. Tests are also performed to detect electrical degradation. Commercial diagnostic systems are now available from several companies which facilitate these tests.

5. MITIGATION OF MOV AGEING

Currently, the mitigation of MOV ageing is normally based on planned preventive and corrective maintenance programmes. The data for planning maintenance comes from knowledge of the degradation processes, from operating experience and periodic tests on a specific valve or type of valve. The maintenance programme must be specific for each valve or type of valve considering the operating history and experience.

A planned maintenance programme can be supported by condition monitoring techniques. These techniques, described in Section 3.5, can provide indications of the start of degradation of MOVs before valve failure.

To mitigate ageing problems, less durable components such as seals or gaskets are systematically replaced. However, guidelines should be developed for timely mitigation of MOV ageing. In addition, operating procedures can sometimes be improved to reduce ageing. The maintenance programme must be regularly updated according to accumulated operational experience.

6. BACKGROUND OF THE CRP

Understanding of MOV ageing

A database on the failure and malfunction of MOVs in plant operation should be developed, maintained, and made available to interested parties. This database would provide information for understanding MOV ageing including degradation and failures, their cause and correction and thus alert operators of nuclear plants to potential problems and potential maintenance needs. Databases of this type are being developed in several countries today specifically for their own operating nuclear plants. However, they lack the common format needed to understand age related degradation processes and to utilize them internationally. One of the first tasks of the pilot study should be the development of a common database format useful to understand MOV ageing.
Monitoring of MOV ageing

A system or device needs to be identified and/or developed that could detect ageing of MOVs, would enable early detection of degradation, and allow for maintenance or correction before failure occurred and thus enhance valve availability and reliability. Such system or device may be able to identify the probable cause of degradation so that appropriate maintenance action could be taken. Research has been undertaken on the use of motor current signature analysis and power measurement. The design of such system or device should be further developed to detect age related degradations in MOVs and should be demonstrated by in-plant experience. The device may have to measure more than the current or power, for example, the thrust on torque controlled valves. Monitoring of age related degradation in limit controlled valves may require technology not yet developed.

Risk and reliability assessment

Stochastic models have been developed for use in probabilistic risk assessments and assessments of valve reliability. Current stochastic models are also available for describing the time dependence of ageing processes accurately. However, more data are needed to assess the degraded MOV conditions and failure for use in probabilistic risk assessments. Maintenance schedules and the need for valve replacement and repair could then be predicted more accurately. Those valves which represent the largest contributor to risk could then be identified and given appropriate attention and maintenance.

Qualification methods

MOVs are currently qualified before installation for environmental, electrical and mechanical (thermal-hydraulic) conditions associated with accident conditions using both testing and analysis. Because the design of MOVs varies and specified accident conditions vary from country to country and in application, it would be difficult to conduct an international test programme. However, there is a need to exchange information on the techniques and results of qualification tests. Universal guidelines for MOV qualification considering ageing effects should be developed.

The degradation of valves in-service is simulated by pre-ageing treatments prior to testing under accident conditions. It would be desirable to remove few selected MOVs after actual in-plant ageing (say 20 years) and determine by a qualification test whether ageing degradation has affected the original qualification. Information could then be exchanged to assist in formulating future modifications in qualification test procedures.

Improved mechanical qualification methods would also enable utilities to specify motors and actuators more closely matched to the thrust force required during valve opening, closure or isolation under accident conditions.

Maintenance procedures

Guidelines should be developed so that maintenance procedures would adequately address known ageing degradation processes. These guidelines would be used by utilities and manufacturers of valves in developing detailed maintenance procedures.
Improved MOV diagnostic systems utilizing current, voltage, valve torque and stem thrust or strain, etc. are now available which can be used to accurately set the limit and torque switches of the valves. However, results of a recent validation programme in the USA revealed a wide range of MOV diagnostic equipment accuracy. Initial calibration of the thrust measurement devices was a problem for most of the systems. Users of test equipment should be aware of its capabilities and limitations.

Better maintenance procedures that alleviates ageing concerns coupled with advanced diagnostic tools to detect degraded condition would reduce the risk of valve failure and give greater assurance of safe operation under accident conditions.

7. STATEMENT OF WORK FOR CO-ORDINATED RESEARCH PROGRAMME ON MANAGEMENT OF AGEING OF MOTOR OPERATED ISOLATING VALVE

I. Objectives

1. To improve understanding of MOV ageing mechanisms and effects, and thus help assure functionality of MOVs under both normal operating and accident conditions.

2. To identify effective and practical methods for monitoring of MOV ageing capable of timely detection of MOV anomalies attributable to age related degradation.

3. To develop guidelines for risk and reliability assessment of MOV ageing.

4. To improve MOV qualification methods and formulate MOV qualification guidelines.

5. To establish guidelines for effective MOV maintenance to alleviate ageing effects and concerns.

II. Scope of work

The CRP includes:

- The entire motor operated valve as defined in Section 3.1 of TECDOC-670. Pilot studies on management of ageing of nuclear power plant components, IAEA TECDOC-670, IAEA, Vienna, Austria (1992). Upon completion of Stage I, recommendations will be developed for selecting one or more specific MOVs and for Stage II work [2].
III. Tasks to be performed

All five CRP objectives will be addressed in parallel.

Phased approach described below will be used to meet the CRP objectives. Such an approach will facilitate achievement in Stage I of intermediate results useful for MOV ageing management and the initiation of appropriate follow-up work under Stage II.

STAGE I

<table>
<thead>
<tr>
<th>Task No.</th>
<th>Task Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>(1) Understanding of MOV ageing</td>
<td></td>
</tr>
<tr>
<td>1.1 Specify a practical common format for collection of incident data and exchange of evaluated data on age related malfunction and degradation of MOVs (including a root cause and origin of incidents, when available). Also specify the means of information exchange to be used.</td>
<td></td>
</tr>
<tr>
<td>1.2 Exchange evaluated data using the format specified under 1.1.</td>
<td></td>
</tr>
<tr>
<td>1.3 Analyze compiled data and identify dominant ageing mechanisms for MOVs.</td>
<td></td>
</tr>
<tr>
<td>1.4 Prepare Stage I report and recommend follow-up work on understanding of MOV ageing for Phase II CRP, if needed.</td>
<td></td>
</tr>
<tr>
<td>(2) Monitoring of MOV ageing</td>
<td></td>
</tr>
<tr>
<td>2.1 Specify a practical common format and means for the exchange of information on MOV monitoring and diagnostic methods and on their effectiveness in detecting MOV degradation before failure. (All ageing mechanisms listed in Table III should be considered).</td>
<td></td>
</tr>
<tr>
<td>2.2 Exchange information specified under 2.1.</td>
<td></td>
</tr>
<tr>
<td>2.3 Analyze compiled data and identify (a) effective and practical MOV monitoring methods, and (b) those significant ageing mechanisms for which effective monitoring methods are not available.</td>
<td></td>
</tr>
<tr>
<td>2.4 Prepare Stage I report and recommend follow-up work on monitoring MOV ageing for Stage II, if necessary.</td>
<td></td>
</tr>
</tbody>
</table>
(3) Risk and reliability assessment of MOV ageing

3.1 Specify a practical common format for the collection and exchange of operating experience data useful for risk and reliability assessment of MOV ageing. (These data should be useful for probabilistic safety assessment (PSA) failure modes and effects analyses (FMEA) and for ranking of MOVs in terms of their risk significance). Also specify the means of information exchange to be used. Tasks 1.1 and 3.1 should be co-ordinated.

3.2 Exchange data specified under 3.1

3.3 Compare existing stochastic models describing MOV ageing in Stage II, if needed.

(4) MOV qualification methods and guidelines

4.1 Specify a practical common format and means for exchange of information on MOV qualification methods and results and, in particular, on techniques used to account for MOV ageing effects.

4.2 Exchange information specified under 4.1.

4.3 Compare compiled information and identify effective MOV qualification methods and techniques.

4.4 Prepare Stage I report and recommend follow-up work on MOV qualification for Stage II, including the development of qualification guidelines and improved qualification methods, if needed.

(5) Guidelines for MOV maintenance

5.1 Specify a practical common format and means for the exchange of information on MOV maintenance methods and their effectiveness in mitigating ageing effects. (For example, current reliability centred maintenance projects used for the optimization of maintenance programmes could provide useful information on the effectiveness of MOV maintenance).

5.2 Exchange information specified under 5.1.

5.3 Compare and analyze compiled information, and identify effective MOV maintenance methods.

5.4 Prepare Stage I report and recommend follow-up work on MOV maintenance, including the development of maintenance guidelines that would address all significant ageing mechanisms identified in Task 1.3.

10

380
STAGE II

Objectives and scope of Stage II will depend upon the findings and the overall progress made in Stage I. It will also depend upon additional needs of participating Member States, the knowledge and technology gaps existing at the end of Stage I, and the availability of resources.

For example, Stage II study may be needed to narrow the knowledge and technology gaps relating to:

- Qualification testing of naturally aged MOVs under accident conditions,
- Selection and prioritization of MOVs based on their susceptibility to ageing and their risk significance, and
- Built-in diagnostic and monitoring features useful for monitoring and predicting MOV performance.

IV. CRP network

Organizations from Canada, Finland, Germany, India and Russian Federation have signed research agreements in the subject area (see Attachment 1). Other Member State Organizations are invited to participate. Actual participants will be organized in a CRP network to facilitate co-operative work.

8. SUMMARY OF THE FIRST CRP MEETING

The meeting was held at the Technical Research Centre of Finland, VTT, Espoo. The meeting was opened with brief presentations by the Director of the Laboratory of Electrical and Automation Engineering of VTT, Professor Wahlström, who welcomed the participants and focused on the capabilities of the Research Centre. He also identified some of the projects that have been worked on at VTT. The IAEA representative welcomed the CRP members and discussed the meeting agenda. The members agreed with the agenda and each member presented reports of research topics that have been conducted, are currently going on, and possible future work addressing MOV ageing technology [3].

The USA representative gave a summary of the Motor-Operated Research Program supported by the US Nuclear Regulatory Commission (NRC). The work presented included the results from tests performed on specific isolation MOVs located in high energy pipes of Boiling Water Reactor (BWR) power plants in the USA. The test were conducted to determine whether these isolation valves would close against high flows that could result from a guillotine pipe break outside containment. Failure of these valves to close would result in harsh environments that would cause other important equipment in the area of the pipe break to fail. Since very little testing had been performed on these MOVs in the past, the NRC decided to conduct these tests. The results showed that the industry formula underpredicts the valve closure thrust for friction factor of 0.3. In addition, most of the tested MOVs experienced damage to internal components during closure. Current and planned efforts on
effects of ageing such as internal corrosion, and valve body erosion on MOV operability were also presented. The results of the MOV research is being utilized by the NRC regulatory staff to evaluate nuclear plant compliance with current NRC regulatory requirements.

In the presentation of the representative of Czech Republic from ARPO a.s., suggestions for implementing valve diagnostic systems were made. By means of several diagnostic measuring systems a whole family of valves could be checked. The scope of work that must be made to implement valve diagnostics into operation was introduced. Some samples diagrams of diagnostic measurements made on main isolating gate valve D 500 P 1600 VEER 440, taken by diagnostic equipment MOATS 3500 were presented. Measurement of motor current, motor power, stem position, and stem thrust, limit and torque switches had been monitored in the real time. Signature analysis has shown the considerable influence of inertia force of the moving parts.

The representative of India described the work done and planned at Bhabha Atomic Research Centre. The valves in the safety systems, such as the Emergency Core Cooling System (ECCS), have to operate, as required, under abnormal/accident conditions and have to perform their intended safety functions over the entire plant life. It is essential to understand the effects of ageing, so that the ageing-related failures can be minimized through proper rectification/replacement. A comprehensive programme would call for plant tests as well as laboratory studies. Testing of the MOVs which perform a safety function is carried out at all NPPs at specified intervals. From the experience with the MOVs in the oldest operating NPP in India it was observed that torque switches and spring loaded sensors for the same have failed due to ageing and need periodic replacement. From regular tests carried out on MOVs in PHWRs, it was noted that one of the causes of failure was tripping, on overload, during first operation of the valve. The valve were found to operate on the subsequent start signal. It is important to identify the cause and its relation to ageing, if any. Results of the preliminary laboratory tests carried out regarding the use of current signature analysis were also presented.

The presentation of the Technical Research Centre of Finland summarized two studies of MOV operating experiences from Finnish NPPs. In the first study, the MOV failure and maintenance data of two BWR units from a time period of nine years were analyzed. Qualitative analyses were performed for the failed mechanical and electrical valve parts, ways of detection of failure modes, failure effects, and repair actions. Quantitative studies include statistical analyses concerning failure trends and repair and unavailability times. The second study was focused on actuators of selected valves of both PWR and BWR units with the objective to apply an experience based reliability centered maintenance methodology on MOV drives. Based on the analysis results, the current testing, preventive and corrective maintenance tasks, as well as the need of monitoring systems and design modifications can be evaluated and the proposed improvements can be considered by the utilities.

The second Finnish presentation was given by the representative of Loviisa NPP. IVO is developing a system of how to be aware of the state of the LOCA-valves and actuators during their operational period (30 years). For instance, IVO has obtained an MOV diagnosis system to check what the current settings for the torque limit switches are and whether something has happened with the valve or actuator during a period of 1-2 years. IVO is also able to check the tightness of LOCA actuators after the assembly, and in the future this test
will be done according to the maintenance procedure in a 1-2 years interval.

The representative of All-Russian Research Institute for Nuclear Power Plant Operation (RINPO) presented two reports. One of them deals with the description of Russian Scientific-Technical programme for NPP equipment lifetime monitoring, estimation, prediction and management. In order to solve the problems of equipment lifetime estimation, prediction, monitoring and management, the Russian Industrial Scientific-Technical programme (ISTP) "RESURS" was developed. The implementation of the programme will allow to work out regulatory-methodological basis providing legal and technical solution of NPP equipment lifetime management problems. Scientific supervisor of the programme is VNIIAES. Leading experts of industrial scientific-research, design, and operating organization as well as NPP equipment manufacturers, nuclear power plants etc. take part in the implementation of the programme. The implementation of the programme began in April 1993. At present in the framework of this programme the review analysis of the existing national regulatory-methodological basis for equipment lifetime management, prediction, estimation and monitoring was completed, allowing to elaborate the structure and list of the necessary regulatory methodological documents. The second report contained the brief description of the approach to analysis of ageing process of valves and prediction of their lifetime characteristics. Valves equipped with electric drive that are installed in level control system of steam generator in VEER-1000 reactor were taken as an example. Main emphasis was made on classification of failures which had taken place during operation, on detection of prevailing mechanisms of ageing and on assessment of operation factors of reliability and methods of their testing, assessment and prediction. Principles of product ageing parameters selection were briefly described as well as mathematical methods used for quantitative assessment of product reliability factors according to its operation data. The report included considerations on procedure of operating evaluation, testing and prediction of complex unique equipment based on testing of state vectors path.

Tasks and responsibilities:

The participants decided to select task leaders for the various tasks mentioned in the Statement of work of the CRP. The responsibility of the task leader is to collect and analyze the research results from the members that are participating in the specific task. Furthermore, the task leaders are required to prepare and present a summary report on the task in the Research Coordinated Meetings.

The responsibility of each participant is to provide a report on their activities under the Research Programme. In addition, the participants are welcome to present their national activities in the subject area in the meetings.

The schedule of each task, mode of communication, and information format should be defined by each of the five task leaders as soon as possible. The participants for each task should also respond with the appropriate information as soon as possible.

At the completion of the CRP on MOV ageing, an IAEA technical report will be written. The Scientific Secretary will have the overall responsibility for the document. However, each task leader will be responsible for writing a section for the report. A draft report will be reviewed by each of the participants and comments submitted for resolution by
the responsible task leaders. The Scientific Secretary and the task leaders will prepare the final report for publication.

According to the areas of interest of the participants, the matrix presented in Table II was prepared. TUV/Germany was suggested to be the task leader of "Understanding MOV ageing" and VTT/Finland agreed to take the responsibility of the task "Risk and reliability assessment of MOV ageing".

Conclusions and recommendations

1. It is admitted among the participants that a long term systematic approach to ageing aspects is needed to ensure the overall safety and operability of nuclear power plant. When ensuring the function of the safety related MOVs the goal is to establish a solid practice useful far into the future.

The participants agreed that the 5 main tasks identified in the statement of work are important for accomplishing a successful report on MOVs and will be important to the Member States.

2. It is recommended that the 5 task leaders establish a consistent format in which the information can be reported by the participants.

3. This report is to be distributed among all CRP participants with a request to specify their areas of interest (see Table 1) and possible leadership of the tasks.

4. New participants from the Member States are invited to join the CRP. Official requests should be sent to the IAEA.

5. The next Research Coordinated Meeting was recommended to be held in October - November 1994. The place could possibly be Prague.
<table>
<thead>
<tr>
<th>TASKS Participants</th>
<th>Understanding of MOV ageing</th>
<th>Monitoring of MOV ageing</th>
<th>Risk and reliability assessment of MOV ageing</th>
<th>MOV qualification methods and guidelines</th>
<th>Guidelines for MOV maintenance</th>
</tr>
</thead>
<tbody>
<tr>
<td>CANADA AECL CANDU</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>FINLAND VTT</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>FINLAND LOVIISA NPP</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>GERMANY TUV Norddeutch.</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>INDIA BARC</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>RUSSIA VNIIAES</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>RUSSIA OKB GIDROPRESS</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>CZECH REP. ARPO</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>USA US NRC</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
</tbody>
</table>

TL - Technical Leader
REFERENCES


## List of Participants

<table>
<thead>
<tr>
<th>Institute Name, Address</th>
<th>Principal Investigator Name, Tel., Fax.</th>
<th>Research Agreement No., Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. AECL CANDU 2251 Speakman Drive Mississauga Ontario L5K 1B2 Canada</td>
<td>Mr. A. Kumar Tel.: (416) 823 9040 Fax: (416) 823 8006</td>
<td>7304/CF &quot;Management of ageing of motor operated isolating valve&quot;</td>
</tr>
<tr>
<td>2. Technical Research Centre of Finland Laboratory of Electrical and Automation Eng. Otakaari 7B SF-02150 Espoo Finland</td>
<td>Ms. K. Simola Tel.: 358 0 4561 Fax: 358 0 456 6475 E-mail: KAISA. <a href="mailto:SIMOLA@VTT.FI">SIMOLA@VTT.FI</a></td>
<td>7307/CF &quot;Pilot studies on selected components for the management of power plant ageing and life: study of motor operated isolation valve&quot;</td>
</tr>
<tr>
<td>3. Imatran Voima Oy Loviisa Power Plant P.O.Box 23 SF-07901 Loviisa Finland</td>
<td>Mr. J. Snellman Tel.: 358 15 550 3119 Fax: 358 15 550 4435</td>
<td>7306/CF &quot;Management of ageing of motor operated isolating valve&quot;</td>
</tr>
<tr>
<td>4. TÜV Norddeutschland e.V. Grosse Bahnstrasse 31 2000 Hamburg 54 Germany</td>
<td>Mr. W. Ressing Tel.: (040) 8557 2566 Fax: (040) 8557 2429</td>
<td>7528/CF &quot;Management of ageing of motor operated isolating valve in Germany&quot;</td>
</tr>
<tr>
<td>5. Bhabha Atomic Research Centre Reactor Safety Division Trombay, Bombay 400085 India</td>
<td>Mr. V. Venkat Raj Tel.: 9122 5550847 Fax: 9122 5560750</td>
<td>7308/CF &quot;Management of ageing of motor operated isolating valve&quot;</td>
</tr>
<tr>
<td>6. All Russian Research Institute for Nuclear Power Plant 109507 Moscow Russian Federation</td>
<td>Mr. V. Emelyanov Tel.: (095) 377 0024 Fax: (095) 377 0024</td>
<td>7338/CF &quot;Management of ageing of motor operated isolating valve&quot;</td>
</tr>
<tr>
<td>7. OKB &quot;GIDROPRESS&quot; Ordzhonikidze 21 142103 Podolsk Russian Federation</td>
<td>Mr. P. Razgarov Tel.: (095) 137 9108 Fax: (095) 137 9108</td>
<td>7529/CF &quot;Management of ageing of motor operated isolating valve in the Russian Federation&quot;</td>
</tr>
<tr>
<td></td>
<td>ARPO A.S.</td>
<td>Mr. R. Josifko</td>
</tr>
<tr>
<td>---</td>
<td>----------</td>
<td>----------------</td>
</tr>
<tr>
<td>8</td>
<td>Do Koutu 3</td>
<td>Tel.: (0422) 402 5437</td>
</tr>
<tr>
<td></td>
<td>14316 Prauge 4 - Modrany Czech Republic</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>US NRC</td>
<td>Mr. G. Weidenhammer</td>
</tr>
<tr>
<td></td>
<td>5650 Nicholson Lane Rockville, Maryland 20852</td>
<td>Tel.: 301-492-3839</td>
</tr>
</tbody>
</table>
USNRC-SPONSORED RESEARCH LOOKS AT NONLINEAR GATE VALVE RESPONSE

J. C. Watkins, R. Steele Jr., and K. G. DeWall

April 25 through 27, 1994

DATE

OECD/IAEA Joint Specialist Meeting on Motor Operated Valve Issues

SUBMITTED TO

This is a preprint of a paper intended for publication in a journal or proceedings. Since changes may be made before publication, this preprint is made available with the understanding that it will not be cited or reproduced without permission of the author.

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, or any of their employees, makes any warranty, expressed or implied, or assumes any legal liability or responsibility for any third party’s use, or the results of such use, of any information, apparatus, product or process disclosed in this report, or represents that its use by such third party would not infringe privately owned rights. The views expressed in this paper are not necessarily those of the U.S. Nuclear Regulatory Commission.
As more and more valve testing is being completed in response to Generic Letter 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance," we have observed a substantial number of cases where the peak force occurs before flow isolation in the closing direction and after unseating in the opening direction. We call this a nonlinear response. All of the current industry stem force prediction models, including the one developed at the INEL, are closing models for valves with a linear response. Because these models assume a uniform pressure both upstream and downstream of the disc, the models are applicable only when the disc is sliding on the seat. The models do not consider disc tippage and flow forces off the seat, and therefore they may not provide conservative predictions or the correct basis for extrapolation for valves with nonlinear responses.

This nonlinear response would not present such a problem if every valve could be tested at design-basis conditions, but this is not the case. This nonlinear response is not related to a single valve manufacturer, but instead depends upon the internal configuration of the valve, the manufacturing tolerances, the dynamic forces due to flow and differential pressure, and the temperature and quality of the fluid. A nonlinear response cannot be predicted without internal valve measurements, and then only estimated. A look at the valve manufacturers' worst-case internal tolerances indicates that almost any installed valve is capable of nonlinear performance. To reduce unnecessary conservatism, some form of testing is necessary to identify which valves and under what dynamic conditions certain valves fall into this nonlinear category. This does not mean that every valve must be tested at design-basis conditions, but some best-effort differential pressure test should be performed to determine whether a valve performs with a linear response or a nonlinear response. In addition, there is a real need for a new method to evaluate those valves that cannot be tested at design-basis conditions. This new method should include the capability of extrapolating estimates of the required stem force at the design-basis conditions from the test measurements made at conditions less severe than design-basis conditions.

This paper presents our current understanding of this nonlinear behavior and discusses performance-based methods for extrapolating closing loads and for predicting opening loads for valves whose peak thrust is influenced by flow forces acting on the disc when it is off the seat.
Introduction and Background

Sponsored by the U.S. Nuclear Regulatory Commission (NRC),* the Idaho National Engineering Laboratory (INEL) has conducted a wide range of valve research over the past 10 years, including full-scale field testing, single effects laboratory testing, and data analysis. Much of this full-scale work was the first of its kind: the butterfly valve testing in 1983 and -84, the motor-operated valve (MOV) flow and pressure operation plus seismic load tests in 1986, -87, and -88, the line break flow MOV tests in 1988, the parametric and multi-fluid MOV testing in 1989, and the integrated valve loader simulator work conducted in 1991, -92, and -93. From this and other industry research, the classic or linear response of a flex-wedge gate valve is better understood. New equations, correlations, and insights for in situ testing evolved from the work, as well as a number of regulatory information notices and several supplements to Generic Letter (GL) 89-10 “Safety-Related Motor-Operated Valve Testing and Surveillance.” Our flexwedge gate valve stem force correlation identified new vertical loads that previously had not been accounted for. In addition to the correlation, the work identified the fact that internal valve friction factors were influenced by fluid conditions and temperature. Initially, the U.S. nuclear industry was skeptical about whether fluid, pressure, and temperature effects influenced valve performance. However, as more and more industry testing is completed, the effects of pressure and temperature are being validated. The industry test program results have not yet validated our fluid effects theory, but our investigation of valve opening response and nonlinear behavior affirms that fluid conditions indeed affect valve response.

Figure 1 presents classic opening and closing stem force histories showing a linear response for a flexwedge gate valve. Figure 2 shows the same information for a nonlinear case. We call this response nonlinear because the stem force is not linear with the exposed disc area and the differential pressure; as a result, this irregular shape (often a hook shape) appears in the stem force history. The primary cause of this nonlinear performance is tipping of the disc, caused by flow and differential pressure forces as the valve opens or closes. Larger-than-necessary internal valve clearances between the disc and the guides allow the flow forces to tip the disc before full seat contact in the closing direction and after coming off the seat in the opening direction. Figure 3 shows what we call a guide-restrained tipped disc; Figure 4 shows what we call a seat-restrained tipped disc. These two figures represent the worst-case tipage for each of the cases shown. The tipping of the disc changes the pressure distribution around the disc and causes the response to deviate from the expected, linear stem force response.

In addition to the change in pressure response, mechanical interference and physical drag can add to the thrust required to operate the valve. Most of the interference and physically induced forces will be in the closing direction. In the guide-restrained case, if the angle is large enough the guide contact area is reduced, increasing the contact stress, which can lead to galling and plastic deformation. In the seat-restrained case, if the angle of the tipped disc is large enough and the leading edges sharp enough, the disc can machine the seat as it closes. If the

---

a. Work supported by the U.S. Nuclear Regulatory Commission, Division of Engineering, Office of Nuclear Regulatory Research, under U.S. Department of Energy contract No. DE-AC07-76ID01570. FIN A6857 Dr. G. H. Weidenhamer, USNRC Program Manager.
Figure 1. Six-in. RWCU system valve, opening approximately 30% at high flow and reclosing. The resulting stem force history is classic linear behavior.

Figure 2. Ten-in. HPCI system valve, opening approximately 30% at high flow and reclosing. The resulting stem force history is typical nonlinear behavior.
Figure 3. Larger-than-necessary disc-to-guide clearances allow disc to tip. In this example, the guides restrain the amount of tipping.
Figure 4. Larger-than-necessary disc-to-guide clearances allow the disc to tip. In this example, the body seat restrains the tipping.
edges are rounded, only mechanical interference will occur, causing added resistance. All of these instances of mechanical interference and physical damage have been observed in actual testing. Posttest inspection of the valve parts and careful analysis of the stem force histories have shown that most damage can be detected from the stem force history. Figures 5 and 6 are stem force histories from our full-scale valve testing, which show damage occurring during the valve closure. Figure 7 is a stem force history from a closing valve test where the disc tipped but no damage occurred. Figure 8 shows a stem force history from an opening test where the forces are less on the seat than they are after flow is initiated. These later stem force histories that occurred without valve damage depict the nonlinear behavior that we wish to discuss in this paper.

Without testing, nonlinear behavior cannot be predicted with any accuracy because it depends on valve-specific tolerance stackups. Worst-case estimations can be made, but they will be very conservative. The good news is that from a best-effort flow test, nonlinear behavior that does not involve damage can be extrapolated to bound the design-basis response without excessive conservatism. The following discussion explains why we think this is true.

All of the known industry models, including the one developed at the INEL, are applicable only after flow isolation when the disc is riding on the seat and the upstream and downstream pressures have stabilized. These models have been used to predict and to extrapolate both opening and closing forces, but because they do not consider flow forces, pressure distribution, and valve disc tippage, they may not always provide conservative predictions or the correct basis for extrapolation of stem force requirements for valves that may experience nonlinear responses. All of the valves we tested have discs that can tip, and the data we have seen from the industry test programs released in the public domain indicate that the discs in those valves can tip. This nonlinear response would not present such a problem if every valve could be tested at design-basis conditions. However, this is not always possible. In addition, the nonlinear response is not related to a single valve manufacturer, but to the ability of the disc to tip and thus change the pressure distribution around the disc. The magnitude of the nonlinear response is also influenced by the dynamic flow forces, differential pressure, temperature and state of the fluid.

Unless very conservative assumptions are made that would result in a very conservative design-basis calculations, some form of testing will be necessary to identify which valves fall into this problem category. This does not mean that every valve that closes or opens against differential pressure must be tested at design-basis conditions, but some best-effort differential pressure test should be performed to determine whether a valve responds in a linear or nonlinear manner. For those valves that can be tested at conditions less severe than design-basis conditions and that exhibit a nonlinear response, there is a real need for a method to extrapolate the test results to design-basis conditions.

What We Know

The materials of construction for valve internals have been the subject of a number of friction studies. Typically, when valve responses obtained from flow and pressure testing are analyzed, the resulting friction factors will be lower than the friction factors obtained in the laboratory from material samples. The effects of temperature and load on the seat and disc hardfacing have been noted in the laboratory; generally, these effects lower the friction factor. However, when evaluating the results of actual valve testing, the temperature, fluid type, and
Figure 5. Six-in. RWCU system valve, design-basis line break flow. Stem force history shows indications of damage during the loaded portion of the closing cycle.

Figure 6. Six-in. RWCU system valve, design basis line break flow. Stem force history shows indication of damage during the loaded portion of the closing cycle.
**Figure 7.** Six-in. service water valve closing on design basis flow. Stem force history shows nonlinear performance but no damage.

**Figure 8.** Ten-in. HPCI system valve, opening approximately 30% at high flow and reclosing. The stem force history in the opening direction shows a higher load after unseating.
pressure all have an effect on the apparent friction factor. Many times, the load components are not easily separated. In such cases, instead of extracting a friction factor from the test results, a disc factor is extracted. The disc factor is typically a multiplying fraction like the friction factor, but it is not simply the result of a normal versus sliding load calculation. In other words, the disc factor contains other unknowns. Generally, the results of test analysis or calculations using a disc factor cannot be compared with those determined from a friction factor analysis.

As stated, a nonlinear response occurs in valves when the disc can tip far enough to change the pressure distribution around the disc before it comes in full contact with the downstream seat. This nonlinear response occurs before flow isolation (and well before wedging) in the closing direction and after unwedging and after flow initiation in the opening direction. The following analysis provides additional insights on the causes of this nonlinear response. Figure 9 shows the horizontal areas and stem rejection areas where the pressure loads can act on the wedge disc. These areas and corresponding pressure loads were the basis of the old industry gate valve sizing equation. The industry equation was basically an area times a differential pressure times a fractional disc factor, plus or minus the stem rejection load (depending on whether the valve was opening or closing), plus a packing friction load. The internal valve pressure is always trying to expel the stem, so the pressure load on the stem area assists valve opening and resists valve closing. The stem rejection load was the only vertical pressure load that was considered in the early gate valve sizing equations. Valve testing and comparisons of results with the sizing equations provided us with the first insights that other forces were acting on the disc in the wedge gate designs.

![Diagram of valve disc cross-section](image)

**Figure 9.** Valve disc cross-section, showing the unbalanced vertical forces acting on a disc that contributes to the maximum stem force before wedging.
Figure 10 shows an additional area that was since identified where the internal valve pressures could produce an additional vertical load on a wedge-shaped disc. This is an elliptical area defined by the diameter of the seat and the angle of the seat (typically 5° nominal). The study that identified the additional vertical forces is documented in NUREG/CR-5720 (1992). In the closing direction, the pressure forces acting on this area produce a net downward loading that tends to offset the stem rejection load. Typically, the stem rejection load is the greater of the two loads in valves smaller than about 6 in. (assuming a 5° wedge angle). In valves larger than about 6 in., the stem rejection load is the smaller of the two loads. All of the loads discussed in these two paragraphs maintain their relationships as long as the disc is closing or opening in an untipped condition.

However, all valve discs will tip when not in full contact with the valve body seats, and many will tip enough for nonlinear behavior to occur. Figure 11 shows how the pressure distribution loads change when the disc tips. The net downward load shown in Figure 10 gets smaller, and a new upstream vertical disc area term becomes larger as the tippage of the disc increases, resulting in a net upward force that resists valve closure. This new upward force occurs because the upstream pressure is greater than the bonnet pressure. The decrease in the stem thrust just before flow isolation, shown in Figure 7, is caused by the disc coming into full contact with the seat and straightening up. This changes the pressure distribution back to a normal distribution.

Every nonlinear response will be different, and no two valves will exhibit the same nonlinear response. However, a valve tested at conditions less severe than design-basis conditions but at the same fluid subcooling should exhibit all the signs of its nonlinear response. The tippage, pressure distribution, and fluid effects will all be there, and bounding the design-basis response should only be a matter of extrapolation.

What We Don’t Know

We do not know (a) why the larger valves appear to be so insensitive to pressure (it may have to do with the vertical area ratios), (b) why the stem force necessary to move the disc across the seat prior to initiating flow in the valves is so low when tested with single-phase fluids, yet after flow initiation the forces return to expected levels, (c) why the valve opening responses are closer to a mirror image of closing responses with fluids that can flash versus the single phase fluid mentioned in (b) above.

The Nonlinear (Hook) Response in the Closing Direction

As stated, the nonlinear closing response or hook response is the result of a tipped disc and may include mechanical interference between the nose of the disc and the downstream seat. This type of response is particularly troublesome because it cannot be predicted with any degree of accuracy without substantial internal valve measurements. Then, predicting the magnitude of the response is further complicated by the pressure distribution (Figure 11) around the disc, which comes from both the extent of disc tippage and the flow paths in the bonnet region. If the disc can tip, and most of the installed valve designs can, a best-effort flow test will reduce the amount of conservatism that needs to be added to the design-basis calculation because of the unknown valve response. Figure 7 is a hooked valve response in the closing direction from an industry test.
**Figure 10.** Valve disc cross-section, showing the unbalanced vertical forces acting on a disc that contributes to the maximum stem force before wedging.

**Flow isolation point, disc on seat**

\[ P_{up} > P_{down} \]

(a)

**Disc on guide**

\[ P_{up} > P_{bonnet} > P_{down} \]

Disc tipped before isolation

(b)

**Figure 11.** Change in disc areas that the various pressures act on as a result of tipage.
program. This valve and several other valves from INEL and industry test programs have shown hooks in the closing direction. Some included an additional force to close before flow isolation of 20 to 50% over the force required at flow isolation. This is not a trivial unknown.

Once a best-effort closing flow test has been performed and a hook response identified, the force history should be examined for a jagged appearance. Figures 6 and 2 (Valves 1 and 6, respectively) show a stem force history where damage occurred during the closing stroke. Figure 1 (Valve 3) shows a stem force history in which no damage is evident. During our testing we were always able to tell from the stem force history when major damage occurred in a closing test. Some of the more subtle damage, such as bent guides, were identified after testing through posttest inspection. That response can also be seen in the stem force history now that we know what to look for; see Figure 2 (Valve 6). A posttest seat leakage test can also help identify damage. Another good indicator of seat damage is to conduct a second flow test; if the hook is smaller in the second test than in the first test, it is likely that damage occurred in the first test. (The first test removes the sharp edges from the valve seat and rounds the machined area on the disc seat, so the valve closes with less mechanical resistance in the second test.) Figure 12 (Valve 5, second test) is a good example of a test where the hook is less pronounced in the second test than in the first (see Figure 13). If no damage is found, one can assume that the hooked response will increase linearly as the valve pressure load is increased; thus, the design-basis load can be estimated by extrapolating the best-effort flow test with the following equation.

\[ F_{stem} = C_{hooking} \Delta P + P_{up} A_{stem} + F_{packing} \]

where

- \( F_{stem} \) = valve stem thrust
- \( C_{hooking} \) = hooking factor
- \( \Delta P \) = valve differential pressure
- \( P_{up} \) = valve upstream pressure
- \( A_{stem} = \frac{1}{4} \pi (\text{valve stem diameter})^2 \)
- \( F_{packing} \) = packing drag

Use data from the best effort flow test as input to calculate a valve-specific hooking factor, then use that hooking factor along with the design basis pressure and differential pressure to estimate the design basis stem thrust. This extrapolation is based on the fact that the disc is tipped as far as it can tip, and that the pressure distribution around the disc and the resistance from mechanical interference have been established and can be conservatively extrapolated with a linear equation. We believe this extrapolation to be conservative because the ratio of stem force to differential pressure typically drops as the pressure increases, all other things remaining the same.

Extrapolation of Nonlinear Opening Response

While valve opening is the subject of other work, predicting opening loads and analyzing test results from opening tests has been a less-than-absolute science in the past. Before discussing the nonlinear opening valve response, we believe it is necessary to discuss valve opening in general,
Figure 12. Ten-in. HPCI system valve, 30% reopen and closing at high flow. Stem force history shows less of a hook in the second line break flow closing load.

Figure 13. Ten-in. HPCI system valve. Closing against line break flow in the first test shows a significant hook just before flow isolation.
because nonlinear opening behavior cannot be extrapolated; it can only be evaluated and then predicted with a correlation based on expected behavior.

With that said, we must admit that we struggled more to understand opening responses, because after figuring out the subtleties of the closing responses, we were unable to put the opening loads in such neat boxes. The opening loads were not a mirror image of closing with a few changes in signs, i.e., stem rejection load, etc. These opening responses presented a completely different problem. Quick-look stem force history plots for the 10-in. valves showed that flow loads after unseating were higher than the unseating loads. This resulted in a peak in the opening trace that looked like a hump in the stem force after unseating. This meant that not only was there a hook in the closing direction, but a corresponding hump in the opening direction. The evaluation was further complicated in some cases by mechanical interference between the disc and the seat, which added resistance in the closing direction but not in the opening direction. In addition, in the opening direction the fluid and pressure effects seemed to dwarf those seen in the closing direction. There were always a number of individual forces influencing the opening response, and separating these individual forces was more difficult for the opening direction than for the closing direction.

We created a model that represented what we thought should be going on within the valve in the opening direction and then analyzed all of our close-to-open-to-close test data against this standard model. The model we developed was based on the INEL closing equation (see NUREG/CR-5720, 1992), refitted for the opening direction, with the added capability to address disc tippage. The model used actual valve dimensions and test pressures on a real-time basis to make response estimations. It was not our intent during this initial effort to keep tweaking the model until it provided exact response predictions, but rather to look at all of our high-flow reopening and reclosing data against a common set of predictions based on what we thought was going on. We made a lot of assumptions, but as long as the assumptions were consistent from test to test, the model met our needs.

In both phases of our full-scale test programs, we wrote the test procedures to require the valve to be reopened about 20 to 30% under full-blowdown conditions and then reclosed. This testing was performed about 5 to 30 minutes after the preceding blowdown test, as soon as the test had been evaluated. We opened and closed the valves under other flow loads, but typically the highest flow tests are better for analysis purposes because the load components are easier to separate. What might have been a subtle influence in a pump flow differential pressure test stood out in the blowdown test.

Figure 14 shows the stem force plotted against calculations for one of our 10-in. valves during the opening and closing cycles discussed above. In this figure, the upper plot is valve opening, and the stem force history travels from left to right on the plot. The lower plot is valve closing, and the stem force history travels from right to left on the plot. The stem force histories have been plotted against stem position, and the fully closed valve position for both opening and closing is on the left. The zero stem position corresponds to the disc position where the visual flow path is first blocked. Figure 14 shows the stem force during the opening and reclosing cycle for Valve 5, a 10-in. Wm. Powell flexedge gate valve. As explained, the traces representing the calculated responses are the estimated responses based on the actual test pressures, assuming a specific disc friction, and accounting for slight disc tippage after the disc came off the seat.
Figure 14. Valve 5, 10-in. HPCI steam valve opening, 30% on a steam flow and reclosing, shown against calculated loads.
Except for the initial, on-the-seat response for the valve in the opening direction, the force estimates for opening and closing are slightly conservative. The on-the-seat stem force response of the valve is very low compared to the prediction. However, once the disc moves off the seat and flow is initiated, the opening response increases and matches the prediction quite well. The valve has a slight hook in the closing direction. We also note that these particular estimations begin to overpredict the stem thrust at about the 1.5-in. stem position. This is the result of assuming a constant disc area throughout the stroke. Our primary interest was in the response of the valve during the initial portion of the valve opening, when the full area of the disc was exposed to flow. Within this limit, what the opening estimation tells us is that the stem thrust is lower than predicted while the disc is riding on the seat, but as soon as flow is initiated, the disc load increases and starts to respond more as the models would predict it to. This means that the hump in the opening direction is not a new load, but rather the appearance of the load we expected to be there. The large valves (10-in.) were tested only with steam, and all of the responses were similar to the example shown here. Further discussion of this type of opening response is presented later in this paper, in the discussion of the extrapolation of nonlinear opening response.

The smaller, 6-in. valves were exposed to a number of fluid conditions and pressure parametric tests. The low on-the-seat opening response observed during the 10-in. valve testing was also present during the 6-in. valve testing, but only with the single phase fluids (cold water and steam), as shown in Figures 15 and 16 for Valve 2, a 6-in. Velan flexwedge gate valve. Conversely, Figure 17 shows the response of the same valve while opening and closing against 10°F subcooled water at about 1000 psid. This opening response is much closer to a mirror image of closing. Other opening responses for Valve 2 with other fluid conditions where the fluid could flash downstream of the closing disc show the same general tendencies; the opening mirrors the closing response.

A 6-in. Walworth flexwedge gate valve was parametrically pressure tested with 10°F subcooled water. This valve exhibited the same general opening shapes as the valve described above did when tested with flashing fluids. These test results show that for a fluid that can flash, the opening direction develops a response much like that seen in the closing direction. The pressure effect was also observed for both 6-in. valves in the opening direction; as we increased the valve inlet pressure, the responding stem thrust was less per pound differential pressure.

The pressure and fluid effects in the opening direction are not unexpected, but the magnitude of the effects while the disc was riding on the seat are surprising. We had previously observed the pressure effect in the closing direction, but to a much smaller degree. The fluid effect in the closing direction required the INEL closing correlation to have two different disc friction factors, one for fluids under 70°F subcooled and another value for fluids over 70°F subcooled. So, while the fluid and pressure effects were not unexpected, what we cannot explain is why the seat friction factors are so low for the single-phase fluids. While it is a curiosity, the fact remains that once flow starts, the response recovers to a more predictable value. This dictates that we must predict the response without regard for the unexplained low friction while the disc is on the seat.
Figure 15. Valve 2, 6-in. RWCU valve opening 30% on ambient temperature water and reclosing shown against calculations.

Figure 16. Valve 2, 6-in. RWCU valve opening, 30% on steam flow and reclosing, shown against calculations.
Figure 17. Valve 2, 6-in. RWCU valve opening, 30% on 10°F subcooled water and reclosing, shown against calculations.
Once we understood that the fluid type and pressure effects were more severe in the opening direction than in the closing direction, and that the hump in the opening direction did not represent a new flow load, but only an increase in the load to the level where it should have been, we were able to start understanding the opening response. After examining our test results in light of the effect that fluid subcooling has on the peak response of the valve, we extracted the results of those tests where the fluid could flash. These results bound the observed responses and avoid the difficulties of dealing with the low on-the-seat friction factors. Using these data, we were able to estimate the normal and sliding loads acting on the disc and to correlate them over a wide range of differential pressure conditions. Figure 18 is a plot of the sliding versus normal loads for opening a gate valve.

Based on this effort, we suggest that one of the following two correlations be used to estimate the peak stem thrust demands of a valve during opening. Above a normalized normal load of approximately 320 psi (for a valve with a seat angle of 5°), the first correlation should be used, whereas below this load, the second correlation should be used.

![Diagram](image)

**Figure 18.** Flexwedge gate valve opening, normal versus sliding correlation.
For $F_{up} \geq 320$ psi:

\[
F_{stem} = F_{packing} + F_{top} - F_{bot} - F_{sr} + \frac{f_f \cos^2 \alpha}{A_{ms}} \left( F_{up} - F_{dn} \right) \frac{f_f \cos \alpha}{A_{ms}} \left( F_{up} - F_{dn} \right)^2 + 60 A_{ms}
\]

For $F_{up} \leq 320$ psi:

\[
F_{stem} = F_{packing} + F_{top} - F_{bot} - F_{sr} + \frac{1.3 f_f \cos^2 \alpha}{A_{ms}} (F_{up} - F_{dn}) - \frac{2.6 f_f \cos \alpha \sin \alpha}{A_{ms}} (F_{up} - F_{dn})^2
\]

where

- $F_{stem}$ = valve stem thrust
- $F_{packing}$ = packing drag
- $F_{top} = P_{up} \cdot A_{ms} \cdot \tan \alpha$
- $F_{bot} = P_{dn} \cdot A_{ms} \cdot \tan \alpha$
- $F_{sr} = P_{up} \cdot A_{stem}$
- $F_{up} = \frac{P_{up}}{A_{stem}}$
- $F_{dn} = \frac{P_{dn}}{A_{ms}}$
- $\alpha$ = valve seat angle
- $A_{ms} = \frac{1}{4} \pi \left( \text{seat MD} \right)^2$
- $A_{stem} = \frac{1}{4} \pi \left( \text{stem dia} \right)^2$
- seat MD = valve mean seat diameter
- stem dia = valve stem diameter
- $P_{up}$ = upstream pressure
- $P_{dn}$ = downstream pressure
- $f_f = 0.63$
- $f_0 = 0.00013$

Unlike the INEL's linear correlation for closing (Figure 19), the opening correlation is a quadratic. In effect, this results in a larger friction factor when the valve is lightly loaded (around 0.63), and the friction factor decreases as the valve load increases. Another difference between the closing correlation and the opening correlation can be seen in the limits of the data scatter at higher loadings. The closing correlation bounded the response with a ±50 psi band, whereas the opening correlation bounds the data with a ±60 psi band. These values represent the term necessary to conservatively bound actual valve performance.

We know from our closing correlation that the data scatter below a disc loading of 400 psi becomes dominant. As such, we used the data we had from the Trojan and Grand Gulf Nuclear Power Plants and the publicly available data we have reviewed from other industry testing. All of these data for low-pressure, low-flow testing indicate that when the disc is lightly loaded, the data scatter should be less than the bounds specified in the paragraph above. Based on those test
Figure 19. Normalized sliding force versus normalized normal force for gate valves.

results, we believe that test results that fall outside the specified bounds represent valve
performance that is not typical of the responses we have observed. This thought is imbedded in
the above correlation when the normalized normal disc loading is less than approximately 320 psi.

Opening responses that are not a mirror image of what we consider a linear closing response
are, to some degree, nonlinear. One of the nonlinear opening responses that we have seen most
often is where the disc load after the initiation of flow is higher than the disc load on the seat;
see Figure 12 (Valve 5, second test). The cause of this response is not completely understood,
and it usually occurs only with high-pressure applications and single-phase fluids. We have not
observed this type of response at loads lower than a 400-psi disc load. This response cannot be
extrapolated; hence, when this response is observed in an in situ test, the equation presented
above for calculating a linear opening load should be used to estimate the design-basis load.

Insufficient data exist to extrapolate these unusually low on-the-seat responses, and the
appearance of such a response is somewhat a moot point, because as soon as flow is initiated the
stem force increases to the level we would expect for the loading just off the seat. Without more
data and additional work, we cannot recommend extrapolating these types of stem force
responses.

Occasionally, an opening stem force history will have a dip or a reduction in the force trace
momentarily after flow is initiated. This response in the opening direction is equivalent to the
closing hook. The disc is coming off of the seat and tipping as the results of the flow forces. The
tipped disc changes the pressure distribution around the disc, and the stem force reflects that
change. The magnitude of the response will usually be less than in the closing direction hook,
because the mechanical interference resistance will be less. This drop in stem force after coming off of the seat is not a consideration in extrapolation because it takes place after the highest force in unseating.

**Conclusion**

Nonlinear behavior, whether in the opening or closing direction, is a manageable condition if it occurs without valve damage. Performing a best-effort differential flow test in either direction will help to reduce unnecessary conservatism.

**Reference**


**Notice**

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, or any of their employees, makes any warranty, expressed or implied, or assumes any legal liability or responsibility for any third party's use, or the results of such use, of any information, apparatus, product or process disclosed in this report, or represents that its use by such third party would not infringe privately owned rights. The views expressed in this report are not necessarily those of the U.S. Nuclear Regulatory Commission.
ASSESSMENT OF NPP OPERATIONAL EVENTS DUE TO VALVE FAILURES
AND MAJOR AREAS OF THE PROGRAM TO IMPROVE NPP VALVES
OPERATION

1. Assessment of NPP Operational Events due to valve failure

The following reactor types are presently operated at NPPs in
CIS member countries: VVER-440, VVER-1000, RBMK-1000, RBMK-1500,
BN-600, EGP-6. The number of valves used in the process systems of
VVER and RBMK units including the I&C and non-nuclear grade valves
amounts to 50 000 per unit /Refs.1,2/. The share of valves in the
total cost of unit equipment is 15%. Data on the number and
content of unit valves is given in Table 1. A peculiar feature of
valves as equipment type is that in many process systems they
perform the system's intended function. This refers for example to
the following systems:

- ECCS safety systems (fast-acting and check valves);
- protection systems of steam generator, steam drum and
  pressurizer (safety and pilot valves);
- steam dumping/relief systems (isolation/throttle valves of
different steam reduction facilities);
- steam generator (SG) and steam drum feedwater systems
  (control valves) etc.

A great number of valves and their role in assuring major
functions of process systems result in relatively high effects of
valve reliability on unit safe and efficient operation.

1.1. Number of events due to valve failures.

Let us review the statistics of NPP operational events due to
valve failures /Refs. 3,5,6,7/. Table 2 and Fig. 1 show the number
of events due to valve failures for the 1987-1992 period.

Analysis of the data shows that during the reviewed period
the number of events due to valve failures has increased as
follows:
<table>
<thead>
<tr>
<th></th>
<th>VVER-440</th>
<th>VVER-1000</th>
<th>RMK-1000</th>
<th>RMK-1500</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Odd unit</td>
<td>Oven unit</td>
<td>Odd unit</td>
</tr>
<tr>
<td>Gate valves</td>
<td>74</td>
<td>246</td>
<td>1820</td>
<td>1145</td>
</tr>
<tr>
<td>Globe valves</td>
<td>5448</td>
<td>6519</td>
<td>8715</td>
<td>8714</td>
</tr>
<tr>
<td>Control valves</td>
<td>80</td>
<td>126</td>
<td>352</td>
<td>265</td>
</tr>
<tr>
<td>Emergency valves</td>
<td>65</td>
<td>172</td>
<td>135</td>
<td>135</td>
</tr>
<tr>
<td>Check valves</td>
<td>338</td>
<td>421</td>
<td>609</td>
<td>429</td>
</tr>
<tr>
<td>Valves for M&amp;TE</td>
<td>5480</td>
<td>21221</td>
<td>17300</td>
<td>17300</td>
</tr>
<tr>
<td>(measuring and</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>test equipment)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total without M&amp;TE</td>
<td>6005</td>
<td>7484</td>
<td>11631</td>
<td>10688</td>
</tr>
</tbody>
</table>
### Table 2

Number of events in units operation due to valves failures

<table>
<thead>
<tr>
<th>Year</th>
<th>VVER-1000</th>
<th>VVER-440</th>
<th>REMK-1000</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>1987</td>
<td>34</td>
<td>11</td>
<td>12</td>
<td>57</td>
</tr>
<tr>
<td>1988</td>
<td>37</td>
<td>4</td>
<td>11</td>
<td>52</td>
</tr>
<tr>
<td>1989</td>
<td>34</td>
<td>1</td>
<td>6</td>
<td>41</td>
</tr>
<tr>
<td>1990</td>
<td>48</td>
<td>8</td>
<td>30</td>
<td>86</td>
</tr>
<tr>
<td>1991</td>
<td>38</td>
<td>12</td>
<td>17</td>
<td>67</td>
</tr>
<tr>
<td>1992</td>
<td>41</td>
<td>13</td>
<td>23</td>
<td>77</td>
</tr>
</tbody>
</table>

### Table 3

Average electricity loss per 1 event due to valve failures, MWh

<table>
<thead>
<tr>
<th>Year</th>
<th>VVER-1000</th>
<th>VVER-440</th>
<th>REMK-1000</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>1987</td>
<td>16333</td>
<td>3187</td>
<td>21285</td>
<td>14838</td>
</tr>
<tr>
<td>1988</td>
<td>22958</td>
<td>9880</td>
<td>49003</td>
<td>27461</td>
</tr>
<tr>
<td>1989</td>
<td>12734</td>
<td>1600</td>
<td>38310</td>
<td>16205</td>
</tr>
<tr>
<td>1990</td>
<td>22744</td>
<td>7393</td>
<td>33474</td>
<td>23834</td>
</tr>
<tr>
<td>1991</td>
<td>35944</td>
<td>4976</td>
<td>22710</td>
<td>27040</td>
</tr>
<tr>
<td>1992</td>
<td>27482</td>
<td>56769</td>
<td>74590</td>
<td>46500</td>
</tr>
</tbody>
</table>

415
Number of events due to valves failures

Fig. 1

415
- for RBMK units - by the factor of 2.5;
- for VVER-440 units - by the factor of 3;
- for VVER-1000 units - by the factor of 1.5.

1.2. Electricity losses due to valve failures.

Table 3 and Fig. 2 show the average electricity loss per one event due to valve failures.

Analysis of the data shows that during the reviewed period the average electricity loss per one event due to valve failures has increased as follows:
- for RBMK units - by the factor of 2;
- for VVER-440 units - by the factor of 10;
- for VVER-1000 units - by the factor of 1.5.

Fig. 3 shows the share of electricity loss due to events caused by valve failures. During the reviewed period this share has increased by the factor of 8 and amounted to 62% in 1992.

1.3. Assessment of VVER-1000 unit operational events (NV NPP Unit 5).

Fig. 4 presents the diagram of distribution of monthly shutdowns of a VVER-1000 unit (NV NPP Unit 5) due to equipment failures during 5.5 years of unit operation since the commissioning date /Ref.3/.. For the purpose of analysis, the unit equipment has been divided into 7 groups:
- reactor equipment;
- turbine equipment;
- heat exchangers;
- pumps;
- I & C;
- piping;
- valves.

Analysis of the data in Fig. 4 shows the following:
- out of 112 unit shutdowns due to equipment failures since the startup 19 shutdowns occurred due to valve failures, i.e. 17%.
Average electricity loss per 1 valve failure

MWh
80000
70000
60000
50000
40000
30000
20000
10000
0

Calendar year

RBMK ○
Total ●
VVER-1000 ●
VVER-440 ○

Fig. 2
Share of electricity loss due to valve failures

Fig. 3
Distribution of unit shutdown due to equipment failures

Fig. 4
a) total number of shutdowns;
b) due to reactor equipment failures;
c) due to turbine equipment failures;
d) due to pump failures;
e) due to heat-exchange equipment failures;
f) due to pipe failures;
g) due to M&TE (measuring and test equipment) failures;
h) due to valve failures.
Electricity loss due to unit shutdowns and power reductions caused by turbine, high & low pressure heater and other component trips due to valve failures amounts to 15% of the annual electricity loss;

- increased failure frequency is observed after a number of SPMs (Scheduled Preventive Maintenance) (vertical dotted lines).

1.4. Valve quality level at different stages of its life.

The stages of valve life cycle can be represented in a similar way as those of an engineering product (Fig. 5).

The results of operational data assessment allow to identify the stages (phases) of valve life cycle at which we "introduced" failures that are revealed during operation.

For example:

design - pressure difference value on the SG control valve as specified in the "order sheet" and in the "technical request" for the development of this valve has turned out to be lower than the actual value; while developing the Steam Reduction Facility valves they failed to specify the activation/turn-on frequency. The Developer (ChZEM), based on his experience with developing steam reduction facility valves for fossil-fuel plants, has used the motor with the actuation frequency of no more than 60-100/hr while in real situation when operating in the control regime the activation frequency reaches several times per second; no requirements existed for locking the isolation valves in extreme positions etc.;

manufacture - serious violations of the processes of welding the nozzles on the main isolation valves (Kola NPP) detected during the in-service metal inspection; frequent detection of cracks in the radial transitions under the flanges, of cracks in the guides' welds etc.;

acceptance and periodic tests - lack of appropriate test facilities in CIS countries that would allow to obtain trustworthy information in conditions very close to the real conditions /Ref.4/;
Stages and phases of valves life.

Design
- Application for development
- TS for a sample specimen
- TS for a sample request for proposal RFP
- R & D
- Test sample specimen
- Test sample acceptance tests

 Manufacture
- TS for series
- Manufacture
- Test
- Periodic tests
- Transportation

Operation
- Storage
- Incoming inspection
- Mounting
- Commissioning
- Purpose-based use
- Maintenance and repair
- Description

Fig. 5
storage - because of non-compliance with the primary circuit valves storage requirements corrosion wear of valve sealing gaskets occurred at Smolensk NPP leading to significant increase of in-service frequency of the "flange joint leak" type failures;

installation - deviations from the design while installing the valves at Zaporozhye NPP Unit 3 has led to complications and, in certain cases, to impossibility of performing valve maintenance including valves installed in the systems of normal and emergency cooling.

1.5. Conclusions

The above results of data analysis which show low operational reliability of valves; appearance of potentially nuclear-hazardous events, e.g. involving MSIV failure to seat properly, leaks in the check valves of distributing headers, failures of isolation-control valves; the results of audits of equipment operation practice at some NPPs /Refs.8,9/; high labour and dose commitments for valve maintenance are the vivid evidence of inadequate technical level of NPP valves.

Inadequate technical level of valves is due to the absence of a single technical policy at all stages of valve life cycle (Fig.5) starting from the submittal of orders for valve development and up to valve discarding. Implementation of a single technical policy is possible provided one has a comprehensive program of improving NPP valve technical level under way. Such a program is currently being developed by a working group in the "Rosenergoatom" concern established in 1993.

Implementation of this program will enable to improve the technical level of valves to be used in the designs of new NPPs.

At the same time, the assessment of above mentioned data also demonstrates inadequate level of valves operation due to valve deficiencies and inadequacy of valve operation practice.

Leaving aside the issues of improving valve technical level, let us address in greater detail the operation stage.
2. Basic Requirements of NPP Valve Operation Practice

2.1. Operation Practice (System)

Addressing the operation stage, one can refer to the operation practice which is a combination of operation objects, operation means/features, operations & maintenance personnel and documentation which specifies the rules of their interaction required and sufficient to fulfil the tasks of operation /Ref.10/. The operation practice or the operation system includes the following subsystems:

- subsystem of "operative" operation (purpose-based use);
- subsystem of technical operation (Fig.6).

The process of "operative" operation is defined by the program of operation (in the nuclear power it is referred to as operation procedure).

The notion "operation program" (operation procedure) means a document which contains basic principles and adopted decisions for the use of most efficient methods and regimes of "operative" operation implemented in NPP designs, in NPP equipments while designing and manufacturing as well as in the operational-technological documentation taking into account the specified requirements and conditions of operation. This document should reflect the strategy of "operative" operation adopted for a given component. Depending on existing possibilities of defining the limit condition of equipment, the following operation strategies (purpose-based use) can be identified:

- up to expiration of lifetime (service life);
- to failure;
- to pre-failure condition.

The process of technical operation can be represented as a change of different conditions of an operated object. To technical operation conditions the following can be assigned: different types of maintenance and repair (according to the adopted strategy), diagnostics, checks and tests, transportation, storage and expectation of coming into each of the above operation
Structure of NPP valves operation system

Fig. 6
conditions /Ref.10/.

The structure and nature of the process of technical operation is determined by the program of Maintenance and Repair (M&R).

Similar to the above definition, the program of M&R means a document which contains a variety of basic principles and adopted decisions to use the most efficient methods and regimes of M&R implemented in NPP designs, in NPP equipments while designing and manufacturing as well as in the operational and maintenance documentation taking into account the specified requirements and conditions of operation. Here, the program, according to requirements /Ref.12/, should define the M&R strategy adopted for a given component depending on the operation strategy chosen for this component. It should also define the quantitative characteristics of M&R types (regimes), the order of their updating throughout equipment lifetime starting from the first operation and up to discarding. Analysis of papers devoted to the problem of development of equipment M&R systems allows to identify the following M&R strategies:

- specified M&R;
- operation time-based M&R;
- status-based M&R (with Reliability Level Control /RLC/ and/or with parameter control).

M&R strategies are related with the operation strategies (purpose-based use) (Table 4), and for each of them we can choose definite and most efficient M&R strategies (marked by +).
Table 4

<table>
<thead>
<tr>
<th>M&amp;R strategy (type)</th>
<th>Operation Strategy</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>to lifetime</td>
</tr>
<tr>
<td></td>
<td>expiration</td>
</tr>
<tr>
<td></td>
<td>MAINTENANCE</td>
</tr>
<tr>
<td>operation time-based</td>
<td>+</td>
</tr>
<tr>
<td>status-based with parameter control</td>
<td>-</td>
</tr>
<tr>
<td>status-based with RLC</td>
<td>-</td>
</tr>
</tbody>
</table>

Repair

| operation time-based |          +    | -             | +         |
| status-based         |          +    | +             | +         |

The strategy of the specified M&R is characterised by the principle of performing M&R operations with a scope and frequency prescribed in the operational documentation independent of the technical status of equipment at the time of M&R start /Ref.10/.

The strategy of the operation time-based M&R is characterised by the principle of assigning identical, for a population of same equipments, scope of disassembly and rejection of equipment components depending on the operation time since the start of operation and/or after overhaul, and the list
of recovery operations is determined taking into account the results of rejection of equipment components /Refs. 11, 12/.

The difference of the scheduled (specified) maintenance from the operation time-based maintenance is that the specifying of the types and frequencies of work is in the first case established for all the same equipments (independent of the operation time and depending on the place of mounting) and in the second case they specify the depth of disassembly and the frequency strictly depending on the operation time of concrete component.

The basic principle of the strategy of the status-based M&R with RLC may be considered to be the principle of following the strict planning while performing different types of M&R. However the planned element here is only a portion of standard specified operation time-based operations, the work to control equipment reliability level and the frequency of work performing. Another important principle is the timely prevention of functional failures of equipment and its components if the maximum possible operation time before replacement has been achieved. The preventive nature is assured by maintaining continuous in-service observation of equipment and its component reliability levels for the timely detection of pre-failure condition and taking decisions on replacement, on recovery and adjustment operations. /Refs. 11, 12, 13/.

The following characteristic features of M&R with RLC can be stated. Each valve is operated till it fails. No inter-maintenance lifetimes are established. The maintenance of a concrete valve lies in the performing of a required scope of work for adjusting, tuning, detection of failures, defects, faults and their elimination. For valves with sophisticated configurations it may be practical to perform replacements of certain components and elements in accordance with the operation time. Reliability level control is performed for the whole population of identical valves (operated at all NPPs). In cases, where the actual reliability level of a valve is below the specified (or earlier adopted) level detailed analysis of the causes of deviation is performed and measures are taken to raise the reliability level. Introduction of
valve maintenance with RLC involves the solution of a number of organizational and technical problems, e.g.: organization of prompt collection and processing of reliability data enabling to identify the actual reliability levels of the operated valve types; development of methods of setting regulated values of reliability levels for each type of valve; organization of prompt comparison of the actual reliability level with the regulated (specified) level and conduct of analysis of potential consequences; taking decision on the possibility to continue the operation of a certain type of valve before failure and disassembly and taking measures to maintain reliability level.

It is practical to restrict the application area of this strategy by the following valves:

- valves, the failures of which do not affect NPP safety, which is determined by the analysis of reliability of unit process systems and do not lead to system or unit failure;

- valves having high operational flexibility, i.e. valve components and parts are easy to remove, accessible, interchangeable (when applying the strategy to valve elements).

The major principle of the status-based M&R with parameter control is the principle of complying with the strict planning while conducting certain types of M&R (the planned operations here are only a part of standard specified operations according to elapsed operation time, parameter control work /diagnostics/) of equipment and the frequency of performing these operations. Another important principle is the timely prevention of equipment and its component failures if the maximum possible operation time before replacement has been achieved. The preventive nature is assured by the organization of constant in-service observation of the results of equipment and its component diagnostics for the timely detection of pre-failure condition and taking decisions on replacement, on the conduct of recovery and adjustment operations /Refs. 11,12,13/.

The strategies of the status-based M&R (with RLC and parameter control) involve assuring of the high level of operational-maintenance flexibility of components; provision in
sufficient scopes of effective diagnostics and non-destructive
testing tools; extension of industrial-technical and experimental
base capabilities of operating and maintenance organizations; high
level of information support.

The perfection level of a strategy is defined by the degree
of interaction between the objective process of the change in the
component's technical status and the process of its technical
operation intended to maintain operability and good condition. M&R
system based on performing the prescribed scopes of work with
advance-planned intervals or depending on operation time for the
whole population of operated valves shows weak interaction between
the above processes and consequently has poor efficiency, which is
illustrated by data given in Section 1. The most effective system
in these terms is the M&R system based on the optimum combination
of valve repair and maintenance strategies (in the common case of
equipment):

- specified M&R;
- operation time-based M&R;
- status-based M&R (with RLC and parameter control) /Ref.12/.

One may note here that presently there are methods of choosing and
optimizing M&R strategies /Refs. 12,14,15,16,17,18/.

The data in the technical papers /Ref.12/ show that when
introducing the status-based M&R strategy the technical operation
costs may be reduced to 30%.

The major share of exposure (more than 75%) and most of
radiation events occur with maintenance (servicing) personnel.
Optimization of M&R allows to reduce exposure (at least twofold)
as compared to initial one /Ref.19/.

M&R system optimization requires to resolve a number of
methodological issues. First of all we have to identify and
appropriately regulate the correct relationships between such
notions as "operation", "maintenance" and "repair".
2.2. Experience of M&R system documentation development in aviation and abroad

M&R system is a main component of technical operation. It represents a variety of interrelated means, documentation of technical maintenance and repair and executors needed to maintain and restore the quality of items included in the system /Ref. 10/.

Documentation of maintenance and repair is to ensure the solution of M&R system main tasks:

- establishment of requirements to M&R program including the given quality at minimum material, labour expenditures and dose rates;
- preparation and implementation of equipment maintenance and repair with the given quality;
- provision of conditions to perform maintenance and repair including setting up and equipping divisions with necessary tools, training of required number of personnel;
- optimization of production base deployment and of material resources.

M&R program /Ref. 24/ is the initial regulatory document to develop and improve M&R system.

M&R program of equipment can be considered as a set of programs belonging to lower level. Depending on M&R strategy applied, one can distinguish the following M&R programs: specified M&R, operating life-based M&R with parameter control and status-based M&R with reliability level control (RLC).

Taking into account the purpose and structural features of equipment, one can distinguish M&R programs according to types of equipment and its components. For valves for instance one can distinguish M&R programs for valve proper and its motor.

Analysis of M&R programs development experience in other industries and abroad /Ref. 14, 17, 19, 23, 24, 25, 26/ enables to formulate general requirements to composition and

432
contents of valves M&R program sections. M&R program must include the following sections:

- introduction;
- prescribed conditions of operation;
- valves description as M&R object;
- M&R plan;
- M&R organization, M&R tools;
- indices of M&R program;
- appendices.

The introduction should include: basis for program development, aim and purpose of the program, stages and timing for development and updating.

The prescribed operation conditions include:

conditions in which valves are operating taking into account activation demand, duration of activation, parameters of working and ambient medium, acceptable values of reliability factors etc.

Valves description as M&R object must include data on design and configuration features of valves (accessibility, easy to remove, interchangeability of components, suitability for inspection), diagrams of regulating devices location, diagrams of oilers and orifices for lubricating materials location, flanges of components, structural adaptability to advanced strategies and M&R methods.

M&R plan includes:

- strategies and quantitative characteristics of M&R types;
- standard structure (nomenclature and frequency) of M&R types;
- design lifetime, service life, average frequency of unplanned replacements, name, frequency, working hours and duration of activities, normal quality of required spare parts and materials;
- structure of repair cycle (for valves with operating life-based M&R and specified M&R);
- parameters determining technical status of valves and
values of the parameters, lead tolerances, list of testing methods and devices (for valves with status-based M&R with parameter control);

- recommendations for the application of new methods to restore components during M&R.

Contents of the section "M&R organization and tools" is a standard one.

The section "M&R system indices" is to include: specific total cost and working hours of M&R; duration of M&R types; average time and probability of recovery; average duration and working hours for each M&R type; cost of spare parts and materials.

2.3. Analysis of existing M&R system documentation.

Analysis of existing M&R system documents makes it possible to distinguish three groups depending on their purpose:

1) documentation of Regulatory Bodies;
2) documentation used as guidelines while organizing, planning and preparing M&R;
3) documentation used as guidelines while organizing, developing repair procedures and norms (repair documents on technical specifications (TS), standards, calculation of required spare parts, technical processes);
4) documentation applied as executive directly during performing the work and also whole planning and preparing repair.

Nomenclature of group documentation is given in Table 4. The requirements stated in the first group documentation are to be met while developing and applying documentation referring to the groups 2-4.

The 4-th group documentation can be grouped depending on area of application into:
- unit system (facilities, components) or conventional systems;
- homogeneous equipment incorporated into different NPP systems;
- same type equipment included into different NPP system;
- similar equipment of the specific system;
- separate M&R operations performed on similar equipment (norms and standards of "emergency" repair, technological processes of "emergency" repair).

Analysis of documents of groups 2-4 shows that according to the composition and contents of sections the documents practically meet the requirements stated in the programs valid in aviation and nuclear industry abroad.

Analysis of M&R documentation (M&R programs) shows that M&R program valid in the industry is oriented to the specified repair of equipment.

According to the requirements of the first and the second group documentation, the development of the following executive documentation is needed:
- norms of planned (specified) repair;
- technical specifications for repair;
- technological processes of repair.

It should be noted that TSSs for repair are applied in M&R system without taking into account the strategy chosen.

At present activities are under way to develop TS repair and technological processes of valves repair for RBMK NPPs. It is necessary to speed up the organization of similar activities for VVER plants.

3. Main areas of work to improve operation level of NPP valves.

The need to develop the program of valves operation system improvement was mentioned in the decision of the Scientific Council dated March 10, 1988 (Minutes N1) on nuclear power issues in 1988. However, due to some reasons such a program has not been developed so far.
- 15 -

Considering all mentioned above it is possible to give main areas of the program for improving the system of valves operation:

1) Development of guiding documents to choose M&R strategy:
   - main guidelines of NPP valves M&R program;
   - order of valve nomenclature selection with M&R strategy and with specified repair;
   - order of valves nomenclature selection (operative symbols) with operating life-based strategy;
   - order of valves nomenclature selection with parameters monitoring strategy with RLC;
   - order of strategy updating.

2) Development of regulatory documents on the chosen strategies:
   - norms of operating life-based M&R;
   - norms of status-based M&R (the above documents can regulate work packages common for all the strategies);
   - types and structure of documents on the chosen strategies;
   - development of regulatory documents.

3) Development of executive documentation on the chosen strategies:
   - types and structure of executive documentation for specific strategy;
   - development of executive documentation.

4) Development of M&R information support system:
   - development of requirements to the system;
   - development of valve register;
   - development of data base "Technical status of valves";
   - development of data base "Regulatory documentation", "Executive documentation".

5) Development of testing tools (TT):
   - development of requirements to TTs;
   - preparation of TT development program.

6) Development of M&R tools:
   - development of requirements to the tools;
   - definition of tools nomenclature;
7) Organization of personnel training:
- development of a training centre project;
- development of programs and training methodologies;
- development of visual aids.

References:

1. Отчет ВНИИАЭС. Сводный перечень потребности в специальной и общепромышленной арматуре для обеспечения АЭС и АСТ, строящихся с СССР и странах СЭВ в XI пятилетке. УДК 621.311.25:621.039. г. Москва, 1981 г.


3. Отчет ВНИИАЭС. Сбор, обработка и анализ данных об отказах, дефектах и повреждениях специарматуры АЭС по данным ССИФ Н "Атомэнерго" г. Москва, 1988 г. 42 с.


5. Отчет ВНИИАЭС. Анализ работы и нарушений в работе АС Минатомэнергопрома СССР. Разработка организационно-технических мероприятий и рекомендаций по повышению надежности и безопасности (за 1989 год). ОЭ-2878/90.


10. ГОСТ 25866-83. Эксплуатация техники. Термины и определения. м.: Изд-во стандартов. 1983 г.

11. ГОСТ 18322-78. Система технического обслуживания и ремонта техники. Термины и определения. м.: Изд-во стандартов. 1986 г.

12. ГОСТ 24212-80. Система технического обслуживания и ремонта авиационной техники. Термины и определения. м.: Изд-во стандартов. 1980 г. (с 01.11.90 ОСТ 54-003-025-089).


15. Ищикович А.А. Повышение эффективности технической эксплуатации самолетов. Обзор по материалам отечественной и зарубежной печати. М.: ЦНИИ ГА, 1982. 48 с.


17. Руя Б. Классификация и выбор при наличии нескольких критериев. Вопросы анализа и процедуры принятия решений. м.: Мир. 1976.


433

23. РД-7-5. Инструкция по надзору за безопасностью при эксплуатации объектов атомной энергетики. Утверждена постановлением Госатомнадзора N 2 от 05.06. 1986 г.

24. Колесникова Н.М. Организация технического обслуживания АЭС Франции. Обзор. М.: ЦНИИАтоминформ, 1991 г. 48 c.


26. Ицкович А.А. Основные принципы построения комплексной программы технического обслуживания и ремонта самолетов. Научные основы построения и реализации программ ТОиР летательных аппаратов. М.: МИИГА, 1982 г.

27. Общие требования к программе технического обслуживания и ремонта самолетов гражданской авиации. М. ЦНИИ ГА, 1985, 20 с.
THE MOV DISC FACTOR & ENHANCEMENT METHODOLOGIES

Dr. Claude L. Thibault, John M. Stubbert, Jr., David W. Kessler

Wyle Laboratories, Huntsville, Alabama

ABSTRACT

Over the past eight years, the nuclear industry has struggled to understand the dynamic phenomena of motor operated valve operation under differing flow conditions. Wyle Laboratories has investigated these characteristics for numerous motor-operated valves, focusing on identifying the apparent valve, disc and stem factors for various families of valves. During several MOV test programs, more than 37 valves and the data from as much as 1000 strokes per valve were evaluated. This massive quantity of data and its resultant analysis, has led to a more complete understanding of the dynamics of valves and their resultant disc factors (μ) under various pressure and flow conditions.

In other research, Wyle investigated technologies for improving the operability, reliability, and predictability of motor operated valves. During this research, Wyle found that for some valves and designs, their operational functionality was predictable; for others, unpredictable. Although much has been accomplished, especially on modeling valve dynamics, Wyle found that the unpredictability of many valves and designs still exists. A few valve manufacturers are focusing on improving design and fabrication techniques to enhance product reliability and predictability. However, their approach does not address these issues for installed and unpredictable valves.

This paper reviews both the primary research conducted in identifying valve disc factors under various flow conditions and explores some of the more promising techniques that may reduce the disc-seat coefficient of friction - techniques that include optimized valve tolerancing, enhanced surface treatment, and surrogate material evaluation. The first step towards making a valve predictable is to minimize disc tilt by refining tolerances, then secondly reduce disc friction factors by improving the sliding surfaces.
INTRODUCTION

NRC Generic Letter 89-10 addresses the need for maintaining predictable valve operability. For the most part, a predictable valve is one whose stem thrust conforms to prediction models for that valve. When the thrust value exceeds the prediction models, it can generally be attributed to either internal valve clearances, excessive sliding friction, or a combination of factors, any of which can cause valve damage.

Wyle Laboratories has, over the past several years, been involved in several Motor Operated Valve (MOV) testing programs in which the identification of the MOV disc factor was of major concern. Review of test data has shown us, and is supported by widely held belief, that while there are certainly predictable valves, there are also many valves that do not conform to predictable models. In an attempt to reduce the number of unpredictable valves already in service, Wyle has focused its research on reducing the disc factor of the unpredictable valves by investigating the reasons for elevated disc factors, and provide approaches to reducing the forces acting on the valve that can be attributed to internal clearances and poor sliding surfaces.

Research has shown that the disc factors recorded during testing of valves under static and/or low flow conditions cannot accurately be extrapolated to design-basis conditions. Several instances have occurred where valves sustained damage during blowdown tests yet operated predictably under less severe tests.

During tests in support of the resolution of Generic Safety Issue 87 (GSI-87), "Failure of HPCL Steam Line Without Isolation", conducted by the Idaho National Engineering Laboratory (INEL), several of the valves, including a 6-inch and a 10-inch flexible wedge gate valve, which had operated predictably in low flow tests, sustained internal damage during blowdown tests.[4]

In another program, which primarily evaluated valve performance as pertains to the Generic Letter 89-10 MOV operability issue, some valves performed predictably. Each valve was subjected to varying test conditions. The program included cold water flow, hot water, and steam blowdown testing over a variety of differential pressures and fluid velocities on selected valve specimens. Many valves, however, did sustain considerable damage during blowdown tests. Included in these were two 6-inch flexible wedge gate valves and a 2 1/2-inch gate valve. The 6-inch valve's body guide
rails and disc guides sustained damage, which was in the form of severe gouging of the sliding surfaces of the rails and guides. This type of damage is the result of excessive clearances in the guide and rail, and/or in soft sliding surface materials. The stem thrust plots of the 2 1/2-inch valve identified a thrust excursion, which was felt to be the result of as-built dimensions being out of tolerance. Larger than necessary clearances allow the disc to tilt under the flow pressure exerted against the disc during closing, thereby allowing the lower edge of the disc to impact the seat.

Studies of test data have shown that evidence of minor disc tilt under low flow conditions is indicative of a greater, perhaps damaging, tilt under increased flow conditions. Additionally, if there is a tendency for elevated disc factors under low flow conditions, studies have shown that one should expect higher than normal disc factors when a valve is subjected to increased flow conditions.

VALVE DISC FACTORS

The reliability of disc-type MOV's to meet their performance requirements has always been dependent on understanding the disc factor and its primary component, which is the coefficient of friction between the sliding surfaces (disc and seat). Disc friction factors are generally dependent on the contact surface materials, the contact surface finish, and the characteristics and temperature of the fluid.

There are several equations, or sets of equations, that are used throughout the nuclear industry for estimating stem thrust. However, the one most widely used is the Standard Industry Equation (SIE), which characterizes the primary loads acting on the valve during opening or closing. The three terms of the equation represent Differential Pressure Load, Stem Rejection, and Packing Drag:

\[
\text{StemThrust (lbf)} = \mu A_{\text{disc}} dP \pm A_{\text{stem}} P + F_{\text{PACK}}
\]
The Nuclear Maintenance Application Center (NMAC)\[7\] stem thrust equation is:

\[
F_{R}(\text{lbs}) = \left[ -F_W + F_{Pack} + \frac{P}{4} D_S^2 + \mu dP A_{Orifice} \frac{1}{\cos \theta - \mu \sin \theta} \right] \cdot \left[ 1 - \frac{\mu_i(F_S)}{\eta} \right]^{-1} \quad \text{[closing]}
\]

and

\[
F_{R}(\text{lbs}) = \left[ +F_W + F_{Pack} - \frac{P}{4} D_S^2 + \mu dP A_{Orifice} \frac{1}{\cos \theta + \mu \sin \theta} \right] \cdot \left[ 1 - \frac{\mu_i(F_S)}{\eta} \right]^{-1} \quad \text{[opening]}
\]

The NMAC equation contains terms that represent the Stem and Disc Weight, the Packing Friction Load, the Piston Effect Load, Differential Pressure Load, Sealing Load and Torque Reaction Factor. The Differential Pressure Load term is expanded to account for the angle between the disc and seat. The Sealing Load is the vertical component of the sealing contact force. And finally, the Torque Reaction Factor is the function of the required stem thrust, where "FS" is the stem factor.

Idaho National Engineering Laboratory (INEL) expresses the thrust equation as:

\[
F_{disk} = \left[ \cos \alpha + \sin \alpha \right] \left[ (P_{Upstream} \cdot \text{Seal Area}) - (P_{Downstream} \cdot \text{Seal Area}) \right] + \left[ C \cdot \text{Seal Area} \right] \cos \alpha - \sin \alpha
\]

In the INEL equation (extracted from the INEL Isolation Valve Assessment computer program, Version 3.10),

- \( f = 0.400 \) for less than 70° F subcooled water, or \( 0.500 \) for 70° F or greater subcooled water, and
- \( C = 0 \) lbf/in for best estimate, or
- \( 50 \) lbf/in for a conservative calculation.

The load that we are concerned with, the one which is dependent in part on the disc coefficient of friction, is the Differential Pressure Load, or disc drag. This is the vertical component of the load that reacts the differential pressure on the valve disc. Historically, the disc friction factor, or \( \mu \) in the above equation, has presented the industry with the most difficulty to accurately predict. Generally thought to be relatively low, (0.2 to 0.3), disc friction factors have, through recent tests on gate valves, been
found to be much higher for many valves. Flow interruption tests have identified trends that affect the differential pressure load. These tests showed that, (1), repeated cycling tends to increase the stem thrust required to open or close the valve, (2) the disc friction factor may vary greatly from the industry standard 0.3, (3), mass flow/momentum and temperature can have significant effect on stem thrust loads. [2]

An example of disc tilt, and its resultant effect on stem thrust may be seen in Figures 1, 2 and 3. These thrust plots all show a hook just prior to flow isolation. Two of these strokes (Figure 1 and 3) were recorded during blowdown tests, and there is also evidence of gouging and galling of the guide and guide rails as the stroke progresses. The valve in Figure 3 received severe gouging and galling in the guides during this stroke. Post-test inspection confirmed that damage had occurred. The stroke in Figure 2 was recorded during cold water pumped flow tests. Maximum Disc Factors calculated during closure for these valves were between 0.3 and 0.7.

Examples of the disc rubbing on the seat are shown in Figures 4, 5, and 6. The strokes were recorded during cold water tests. Disc Friction Factors ranged from 0.4 to 0.6.

OPTIMIZED VALVE TOLERANCING

As the disc moves into the flow it is subjected to the fluid dynamic forces in the direction of flow. As the disc continues to move into the fluid, the flow resistance increases, and there is a corresponding increase in the differential pressure load on the disc. With some valve designs, the disc may begin to tilt at mid-stroke, while with other valve designs, the disc may not tilt in a downstream direction until the disc is near the bottom of the stroke.

The effect of high flow and differential pressure on the disc increases the likelihood that the disc will tilt enough downstream to cause damage. Should the tilt occur during mid-stroke, galling of the disc guides or slots is likely. When the tilt of the disc occurs near the end of the stroke, there is an increased possibility that galling of the seat surface will occur. Either of these types of damage can substantially increase the sliding friction and thus increase the stem force.

Larger than necessary clearances allow the disc to tilt, and determine the degree of tilt. Reducing the manufacturing tolerances in disc/body guide clearances will reduce the
tilt. There is an increased likelihood for potentially damaging tilt (observing a hook in stem thrust test data) in smaller valves than with large valves. It has been found that the amount of tilt in larger valves is less than with small valves when they both have been built to the same tolerances. The reduction of dimensional tolerances should be relative to the valve size, however, sufficient clearance should remain for thermal relative expansion of the valve components. Also, increasing the guide length diminishes the effect that larger clearances have on disc tilt. Softening the increased angle contact of the wedge against the downstream seat by beveling the bottom edge of the disc would reduce "plowing" the disc into the seats during closure. Machining chamfers on the lower edges of the disc guides would reduce gouging and galling of guide rails.[1]

As part of a research program conducted by Wyle Laboratories for Virginia Electric Power Company, Wyle developed a gate valve model, using the design parameters of a high quality 3-inch wedge gate valve, to study the effects of valve dimensional tolerancing on valve operability. This mathematical model was developed to identify where binding or premature contact might occur during a stroke. This model was integrated into a computer program entitled "MODGVALV", which was developed in IBM Basic to ensure compatibility with most personal computers. The program, which has been validated, calculates critical valve clearances to determine the relative positions of the disc and seat during valve closure, and thereby identifies any premature, and unwanted, contact of the disc with the seat. MODGVALV was designed to allow direct operator input, which provides a means, through adjusting input values, to fine-tune dimensions to optimize the clearances, or to study the effects of operational wear, such as guide wear, by progressively increasing the clearances. Through the direct operator input, the program will also accept reasonable gate valve parameters from other gate valves. The gate valve modeling program also considers linear thermal expansions and guide-offset impact.[3]

As the MODGVALV program predicts the point of disc/seat contact, it can be used to identify potential points on the disc and seat where excessive loads may occur, possibly causing surface damage, which can potentially reduce the operability of a valve.
ENHANCED SURFACE TREATMENTS

A variety of surface treatment techniques were evaluated by Wyle to determine their application and potential for reducing the friction and wear associated with valve operation. Of these, two material modification technologies, ion implantation, and a multi-layer infusion coating process called Magnaplate Hi-T Lube™, appeared to offer good, if not dramatic improvement potential and are therefore reviewed first. Immediately following, an overview of other enhancement technologies that were considered but eliminated from further research is presented, along with the reasons for their inclusion.

Ion Implantation

Ion implantation is a process whereby energetic ions of selected materials are accelerated and made to strike the surfaces of workpieces in a vacuum chamber [5]. The ions, typically with energies to hundreds of kiloelectron-volts (Kev), penetrate hundreds of atomic layers into the surface, where they are slowed down and eventually stop through collisions with atoms of the host material.

The collision cascade upon host atoms creates a region of extensive radiation damage within the surface layer of the target. A term borrowed from solid-state physics, radiation damage refers to alterations produced in the crystal structure of a material. In ion implantation of metal workpieces, radiation damage is extremely desirable because it, along with the foreign implanted ions, contributes to alter surface properties in a number of desirable ways.

The resultant combination can create an amorphous layer with no grain boundaries, which is believed to provide superior wear performance and low friction in tribological service. In ferrous alloys, an amorphous surface can be responsible for reducing corrosion, which often initiates at grain boundaries.

From work performed by Mr. Piran Sioshansi, Spire Corp. [5], the ion implantation process has been applied to a variety of industrial products and tools such as:

- Bearings - To impart corrosion resistance and to improve wear resistance of precision bearings like those used in gyroscopic inertia systems, etc., a
combination of Titanium and Carbon has provided the best results to date. The amorphous, glassy, Titanium carbide precipitates formed on the surface has been shown to decrease the coefficient of friction by a factor 2 and provide superior wear performance.

- Nuclear Reactor Components - Because of its excellent corrosion resistance, Zirconium has been used in this highly corrosive environment. However, Zirconium surfaces are susceptible to fretting and adhesive wear. High dose implantation of Carbon and Nitrogen increases the micro hardness and drastically improves the resistance of Zirconium to both of these types of wear by creation of Zirconium Carbonitride layers. Other ions implanted such as Chromium and Carbon have been chosen for evaluation of Zircaloy components for Nuclear Reactor applications.

- Prosthetics - Many orthopedic implant devices are made from Titanium based alloys. Ion implantation has proven to be valuable for increasing the wear resistance of Titanium alloys, while also improving fatigue performance and corrosion resistance.

- Stamping, Cutting Tools, Injection Molds, Ceramic Parts - are some other applications that have utilized ion implantation technology to improve various performance factors.

Magnaplate Hi-T Lube™

Magnaplate Hi-T Lube™ is a patented dry film lubricant consisting of a multi-layer system that is applied to wear surfaces by means of a series of "synergistic" electrodeposited metals and alloys which are permanently bonded to the substrate metal [6]. Hi-T Lube™ is a registered trademark of the General Magnaplate Corporation. It consists of metallic and non-metallic layers that minimize dimensional changes.

"Synergistic" coatings are not true coatings in the conventional sense. These coatings become an integral part of the top layer of the base metal rather than just a surface cover. These coatings are referred to as "synergistic" because the resulting surfaces are superior in performance to both the base metal and the individual components of
the coating. Each layer of the Hi-T Lube™ matrix has beneficial features. However, upon final diffusion, they form a metallic/oxide matrix that is significantly better than any one of the individual layers or the base metal.

The first layer applied is an extremely hard coating. Before application of this first layer, the surface of the base metal is metallurgically cleaned. During the pre-cleaning process, special provisions are made to avoid hydrogen embrittlement of the metal since it not only contributes to the hardness of the total matrix structure, but actually forms the critical interface with the base metal. The second layer applied is a semi-soft, compressible metal layer. It is composed of metals that can withstand high temperatures and loads. The final surface layer is composed of a blend of highly effective lubricants. The applied layers are then diffused in a controlled atmosphere chamber [6].

Testing Process

To determine the feasibility and potential of these two material surface enhancement technologies, Wyle established a test program that would subject treated and untreated samples of common valve material to a form of testing that would yield friction test data. For this program, a pin and V-block testing process was selected because of its universal acceptability, economy, and usability in measuring friction and wear simultaneously. Although the objective was to determine friction factors, the benefit of obtaining wear factors was obvious. Twenty sets of material samples (SS 316 and Stellite No. 6) were manufactured in addition to spare pins and V-blocks. Each set was fabricated to conform to ASTM specifications D2625, D2670, and D3233, and was comprised of two V-blocks and one pin. The fabricated samples were as follows:

Ion implantation was performed by the Spire Corporation using their "IONGUARD 2001" process. Titanium and Carbon ions (Ti⁺ and C⁺) were implanted to a nominal depth of penetration of 1000 Angstroms without increasing the dimensions of the piece. Hi-T Lube™ processing was performed by General Magnaplate of Texas to a coating of 0.001 inch, plus or minus 0.0003 inch.

Testing was conducted on both untreated stainless steel and stellite samples and the ion implanted and Hi-T Lube™ processed samples. The test matrix in Table 1 shows the various pin and V-block combinations. Each test was run for a total of 20 minutes
at a linear speed of 3.8 inches per second, which is equivalent to 4,560 inches of travel. Each test was run at a linear speed of 3.8 inches per second. The samples were subjected to an initial load of 30 lbf, and the coefficient of friction was measured at 1, 2, 3, 4, and 5 minutes. The load was then increased to 50 lbf and an initial measurement was recorded at time $t = 5$ minutes, followed by measurements at 10, 15, and 20 minutes. All samples were lubricated with distilled water. A wear measurement was taken at time $t = 20$ minutes.

Test Results

The test results presented in Tables 2 and 3 clearly show that Stellite improves wear when compared to stainless steel. However, among the surface-modified samples, the results are not as clear (see Figures 7 through 10). Without further investigation, the Hi-T Lube™ process "appears" to be superior to the ion implantation process. It shows apparent improvement in both friction and wear for both stainless steel and Stellite "hardfaced" samples. This improvement is greater if only one of the two material components comprising a set is surface treated. The ion-implanted samples showed no improvement in friction but showed good improvement in wear. The stainless steel/stainless steel reference sample results are not clear. On one hand, the wear factors are as anticipated but on the other hand, the friction factors appear to improve (decrease).

Note: Using ASTM specification 2670 for Pin on Disc, the load per unit area and the velocity will better simulate actual valve conditions. Further investigation using the Pin on Disc method should improve confidence in the ion implantation method, and confirm its application for improving valve sliding surfaces.

Other Enhancement Techniques Considered

Several other enhanced surface treatment techniques were considered by Wyle but were eliminated from further research and sample testing. Plasma surfacing was discarded as a viable surface treatment technique for major internal valve components as its application would increase part dimensional limits beyond acceptable tolerances without major component modification, and its adhesive lamination properties are suspect for the intended use in the program. Micro sealing may prove viable for reducing friction at the stem-stem nut interface (stainless steel on brass); however, the
process does not result in true surface hardening and therefore was considered not suitable for use on wear-related internal components. And finally, vapor deposition was eliminated from further consideration because it is believed that it is not suitable for long term wear in the extreme operating and postulated design basis environments.

IMPROVED GUIDE AND GUIDE RAIL MATERIALS

The softer guide and guide rail materials in some gate valves tend to increase the potential for galling and gouging, thereby requiring greater thrust to open or close the valve. By using materials on guiding surfaces that are able to support the contact stresses without damage, significant improvement of disc movement is accomplished by "hardening" the guide and guide rail materials. This can be accomplished by replacing soft guide materials with harder materials, or with surface hardening techniques, such as use of a Stellite overlay on all friction surfaces: seats and discs tracks, internal guide sides, and the exposed faces of guide bars.

CONCLUSIONS AND RECOMMENDATIONS

Regarding the optimal valve tolerancing issue, MODGVALV is a simple yet effective tool that designers and manufacturers can use to statistically model gate valve kinematics. Utilizing Basic as the programming language, MODGVALV can predict the location of contact on a gate valve's disc and downstream seat and thus denote potential locations of surface damage which can lead to abnormal closing forces and valve failure. The program also provides a method to study the generic effects of operational wear by selectively increasing the clearances between guiding surfaces which, of course, will increase the likelihood of premature contact.

Regarding enhanced surface treatment techniques, the research results are very encouraging. The Hi-T Lube™ process showed apparent improvement in both friction and wear for both stainless steel and Stellite hardfaced samples. This improvement was greater when only one of the two material components comprising a set was surface treated. The ion-implanted samples showed no improvement in friction, but showed good improvement in wear. The data for the untreated stainless steel and Stellite hardfaced samples clearly support the conclusion that sliding surfaces in all stainless steel valves should be "hardfaced" where possible.
For unpredictable valves, enhanced surface treatments may be used to decrease friction and improve wear factors of MOV sliding surfaces. Such sliding surfaces include the disc guides and body guide rails, the stem, and the disc seats and body seat rings. Improvement of these surfaces, along with proper tolerancing, can prevent or mitigate internal damage such as gouging, galling, and excessive wear, which testing has shown can occur in some MOVs under severe operating conditions (high flow and/or high differential pressure). Prevention of these types of internal damage can significantly contribute towards making unpredictable valves predictable.

Further, these enhanced surface treatments and technologies may be suitable for applications involving predictable valves as well. To illustrate, the EPRI MOV Performance Prediction Program recently completed by Wyle demonstrated that even the valves that performed predictably and consistently exhibited disc coefficients of friction somewhat higher than the widely accepted value of 0.3 during cold water pumped flow testing (after preconditioning) as well as during hot water (530°F) and steam blowdown testing. Since the maximum thrust required to open or close a valve is directly related to the value of disc coefficient of friction, the higher the disc μ, the greater the stem thrust required. Application of one or more of the enhanced surface techniques to the disc, guides and/or valve body sealing surfaces would reduce the disc sliding coefficient of friction to 0.3 (or less), thereby avoiding more costly modifications required to counteract the effects of a high apparent disc μ. Further research and testing would be required to determine the actual improvement in friction and wear for a full-scale valve. And, other factors such as cost effectiveness, feasibility of application, life span of the proposed treatment (i.e., will the proposed treatment continue to be effective after 100, 200, 1000 strokes?), and, of course, retrofitting techniques need to be evaluated. However, the benefits associated with even modest improvements in friction and wear factors for installed valves are readily apparent.

Tighter dimensional tolerances, and improved sliding surfaces, thereby reducing the potential for valve damage, and decreasing the disc sliding friction factor to a predictable value, increases the likelihood that the stem force will be able to overcome other forces, such as thermo-hydraulic, that act on the disc. Therefore, a combination of actions will have the best chance for success of improving the operability of a valve.
REFERENCES


### Table 1. Pin and V-Block Test Matrixa

<table>
<thead>
<tr>
<th>Ion Implantation Process</th>
<th>Pin Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Block Set Material</td>
<td>ST</td>
</tr>
<tr>
<td>ST</td>
<td>•</td>
</tr>
<tr>
<td>SS</td>
<td></td>
</tr>
<tr>
<td>ST-II</td>
<td></td>
</tr>
<tr>
<td>SS-II</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Hi-T Lube™ Process</th>
<th>Pin Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Block Set Material</td>
<td>ST</td>
</tr>
<tr>
<td>ST</td>
<td></td>
</tr>
<tr>
<td>SS</td>
<td></td>
</tr>
<tr>
<td>ST-HT</td>
<td></td>
</tr>
<tr>
<td>SS-HT</td>
<td></td>
</tr>
</tbody>
</table>

---

a. SS = Stainless Steel
ST = Stellite No. 6
II = Ion Implantation Processed
HT = Hi-T Lube™ Processed
<table>
<thead>
<tr>
<th>Set</th>
<th>Pin Material &amp; Treatment&lt;sup&gt;a&lt;/sup&gt;</th>
<th>Block Material &amp; Treatment&lt;sup&gt;a&lt;/sup&gt;</th>
<th>Coefficient of Friction at time (minutes)&lt;sup&gt;b&lt;/sup&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>A</td>
<td>ST</td>
<td>ST</td>
<td>0.297</td>
</tr>
<tr>
<td>C</td>
<td>SS</td>
<td>SS</td>
<td>0.495</td>
</tr>
<tr>
<td>D</td>
<td>SS-II</td>
<td>SS</td>
<td>0.495</td>
</tr>
<tr>
<td>E</td>
<td>ST</td>
<td>ST-II</td>
<td>0.396</td>
</tr>
<tr>
<td>F</td>
<td>SS</td>
<td>ST-II</td>
<td>0.396</td>
</tr>
<tr>
<td>G</td>
<td>ST-II</td>
<td>ST-II</td>
<td>0.396</td>
</tr>
<tr>
<td>H</td>
<td>SS-II</td>
<td>ST-II</td>
<td>0.396</td>
</tr>
<tr>
<td>I</td>
<td>ST</td>
<td>SS-II</td>
<td>0.297</td>
</tr>
<tr>
<td>J</td>
<td>SS-II</td>
<td>SS-II</td>
<td>0.396</td>
</tr>
<tr>
<td>K</td>
<td>SS-HT</td>
<td>SS</td>
<td>0.099</td>
</tr>
<tr>
<td>L</td>
<td>ST</td>
<td>ST-HT</td>
<td>&lt;0.099</td>
</tr>
<tr>
<td>M</td>
<td>SS</td>
<td>ST-HT</td>
<td>&lt;0.099</td>
</tr>
<tr>
<td>N</td>
<td>ST-HT</td>
<td>ST-HT</td>
<td>0.198</td>
</tr>
<tr>
<td>O</td>
<td>SS-HT</td>
<td>ST-HT</td>
<td>0.099</td>
</tr>
<tr>
<td>P</td>
<td>ST</td>
<td>SS-HT</td>
<td>0.198</td>
</tr>
<tr>
<td>Q</td>
<td>SS-HT</td>
<td>SS-HT</td>
<td>0.147</td>
</tr>
</tbody>
</table>

<sup>a</sup> SS = Stainless Steel  
ST = Stellite No. 6  
II = Ion Implantation Processed  
HT = Hi-T Lube™ Processed

<sup>b</sup> The coefficient of friction at time 0 indicates the start of testing with a load of 30 lbf. Additional friction data was recorded at 1, 2, 3, 4, and 5 minutes thereafter.
Table 3  Pin and V-Block Friction and Wear Test Results (50 lbf Results)

<table>
<thead>
<tr>
<th>Set</th>
<th>Pin Material &amp; Treatment(^a)</th>
<th>Block Material &amp; Treatment(^a)</th>
<th>Coefficient of Friction at time (minutes)(^b) 5</th>
<th>10</th>
<th>15</th>
<th>20</th>
<th>Wear (Teeth)(^c)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>ST</td>
<td>ST</td>
<td>0.327</td>
<td>0.357</td>
<td>0.357</td>
<td>0.386</td>
<td>21</td>
</tr>
<tr>
<td>C</td>
<td>SS</td>
<td>SS</td>
<td>0.416</td>
<td>0.327</td>
<td>0.297</td>
<td>0.268</td>
<td>540</td>
</tr>
<tr>
<td>D</td>
<td>SS-II</td>
<td>SS</td>
<td>0.535</td>
<td>0.505</td>
<td>0.535</td>
<td>0.565</td>
<td>591</td>
</tr>
<tr>
<td>E</td>
<td>ST</td>
<td>ST-II</td>
<td>0.446</td>
<td>0.416</td>
<td>0.416</td>
<td>0.416</td>
<td>24</td>
</tr>
<tr>
<td>F</td>
<td>SS</td>
<td>ST-II</td>
<td>0.446</td>
<td>0.505</td>
<td>0.624</td>
<td>0.594</td>
<td>318</td>
</tr>
<tr>
<td>G</td>
<td>ST-II</td>
<td>ST-II</td>
<td>0.327</td>
<td>0.386</td>
<td>0.416</td>
<td>0.416</td>
<td>9</td>
</tr>
<tr>
<td>H</td>
<td>SS-II</td>
<td>ST-II</td>
<td>0.446</td>
<td>0.446</td>
<td>0.446</td>
<td>0.476</td>
<td>295</td>
</tr>
<tr>
<td>I</td>
<td>ST</td>
<td>SS-II</td>
<td>0.297</td>
<td>0.297</td>
<td>0.297</td>
<td>0.327</td>
<td>5</td>
</tr>
<tr>
<td>J</td>
<td>SS-II</td>
<td>SS-II</td>
<td>0.594</td>
<td>0.594</td>
<td>0.594</td>
<td>0.594</td>
<td>495</td>
</tr>
<tr>
<td>K</td>
<td>SS-HT</td>
<td>SS</td>
<td>0.178</td>
<td>0.178</td>
<td>0.178</td>
<td>0.178</td>
<td>1</td>
</tr>
<tr>
<td>L</td>
<td>ST</td>
<td>ST-HT</td>
<td>0.060</td>
<td>0.060</td>
<td>0.060</td>
<td>0.089</td>
<td>4</td>
</tr>
<tr>
<td>M</td>
<td>SS</td>
<td>ST-HT</td>
<td>0.238</td>
<td>0.327</td>
<td>0.297</td>
<td>0.386</td>
<td>9</td>
</tr>
<tr>
<td>N</td>
<td>ST-HT</td>
<td>ST-HT</td>
<td>0.268</td>
<td>0.297</td>
<td>0.297</td>
<td>0.297</td>
<td>6</td>
</tr>
<tr>
<td>O</td>
<td>SS-HT</td>
<td>ST-HT</td>
<td>0.268</td>
<td>0.416</td>
<td>0.446</td>
<td>0.505</td>
<td>146</td>
</tr>
<tr>
<td>P</td>
<td>ST</td>
<td>SS-HT</td>
<td>0.178</td>
<td>0.178</td>
<td>0.178</td>
<td>0.178</td>
<td>2</td>
</tr>
<tr>
<td>Q</td>
<td>SS-HT</td>
<td>SS-HT</td>
<td>0.238</td>
<td>0.238</td>
<td>0.238</td>
<td>0.238</td>
<td>4</td>
</tr>
</tbody>
</table>

\(^a\) SS = Stainless Steel  
ST = Stellite No. 6  
II = Ion Implantation Processed  
HT = Hi-T Lube™ Processed

\(^b\) The coefficient of friction at time 5 indicates the start of testing with a load of 50 lbf. Additional friction data was recorded at 10, 15, and 20 minutes thereafter. The wear measurement was taken at time t=20 minutes, upon completion of testing.

\(^c\) 14.4108 teeth = 0.001 inches of total wear.
Figure 1. 10-Inch Gate Valve Showing Hook

Figure 2. 4-Inch Gate Valve Showing Hook
Figure 3. 10-Inch Gate Valve Showing Excessive Guide Wear and Hook

Figure 4. 12-Inch Gate Valve Showing Disc/Seat Rubbing
10-INCH GATE VALVE OPENING

Figure 5. 10-Inch Gate Valve Showing Disc/Seat Rubbing

10-INCH GATE VALVE CLOSING

Figure 6. 10-Inch Gate Valve Showing Disc/Seat Rubbing
Figure 7. Friction Results for Untreated Stainless Steel and Stellite Samples

Figure 8. Friction Results for Samples with Matching Pin and V-Block Materials and Treatments
Figure 9. Friction Results for Ion Implant Processed Samples

Figure 10. Friction Results for Hi-T Lube™ Processed Samples
Effects of Internal Corrosion on Motor-Operated Valve Operability

Gerald H. Weidenhamer
U.S. Nuclear Regulatory Commission
Washington, D.C. 20555

ABSTRACT

The U.S. Nuclear Regulatory Commission (NRC) initiated research to determine the effects that internal corrosion may have on thrust requirements and motor-operated valve (MOV) operability. The results reported in this paper focus on friction experiments conducted on material specimens typical of a certain containment isolation valve in Reactor Water Cleanup pipes in Boiler Water Reactor (BWR) plants. These small material specimens were subjected to a hot water environment typical of BWR primary and Pressurized Water Reactors (PWR) secondary systems. The corrosion period was accelerated to approximate 18 months of actual operation. Following the corrosion cycle, the corroded specimens were subjected to friction tests in accordance with accepted standards. In addition to the corroded specimens, uncorroded specimens of the same materials were subjected to the same friction tests and differences in friction coefficients quantified. Other specimens typical of other MOV materials will also be tested in the future and friction changes determined.

The results of these experiments are intended to address questions regarding the impact of this particular type of aging degradation on MOV operability. Specifically, the amount of margin to be added to torque switch settings to account for these types of effects are expected to be quantified.

The results reported here are preliminary and are intended to describe the corrosion effects phenomena and the extent of the research programs the NRC is supporting to understand these effects. These results will provide the NRC regulatory staff with the technical basis for evaluating licensee's MOV torque switch settings that should accommodate increased thrust needs for overcoming corrosion effects. This work addresses the periodic verification part of Generic Letter 89-10.

Background

On June 28, 1989, the NRC issued Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance." The reason for issuing GL 89-10 was that assessments of the reliability of safety-related MOVs, based on extrapolations of the available information from valve surveillances at that time, indicated that more than expected MOV malfunctions were occurring and that additional measures should be taken to ensure operation under design basis conditions for the life of the plant. One of the main items identified in GL 89-10 is that licensees need to periodically verify torque switch settings to ensure that MOVs will accomplish their intended functions over the life of the nuclear power plants in the USA.
The intent of periodic verification is to assure that the licensees consider the effects of aging degradation that can cause reduced performance of motors, operators, and/or the valves after long periods of operation. These aging degradations can cause reductions in performance and result in either premature trips or motor stalls such that the MOV discs can stop at partial stroke positions. When required to operate, most MOVs must change to either the fully open or the fully closed positions; therefore, premature trips at partial-stroke positions prevent the MOVs from performing their intended functions.

Although a considerable amount of friction testing has been conducted in the past, most of this work was directed at quantifying friction coefficients for new materials. As far as this author has been able to determine, there has been little friction testing on corroded materials for the purpose of quantifying torque switch margin.

The NRC has been supporting MOV operability research over the past 10 years. The Electric Power Research Institute (EPRI) more recently, has also supported MOV research. These research programs included performing a number of flow tests, up to blowdown conditions, to understand MOV technology when subjected to various flow loads. The EPRI MOV research program also supported special research on quantifying frictional effects between certain valve components. However, there is no specific data available from either the NRC or the EPRI research programs for determining MOV performance under corroded conditions.

Discussion

The aging degradation mechanism considered in this paper is internal corrosion. The objective of this work is to determine the changes in friction that will occur after an MOV has been subjected to long periods of operations in hot water environments. Specifically, the additional thrust needed to overcome the increased friction effects due to corrosion must be quantified such that margins to MOV torque switch settings can be added to account for incremental changes in friction. Since the MOV must operate over the period after it is reverified, the MOV must be capable of accomplishing its intended functions at any time during the next operation period. Determining the magnitude of this margin to ensure operation during this period is the intent of this NRC project.

In 1993, the NRC Office of Nuclear Regulatory Research started a project to develop a data base for determining the corrosion effects on friction. At that time it was decided that typical MOV materials would be corroded in environments replicating LWR conditions. It was also decided that the containment isolation MOVs utilized in the Reactor Water Cleanup Pipe in Boiling Water Reactors (BWRs) would be investigated first. Specimens of materials for these MOVs were machined and subjected to environments simulating water temperature, pressure, and water chemistry for this particular pipe. Half of the specimens were subjected to the environmental effects while the other half of the specimens were not corroded. The corrosion environment was accelerated to simulate a period of approximately 18 months. This period of time (18 months) is a typical time between outages for nuclear power plants in the USA. Most MOVs that are to be periodically
verified will be tested during these outages. During the corrosion cycle, conditions were monitored to maintain control on the variables and reduce anomalies.

Following the corrosion phase, all specimens were subjected to friction tests. These tests were conducted in accordance with accepted practices.

Results

Table 1 contains some of the information and results of these first tests. Although this data is preliminary and has not been completely evaluated, some of the important trends and findings can be reported.

The data in Table 1 show that there is a 120% increase in the friction factor between the carbon steel guide material (A36) and the carbon steel disc material (A105). The friction factor for the cast carbon steel guide material (A216) and the same disc material (A216) shows an increase of 200% above the uncorroded material. The hardfacing material on the disc sealing surface (stellite) acting on the same stellite hardfacing of the valve body sealing surface shows that the friction factor for the corroded stellite increases 135% above the uncorroded material. All test results reported here were conducted under wet (cold water) conditions.

These changes in friction factor values appear to be very significant; however, the friction factors for all three uncorroded materials have absolute values of around 0.2. Therefore, the friction factors for the three corroded specimens ranged from around 0.45 to 0.60.

**TABLE 1. SPECIMEN AND TEST INFORMATION AND RESULTS**

<table>
<thead>
<tr>
<th>TEST SERIES</th>
<th>DRY OR WET</th>
<th>MATERIAL</th>
<th>VELOCITY (IN/MIN.)</th>
<th>% INCREASE** IN FRICTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>W</td>
<td>A36/A105*</td>
<td>2.5</td>
<td>120%</td>
</tr>
<tr>
<td>II</td>
<td>W</td>
<td>A216/A216</td>
<td>2.5</td>
<td>200%</td>
</tr>
<tr>
<td>III</td>
<td>W</td>
<td>STELLITE/STELLITE</td>
<td>2.5</td>
<td>135%</td>
</tr>
</tbody>
</table>

*Material of shoe (stationary) specimen/material of pull specimen
**% Increase = \((C - U) \times 100\)

C is defined as corroded friction value
U is defined as uncorroded friction value
The transformation of this specimen data to actual valve hardware has yet to be determined. There are still some questions that will require answering before any usage is possible. These questions and others are identified in the "Conclusions" part of this paper.

Problems Encountered

The tests on the carbon steel specimens (A36 on A105), showed that galling occurred when large bearing loads (similar to blowdown conditions) were applied as the normal forces. This galling problem also occurred on the uncorroded specimens because the stationary upper specimen shoe tilted slightly, due to the horizontal friction load, such that the sharp, rear, lower edge of the shoe started to gall the surface of the moving specimen. Although galling occurred on both the corroded and uncorroded carbon steel specimens, the corroded specimens were more susceptible to this condition. The bearing loads were reduced by approximately 50% (for the uncorroded) and by 85% for the corroded specimens. In addition, the stationary shoe support structure was adjusted to reduce the tilt angle of the specimen and minimize the edge galling under loading conditions. These reduced loads and the adjustment eliminated the galling problems; however, other questions on whether actual bearing stress should be applied to simulate the MOV loading condition must be answered. The friction factor results reported herein were recorded during sliding, i.e., without galling.

Conclusions

As stated earlier, there were difficulties in conducting these tests because of the galling tendencies on the carbon steel specimens. This problem was eliminated by adjusting the contact angle of the specimen and by reducing the bearing loads on the specimens. Although the changes in friction are of primary interest in this project, changing test parameters such as reduced bearing stress to eliminate galling raises questions that must be answered. Some of these questions are:

- For corroded specimens, does increased normal force cause increases in friction and is galling a typical occurrence for carbon steels?
- Does relative velocity of specimens cause changes to friction for corroded specimens?
- Is either forward edge or rear edge friction different than surface friction and does this introduce additional variables that must be controlled during testing?
- Can temperature of fluid (for wet test conditions) affect changes in friction?
- Can the changes in friction be directly transformed to actual valve hardware or are the changes dependent on geometry?

There are other important questions that must also be addressed; however, the five questions above must be resolved first.
The test results reported here are preliminary and should not be used for estimating thrust requirements for MOVs having similar materials as those listed in Table 1.

Follow-on and Other Related NRC Research

Potential follow-on work related to this project is being considered by the NRC. Although these first friction tests consider the changes from new materials to 18 months corroded materials, additional specimens of the same materials are being subjected to hot water environments to simulate a 36 months interval. These specimens will be tested to identify whether there are additional increases in friction (above the friction values reported in this paper) or whether the friction may reach a threshold beyond which there will be no additional increases in friction. If the friction is still increasing, another 18 months corrosion cycle may be necessary.

Other MOV aging work related to this corrosion activity is also being supported by the NRC. Recently, an MOV test was started to determine whether changes in thrust due to corrosion can be detected using strain-gage type instruments located on the valve stem. The valve is installed in a pipe at a testing laboratory where it experiences corrosion. The MOV is tested about every 6 months to determine whether a change in thrust can be detected. It is hoped that strain-gage instruments will be sensitive enough to detect and trend small changes in corrosion effects. The first set of test data are currently being evaluated; however, no conclusions can be made at this time.
FAIL SAFE ELECTRIC SPRING RETURN ACTUATORS FOR SAFETY APPLICATIONS IN NUCLEAR REACTORS

WHILE HUNDREDS OF ELECTRIC ACTUATORS ARE USED ON ALL CIRCUITS OF FRENCH PWR REACTORS, CONTROL AND FAIL SAFE FUNCTIONS ARE PROVIDED TODAY BY PNEUMATIC ACTUATORS. THESE PRESENT HOWEVER VARIOUS DRAWBACKS AMONG WHICH THE CONSTANT RELEASE OF AIR IN THE DEPRESSURIZED CONTAINMENT BUILDING, IS THE MOST IMPORTANT.

AS A RESULT THE STUDY DEPARTMENT OF EDF HAD QUESTIONED SEVERAL YEARS AGO SUPPLIERS OF QUALIFIED EQUIPMENT IN VIEW OF PROVIDING FAIL SAFE ELECTRIC ACTUATORS ABLE TO REPLACE PNEUMATIC UNITS. FIRST ATTEMPTS WERE MADE BY ADDING A CLUTCH AND SPRING SYSTEM TO EXISTING ACTUATORS. BUT THE RESULTS WERE QUITE DISAPPOINTING, BOTH FOR SAFETY REASONS BECAUSE OF THE CLUTCH AND BECAUSE OF THE LACK OF COMPACTNESS.

MEANWHILE AND FOR A LONG TIME BERNARD HAD DEVELOPED A SPECIFIC CLUTCHLESS TECHNOLOGY AIMED AT SOLVING THE PROBLEM PRESENTED BY FAIL SAFE ELECTRICS.

WHY IS IT SUCH A PROBLEM TO MAKE ELECTRIC FAIL SAFE WHILE PNEUMATIC ACTUATORS HAVE ALWAYS BEEN ABLE TO OFFER THIS FUNCTION ?

THE ANSWER IS QUITE OBVIOUS WHEN ONE LOOKS AT THE BASIC DESIGN DIFFERENCE BETWEEN PNEUMATIC AND ELECTRIC ACTUATORS.

THE FIRST ARE SIMPLY A RIGID OR FLEXIBLE SURFACE MOVED BY AIR PRESSURE. THE MOVEMENT IS LINEAR AND IT IS EASY TO OPPOSE IT WITH A SPRING WHICH WILL ASSURE A SAFETY POSITION IN CASE OF LACK OF POWER. HOWEVER IT IS DIFFICULT TO CONTROL SPEED OF OPERATION WHICH IS DEPENDANT OF THE VALVE OPERATING FORCE AND FAIL IN PLACE FUNCTION CANNOT BE ASSURED.

ELECTRIC ACTUATORS COMPONENTS ARE AN ELECTRIC ROTARY MOTOR COUPLED TO A GEARBOX PROVIDING A ROTARY MOTION AND SPEED IS DEFINED BY THE CHOICE OF COMPONENTS AND IS ASSURED IN A WIDE RANGE. BY CONSTRUCTION, THESE ACTUATORS ARE SELF LOCKING SO THAT IN CASE OF POWER LOSS THEY ENSURE A FAIL IN PLACE FUNCTION.

SEVERAL SOLUTIONS FOR FAIL SAFE HAVE BEEN CONSIDERED AND IMPLEMENTED FOR A LONG TIME SUCH AS DC POWERED ACTUATORS OR ELECTRO-HYDRAULIC OR ELECTRO-PNEUMATIC SYSTEMS BUT PARTICULARLY IN HARSH ENVIRONMENTS, THE METALLIC SPRING OFFERS THE HIGHEST LEVEL OF RELIABILITY.

SO THE QUESTION TO WHICH WE HAVE BEEN ABLE TO ANSWER IS HOW TO AUTOMATICALLY SWITCH FROM A SELF LOCKING MECHANISM TO A FREE ONE IN CASE OF LOSS OF POWER AND AT THE SAME TIME ENSURE SMOOTH OPERATION TO PROTECT VALVE AND MECHANISM WITHOUT THE USE OF A CLUTCH.
BUT THISIMPORTANTSTEP IS ONLY A PARTIAL SOLUTION TO THE PROBLEM OF REPLACING PNEUMATIC ACTUATORS.

AS WESHAVE SEEN EARLIER PNEUMATIC AND ELECTRIC ACTUATORS HAVE QUITE DIFFERENT OPERATING CHARACTERISTICS AND THE DIRECT REPLACEMENT OF ONE TYPE OF ACTUATOR BY ANOTHER CANNOT BE ACHIEVED EFFICIENTLY IF ONE DOES NOT TAKE INTO ACCOUNT FUNCTIONAL REQUIREMENTS OF THE PROCESS.

IWISH TO PROPOSE YOU A FRAME OF REFERENCE SPECIALLY DESIGNED TO HELP IN THE CHOICE OF THE PROPER ACTUATOR. OF COURSE, DIMENSIONAL AND POWER PARAMETERS OF A VALVE ARE IMPORTANT FACTORS BUT IT IS ALSO ESSENTIAL TO DEFINE FUNCTIONAL CLASSES:

WE HAVE CHOSEN TO DIVIDE USER NEEDS IN 4 CATEGORIES:

- ON-OFF VALVES (SELF EXPLANATORY)
- MODULATING VALVES SEPARATED IN 3 CLASSES:
  . POSITIONING (CLASS III)
  . REGULATION (CLASS II)
  . HIGH-SPEED REGULATION (CLASS I)

THE 3 CLASSES OF MODULATION DIFFER THROUGH VARIATIONS IN FUNCTIONAL REQUIREMENTS FOR PRECISION (INERTIA), SPEED OF OPERATION AND FREQUENCY OF OPERATION.

<table>
<thead>
<tr>
<th>CLASS</th>
<th>PRECISION</th>
<th>SPEED OF OPERATION</th>
<th>FREQ. OF OPERATION</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>%</td>
<td>degrees or mm/sec</td>
<td>Starts / Hour</td>
</tr>
<tr>
<td>III POSITIONING</td>
<td>MODERATE (&lt; 2 %)</td>
<td>LOW</td>
<td>LOW</td>
</tr>
<tr>
<td>II REGULATION</td>
<td>HIGH (&lt; 1 %)</td>
<td>MODERATE</td>
<td>MODERATE</td>
</tr>
<tr>
<td>I HIGH-SPEED REGULATION</td>
<td>VERY HIGH (&lt; 0,5 %)</td>
<td>HIGH</td>
<td>HIGH</td>
</tr>
</tbody>
</table>

THESE VARIOUS REQUIREMENTS BRING DESIGNERS TO OFFER DIFFERENT TECHNICAL SOLUTIONS.

WHILE STANDARD ELECTRICAL MOTORS ARE USED FOR ON-OFF AND CLASS III (POSITIONING), SPECIAL 100 % DUTY RATING MOTORS EQUIP REGULATION ACTUATORS.
FOR HIGH SPEED REGULATION IT IS EVEN NECESSARY TO USE BRUSHLESS DC MOTORS ELECTRONICALLY CONTROLLED.

SIMILARLY THE REDUCTION GEARS ARE CHOSEN IN VIEW OF THE DESIRED MOTION AND SPEED OF OPERATION.

THANKS TO THE PROPER CHOICE OF COMPONENTS WE ARE NOW ABLE TO OFFER A WIDE ARRAY OF PRODUCTS INCLUDING A FAIL SAFE FUNCTION.

WE WOULD LIKE TO SHOW YOU SOME OF THEM:

- LINEAR ON-OFF SPRING-RETURN (FOR GATE VALVE)
- QUARTER-TURN ON-OFF SPRING RETURN FOR BUTTERFLY OR BALL VALVES
- LINEAR REGULATION ACTUATOR
- LINEAR HIGH SPEED REGULATION ACTUATOR

AT THIS TIME TESTS ARE CONDUCTED AT EDF'S "LES RENARDIERES" LABORATORY IN ORDER TO EVALUATE FUNCTIONALLY THESE ACTUATORS.

SIMULTANEOUSLY, EVALUATIONS ARE CONDUCTED AT THE EUROPEAN LEVEL IN ORDER TO ASSESS THE FEASIBILITY OF AN ALL ELECTRIC SOLUTION FOR THE E.P.R. PROGRAM.

WE EXPECT THE OUT COME TO BE POSITIVE, IN WHICH CASE THE NEW ACTUATORS WILL NEED TO BE QUALIFIED FOR THE E.P.R. SEISMIC AND THERMODYNAMIC REQUIREMENTS.

THROUGH OUR PAST EXPERIENCE IN THIS FIELD WE DO NOT FORESEE ANY MAJOR OBSTACLES.

SO IN THE NEAR FUTURE IT WILL BE POSSIBLE TO EQUIP PWR NUCLEAR REACTORS WITH ACTUATORS EXCLUSIVELY ELECTRIC.

WHAT WE HAVE TRIED TO EVIDENCE HOWEVER IS THAT IN ORDER TO DO THIS EFFECTIVELY IT IS IMPORTANT TO ASSESS THE FUNCTIONAL REQUIREMENTS OF EACH CIRCUIT BECAUSE STRAIGHT FORWARD REPLACEMENT OF PNEUMATIC ACTUATORS BY ELECTRIC ONES IS IN FACT AN AKWARD PROCESS AND BETTER RESULTS WILL BE ACHIEVED BY MAXIMIZING THE FIT BETWEEN ACTUAL NEEDS AND ELECTRIC ACTUATORS PERFORMANCES.
NON-INTRUSIVE MOTOR OPERATED VALVE
DIAGNOSTIC EQUIPMENT
AND
TESTING EXPERIENCES

JOINT NEA/IAEA SPECIALIST
MEETING ON MOTOR OPERATED VALVE ISSUES

PARIS, FRANCE
APRIL 27, 1994

WRITTEN BY
JAMES J. BALASCHAK

Teledyne Brown Engineering
Engineering Services
513 MILL STREET • MARION, MASSACHUSETTS 02738-0258
INTRODUCTION

Companies operating nuclear power plants recognize that the correct functioning of all motor operated valves, and particularly those in safety-related systems, is of paramount importance. In the United States, the Nuclear Regulatory Commission (NRC) has issued Generic Letter 89-10 (Reference 1) relative to this concern. Operability must be demonstrated, under design-basis conditions, if practical.

Over the past several years, several valve diagnostic systems have been used to determine the condition of the valve and actuator by monitoring signals representing stem thrust, spring pack deflection, motor current, voltage or power, and switches, usually in some non-inclusive combination. This information is used both to determine operability and to pin-point problem areas of the valve and actuator that require maintenance or repair.

This paper describes a motor operated valve testing and diagnostic system that differs from other systems in several significant areas, including operational modes and flexibility. The technical approach varies primarily in the technique used to sense direct stem thrust and torque and in the system used to acquire the test data.

The paper will present the technology and review several typical motor operated valve test experience results.
BACKGROUND

On June 28, 1989, the U.S. Nuclear Regulatory Commission (NRC) issued Generic Letter 89-10 (Reference 1) to all holders of nuclear power plant operating licenses and construction permits. The generic letter requested the plant owners and operators establish a program to provide for the testing, inspection, and maintenance of safety-related Motor Operated Valves (MOVs). The program was deemed necessary by the NRC to provide assurance that the MOVs will function when subjected to design-basis conditions that occur during normal operations and abnormal predictable events.

Although MOV problems had been recognized in the industry for many years, the breadth of these problems were not widely realized until a loss-of-feedwater event occurred at the Davis-Besse Nuclear Power Plant on June 9, 1985. On that date, the pressurized water reactor was operating at a high power level when the main feedwater pumps automatically tripped and the auxiliary feedwater (AFW) system was activated. An operator initiated the steam and feedwater rupture control system by mistake on a low steam pressure signal rather than the required low steam generator level signal. This caused the AFW isolation valves to close automatically. The operator realized his error and reset the control system. The MOVs in the AFW system, however, were not able to be re-opened electrically from the control room. The operators were able, however, to open the MOVs manually and to re-initiate flow to the steam generators. Later it was found the MOVs failed to open since the torque bypass-switch set points prevented the signal bypassing the torque switch for a sufficient period to allow the actuator to open the valve with a high differential pressure. This event highlighted a significant concern regarding the ability of MOVs to operate under design-basis conditions.
As a consequence of the Davis-Besse event, and other industry research, the NRC developed Bulletin 85-03 (Reference 2) which provided for the establishment of programs to ensure that switch settings on MOVs located in the high-pressure coolant injection/core spray and emergency feedwater systems (reactor core isolation cooling in boiling water reactors) were selected, set, and maintained properly. The program anticipated this effort to be completed within about two years.

The implementation of Bulletin 85-03 determined that many more MOVs than expected would not have been able to operate under the design-basis conditions. The failure rate, suggested by the results from the implementation of the bulletin, was much higher than probabilistic risk assessments had assumed. Additional deficiencies beyond switch settings were also identified.

By the Generic Letter 89-10, the NRC extended the scope of the program outlined in Bulletin 85-03 to include all safety-related MOVs. The NRC recommended that licensees develop and implement a program to ensure that motor-operated valve switch settings (torque, torque bypass, etc.) be selected, set and maintained for the life of the plant.

One of the major differences between Generic Letter 89-10 and Bulletin 85-03 is in the scope of the two documents. Generic Letter 89-10 affects more MOVs than Bulletin 85-03 because of the greater number of systems within its scope. It was estimated that Bulletin 85-03 covered about 25 to 30 MOVs per reactor unit and that Generic Letter 89-10 would address approximately 175 MOVs per unit.
Idaho National Engineering Laboratory (INEL), at the behest of the NRC, conducted full scale blowdown testing (Reference 3) which indicated that the industry sizing equations for MOVs were not conservative for design-basis conditions. INEL noted that measurement of both stem thrust and motor torque were necessary to ensure prediction of the MOV. The INEL tests at that time, however, did not look at the limitations of specific MOV diagnostic systems. This was to be later addressed by industry sponsored tests, NRC supplements to GL 89-10 and NRC information notices.

Industry sponsored testing at INEL (Reference 4) initiated events leading to the issuance of Supplement 5 to GL 89-10. The supplement addressed the inadequacy of test equipment which rely on spring-pack displacement or valve yoke strain to estimate stem thrust. As a result, utilities in turn were obliged to re-evaluate test methods. Utilities then insisted on full documentation of MOV diagnostic vendor accuracy claims. In fact, two diagnostic equipment vendors notified the NRC in accordance with 10CFR21 that previous use of those vendors' thrust measurement devices may create a safety hazard as defined by NRC regulations. Licensees' responsiveness to Supplement 5 to GL 89-10 lead to the publication of Information Notice 94-18 to alert licensees of accuracy problems (Reference 5) which described specific diagnostic equipment problems. The Teledyne Brown Engineering (TBE) sensing technology described in this paper does not contain any of these limitations.
TECHNICAL APPROACH

The approach described below differs from other MOV diagnostic systems primarily in the technique used to sense stem thrust and in the system used to acquire the test data.

Sensing Stem Thrust: Test the Stem

Our approach measures direct stem thrust, and/or torque, using bonded resistance strain gages mounted on the valve stem. This is the most direct and accurate means of measuring the loads experienced by the stem. Indeed, an Electric Power Research Institute (EPRI) study (Reference 6) stated the best way to monitor events on a motor-operated valve (MOV) was the "direct measurement of valve stem load through the use of strain gages attached to the stem".

There are many inherent and practical advantages to using this type of sensor over those that rely on attachment points or clamping to obtain a measurement. These advantages include the following:

1. Direct measurement as a function of material strain.

2. Totally non-intrusive.

3. No moving parts and no relative motion required.

4. Demonstrated accuracy, reliability and linearity.

5. Proven environmental protection techniques.

6. Permanent sensor installation reduces the time required for subsequent tests.
Another major advantage of direct strain gage to stem attachment is the opportunity to completely cancel unwanted mechanical effects, so that the thrust bridge, for example, does not depend on an elaborate calibration procedure to determine the effect of indirect load paths or non-uniform geometries on the sensor output. By installing the thrust gages directly on the valve stem in diametrically opposed pairs, and connecting each set of four strain gages correctly in a full Wheatstone bridge configuration, the thrust bridge responds to stem tension and compression only, the torsional and bending effects are cancelled. These techniques, used for many years in the transducer industry, are applied daily in standard load cell and torque transducer product lines, and are well documented (References 7, 8 and 9).

The valve stem, when instrumented with a full Wheatstone bridge strain gage circuit, is transformed into a column load cell. The column load cell is one of the earliest strain gage transducers. The design has four strain gages, two in the longitudinal direction and two oriented transversely to detect Poisson strain. The gages are oriented 180° apart and connected in a full Wheatstone bridge circuit (Figure 1). A full bridge for torque measurement is applied in the same manner. The completed system has the ability to conveniently measure static and dynamic strains.

The strain gage sensitivity for determining uncalibrated thrust and torque are of the general forms:

\[
\text{Thrust} = C_T D^2 \frac{E}{1+\nu} \text{ (LBS) } \frac{\text{mV/V}}{}
\]

\[
\text{Torque} = C_Q D^3 \frac{E}{1+\nu} \text{ (FT-LBS) } \frac{\text{mV/V}}{}
\]

Where:  
\(C_T, C_Q\) are Constants  
\(D\) = Stem Diameter (IN)  
\(E\) = Young’s Modulus of Stem Material/10^6 PSI  
\(\nu\) = Poisson’s Ratio of Stem Material
FIGURE 1. PRINCIPLE OF THE COLUMN LOAD CELL
Quick Stem Sensor™

The Quick Stem Sensor™ (QSS) (Figure 2) is a patented device which is applied to the unthreaded part of a valve stem using a layer of adhesive. Strain gages, circuitry, and pre-wired connectors are included in the one-piece transducer. The QSS™ makes possible an accurate nonintrusive measurement and enhances installation by reducing time without compromising the quality of the measurement. After mounting and curing, the QSS™ can be used with or without calibration depending on accuracy requirements.

There are valves having no unthreaded stem available for the installation of Quick Stem Sensors. For those valves having no unthreaded stem available, we have developed a tool to remove threads to create a smoothed area. Alternately, a fully instrumented replacement stem, or SMARTSTEM™ can be installed.

SMARTSTEM™

The patented SMARTSTEM™ transducer (Figure 3) is a direct replacement of the valve stem supplied by the manufacturer. The SMARTSTEM™ measures loads using a strain gage sensing element sensitive to thrust and/or torque loads. Strain is related to thrust or torque during a laboratory calibration. This calibration is traceable to the National Institute of Standards and Technology (NIST). The patented SMARTSTEM™ is the most direct and accurate means of measuring loads, and has a calibrated accuracy to 0.50%, validated at Idaho National Engineering Laboratories (INEL).
FIGURE 2. QUICK STEM SENSOR™.
FIGURE 3. SMARTSTEM™.
ACCURACY VALIDATION TESTING RESULTS AND QUALITY ASSURANCE

The in-situ valve stem gaging techniques described above and the diagnostic system were validated independently by TBE and at INEL in January 1991. Validation test reports (References 10-14) are available on a proprietary basis to clients which contains both analytical accuracy determination and experimental validation of the transducer accuracies. The SMARTSTEM™ is a NIST traceable calibrated transducer. The calibration program meets the requirements of 10CFR50, Appendix B and the SMARTSTEM™ is provided with a certificate of calibration.

We also participated in the Motor Operated Valve Users’ Group (MUG) and NRC sponsored validation tests at INEL during the week of May 20, 1991. The results of this validation test have been issued to all licensees. The validation results met or exceeded published accuracy claims. The accuracies are given below.

<table>
<thead>
<tr>
<th>Method</th>
<th>Inaccuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMARTSTEM™ Transducer</td>
<td>±0.5% Full Scale</td>
</tr>
<tr>
<td>Direct Stem Gaging, using TBE System, End-to-end</td>
<td>±8.2% Reading</td>
</tr>
<tr>
<td>Quick Stem Sensor, Uncalibrated</td>
<td>±8.2% of Reading</td>
</tr>
<tr>
<td>Quick Stem Sensor, Calibrated In-situ</td>
<td>±3.0% of Reading</td>
</tr>
</tbody>
</table>

The TBE Quality Assurance Program for diagnostic test equipment, transducers, and valve testing meets the requirements of 10CFR50, Appendix B, 10CFR21, ANSI/ASME NQA-1, ANSI N45.2 and ASME BPVC Section III.
DATA ACQUISITION AND ANALYSIS

There are a variety of optional methods to monitor and record strain gage data. These range from a simple strain indicator for static monitoring to high-speed digital computer data acquisition systems. Currently, the preferred system for MOV testing is a personal computer based recording system which uses signal conditioning. In fact, the QSS™ and SMARTSTEM™ have been used by all MOV systems available in the U.S. market today.

The second area differentiating our approach from others, and building on the permanent sensor capability described above, is data acquisition, capable of flexible operating modes.

When the sensors are installed permanently, several advantages accrue from the portable capability:

- Waiting time is reduced from the test. All that is necessary is to stroke the valve once the test conditions have been achieved. Multiple strokes can be accommodated with the on-board memory.

- ALARA concerns are addressed directly by the approach. The result is drastically reduced exposure, particularly after the baseline testing phase is over and set point verification tests are performed.

- It is now feasible to send MOV sensor data to some central point via cable or optical fiber. Having the sensors permanently installed and cabled to a common point will greatly reduce the cost and effort of tests.

- In the future, the SMARTSTEM™ will allow MOVs to be controlled by actual stem thrust rather than by estimated torque derived from spring pack displacement through a Digital Thrust Switch™ (DTS™).
QUIKLOOK™ (Figure 4) is a personal computer (PC) based software and hardware package used to acquire data from strain gages and other sensors attached to MOVs. Program QUIKLOOK™ uses a high speed data acquisition PC interface card. The interface card allows direct sensor input without external signal conditioning. There are no external break-out, signal conditioning or junction boxes. Hence, the unit requires only one operator.

QUIKLOOK™ acquires up to eight channels of strain gage or voltage based signal data at a rate of 1000 samples per second. The duration of sampling is approximately four (4) minutes at 1000 sps.

QUIKLOOK™ allows for instantaneous review of signals after stroking the valve. This permits valve diagnostics and analysis without extensive delays for signal processing. Signatures are reviewed and values such as thrust at torque switch trip or final load are extracted as well as switches, motor current or differential pressure. Sensor channels are user defined to allow flexibility for applications other than MOVs. The signatures are stored to allow extensive analysis as required by the user.

We have developed custom command and menu files to automate specific applications for MOV data analysis. This approach complements the flexibility inherent in the QUIKLOOK™ concept; additional analyses can easily be configured to analyze QUIKLOOK™ data from any type of test.

Software program verification has been performed to the quality requirements of 10CFR50 Appendix B. The software supports simultaneous display of one to one hundred windows defined by the user. Individual traces may be zoomed in x and/or y, scrolled, scaled, or printed out. FFT functions are available to determine frequency content of valve signatures.
FIGURE 4. QUIKLOOK™.
Data in one series may be plotted against another, in xy fashion, or may be overplotted. In addition to the many functions available the software is also be used for trending. A thrust trace from last month or year may be plotted directly over one from today. There is also no limit on data series length; i.e., multiple open/close cycles may be displayed on one plot.

Program QUIKLOOK™ is calibrated within a 1% accuracy traceable to N.I.S.T.

**EXAMPLES OF TESTING EXPERIENCES**

In order to illustrate the capabilities of the equipment and sensors described above, we have selected several representative examples of test programs conducted in response to GL 89-10.

**Butterfly Valve Full Differential Pressure Tests**

TBE was engaged by a midwestern U.S. utility to validate actuator sizing criteria for their safety-related butterfly valves. The pressurized water reactor has a large number of butterfly valves in critical plant service. In response to GL 89-10, a test program was organized to measure actual stem torque at near design basis differential pressure and flow condition. The butterfly valves were 150# Standard Rated (8", 12", 14", 20") with a non-symmetric disc and single offset stem. The measured torque was compared to the analytical requirements of the valve manufacturer.

The actual stem torque was measured using the QSST™ mounted directly on the valve stem. In addition, upstream and downstream pressure, flow and valve disk position were also measured. The QUIKLOOK™ data acquisition system was used to record and analyze the signals.
The measurement devices used in the tests are shown in Figure 5. Typical results of the analyses are shown in Figures 6 and 7 which illustrates the actual required torque to open and close the valve under static (no flow, no pressure) and dynamic (full flow and pressure) condition. Additional analyses allows use of the measured differential pressure to determine the analytical sizing criteria requirements of the valve manufacturer (Figures 8 and 9).

The results indicate that the current performance prediction methodology may not adequately predict require torque. In-situ testing is recommended to determine torque requirements, especially when the valves are located near upstream flow disturbances.

**Flow Tests of Globe and Gate Valves**

An owner of three PWRs in the southwest U.S. was one of the first plants to recognize the inherent qualities of direct measurement of MOV stem torque and thrust. The QSS™ was used as the primary sensing device for their GL 89-10 program. As stated earlier, the strain gage circuit can be input to all available MOV diagnostic systems. This utility utilized the ITI MOVATS 3000 system, another PC based data acquisition system. The test programs were performed at full design conditions to meet the requirements of GL 89-10. Actual stem thrust was compared to analytical predictions to determine operability and correct switch settings. Additional values such as stem factor (ratio of operator output torque to delivered thrust) and stem friction were determined. The QSS™ is the only available device which accurately measures these variables. Indeed, research has shown that these variables have importance in predictable valve behavior (Reference 15). Highlights of this program were presented at the MUG meeting on February 2, 1994.
## IN-SITU TRANSDUCERS

**QUIKLOOK™ Data Acquisition System**

<table>
<thead>
<tr>
<th>Channel</th>
<th>Measurement</th>
<th>Transducer</th>
<th>Output</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Torque</td>
<td>QSS</td>
<td>±2.5mV/V</td>
<td>±9.8% Reading</td>
</tr>
<tr>
<td>2</td>
<td>Upstream Pressure</td>
<td>Omega PD</td>
<td>1-5Vdc</td>
<td>±1.0% Reading</td>
</tr>
<tr>
<td>3</td>
<td>Downstream Pressure</td>
<td>Omega PD</td>
<td>1-5Vdc</td>
<td>±1.0% Reading</td>
</tr>
<tr>
<td>4</td>
<td>Flow</td>
<td>Ultrasonic Flowmeter</td>
<td>0-10Vdc</td>
<td>±3.0% Reading</td>
</tr>
<tr>
<td>5</td>
<td>Rotation</td>
<td>Linear String Pot</td>
<td>0-4Vdc</td>
<td>±0.1% Reading</td>
</tr>
<tr>
<td>6</td>
<td>Unused</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Unused</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Unused</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**FIGURE 5. TRANSDUCERS FOR BUTTERFLY VALVE.**
FIGURE 6. BUTTERFLY VALVE TORQUE RESULTS.

FIGURE 7. BUTTERFLY VALVE TORQUE RESULTS.
FIGURE 8. DIFFERENTIAL PRESSURE VS. T5 PARAMETER.

FIGURE 9. SIZING PARAMETERS BASED UPON DIFFERENTIAL PRESSURE.
Figures 10, 11, and 12 show the test results for a High Pressure Safety Injection (HPSI), Shutdown Cooling Isolation (SCI), and Auxiliary Feedwater (AFW) valves, respectively. The HPSI valve is a globe type with a unique rising rotating valve stem. The SCI and AFW valves are gate types with typical rising stems.

Measurements of Valve Stem Factor and Friction Coefficient

One of the important parameters for rising stem valves is the stem factor. This is valuable in order to determine the conversion of actual output torque of the operator to stem thrust. The stem factor is a function of stem thread geometry, machined finish, and lubrication. The stem factor can be determined by simultaneously measuring stem torque and thrust with the QSS\textsuperscript{tm} or SMARTSTEM\textsuperscript{tm}. Similarly, the stem coefficient of friction can also be determined. The friction coefficient indicates the quality of lubrication at the valve stem and stem nut surfaces and finish of contact surfaces.

Figures 13 and 14 depict the valve stem factor and coefficient of friction through a complete valve stroke, open to close.

SMARTSTEM\textsuperscript{tm} Experiences

The SMARTSTEM\textsuperscript{tm} is a direct replacement of the manufacturer’s valve stem. In fact, in the United States, SMARTSTEM\textsuperscript{tm}s have been supplied for replacement valves through all major valve manufacturers to utility clients. Several U.S. utilities specify SMARTSTEM\textsuperscript{tm}s for any new valve in a safety-related function. Additionally, The TBE SMARTSTEM\textsuperscript{tm} was selected by the Electric Power Research Institute for the MOV Performance Prediction Program due for accuracy requirements.
FIGURE 10. CLOSE-TO-OPEN-TO-CLOSE FLOWING TEST.
HIGH PRESSURE SAFETY INJECTION RISING/ROTATING GLOBE VALVE.

FIGURE 11. CLOSE-TO-OPEN AUXILIARY FEEDWATER ISOLATION GATE VALVE.
FIGURE 12. CLOSE-TO-OPEN HYDRO-PUMP TEST. SHUTDOWN COOLING ISOLATION GATE VALVE.

FIGURE 13. STEM FACTOR VERSUS STEM POSITION.
FIGURE 14. STEM FRICTION COEFFICIENT VERSUS STEM POSITION.
For new plant construction outside the U.S., plant designers are considering specifying SMARTSTEM™s for all safety-related valves. These SMARTSTEM™s would be linked to a central data acquisition point, where valve thrust and torque data would be recorded automatically during valve operation.

**SMARTSTEM™ for Testing Valve Performance**

The Electric Power Research Institute (EPRI) conducted an MOV performance prediction program to validate the analytical methodology to determine required valve thrust at design basis conditions (Reference 16). The SMARTSTEM™ was selected for the program for use in approximately forty (40) gate, butterfly and globe valves to measure direct stem torque and thrust.

After multiple preconditioning valve operations, and numerous d.p. tests, the valves were disassembled and the SMARTSTEM™s returned for post-test calibrations. Time between initial and final calibrations ranged from three to seventeen months. The results of the recalibrations indicated an average change in thrust/torque sensitivity of 0.02% for thirty-seven SMARTSTEM™s.

Figures 15 and 16 show closing and opening results respectively for a similar flow performance during accident blowdown tests for a flex wedge gate valve (Reference 17). The testing demonstrated the reliability of the double disc valve design to close and isolate flow under blowdown conditions. The stem thrust required was determined by the standard industry equations:
FIGURE 15. 6" FW GATE VALVE CLOSING TEST.

FIGURE 16. 6" FW GATE VALVE OPENING TEST.
Ts = Tp + Te + (μ * As * DP)

where

Ts = stem thrust required to isolate (lbs)
Tp = packing friction (lbs)
Te = stem load (lbs)
μ = valve factor
As = seat area, based on mean seat diameter (in²)
DP = differential pressure (psi)

The SMARTSTEM™ was used to measure stem thrust and torque during the test. The test conditions approximated PORV isolation service for a PWR, using water at 2450 psig and 650° with flow up to 230,000 lb/hr.

The response of the SMARTSTEM™ strain gages showed no effect of dynamics due to flow. The thrust signature obtained was repeatable throughout the multiple tests.

Measuring Torque Conversion Efficiency

Figures 17 and 18 show results of SMARTSTEM™s used to determine torque conversion efficiency for valve stem nuts (Reference 18). By measuring stem torque and thrust simultaneously, it was determined that the roller screw provided low friction and high torque to thrust efficiencies compared to the ACME screw.
Figure 17. Torque-to-Thrust Conversion.

Figure 18. Thrust-to-Torque Stem Factor.
CONCLUSION

The most accurate means to measure a parameter is directly. The key to accuracy is isolating the force to measure. By mounting strain gages directly on the valve stem, stem torque and thrust are accurately measured. The Quick Stem Sensor™ and SMARTSTEM™ are practical methods to ensure the correct functioning of a motor operated valve, when using diagnostic equipment such as QUIKLOOK™, to sense MOV information. The QST™ and SMARTSTEM™ and QUIKLOOK™ can be used to determine a variety of MOV characteristics, including but not limited to, required stem thrust and/or torque for stem factor, stem-to-stem nut coefficient of friction, design basis conditions, and torque-to-thrust conversion efficiency, and running loads for globe, gate and butterfly motor operated valves.
REFERENCES


5. U.S. NRC Information Notice 94-18: Accuracy of Motor-Operated Valve Diagnostic Equipment (Responses to Supplement 5 to Generic Letter 89-10).


ACKNOWLEDGEMENTS

Data traces were generously supplied by Mssrs. Eddie Edmonds and Michael Richard.
"EMG-DREHMO-actuator with controlled friction clutch"

Limitation of torque excess

Author: ELEKTRO-MECHANIK GmbH
W. Händel
57482 Wenden

AEG/EMG manufactures electromechanical actuators for more than 40 years in torque range up to 10 000 Nm and in a power range up to 27 kW (foil 1).

The first actuators for nuclear power stations were delivered at the beginning of the 60th. The gear notes base in all lines on worm gear pairs combined with spur gears. A special element of the EMG-actuators for power stations is the controlled friction clutch.

The controlled friction clutch is a design element for the effective limitation of excessive torques during operational disconnection procedures and disconnection failures.

It is to be found in the actuator series DA and DB (foil 2 and 3).

The power flux is transmitted from the motor shaft to the clutch shaft and via the corresponding feather or serrated connection to the multi-disc cage. The torque is transmitted to the inner discs via the dogs of the outer discs, which engage into the grooves of the multi-disc cage. The dogs of the inner discs engage into the corresponding worm grooves. A sufficient excess of transmittable torque of the multi-disc clutch relative to the motor torque to be transmitted is provided via the pretensioned springs (foils 4 and 5).

As in the case of an actuator with a traverse worm, upon torque build-up the worm effects an excursion against the force of the torque spring proportionately to the peripheral force acting on the worm wheel. This excursion does not begin until a certain limit torque is reached, this being specified by the pretensioning of the spring and hence the preset disconnection torque.

In the process the ball ring is initially shifted onto a cylindrical track together with the multi-disc cage. The torque disconnection is effected after a certain excursion path.

If even greater torques build up on the worm wheel due to disconnection failure or after-running, the traverse process continues. The ball ring reaches the tapered part of the guidance groove and the balls are forced onto a track or larger diameter. Due to this they are pressed into the conical gap between the ball thrust ring of the worm and the multi-disc cage. This brings about a separation of the discs and hence of the power flux. The worm stops and the multi-disc cage continues to rotate at the speed of the motor. Only the roll friction of the ball ring is then transmitted to the worm, the former now acting like a ball-bearing.

Due to the breakdown in power flux the worm performs a return excursion. Contact pressure with the disc pack now comes about and at a constant counter-torque an oscillating engagement and disengagement of the clutch thus sets in. This procedure represents a control process.
A constant slide torque is measurable at the output drive of the actuator. In terms of the excessive torque a drive with a controlled friction clutch behaves like drive with a traverse worm (foil 6) until the slide torque is reached. Any torques in excess of this are cut off.

The controlled friction clutch is not a continuous slip clutch. Its purpose is to absorb the residual energy arising from normal disconnection procedures. If the slip procedures take longer the disc packs wear and the transmittable torque diminishes (foil 7). Under very unfavourable conditions a baking of the discs may even come about, which can never occur during the first slippage. The disc packs have to be replaced following an unacceptably long slip procedure. This presumes that unacceptable slippage is reliably recognized.

An electronic slippage monitoring system is supplied for drives with a controlled friction clutch in series DB2, 3, 4 and DA35, 36, 106, 107.

The speed is measured before the clutch (at the motor or clutch shaft) and behind the clutch (in the limit switch assembly, foil 8). If operation is free of slippage, a constant ratio, must exist between these two values. If a deviation is measured for a longer period than the adjustable value $t_{\text{limit}}$, a signal is emitted by the evaluation device located outside of the drive.

No signal is emitted in the start-up process when the gearbox backlash is modulated, if operation is by handwheel and if operational slippage is desired at the limits for the purpose of torque limitation.

In order to acquire the requisite speeds, crown gears are mounted on the output drive shaft of the motor and on the limit switch assembly. Robust, maintenance-free inductive pulse generators (initiators) on the NAMUR principle allow reliable acquisition of the rotational movement. The rushing past the initiators of the teeth leads to a corresponding sequence that is a dimension of the speed.

The initial intention is to work out the quantitative difference with respect to excessive motor torque between drives with a traverse worm and drives with a controlled friction clutch.

For the motor rating the requisite start-up and stalling torques are calculated from the armature torque demanded (with the additions and safety margins already concealed therein) and the reserves for subvoltage operation, temperature and efficiency uncertainty factors (foil 9).

The maximum actuator torque of a drive with a traverse worm technique is calculated in the case of disconnection failures in accordance with the equation shown on foil 10. Excesses can occur as a consequence of motor and gearbox efficiency fluctuations and as a consequence of dynamic effects in the limit range.

This simple example shows a ratio of 3.5 between the actually requisite torque and the maximum possible torque (foil 11).

The controlled friction clutch limits this ratio to the value 1.5 (foil 12).

The advantages associated with this for the design-rating of components located behind the actuator in the force flux are apparent.
EMG-actuators in nuclear power stations
DREHMO-actuators "heavy duty type"

<table>
<thead>
<tr>
<th>program</th>
<th>size acc. to DIN 3210</th>
<th>type</th>
<th>min (^{-1})/speed</th>
<th>max. starting torque Nm</th>
<th>torque range Nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>●</td>
<td>DCO</td>
<td>15...190</td>
<td>100</td>
<td>20 - 60</td>
<td></td>
</tr>
<tr>
<td></td>
<td>15...190</td>
<td>150</td>
<td>40 - 120</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>DC1</td>
<td>15...190</td>
<td>150</td>
<td>40 - 120</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>DC2</td>
<td>8...95</td>
<td>300</td>
<td>80 - 240</td>
<td></td>
</tr>
<tr>
<td></td>
<td>5...65</td>
<td>450</td>
<td>120 - 360</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>DC3</td>
<td>4...48</td>
<td>600</td>
<td>150 - 450</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2...30</td>
<td>950</td>
<td>250 - 750</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>DB2</td>
<td>20...513</td>
<td>-</td>
<td>80 - 200</td>
<td></td>
</tr>
<tr>
<td></td>
<td>20...513</td>
<td>-</td>
<td>200 - 400</td>
<td></td>
<td></td>
</tr>
<tr>
<td>●</td>
<td>DB3</td>
<td>6...168</td>
<td>-</td>
<td>550 - 1100</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>DB4</td>
<td>4...103</td>
<td>-</td>
<td>1000 - 2000*</td>
<td></td>
</tr>
<tr>
<td>5 u. 6</td>
<td>DA35/DA36</td>
<td>12...51</td>
<td>-</td>
<td>2000 - 5000</td>
<td></td>
</tr>
<tr>
<td>6 u. 7</td>
<td>DA106/DA107</td>
<td>12...47</td>
<td>-</td>
<td>4000 - 10000</td>
<td></td>
</tr>
</tbody>
</table>

* 1700 Nm according to KTA 3504
EMG-actuators in nuclear power stations
sectional presentation DREHMO-actuator DB 2 - DB 4
EMG-actuators in nuclear power stations
funktion
EMG-actuators in nuclear power stations
funktion
EMG-actuators in nuclear power stations

\[ M_{\text{dmax}} \text{ (Nm)} \]

\[ M_{\text{slide}} \]

2-pole motor
(determined by computation)

6-pole motor
(determined by computation)

\( X = \) measured quantities

\[ tv \text{ (ms)} \]
EMG-actuators in nuclear power stations funktion

Test. Friction clutch (continuous slip)
Typ: DB 4 B 114 f
EMG-actuator in nuclear power stations
controlled friction clutch
slip monitoring
EMG-actuators in nuclear power stations
Motor torque-excess balance

\[
M_{K\text{motor}} = \frac{M_{\text{drive}}}{i_{\text{Get}} \cdot \eta_{\text{Get}}} \cdot \frac{1}{0.8^2} \cdot 1.35
\]

subvoltage reserve
reserve for emergency temperature and efficiency tolerance
EMG-actuators in nuclear power stations
Motor torque-excess balance

\[ M_{\text{Max}} = M_{\text{Kmotor}} \cdot i_{\text{Get}} \cdot \eta_{\text{Get}} \cdot \frac{1,1^2 \cdot 1,15 \cdot k_{\text{dyn}}}{C_A, C_F, \ldots} \]

- Motor torque due to overvoltage
- Motor and efficiency tolerance
- Kinetic residual energy

\[ M_{\text{Max}} = M_{\text{drive}} \cdot 3.5 \]

With \( k_{\text{dyn}} = 1.20 \) as example

---

512 - 10 -
EMG-actuators in nuclear power stations
Motor torque-excess balance

\[
\frac{M}{M_{erf}} \cdot 100
\]

- 400%
- 300%
- 200%
- 100%

- Excessive torque by kinetic residual energy
- Excessive torque by tolerance
- Excessive torque by overvoltage
- Reserve for motor and efficiency tolerance and temperature
- Reserve for voltage drop
- Necessary torque
150% motor torque delimitation by means of controlled friction clutch
RESEARCH AND DEVELOPMENT

Chairman: A. Perez (CSN, Spain)

Mr. Bahir Eldiwayi (Kalsi Engineering, U.S.A.) presented the methodology for torque prediction of butterfly MOVs; this development has been sponsored by the Electric Power Research Institute (EPRI) as part of the Motor Operated Valve Performance Prediction Program. Mr. Eldiwayi explained the different torque components, the disk shapes under consideration, the equations used and the involved coefficients, under the postulated flow conditions. He concluded by illustrating the good agreement of the model with the test data gathered from a comprehensive flow loop and in situ test program.

Mr. Anton Scheuer (TÜV Rheinland e.V., Germany) presented the guidelines, contained in VdTÜV resolution number 171, to calculate the stem forces for gate and globe valves; such guidelines aim to ensure that a uniform procedure is adopted by the experts of the VdTÜV (technical inspectorate) and the GRS for evaluation of the safety related MOVs in the German NPPs. He presented the equations used to determine the thrust/torque requirements for valve opening and closing, the margin for disk unwedging and the permissible stresses on valve parts.

Mr. John F. Hosler (EPRI, U.S.A.) presented an overview of the EPRI Motor Operated Valve Performance Prediction Program. The objectives of the program, that intends to cover up to the 90 percent of the safety related MOV population, are to better understand the factors affecting the performance of the valves, and develop and validate methodologies to predict such performance. He summarized the very broad test program performed, highlighting some of its results: the big dispersion of the observed sliding friction coefficients (0.2 to 0.6 for gate valves, in cold water flow conditions); the importance of the ROL effect, the low potential for damage in pumped flow conditions, and the increased damage potential in blowdown situation.
Improvement in Butterfly Valve Torque Prediction Models Based on Recent Research

Bahir H. Eldiwany
M. S. Kalsi
Vinod Sharma
Kalsi Engineering, Inc.

ABSTRACT
As part of the Motor-Operated Valve Performance Prediction Program, the Electric Power Research Institute (EPRI) has sponsored the development of methodologies for predicting thrust and torque requirements of gate, globe, and butterfly MOVs. This paper presents the butterfly valve methodology which will be used by utilities to calculate the dynamic torque requirements for butterfly valves.

The total dynamic torque at any disk position is the sum of the hydrodynamic torque, bearing torque (which is induced by the hydrodynamic force), as well as other small torque components (such as packing torque). The hydrodynamic torque on the valve disk, caused by the fluid flow through the valve, depends on the disk angle, flow velocity, upstream flow disturbances, disk shape, and the disk aspect ratio. The butterfly valve model provides sets of nondimensional flow and torque coefficients that can be used to predict flow rate and hydrodynamic torque throughout the disk stroke and to calculate the required actuation torque and the maximum transmitted torque throughout the opening and closing stroke.

The scope of the model includes symmetric and nonsymmetric disks of different shapes and aspect ratios in compressible and incompressible fluid applications under both choked and nonchoked flow conditions.

The model features were validated against test data from a comprehensive flow loop and in situ test program. These tests were designed to systematically address the effect of the following parameters on the required torque: valve size, disk shapes and disk aspect ratios, upstream elbow orientation and its proximity, and flow conditions. The applicability of the nondimensional coefficients to valves of different sizes was validated by performing tests on a 42-inch valve and a precisely scaled 6-inch model. The butterfly valve model torque predictions are found to bound test data from the flow loop and in situ testing, as shown in the examples provided in this paper.

INTRODUCTION
The Electric Power Research Institute (EPRI) is developing improved and validated models for predicting the thrust required to operate gate and globe valves, and the torque required to operate butterfly valves in nuclear power plant applications. The focus of this paper is to present the key aspects of the butterfly valve model. The model will be integrated into a PC-based computer program which will be used by the utilities to assess the thrust or torque requirements of motor-operated valves (MOVs) under user-specified design basis conditions.

The basic equations used in the butterfly valve model and the associated assumptions and limitations are summarized. The model predictions have been compared to test data to validate the prediction methodology over a wide range of valve design features and test conditions. Typical examples of these model/data comparisons and conclusions regarding the scope and applicability of the butterfly valve model are presented.

Presented at the Joint NEA/IAEA Meeting
Paris, France — April 25 - 27, 1994

523


**Previous Research**

Prior to starting the development of the EPRI butterfly valve model, an extensive literature search was performed to document the technical state-of-the-art and to identify areas of needed improvement. Results of the literature search are documented in the Application Guide for Motor-Operated Butterfly Valves in Nuclear Power Plants (Eldiwany and Kalsi, 1993). A thorough evaluation of data from previous analytical and experimental research, manufacturers' recommendations for actuator sizing, and the commonly used industry standard (ANSI/AWWA Standard for Rubber-Seat ed Butterfly Valves, 1988) revealed that the following areas needed improvements in developing a butterfly valve torque prediction model suitable for nuclear power plant applications:

- Inclusion of the effect of disk shape and disk aspect ratio (defined as disk thickness/disk diameter) on torque;
- Improved prediction of the effect of flow disturbance caused by an upstream elbow on the torque requirements, with proper accounting for the elbow configuration, its distance from the valve, and the direction of disk rotation;
- Verification of the torque scaling equations used to predict performance of large valves based on tests performed on small scale models; and
- Validation of the model against test data obtained under rigorous quality assurance programs (e.g., satisfying 10CFR50 Appendix B requirements) with documented measurement uncertainties for the key parameters.

In developing the EPRI butterfly valve model, all of these areas of needed improvement were systematically addressed by appropriate theoretical and experimental development.

**TORQUE PREDICTION MODEL**

The objective of the butterfly valve model is to determine stem torque from two standpoints:

- **Required actuation torque:** This is the maximum torque required to operate (open or close) the valve through its entire stroke, including total seating/unseating torque and total dynamic torque.

- **Maximum transmitted torque:** This is the maximum torque that can occur in the valve stem. The capability of the weak link in the valve (e.g., stem, disk-to-stem connection) and the actuator torque rating should both exceed the maximum transmitted torque.

The model provides predictions of both of these values.

**Scope**

The scope of the butterfly valve model includes the following valve design features, installation details, and operating conditions:

- **Disk Designs:** Symmetric disk and nonsymmetric disk with single offset designs shown schematically in Figure 1. Geometric variations in the disk shapes prevalent in nuclear power plants are shown in Figure 2. The butterfly valve model has been validated against test data for all disk shape variations in Figure 2, except Disk Shape 3. This disk shape constitutes a very small fraction of the total butterfly valve population in the nuclear power plants.

- **Operating Conditions:** The model is applicable to all operating conditions including a postulated pipe break immediately downstream of the valve.

- **Seat Designs:** Interference type and pressure energized seat designs (see Reference 1 for different types of seat designs).

- **Valve Stroke:** Both opening and closing stroke directions. The model is applicable for full or partial strokes for incompressible flow; full stroke analysis must be performed for compressible flow.

- **Flow Direction:** For nonsymmetric valves, both shaft upstream and shaft downstream flow directions (Figure 3).

- **Flow Condition:** Fully turbulent flow conditions for incompressible flow (both choked and unchoked) and compressible flow (choked).
• Upstream Flow Disturbances: The model accounts for the effect of an upstream 90-degree elbow on the hydrodynamic torque.

• Valve Condition and Behavior: The model assumes that the valve components have been properly maintained and are in good working order. It does not take into account anomalous or unpredictable behavior due to damaged or degraded seats, bearings, and packings.

**Stem Torque Prediction**

The model calculates two types of stem torque required to operate the valve:

- Total seating/unseating torque \((T_{TS})\)
  \[
  T_{TS} = T_b + T_p + T_s + T_h
  \]
  \[\text{(1)}\]

- Total dynamic torque \((T_{TD})\)
  \[
  T_{TD} = T_b + T_p + T_{hub} \pm T_{hyd}
  \]
  \[\text{(2)}\]

  Total dynamic torque applies throughout the stroke \((5^\circ < \alpha \leq 90^\circ)\) and is a function of disk position.

The torque required to actuate the valve is the larger of the total seating/unseating torque and the total dynamic torque.

**Sign Convention for Torque:** A torque that is applied to the stem by the actuator to cause disk rotation in either the opening or closing direction is a positive (+) torque. A torque that must be applied to the stem to restrain disk rotation in either the opening or closing direction is a negative (-) torque.

**Torque Components**

A brief description of the individual torque components in equations 1 and 2, including the method to calculate each one, is provided below.

**Bearing Torque** \((T_b)\): This is the torque created by the friction force between the stem and the bearings, and is calculated by Equation 3.

\[
T_b = \mu_b \times \pi \times d_{disk}^2 \times d_s \times \frac{\Delta P_v}{96} \text{ ft-lb} \]
  \[\text{(3)}\]

For bronze bearings in clean systems, the model recommends a value of \(\mu_b = 0.25\). The value of \(\mu_b\) is, however, a user input and lower values of \(\mu_b\) (e.g., for Teflon fabric bearings used in clean systems) can be used if test data are available to justify the assumption. For bronze bearings in dirty systems, and non-bronze metal bearing combinations, e.g., 17-4 PH stainless steel against hardened austenitic or martensitic stainless steel, \(\mu_b\) values can be higher than 0.25. The model recommends use of \(\mu_b = 0.6\) in such cases where test data are not available.

**Packing Torque** \((T_p)\): This is the torque created by the friction between the stem and the packing. Packing torque can vary significantly depending upon packing design, material, and the packing gland preload or torque; therefore, the user of the model is required to supply a value for packing torque. The Application Guide (Eldiway and Kalsi, 1993) provides guidance for estimating packing torque for butterfly valves. In situ tests with no flow and no \(\Delta P\) (static tests) can be used to accurately determine packing torque, or to confirm that packing torque is within design values.

For symmetric disk design butterfly valves, the hub remains in contact with the elastomer liner in the body throughout the stroke to provide a seal around the stem. This creates a frictional torque component called the hub seal torque, \(T_{hub}\), which remains nearly constant throughout the stroke, in a manner similar to the packing torque. The user-input value of packing torque must include the contribution from the hub seal.

**Seat Torque** \((T_s)\): For interference type seats, seat torque is calculated by Equation 4.

\[
T_s = \frac{1}{12} \times A \times d_{disk}^2 \text{ ft-lb}
\]
  \[\text{(4)}\]

where \(A\) is a constant which depends on fluid medium. The model recommends bounding values for the constant \(A\) for undamaged seats maintained in accordance with manufacturer's recommendations for incompressible flow media (wet service) and for compressible flow media (dry service) applications. The model also accepts a user-supplied value of seating torque which might be based on in situ test data or manufacturer's data.
Hydrostatic Torque ($T_h$): This is the torque caused by the static pressure difference across the valve created when there is fluid on only one side of the disk. This torque component is calculated by Equation 5.

$$T_h = \frac{\pi \rho}{64} \times \left( \frac{d_{disk}}{12} \right)^4 \times \sin \phi \quad \text{ft-lb} \quad (5)$$

This torque component is significant only for large valves (typically 30" and larger) used in incompressible flow applications with stem in non-vertical orientation.

Hydrodynamic Torque ($T_{hyd}$): This is the torque created by the hydrodynamic force imposed on the disk by the fluid flow, and is present only at disk positions other than fully closed. Hydrodynamic torque is calculated by equations 6 and 7.

- Incompressible Flow

$$T_{hyd} = \frac{1}{12} \times C_t \times d_{disk}^3 \times \Delta P_v \quad (6)$$

where $\Delta P_v$ = smaller of actual pressure drop across valve and $FL^2 (P_1 - 0.96 P_v)$.

- Choked Compressible Flow

$$T_{hyd} = \frac{1}{12} \times C_t \times d_{disk}^3 \times Y^2 \times (xT \times P_1) \quad (7)$$

In these equations, the nondimensional hydrodynamic torque coefficient, $C_t$, is a function of disk position, and depends upon disk geometry. Unlike the frictional torque components which act in a direction to oppose disk motion, hydrodynamic torque always acts in the same direction for a given disk angle. Hydrodynamic torque always tends to close the valve (self-closing), except in the following cases where hydrodynamic torque tends to open the valve:

- For offset disk designs with shaft downstream in choked compressible flow, hydrodynamic torque is self-opening throughout the stroke.

- For offset disk designs with shaft downstream in incompressible flow, the hydrodynamic torque becomes self-opening at disk positions near full open.

Dependence of Torque Coefficient ($C_t$) on Disk Design. The torque coefficient, which is a function of disk position ($\alpha$), depends upon the disk design, disk aspect ratio, and, for nonsymmetric valves, the flow direction (shaft upstream or shaft downstream). To make torque predictions, the model requires the torque coefficient, $C_t$, and the related "consistent" set of flow coefficients: $C_V$ or $K_v$, $F_L$ (for incompressible flow), or $xT$ (for choked compressible flow) as a function of disk position for a given disk design. "Consistent" means that the torque coefficient, $C_t$, and the flow coefficient, $C_V$, are at the same disk position for a particular disk design. Using inconsistent values of $C_t$ and $C_V$ (for example, from different sources) can result in significant errors in the calculated hydrodynamic torque. The flow coefficients are used by a companion computer program (System Flow Model, not discussed in this paper) to calculate the value of flow rate and $\Delta P$ vs. disk position which are used by the butterfly model to calculate torque.

The butterfly valve model provides a consistent set of default torque coefficients and flow coefficients (Table 1) for the symmetric and single offset disk shapes commonly used in nuclear service. The model also accounts for the effect of disk thickness on the hydrodynamic torque and generates a consistent set of $C_t$ and $K_v$ values for different disk aspect ratios.

Effect of Upstream Elbow ($C_{up}$): The butterfly valve model includes a factor, $C_{up}$, to account for the effect of an upstream elbow on hydrodynamic torque. Velocity skew generated by the elbow tends to increase or decrease the hydrodynamic torque. The amount of torque created by this velocity skew depends on the proximity of the elbow to the valve and the orientation of the elbow relative to the axis of the disk stem (see Figure 4). The equations used to calculate the effect of the elbow on hydrodynamic torque are as follows:

$$T'_{hyd} = C_{up} \times T_{hyd} \quad (8)$$

$C_{up}$ is the larger of 1.1 or the value calculated by the following equation:

$$C_{up} = 1 + (C_{up,max} - 1) \left( \frac{\alpha}{90} - \frac{n}{8} \right), \quad \frac{\alpha}{90} > \frac{n}{8} \quad (9)$$

Note: $C_{up} = 1$ when $n \geq 8$
This equation accounts for the fact that the effect of the elbow is most significant at close proximity and at high flow rates.

C_{up,max} depends on flow conditions (compressible or incompressible), orientation of elbow relative to axis of disk stem, and orientation of stem relative to flow direction (upstream or downstream).

MODEL VALIDATION

Test Data

The butterfly valve model was validated by comparing stem torque predicted by the model to that obtained during flow testing of valves. Table 2 shows the matrix of test data used for model validation. The matrix includes several hundred strokes of test data on 15 valves from different flow loop test facilities and in situ tests performed by utilities. Note that test valves 6 through 11 in Table 2 were specifically designed to supplement the data from other sources by providing a systematic variation in disk shapes, disk aspect ratios, and the effect of upstream elbows in incompressible flow. Using this approach, the model could be validated against a range of disk shapes representative of those most prevalent in nuclear service. Figure 5 shows the disk geometries of test valves 6 through 11.

For all tests in incompressible flow (water), both opening and closing stroke data were obtained. For compressible flow tests obtained from the NRC/INEL containment isolation valve test program (Watkins et al., 1986), only closing stroke data were available.

The test matrix used for validation permitted the butterfly valve model to be validated for the following range of design variations and conditions:

- Symmetric and single offset disk designs,
- Common disk shapes used in nuclear power plants,
- Disk aspect ratios from 0.15 to 0.47,
- Valve sizes from 6" to 42",
- Both flow directions for nonsymmetric disk valves,
- Opening and closing strokes,
- A range of flow rates,
- A range of valve differential pressures,
- Geometrically similar disks of 6" and 42" size,
- Upstream elbows with different proximities and orientations,
- Compressible and incompressible flow media,
- Continuous flow and pipe rupture immediately downstream of the valve.

The validation matrix included comparisons of model predictions against test data for dynamic torque as well as for the seating/unseating torques. The following section presents a summary of the technical approach and comparison of dynamic torque results which demonstrate key features of the model that overcome the shortcomings identified in the Introduction.

Technical Approach

The validation of the butterfly valve model included evaluation of the dynamic torque as well as the seating/unseating torque predictions. Comparisons of total dynamic torque predictions against the test data were made using one or more of the following four approaches, depending upon the details of the data available:

1. **Forward approach** in which the total dynamic torque from test results is compared to model predictions. The predictions are based on model torque coefficients, flow coefficients, and bearing coefficient of friction. Torque values are compared at the same value of valve flow resistance coefficient (Kv) for the model and the test valve, rather than at the same disk angle. This approach requires that the test data include ΔP and flow rate information (to determine Kv), as well as torque data, throughout the valve stroke.

2. **Inverse approach** in which data from the opening and closing strokes are used to inversely calculate hydrodynamic and bearing friction torque components. These components are used to determine hydrodynamic torque coefficient and the bearing friction coefficient, which are then compared to the coefficients in the model. This approach requires data for ΔP, flow rate,
and torque as a function of disk position throughout the disk stroke in both opening and closing directions. This approach was used because it provides a validation of individual components in the dynamic torque and it provides a more direct indication of the amount of conservatism in the key model features.

3. **Equivalent resistance approach** in which total dynamic torque predictions are made using an "equivalent resistance" model of the piping system determined from the maximum shut-off head in the fully closed position and maximum flow rate in the fully open position. This approach is somewhat approximate, but is used when ΔP and/or flow rate data are not available throughout the disk stroke. The procedure is described in *The Application Guide for Motor-Operated Butterfly Valves in Nuclear Power Plants* (Elidiwany and Kalsi, 1993).

4. **Normalization of upstream pressure approach** in which the test data were normalized to the nominal upstream absolute static pressure, P₁. This approach is used with compressible flow blowdown test results, for which the butterfly valve model dynamic torque predictions are calculated at the nominal value of P₁.

The packing torque component used in the predictions was based on measured data from opening and closing test strokes with no differential pressure. For symmetric disk valves the packing torque determined from the data includes the hub seal friction. The bearing friction coefficient used was selected using model guidelines.

**TYPICAL EXAMPLES OF COMPARISONS**

Detailed model validation included numerous comparisons to data for various test strokes, ΔPs, flow rates, flow directions, etc. obtained from different test facilities. In all cases, the model appropriately bounded the test data. This section presents a few typical examples of comparisons and highlights of the results pertaining to the following aspects of the model:

- Total dynamic torque validation
- Hydrodynamic torque component validation
- Upstream elbow effect validation
- Scaling validation

**Total Dynamic Torque Validation**

The total dynamic torque predicted by the model was validated by comparing the predicted torques with measured torques for eight valves (1 through 5 and 12 through 14). Both incompressible and compressible flow example comparisons are presented below.

**Valve No. 1 (Incompressible Flow)**

Valve No. 1 is a 6-inch, 150-pound butterfly valve manufactured by Henry Pratt. This valve has a symmetric disk of 0.31 aspect ratio. Tests were performed at three differential pressures: 50, 100, and 150 psi. Maximum flow velocity with the valve in the fully open position was 15 ft/s. Total dynamic torque predictions for this valve were performed using the forward approach for both the opening and closing directions, and the results are shown in figures 6 and 7.

Figure 6 shows that the model bounds the test data for all three differential pressures in the opening stroke direction. The torque results are plotted versus the butterfly valve model disk angle in degrees. The total dynamic torque comparisons are valid only outside the seating/unseating zone, which typically covers up to 10 degrees of disk opening. As described in the sign convention, the positive torque sign indicates that the actuator was required to supply torque to operate the valve.

Figure 7 shows closing stroke comparisons for three differential pressures. For the closing stroke, the model predicts the required total dynamic torque which must be supplied by the actuator. This is conservatively obtained by not taking credit for the self-closing hydrodynamic torque component. The model predictions for the required torque bounds the test data for all three ΔPs. A comparison of the model torque signature prediction, which does take into account the self-closing hydrodynamic torque, is also shown for the 150 psi DP case only. The torque signature prediction by the model closely matches the test results. It should be noted that torque signature prediction for the closing stroke
is provided for comparison and interpretation of test data only; it is not the required actuator torque.

As predicted by the model and found by testing, the required torque in the opening stroke direction is higher than the required torque in the closing stroke direction. This is the case with all valve applications that exhibit self-closing hydrodynamic torque.

Valve No. 2 (Incompressible Flow)
Valve No. 2 is a 6-inch, 150-pound butterfly valve manufactured by Henry Pratt. This valve has a nonsymmetric disk (single offset) of 0.47 aspect ratio. Tests were performed in both shaft upstream and shaft downstream orientations. Figures 8 and 9 show the results for the opening and closing strokes for the shaft upstream flow direction only (the higher torque direction).

The model predictions bound the test data for both the 50 and 100 psi differential pressure test conditions in the opening and closing directions. Similar comparisons were found for the shaft downstream condition.

Valve No. 3 (Incompressible Flow)
Valve No. 3 is a 42-inch, Class 150 butterfly valve manufactured by the Posi-Seal Division of Fisher Controls. This valve has a nonsymmetric disk (single offset) of 0.17 aspect ratio and a conical backface shape similar to Valve Assembly No. 6 in Figure 5. Tests were performed at the Utah State University Water Research Laboratory, using ambient water as the flow medium, in both shaft upstream and shaft downstream orientations with respect to flow. The maximum differential pressure used in these tests was 14 psi, and maximum flow velocity with the valve in the fully open position was 12 fps (55,000 gpm). The torque predictions were performed using the equivalent resistance approach, and total dynamic torque comparisons for the shaft upstream and shaft downstream orientation are shown in figures 10 and 11. The comparisons show that the model bounds the test results for both orientations with good margin.

It is noted that, for this low aspect ratio nonsymmetric valve with a conical backface, the model is very conservative, especially in the shaft upstream orientation. However, since the model is intended to cover the typical variations in disk shapes offered by other manufacturers, a reduction in the model conservatism is not justified.

Valve No. 12 (Compressible Flow)
Validation against compressible flow data was performed using the normalization of upstream pressure approach.

Data for validating the butterfly valve model in compressible flow were obtained from NUREG/CR-4648 (Watkins et al, 1986). Supplemental information was provided by INEL in the form of hard copy plots of torque signatures, pressures, temperatures, disk positions, and additional details pertaining to valve designs and the test matrix. The objective of the NRC/INEL test program was to assess the ability of the valve to close under design basis accident conditions in containment isolation applications; therefore, the digital data provided by INEL (Watkins et al, 1986) are for closing strokes only. The supplemental information provided some data pertaining to the opening stroke; however, during the opening stroke, the upstream pressure decayed rapidly throughout the stroke. Opening stroke data were used to estimate bearing coefficients of friction for the test valves. The bearing coefficient was found to be approximately 0.15, which is bounded by the 0.25 value used in the model.

The objective of the compressible flow model is to predict the peak total dynamic torque that occurs anywhere in the stroke. It assumes that the maximum upstream pressure held constant throughout the stroke. The small variations in the nominal upstream target pressure for each closing stroke in the INEL test matrix were accounted for by normalizing the data to a constant upstream pressure. Closing stroke tests were performed in both the shaft upstream and shaft downstream orientations. Tests were performed using nitrogen as the flow medium with maximum differential pressures maintained at nominal values ranging from 5 psi to 60 psi.

Note: The sign convention used for all compressible flow comparisons has been kept consistent with the sign convention adopted by INEL, which is opposite to that of the model and that employed for the incompressible data comparisons. In compressible comparisons, a negative torque indicates that the actuator has
to supply the torque to close the valve, and a positive torque magnitude indicates that the actuator is restraining the torque imposed on the shaft by the hydrodynamic forces.

Valve No. 12 is a 24-inch, 150-pound butterfly valve manufactured by Henry Pratt. This valve has a nonsymmetric (single offset) disk of 0.26 aspect ratio and a shape similar to Valve Assembly No. 4 in Figure 5.

Figure 12 shows the total dynamic torque predictions by the model against the test results under 60 psi differential pressure with the shaft in the upstream orientation. The total dynamic torque signature prediction envelope is defined by two values of bearing coefficient of friction: 0.25 and 0.0. The figure also shows the model prediction for the total required torque to actuate the valve. It can be seen that the maximum transmitted total dynamic torque predicted by the model bounds the test data. This maximum transmitted torque is recommended by the model to be used for evaluating the structural integrity of the MOV. The model prediction for the required actuation torque is only 700 ft-lb, which bounds the zero ft-lb found by actual testing.

Figure 13 shows comparisons of the total dynamic torque predictions by the model against the test results at 60 psi in the shaft downstream orientation. In this case the torque is negative, which means that the actuator has to supply the torque to close the valve. The peak total dynamic torque predicted by the model for 60 psi ΔP test condition bounds the test data. For this shaft orientation, this is also the maximum transmitted torque.

In the shaft downstream orientation, Valve No. 12 was also tested under three additional nominal upstream pressure conditions: 15, 30, and 45 psig (30, 45, and 60 psia). The peak dynamic torque comparisons of the model predictions against the test results for all of the ΔP conditions are summarized in Figure 14. It can be seen that the model bounds the test results for all pressure conditions.

Hydrodynamic Torque Component Validation

The hydrodynamic torque coefficients used in the model were validated by comparing the model coefficients with coefficients extracted from test data for nine valves (1 through 3 and 6 through 11) using the inverse approach. Only incompressible flow data were used for this validation.

Figure 15 shows typical comparisons of the torque coefficients used in the model for the symmetric disk designs with aspect ratios of 0.15, 0.25, and 0.31 to those determined from test data. The comparisons show that the model bounds test data for all three aspect ratios. Similar comparisons between model predictions and test data were made for the single offset disk designs with aspect ratios of 0.15, 0.17, 0.25, 0.35, and 0.47. The model predictions for all aspect ratios tested were found to bound the test data in both the shaft upstream and the shaft downstream orientations.

Scaling Validation

The nondimensional hydrodynamic torque coefficients, C_t, were extracted from test data for the 42-inch Posi-Seal valve (Valve No. 3) and its precisely scaled 6-inch model (Valve No. 11) using the inverse approach. The comparison of results for the shaft upstream and shaft downstream orientations for the 6-inch and the 42-inch valves is shown in Figures 16 and 17. The results show that the C_t for the 6-inch precisely scaled model is in excellent agreement with C_t obtained for the 42-inch valve in both flow directions. From this comparison it can be concluded that C_t is independent of the valve size and that the hydrodynamic torque component is proportional to the cube of the disk diameter as stated in equations 6 and 7. Furthermore, the scaling validation results provide the basis for applying the butterfly valve model to all valve sizes found in nuclear power plant applications.

Upstream Elbow Effect Validation

The upstream elbow model was validated by performing comparisons against a comprehensive matrix of test data for incompressible and compressible flow.

In the incompressible flow test matrix, the valve type, elbow orientation, flow direction, and elbow distances were systematically varied to provide 27 unique upstream elbow test configura-
tions. The matrix included symmetric and single offset disk designs of 0.25 aspect ratio tested in each of the elbow orientations shown in Figure 4 with three different spacings (0, 3, and 7 pipe diameters) between the elbow and the test valve. For each test configuration, the elbow effect was evaluated by opening and closing the valve under the following four flow conditions of maximum ΔP in the closed position and maximum flow velocity in the fully open position: 30, 60, and 90 psi with 15 ft/sec, and 90 psi with 30 ft/sec.

Figure 18 shows typical hydrodynamic torque results for the symmetric disk valve tested in elbow configuration 1(A) identified in Figure 4 with distances of 0, 3, and 7 pipe diameters between the elbow and the valve. For comparison, the baseline test results (with 20 diameters of straight pipe upstream of the valve) are also shown in this figure. The peak torque in these tests occurred in the 55- to 60-degree range of disk position, and the increase in peak torque due to the elbow effect was less than 10 percent. However, it is important to note that the actual location of the peak torque can vary, and the actual increase in torque caused by the presence of an elbow can be significantly higher for lower piping resistance systems which tend to shift the peak hydrodynamic torque towards the fully open disk position.

To account for this effect and extend the applicability of the elbow test results to other piping systems regardless of their piping resistance, the torque results were reduced to nondimensional upstream disturbance effect factors, \( C_{\text{Up}} \), defined by the torque ratio (\( T_{\text{Hyd}} / T_{\text{Hyd}} \)) at each disk position. Typical torque ratio plots for the test data and model predictions for the symmetric disk valve in Elbow Configuration 1(A) are shown in Figure 19. It can be seen that the elbow has the most significant influence at zero pipe diameter distance and full disk opening angle. The torque ratio plot also shows that the model bounds the test data for all disk openings except in the vicinity of fully open and fully closed positions, where the baseline torque approaches zero magnitude and the ratios become very high (theoretically infinite). The larger values of torque ratios are meaningless from a practical torque requirement standpoint since the peak hydrodynamic torque occurs in the 20- to 75-degree range of disk opening angle.

Figures 20 and 21 show that the elbow effect results for the two other possible elbow configurations (2A and 3 identified in Figure 4) for the symmetric disk valve in incompressible flow are also bounded by the model predictions. Elbow Configuration 2A creates a skew in the velocity profile which has a favorable effect (i.e., it tends to reduce the hydrodynamic torque imposed on the disk) at the zero diameter spacing; however, the model does not take credit for the reduction because this effect is very sensitive to the exact spacing between the elbow and valve and it diminishes very rapidly. Figure 22 shows the results for Elbow Configuration 3 in which the valve stem is in the plane of the elbow bend. In this configuration, the velocity skew caused by the elbow is symmetrical about the stem axis; therefore, the flow disturbance caused by the upstream elbow has the least effect on the hydrodynamic torque.

Typical comparisons from the compressible flow based on data from the NRC/INEL tests on three different valves with Elbow Configuration 1B at zero pipe diameter upstream of the valve are shown in Figure 22. The peak dynamic torque for compressible, choked flow occurs at disk openings of around 70 to 75 degrees. As shown in Figure 22, the compressible elbow effect model comfortably bounds the test results for these valves in the range of disk angles corresponding to the peak torque locations.

It should be noted that the \( C_{\text{up}, \text{max}} \) factors provided in the model are different for different elbow configurations and incompressible or compressible flow to suitably bound the performance for different applications.

Observations Regarding Manufacturers’ Data

Comparisons of actual test results against the predictions based on available manufacturers’ data showed that the total dynamic torque predictions by some of the manufacturers were unconservative, whereas other manufacturers’ predictions were extremely conservative. In general, it was found that some manufacturers’ predictions do not take into account the variations in disk aspect ratios and disk shapes. From a survey of manufacturers’ data, it was found that some manufacturers have typically performed a limited number of tests on one or two
valves and extended the results to valves of other sizes and pressure ratings that have disks of different aspect ratios and shapes. The EPRI butterfly valve model overcomes this deficiency and provides the appropriate flow and torque coefficients for a given disk shape and aspect ratio.

CONCLUSIONS
The research program described in this paper has led to the development of an improved and validated butterfly valve torque prediction methodology. For dynamic torque predictions, the model provides the appropriate torque and flow coefficients based on the disk design (symmetric or single offset), disk aspect ratio, flow direction, and fluid media (compressible or incompressible). Model predictions for the total dynamic torque have been compared against test data for a number of valves ranging in size from 6 to 42 inches having symmetric and single offset disk designs and aspect ratios ranging from 0.15 to 0.47 that were tested in incompressible and compressible flow applications. The model was found to bound the test results for the required actuation torque and the maximum transmitted torque in all cases. The elbow model provides the appropriate upstream elbow effect factor, \( C_{up} \), which depends upon elbow orientation, elbow distance, disk opening angle, fluid media (incompressible or compressible), and flow direction. The elbow model predictions were compared against test data from a large matrix of tests in incompressible and compressible flow media. The elbow model predictions were found to bound test results in all cases.

Evaluation of test data against previously reported generalized disk model (e.g., the Application Guide for Butterfly Valves, Eldiwanay and Kalsi, 1993) shows that the earlier model did not bound some test results, especially for higher disk aspect ratios. Furthermore, the earlier elbow model did not bound the test results in some elbow configurations and was overly conservative for others. The current research eliminates these shortcomings of the earlier generalized disk model and limitations of some manufacturers' torque prediction methods and provides a validated model for butterfly valve torque prediction.

ACKNOWLEDGEMENTS
The authors are grateful for the guidance and technical reviews provided throughout the execution of this research program by all members of the MOV Technical Advisory Group chaired by Mike Eidson and Bob Elfstrom; and to John Hosler, the EPRI program manager.

REFERENCES


NOMENCLATURE

\( A \) Pressure-independent seat torque coefficient, in-lb/in

\( C_t \) Hydrodynamic torque coefficient, dimensionless

\( C_{up} \) Upstream disturbance factor, dimensionless

\( C_{up,\text{max}} \) Maximum value of upstream disturbance factor, dimensionless

\( C_{v,\text{max}} \) Valve flow coefficient in the fully open position, gpm/\sqrt{psi}

\( d_{\text{disk}} \) Disk diameter, inch

\( d_i \) Valve inlet diameter, inch

\( d_s \) Stem diameter, inch

\( F_L \) Valve liquid pressure recovery factor, dimensionless

\( K_{v,\text{min}} \) Valve head loss coefficient in the fully open position, dimensionless
n  Length of straight pipe between an upstream elbow and valve inlet expressed in pipe diameters, dimensionless

$\Delta P_{\text{choke}}$  Pressure differential across valve (valve/fitting assembly) at onset of choking, psi

$\Delta P_{\text{max}}$  Maximum pressure differential at shutoff, psi

$\Delta P_V$  Pressure differential across the valve or valve-fitting assembly, psi

$R_a$  Disk aspect ratio (disk thickness/disk diameter), dimensionless

$T_b$  Bearing torque, ft-lb

$T_h$  Hydrostatic torque, ft-lb

$T_{\text{hub}}$  Hub seal friction torque, ft-lb. This component is present only in rubber seated symmetric disk butterfly valves

$T_{\text{hyd}}$  Hydrodynamic torque (without an upstream flow disturbance), ft-lb

$T_{\text{hyd}'}$  Hydrodynamic torque (with an upstream flow disturbance), ft-lb

$T_p$  Packing torque, ft-lb

$T_s$  Seat torque, ft-lb

$T_{\text{TD}}$  Total dynamic torque at disk angle, $\alpha$, ft-lb

$T_{\text{TD, max}}$  Maximum total dynamic torque, ft-lb

$T_{\text{TS}}$  Total seating/unseating torque, ft-lb

$T_{\text{WL}}$  Maximum transmitted torque for weak link analysis, ft-lb

$x_T$  Valve pressure drop ratio factor, dimensionless

$\alpha$  Disk opening angle, degree

$\mu_b$  Bearing coefficient of friction, dimensionless

$p$  Fluid density at valve inlet conditions, lb/ft$^3$

$\phi$  Stem angle measured from vertical, degree
<table>
<thead>
<tr>
<th>Valve Resistance Coefficient, $K_v$ (min = 0.53)</th>
<th>Disk Opening Angle, $\alpha$ (deg.)</th>
<th>x_T</th>
<th>$\text{FL}^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 (Closed)</td>
<td>928698989</td>
<td>0.000</td>
<td>0.000</td>
</tr>
<tr>
<td>1</td>
<td>115272617</td>
<td>0.009</td>
<td>0.000</td>
</tr>
<tr>
<td>2</td>
<td>7186703</td>
<td>0.009</td>
<td>0.000</td>
</tr>
<tr>
<td>3</td>
<td>1416391</td>
<td>0.009</td>
<td>0.000</td>
</tr>
<tr>
<td>4</td>
<td>446391</td>
<td>0.009</td>
<td>0.000</td>
</tr>
<tr>
<td>5</td>
<td>182717</td>
<td>0.009</td>
<td>0.000</td>
</tr>
<tr>
<td>6</td>
<td>52052</td>
<td>0.009</td>
<td>0.000</td>
</tr>
<tr>
<td>7</td>
<td>15003</td>
<td>0.009</td>
<td>0.000</td>
</tr>
<tr>
<td>8</td>
<td>44013</td>
<td>0.009</td>
<td>0.000</td>
</tr>
<tr>
<td>9</td>
<td>13271</td>
<td>0.009</td>
<td>0.000</td>
</tr>
</tbody>
</table>

Table 1: Butterfly Valve Model Coefficients for Hydrodynamic Torque Calculations

Presented at the Joint NEA/IAEA Meeting
<table>
<thead>
<tr>
<th>Valve No.</th>
<th>Valve Description</th>
<th>Disk Design (Note 1)</th>
<th>Media</th>
<th>Data Source</th>
<th>Flow Direction</th>
<th>ΔP, psi</th>
<th>Max Flow, gpm or l/sec</th>
<th>Max V ft/sec</th>
<th>Test Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>6&quot; Henry Pratt (EPRI #54)</td>
<td>Sym</td>
<td>Water</td>
<td>Wyle</td>
<td>N.A.</td>
<td>50, 100, 150</td>
<td>1,500</td>
<td>15</td>
<td>Yes</td>
</tr>
<tr>
<td>2</td>
<td>6&quot; Henry Pratt (EPRI #55)</td>
<td>SO</td>
<td>Water</td>
<td>Wyle</td>
<td>Both</td>
<td>50, 100, 150</td>
<td>1,500</td>
<td>15</td>
<td>Yes</td>
</tr>
<tr>
<td>3</td>
<td>42&quot; Posi-Seal</td>
<td>SO</td>
<td>Water</td>
<td>Duke</td>
<td>Both</td>
<td>14</td>
<td>46,000</td>
<td>11</td>
<td>Yes</td>
</tr>
<tr>
<td>4</td>
<td>18&quot; Fisher (2-HV-4572)</td>
<td>Sym</td>
<td>Water</td>
<td>TU</td>
<td>N.A.</td>
<td>130</td>
<td>8,000</td>
<td>11</td>
<td>Yes</td>
</tr>
<tr>
<td>5</td>
<td>24&quot; Fisher (2-HV-4512 and 1-HV-4286)</td>
<td>SO</td>
<td>Water</td>
<td>TU</td>
<td>Upstream</td>
<td>85</td>
<td>15,000</td>
<td>12</td>
<td>Yes</td>
</tr>
<tr>
<td>6</td>
<td>6&quot; model, t/d = 0.15</td>
<td>Sym</td>
<td>Water</td>
<td>Kalsi</td>
<td>N.A.</td>
<td>90</td>
<td>2,700</td>
<td>30</td>
<td>No</td>
</tr>
<tr>
<td>7</td>
<td>6&quot; model, t/d = 0.25</td>
<td>Sym</td>
<td>Water</td>
<td>Kalsi</td>
<td>N.A.</td>
<td>90</td>
<td>2,700</td>
<td>30</td>
<td>No</td>
</tr>
<tr>
<td>8</td>
<td>6&quot; model, t/d = 0.15</td>
<td>SO</td>
<td>Water</td>
<td>Kalsi</td>
<td>Both</td>
<td>90</td>
<td>2,700</td>
<td>30</td>
<td>No</td>
</tr>
<tr>
<td>9</td>
<td>6&quot; model, t/d = 0.25</td>
<td>SO</td>
<td>Water</td>
<td>Kalsi</td>
<td>Both</td>
<td>90</td>
<td>2,700</td>
<td>30</td>
<td>No</td>
</tr>
<tr>
<td>10</td>
<td>6&quot; model, t/d = 0.35</td>
<td>SO</td>
<td>Water</td>
<td>Kalsi</td>
<td>Both</td>
<td>90</td>
<td>2,700</td>
<td>30</td>
<td>No</td>
</tr>
<tr>
<td>11</td>
<td>6&quot; model of 42&quot; Posi-Seal</td>
<td>SO</td>
<td>Water</td>
<td>Kalsi</td>
<td>Both</td>
<td>90</td>
<td>2,700</td>
<td>30</td>
<td>No</td>
</tr>
<tr>
<td>12</td>
<td>24&quot; Henry Pratt</td>
<td>SO</td>
<td>Nitrogen</td>
<td>INEL</td>
<td>Both</td>
<td>5 → 60</td>
<td>N/A</td>
<td>Choked</td>
<td>Yes</td>
</tr>
<tr>
<td>13</td>
<td>8&quot; Henry Pratt</td>
<td>SO</td>
<td>Nitrogen</td>
<td>INEL</td>
<td>Both</td>
<td>5 → 60</td>
<td>N/A</td>
<td>Choked</td>
<td>Yes</td>
</tr>
<tr>
<td>14</td>
<td>8&quot; Allis Chalmers</td>
<td>SO</td>
<td>Nitrogen</td>
<td>INEL</td>
<td>Both</td>
<td>5 → 60</td>
<td>N/A</td>
<td>Choked</td>
<td>Yes</td>
</tr>
<tr>
<td>15</td>
<td>10&quot; Henry Pratt</td>
<td>SO</td>
<td>Water</td>
<td>APS</td>
<td>Downstream</td>
<td>120</td>
<td>3,815</td>
<td>16</td>
<td>Yes</td>
</tr>
</tbody>
</table>

**Note 1.** Sym = symmetric disk  
SO = single offset disk

**Note 2.**
Figure 1  Symmetric and Single Offset Disk Butterfly Valves
Figure 3  Valve Orientation with Respect to Flow
Figure 4  Upstream Elbow Orientation for Symmetric and Nonsymmetric Valves
Figure 5. Disk Shapes Used in Matrix of Tests Performed at Kalas Engineering Flow Loop.
Figure 6  Model Predictions and Test Results for Opening Strokes for Valve No. 1 (6-Inch Pratt Symmetric Disk) in Incompressible Flow
Figure 7  Model Predictions and Test Results for Closing Strokes for Valve
No. 1 (6-Inch Pratt Symmetric Disk) in Incompressible Flow

Presented at the Joint NEA/IAEA Meeting 20

542
Figure 8  Model Predictions and Test Results for Shaft Upstream, Opening Strokes, for Valve No. 2 (6-Inch Pratt Single Offset Disk) in Incompressible Flow
6" Pratt, Single Offset Disk, Shaft Upstream
Aspect Ratio = 0.47, Closing Strokes

Figure 9  Model Predictions and Test Results for Shaft Upstream, Closing Strokes, for Valve No. 2 (6-Inch Pratt Single Offset Disk) in Incompressible Flow

Presented at the Joint NEA/IAEA Meeting
Figure 10  Model Predictions and Test Results for Shaft Upstream, Opening Strokes, for Valve No. 3 (42-Inch Posi-Seal Single Offset Disk) in Incompressible Flow

Figure 11  Model Predictions and Test Results for Shaft Downstream, Opening Strokes, for Valve No. 3 (42-Inch Posi-Seal Single Offset Disk) in Incompressible Flow
24" Pratt, Single Offset Disk, Shaft Upstream
Aspect Ratio = 0.26, Closing Stroke

Note: The actuator is required to supply the driving torque when torque values are negative per INEL torque sign convention.

Figure 12  Model Predictions and Test Results for Shaft Upstream, Closing Strokes, for Valve No. 12 (24-Inch Pratt Single Offset Disk) in Compressible Flow
Figure 13  Model Predictions and Test Results for Shaft Downstream, Closing Strokes, for Valve No. 12 (24-Inch Pratt Single Offset Disk) in Compressible Flow

Note: The actuator is required to supply the driving torque when torque values are negative per INEL torque sign convention.
24" Pratt, Single Offset Disk, Shaft Downstream
Aspect Ratio = 0.26, Closing Stroke

![Graph]

**Note:** The actuator is required to supply the driving torque when torque values are negative per INEL torque sign convention.

**Figure 14** Model Predictions and Test Results for Shaft Downstream, Closing Strokes, for Valve No. 12 (24-Inch Pratt Single Offset Disk) in Compressible Flow under Different ΔP Conditions

Presented at the Joint NEA/IAEA Meeting
Figure 15  Comparison of Model Torque Coefficients and Flow Coefficients against Test Data for Symmetric Disk Valves of Different Disk Aspect Ratios
Figure 16  Comparison of Test Results from 6-Inch Scaled Model Test and 42-Inch Valve with Shaft Upstream Orientation

Comparison of C_t versus K_v from the 42-inch valve Test, the 6-inch Model, and the Butterfly Valve Model (Shaft Downstream)

Figure 17  Comparison of Test Results from 6-Inch Scaled Model Test and 42-Inch Valve with Shaft Downstream Orientation
Figure 18  Hydrodynamic Torque Results for the Symmetric Disk Valve Tested in Elbow Configuration 1A at 0, 3, and 7 Pipe Diameters Upstream in Incompressible Flow
Figure 19  Torque Ratio Plots for Valve No. 7 Tested in Elbow Configuration 1A at 0, 3, and 7 Pipe Diameters Upstream in Incompressible Flow
Symmetric Disk
Aspect Ratio = 0.25, Elbow Configuration 2A

Figure 20  Torque Ratio Plots for Valve No. 7 Tested in Elbow Configuration 2A at 0, 3, and 7 Pipe Diameters Upstream in Incompressible Flow
Figure 21  Torque Ratio Plots for Valve No. 7 Tested in Elbow Configuration 3 at 0, 3, and 7 Pipe Diameters Upstream in Incompressible Flow
Figure 22  Torque Ratio Plots for Valve Nos. 12, 13, and 14 Tested in Elbow Configuration 1B at Zero Pipe Diameters Upstream in Compressible Flow
Mechanical Design Calculations for Stem Forces in Safety-Related Gate and Globe Valves with Regard to Proper Valve Function and to Load Effects on Valve Components
R. Kubosch, A. Scheuer / TÜV Rheinland e.V., Köln

Introduction

The aim of these guidelines introduced by us for calculating gate and globe valves is to ensure that a uniform procedure is adopted by the experts of the technical inspectorates (VdTÜV) and the Association for Reactor Safety (GRS) in the evaluation of gate and globe valves important for safety in German nuclear power plants. The guidelines are applicable to existing and new valves.

Various incidents in German as well as foreign nuclear power plants, in which shut-off valves did not close or open fully because the actuating force of the valve actuators was inadequate, and the recommendations of the GRS in memorandum No. 14/90 "Recent experience with actuating force reserves of shut-off valves - results of US large-scale tests" were the reason why greater attention had to be paid to ensuring adequate reserves with regard to the actuating forces to provide reliable operation when required as well as strength of the valve parts affected by the actuating force during valve movement.

These incidents in American nuclear power plants involving gate valves in locations where they affect plant safety induced the NRC (Nuclear Regulatory Commission) to conduct comprehensive investigations and tests in order to examine the causes in more detail.

The most important result of these NRC investigations was that at full differential pressure, i.e. with maximum forces acting upon the valve in a model accident situation, significantly higher actuating forces may occur than were taken into account in the calculation of the valve actuating torque requirements.

With regard to the applicability of the above-mentioned NRC results to the valves installed in German plants, in particular gate valves, the German Commission "Leitstelle - Kerntechnik" instructed the "Components and Materials" working group and its ad hoc "Shut-off Valves in Pressurised Water and Boiling Water Reactors" working party to check this situation. As the above-mentioned aspects have so far not been dealt with in the German KTA regulations, the drawing up of a calculation specification was to supply the required details. After a number of discussions both in the ad hoc working party and also with the plant operators a calculation concept, in which, in the main, factors which vary within given tolerances and have an influence on MAXIMUM/MINIMUM parameters were to be dealt with
and which also takes into account whether the respective calculation will determine either the required torque for actuating the valve or the strength of the components exposed to the actuating force, appeared appropriate to the ad hoc working party in view of recurring problems with regard to the available actuating forces on valves.

The valve spindle force which includes, in particular, the frictional resistance of the stuffing box in this case, and a value representing the actuating force tolerance, were regarded as important tolerated parameter factors to be taken into account. The frictional resistance of the valve disc (also important parameter factor) was taken as a basis for the calculation concept only as a mean value for lack of sufficient experimental data. Adequate allowance was made for tolerance variations of the above parameters by introducing a safety factor S. Furthermore, the effects of various actuator stop situations were taken into account for the determination of the actuating force and strengths of the parts affected by the actuating force during valve movement.

When applying this calculation specification it is assumed that basis requirements are met in the design of the valves (e.g. all edges of the gate and the seating ring/casing should be radiuses and the disc guide should be armoured and have narrow clearances.

A corresponding schedule is currently being prepared.

It is likewise assumed that the servicing and maintenance requirements are being met throughout the operating life of the valves and verified by performance tests in order to maintain the initial level of valve performance.

A simplified calculation method is given (Section 7) for verifying that valves already installed are within the new calculation requirements. An adequate actuation force, i.e. reliable operation, takes priority over the strength calculation in this case.
Contents

0. Introduction

1. Field of application

2. Determination of the required torque $M_{req}$, for valve opening and closing
   2.1 Gate valves with wedge-shaped and parallel discs
   2.2 Globe valves

3. Determination of the required release torque $M_{req, release}$ for opening jammed gate valves and globe valves on failure of the movement limit switches or the excess torque protection switch

4. Torque setting conditions for gate valves and globe valves
   4.1 Opening torque setting $M_{set, OPEN}$
   4.2 Closing torque setting $M_{set, CLOSE}$

5. Determination of calculation torques to suit various stop situations for the purpose of calculating the strength of valve parts affected by the actuating force

6. Proof of strength (permissible stress) of valve parts affected by the actuating force under operating conditions and at the various actuator stop situations

7. Simplified calculation method for valves already used in a plant

552
1. **Field of application**

These guidelines for calculating gate valves and globe valves apply to valves in locations where they directly affect the safe operation of the plant. The use of these calculations presupposes that the opening and closing movements of gate valves are limited by mechanical means, and the opening movement of globe valves by mechanical means and the closing movement by torque limitation.

The guidelines cover the following:

- calculation of the required torque settings to suit operating conditions,
- strength calculations for valve parts affected by the actuating force during valve movement,
- verification of casing integrity in the event of overload conditions on closing due to failure of torque limitation including verification of non-leakage from the valve even if the valve function is impaired,
- verification of stress conditions of the valve parts subjected to operating forces during normal operation and in the event of overload conditions on closing due to failure of torque limitation.

2. **Determination of the required torque \( M_{req} \) for valve opening and closing**

2.1 **Gate valves with wedge-shaped and parallel gates**

The required torque can be calculated employing the following formula:

\[
M_{req} = S \cdot F_s P \cdot \frac{d^2}{2} \cdot \tan(\rho + \alpha) \cdot \frac{f}{\mathrm{torque} \cdot \eta_{overall}}
\]

The spindle force \( F_s P \) is determined in the

- closing direction: \( F_s P = F_{open} + F_{gland} + \text{Friction} \)
- opening direction: \( F_s P = \text{Friction} + F_{gland} - F_{OPEN} \)

From this, the torque required can be calculated thus:
for closing: \[ M_{req. close} = S(F_{open} + F_{friction} + F_{gland}) \frac{d^2}{2} \cdot \tan(\rho + \alpha) \cdot \frac{f}{\text{inves.} \cdot \eta_{overall}} \]

for opening: \[ M_{req. open} = [S \cdot (F_{friction} + F_{gland}) - F_{open}] \frac{d^2}{2} \cdot \tan(\rho + \alpha) \cdot \frac{f}{\text{inves.} \cdot \eta_{overall}} \]

**Stuffing box frictional force \( F_{gland} \)**

\[ F_{gland} = F_B \cdot \frac{r_i}{r_a + r_i} \left[ \frac{-2 \mu_s \cdot K \cdot L}{1 - e} \frac{r_a - r_i}{r_a - r_i} \right] \]

for \( r_a = r_i \) the following simplified calculation can be made:

\[ F_{gland} = \frac{F_B}{2} \left[ \frac{-2 \mu_s \cdot K \cdot L}{1 - e} \frac{r_a - r_i}{r_a - r_i} \right] \]

- \( L \) = length of the gland packing
- \( r_a \) = outside radius of the packing
- \( r_i \) = inside radius of the packing
- \( K \) = factor for converting gland pressure to a frictional force (Table 1)
- \( \mu_s \) = coefficient of friction between packing and spindle (Table 1)
- \( F_B \) = maximum gland pressure taken from installation instructions

**Table:** Guide values for material-related characteristic values (apply only to one-piece packing)

| Type of packing  | Coefficient of friction between packing and spindle \( \mu_s \) 1) | Characteristic value 1) | K |
|------------------|---------------------------------------------------------------|-------------------------|
| Pure graphite    | 0.13                                                          | 0.60                    |
| PTFE silk        | 0.13                                                          | 0.58                    |

1) Deviating manufacturer's data, which have been proved by tests, are permissible
Remarks:
The above calculation of the stuffing box frictional force $F_{gland}$ applies to a full set of packing only.
For individual packing rings equations specific to the packing should be used for $F_{gland}$

\[
f = \text{correction factor for remote drives}
\]

\[
\eta_{\text{overall}} = \text{overall efficiency of gears and bearings}
\]

\[
\eta_{\text{trans.}} = \text{ratio of intermediate gears}
\]

**Disc force friction**

When closing:

\[
F_{\text{friction}} = \frac{\mu_p \cdot F_{PB} + 2F_F \cdot (\mu_p \cdot \cos \psi + \sin \psi)}{\cos \psi - \mu_p \sin \psi}
\]

When opening:

\[
F_{\text{friction}} = \frac{\mu_p \cdot F_{PB} + 2F_F \cdot (\mu_p \cdot \cos \psi - \sin \psi)}{\cos \psi + \mu_p \sin \psi}
\]

where

\[
F_{PB} = \text{Pressure force on the valve disc due to maximum differential pressure}
\]

\[
F_{PB} = \frac{\pi}{4} d_m^2 \cdot \Delta p
\]

\[
d_m = \text{mean disc seat diameter}
\]

\[
\Delta p = \text{differential pressure on closing in case of demand}
\]

\[
F_F = \text{spring force due to gate design if applicable, e.g. from cup springs or bellow}
\]

\[
\psi = \text{wedge disc angle (\psi = 0 for parallel discs)}
\]

\[
\mu_p = \text{coefficient of friction between valve disc and casing seat (value to be inserted}
\]

\[
\mu_p = 0.4 \text{ for material pairing stellite 6/6 or other confirmed values}
\]
Forces and direction of forces acting upon the valve gate are shown in the following picture:
Spindle raising force $F_{open}$

$$F_{open} = \frac{\pi}{4} \cdot d_{sp}^2 \cdot p$$

$d_{sp}$ = spindle diameter (in stuffing box)

$p$ = maximum operating pressure in normal operation (alternatively permissible overload pressure)

$S$ = safety factor

Remark:

For ratio $\frac{F_{friction}}{F_{sp}} \geq 0.8$ the safety factor $S = 1.5$

$S$ may be reduced by up to 10% by linear interpolation if the ratio $\frac{F_{friction}}{F_{sp}}$ is between 0.8 and 0.5

$d_2$ = mean spindle thread diameter

$\alpha$ = helix angle $\alpha = \frac{h}{d_2 \cdot \pi}$

$h$ = pitch of thread

$\rho$ = friction angle $\rho = \frac{\mu G \text{(max/min)}}{\cos \beta}$

$\beta$ = half flank angle

$\mu G$ = coefficient of friction between spindle thread and nut

The coefficient of friction varies between a maximum ($\mu_{\text{max}}$) and minimum ($\mu_{\text{min}}$) value and should be advised and proved by the material manufacturer.
$\mu G_{\text{max}}$ when calculating the required actuating torque
$\mu G_{\text{min}}$ when calculating the strength of the components affected by the actuating force

In accordance with test results so far available the coefficients of friction $\mu G_{\text{min}} \geq 0.1$ and $\mu G_{\text{max}} = 0.17$ apply to the following materials:

- spindle: Mat. No. 1.4057, 1.4542 and 1.4980
- spindle nut: Mat. No. 2.0550, 2.0966

These values are valid for standard trapezoidal threads with a diameter/pitch ratio of up to about 9, proper lubrication and with the spindle nut mounted on anti-friction bearings. The above-mentioned values do not embrace all available test results, because the safety factors used are regarded as adequate to cover values occurring only infrequently.

The $\mu G_{\text{max}}$ values so far available for spindle nut material 2.0550 are lower (generally $\mu G_{\text{max}} = 0.15$) than for material 2.0966; in particular under coolant loss conditions values exceeding $\mu G_{\text{max}} = 0.17$ may also occur for material 2.0966.

Valve manufacturer's data deviating from the above values and confirmed by tests can be used.

In the case of coolant loss conditions a nut thread lengths $> 2$ should be proved to be suitable functionally.

2.2 Globe valves

The required torque for valve actuation can be calculated thus:

$$M_{\text{reqy, close}} = S \cdot F_{sp} \cdot \frac{d^2}{2} \cdot \tan(\rho + \alpha) \cdot \frac{f}{\text{Irres} \cdot \eta_{\text{overall}}}$$

The spindle force $F_{sp}$ for opening/closing is determined as follows:

$$F_{sp} = F_P + F_{gland} \pm F_K$$

where
\[ F_p = \text{spindle raising force} \]
\[ F_k = \text{bellow force} \]
\[ F_{gland} = \text{gland packing frictional force} \]

The spindle raising force \( F_p \) is determined as follows:

a) **When closing**

\[ F_p = p_1 \cdot A_1 - p_2 \cdot A_1 + p_2 \cdot A_3 \]
(when the closed position is achieved)

\[ F_p = p \cdot A_3 \) (effective pressure during the stroke) \]

\[ p = \text{maximum operating pressure under working conditions} \]
(alternatively permissible overpressure)

b) **When opening**

\[ F_p = -p_1 \cdot A_1 + p_2 \cdot A_1 - p_2 \cdot A_3 \]
(when moving away from the closed position)

Pressure and direction of pressures acting upon the valve seat are shown below.
Fig. 2: Pressures and pressure geometries on the valve cone

\[ A_1 = \frac{\pi}{4} \cdot d_m^2 \]

\[ A_2 = A_1 - A_3 \]

\[ A_3 = \frac{\pi}{4} d_p^2 \quad \text{or for bellow} \quad A_3 = \frac{\pi}{4} d_B^2 \]

\[ P_1 \quad \text{pressure from under the valve seat} \]

\[ P_2 \quad \text{pressure from above the valve seat} \]

\[ d_{sp} \quad \text{spindle diameter through stuffing box} \]

\[ d_{FB} \quad \text{effective diameter of bellow in bellow-type valves} \]

\[ d_m \quad \text{mean valve seat diameter} \]

The bellow force \( F_k \) may be + or -, depending on the arrangement of the bellow.
Determination of the gland packing frictional force has already been dealt with in the section on gate valves (see formula).

The safety factor $S$ must be at least $1.2$. Design parameters which cannot be ascertained (e.g. friction due to seat cone guide) should be taken into account in a safety factor $S > 1.2$.

All other parameters of the formula have already been dealt with in the section on gate valves.

3. **Determination of required release torque $M_{\text{req.}}$ for opening jammed gate valves and globe valves on failure of movement limit switches or excess torque protection switch**

$$M_{\text{req.}} = F_s \cdot \frac{d_2}{2} \cdot \frac{\tan(\rho - \alpha)}{\tan(\rho + \alpha)} \cdot \frac{f}{\text{trans} \cdot \eta_{\text{overall}}}$$

$$\tan \beta = \frac{\mu g}{\cos \beta} \quad \text{with} \quad \mu_{\text{max}} = 0.18^*$$

The spindle force required for opening valve from a jammed position $F_{s \text{release}}$ is determined as follows:

$$F_{s \text{jam}} = \frac{2 \cdot M_{\text{jam}}}{d_2 \cdot \tan(\rho + \alpha)}$$

**Torque $M_{\text{jam}}$ is determined thus:**

$$M_{\text{jam}} = (M_{\text{req.}} + \Delta M) \cdot C_u \cdot \frac{\text{trans} \cdot \eta_{\text{overall}}}{f}$$

where

$$C_u = \text{factor for excess torque; should be determined from the valve actuator response time delay, the flywheel effect and the end position stiffness of the valve as given by the manufacturer.}$$

$$\Delta M = 0.1 \cdot (\text{max. adjustable torque})$$
ΔM takes into account the maximum permissible tolerance of ± 10 % of the maximum adjustable torque at which movement stops according to KTA 3504.

If confirmed values are available for different tolerances, they may be used.

\[ \tan \beta = \frac{\mu_0}{\cos \beta} \quad \text{with} \quad \mu_{0\min} = 0.10^* \]

- The limits of the coefficient of friction \( \mu_{\text{Gmax}} = 0.1 \) and \( \mu_{\text{Gmax}} = 0.18 \) refer to an individual spindle (not the total number of spindles). The difference between \( \mu_{\text{Gmin}} \) and \( \mu_{\text{Gmax}} \) should cover release from jammed also after prolonged stoppages.

4. Torque setting condition for gate valves and globe valves

4.1 Opening torque setting:

\[ \text{Mset OPEN} \geq \text{Mreq.} + \Delta M \]

\( \text{Mreq.} \) is the maximum value from:

\[ \text{max.} \ \{(\text{Mreq. OPEN}; \text{Mreq. rel.;}\ \text{Mreq. CLOSE}) \]

4.2 Closing torque setting:

\[ \text{Mset CLOSE} \geq \text{Mreq.} + \Delta M \]

end of opening stroke reached
(no jamming in uppermost position)

\[ \text{with Mreq.} = \text{Mreq. CLOSE} \]

or
Opening torques reached
opening movement stops

with Mreq. as maximum value from:

\[ \text{max. (Mreq. CLOSE: Mreq. rel.)} \]

Mreq. rel. = required release torque from the upper end position

In the case of valves incorporating a friction clutch, when actuated by by-passing the actuator overload protection, the torque for breaking away from the upper end position must not reach the magnitude of the torque at which the friction clutch starts slipping. This can be achieved by setting the break-away (release) torque well below the slipping torque of the clutch.

5. Determination of calculation torques of suit various actuator stop situations for the purpose of calculation of valve parts

For determining the size of valve parts affected by the actuating force, the torques for the various actuator stop situations must be taken into account. These torque ratings can be calculated as follows:

5.1 Stop situation

• End of stroke (Limit switch reached)

\[ M_{\text{calc}} = (M_{\text{set \ \text{class}}} + \Delta M) \cdot C_s \cdot \frac{\eta_{\text{trans}} \cdot \eta_{\text{overall}}}{\eta_f} \]

\[ M_{\text{set}} = \text{set actuator overload protection} \]

\[ C_s = 1.0 \]

5.2 Stop situation

• Operating torque reached (Rotary switch)
• End of stroke (Limit switch overrun)

Switch-off by overload protection

\[ M_{\text{calc}} = (M_{\text{set \ \text{class}}} + \Delta M) \cdot C_s \cdot \frac{\eta_{\text{trans}} \cdot \eta_{\text{overall}}}{\eta_f} \]
CU = increasing factor as a function of signal response time and flywheel effect as advised and proved by manufacturer.

5.3 Stop situation

- Limit and rotary switch overrun

Stop due to friction clutch slipping

\[ M_{cal} = (M_{max\clutch} + \delta M_{clutch}) \cdot Cd \cdot \frac{i_{trans} \cdot \eta_{overall}}{f} \]

\( \delta M_{clutch} = \) torque rating tolerance within which friction clutch starts slipping as advised by manufacturer

Cd = dynamic increasing factor due to flywheel effect for actuators with friction clutches.

Manufacturer's data with proof should be used.

Mclutch = friction clutch torque setting

5.4 Stop situation

- Limit and rotary switch overrun

No friction clutch fitted

In the case of actuators without friction clutch the motor stalling torque with 110%. \( \eta_{rated} \) at the motor terminals and the dynamic factor Cd are taken into account in determining Open/Close torques.

\[ M_{cal} = M_{max\ motor} \cdot (1,1)^2 \cdot Cd \cdot \frac{i_{trans} \cdot \eta_{overall} \cdot \eta_{act}}{f} \]

or

\[ M_{cal} = M_{max\ drive} \cdot \frac{i_{trans} \cdot \eta_{overall}}{f} \]

\( M_{max\ drive} = \) maximum drive torque as advised by manufacturer (possibly data from individual test) with consideration of the existing valve stiffness and the maximum terminal voltage.
In the case of drive motors with terminals connected to a controlled busbar the most unfavourable limit of the controlled voltage should be used in the torque equation.

\[ C_d = \text{dynamic increasing factor due to flywheel effect. Manufacturer's data with proof should be used.} \]

The maximum motor torque \( M_{\text{max.motor}} \) (starting or stalling torque) should be confirmed by the motor manufacturer.

5.5 Stop situation

• Bridging

In "bridging" the actuator torque setting is rendered inoperative. In this case the valve parts affected by the actuating force should be made of sufficient strength to withstand maximum torque conditions i.e. either the motor stalling torque of 110 % \( U_{\text{emf.}} \) at the motor terminals or the friction clutch torque.

\( M_{\text{calculation}} \) see 5.4

For checking the strength of the valve parts affected by the actuating force, the force acting upon the valve spindle must be derived from the design torque \( M_{\text{design}} \). \( \mu_{\text{min}} \) should be used as coefficient of friction between spindle and spindle nut in this case.

6. Proof of strength (permissible stress) of valve parts affected by the actuating force under operating conditions and at the various actuator stop situations

The stress values of valve parts affected by the actuating force should be determined by using the torque ratings as per Para 5.

We have drafted recommendations for the individual load conditions so that valve parts affected by the actuating force during valve movement will not be overstressed. However these are currently under discussion for some components and will be submitted later, if required.
7. Simplified calculation method for valves already used in a plant

Determination of the required torque $M_{\text{set requ.}}$ is based on the following:

- the mean value of the coefficient of friction between spindle and nut $\mu_a = 0.15$,
- the mean value of the coefficient of friction $\mu_p$ for the valve gate ($\mu_p = 0.4$),
- a safety factor $S = 1.5$ for gate valves and $S \geq 1.2$ for globe valves,
- a factor $C = 1.1$ to take into account the setting tolerance for $M_{\text{set requ.}}$ and
- proper compression of the gland packing (this should be in accordance with assembly Instructions if new packing is used).

The required torque $M_{\text{req}}$, calculated in this way is then compared with the existing torque setting $M_{\text{set actual}}$.

If $M_{\text{set requ.}} \leq 0.85 \times M_{\text{set actual}}$ then no further detailed proof is required.

If $M_{\text{set requ.}} > 0.85 \times M_{\text{set actual}}$ then the following additional calculations are required as proof of suitability:

- determination of the required torque $M_{\text{req.}}$ for $\mu_{\text{Gmax}}$ in the thread, spindle and nut
- proof of strength of the components affected by the actuating force for $\mu_{\text{Gmin}}$ in the thread, spindle and nut, and
- consideration of the torque setting tolerance according to KTA 3504

as dealt with in Sections 2 and 3, should be applied.

The following are taken as a basis for proof of the strength of the components affected by the actuating force with the condition $M_{\text{set requ.}} \leq 0.85 \times M_{\text{set actual}}$

- a factor $C = 1.1$ to take into account the setting tolerance for $M_{\text{set actual}}$
- the mean value ($\mu_a = 0.15$) of the coefficient of friction for the spindle/spindle nut.

If $M_{\text{set requ.}} > 0.85 \times M_{\text{set actual}}$ then Section 5 should be applied.
EPRI MOV PERFORMANCE PREDICTION PROGRAM
J. F. Hosler
Electric Power Research Institute, 3412 Hillview Avenue, Palo Alto, CA 94303
P. S. Damerell
MPR Associates, Inc., 320 King Street, Alexandria, VA 22314
M. G. Eidson
Southern Nuclear Operating Co., 40 Inverness Center Parkway, Birmingham, AL 35242
N. E. Estep
Duke Power Company, 526 S. Church Street, Charlotte, NC 28201

ABSTRACT

An overview of the EPRI Motor-Operated Valve (MOV) Performance Prediction Program is presented. The objectives of this Program are to better understand the factors affecting the performance of motor-operated valves and to develop and validate methodologies to predict MOV performance. The Program involves valve analytical modeling, separate effects testing to refine the models and flow loop, and in plant MOV testing to provide a basis for model validation. The ultimate product of the Program is an MOV Performance Prediction Methodology applicable to common gate, globe, and butterfly valves. The methodology predicts thrust/torque requirements at design basis flow and differential pressure conditions, assesses the potential for gate valve internal damage and provides test methods to quantify potential variations in actuator output thrust with loading condition. Key findings and their potential impact on MOV design and engineering application are summarized.

BACKGROUND

During the mid to late 1980's, motor-operated valve failures/incidents in US nuclear power plants resulted in an increased emphasis by both the US nuclear industry and the US Nuclear Regulatory Commission (NRC) on improving the performance, reliability and predictability of MOVs. In response, the Electric Power Research Institute (EPRI) initiated efforts to document existing MOV maintenance and engineering evaluation technology in the form of technical repair and engineering application guides and initiated a study to assess long-term industry needs.
MOV Application Guides

In the 1988 to 1990 time frame, EPRI’s Nuclear Maintenance Application Center (NMAC) worked together with utility experts to develop several technical repair guidelines for Limitorque motor actuators. In addition, NMAC developed the Application Guide for Motor-Operated Valves in Nuclear Power Plants (Grant, 1990) to document the existing state-of-the-art in conducting engineering evaluations of MOV applications. The scope of this document included rising stem gate and globe valves powered by Limitorque motor actuators. The Application Guide addressed definition of MOV requirements for an application, determination of required stem thrust for gate and globe valves, and determination of available thrust from Limitorque actuators. A similar guide covering butterfly valves was developed by NMAC in 1993 (Eldiwany, 1993).

At the time these guides were developed, it was recognized that there existed areas of uncertainty in MOV performance prediction. Examples included disc-to-seat friction coefficients for gate valves, guide friction coefficients for gate valves, flow loading and mid-stroke effects for gate valves, unwedging thrust for gate valves, sealing load for gate and globe valves, stem-to-stem nut friction coefficients, and variation in operator output thrust at torque switch trip with DP loading condition. In addition, uncertainty existed in methods for predicting hydrodynamic torque loading on butterfly valve discs.

Definition Study for EPRI MOV Performance Prediction Program

During 1989, a planning study was conducted by EPRI to determine the extent to which the known areas of uncertainty in existing predictive methods could be addressed by a generic test program. This study also examined available data from other test programs and from in-plant tests to assess how these data could be utilized in such a program. To ensure industry needs were appropriately identified, the study was coordinated with the utility MOV User’s Group (MUG), as well as the Nuclear Utility Management and Resources Council (NUMARC). The result of the study was a recommendation for a generic MOV research program.
As part of the study, a preliminary database of MOV applications in nuclear power plants was created. Review of the database indicated that about 50% of the 16,000 safety-related MOVs in US nuclear units were gate valves, and about 20% each were globe and butterfly valves. Valve population distributions were also defined in terms of size, pressure class, manufacturer, system, DP, etc. This distribution demonstrated that the valve population was very diverse, with only limited instance of identical or similar valves in service at multiple locations. This finding meant an approach to assess MOV performance using "type testing" would be impractical and cost-prohibitive.

The study recommended that an MOV test and analysis program be conducted with the objective of providing improved and validated methods for predicting MOV performance. The scope of the program was to cover common gate, globe and butterfly valves used in safety-related applications. It was recommended that the necessary modeling to develop improved methods follow a first-principles approach which addressed known areas of uncertainty identified during development of the application guides. A combination of separate effects testing, flow loop testing and enhanced in-plant testing was recommended as the most effective approach to provide the needed data. These recommendations laid the fundamental groundwork for the EPRI MOV Performance Prediction Program (PPP). Based on the results of this study, EPRI formed a utility Technical Advisory Group (TAG) comprised of utility industry MOV experts to provide guidance to EPRI in the detailed formulation and execution of the Program. The Program was formally initiated in the fall of 1990 and is scheduled for completion in July 1994.

OBJECTIVES AND SCOPE

The objectives of the EPRI MOV Performance Prediction Program are to:

1) Provide short-term products to utilities to allow expeditious evaluation of MOVs based on existing technology. The short-term program includes the following activities.

- Development of an In-Situ Test Guide
- Development of a computerized MOV General Information Database
- Development of an MOV Margin Improvement Guide
• Review of an NRC sponsored Gate Valve Test Program conducted by the Idaho National Engineering Laboratory (INEL)

and

2) Conduct a long-term program to develop and validate improved methods for predicting motor-operated valve performance. Such methods can be used to demonstrate the design basis capability of MOVs in cases when valve unique design basis test data is not available. Key aspects of the long term Program are summarized below.

• Development of improved methods for prediction or evaluation of:
  a) System flow parameters - calculation of differential pressure versus stroke position,
  b) Gate valve performance - calculation of required thrust and potential for internal damage,
  c) Globe valve performance - calculation of required thrust,
  d) Butterfly valve performance - calculation of required torque, and
  e) Motor Operator dynamic performance - quantification of variations in actuator output thrust with DP loading condition;

• Separate effects tests providing information to refine the gate valve and operator methods:
  a) Gate valve friction - determine friction coefficients and damage thresholds for gate valve internal components,
  b) Gate valve design effects - understand the interaction of gate valve internal components,
  c) Operator separate effects - understand observed variations in actuator output thrust with DP loading condition, and
  d) Operator stem/stem-nut lubricant performance - establish qualitative lubricant comparison data;

• MOV tests providing data for model/method development and validation:
  a) Flow loop testing of 34 gate, globe and butterfly MOVs under a wide range of flow conditions,
b) Flow loop testing of six butterfly valve disc designs to assess flow and inlet piping effects, and
c) In-plant (i.e., in situ) tests of 35 MOVs.

SHORT-TERM PROGRAM

The products developed to support near-term utility evaluation and testing of MOVs and definition of the scope and focus of the EPRI Program are reviewed below.

In Situ Test Guide

The In Situ Test Guide for Motor-Operated Valves was prepared to provide guidance on the requirements for in-plant test data. The test guide considered existing industry experience by incorporating elements of the in situ testing guide developed by the UG Thrust Calculations and Switch Settings Committee. The In Situ Test Guide includes requirements for test instrumentation accuracy and recording speed; valve inspection, measurement and documentation; as well as overall documentation of the data package. For the enhanced in situ tests utilized in the EPRI MOV PPP, additional requirements include measurement of time-history DP across the valve, as well as internal dimensions for gate valves.

MOV General Information Database

Utilities identified that an MOV database would be helpful to facilitate communication among nuclear utilities on MOV-related issues. In response to this need, EPRI prepared a PC-driven database covering over 5000 nuclear safety-related MOVs. The database contains over 40 fields of information for each MOV and is updated approximately semi-annually. Each participating utility has a copy of the database on diskette. In addition to facilitating communication among utilities, the database also provided a basis to assess the MOV population for selecting valves for the Flow Loop and In Situ Test Programs.

NRC/INEL Gate Valve Test Review

During 1988-1990, INEL conducted gate valve testing for the NRC (Steele, 1990). The primary emphasis was to evaluate gate valve performance in BWR blowdown
isolation service in resolution of NRC Generic Issue 87. One key conclusion from the
test program was that use of the standard industry equation in combination with disc
factors historically assumed by the valve vendors may under-predict gate valve stem
thrust requirements under certain conditions. In addition, certain gate valve designs
were found to be susceptible to damage during closure under blowdown flow
conditions. The damage was attributed to high contact loading on the guides and the
seats due to disc tipping. The tests also provided strong evidence that there were direct
loads applied by the flow to the disc in not only the pipe-axis direction, but also the
stem-axis direction. These loads resisted the valve's opening motion and are considered
to be the result of Bernoulli forces.

EPRI performed a review of the NRC/INEL tests. This review identified the
apparent disc factors associated with various valve designs and conditions tested. This
effort also included detailed inspections of the INEL valves, and measurements of key
dimensions and material properties. Finally, the review identified specific areas where
the information and experience from this testing could be factored into the modeling
and test activities being planned as part of the EPRI MOV Performance Prediction
Program.

Margin Improvement Guide

The Motor-Operated Valve Margin Improvement Guide supplements the Application
Guides by identifying specific actions which can be implemented to increase available
actuator margin and references the appropriate sources where additional information to
support evaluation of specific situations can be obtained. It also incorporates detailed
guidance for MOV limit switch configurations and settings, which was developed by
the MUG Thrust Calculations and Switch Settings Committee.

LONG TERM PROGRAM

The long term program includes development of improved methods for
prediction of valve performance and motor operator dynamic effects, separate effects
testing to provide a basis for method refinement, and flow loop and in plant MOV
testing to support method validation. The following is a description of each long-term
program activity and, where appropriate, a summary of key findings.
PREDICTIVE METHODS

System Model

To predict the performance of valves it is desirable to accurately predict the differential pressure across the valve disc over the full range of stroke positions. To accomplish this task, a computer-based system model was developed. The model allows simulation of a variety of piping configurations, including single and parallel line pumped flow configurations with up to two active MOVs, as well as a single line blowdown configuration.

The system model requires plant engineering inputs to define the frictional and flow driver characteristics of the piping system in which the MOV is installed. The system model predicts the DP across the MOV at all stroke positions. The predicted DP versus disc position relationship is used as input to the gate, globe and butterfly valve models to determine disc loading at all positions. The system model is separately validated by comparing predicted DP versus position to that measured during flow loop and in plant testing.

Gate Valve Model

The development of a computer-based gate valve model represents the most challenging goal of the EPRI MOV Performance Prediction Program. At the outset, it was determined that the gate valve model would need to be able to address the fluid loading on the disc as well as the detailed mechanical interaction between the stem, disc, guides and seat, including the potential for material damage at sliding interfaces.

Computational fluid dynamics analyses were performed to evaluate fluid loading. A simplified algorithm was incorporated into the model, based on the results of these analyses, to compute vertical and horizontal forces on the disc at all disc positions as a function of valve DP. A detailed mechanical model which determines disc force equilibrium over the full range of disc positions was then developed. The model accounts for disc tipping within the constraints of the guides. Results from gate valve design separate effects testing were used to refine the model and to verify that the disc behavior is being properly calculated. Based on the results of friction separate effects testing, a friction algorithm was added to the mechanical model to determine
friction coefficient as a function of the material pair, contact mode, contact load (or stress) and fluid temperature. Material damage is also predicted under modes where damage is likely to occur. The resultant model is able to predict the stem thrust to move the disc for both opening and closing strokes using DP as a function of stroke position as input.

The sliding friction coefficients in the gate valve model are intended to be bounding values. The model allows the user to manually input a value for the disc-to-seat sliding friction coefficient to replace the value determined by the friction algorithm. Methods are provided to determine disc-to-seat sliding friction coefficients from valve specific test results. These methods include use of data obtained from full DP tests, partial DP tests, hydropump DP tests and static (zero DP) tests.

The gate valve model is applicable to solid and flexible wedge gate valves with single piece discs, a conventional guiding arrangement with guide rails and slots, and a stem-to-disc connection consisting of a T-head and a T-slot. Required valve internal information (dimensions and materials) for input can be obtained from the valve manufacturer by use of a specification provided as part of model documentation. Alternately, valve specific internal measurements, if available, can be used to define input parameters. The gate valve model is validated by comparing predictions of thrust to data obtained from flow loop, in-plant and previous INEL (Steele, 1990) testing.

Globe Valve Model

The globe valve model predicts thrust requirements under differential pressure loading for the full range of the valve stroke positions. The model is applicable to globe valves with T-pattern or Y-pattern bodies, rising or rising rotating stems, and balanced or unbalanced discs. Both underseat and overseas flow configurations can be accommodated. The model can compute required thrusts under incompressible, pumped flow, conditions. The model is computer based and is validated by comparison of predicted thrust to that measured during flow loop and in-situ MOV testing.

It is necessary to select the appropriate disc area (either disc seat or disc guide area) for DP application in order to accurately predict required thrust. The globe valve
methodology provides guidance in selecting the appropriate disc area based on valve internal design characteristics.

Under hot water blowdown conditions, significant disc side loading can potentially occur due to pressure variations within the valve body. Although disc side loading is not currently evaluated by the model, non-computer based methods to account for this effect are under development and will be included as part of the final Program documentation package.

**Butterfly Valve Model**

The butterfly valve model determines the required torque to operate butterfly valves through their full range of stroke positions. Two types of torque calculations are performed: a seating/unseating torque which applies when the disc is near the fully closed position and a total dynamic torque which applies throughout the remainder of the stroke. Seat torque can be predicted for seats which are in a new or well-maintained condition; however, use of valve test results is recommended for valves where the seats may have degraded or aged. Total dynamic torque is predicted using a bearing torque component and a hydrodynamic torque component. The key advances in this technology are the development of generic hydrodynamic torque coefficients which account for disc type, orientation and aspect ratio, as well as multipliers to account for the influence of upstream elbows.

The butterfly valve model is computer based and is applicable to symmetric, single offset and double offset valves with circular discs. The model can predict torque requirements with flow in either direction. The model applies to incompressible flow and compressible choked flow and is validated by comparing torque predictions to data obtained from flow loop, in situ and previous INEL (Steele, 1986) testing.

**Gate Valve Empirically Based Methods**

In addition to the computer-based model which addresses conventional solid and flexible wedge gate valve designs, manual calculational methods are being developed to address the following unique gate valve design configurations:
• Parallel double-disc with internal wedge;
• Flexible wedge with pin-and-link stem connection;
• Split wedge valve with ball and socket joint;
• Split wedge valve with spacer ring joint; and
• Parallel expanding valve.

These methods provide guidance in applying EPRI flow loop and in-situ testing results to plant specific MOV applications of these valve designs.

**Operator Dynamic Effects Methods**

EPRI and industry MOV testing revealed that a significant reduction in actuator output thrust at torque switch trip can occur when the valve is loaded slowly (i.e., under differential pressure conditions) relative to the observed thrust output at torque switch trip under static (no DP) conditions. This phenomena has been called the "rate-of-loading" effect or "load sensitive behavior". Since torque switches are generally set under static conditions, the potential exists that insufficient thrust capability will exist when the valve is subjected to design basis DP and flow conditions.

Separate effects testing was conducted to better understand the root cause for the "rate-of-loading" phenomenon and to develop methods for quantifying the potential effect for a given installed MOV. Based on this testing and testing conducted by Steele (1992), it was concluded that this phenomenon is attributable to specific characteristics of the stem/stem-nut and lubricant combination and is not amenable to analytical treatment. Some level of MOV unique testing is necessary to accurately assess the effect for a given MOV.

Several alternative approaches are being assessed to provide utilities with means for accommodating potential "rate-of-loading" effects. These methods are summarized below.

• Impose a margin penalty if no valve specific data is available.

• Set the torque switch with a reduced loading rate i.e., by use of handwheel or gear reduction on the motor. This method requires thrust measurement but torque measurement is not required. Some
margin penalty may still be required with this approach but can be
minimized if torque measurements are also made.

- Set the torque switch with a DP load simulator device. If successful,
  this device will accurately reproduce the maximum coefficient of
  friction that could occur at the stem/stem-nut interface under design
  basis DP loading conditions. This method requires only a thrust
  measurement.

- Use one of two approaches recommended by INEL. These methods
  are deemed the "threshold" method and the "fold line" method. These
  methods require the measurement of both thrust and torque and
  involve testing under static or relatively low DP conditions.

- Modify the control switch logic to bypass the torque switch until flow
  isolation, but not necessarily leak tightness, is achieved. The torque
  switch would be set at a nominal setting so as not to impose excessive
  thrust loading during static tests. This approach would eliminate the
  need to add margin to accommodate potential "rate-of-loading" effects
  but would still require evaluation of actuator capability to achieve the
  required thrust while the torque switch is bypassed.

SEPARATE EFFECTS TESTING

Friction Testing

- Test fixtures were fabricated to allow determination of sliding friction
  coefficients and damage threshold load levels for the range of material pairs, contact
  geometries and stresses, and water/steam temperatures and pressures typically found
  in gate valves installed in nuclear power plants. Four predominant material
  combinations were tested: Stellite 6 on Stellite 6; Stellite 6 on carbon steel; Stellite 6 on
  stainless steel; and carbon steel on carbon steel. Tests were conducted under water and
  steam conditions at temperatures ranging from room temperature to 650°F.

  Stellite on Stellite sliding friction coefficients under room temperature water
  conditions were found to increase significantly, from approximately 0.2 to greater than
0.6, and eventually reach a "plateau" maximum value as the number of strokes is increased. Hot water and steam friction coefficients for Stellite on Stellite were found to be lower than cold water "plateau" sliding friction coefficients and did not vary significantly with stroke number. Carbon steel on carbon steel friction coefficients and the potential for gouging damage were found to increase significantly as temperature increased from 70 to 120°F.

**Gate Valve Design Effects Testing**

To ensure that the gate valve model would accurately predict disc orientation, contact points and damage threshold levels, a test fixture was fabricated in which actual gate valve internal parts could be transiently loaded with hydraulic pistons to simulate DP loading conditions. A comprehensive set of parametric tests were carried out to assess the influences of variation in guide lengths, guide clearances, guide materials, and disc and body seat edge radii. The results revealed that disc and body seat edge radius or chamfer are critical parameters affecting valve performance and the potential for valve internal damage. The results of this testing were used as a basis for refinement of the assumptions made in the gate valve model.

**Operator Dynamics Testing**

To support development of the Operator Dynamic Effects Methods described earlier, a test fixture was fabricated to simulate the full range of MOV loading conditions. The test fixture incorporated a hydraulic cylinder capable of providing a pre-programmed back loading on the end of the stem as the actuator attempts to move the stem in the closing direction. The test fixture also includes a hard stop to simulate high loading rates typical of those which occur in gate or globe valves during wedging or seating. A comprehensive set of parametric tests is being conducted in which the effects of variation in loading level, loading rate, stem/stem-nut combinations, and stem/stem-nut lubricants are evaluated. The Operator Test Fixture is being used to assess and validate a variety of approaches to account for the "rate-of-loading" effect in MOV switch set up and margin determination.

Detailed assessment of the test results indicates that the "rate-of-loading" effect is caused by a transient reduction in the stem/stem-nut coefficient of friction when the stem is loaded at a high rate i.e., during a static closure. This effect is postulated to
result from the fact that under high loading rate conditions, the load is increased to a high level in a very short time before the lubricant can be fully squeezed from the stem nut threads. During this short time period (<100 ms), a mixture of hydrodynamic and boundary lubrication modes is in effect and can result in transiently very low friction coefficients. If the actuator has been running for sufficient time at a sufficiently high load, most of the grease is squeezed out resulting in predominantly boundary lubrication and somewhat higher friction coefficients typical of those expected under DP loading conditions. Only a fraction of the stem/stem-nut combinations tested exhibited a significant "rate-of-loading" effect. These findings and general conclusions regarding the cause of the phenomenon are consistent with those documented by INEL (Steele, 1992).

Stem/Stem-Nut Lubricant Testing

A separate effects test program was conducted to assess the friction and wear characteristics of various greases and solid films that are now or could be applied as stem/stem-nut lubricants in MOVs. A total of 21 lubricants were evaluated in a test fixture designed to simulate a motor-operated valve application. The effects of stroke number and loading level on friction and wear were evaluated. Although the maximum friction coefficients ranged from approximately 0.1 to 0.2 for the grease-type lubricants tested, most were bounded by a value of approximately 0.15. Friction coefficients were generally observed to decrease with increasing stroke number. The solid lubricants tested exhibited poor performance.

Data from these tests can be used by utilities as a basis for qualitatively comparing the performance of the lubricants tested. The results are not used directly as part of the predictive methodology.

FLOW LOOP TESTING

Full Scale MOV Testing

Flow loop testing was conducted in four flow loops, three of which were located in the United States and the fourth in Karlstein, Germany. Thirty-four MOVs were subjected to a combined total of more than 1200 formal test strokes. The text matrix covered a wide range of flow, temperature and differential pressure conditions.
Twenty-eight gate, four globe and two butterfly valves were tested. Valve test candidate size, pressure class and manufacturer were selected based on their predominance in the MOV General Information Database and on availability. Gate valves tested ranged in size from 2-1/2 to 18 inches, and globe valve sizes ranged from 2-1/2 to six inches. Two six inch butterfly valves were tested.

Prior to testing, all MOVs were disassembled, and comprehensive internal measurements and photos were taken. The valves and actuators were then reassembled and prepared for testing. All MOVs were tested under "baseline", 15 feet per second, cold water flow conditions over a range of DPs up to the maximum expected in nuclear power plant applications. After cold water testing, selected valves received parametric testing to assess the effects of variation in fluid temperature, fluid velocity, and flow media. Tests were conducted with cold water flow velocities up to 50 feet per second, as well as hot water and steam blowdown flow conditions. Differential pressures ranged from zero to 2650 psid. Internal valve inspections were conducted at regular intervals to assess any valve internal damage.

High speed data was acquired to record the following parameters.

- valve DP
- valve inlet pressure
- fluid temperature
- fluid flow rate
- direct stem force
- direct stem torque
- spring pack displacement
- disc position
- motor voltage
- motor current
- motor active power
- limit and torque switch actuation

Friction separate effects testing results indicated that under cold water conditions, Stellite on Stellite sliding seat friction coefficients ranged from 0.2 to greater than 0.6. This variation is attributed to the number of loaded strokes applied and the contact stress level. Although not evaluated under this Program, some industry experience indicates that the duration of exposure to the fluid may also play a role. The friction coefficients were found to eventually stabilize at a maximum value. To minimize the potential impact of the "stroke effect" on valve performance during planned flow loop parametric testing, all gate valve seats were "preconditioned" by short stroking the valve into the seats under DP loading until the sliding friction coefficient reached a maximum "plateau" level. The number of strokes required to reach
the "plateau" level of friction varied widely from approximately 100 to as many as 900 strokes.

Once gate valves had been "preconditioned" they were subjected to full flow tests at cold water flow velocities ranging from 15 to 50 feet per second. Apparent friction coefficients ranged from 0.2 to 0.9 during these tests. These "apparent" disc coefficients of friction include all valve performance phenomena and are not necessarily representative of sliding friction alone. Although a significant range of apparent disc friction coefficients was observed, only a single gate valve sustained internal damage under cold water, 15 feet per second flow conditions. In this case, the disc was pushed through the seats allowing leakage above the disc on an 18 inch valve with a 3 degree seat half angle.

Hot water, pumped flow, testing resulted in lower sliding friction coefficients than for cold water, generally ranging from 0.3 to 0.5. The "stroke" effect on sliding friction was not observed to be significant under hot water conditions. Under hot water and steam blowdown conditions at 1200 psid, apparent disc friction coefficients ranging from approximately 0.3 to 0.8 were observed. Although some valve designs were undamaged, others sustained significant guide and/or seat damage.

Disc to body seat friction sliding coefficients were found to decrease with increasing DP. This finding confirms friction separate effects testing results and supports the use of sliding friction coefficients obtained from reduced DP test results in the evaluation of thrust requirements under full DP conditions for gate valves in pumped flow systems.

Testing of globe valves under incompressible flow conditions revealed that it is necessary to select the appropriate area (either disc seat or guide area) for DP application, in order to accurately predict required thrust. Under compressible flashing flow conditions, excessive thrust loading was observed which exceeded even guide area based predictions. Side loading of the disc due to pressure variations within the valve body appears to play a role in this phenomenon.

Vendor methodologies for predicting required hydrodynamic torque for butterfly valves are generally proprietary, and as a result, little can be concluded at this time regarding the suitability of such methods based on Program test results. The EPRI
butterfly valve model accurately predicts butterfly valve performance for the valves tested in the flow loops.

**Subscale Butterfly Valve Parametric Testing**

To comprehensively assess the effects of butterfly valve disc design and upstream elbow effects a parametric test series was conducted at small scale. Selected results were compared to large scale data for the same disc design to confirm scaling relationships. An existing test facility was modified to allow the insertion of six different butterfly disc designs. Baseline testing was conducted on each disc design to assess flow and DP effects on required hydrodynamic torque. In addition, selected disc designs were parametrically tested to assess upstream elbow distance and orientation, as well as flow direction, effects.

**In Situ MOV Testing**

To supplement the flow loop testing of MOVs, data from approximately 35 MOV tests conducted in nuclear plants are being obtained and formally documented. In situ test data is being obtained for 19 gate, seven globe and nine butterfly valves. Gate valves tested range in size from three to 18 inches, globe valves from two to 18 inches and butterfly valves from ten to 42 inches. Tested differential pressures ranged from zero to 2880 psid. The test data obtained generally includes high speed data acquisition to measure and record the following parameters.

- stem thrust (gate and globe valves)
- stem torque (butterfly valves)
- valve upstream pressure
- valve differential pressure
- motor current
- spring pack displacement
- actuator control switch actuation

In addition, internal measurements were obtained on gate valves to support validation of the gate valve model.

The in situ data are used to demonstrate the capability of the MOV PPP methodologies to predict the performance of "real world" valves installed in nuclear power plants.
CONCLUSIONS

The research conducted as part of this Program has resulted in a giant step forward in the general understanding of motor-operated valve behavior and the ability to accurately predict MOV performance. As a result of this Program, fully validated methods will, for the first time, be available to confirm the adequacy of existing MOV installations and control switch settings and to support the evaluation of MOV modifications or replacements. The lessons learned from this Program should be factored into future valve and actuator development and into the design of advanced nuclear plant systems.

Specific conclusions are summarized below:

- Cold water Stellite on Stellite sliding friction coefficients can be highly variable, ranging from less than 0.2 to greater than 0.6.
  - Friction coefficient variation appears to be based on the number of loaded strokes applied and the contact stress level. Duration of exposure to the fluid may also play a role.
  - Friction coefficients increase with stroke number to a maximum "plateau" level, then stabilize.

- Hot water Stellite on Stellite sliding friction coefficients are less variable, generally ranging from 0.3 to 0.5

- Under pumped flow (~15 feet/second) conditions, the potential for internal damage to gate valves is extremely low.

- Under high velocity flows (> 50 feet/second up to blowdown conditions) the potential for valve internal damage increases significantly for some gate valve designs.

- Gate valve disc and body seat, as well as disc guide slot and body guide, edge radii or chamfers can have a profound impact on the potential for valve internal damage. Sharp edges should be avoided.
• Under pumped flow conditions, gate valve disc to seat sliding friction coefficients tend to decrease with increasing DP. This finding supports the use of friction coefficients measured under reduced DP conditions when evaluating thrust requirements at higher DPs.

• Under incompressible flow conditions, globe valve thrust requirements can be predicted accurately if the appropriate area (disc vs. guide) is assumed for DP application. The EPRI methodology provides guidance in selection of the appropriate area based on specific globe valve internal design features.

• Under compressible flow conditions, globe valve thrust prediction requires consideration of potential disc side loading.

• For some butterfly valve designs and flow combinations, hydrodynamic torque loading can dominate total torque requirements.

• For some gate and globe valves, significant reductions in motor-operator output thrust can occur under dynamic (DP loading) conditions relative to those which occur under static (no DP) conditions.

• The EPRI MOV Performance Prediction Program provides validated methods to appropriately bound thrust/torque requirements for common gate, globe and butterfly valves and several alternative approaches for accommodating potential "rate-of-loading" effects on actuator output thrust.

ACKNOWLEDGMENTS

Many individuals have played significant roles in the development and implementation of this major research effort. Mr. Boyd Brooks (EPRI retired) initiated the planning study which led to the Program. Mr. Robert Elstrom, then employed by Toledo Edison Company, served as the initial chairman of the utility Technical Advisory Group (TAG) formed to guide Program development and was later succeeded by Mr. Michael Eidson of Southern Nuclear Company. The support and guidance provided by the TAG was vital to the successful completion of the Program. The following is a listing of those people who have served on the TAG:
Mr. John Allen, Tennessee Valley Authority
Mr. Denver Atwood, Southern Nuclear Company
Mr. Bill Black, TU Electric
Mr. Clive Callaway, NUMARC
Mr. Brian Curry, Philadelphia Electric Company
Mr. Michael Eidson, Southern Nuclear Company (TAG Chairman)
Mr. Robert Elfstrom, Toledo Edison Company (former TAG Chairman)
Mr. Neal Estep, Duke Power Company (TAG Vice-Chairman)
Mr. Chris Hansen, Yankee Atomic
Mr. Sam Henry, Tennessee Valley Authority
Mr. Robert Kershaw, Arizona Public Service Company
Mr. Nick Konstantinou, Commonwealth Edison Company
Ms. Stephanie Lane, Arizona Public Service Company
Mr. Fred Martisen, New York Power Authority
Mr. Robert McPherson, Southern California Edison Company
Ms. Susan Montgomery, Pennsylvania Power & Light Company
Mr. Robert Prato, Baltimore Gas and Electric Company
Mr. Michael Rose, Pennsylvania Power & Light Company
Mr. Ron Scherman, Cleveland Electric Illuminating Company
Mr. Robert Woehl, Pacific Gas & Electric Company

The efforts of the EPRI MOV Program staff including Mr. Larry Dorfman, Mr. William Kennedy, Mr. William McDaniel, Ms. Jenny Preciado and Mr. Kenneth Wolfe and are acknowledged for, without their tireless efforts, the Program could never have been accomplished.

Finally, the contractors' who actually conducted and supported the research must be acknowledged. These include:

Battelle Columbus
Bolt and Associates
Continuum Dynamics
Kalsi Engineering
Liberty Technologies, Inc.
MPR Associates
Siemens/KWU
S. Levy, Inc.
Teledyne Engineering Services
Toledo Edison Company
Wyle Laboratories
Vectra Corporation

-19-

593
REFERENCES


Nuclear Power Plants," EPRI Report NP-6660-D.

Steele, R. and DeWall, K. G., 1990, "Generic Issue 87 Flexible-Wedge Gate Valve Test
Engineering Laboratory.

CR-1023, Proceedings, Second NRC/ASME Symposium on Pump and Valve
Testing, pp. 25-37.

Steele, R. et al., 1986, "A Study of Typical Nuclear Containment Purge Valves in an
Accident Environment", NUREG/CR-4648.
Session #5

TESTING AND MAINTENANCE

Chairman: Mr. T. Scarbrough (U.S. NRC)

The need to re-evaluate the capability of MOVs to perform their safety function under design-basis conditions is well-recognized in light of operating experience and research results. After completing the re-evaluation and taking action to ensure MOV design-basis capability, utilities will need to establish and implement programs to maintain the capability of MOVs to perform their safety functions under design-basis conditions. During Session #5, four speakers discussed testing and maintenance activities to help ensure the continued design-basis capability of MOVs. In addition, one speaker discussed maintenance activities for solenoid-operated valves (SOVs).

Mr. M. Dubois, of AIB Vinçotte Nuclear, discussed the activities of the nuclear utilities in Belgium to improve the operational readiness of MOVs. Mr. Dubois stated that Belgian utilities requested the architect-engineer (TRACTEBEL) to perform a design-basis review for all safety-related MOVs and a validation of the actuator sizing. From this re-evaluation, the Belgian utilities developed a program to set, verify and maintain the correct torque switch settings for the MOVs. Mr. Dubois reported that a significant amount of adjustments or modifications were performed on the MOVs as a result of the program. He stated that preventive maintenance is performed periodically on each MOV. He also described the increased use of diagnostic equipment of monitor periodically MOV performance. He believed that MOV diagnostic equipment provided a more realistic view of MOV performance than ASME Section XI stroke time testing.

Mr. S. Takeda, of TOA Valve Co., Ltd., discussed a diagnostic system for MOVs being developed and used in Japan. Mr. H. Sakamoto of Hokkaido Electric Power Co. discussed the use of MOV diagnostic equipment at the Tomari Power Station in the restoration of MOV capability after overhaul of the MOV.

Mr. W. Reiger, of TÜV Südwest, discussed operating experience with MOVs that showed that the requirements for a safe function are not fulfilled during all operating and test conditions. Mr. Reiger provided examples of MOV problems, their causes, and actions taken to resolve those problems. He discussed recommendations to ensure the capability of MOVs and to maintain that capability. He also described new designs for gate and globe valves to help improve MOV performance and minimize MOV problems.

Mr. V. Varma, of the Electric Power Research Institute, provided helpful information on the application, use and maintenance of solenoid-operated valves.
Presentation at the OECD

Working Group No.1 on the Specialist Meeting on MOVs
25-27th April 1994

Belgian Experience on MOV Diagnostic Methods

P. COENRAETS
J. MAES
R. VANDENBUSSCHE
M. DUBOIS

ELECTRABEL  Tihange NPP
ELECTRABEL  Doel NPP
TRACTEBEL
AIB-VINCOTTE NUCLEAR

597
ABSTRACT

Since the publication of the USNRC IE Bulletin 85-03 following the DAVIS-BESSE total loss of feedwater, in depth research in laboratory and in situ experiences have extended the know-how about the MOVs and led to further recommended actions in Generic Letter 89-10 and its supplements.

The GL 89-10 action program concerns all safety-related MOVs and must be implemented in successive stages:

- verification of the correct design for operation under normal and abnormal conditions and in situ adequate settings of the parameters
- definition of a surveillance and maintenance program in order to ensure the reliability and the operability of the MOVs along the plant life

AIB-VINCOTTE NUCLEAR, which is in charge of Belgian nuclear power plants safety inspections, has requested all the Belgian utilities to apply the recommended actions in the USNRC IE Bulletin 85-03.

This presentation will give an overview of the present state in the Belgian nuclear power plants about the MOV problems, in particular:

- the maintenance policy decided by the Belgian utilities, to meet the intent of GL 89-10
- the diagnostic methods finally adopted on the two Belgian sites and their performances
- the identification of the MOV- operational problems and their resolution.
Contents

1. Introduction and Background

2. MOV operating principles

3. Program of Belgian utilities
   3.1. Requirements of Generic Letter 89-10
   3.2. Design-basis review
   3.3. MOV diagnostic method
   3.4. Maintenance Policy
      3.4.1. Preliminary actions
      3.4.2. Torque switch settings after design-basis review
      3.4.3. Periodic maintenance
   3.5. Results with the diagnostic methods

4. Conclusions

Appendix

Figures
1. Introduction and Background

Since 1980, numerous MOV problems have been reported in the United States by NRC Information Notices and Bulletins. IE Bulletin 85-03 (issued November 1985) recommended that American Utilities determine actions to improve the reliability of safety-related isolation valves and more precisely to develop a program to ensure that switch settings will operate under design-basis conditions for the life of the plant.

In Belgium, the Belgian Regulatory Body, AIB-VINCOTTE NUCLEAR (AVN), has considered this problem very closely and has asked the utilities to apply the recommended "ACTIONS" of IE Bulletin 85-03.

As a result, the utilities of DOEL and TIHANGE have responded to the NRC recommendations. The results of the respective programs have been examined and described in a synthetic document that was reported at the IAEA in Vienna in May 1990 (IAEA-SR-169/4).

On the basis of the results of the inquiries recommended by the IE Bulletin 85-03, the NRC indicated that the MOV failures have a frequency higher than previously estimated.

In June 1989, the NRC issued Generic Letter 89-10 "Safety-Related Motor-Operated Valves Testing and Surveillance" which supersedes the recommendations of IE Bulletin 85-03 and extends its scope to include all safety-related MOVs as well as position-changeable MOVs, i.e. which may be inadequately positioned and actuated afterwards in the most severe operating conditions.

The NRC warned the utilities that ASME Section XI testing is not sufficient to ensure MOV operability under design-basis conditions because it cannot verify switch settings.

The NRC drew the attention of the NPP operators on the importance of design-basis degraded voltage effects on MOV performance.

Furthermore, it is not demonstrated that the test results with a differential pressure lower than the design-basis differential pressure can be extrapolated to determine the correct torque switch setting values by means of any diagnostic method.

The purpose of this lecture is to explain the way followed by Belgian utilities to improve the operational readiness of the motor-operated valves (MOV) in accordance to the requirements of the Generic Letter (GL) 89-10.

2. MOV operating principles. (figure 1)

Through a gear reduction box, an electrical motor drives a splined shaft on which slides a worm. The latter moves freely and compresses axially a spring pack, which is composed of a serie belleville washers. The worm rotates a worm gear that is linked to the valve drive sleeve.

During the valve stroke, the opposite forces are generally constant and the worm stays in its initial position imposed by both preloaded spring packs. When the worm gear-torque increases, the worm moves axially and rotates the torque switch by means of a cam-follower arrangement. The amplitude of its rotation is proportional to the forces that oppose valve stem movement.
From the moment the torque reaches a preset value, the cam activates the torque switch contacts, which cut off the electrical power to the motor. The cut-off torque is set by rotating the angular position of the torque switch cam.

The motor operator is also equipped with a limit switch, working according to the turn-counter principle.

When the rotation of the motor is reversed, the valve stem does not move as long as the stem nut does not contact the worm gear lugs. That allows the motor unit to start unloaded, to reach quickly its nominal speed and to obtain sufficiently inertia for the valve unseating.

3. Program of Belgian Utilities

After the publication of IE Bulletin 85-03, AVN has suggested to the utilities to examine at first a sample of MOVs in the main safeguards systems. The action plans carried out in the different units have all followed the requirements of the Bulletin although they have been adapted to the measurement methods available in each unit.

In 1987, as required afterwards by GL 89-10, the utilities have already asked the architect engineer, TRACTEBEL, a design-basis review for all safety-related MOVs, and a validation of the actuator sizing. From this revaluation, utilities have developed a large program to set, verify and maintain the correct torque switch values for the MOVs. At present, the important goals of this program are defined in a maintenance policy based on the experience already gained with the chosen diagnostic methods.

3.1 Requirements of Generic Letter 89-10

This Letter extends the scope of IE Bulletin 85-03 to include all safety-related MOVs as well as all position-changeable MOVs in safety-related systems. GL 89-10 recommends that utilities implement a program to ensure that switch settings for safety-related MOVs are selected, set, and maintained so that the MOVs will operate under design-basis conditions for the life of the plant.

The main requirements of the program are:

a. Review and documentation of the design-basis for each MOV.
   At this stage, the maximum differential pressure must be defined for both opening and closing of the MOV during both normal operations and abnormal events. Moreover, other design criteria used in choosing the particular MOV, such as the design-basis degraded voltage, should be examined.

b. Establishment of the correct switch settings.
   It includes a program to examine the adequacy of the methods for setting the different switches and, if necessary, to modify them.

c. Verification and correction of the settings in situ.
   A stroke test is required to demonstrate the operability of the MOV at the design-basis differential pressure and/or flow. If such a test is not possible, a justification should be documented.
   In each case, the MOV must be stroke tested at no-pressure and no-flow conditions.
d. Write or revise procedures to ensure and maintain the correct settings.

e. Document, analyse and justify each MOV failure and corrective action. The MOV data file should be periodically examined to establish the trending of the MOV operability.

3.2. Design basis-review

The architect engineer, TRACTEBEL, has reviewed the design-basis for all the safety-related motor-operated valves and has re-evaluated the operating conditions under design-basis degraded voltage.

The methodology of the design-basis review includes the following stages:

a. Determination of the design-basis conditions

Generally, three data sources have been considered such as the valve manufacturer, the actuator manufacturer and the architect engineer. The data have been collected for each valve in order to define the functional parameters (ΔP, Q) in normal and accidental conditions or, if necessary, in specific configurations.

b. Definition of the equipment type and the switch logic

Depending on the valve type, the switch logic can differ for the opening and/or the closing. The general principles are summarised in table 1.

<table>
<thead>
<tr>
<th>valve type</th>
<th>opening</th>
<th>closing</th>
</tr>
</thead>
<tbody>
<tr>
<td>slide gate</td>
<td>LS</td>
<td>LS (1)</td>
</tr>
<tr>
<td>globe</td>
<td>LS</td>
<td>TS</td>
</tr>
<tr>
<td>wedge</td>
<td>LS</td>
<td>TS</td>
</tr>
<tr>
<td>butterfly</td>
<td>LS</td>
<td>LS/TS (2)</td>
</tr>
<tr>
<td>cage</td>
<td>LS</td>
<td>TS/LS (2)</td>
</tr>
</tbody>
</table>

LS : limit switch
(1) TS as backup
(2) LS or TS according to valve type

TS : torque switch

This stage includes the consultation and the comparison of all the data and formulae used in the literature, by the manufacturers, and by the architect engineer in order to estimate the correct values for several parameters: valve friction coefficient (seat/slide), stem effects and packing loads, supplementary loads (wedgeing, remote control), conversion stem force/torque, gear effects. The calculation methods differ with valve type.
d. Determination of the minimum required torque value

For the calculation of the actuator output torque, the architect engineer has considered, in addition to the usual technical data, the torque switch tolerances, the torque value (85% Un), the closing switch logic, the available torque range, the motor sizing, the gear box, the remote control efficiency.

Only the most severe operating conditions are considered in order to calculate the minimum required torque value.

e. Validation of the actuator sizing

The actuator sizing is adequate if the torque switch trip value determined for a correct operation of the MOV is lower than the actuator output torque available at 85% nominal voltage.

f. Definition of the recommended torque setting value:

The equipment type, the function and the opening/closing logic have been considered. (see appendix 1)

3.3. MOV diagnostic method

In order to implement their maintenance policy, the Belgian utilities have acquired diagnostic systems that monitor the following key actuator parameters:

- actuator output torque
- torque switch rotation
- motor load
- control switch actuation

The analysis of these parameters and their change allows to establish the status of the MOV and detect eventual anomalies that will affect MOV operability.

The diagnosis strategy includes three stages:

Calibration

The calibration test is performed when the actuator is removed from the valve.

The calibration traces allow to correlate the motor load and the torque switch rotation with the actuator output torque.

On the test bank, a brake equipped with a torque cell is linked to the actuator output shaft. To simulate various loading conditions, the supply to the brake is regulated while the torque cell measures the actuator output torque. Therefore, this test translates the actuator's response to normal operating conditions.
In order to calibrate the motor load and the torque switch rotation to the actuator output torque, the torque brake is used respectively, in conjunction with:

- a torque switch transducer which is mounted directly to the torque switch mechanism and measures the torque switch rotation in both open and close directions. Normally, this rotation translates the linear movement of the spring pack when the actuator is subject to a load.

- and a motor load measurement device which monitors the voltage of each motor phase and the motor current to yield the power consumed by the motor.

Both calibration traces make up the fundamental basis of the diagnostic method. They predict eventual degradation of the actuator.

The actuation of the actuator's control switches is normally monitored together with the other parameters. This allows to determine the valve stroke time and the torque and limit switches trip values.

**Full stroke test**

After the calibration data have been registered, the actuator is replaced on the valve. In Doel, the full stroke test is performed locally with monitoring of the torque switch rotation, the motor load and the control switch actuation. In Tihange, only the motor load and the motor switch-off are registered from the MCC during this test.

Once the actuator has been correctly calibrated and the entire MOV has been full stroke tested without any degradation, the full stroke data together with the calibration traces will serve as baseline test data for the future diagnostic tests.

**Periodic diagnosis**

In Belgium, the monitoring of the motor load from the motor control centre has been adopted as MOV trending tool. The motor switch-off is also registered.

**3.4. Maintenance policy**

In order to respond to the requirements of the Generic Letter 89-10, the utilities of Doel and Tihange have decided the main goals of their maintenance policy for all safety-related MOVs.

After the design-basis review (DBR), the switches were correctly set and verified with a test bank.

In most cases, full stroke tests were difficult to perform under design-basis conditions. The correct settings have been justified by the DBR calculations led by the architect engineer, TRACTEBEL.

Finally, full stroke tests are periodically performed under the same normal operating conditions in order to follow the behaviour of the MOV and to confirm its operability.
3.4.1. Preliminary actions

The first stage was the revaluation of the MOV design-basis carried out by the architect engineer. This study has led to the modification of several torque switch setting values and even, in some cases, to changes of actuator sizing. Then, the utilities have built a database with all the critical characteristics of the actuators.

3.4.2. Torque switch settings after design-basis review

The initial setting has to meet a double goal:

- each MOV setting is adequate with respect to the design-basis review.
- each MOV signature is evaluated to verify that the DBR settings allow correct opening and closing under normal operating conditions.

This work is carried out theoretically once for each MOV with the following stages:

- Inspection of the actuator status
- Torque switch trip settings and calibration of the actuator output torque on a test bank
- Full stroke test of the MOV either locally, with monitoring of actuator parameters: motor load, torque switch rotation, control switch actuation, or from the MCC, only with monitoring of the motor load. The pressure and flow in the circuit are also registered.

The motor load traces (opening/closing) serve as baseline signatures with which all the future monitored signatures of the MOV may be compared.

Particularly, in Doel, the MOVs are distributed in groups. The membership to a group depends on the following criteria:

<table>
<thead>
<tr>
<th>VALVE</th>
<th>ACTUATOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>manufacturer</td>
<td>actuator type</td>
</tr>
<tr>
<td>diameter</td>
<td>output speed</td>
</tr>
<tr>
<td>stroke time</td>
<td>motor type</td>
</tr>
<tr>
<td>gear box</td>
<td>close on TS/LS</td>
</tr>
</tbody>
</table>

In a same group, the motor load signatures of the different MOVs are practically comparable (fig.6). Therefore, for each group, a baseline signature can be defined. The group baseline signature is considered as the "ideal signature" of an identical MOV working in the same operating conditions.

The functional operability of the MOV is confirmed if no appreciable drifts are noted between its own signature and the baseline signature.
3.2.3. Periodic maintenance

In order to ensure and improve the reliability of the safety-related MOVs, the NPP operators have defined a maintenance policy including the three following aspects: prediction, prevention and correction.

Predictive maintenance

A diagnostic test from the motor control centre is performed on each MOV in the most representative operating conditions (if practicable). The frequency of this test is 3 years for Tihange and 5 years for Doel. In order to keep functional operability, the motor load signature has not to show drift in comparison with the baseline signature.

In Tihange, a sample of grease is taken from the actuator of 6 pilot-MOVs and analysed every 3 years. These pilot-MOVs will be chosen in different representative environments (temperature, radiation, ...).

Preventive maintenance

Every 5 years, the general status of each actuator is externally inspected.

In Doel, every 10 years, one actuator of each group is completely dismounted for control of the wear of the internal parts (worm, spring pack, gear,...). A calibration of this actuator is also performed with the MOVATS system. If any anomaly is noted during these periodic inspections, corrective actions are also performed on the other members of the same group.

In the future, additional preventive actions will be considered based on the experience from the present predictive and preventive maintenance program.

Corrective maintenance

In case of a MOV failure, two signatures are monitored, one before and one after repair. The first signature allows to orientate the investigation of the root causes and to decide the corrective action. The second must be evaluated according to the first and to the baseline signature in order to accept the functional availability of the MOV.

3.5. Results with the diagnostic methods

TIHANGE

Actuator's manufacturers: AUMA and JOUCOMATIC
Diagnostic method: SIEMENS

In 1992, the Tihange 2 NPP has first experimented the diagnostic method. 26 actuators have been calibrated on the test bank and afterwards, all the MOVs have been full stroke tested locally with monitoring of the key actuator parameters.

After this first experience, in order to determine the baseline data for future diagnostics, the utilities have decided to measure only the motor load from the motor control centre together with the calibration traces.
During these first tests, the following general observations have been made concerning the actuator's status:

- the torque switch setting scale is incompatible with the trip setpoint value recommended by the architect engineer.
- inability to set the recommended torque value with an acceptable accuracy.
- no reproducibility of the torque switch actuation.
- no linearity between the torque switch angular position and the opposite force imposed to the actuator.

In 1993, 87 calibrations and 110 diagnostics have been carried out during the outage. The results have still to be further examined by the maintenance department but are available in the MOV individual data sheet.

The other NPP's have also used the diagnostic system but it is too soon to observe relevant statistical findings about the MOV failures.

The utilities of Tihange are confident in the performances of the SIEMENS diagnostic system. Such a method will certainly become a complementary investigation tool for MOV maintenance.

However, the implementation of an efficient preventive maintenance policy requires the collection of numerous measurement, comparison and evaluation results. The utilities have intended to adapt the SIEMENS system in order to compare directly signatures on a same record (for example, before and after repair).

DOEL

Actuator's manufacturers: AUMA and LIMITORQUE
Diagnostic method: MOVATS

The pilot-units are the Doel 3 and 4 NPP's. Since 1988, all the safety-related MOVs are periodically diagnosed with the MOVATS system. Until now, more than 30 different anomalies have been observed on more than 600 MOVs (safety-related and not safety-related). In this synthesis, we have considered only those with a direct impact on the reliability of the valve operating.

The following statistical data come from the results of the campaign 1988-1991 (figures 2 and 3). They are only related to AUMA actuators.

a) Actuator design modifications

a.1) Spring pack star washers (figure 4)

Concerning the actuators type SA6 and SA12, and especially for those with a small output speed, it was noticed that, during the first test, the torque limit switch was actuated at a measured value higher than the initially preset value. During the successive tests, the measured trip values became more and more comparable to the preset value.
The Maintenance Department of Doel has solved this problem by equipping the affected MOVs with a worm type composed of an alternative succession of star and belleville washers. At present, the hypothesis following which the elasticity coefficient of the spring pack is modified by the viscosity of the grease trapped between the washers has been demonstrated.

63.2 % of the MOVs have been modified.

a.2) O-rings and seals

The EPDM material of these O-rings was not compatible with the grease used in the crankcase of the MOVs. The NBR material is now used.

58.3 % of the MOVs have been modified.

b) Torque switch settings

b.1) Torque switch scale calibration error

This problem has been found for 82.7 % of the actuators. The torque switch is linked to a graduated setting scale. The torque value preset on the scale did not correspond to the torque value measured with the MOVATS device. After the recommended torque value was set and verified with the MOVATS bank, the torque switch scale was calibrated so that the value indicated by the scale corresponds to the preset value.

b.2) Switch trip found too low

Switch trip measured too low at the opening and/or closing with respect to the recommended setting value. These values have been corrected for 35.5 % of the MOVs.

b.3) Switch trip found too high

Switch trip measured too high at the opening and/or closing with respect to the recommended setting value. These values have been corrected for 58.5 % of the MOVs.

c) Design-basis review

c.1) Torque requirement change

For 80 % of the MOVs, the initial recommended torque switch value has been modified due to the design-basis review.

c.2) Design change actuator/motor

The actuator and/or the motor have been changed because of the design-basis review for 10 % of the MOVs.

d) Failures causes (figure 3)

The analysis of the MOV failures has demonstrated obviously that the most affected element is the valve assembly (24 %). Because of its direct contact with the process, this component of the MOV is submitted to a lot of phenomena such as corrosion, mechanical wear, etc...
The valve seating and/or unseating is the predominant element (14.9%) of misfunctioning of the MOV. Another appreciable problem comes from the valve packing. Valve packing concerns were observed for 7.2% of the MOVs.

Consequently, it seems that more maintenance effort has to be directed towards the mechanical aspects of the valve than towards the components of the actuator. However, among the actuator failures, the torque switch seems also to be a weak component for 5.2% of the MOVs.

Examples of diagnostic signatures

The Maintenance Department of Doel has already collected many motor load signatures in different maintenance circumstances in order to link the signature with a well-defined MOV failure and to identify the typical failures.

Three diagnostic signatures are presented on the figure 5. These motor load traces have been monitored from the MCC in the following specific cases:

- postmaintenance test:
  This example (fig 5a) shows the difference of a MOV behaviour before and after a maintenance activity: a new packing has been replaced. The new adequate packing causes more friction than the old one.

- degradation determination:
  During an ASME XI testing or current operating conditions, the operators of the control room can observe some MOV misfunctioning problems when trying to operate the valve.

  The figure 5b shows that, after three attempts, the double gate valve finally succeeds to leave its seat position. It identifies clearly an unseating problem. A maintenance inspection confirms a mechanical degradation of both slides. After the corrective action, the signature has a normal behaviour

- static or dynamic testing:
  Different MOVs from the same group are compared under different pressure and flow conditions. Under no-pressure, no-flow conditions, the torque value is lower at the valve closing than under a differential pressure.

- MOV group:
  The figure 6 shows the similar behaviour of the motor load traces of different MOVs from the same group.

The developments with the MOVATS diagnostic system have provided to the utilities of Doel a more realistic trending tool in order to increase the functional operability of safety-related MOVs. In comparison with the pre-MOVATS period, when the predictive MOV maintenance was principally based on the ASME XI requirements, the diagnosis quality is now improved and allows to estimate the degree of the operational readiness.

However, the utilities should define and adopt more accurate criteria and tolerances for the analysis and the interpretation of the monitored signatures before confirming the availability of a MOV.
4. Conclusions

The Belgian utilities have responded efficiently on the whole to the first recommendations of IE Bulletin 85-03. Therefore, at the publication of GL 89-10, they have extended the majority of the recommended actions to all safety-related MOVs and even to others critical for operation. They have established a common maintenance strategy covering the important stages mentioned in the GL.

Through extended use of diagnostic methods, the most frequent incipient failures of MOVs will be recognised. This, in turn, will allow the optimisation of preventive maintenance, in order to avoid common-mode failures of critical MOVs.
Appendix 1

Calculation methods of the torque switch settings

A note issued by the architect engineer, TRACTEBEL, describes the methodology for the verification of the actuator sizing and the determination of the advised torque setting values.

a. Calculation of the actuator output torque ($C_{alc}$)

The used calculation methods depend on the valve type (parallel slide valve, globe valve, etc.). These methods may be prescribed by the manufacturer when the valve type is very particular. The stem valve thrust is the sum of the three well-known loads:

- the differential pressure load:
  - The average diameter of the seating contact zone ($= 0.5 \times (\text{minimum external diameter} + \text{maximum internal diameter})$) is taken to calculate the seat thrust.
  - The choice of a correct valve friction coefficient (seat/slide) is discussed very much in many countries. The following values have been used according to each case: $0.3 - 0.4 - 0.6$ (steam). Recent research seems to demonstrate that a value of 0.4 could be taken in each case (even for steam). This value is becoming an European standard value and is now adopted in Belgium.
  - Sometimes, supplementary loads have been added to consider the frictions between the stem and the slides (for blockable slides).

- the packing load:
  - Two complementary effects have been considered: the packing tightness preload and the decreasing fluid pressure applied on the packing.

- the stem rejection load:
  - For each valve type, this load has always been considered. This load factor helps the valve during an opening stroke and is opposite during a closing stroke.

The valve stem thrust is translated in an actuator output torque value $C_{alc}$ by means of:

- stem valve thread
- average thread radius
- thread angle
- friction coefficient ($= 0.15$, European standard)

b. Determination of the minimum required torque values ($C_{min}$)

From the value calculated in item a., the minimum required torque setting value is determined in relation to the closing switch logic.

b.1. End of closing on Position Limit Switch (L.S.)

The inaccuracy of the torque switches (series of belleville washers) and the hysteresis of the system have been considered. Therefore, the minimum recommended value $C_{min}$ is equal to the sum of the calculated value $C_{alc}$ and the torque switch tolerance. This tolerance is equivalent to 10% of the maximum value of the setting range and is based on the drift noted during the calibration tests.
b.2. End of closing on Torque Limit Switch (T.S.)

The value $C_{min}$ is chosen equal to the value $C_{calc}$. This choice is justified by preventing unnecessary loads on the internal pieces of the MOV (crashing of the belleville washers). This philosophy can ensure the operability but not necessarily the tightness of the MOV.

b.3. Start of opening on Torque Limit Switch (T.S.)

The torque limit switch is set for the opening at 1.5 times the closing value to ensure the opening function. Indeed, a supplementary force is needed to overcome the load resulting from the MOV inertia at the end of closing on T.S. This estimation can be modulated depending on the torque value available at 85% Un and the equipment characteristics in order to prevent a power inflation of the actuator sizing.

$$C_{min} \text{ (opening)} = 1.5 \times C_{calc} \text{ (closing)}$$

c. Verification of the actuator sizing

For the validation of a correct actuator sizing, two conditions are necessary:

1) $C_{min} < C_{gar} < C0.85 \ (*)$
   where $C_{gar}$ is the torque certified at 85% Un by the manufacturer.

2) the torque switch setting range must be situated between $C_{min}$ and $C_{gar}$.

(*) the calculation note introduces a value $C0.85$ that corresponds to the available torque at 85% Un measured during testing.

d. Setting of the recommended torque value ($C_{cons}$)

Theoretically, this value can be set anywhere between $C_{min}$ and $C0.85$.
In order to consider the operating conditions and the equipment wear, the architect engineer determines an optimum advised torque setting $C_{cons}$ that differs depending on the opening/closing switch logic:

d.1. End of opening/closing on L.S.

In this case, the T.S. must protect the MOV if the L.S. fails.
The architect engineer advises to set the torque switch at a high value, usually close to 85% of the certified value $C_{gar}$ with a maximum value of 2 times $C_{calc}$ to prevent deterioration of the internal pieces.

$$C_{cons} = 0.85 \times C_{gar} < 2 \times C_{calc}$$

d.2. End of closing on T.S.

Here, the T.S. is actuated at each valve closing stroke. Therefore, the stresses have to be minimised in the internal pieces of the MOV.
The architect engineer advises to set the T.S. at the value $C_{min} = C_{calc}$ augmented with the half of the T.S. tolerance (i.e. 5% of the torque switch setting range).

$$C_{cons} = C_{calc} + \frac{1}{2} \text{ of the TS tolerance} = C_{calc} + 0.05 \times \text{TS setting range}$$
BASIC PRINCIPLE OF OPERATION
### Kerncentrale Doel Units Three and Four

MOV Diagnostic Summary
February 1988 to June 1991

#### Itemization Of Findings

<table>
<thead>
<tr>
<th>Actuators tested</th>
<th>317</th>
<th>297</th>
<th>614</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Torque switch settings</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Switch trip found too low</td>
<td>36.6</td>
<td>35.0</td>
<td>35.0</td>
</tr>
<tr>
<td>Switch trip found too high</td>
<td>59.9</td>
<td>56.9</td>
<td>58.5</td>
</tr>
<tr>
<td>Torque switch scale calibration</td>
<td>82.7</td>
<td>82.8</td>
<td>82.7</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Actuator design modifications</strong></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Replace O-rings and/or seals</td>
<td>65.9</td>
<td>50.2</td>
<td>58.3</td>
</tr>
<tr>
<td>Spring pack star washers</td>
<td>65.3</td>
<td>60.9</td>
<td>63.2</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Design Basis Review</strong></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Control logic change</td>
<td>0.0</td>
<td>1.6</td>
<td>0.8</td>
</tr>
<tr>
<td>Design change actuator/motor</td>
<td>9.8</td>
<td>10.1</td>
<td>10.0</td>
</tr>
<tr>
<td>Torque requirement change</td>
<td>84.3</td>
<td>75.8</td>
<td>80.0</td>
</tr>
</tbody>
</table>

(diagmov2)
**Kerncentrale Doel Units Three and Four**  
MOV Diagnostic Summary  
February 1988 to June 1991

**Itemization of Failure Causes**

<table>
<thead>
<tr>
<th>Actuators tested :</th>
<th>317</th>
<th>297</th>
<th>614</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Actuator : electrical</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Motor</td>
<td>1.3</td>
<td>0.3</td>
<td>0.8</td>
</tr>
<tr>
<td>Wiring and cables</td>
<td>0.6</td>
<td>1.4</td>
<td>1.0</td>
</tr>
<tr>
<td>Total</td>
<td>1.9</td>
<td>1.7</td>
<td>1.8</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Actuator : mechanical</strong></th>
<th>%Unit3</th>
<th>%Unit4</th>
<th>% 3+4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manual operation</td>
<td>0.0</td>
<td>0.3</td>
<td>0.2</td>
</tr>
<tr>
<td>Grease</td>
<td>1.0</td>
<td>0.0</td>
<td>0.5</td>
</tr>
<tr>
<td>Limit switch</td>
<td>0.0</td>
<td>1.3</td>
<td>0.6</td>
</tr>
<tr>
<td>Bearings or gears</td>
<td>0.3</td>
<td>1.0</td>
<td>0.6</td>
</tr>
<tr>
<td>Torque switch</td>
<td>5.4</td>
<td>5.0</td>
<td>5.2</td>
</tr>
<tr>
<td>Total</td>
<td>6.6</td>
<td>7.8</td>
<td>7.1</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Valve assembly</strong></th>
<th>%Unit3</th>
<th>%Unit4</th>
<th>% 3+4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reduction gear</td>
<td>0.0</td>
<td>1.0</td>
<td>0.5</td>
</tr>
<tr>
<td>Stem</td>
<td>1.6</td>
<td>1.4</td>
<td>1.5</td>
</tr>
<tr>
<td>Valve packing concern</td>
<td>10.7</td>
<td>3.7</td>
<td>7.2</td>
</tr>
<tr>
<td>Valve seating and/or unseating</td>
<td>18.9</td>
<td>10.4</td>
<td>14.9</td>
</tr>
<tr>
<td>Total</td>
<td>30.6</td>
<td>16.2</td>
<td>24.0</td>
</tr>
</tbody>
</table>
An Auma spring pack prior to the star washer modification.

The SA6 and SA12 spring packs utilize star washers to prevent grease from interfering with spring pack compression.
Motor load traces can be compared with MOV's from the same family to determine a MOV's status.
Adoption of Automatic Diagnostic System for Electric Motor Operated Valves in Nuclear Power Plant

Kansai Electric Power Co. Inc.
Mutsuo TAKAI
Toa Valve Co., Ltd.
* Sadao TAKEDA
Yoshihisa MANABE

1. Outline

In Nuclear Power Plant, various kinds of valves are installed for system controlling, preserving the safety of plant, managing, etc., therefore evaluation on integrity of these valves is the most important assignments to secure safety and reliability of the plant, and it is one of the most important responsibilities for maintenance personnel at the Power Plant.

For this reason, in the Nuclear Power Plant, disassembly checking, appearance and dimensional inspections and operational performance test for the Motor Operated Valves (called as "MOVs") are conducted periodically at the plant outage, and the integrity of valve is confirmed in accordance with the procedure indicated in Fig. 1 in Japan.

Furthermore, demand for a rational management method is considerably increased recently to shorten periodic inspection time and to upgrade the utility rate of the plant.

To cope with these problems and trends in recent years, it is required that both technique for valve maintenance and management are reconfirmed and reevaluated.

The research for newly developed measuring technology of instrumentation, the data processing technology and the applying technology have been positively conducted in all fields of industries. The applicable scope of diagnosis technology has been enlarged.
Currently manual evaluation technique

Sound
Acoustically comprehensive judgment

Current
Measurement by ammeter

Lift
Measurement by scale, scale calipers, etc.

Output TRQ
Uncheck

SW signal
Sequence check and confirming operation

Measurement and record preparation by technical engineers with high technology and abundant experiences

Contents of operation
Long period operation in the dense atmosphere Many operators are required.

Inspection method
Many points are relied on skilled operator's sense.

Judging method
Visual, appearance, indicating lamp, etc. Quantitative evaluation are difficult to be made.

Management method to be expected
- Upgrade of reliability Effective maintenance management of latent phenomena by amplifying sampled data in the engineering safety line valves.
- Upgrade of operation Mechanization of inspection method and reduction of inspecting time.

Fig.-1 Maintenance Management by Currently Manual
To upgrade reliability and operability of valve maintenance in the background of technological innovation, Automatic Diagnostic System shown in Fig. 2 for MOVs has been developed by Toa Valve Co., Ltd., under the system conception in which personal computer is suitably employed for diagnosing.

This system (called as "TACS") is highly sophisticated, and used as a performance test device for MOVs. All data collection necessary for diagnosing is automated, human error is always prevented, maintenance management is also unified and checking history is managed. The basic concept on TACS system is that everybody can use this system without any special engineering knowledge. This TACS system has already been applied for Japanese nuclear power plant as trial use. Its function and status of operation at Power Plant are as described hereunder.

2. Outline of TACS System

Data collectable through the visual and dimensional inspections after disassembling the valve, and through operational performance test after final assembly is not enough to manage the MOVs at the present. Therefore, it is hard to say that quantitative evaluation is necessarily done.

To resolve these problems, lots of data should be collected, whereas it is the most desirable that the data is evaluated comprehensively. As the resolution of this problem, Automatic Diagnostic System enabling to collect data automatically and to process immediately has been considered as desirable to develop in combination of personal computer with various types of high precision sensors. The TACS system diagnosis system flow chart is shown in Fig. 3.
1 Valve Lift Sensor
2 Torque SW Contact Sensor
3 Limit SW Contact Sensor
4 Torque Bypass SW Contact Sensor
5 Motor Current Sensor
6 Torque SW Displacement Angle Sensor
7 Solid Sound Sensor

Diagram: Diagnosis Computer Unit
Data Collection Unit

Evaluation technique by Automatic Diagnostic Device for MOV

- Solid sound sensor
- Motor current sensor
- Valve lift sensor
- Torque SW displacement angle sensor
- Torque Limit By-pass SW contact sensor
- Detection of actuator's vibration
- Detection of motor current
- Detection of stem movement
- Detection of actuator output torque
- Detection of each SW's ON-OFF

Analysis by personal computer

- Diagnosis
- Graphic display
- Record of data
- Data management Trend management

Contents of operation
Possible to fix sensor without disassembling the valve to be examined. Skilled operator not required, and capable of remote controlling valve to be examined from far distance.

Inspection method
If the valve to be examined to which sensors are fixed is opened and closed, the data can be collected automatically without human's hands.

Judging method
Detected data can be evaluated by computer automatically and in a short time. Exclusion of human's sensual evaluation.

Management method
- Upgrade of reliability
  Dynamic data can be collected, and detailed evaluation can be made effectively.
- Upgrade of operation
  Data can be accumulated and controlled automatically, so data maintaining management can be reasonably made.

Fig.-2 Management Method by Automatic Diagnostic System
Fig.-3 Diagnostic system flow
Managing the integrity of MOVs, this TACS system seizes the indispensable physical quantity as much as possible by sensors, and these are processed by personal computer. The integrity is checked comprehensively after comparing various data of the design basis and diagnosing abnormality. Moreover, each valve has its floppy disc in which valve life data is appended so that aging degradation can be grasped.

3. Composition of TACS System

The maintenance management method of the valve conventional inspection was systematically analyzed. For the purpose of solving the current various problems known through the analysis, automation by making use of sensors and personal computer was investigated. The composition of system is shown in Fig. 4. This system consists of the following three subsystems. And, output torque diagnosis shown in Fig. 5 evaluates torque switch settings.

1) Detection Subsystem

The current maintenance management method is such that sound, vibration, motor current, stroke, switch signal, etc. are confirmed by the five senses of human and measured by usual instruments. The detecting element at measured location is investigated individually. Special sensors were developed for each purpose. Seven types of sensors for normal diagnosis are used in the TACS system. Additional one is only used when measuring output torque out of actuator.
Fig. 4 Composition of Automatic Diagnostic System
Fig. 5  System composition when diagnosing output torque
2) Data collecting subsystem

Data collecting system consists of amplifying amp. through an insulation amplifier and A/D converter which converts to specified transmitting signal.

3) Diagnostic computer subsystem

This subsystem is a nucleus of system for processing the following items.

1) To analyze the data converted from the data collecting subsystem
2) To compare the standard values of preset diagnosis items and collected data.
3) To diagnose fault mode and degradation mode.

4. Specification of System

The system consists of seven types of sensors, data collecting unit and diagnostic computer unit. Sensors can be easily attached to Motor Operator, even if the operator wears protective clothes against radiation. The only one thing to do is that the stem cover of actuator and limit switch shall be removed.

After attaching sensors, opening and closing the valve by motor actuator only once, dynamic data can be collected automatically into data collecting subsystem. Analyzed evaluation of collected data is to be processed referring to judgment of fault modes of 21 items, and performing history management and tendency monitoring for collected data as shown in Fig. 6.

Sample display of diagnosed result of MOVs

List of diagnosed result for each diagnosed item is displayed.
Fig. - 6 Automatic Judgment of Valve Fault Modes
Detailed display of diagnosed result Setting of Torque Bypass SW

This example indicates detailed diagnosed results for adjustment of torque by-pass switch provided to preventive measures against malfunction of valve. With this example, since the position of disc parting from valve seat could not be measured earlier, the management was performed by only L4 set position, but now the position of disc parting from valve seat, L3 is detected, and the management value is diagnosed whether the value, L4 minus L3 (L4-L3), satisfies 5.2 mm or not.

5. Reliability of System

Reliability assessments of the TACS system were established based on diagnosis principle from the relations among inspection specifications required for design basis as a valve manufacturer, and the fault mode and physical mode. The reliability of the TACS system was evaluated in accordance with the block diagram in Fig. 7.

5.1 Judgment principle

The abnormality judgement logic was considered as the relation between the fault of each diagnosis item and physical mode that was evaluated based on the past fault modes.

Each diagnosis item is complicated because judgment logic is different from each other in valve type, nominal pressure, nominal size, etc. The logic of the various abnormality judgment is confirmed to be appropriate by generating each simulating fault modes on valve.
Reliability of MOV Function Abnormality Diagnostic System

Judging Principle
Relation between fault modes and physical mode for each diagnostic item is searched and judging principle for automation is set. Current management method is also automated.

Measurement Precision
The precision of sensor shall be equivalent to or more than the precision of measuring instrument used currently for inspection and inspection precision required for design.

Adaptability to actual unit
The system shall be adaptable as functional inspection device for MOV.

Validity of logic for abnormality judgment
Precision of sensor
Precision of system
Work-ability
Operation-ability

Fig.-7 Function Reliability Evaluation Block Figure
5.2 Precision of Measurement

Precision of measurement of each sensor is equivalent to or better than the precision of measuring instruments used for present inspection. Moreover, precision of measurement in the each sensor was confirmed to meet the precision of measuring under the load of standard calibration value on each sensor. In addition, in the measurement in combination with valve, it was confirmed that significant difference is not occurred when comparing with measured value by the present measuring method.

5.3 Adaptability to Actual Unit

Adaptability to the valves being used at actual plants was evaluated when using at the time of periodic inspection at Nuclear Power Plant. Tests were conducted in two periodic inspections at nine plants, the numbers of tested valves reached over 500 sets in all.

1) Workability

Many improvements are adopted for operator such as easy fixing and removing of sensors, compactness of sensor, and simplification and light weight of sensor because working conditions are very hard in the containing vessel. Moreover, in the diagnosis for actuator with DC motor, surge noise which occurs at the time of switching ON and OFF may effect on diagnosed data, so that an insulation amplifier is provided for anti-noise measures. These many improvements were implemented in the course of investigation for adaptability evaluation to the actual plant. After improved, it is concluded that these improvements do not affect detrimentally on the system and valve functioning.
2) Operability

Automatic detecting software used for detection of diagnosis logic such as starting point of valve stem movement and hammer blow occurring point in actuator is correctly recognized for many valves. However, for specific valves there were cases where the software did not recognize the hammer blow sounds in actuator due to the factors of valve's inherent difference.

For improving the problems, the method of recognizing the hammer blow sounds was combined with the use of improved automatic detecting software and addition of manual detecting method. Also, the operability is upgraded for preventing misfixing the sensors by operator and developed software avoiding human error. The operators evaluated that the operation of the TACS system could be mastered for a couple of days's indoctrination and for about one week's training for diagnosing the actual unit.
6. Expected Effect by Adoption of the TACS System

The TACS system can not detect seat leakage, gland leakage, grease degradation, oil leakage and loosening of terminal’s set screws at actuator portion which are checked by the present checking operation. For this reason, the effect obtained by adoption of the TACS system is the most effective in case where the system is used in combination with the present visual appearance inspection and disassembling inspection at the time of periodic inspection and then the effect shown in Fig. 8 can be expected.

The application method of the actual plant can be considered as follows.

(1) The TACS system is applied at the time of functional test during the periodic inspection and data are suitably collected, then integrity is confirmed.

(2) The TACS system is applied before checking for MOVs at the time of plant outage or surveillance test, and the checking plans for valve is rationally fixed. However, this method is expected to cause limitation from processing management of periodic inspection at plant outage and when surveillance testing, fixing of sensor is hot-line work so that it greatly risks safety of human body and plant operation. Therefore application of the TACS system is mostly effective at the time of functional test during the periodic inspection at plant outage. After adoption, the accumulated data can be used as additional preventive maintenance data.
Automatic Diagnostic System collects dynamic data, and is capable of high maintenance management.

Upgrade of visual checking conducted by conventional technique is limited.

Maximum effect in the context of combining with conventional technique.

Fig.-8 MOVs Automatic Diagnostic System Adoption
1. Dynamic data can be collected in no time and the maintenance management data is expanded.

2. By accumulating data, utilization as preventive maintenance device is expected.

3. Fault modes which can not be found through only disassembling check is expected to detect beforehand, and rational checking program can be expected to form.

4. Prevention of human error by disassembling check for valve, checking management value and the maintenance management can be unified, and rational management can be made.

5. By adoption of this system, maintaining and securing reliability of MOVs can be expected for long time without disassembling actuator by also applying conventional inspection methods.

7. **Utilizing Status at Power Plant**

Utilizing status at Power Plant is shown in Fig. 9. Data at normal time of each valve and standard value of diagnosed judgment are recorded in the floppy disc. At the time of periodic inspection, detection data, called present status data, picked up from the valve to be diagnosed at site are recorded automatically in the floppy disc as the present status data. Simultaneously, the data are compared instantly with standard value of diagnosed judgment in the floppy disc for diagnosis and diagnosis is conducted for abnormality. If detecting abnormal spots, repairing will be conducted at once.

The present status data recorded in the floppy disc is brought back to the office to record in optical disc. In such a way, the trend management of various data in each valve can be made.
Diagnostic operation at Power Plant

Automatic Diagnostic Device for MOVs

Detected data by sensor

M

Valve to be diagnosed

F.D. for diagnosis F.D. for the present status data

Result to be expected

1) Upgrade of limit SW adjusting precision
2) Upgrade of torque value management precision
3) Improvement of aging degradation
4) Human error prevention when disassembly checking

* Diagnosis in function test at the time of periodic inspection

Diagnosis data and MOVs maintenance management work in office

Data management device

Personal computer

Optical disk unit

Diagnostic data recording clear unit

Optical disk unit

The present status data recording clear unit

Printer

F.D. for diagnosis F.D. for the present status data

Personal computer body

Data base process

(Diagnosis data file)
- Diagnosis data
- Retrieving information for diagnosis result
- Retrieving information for specifications of valve and actuator
- Information for preventive maintenance data
- Information for check record
- Information for diagnosis evaluation standard

(The present status data file)
- Information for the present status data

Results to be expected

The system is able to be used as a monitor of data's aging change and back up data of life extension based on the time of the device introduction. Periodic inspection data and valve information was managed by system, so it can be retrieved in no time.

Fig.-9 Introduction Form of Automatic Diagnostic System at Power Plant
By adopting TACS system, the following advantages are expected.

- Adjusting management of limit switch and torque switch required for mechanical skill can be done easily.

- Hitherto, inspection equivalent to the items examined or inspected during the periodic inspection can be conducted by this system, therefore frequency of disassembling check on actuator is reconsidered.

(Table-1)

- Prevention of human error such as mistaken wiring at the time of reassembling, and in addition, data picked out can be accumulated, and tendency of aging degradation can be easily grasped.

8. Conclusion

In the operation of Plant, quality of maintenance plays a very important role to secure reliability and safety which are required for the valves at Nuclear Power Plant. At the time of periodic inspection, lots of valves are repaired and maintained, and various data are collected and controlled to prevent accidents from occurring beforehand. We introduced the Automatic Diagnostic System for MOVs with which the effect of preventive maintenance judged on the basis of data, and rationalized operation can be expected. It is believed that many effects can duly be expected and it contributes upgrading of the utilization rate of Power Plant.
Table 1: Example for Review of Disassembly checking Cycle

<table>
<thead>
<tr>
<th>Item</th>
<th>Year</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>Actuator (Important valve)</td>
<td>Before intro.</td>
<td>Dis</td>
</tr>
<tr>
<td></td>
<td>After intro.</td>
<td>Dis</td>
</tr>
<tr>
<td>Actuator (General valve)</td>
<td>Before intro.</td>
<td>Dis</td>
</tr>
<tr>
<td></td>
<td>After intro.</td>
<td>Dis</td>
</tr>
<tr>
<td>Valve body</td>
<td>Before intro.</td>
<td>Dis</td>
</tr>
<tr>
<td></td>
<td>After intro.</td>
<td>Dis</td>
</tr>
</tbody>
</table>

Legend

Dis.: Disassembly checking

T: Automatic diagnostic system and operation confirmation

O: Check for current status visually and by confirming operation
1. Introduction

The way of thinking of preventive maintenance is adopted for nuclear plant maintenance exhaustively in Japan. It requires us to have a plant outage for two or three months in approximately one year interval, in which we inspect or repair the plant components.

In this paper, I am introducing a series of processes for complete restoration of MOV operating function. We adopt it when we overhaul MOVs in TOMARI power station, which is owned and operated by HOKKAIDO Electric Power Company Inc. TOMARI consists of two nuclear power units and is located in the most northern part of Japan.

Furthermore, probably it results in that I also describe some of the standard methods for restoration of MOVs performed in Japanese nuclear plants.

Fig-1 shows the operational history of TOMARI unit 1 since its commissioning. It is also a standard pattern of nuclear power plant operation in Japan.
During maintenance outages we carry out periodical inspections of the plant. A lot of components in the plant are maintained then. MOVs are also inspected in various levels, from the simple function confirmation to the regular overhauling. And now it goes without saying that it is very important that the function which is required to operate a MOV is restored completely after its inspection. Especially, we have to take care of their restoration when we decompose MOVs, because we cut off a lot of their circuit wires and remove their parts like limit and torque switches then. If their functions are lost by their inspection, it’s just the other way round.

Thus, when we inspect MOVs, especially with decomposition, it is necessary to establish the precise method which guarantees complete restoration of MOV’s operating function after inspection.

The method adopted in TOMARI is described as follows.

2. The Processes for the Restoration of MOVs’ Actuator Function

2 - 1 The Flow of the Regular Inspection of MOVs

We have about 130 MOVs in primary side, and about 70 in secondary side in each unit in TOMARI. And most of them have Limitorque type actuators. So I am going to describe the processes about this type.

For MOVs in TOMARI we carry out the simple inspection in 1-3 year interval, in which we confirm the valve’s operating function without decomposing it. Furthermore, we do the regular one in 6-10 year interval, in which we decompose both actuator and the valve itself and overhaul then. We decide the frequency of inspections according to the importance of the valves. Especially, we apply the shortest intervals for the main systems of primary-side loop (RCS, CVCS, RRHS) and safety systems (SIS, CSS). Fig-2 shows the history of the number of the MOVs which have been inspected in TOMARI unit 1.

<table>
<thead>
<tr>
<th>Number of MOVs Inspected in the Outage</th>
<th>1st Outage</th>
<th>2nd Outage</th>
<th>3rd Outage</th>
<th>4th Outage</th>
</tr>
</thead>
<tbody>
<tr>
<td>without Decomposition</td>
<td>73 / 15</td>
<td>93 / 16</td>
<td>48 / 15</td>
<td>55 / 15</td>
</tr>
<tr>
<td>with Decomposition</td>
<td>21 / 8</td>
<td>15 / 16</td>
<td>28 / 15</td>
<td>30 / 7</td>
</tr>
</tbody>
</table>

*Primary Side*

*Secondary Side*

Fig-2 MOV Inspection History in TOMARI Unit 1

640
The standard flow of the MOV's regular inspection in TOMARI power station is as follows.

(1) Neggering before the inspection
(2) Data measurement about the valve function before the inspection.
(3) Cutting off the circuit wires
(4) Removing the actuator from the valve
(5) Overhauling of the valve itself and the actuator
(6) Setting up the actuator to the valve
(7) Connecting the circuit wires
(8) Neggering after the inspection
(9) Data measurement after the inspection and sequence checking

In the above flow steps, (1), (2), (8) and (9) are performed only for the purpose of confirming that there are not any differences of valve operating functions between before and after the inspection. Besides, when we perform step (3) and (7) above, we must take good care that those are done accurately in order to restore the valve function.

In the following clauses, I am describing the methods to guarantee complete restoration of the MOV's operating function we apply in those steps.

2 - 2 The treatment for cutting off and restoring electrical circuits

When we remove an actuator from an MOV itself for the purpose of decomposing it, it is necessary to cut off all the circuit wires which are laid between the actuator and the control area. Fig-3 shows the points we need to cut off before removing the actuator from the valve body. When this work is done, it is essential that we prepare for restoring those circuits correctly.

There seem to be some ways to do that. And we use the standard format sheets whose example is shown in Fig-4. This type of sheets is applied for all the works concerning to cutting off and restoring electrical circuits in our plant.

As Fig-4 shows, we fill the following items in the list in advance when we use the sheets.

(1) The tag number of the valve
(2) The ID (identification) numbers of the wires which we cut off
(3) The ID numbers of the terminals to which these wires are connected
(4) The treatments (lifting or jumping) we should do.

These entries are made according to the sequence diagrams. Then we sign on the sheets when we cut off the circuits and restore them.

By taking these steps, we can prevent this work from forgetting to restore the
circuits or restoring them incorrectly.

Besides, we always attach yellow tags to the wires we cut off. On these tags, we note ID numbers of the wires and the terminals, the name of the work, and the name of responsibility person. These ID numbers are described also on the wires and the terminals themselves, so the above tags are not necessarily needed to attach. However, we can easily distinguish the points we cut off in the field by doing that. Moreover, it becomes evident that the wires are cut off through the formal procedures.

![Diagram]

**Fig-3** Cutting off and Winding Points in a Circuit

**Fig-4** An Example of a Checking Sheet
2 - 3 Meggering

In regular inspection of MOVs, we pull circuit wire cables out of an actuator, or re-lay the wires in an actuator. Thus, we need to confirm that there is not any short circuit or ground-fault after restoring it.

In TOMARI we confirm the isolation to the earth of the circuits before cutting off the circuit wires and after restoring them. We measure resistance between the circuits which include the points we cut off and the earth by using of a Megger. Fig-3 shows the points where we make measures then. Meggering is performed in a lump in the valve control center, in which the power sources and relays are concentrated.

We had some experiences to find that the isolation of the circuits was broken because of inappropriate works.

2 - 4 Data measurement before and after decomposition

Valve strokes, driving times and the other parameters can be changed during regular inspection of a MOV, because we fit or cut the valve plug, or we decompose and restore parts in an actuator then.

Therefore, in TOMARI, we measure these data before decomposition and after restoration. Moreover, we fit the values to the former data after restoration if we can adjust them. For example, limit switch setting points. Finally, we confirm that there is not any remarkable difference between the former and the latter. Of course we also confirm whether the data satisfy their criteria or not. The data we measure in MOV inspection are, in the concrete, as follows.

\[ E = D + a \]

(Motor Drive Stroke)

\[ a : \text{Over Run} \]

\[ E \text{ (Full Stroke)} \]

Fig-5 Data Measurement Points about Limit Switches
(1) The insulation resistance of the circuits we mentioned in previous clause
(2) Opening and closing time of the valve
(3) Motor Driving current
(4) Motor Driving strokes
(5) Limit switch setting points which are shown in Fig-5.

When we measure the data shown in Fig-5, we operate the valve by hand. Then we
detect the position of top of the valve stem by using of a depth gage. To know the
point in which the limit switch works, we watch the rotor of the switch turn round
with our own eyes.

Of course we also do confirm that each part which compose the valve works without
unusual sound, vibration, and any other wrong matters.

On the other hand, we cannot measure torque which acts on a valve unless we use
a special instrument. Torque is one of the important parameters about MOVs. Now,
concerning this parameter, we get the data of some numbers of MOVs we choose in
each outage. When we measure them, we use an automatic diagnostic instrument
system and detect turning angle of a torque switch.

This system was developed by a valve vendor. It has functions to measure a lot of
parameters of a MOV including torque all at once. However, we don not apply it to
all the MOVs because it consists of a lot of parts and cables and takes rather
long time to set up or move.

Data measurement and confirmation of the valve operating function described in
this clause are performed not only in regular inspection (with decomposition) but
also in simple one (without decomposition).

2-5 Sequence Checking

We cut off lots of circuit wires and remove some parts like limit and torque
switches from an actuator body in regular inspection.

So we carry out "Sequence checking" in order to confirm totally whether the
circuits was restored and functions correctly. We carry out it only when we do
regular inspection of an MOV.

Fig-6 shows main area of standard control circuits of an MOV. And Fig-7 shows the
processes of sequence checking of the circuits shown in Fig-6.

As preparation of this work, we do the following steps.

(1) Three people take up their positions. Namely, the main control room to
operate valve control switches, the control center to confirm relays' action,
and the field to lift and jump the circuits.
(2) We prepare portable telephones to be able to keep talking each other.
(3) We make the circuits' condition that power source of an actuator is dead,
and only control circuits is alive.
After above preparation, we do sequence checking by repeating following steps.

(1) We make a condition which simulate the action of a limit or torque switch by lifting or jumping the circuits.

(2) We confirm that the suitable relay work and the suitable lamp is lighted correctly according to the operation of the control switch.

---

**Fig-6 Main Area of Standard control circuits of a MOV**

<table>
<thead>
<tr>
<th>Operation</th>
<th>Reaction of the Relay</th>
<th>Confirmation line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Driving the Valve to the Middle Position (by hand)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Changing the Circuit to the test mode (Normal/Off/Testing)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operating the Opening Switch</td>
<td>Action of “O” Relay</td>
<td>Function of “O” Relay</td>
</tr>
<tr>
<td>Operating the Closing Switch</td>
<td>No Change (“O” Relay’s Self-holding)</td>
<td></td>
</tr>
<tr>
<td>Lifting the Over-side Limit Switch “L1”</td>
<td>Returning of “O” Relay</td>
<td></td>
</tr>
<tr>
<td>Retuning “L1”</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operating the Opening Switch</td>
<td>Action of “O” Relay</td>
<td></td>
</tr>
<tr>
<td>Opening the Torque Switch “T1”</td>
<td>Returning of “O” Relay</td>
<td></td>
</tr>
<tr>
<td>Retuning “T1”</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operating the Closing Switch</td>
<td>Action of “C” Relay</td>
<td>Function of the Close-side Limit Switch “L1”</td>
</tr>
<tr>
<td>Lifting the Over-side Limit Switch “L2”</td>
<td>Returning of “C” Relay</td>
<td></td>
</tr>
<tr>
<td>Retuning “L2”</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operating the Closing switch</td>
<td>Action of “C” Relay</td>
<td></td>
</tr>
<tr>
<td>Lifting the Torque switch “T2”</td>
<td>Returning of “C” Relay</td>
<td></td>
</tr>
<tr>
<td>Retuning “T2”</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Driving from “J1” to “J2” (Operation of the Torque switch “T1” action)</td>
<td>Action of “T” Relay</td>
<td>Function of the Over-side Limit Switch “L1”</td>
</tr>
<tr>
<td>Lifting the Over-side Limit switch “L1”</td>
<td>Returning of “T” Relay</td>
<td></td>
</tr>
<tr>
<td>Retuning “L1”</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lifting the Close-side Limit switch “L2”</td>
<td>Action of “T” Relay</td>
<td></td>
</tr>
<tr>
<td>Retuning “L2”</td>
<td>Returning of “T” Relay</td>
<td></td>
</tr>
<tr>
<td>Reversing the Jumping Wire “J1-J2”</td>
<td>Returning of “T” Relay</td>
<td></td>
</tr>
<tr>
<td>Jumping from “J1” to “J2” (Operation of the Torque switch “T1” action)</td>
<td>Action of “T” Relay</td>
<td></td>
</tr>
<tr>
<td>Lifting the Close-side Limit switch “L2”</td>
<td>Returning of “T” Relay</td>
<td></td>
</tr>
<tr>
<td>Retuning “C”</td>
<td>Action of “T” Relay</td>
<td></td>
</tr>
<tr>
<td>Reversing the Jumping Wire “J1-J2”</td>
<td>Returning of “T” Relay</td>
<td></td>
</tr>
<tr>
<td>Changing the circuits to the normal mode</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

---

**Fig-7 The Processes Sequence Checking**
Furthermore, after we finish sequence checking, we restore power source and confirm real action of the valve. When the valve is being put in a logic of safety injection or C/V isolation systems, we confirm that the valve moves correctly according to SI or C/V isolating signal.

3. Conclusion

The processes I have described above are, individually, not necessarily advanced methods, or rather they can be conservative and time consuming ones. As we know, inspection of a MOV with decomposition takes a long time and lots of money, but we carry out regular inspection for 30-50 MOVs during a plant outage from the viewpoint of preventive maintenance. And we apply these restoring processes for all the MOVs we decompose.

Maybe this way of MOV maintenance is excessive for only securing valve operating function.

But it is one fact that we have been operating TOMARI unit 1 for 5 years and unit 2 for 3 years since their commissioning, and we have never experienced a trouble concerning MOVs.

Thus, we think that there are something useful in our processes for foreign plants.
MOTOR OPERATED VALVE ISSUES

Functioning, Leak Tightness, Maintenance of Gate Valves and Globe Valves in Nucl. Power Stations

Expert Experience from 5 Operating NPPs (PWR and BWR) in SW Germany

by

W. RIEGER

Presentation Paris, April 25th-27th 1994
Summary

The requirements for safety relevant wedge gate valves and globe valves in NPPs are generally known, but are briefly stated. The operating experience of the currently operating NPPs shows that the requirements for a safe function are not fulfilled during all operating and test conditions.

Using several examples of gate valves and globe valves it is shown that there is a wide range of causes for functional failures and how specific remedial measures can be applied.

A number of different technical corrective activities is presented.

For a DN 500/600 steam line wedge gate valve the specific causes for a shaving action of the disk are established and it is presented how design improvements can be applied.

Additional sensitive areas, which are related to the function of gate valves, are mentioned. These are used to establish some general technical demands and requirements for the use of safety relevant valves.

Substantial maintenance activities are necessary for both gate and globe valves to guarantee a safe function during operation.

Several established maintenance activities are explained and discussed.

In the past it was not possible to eliminate all shaving related deficiencies of gate and globe valves, because the old design could not fulfill all new requirements. It is shown how newly developed and just built gate valves are designed to avoid these deficiencies.

As a final conclusion some demands are presented, which are currently discussed by experts in (SW) Germany as an outlook on valves built in the future.
Requirements for Safety Relevant Wedge Gate Valves in nuclear power stations

1. Integrity
   - no destruction/no leakage of pressure retaining components

2. Functionability
   - close safely
   - open safely
   - keep closed
   - guarantee the leak tightness in the seat region
   - still in good function after one year in closed position

Function
- at reduced motor voltage (0,8 x U)
- at multiple operation
- at maximum differential pressure Δp
- at 5 bar external pressure steam atmosphere at 140°C during 1 hour in the containment
- after earthquake, after aircraft crash
Examples for Malfunction

Wedge gate valve in the volume control system
(demineralized Water + 4% boric acid)

Motor

Trouble:
fretting valvestem/ stemnut

Cause:
angle of screw thread between nut and stem was not in accordance with standard (manufacturing fault)

DN 150
Example: Malfunction of Wedge Gate Valves

Trouble:
Closing problems at the beginning of the flow interruption (after 95% of the lift) and at the maximum of differential pressure $\Delta p = 110$ bar

Cause:
Unfavourable tolerances causing a bracing of the right disc by plastification areas at the deformed region
Example: Malfunction of Wedge Gate Valves

Trouble:
At a maximum of $\Delta p = 15$ bar (test state) the valve doesn't open and doesn't reach the end position in the closing direction.

Cause:
The increased friction values between stem nut made of mild steel CK35N in combination with the valve stem made of stainless steel 1.4571
Main Steam Wedge Gate Valve

First design of the disc

Changed design of the disc

Spade disc

Changes:
- more seat area for overlapping
- small deflection of the disc
- acceptable radius $R$
- nitrided seat areas
Delicate Sectors of Wedge Gate Valves with Influence on Function

- minimum of overlapping between the gliding surfaces
- tilting margin of the disks
- material of the gliding surfaces
- the outline, that means the radius of the gliding surface edges
- small holes for pressure equalizing or pressure protection devices
- sufficient stiffness of the casing
- possibility of mechanical reworking of the gliding surfaces
- no tight bracing of the discs
  use of bellows or spring plates
- stem/nut-bearing with mechanical shock absorber
Preconditions, Minimum Requirements for the Application of Functioning Wedge Gate Valves

- Test / operation results of function with \( \Delta p \) are required
- Run the wedge gate valves in the closed position only via position stop
- registration of the torque thrust
- Inservice inspections, visual inspection of the gliding surfaces registration of the torque thrust as a function of the lift at the end of every maintenance
- A good spare part depot for wearparts
- Take notice of permissible operating modes
  E.g. for use of the switch off mode "bypass of torque switch"
  the wedge gate valve must be designed for the loads of switch off failure in level B
2 Examples out of maintenance instructions
Volume: 15 - 30 pages for every valve

Control of function

The final test is the "active power measurement" beginning from the endpositions of the valves from "open" to "closed" and from "closed" to the "open" position.

The controlling signal is carried out from the shift supervisor's desk, the measurement in the switch gear building is done by active power measure device. Simultaneously at the valve the sense of rotation and the function of the torque switch are tested.

Determination of the opening torque thrusts:
The valves are closed by the temporary power supply. The opening torque thrusts are determined by the rotation equipment and by the stem thrust equipment.

Leak Tightness Test:
Tightness of the casing
The casing is pressurized with 2 bar of nitrogen and the tightness is checked. The valves are in the back seat position.

Leak Tightness of the Seats:
Every seat is pressurized with 88,3 bar demineralized water and the leak rate is determined by an over flow pipe at the next flange.
Example: KSB Optimized Wedge Gate Valves

- Fixing final position of disc. At switch off failure no additional loads on pressure retaining parts
- Separate cylinder for guide ribs demountable (patent pending)
- Spade disc (min. deflection)
- Stellite hardfaced guide ribs
Example: Sempell / Babcock
Optimized wedge gate valves

- Stellite-hard-faced sealing and guide ribs

- Changed (long) guide ribs, reworking is possible

- Fixing final position of disc

- Spade disc (min. deflection, no tilting)
Example: Sempell Optimized Globe Valve

- Bearing lubrication
- Necked down bolts designed for switch off failure and pressure
- Hardmaterial welded guide of the valve cone
Recommendations, Design Instructions
(Discussion in South-West Germany)

- Design of the valves including actuator according to switch off failure in level B

- Functioning test of valves which are relevant for function at maximum Δp after a valve- or actuator overhaul as far as possible in the plant registration of the torque thrust

- If the plant requirement is leak tightness for low pressures, no wedge gate valves, only globe valves, (for safety relevant valves)

- Proof of self locking for all valves

- Take only pressure protection devices which can be tested

- For wedge gate valves: determination of tilting backlash
MAINTENANCE OF SOLENOID OPERATED VALVES

by

Vic Varma

Nuclear Maintenance Applications Center (NMAC)

Background

Solenoid operated valves (SOV) are widely used in the nuclear power industry in the United States. Solenoid Valves are economical in their first cost and extremely easy to operate. Though some solenoid valves are used in selected hydraulic applications, by and large most solenoid valves in power plants are used in instrument air applications. Solenoid valves can be ac or dc powered and can be used to control fluid flow directly (in line application) or indirectly (as pilot controller). Utilities estimate that there may be as many as 1000 - 2500 solenoid operated valves in a typical light water reactor nuclear power station. Boiling Water Reactors (BWR) generally use more SOVs than Pressurized Water Reactors (PWR). The BWR plants have more hydraulically actuated systems that tend to use more SOVs. Solenoid operated valves are used both in safety and non-safety related systems of the plant.

Because of some reported failures of solenoid valves, US Nuclear Regulatory Commission conducted a study on the operating experience of these valves, which is published as NUREG 1275, vol. 6. Simultaneously, Nuclear Maintenance Applications Center (NMAC), operated by Electric Power Research Institute, organized an industry effort in the form of a utility Technical Advisory Group (TAG), to prepare a guide on solenoid valve maintenance and application. This guide has been published as EPRI Report NP-7414 and is available to all EPRI members.

Failure Analysis

In order to recommend certain maintenance actions it was necessary first to determine the various modes and mechanisms that could result in a solenoid valve failure. One source of this data for the nuclear industry is the Nuclear Plant Reliability Data System (NPRDS) maintained by the Institute of Nuclear Power Operations (INPO). NPRDS data was supplemented with actual maintenance data from various power plants. A number of utilities cooperated by providing their maintenance reports. This data, however, needed adjustment and detailed analysis to account for the fact that maintenance of solenoid valves is usually recorded under the primary equipment. For example, a solenoid air-pilot valve may not be identified individually but listed as part of the air operated valve on which it is mounted. Failure mechanisms and causes for air-pilot SOVs and process SOVs are shown in the charts (Figures 1 and 2) below. Electrical coil failures, degradation of seating surfaces and accumulation of debris/corrosion products are the major failure mechanisms, while wear/aging, contamination and human error were found to be the dominant failure causes.
MAINTENANCE OF SOLENOID OPERATED VALVES

Figure 1
SOV Failure Mechanisms

Figure 2
SOV Failure Causes

Technical Description

Simply stated a solenoid operated valve is actuated by energizing a solenoid coil with sufficient voltage. When the coil is energized, it produces a magnetic field to attract a plunger assembly. Depending on the mechanism, the plunger will open or close the valve attached to it. However, such direct acting solenoid valves can be economically manufactured only in small sizes and are designed for low pressure applications. For large SOVs or for high pressure systems, force developed by the solenoid coils is inadequate to operate the valves. Therefore, in such applications a piloted solenoid valve is required. Figure 3 is a simplified diagram of a piloted solenoid valve. When the main disc and the pilot valve are closed, the system inlet pressure provides the main disc seating force and tightly closes the valve,
assuming that outlet pressure is zero. When the pilot valve is opened, the main disc chamber rapidly depressurizes and the inlet pressure acting below the main disc unseats the valve. A small spring is often used to help alignment, seat the disc properly and provide operational stability. This spring force is generally not significant compared to the pressure forces acting on the main disc.

![Diagram of pilot solenoid valve]

Figure 3
Piloted Solenoid Valve

As shown in figure 3, most pilot valves require Minimum Operating Differential Pressure (MOPD) to operate reliably. If the MOPD falls below the specified minimum, the valve may not seat properly and leak internally. Also, piloted solenoid valves are uni-directional. If valve installation is inadvertently reversed, this type of valve will invariably leak. Flow direction on all solenoid valves is indicated either by an arrow or labeling IN and OUT (for 2-way valves), and by designating "pressure," "cylinder" and "exhaust" ports on a 3-way valve.

![Operating time variations graph]

Figure 4
Stroke Time Variations
(Pilot Assisted High Pressure SOV)
Another important operational characteristic of a piloted solenoid valve is the change in closing and opening times as the system pressure changes. This phenomenon, however, only occurs in fluid systems, not in gas or steam service. To open a piloted solenoid valve, the fluid in the main disc chamber must be displaced (see figure 3) before the main disc can open fully. The rate of this displacement is a function of system pressure and the orifice size. If the system pressure drops, opening time will be longer. Figure 4 shows some typical valve operating times at various system pressures from a valve manufacturer.

**Burping in Piloted SOVs**

When a piloted SOV is closed under normal conditions, the pressure in the main disc chamber will remain equal to the inlet pressure (see figure 3). However, certain transient conditions can increase the inlet pressure significantly before flow through the inlet orifice balances the main disc pressure. If the transient is rapid enough, the inlet pressure at the bottom of the disc can momentarily open the valve. This phenomenon is called "burping." It should be noted that burping does not occur if,

i) the valve design or application allows rapid equalization of the main disc chamber pressure with the inlet pressure,

ii) the process medium is incompressible fluid and no air or gas is present in the disc chamber.

Proper air venting from the system and orientation of valves to prevent entrapment of air can reduce possibility of burping.

**Valve Seats**

Valves can be hard or soft seated. Hard seated valves will have a metal-to-metal seating to close the valve port. Soft seat refers to the use of elastomer or plastic material at the seating surface. Soft seats tend to be more effective than hard seats in blocking the leakage path. However, use of soft seated valves is limited to temperatures below 350°F approximately. Leak tightness on hard seated valves is achieved by highly finished mating surfaces. These surfaces can easily be damaged by trapping contaminants between the closing surfaces. In high pressure and high temperature systems, once a seat is scored and leakage developed, it can quickly and severely damage the valve seat by a "wire drawing" effect of the fluid flow through the leakage path. It is possible to repair hard seats by lapping at early stages of damage, but if the leakage goes unchecked replacement of discs and seats may become necessary.

**High Temperature Operation**

Failure analysis indicates that a large number of solenoid operated valve failures occur due to prolonged operation at high temperatures or moisture intrusion, or both. When a solenoid coil is energized, heat is generated. In DC coils the heat generated is simply based on the coil resistance and operating voltage (V²/R). In AC coils additional heat is generated by current circulating in shading ring and eddy current losses. If an obstruction prevents a solenoid plunger from traveling fully, higher currents will continue to flow through the coil, generating additional heat. As a rule of thumb, coil material life is halved for every 10°C
increase in coil temperature. This rule of thumb will also apply when a solenoid valve is heat traced or inadvertently insulated along with piping.

**Valve Body Material**

Brass is the most common valve body material for air pilot SOVs and smaller process valves. Certain larger valves may be made of bronze. Stainless steel is used for all other SOVs. However, neither brass nor bronze is acceptable ASME section III material. Since certain acids and corrosion products can attack brass and bronze, their use is not recommended in safety related systems or with certain hydraulic fluids like FYRQUEL.

**Plastics & Elastomers**

A wide variety of plastics and elastomers are used in the manufacturing of solenoid valves, either as a molded part or as gaskets and seals. Some of these materials can severely degrade with heat and petroleum based lubricants. Sometimes the degraded elastomer seat combined with petroleum lubricant will form a sticky substance that will prevent the valve from opening. Valves installed in radioactive environment must also be evaluated for the long term effect of radiation on the elastomer parts. Table 1 gives properties of various plastics and elastomers and their resistances to the above elements.

<table>
<thead>
<tr>
<th>Material</th>
<th>Resistance to Petroleum</th>
<th>Temperature Limit °F</th>
<th>Radiation Limit 10^-6 Rads</th>
</tr>
</thead>
<tbody>
<tr>
<td>Buna-N</td>
<td>Good</td>
<td>180</td>
<td>100</td>
</tr>
<tr>
<td>Neoprene</td>
<td>Fair</td>
<td>200</td>
<td>100-200</td>
</tr>
<tr>
<td>EPDM</td>
<td>Poor</td>
<td>300</td>
<td>100-200</td>
</tr>
<tr>
<td>Viton</td>
<td>Excellent</td>
<td>400</td>
<td>10-20</td>
</tr>
<tr>
<td>Silicone</td>
<td>Good</td>
<td>450</td>
<td>50-200</td>
</tr>
<tr>
<td>Teflon</td>
<td>Excellent</td>
<td>350</td>
<td>0.01</td>
</tr>
<tr>
<td>Polyimides</td>
<td>Excellent</td>
<td>400+</td>
<td>500</td>
</tr>
</tbody>
</table>

**Applications Overview**

The vast majority of solenoid operated valves in a power generating plant are used in air-pilot applications to operate air operated valves. Virtually all air-pilot valves are brass bodied, although some other materials (like stainless steel) may be found in specialized applications. Also, soft seats are typical in these applications to minimize air leakage from the instrument air system. The most common seat material used is Buna-N. EPDM and Viton are also extensively used in safety related applications.
The second most common generating plant SOV application is as main process valves in steam, water, gas and other fluid systems. These valves can range in size from small valves used on 100 psig systems to 8" valves rated at 2500 psig and 5000 gpm flow. Virtually all the process SOVs are two-way on-off style. While a large percentage of the high pressure steam valves are of piloted globe construction, the smaller valves used to control flow in low pressure systems (cooling water, fuel transfer, etc.) are often soft-seated, piloted diaphragm or piston types.

Power operated relief valves (PORV) are unique application of an SOV. These are designed to provide short term blowdown of high pressure, high temperature steam/water systems. PORVs are used on pressurizers, main steam headers and BWR Automatic Depressurization Systems (ADS).

Valve flow coefficient ($C_v$) is a critical factor in properly sizing a valve for its application. This factor is calculated for each valve based on its flow capacity and pressure drop within the valve. Designers tend to specify valves larger than those required for a particular application. This may not be a good practice in case of piloted SOVs. Oversizing a piloted SOV may,

i) reduce its tolerance to reverse pressurization,

ii) increases possibility of leakage due to larger seating area,

iii) increase the required minimum operating pressure differential (MOPD) to a level above that available in the system.

**Maintenance Recommendations**

Maintenance of solenoid valves does not always involve repair of the defective components. In fact, Automatic Switch Company (ASCO), a major supplier of nuclear grade solenoid valves, discourages any repair of their valves used in nuclear power plant safety systems. A large number of solenoid valves used in the balance of plant systems are small and inexpensive. It is more cost effective to replace these than to repair them. However, it is highly recommended that the valves that are repaired or replaced be analyzed for failure causes. The failure cause may be external to the valve itself, like dirt or debris in the pipeline. If the root cause is not removed, the replacement valve is also likely to fail. Periodic troubleshooting observations will reduce instances of sudden operational failures. Table 2 shows some of items to look for when troubleshooting.

Use of incorrect replacement parts is reported to be a major maintenance problem. There are valves which are supplied with similar model number and are available both in AC or DC versions. An AC coils has much lower resistance than a DC coil. If inadvertently an AC coil is installed in a DC circuit, the valve may appear to operate perfectly at the beginning but the coil is likely to burn out within a short period. On the other hand, if a DC coil is installed in an AC solenoid, the valve may not operate or operate sluggishly due to inadequate magnetic force developed. When ordering replacement parts, part numbers should be carefully reviewed with the supplier. If the valve type is in question, the coil resistance should be measured to determine if the proper coil is being installed.
Table 2
Troubleshooting Guide

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Visually inspect the condition of the valve for physical damage; loose electrical or piping connections; leakage, including any obvious water, moisture and chemical deposits.</td>
</tr>
<tr>
<td>2</td>
<td>Verify the direction of flow with respect to the markings on the valve.</td>
</tr>
<tr>
<td>3</td>
<td>Watch for burned coil insulation odor. This may be indicative of high coil temperature. Infrared thermography can be used to detect high temperature operation.</td>
</tr>
<tr>
<td>4</td>
<td>If possible, remove the cover and look for signs of electrical arcing, insulation cracking or other signs of age. Also, any signs of rust or water rings are indicative of moisture intrusion.</td>
</tr>
<tr>
<td>5</td>
<td>Energize the coil and listen for its characteristic click. Absence of the clicking sound indicates restricted travel of the solenoid plunger. Excessive hum or chatter is a clear indication of a potential electrical arcing or mechanical problem.</td>
</tr>
</tbody>
</table>

Solenoid valve failures due to defective coils are rare. However, coil failures may occur due to high ambient temperature, or water and condensation intrusion into the coil housing. Coils may also burn up due to internally generated heat when the valves fail to open fully. This is identified by excessive hum when opened. Coil life generally ranges from 4-10 years when operated within the rated temperature for the insulation class. Usually non-safety related valves are supplied with general purpose coil enclosures which do not provide protection against dust and moisture intrusion. It is recommended that for most applications in a power plant NEMA 4 enclosures (water and dust tight) be specified. Also, note that a continuously energized coil may become too hot for a standard 90°C rated cable termination.

One of the best SOV preventive maintenance techniques available is periodic valve cycling. Since all valves in any power plant cannot be cycled at defined frequency, it is recommended that the cycling frequency be determined by the plant personnel based on operating experience. However, as a rule of thumb, all air system SOVs should be cycled quarterly. Longer intervals can be justified when experiences indicate no prior operating problem.

Since there are no proven SOV condition monitoring techniques available, it is recommended that in addition to corrective maintenance, periodic replacement of age sensitive parts and rebuilding of selective SOVs be performed to maintain their long term operability. Two major components which may benefit from periodic replacement are the coil and the elastomeric components (viz., seats, diaphragms, seals, etc.). Most periodic replacements are specified in plant EQ programs per 10CFR50.49. Periodic replacements of the SOVs which are not controlled under EQ can be based on manufacturers’ recommendation or operating experience.

In developing a maintenance program, criticality of each SOV should be established. Valves typically fall under three categories,
i) Safety related,

ii) Important to power production,

iii) Others.

Valves in item (iii) may not warrant periodic maintenance. Replacement on failure may be the appropriate maintenance strategy for this class of valves. For safety related valves, maintenance requirements are defined in licensing or EQ documents. These should form the "minimum requirements". For valves important to safety or power production, each one should be evaluated for its credible failure mode for determining periodic maintenance requirements. For example, a normally closed, de-energized SOV's only safety function is to remain in that state. For such a valve periodic maintenance may be unnecessary. For other valves which are required to change state and maintain a minimum seat leakage, periodic maintenance is appropriate for performance and reliability.

**Post Maintenance Testing**

To ensure that proper maintenance work has been performed, every repair or replacement should be followed by a post maintenance testing. Testing should be designed to demonstrate that,

i) the original problem has been corrected,

ii) normal operation has been maintained,

iii) the equipment is capable of performing its design functions.

**References:**


For additional information, contact:

Vic Varma  
EPRI-NMAC  
1300 Harris Blvd.  
Charlotte, NC 28262  
Ph:(704)547-6056  
Fax:(704)547-6035
Maintenance of Solenoid Valves

Presented to

Joint IAEA & NEA Meeting
Paris, France
April 1993

Vic Varma
(704) 547-6056

---

What is NMAC?

- Organized by EPRI in 1988 to:
  - Assist utilities with plant maintenance
  - Synthesize Research Results
  - Apply Industry Experience into Practical Applications
- Membership
  - US Domestic Utilities: 40
  - International Utilities: 8
- Support
  - Supported by utilities through annual contributions
- Charter
  - Provide Technical Assistance to members
  - Facilitate Technology Transfer
    - Technical Guides
    - Workshop/Seminars
    - Plant Visits
    - Hot Line Calls
Solenoid Valve Maintenance Issues

- **BACKGROUND**
  - AEOD REPORT # C-90-01
  - NUREG 1275, Vol. 6, Feb. 1991

- **IDENTIFIED PROBLEM AREAS**
  - Misapplication
  - Lack of knowledge
  - Maintenance/Surveillance Errors
  - Hidden Valves

---

Plan of Attack

- **Technical Advisory Group:**
  - Utilities 9
  - Manufacturers 5
  - EPRI/NMAC 1
  - NUMARC 1
  - AEOD (USNRC) 1
  - Contractor 1
  - TOTAL 18
SOV Application Overview

- Majority used in air-pilot applications
- Second most common use - 2-way main process valves (steam, gas, water, etc.)
- Few are used in hydraulic applications (MSIVs, MFWIV)

Piloted Solenoid Valve

- Piloted SOV is unidirectional
- Minimum DP required for proper operation
Reverse pressurization Problem
(an example)

Properties of Elastomers used in SOVs

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Buna-N</td>
<td>Good</td>
<td>180</td>
<td>100</td>
</tr>
<tr>
<td>Neoprene</td>
<td>Fair</td>
<td>200</td>
<td>100-200</td>
</tr>
<tr>
<td>EPDM</td>
<td>Poor</td>
<td>300</td>
<td>100-200</td>
</tr>
<tr>
<td>Viton</td>
<td>Excellent</td>
<td>400</td>
<td>10-20</td>
</tr>
<tr>
<td>Silicone</td>
<td>Good</td>
<td>450</td>
<td>50-200</td>
</tr>
<tr>
<td>Teflon</td>
<td>Excellent</td>
<td>350</td>
<td>0.01</td>
</tr>
<tr>
<td>Polyimides</td>
<td>Excellent</td>
<td>400+</td>
<td>500</td>
</tr>
</tbody>
</table>
Valve Seats

Soft Seats
- Elastomers
- Effective against leaks
- Limit below 350° F

Hard Seats
- Metal to metal contact
- Highly polished surface
- Effective in high temperature applications
- Easily damaged by contaminants
- Possible wire drawing

Lubrication

- Lubricants
  - Applied to facilitate assembly or reduce friction
  - Excessive lubrication may result in valve failure
    - Migrate to undesirable areas
    - Retain debris/contaminants

- Do's and Don'ts
  - Do not use petroleum based lubricants
  - Do not use silicone grease on silicone O-rings
  - Dry type graphite and molybdenum disulfide with volatile carrier (e.g., alcohol) is recommended
Recommended O-ring Lubricants

<table>
<thead>
<tr>
<th>O-Ring Material</th>
<th>Recommended Lubricant</th>
</tr>
</thead>
<tbody>
<tr>
<td>Silicone</td>
<td>Fluorinated Oil (Krytox) or Water</td>
</tr>
<tr>
<td>Viton</td>
<td>Fluorinated Oil (Krytox) or Water</td>
</tr>
<tr>
<td>Natural Rubber</td>
<td>Silicone Based Fluid/Grease</td>
</tr>
<tr>
<td>Neoprene</td>
<td>Silicone Based Fluid/Grease</td>
</tr>
<tr>
<td>Buna-N</td>
<td>Silicone Based Fluid/Grease</td>
</tr>
<tr>
<td>EPDM</td>
<td>Silicone Based Fluid/Grease</td>
</tr>
<tr>
<td>Urethane</td>
<td>Silicone Based Fluid/Grease</td>
</tr>
<tr>
<td>Butyl</td>
<td>Silicone Based Fluid/Grease</td>
</tr>
</tbody>
</table>

High Temperature Application

- High Temperature Operation
  - Incorrect application in an high temp. ambient
  - Incorrect application for system fluid (viz., hot water)
  - Caused by steam lines in the vicinity
  - Caused by plunger obstruction (high current will continue to flow through coil)
  - Temperature will increase if the coil is insulated or heat traced
  - Coil life is reduced by half for every 10° C increase in temperature
  - Operational condition (continuously energized solenoids will run hotter)
Trouble Shooting

- Listen for “click” during operation.
  - Absence of “click” sound may be due to plunger restriction
  - Gas sensor may have been successfully used to verify SOV operation
- Listen for chatter or hum
  - Excessive chatter or hum indicates potential problem
- Observe smell
  - Smell of burnt insulation will indicate high coil temperature
  - Examine coil for electrical arc or physical damage

SOV Application Recommendations

- Sizing: oversizing is not the best choice
  - Reduced capability to tolerate reverse pressurization
  - Larger seat area promotes higher leakage
- Almost all SOV are unidirectional, unless otherwise stated
- SOV materials in contact with system fluids must be compatible with the fluid
- Use highest temperature class coil available
- Use molded coils and water tight enclosure where moisture may be present
## Summary of Maintenance Recommendations

<table>
<thead>
<tr>
<th>Operating/Testing Condition</th>
<th>Operating/Testing Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>High System Contamination Level</td>
<td>Dis assemble and clean system</td>
</tr>
<tr>
<td>Normally Energized</td>
<td>Replace every 5-10 years</td>
</tr>
<tr>
<td>High Process Temperature</td>
<td>Should be replaced twice as often for each 10°C temp rise.</td>
</tr>
<tr>
<td>Normally Energized Coils</td>
<td>Rebuild more often than normally de-energized</td>
</tr>
<tr>
<td>All Valves</td>
<td>Cycle Periodically</td>
</tr>
<tr>
<td>Valve Diagnostics</td>
<td>No substantiated method</td>
</tr>
</tbody>
</table>
Contents

1. Introduction
2. MOV-Types and Numbers
3. Test Concept and Cycles
4. Detailed Information and Test Steps
5. Final Remarks
6. Appendixes

1 Introduction

The NPP Beznau is working with an integrated MOV test concept since the end of 1989. The start up for this concept was in 1986 after the establishment of a working group. In this group are representatives from all departments which are involved in MOV problems. They are specialists from mechanical and electrical maintenance, engineering groups, operation and quality assurance.

In 1992 we replaced the former used 6 channel paper recorder for MCC-Monitoring with a new computerised diagnostic system. For the new system we installed in each class-1E-MCC-insert (MCC = Motor control center) for MOV's a diagnostic plug which is wired with all important test points, like MOV-current and -voltage, signals from switches and coils.

Derived from this electrical data: resistance of the cable including motor winding, and the basic datas from the test bench recording, now we are able to determine the mechanical torque at the stemnut under real plant conditions in both directions, open and close.
2 MOV-Types and Numbers

- In our two identical power plants there are approximately 120 pieces of class 1E actuators installed.

- They are manufactured by Limitorque, USA Types SMB 000 ... 2 (the old types SMA will be replaced)

- All MOV's are equipped with 3-phase AC-Motors and electrical plug-connections close to the valve-body.

3 Test Concept and Cycles

A) Test bench (Test A)

**topic:**
- Complete overhaul of the actuator (replacement of all weak and aged parts) performed
- Measuring of the actuator basic datas for later comparison with tests B and C
- Adjustment of the torque switch

**interval:**
- 8 years

**aim:**
- Actuator-condition practically new.

B) On site, valve-location (Test B)

**topic:**
- Adjustment of the bypass- and limit switches
- Mechanical examination run behaviour of the value (load, noise and vibration)
- Basic datas to compare later with test C

**interval:**
- 4 years
  First test in the same year as Test A

**aim:**
- Exact adjustment of the limit switches

C) Monitoring at Motor Control Center (MCC) (Test C)

**topic:**
- Check the load behaviour (torque) in comparison to the last year result of test C
- Examination of the set point from the switches
interval:
- Every year
  Last MOV-test just prior to plant startup after yearly revision-shutdown.
- After this test, it is strictly forbidden to open a cover, to adjust the stuffing-box or to disconnect electrical installations at the MOV.

aim:
- Correct function of the MOVs (load and switches) prior to plant startup.

Appendixes 2 und 3 show an overview of the test-concept.

4 Detailed Information and Test Steps

4.1 Test at the bench (Test A)
For recording the basis datas, each MOV has to be fixed on the test-bench after the overhaul.

a) The following measurements will be performed in open and close direction
   - Torque
   - Displacement of worm
   - Power
   - Current
   - Voltage

Open and close operation for adjusting or checking the set points of the torque switch. The switching points should be ± 10 % of the rated value.

After the torque switch has been checked and the set point is o.k. the open and close operation will be checked with 80 % of rated voltage (304 V) of the motor.

b) Characteristic of torque and power in comparison to the worm displacement.

c) Quick evaluation will be made in order to make sure that the values measured are in the tolerances.

The detailed evaluation will be made later at the engineering-office.

4.2 On site, valve location (Test B)
a) The diagnostic test is carried out directly at the valve. The test equipment will be connected to the plugs and sockets, it is not necessary to remove any wires or connections.

b) For checking the displacement of the worm, the displacement receiver has to be fitted and calibrated to the 0-position. After one open and one close operation, the result has to be recorded.

c) Quick evaluation
   - Torque/Limit switch operation
   - Bypass switch

Detailed evaluation will be made at the engineering-office.
4.3 Monitoring at the MCC (test C)

The recordings at the MCC are carried out after completion of all maintenance work at the valves (so called final checks).

One open and one close operation will be performed to record the following measurements:
- Power
- Current (3 phases)
- Voltage
- Control voltage
- Open / close time
- Insulation resistance
- Cable and motor-winding resistance

The measurements are recorded by a computerised diagnostic system.

The test equipment will be connected to a diagnostic plug which is installed in the front plate of the MCC. That means, it is not necessary to disconnect or interrupt any circuit.

As we mentioned previously the computer calculates the real stemnuttorqve at each valve position.

With this information, we can decide how serious additional friction is and if there is a need to disassemble the valve for an internal inspection or overhaul.

In appendix 1 you will find the dependencies for the calculation of the stemnuttorqve.

Quick evaluation will be made by comparing it with last year results. Detailed evaluation will be made the same day by the quality assurance-specialist.

The valve will not be released for operation until the check is O.K.

5 Final Remarks

1) MOV Testing is a teamwork between mechanical and electrical specialists.
2) The education of the test crew must be as high as the quality of the test system.
3) It is essential to have statements concerning the run-behavior of the MOV very quickly after the test run.
4) The better your knowledge of MOV is, the lesser you will have frictions with the authority!
5) You can be sure that MOV's follow physical laws and not nightmares!

6 Appendixes

1) Dependencies for calculation of the torque (1 sheet)
2) Arrangement of tests and test equipment (1 sheet)
3) Test intervals (1 Sheet)
Dependence of Measurement Value

\[ F \sim M \sim \Delta s \sim P' \sim P1 \]

**Diagram:**
- \( \Delta s \) axis
- \( M = f \times F \) on the right side
- \( P' \) on the horizontal axis
dashed lines showing dependencies

**Legend:**
- R (cable + motor winding)
- Voltage supply
- Example:
- \( P' \approx P_{\text{air gap motor}} \)
- \( P' = P1 - [I^2 (R_L + R_I)] \)

**Abbreviations:**
- F = force stem
- M = torque stem nut
- \( \Delta s \) = displacement worm
- \( P_1 \) = power measured at MCC
ARRANGEMENT OF TESTS AND TEST EQUIPMENT

DIAGNOSTIC
4 year interval

ON SITE

MONITORING
1 year interval

MCC

CALIBRATION
8 year interval

TEST BENCH

DETAILED EVALUATION

684
TEST INTERVALS

8 Years
1990
1994
1998

4 Years
1990
1994
1998

Yearly
1990
1994
1998

Yearly
1991
1995

Yearly
1992
1996

Yearly
1993
1997

Testbank
1) Basic data
2) Setting torque switch
3) Switch off by 0.8 U

Local-Diagnostic
1) Setting Limit Switch
2) Setting Bypass Switch
3) Additional Torque by Inertia

MCC-Recording
1) Run behaviour
2) Switch off
3) Control
4) Insulation

Measured values
a) Torque Spring feature
b) Stall Torque
c) Set point Torque Switch

Measured values
a) Bypass time
b) Add. Torque
c) Run time
d) Run power
e) Switch off power
f) $\Delta$ I spring

c) Run time
b) Add. Torque
c) Run time
d) Run power
e) Switch off power
f) $\Delta$ I spring

Run time ($t$)
Run power ($Pr$)
Switch off power ($Ps$)
Insulation ($Ri$)
Delay time Cont. ($td$)
Resistance Cable+Motor ($Rm$)
Derived Torque ($M$)

$Pr$
$Ps$
$Ri$
$td$
$Rm$
$M$

Appendix 3) to AN-402-EQ94009
SOME PROBLEMS OF THE UKRAINIAN STATE COMMITTEE ON NUCLEAR AND RADIATION SAFETY REGULATORY ACTIVITY AS TO NPP ARMATURE OPERATION

V. Glygal, V. Kovyrshin, N. Zaritsky, I. Privalko
(UkrSCNRS' Scientific and Technical Centre, Kiev)

Up to now in Ukraine (and in former USSR countries too) power machine-building equipment was operating either to extreme state or during the predetermined servicing term in application conditions established by design and operation documentation. The problem of an equipment technological state change investigation during operation (ageing) and its servicing term prolongation possibility assessment was practically out of attention. Such an approach was used while NPP armature operation as one of numerous elements of different technological systems, including safety significant systems, too.

According to the standard document OTT-87 ("Armature for NPP equipment and pipings. General technical requirements.") which acts in Ukraine, the armature, excluding check valves of unremovable structure, belongs to the class of repaired, recovered articles with regulated discipline of recovery and assigned duration of operation. Assigned servicing term of removable (movable) parts is 10 years while the armature frame parts - 30 years. Besides, OTT-87 establishes the assigned turnout (resource) for the different types of armature as follows:

- gate valves - 1000 cycles;
- shut-off valves - 2700 cycles;
- preventive armature - 200 cycles, etc.

From the other hand, NPP operation experience shows that many types of armature (naturally, depending upon a technological system where they are installed) do not work their assigned cyclic resource out. Besides, armature is operating at NPP with regulated discipline of technical servicing, operability checking, metal state control, and repairing. In a connection with it there are works being carried out at many NPP blocks on the removable parts of armature technical state and servicing term prolongation possibility assessment.

The present information is aimed in:

- to deal with some problems related to creation of standard base on NPP equipment ageing and servicing term prolongation;
- to characterize in general the works which are being carried out presently at the ChNPP on Du-800 gate valves removable parts technical state investigation.

Block double-disc valves Du-800 are made at Alexyn plant (Russia) and used as gate bodies. They are usually being installed at pressure (suction) pipings of RBMK type reactor facilities MFCC.

There are 16 of them installed at one unit. Conducted medium - circulating water, working pressure - 70 kgf/cm², temperature -270 °C.

Operation conditions:
- permanently opened on working MCPs;
- permanently closed on switched off MCPs.

687
To determine the technical state of Du-800 gate valves removable parts the program was developed which consists of the main tasks as follows:

1. Collection and analysis of information on damages, failures and substitutions of gate valves removable parts during the whole operation period.

2. Complex investigation of the gate valve removable parts metal state.

Here it is necessary to note that gate valves metal state operation control is carried out mainly for frame parts. That is why the main attention in the program is paid to the removable parts conjugation points as well as glands and weld overlays defectoscopy.

3. Mechanical testings (while normal and elevated temperatures) of samples cut from the main removable parts details out which worked in an unit from the beginning of its operation.

4. Assessment of normalized reliability indices (unfailing operation probability, working through to the failure).

5. Verificative calculation for cyclic strength and assessment of permissible loading cycles number for the main removable parts of gate valves details.

The necessity of strength calculation to be carried out is stipulated by that the gate valves were elaborated in 70th.

Presently there are new norms for strength calculation of NPP equipment and pipings established.

In the present time the program on Du-800 gate valves removable parts technical state assessment is already partially carried out. During the last planned the ChNPP units shutdowns some of gate valves being operated were opened, and removable parts defectoscopy was carried out. Besides, the mechanical testing (tension and shock bending) of spindle and discs being in operation during 11 and 14 years, respectively, was carried out. It was established that mechanical characteristics of removable parts metal satisfy the requirements of the technical documentation.

The conclusions on removable parts technical state would be done after the whole complex of works fulfilling.

One of the suppressive factors as to the problem of the NPP equipment technical state assessment and servicing term prolongation to be solved is the lack of sufficiently flexible and operable standard base.

Presently several organizations, including the UkrSCHNRS’STC, are carrying out the practical works on the development of relevant standard documentation. The practical experience gathered while carrying of such works out allows to distinguish three levels of standard and technical documentation (STD) which are to be considered while solving of the problem.

Level 1. The rules and norms in nuclear energetics

At this level we consider as necessary to introduce some supplements and changes into an acting STD which would give a principle
possibility and legislative base for the servicing term of all kinds of equipment (thermotechnical, electrotechnical, automatic means, etc.) prolongation with taking into account the safety groups and classes as well as the operative organization responsibility for the works on technical state assessment and establishing of a new assigned equipment servicing term with sufficient substantiation of its safe operation organizing.

The certain elaborations and supplements to the acting STD were already considered at the UkrSCNRS board meeting.

Level 2. Departmental leading documents

Here we consider as necessary to develop the documents regulating the general order and volume of conducted works, requirements to inspection and testing methods and programs as well as requirements to account documentation on technical state assessment and substantiation of further operation possibility with provision of the required safety levels.

The 1st edition of such document developed by the Ukrainian State Committee on Nuclear Power Utilization was already considered in the UkrSCNRS.

Level 3. Operational organizations documentation

At this level it is practically expedient to elaborate the "Statute on resource (servicing term) prolongation" for certain kinds of equipment at certain NPP. Presently there already exist some drafts of such statutes on thermomechanical equipment (the removable parts of armature), systems for radiation control, etc.

It should be noted that above mentioned STD levels have to functionate within the frames of national programme for NPP components ageing and resource prolongation. The development of the Ukrainian national programme will, naturally, demands the financial and intellectual expenses.
## Joint Specialist Meeting CSNI/PWG1
### IAEA/Division of Nuclear Safety

**on Motor Operated Valve Issues in Nuclear Power Plants**

April 25th-27th, 1994 in Paris, France

**REGISTRATION LIST**

<table>
<thead>
<tr>
<th>COUNTRY</th>
<th>NAME</th>
<th>ORGANISATION/COMPANY</th>
<th>NUMBERS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Belgium</td>
<td>Mr. M. DUBOIS</td>
<td>AIB - Vinçotte Nuclear</td>
<td>Tel. +32 2 536 83 85</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Ave. du Roi, 157</td>
<td>Fax. +32 2 536 85 85</td>
</tr>
<tr>
<td></td>
<td></td>
<td>B-1060 Brussels</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Mr. G. JACQUEMIN</td>
<td>Westinghouse European Service</td>
<td>Tel. +32 67 287 864</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Rue de l'Industrie, 43</td>
<td>Fax. +32 67 287 821</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1400 Nivelles</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Mr. R.G.L. VANDENBUSSCHE</td>
<td>Tractebel</td>
<td>Tel. +32 2 773 77 43</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Ave. Arlane, 7</td>
<td>Fax. +32 2 773 89 00</td>
</tr>
<tr>
<td></td>
<td></td>
<td>B-1200 Brussels</td>
<td></td>
</tr>
<tr>
<td>Canada</td>
<td>Mr. B. FERGUSON</td>
<td>Ontario Hydro</td>
<td>Tel. +1 519 361 4716</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Bruce NGS &quot;A&quot;</td>
<td>Fax. +1 519 361 4759</td>
</tr>
<tr>
<td></td>
<td></td>
<td>P.O. Box 3000</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Tiverton, Ontario NOG 2TO</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Mr. W. FITZGERALD</td>
<td>Ontario Hydro</td>
<td>Tel. +1 519 361 47 16</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Bruce NGS &quot;A&quot;</td>
<td>Fax. +1 519 361 47 59</td>
</tr>
<tr>
<td></td>
<td></td>
<td>P.O. BOX 3000</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Tiverton, Ontario NOG 2TO</td>
<td></td>
</tr>
<tr>
<td>Czech Rep.</td>
<td>Mr. J. HULIN</td>
<td>CEZ Jadema Elektrama Dukovany</td>
<td>Tel. +42 509 9231 1. 3528</td>
</tr>
<tr>
<td></td>
<td></td>
<td>675 50 Dukovany</td>
<td>Fax. +42 509 922 495</td>
</tr>
<tr>
<td></td>
<td>Mr. J. BURKERT</td>
<td>CEZ Jadema Elektrama Dukovany</td>
<td>Tel. +42 509 9231 A. 5447</td>
</tr>
<tr>
<td></td>
<td></td>
<td>675 50 Dukovany</td>
<td>Fax. +42 509 922 495</td>
</tr>
<tr>
<td>Finland</td>
<td>Mr. J. HONKANEN</td>
<td>TVO</td>
<td>Tel. +358 938 3811</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Senior Engineer, Process Systems</td>
<td>Fax. +358 938 3814 209</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SF-27160 Olkiluoto</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Mr. J. SNELLMAN</td>
<td>Imatran Voima OY</td>
<td>Tel. +358 1819 inolo fi</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Loviisa Power Plant</td>
<td>Fax. +358 15 550 4435</td>
</tr>
<tr>
<td></td>
<td></td>
<td>P.O. Box 23</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>FIN-07901 Loviisa</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Mr. J.T. LAMMI</td>
<td>Finnish Centre for Radiation and Nuclear</td>
<td>Tel. +358 0 759 881</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Safety</td>
<td>Fax. +358 0 7599 8382</td>
</tr>
<tr>
<td></td>
<td></td>
<td>P.O. Box 14</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>FIN-00881 Helsinki</td>
<td></td>
</tr>
<tr>
<td>Country</td>
<td>Name</td>
<td>Address</td>
<td>Phone</td>
</tr>
<tr>
<td>---------</td>
<td>-----------------</td>
<td>-------------------------------------------------------------------------</td>
<td>------------</td>
</tr>
<tr>
<td>France</td>
<td>Mr. K. ELLIS</td>
<td>Ontario Hydro Liaison Engineer to EDF c/o Electricité de France EdF PT/EPN/Inspection nucléaire Place Marcel Paul 92003 Nanterre Cedex</td>
<td>+33 1 47 25 89 69</td>
</tr>
<tr>
<td></td>
<td>Mr. J. THOMMASON</td>
<td>Electricité de France 2, av. du Gal. de Gaulle 92141 Clamart Cedex</td>
<td>+33 1 47 65 47 71</td>
</tr>
<tr>
<td></td>
<td>Mr. CACHELOU</td>
<td>13/27, Esplanade Ch. de Gaulle Département de la Maintenance 92060 Paris la Defense Cedex 57</td>
<td>+33 1 49 02 05 96</td>
</tr>
<tr>
<td></td>
<td>Mr. GRENET</td>
<td>Electricité de France 12-14, av. Dutrévoz 69628 Villeurbanne Cedex</td>
<td>+33 1 67 82 73 18</td>
</tr>
<tr>
<td></td>
<td>Mr. C. FICHTENBERG</td>
<td>Ets. L. Bernard S.A. 60, av. du Pdt. Wilson 93211 La Plaine St. Denis</td>
<td>+33 1 48 09 25 29</td>
</tr>
<tr>
<td></td>
<td>Mr. J. SERIN</td>
<td>GEC Alsthom Sapag Ave. P. Brossolette 59260 Armentières</td>
<td>+33 1 62 20 10 56 77</td>
</tr>
<tr>
<td></td>
<td>Mr. P. JAMET</td>
<td>Département d'évaluation de sécurité IPSN-CEN/FAR B.P. 6 92260 Fontenay aux Roses</td>
<td>+33 1 46 54 70 92</td>
</tr>
<tr>
<td></td>
<td>Mr. R. ZERMIZOGLOU</td>
<td>DES/GEREP Institut de protection et de sécurité nucléaire - CEA Centre d'études nucléaires de Fontenay aux Roses B.P. 6 92260 Fontenay aux Roses</td>
<td>+33 1 46 54 78 24</td>
</tr>
<tr>
<td></td>
<td>Mr. D. PANNEFIEU</td>
<td>BCCN 15-17, av. Jean Berin 21000 Dijon</td>
<td>+33 1 62 80 29 40 26</td>
</tr>
<tr>
<td></td>
<td>Mr. M. MONIER</td>
<td>GEC Alsthom Velan 90, rue Chellemel la cour 69367 Lyon Cedex 7</td>
<td>+33 1 67 86 67 14</td>
</tr>
<tr>
<td></td>
<td>Mr. J.P. RENAUDIER</td>
<td>EDF Septien 12-14, av. Dutrévoz 69628 Villeurbanne Cedex</td>
<td>+33 1 67 82 74 96</td>
</tr>
<tr>
<td></td>
<td>Mr. D. PINIER</td>
<td>EdF/Direction des Etudes et Recherches Route de Sens - Ecuelles 77250 Moret/Loing</td>
<td>+33 1 67 73 74 58</td>
</tr>
<tr>
<td></td>
<td>Mr. B. DEVERLY</td>
<td>Rotork Motorisation 75, rue Rateau 93120 La Courneuve</td>
<td>+33 1 48 35 44 99</td>
</tr>
<tr>
<td></td>
<td>Mr. P. DICOQUEMARE</td>
<td>Framatome Nuclear Services 10, rue Juliette Recamier 69006 Lyon</td>
<td>+33 1 67 72 74 84 43</td>
</tr>
<tr>
<td>Name</td>
<td>Address</td>
<td>Telephone</td>
<td>Fax</td>
</tr>
<tr>
<td>-----------------</td>
<td>--------------------------------------------------------------------------</td>
<td>-----------</td>
<td>-------------------</td>
</tr>
</tbody>
</table>
| Mr. P. BOURG    | Framatome Nuclear Services  
10, rue Juliette Recamier  
69006 Lyon                  | Tel. +33 16 72 74 84 43  
Fax. +33 16 72 74 86 56    |
| Mr. COPPOLANI   | Deputy Manager Electromechanical Dept  
Framatome  
Tour Fiat Cedex 16  
92084 Paris la Defense   | Tel. +33 1 47 96 39 19  
Fax. +33 1 47 96 32 69    |
| Mr. J. STEINLEN | Fisher Rosemount  
Rue Paul Baudry  
B.P. 10  
68700 Cernay                 | Tel. +33 16 89 37 65 95  
Fax. +33 16 89 75 43 26    |
| Mr. J.L. MAZEL  | GEC Alsthom Velan  
90, rue Challemel Lacour  
69367 Lyon Cedex 7            | Tel. +33 16 78 61 67 00  
Fax. +33 16 78 72 12 18    |
| Mr. P. HENRY    | GEC Alsthom Velan  
90, rue Challemel Lacour  
69367 Lyon Cedex 7            | Tel. +33 16 78 61 67 00  
Fax. +33 16 78 72 12 18    |
| Mr. J. CHANOIS  | GEC Alsthom Velan  
10-12, Ave. des Olympiades  
94132 Fontenay aux Bois Cedex | Tel. +33 1 43 94 32 90  
Fax. +33 1 43 94 19 61    |
| Ms. PAYEN       | Direction de la sûreté des installations nucléaires  
CEN/FAR B.P. 6  
92260 Fontenay aux Roses | Tel +33 1 46 54 71 69  
Fax +33 1 42 53 66 15    |
| Mr. M. BLOT     | Direction de la sûreté des installations nucléaires  
CEN/FAR B.P. 6  
92260 Fontenay aux Roses | Tel +33 1 46 54 71 69  
Fax +33 1 42 53 66 15    |
| Germany         |                                                                         |           |                   |
| Mr. R. KUBOSCH  | TÜV Rheinland e.V.  
P.O. Box 91 09 51  
D-51101 Cologne               | Tel. +49 221 806 0  
Fax. +49 221 806 114   |
| Mr. A. SCHEUER  | TÜV Rheinland e.V.  
P.O. Box 91 09 51  
D-51101 Cologne               | Tel. +49 221 806 0  
Fax. +49 221 806 114   |
| Mr. G. SASONOW  | Kemkraftwerk Brunsbuttel  
Otto Hahn Strabe  
D-25541 Brunsbuttel            | Tel. +49 4852 892 480  
Fax. +49 4852 892 011   |
| Mr. A. GAERTNER | Kemkraftwerk Brunsbuttel  
Otto Hahn Strabe  
D-25541 Brunsbuttel            | Tel. +49 4852 892 400  
Fax. +49 4852 892 011   |
| Mr. W. RIEGER   | TÜV Südwest  
Gottlieb Daimler Str. 7  
D-70794 Filderstadt             | Tel. +49 711 7706 243  
Fax. +49 711 7706 201   |
| Mr. W. HEMPELMANN | Elektro Mechanik GmbH  
Industriestr. 1  
D-57482 Wenden                | Tel. +49 2762 612 373  
Fax. +49 2762 612 217   |
<table>
<thead>
<tr>
<th>Name</th>
<th>Address</th>
<th>Phone</th>
<th>Fax</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mr. W. HANDEL</td>
<td>Elektro Mechanik GmbH</td>
<td>Tel. +49 2762 612 373</td>
<td>Fax. +49 2762 612 217</td>
</tr>
<tr>
<td></td>
<td>Industriestr. 1</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-57482 Wenden</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. P. ZIMMERMANN</td>
<td>Elektro Mechanik GmbH</td>
<td>Tel. +49 2762 612 250</td>
<td>Fax. +49 2762 612 359</td>
</tr>
<tr>
<td></td>
<td>Industriestr. 1</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-57482 Wenden</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. N. RAUFFMANN</td>
<td>Siemens AG</td>
<td>Tel. +49 69 807 26 47</td>
<td>Fax. +49 69 807 47 98</td>
</tr>
<tr>
<td></td>
<td>KWU - NDM 3</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Berliner Str. 295-303</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-63067 Offenbach (Main)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. P. KRADEPOHL</td>
<td>Siemens AG</td>
<td>Tel. +49 69 807 32 59</td>
<td>Fax. +49 69 807 39 21</td>
</tr>
<tr>
<td></td>
<td>KWU - NDM 3</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Berliner Str. 295-303</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-63067 Offenbach (Main)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. K. KOTTHOFF</td>
<td>Gesellschaft für Anlagen und Reaktorsicherheit</td>
<td>Tel. +49 221 2068 417</td>
<td>Fax. +49 221 2068 442</td>
</tr>
<tr>
<td></td>
<td>Schwertnergasse 1</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-50667 Köln</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. P. KEMERY</td>
<td>TÜV Südwest</td>
<td>Tel. +49 711 7706 242</td>
<td>Fax. +49 711 7706 201</td>
</tr>
<tr>
<td></td>
<td>FB Kerntechnik &amp; Strahlenschutz</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Raiffeisenstr. 30</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-70794 Filderstadt Bonlander</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. S. STEINER</td>
<td>TÜV Bayern Sachsen</td>
<td>Tel. +49 89 5791 1606</td>
<td>Fax. +49 89 5791 2157</td>
</tr>
<tr>
<td></td>
<td>Abt. G2 ETL</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Westendstrasse 199</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-80886 München</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. G. KONIG</td>
<td>TÜV Bayern Sachsen</td>
<td>Tel. +49 89 5791 1215</td>
<td>Fax. +49 89 5791 2157</td>
</tr>
<tr>
<td></td>
<td>Abt. G2 ETK 50</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Westendstrasse 199</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-80886 München</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. M. BOMBECK</td>
<td>TÜV Bayern Sachsen</td>
<td>Tel. +49 89 5791 2609</td>
<td>Fax. +49 89 5791 2157</td>
</tr>
<tr>
<td></td>
<td>Abt. G2 ETM 10</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Westendstrasse 199</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-80886 München</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. G. MULLER</td>
<td>Technischer Überwachungs-Verein</td>
<td>Tel. +49 511 986 1833/1826</td>
<td>Fax. +49 511 986 1848</td>
</tr>
<tr>
<td></td>
<td>Hannover/Sachsen Anhalt</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Abt. AF</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Am TÜV 1</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-30519 Hannover</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. R. SCHRAMM</td>
<td>TÜV Nord</td>
<td>Tel. +49 40 8557 2512</td>
<td>Fax. +49 40 8557 2429</td>
</tr>
<tr>
<td></td>
<td>Grosse Bahnstr. 31</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-22525 Hamburg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. G. GUNKEL</td>
<td>TÜV Südwest o.V.</td>
<td>Tel. +49 621 395 534</td>
<td>Fax. +49 621 333 928</td>
</tr>
<tr>
<td></td>
<td>Dudenstr. 28</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-68167 Manheim</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. H. DANNATH</td>
<td>TÜV Nord</td>
<td>Tel. +49 40 8557 0</td>
<td>Fax. +49 40 8557 2295</td>
</tr>
<tr>
<td></td>
<td>Grosse Bahnstr. 31</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>D-22525 Hamburg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Name</td>
<td>Company/Address</td>
<td>Phone</td>
<td>Fax</td>
</tr>
<tr>
<td>---------------------------</td>
<td>---------------------------------------------------------------------------------</td>
<td>----------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>Mr. P. HESS</td>
<td>Ministerialrat de Minister für Finanzen und Energie des landes Schleswig Holstein Abt. Reaktorsicherheit Brunswikerstrasse 16-22 24105 Kiel</td>
<td>Tel. +49 431 596 5035 Fax. +49 431 596 5386</td>
<td></td>
</tr>
<tr>
<td>Mr. G. SOMMER</td>
<td>TÜV Südwest e.V. Dudenstrasse 28 D-68167 Mannheim</td>
<td>Tel. +49 621 395 490 Fax. +49 621 333 928</td>
<td></td>
</tr>
<tr>
<td>Mr. K.D. BANDHOLZ</td>
<td>Energiesystem Nord Gmbh Walderdamm 17 23000 Kiel</td>
<td>Tel. +49 431 66 00 152 Fax. +49 431 66 00 119</td>
<td></td>
</tr>
<tr>
<td>Mr. H. GRAVENHOFFF</td>
<td>Preussen Elektra, Tresckowstr. 3 30457 Hannover</td>
<td>Tel. +49 511 439 4347 Fax. +49 511 439 2551</td>
<td></td>
</tr>
<tr>
<td>Mr. M. SCHMITZ</td>
<td>Assistant Manager Nuclear Services Siemens AG, KWU NV21 P.O. Box 3220 D-91050 Erlangen</td>
<td>Tel. +49 9131 18 3695 Fax. +49 9131 18 7108</td>
<td></td>
</tr>
</tbody>
</table>

**Hungary**

<table>
<thead>
<tr>
<th>Name</th>
<th>Company/Address</th>
<th>Phone</th>
<th>Fax</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mr. Z. UJARRY</td>
<td>NPP Paks I&amp;C Dept., Valves Dept. Paks Pl. 71. H-7031</td>
<td>Tel. +36 75 317 738 Fax. +36 75 317 717</td>
<td></td>
</tr>
<tr>
<td>Mr. G. NEMETH</td>
<td>NPP Paks I&amp;C Dept., Valves Dept. Paks Pl. 71. H-7031</td>
<td>Tel. +36 75 318 748 Fax. +36 75 317 717</td>
<td></td>
</tr>
</tbody>
</table>

**India**

<table>
<thead>
<tr>
<th>Name</th>
<th>Company/Address</th>
<th>Phone</th>
<th>Fax</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mr. G.P. SRIVASTAVA</td>
<td>Bhabha Atomic Research Centre Reactor Control Division</td>
<td>Tel. +9122 551 18 83 Fax. +9122 556 07 50</td>
<td></td>
</tr>
</tbody>
</table>

**Japan**

<table>
<thead>
<tr>
<th>Name</th>
<th>Company/Address</th>
<th>Phone</th>
<th>Fax</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mr. M. KIMURA</td>
<td>Safety Information Research Ctr. Nuclear Power Engineering Corp. Fujita Kanko Toranomon Bldg. 3-17-1 Toranomon, Minato-ku Tokyo 105</td>
<td>Tel. +81 3 5470 5500 Fax. +81 3 5470 5524</td>
<td></td>
</tr>
<tr>
<td>Mr. H. SAKAMOTO</td>
<td>Tomari Power Station Electrical Maintenance Section Hokkaido Electric Power Co. 726 Ohaza Horikapp-Mura, Tomari-Mura Furu-u Gun, Hokkaido 045-02</td>
<td>Tel. +81 135 75 3331 Fax. +81 135 75 3931</td>
<td></td>
</tr>
<tr>
<td>Mr. T. KAMIYAMA</td>
<td>Technical Department R&amp;D Dept. Toa Valve Co. Ltd. 5-12-1 Nishitachibana-cho Amagasaki Hyogo 660</td>
<td>Tel. +81 6 416 8876 Fax. +81 6 419 8862</td>
<td></td>
</tr>
<tr>
<td>Mr. S. TAKEDA</td>
<td>Technical Department R&amp;D Dept. Toa Valve Co. Ltd. 5-12-1 Nishitachibana-cho Amagasaki Hyogo 660</td>
<td>Tel. +81 6 416 8876 Fax. +81 6 419 8862</td>
<td></td>
</tr>
</tbody>
</table>

695
<table>
<thead>
<tr>
<th>Country</th>
<th>Name</th>
<th>Address/Details</th>
<th>Telephone</th>
<th>Fax</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mexico</td>
<td>Mr. ARREVILLAGA F.</td>
<td>C.F.E. Carretera Cardel Nautia -km 425 Veracruz MEXICO</td>
<td>Tel. +29 34 02 66</td>
<td>Fax. +29 34 35 72</td>
</tr>
<tr>
<td>Netherlands</td>
<td>Mr. J. OFFERMAN</td>
<td>N.V. Elektriciteits Produktie maatschappij Zuid Nederland EPZ Lokatie Zeeland P.O. Box 130 NL-4380 AC Vlissingen</td>
<td>Tel. +31 1105 6000</td>
<td>Fax. +31 1105 2434</td>
</tr>
<tr>
<td></td>
<td>Mr. A.J. ROOSEBOOM</td>
<td>Nuclear Safety Department (KFD) P.O. Box 90804 NL-2509 LV Den Haag</td>
<td>Tel. +31 70 333 55 41</td>
<td>Fax. +31 70 333 40 18</td>
</tr>
<tr>
<td>Russia</td>
<td>Mr. A.A. ZRELKIN</td>
<td>All Russia Research Inst. for Nuclear Power Plant Operation 25 Ferganskaya St. 109507 Moscow</td>
<td>Tel. +7 095 376 1540</td>
<td>Fax. +7 095 274 0075</td>
</tr>
<tr>
<td></td>
<td>Mr. B.N. TIUNIN</td>
<td>All Russia Research Inst. for Nuclear Power Plant Operation 25 Ferganskaya St. 109507 Moscow</td>
<td>Tel. +7 095 376 1540</td>
<td>Fax. +7 095 274 0075</td>
</tr>
<tr>
<td>Slovakia</td>
<td>Mr. M. LIPAR</td>
<td>Nuclear Regulatory Authority of the Slovak Republic Okruzna 5 918 64 Tmava</td>
<td>Tel. +42 805 43251/43013</td>
<td>Fax. +42 805 43014</td>
</tr>
<tr>
<td>Slovenia</td>
<td>Mr. A. KUNEJ</td>
<td>Krsko Nuclear Power Plant Vrbina 12 68270 Krsko</td>
<td>Tel. +386 0608 21814</td>
<td>Fax. +386 0608 21528</td>
</tr>
<tr>
<td>Spain</td>
<td>Mr. V. BARBERO</td>
<td>Iberdrola Goya 4 28001 Madrid</td>
<td>Tel. +34 1 5758 151</td>
<td>Fax. +34 1 5782 107</td>
</tr>
<tr>
<td></td>
<td>Mr. A. PEREZ RODRIGUEZ</td>
<td>Consejo de Seguridad Nuclear Calle Justo Dorado 11 28040 Madrid</td>
<td>Tel. +34 1 3460 486</td>
<td>Fax. +34 1 3460 588</td>
</tr>
<tr>
<td></td>
<td>Mr. L. PASCUAL</td>
<td>ENWESA Supervisor Valve Maintenance &amp; Diagnostic Orense 70 28020 MADRID</td>
<td>Tel. +34 1 572 10 43</td>
<td>Fax. +34 1 572 10 43</td>
</tr>
<tr>
<td>Name</td>
<td>Address</td>
<td>Phone</td>
<td>Fax</td>
<td></td>
</tr>
<tr>
<td>------</td>
<td>---------</td>
<td>-------</td>
<td>-----</td>
<td></td>
</tr>
<tr>
<td>Mr. A. SANTIAGO</td>
<td>Framatome Iberica&lt;br&gt;Calle Serrano 43, 6ª oficina 21&lt;br&gt;28006 Madrid</td>
<td>Tel. +34 1 435 88 63&lt;br&gt;Fax. +34 1 576 64 14</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ms. C. RODRIGUEZ</td>
<td>Framatome Iberica&lt;br&gt;Calle Serrano 43, 6ª oficina 21&lt;br&gt;28006 Madrid</td>
<td>Tel. +34 1 435 88 63&lt;br&gt;Fax. +34 1 576 64 14</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. L.M. GARCIA FERRER</td>
<td>EUMYNSA&lt;br&gt;Ctra Castellon&lt;br&gt;Km 5,6&lt;br&gt;ZARAGOZA</td>
<td>Tel. +34 76 500 900&lt;br&gt;Fax. +34 76 500 416</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Sweden</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. G. DENGORS</td>
<td>OKG Aktiebolag&lt;br&gt;Oskarshamnverket&lt;br&gt;57093 Figeholm</td>
<td>Tel. +46 49 18 63 91&lt;br&gt;Fax. +46 49 18 60 90</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. S. ERIXON</td>
<td>Swedish Nuclear Power Inspectorate&lt;br&gt;Box 27106&lt;br&gt;S-10252 Stockholm</td>
<td>Tel. +46 8 665 4400&lt;br&gt;Fax. +46 8 661 9086</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. L.E. FOLKESON</td>
<td>Vattenfall Ringhals&lt;br&gt;Avd. RPT&lt;br&gt;S-43022 Våröbacka</td>
<td>Tel. +46 340 66 76 77&lt;br&gt;Fax. +46 340 66 73 05</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. F. STENBERG</td>
<td>ABB Atom AB&lt;br&gt;Nuclear Services Division&lt;br&gt;S-721 63 Vasteras</td>
<td>Tel. +46 21 348 066&lt;br&gt;Fax. +46 21 139 138</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. B. PIHLGREN</td>
<td>Forsmarks Kraftgrupp&lt;br&gt;S-742 03 Östhammar</td>
<td>Tel. +46 173 810 00&lt;br&gt;Fax. +46 173 551 16</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. K.F. INGEMARSSON</td>
<td>Forsmarks Kraftgrupp&lt;br&gt;S-742 03 Östhammar</td>
<td>Tel. +46 173 810 00&lt;br&gt;Fax. +46 173 551 16</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. B. ANDERSSON</td>
<td>Forsmarks Kraftgrupp&lt;br&gt;S-742 03 Östhammar</td>
<td>Tel. +46 173 810 00&lt;br&gt;Fax. +46 173 551 16</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Switzerland</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. E. LAMPRECHT</td>
<td>Kemkraftwerk Keibstadt AG&lt;br&gt;CH-4353 Leibstadt</td>
<td>Tel. +41 56 47 71 11&lt;br&gt;Fax. +41 56 47 77 90</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. M. WETTSTEIN</td>
<td>Division principale de la sécurité des installations nucléaires&lt;br&gt;CH-5232 Villigen HSK</td>
<td>Tel. +41 56 99 39 25&lt;br&gt;Fax. +41 56 99 39 07</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. H. BURREN</td>
<td>Kemkraftwerk Bezau&lt;br&gt;CH-5312 Dottingnen</td>
<td>Tel. +41 56 99 76 84&lt;br&gt;Fax 41 56 99 77 02</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. K. THOMA</td>
<td>Kemkraftwerk Bezau&lt;br&gt;CH-5312 Dottingnen</td>
<td>Tel. +41 56 99 75 12&lt;br&gt;Fax +41 56 99 77 01</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Ukraine</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. ZORITSKII</td>
<td>Ukrainian State Committee on Nuclear &amp; Radiation Safety Scientific and Technical Centre&lt;br&gt;17 Kharkovskoye Shosse&lt;br&gt;Kiev 253160</td>
<td>Tel. +7 044 559 38 80&lt;br&gt;Fax. +7 044 559 98 06</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

697
<table>
<thead>
<tr>
<th>Name</th>
<th>Organization</th>
<th>Address decimals</th>
<th>Phone</th>
<th>Fax</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mr. PRIVALKO</td>
<td>Ukrainian State Committee on Nuclear &amp; Radiation Safety Scientific and Technical Centre</td>
<td>17 Kharkovskoye Shosse Kiev 253160</td>
<td>Tel. +7 044 559 38 80</td>
<td>Fax. +7 044 559 98 06</td>
</tr>
<tr>
<td>United Kingdom</td>
<td>Mr. D.C. ANDERSON</td>
<td>Nuclear Installations Inspectorate Health &amp; Safety Executive St. Peter's House Balliol Road Bootle, Merseyside L20 3LZ</td>
<td>Tel. +44 051 951 4000</td>
<td>Fax. +44 051 922 5980/1158</td>
</tr>
<tr>
<td></td>
<td>Mr. T. DAVENPORT</td>
<td>Nuclear Installations Inspectorate Health &amp; Safety Executive St. Peter's House Balliol Road Bootle, Merseyside L20 3LZ</td>
<td>Tel. +44 051 951 4000</td>
<td>Fax. +44 051 922 5980/1158</td>
</tr>
<tr>
<td></td>
<td>Mr. I. BURNELL</td>
<td>Rotork Brassmill Lane Bath BA1 3JQ</td>
<td>Tel. +44 225 428 451</td>
<td>Fax. +44 225 333 467</td>
</tr>
<tr>
<td></td>
<td>Mr. T.F.P. MAY</td>
<td>Limitorque International Bone Lane, Newbury BG14 5FH Berks</td>
<td>Tel. +44 635 46 999</td>
<td>Fax. +44 635 521 680</td>
</tr>
<tr>
<td>United States</td>
<td>Mr. B. VARMA</td>
<td>Electric Power Research Inst. (EPRI) 1300 Harris Blvd. Charlotte, North Carolina 28262</td>
<td>Tel. +1 301 547 6056</td>
<td>Fax. +1 301 547 6035</td>
</tr>
<tr>
<td></td>
<td>Mr. R. STEELE</td>
<td>Idaho National Engineering Lab. P.O. Box 1625 Idaho Falls, Idaho 83415-3870</td>
<td>Tel. +1 208 526 6409</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Mr. T.G. SCARBROUGH</td>
<td>U.S. Nuclear Regulatory Commission Mail Stop OWFN 7E23 Washington D.C. 20555</td>
<td>Tel. +1 301 504 2794</td>
<td>Fax. +1 301 504 2444</td>
</tr>
<tr>
<td></td>
<td>Mr. J.E. ROSENTHAL</td>
<td>U.S. Nuclear Regulatory Commission Office for Analysis &amp; Evaluation of Operational Data Washington D.C. 20555</td>
<td>Tel. +1 301 492 4440</td>
<td>Fax. +1 301 492 8931</td>
</tr>
<tr>
<td></td>
<td>Mr. E.J. BROWN</td>
<td>U.S. Nuclear Regulatory Commission Office for Analysis &amp; Evaluation of Operational Data Washington D.C. 20555</td>
<td>Tel. +1 301 492 4491</td>
<td>Fax. +1 301 492 8931</td>
</tr>
<tr>
<td></td>
<td>Mr. H.L. ORNSTEIN</td>
<td>U.S. Nuclear Regulatory Commission Office for Analysis &amp; Evaluation of Operational Data Washington D.C. 20555</td>
<td>Tel. +1 301 492 4439</td>
<td>Fax. +1 301 492 8931</td>
</tr>
</tbody>
</table>

698
<table>
<thead>
<tr>
<th>Name</th>
<th>Company/Address</th>
<th>Phone/Mobile</th>
<th>Fax</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mr. M.A. SMITH</td>
<td>Commonwealth Edison Co.</td>
<td>Tel. +1 815 357 6761</td>
<td></td>
</tr>
<tr>
<td></td>
<td>LaSalle County Nuclear Station</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>PR#/1, Box 220</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>2601 21st Road</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Marseilles, Illinois 61341</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. M.P. MURPHY</td>
<td>Technicon Enterprises Inc.</td>
<td>Tel. +1 708 971 2700</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1401 Branding Lane, Suite 245</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Downers Grove, Illinois 60515</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. R.A. HILL</td>
<td>GE Nuclear Energy</td>
<td>Tel. +1 408 925 5388</td>
<td>Fax +1 408 925 2476</td>
</tr>
<tr>
<td></td>
<td>175 Curtner Ave., M/C 482</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>San Jose, California 95125</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. C.L. THIBAULT</td>
<td>WYLE Laboratories</td>
<td>Tel. +1 205 837 4411</td>
<td>Fax +1 205 721 0144</td>
</tr>
<tr>
<td></td>
<td>WYLER Laboratories</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>7600 Governors Drive</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>P.O. Box 7777</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Huntsville, Alabama 35807-7777</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. J.J. BALASCHAK</td>
<td>Teledyne Brown Engineering</td>
<td>Tel. +1 508 748 0103</td>
<td>Fax +1 508 748 2029</td>
</tr>
<tr>
<td></td>
<td>513 Mill Street</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Marion, IA 02738</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. R.C. CALLAWAY</td>
<td>Nuclear Management &amp; Resources Council</td>
<td>Tel. +1 202 872 1280</td>
<td>Fax +1 202 785 1898</td>
</tr>
<tr>
<td></td>
<td>177E Eye Street, N.W., Suite 300</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Washington D.C. 20006-3706</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. M.S. ELIDIWAN</td>
<td>Kalsi Engineering Inc.</td>
<td>Tel. +1 713 240 6500</td>
<td>Fax +1 713 240 0355</td>
</tr>
<tr>
<td></td>
<td>745 Park Two Drive</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>SugarLand, Texas 77478</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. J. HOSLER</td>
<td>EPRI</td>
<td>Tel. +1 415 855 2785</td>
<td>Fax +1 415 855 2774</td>
</tr>
<tr>
<td></td>
<td>3412 Hillview Ave.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Palo Alto, California 94303</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. P. McQUILLAN</td>
<td>Limitorque Corp.</td>
<td>Tel. +1 804 528 4000</td>
<td>Fax +1 804 845 9736</td>
</tr>
<tr>
<td></td>
<td>5114 Woodall Rd.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Lynchburg, Virginia 24551</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. J. NADEAU</td>
<td>ITI Movats/Westinghouse</td>
<td>Tel. +1 404 424 6343</td>
<td>Fax +1 404 429 4752</td>
</tr>
<tr>
<td></td>
<td>2825 Cobb International Blvd.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Kennesaw, Georgia 30144-4352</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. R. BAKER</td>
<td>ITI Movats/Westinghouse</td>
<td>Tel. +1 404 424 6343</td>
<td>Fax +1 404 429 4752</td>
</tr>
<tr>
<td></td>
<td>2825 Cobb International Blvd.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Kennesaw, Georgia 30144-4352</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. R. BARBIN</td>
<td>ITI Movats/Westinghouse</td>
<td>Tel. +1 404 424 6343</td>
<td>Fax +1 404 429 4752</td>
</tr>
<tr>
<td></td>
<td>2825 Cobb International Blvd.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Kennesaw, Georgia 30144-4352</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. G.H. WEIDENHAMER</td>
<td>U.S. Nuclear Regulatory Commission</td>
<td>Tel. +1 301 492 3839</td>
<td>Fax +1 301 492 3586</td>
</tr>
<tr>
<td></td>
<td>Office of Nuclear Regulatory Research</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Washington D.C. 20555</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mr. B. BLACK</td>
<td>TU Electric</td>
<td>Tel. +1 817 897 05545 or 0963</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Comanche Peak Steam Electric Station</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>P.O. Box 1002</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Glen Rose, Texas 76043</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Name</td>
<td>Organization</td>
<td>Address</td>
<td>Phone</td>
</tr>
<tr>
<td>--------------------</td>
<td>----------------------------------------</td>
<td>----------------------------------------------</td>
<td>-------------</td>
</tr>
<tr>
<td>Mr. R. W. MASSON</td>
<td>Manager Products</td>
<td>Teledyne Brown Engineering 513 Mill St. P.O. Box 288 Marion, Mass 02738-288</td>
<td>Tel 1 508 748 0103</td>
</tr>
<tr>
<td>IAEA</td>
<td>Mr. V. TOLSTYKH</td>
<td>International Atomic Energy Agency Wagramestrasse 5 P.O. Box 100 A-1400 Vienna Austria</td>
<td>Tel. +43 1 2360 6074</td>
</tr>
<tr>
<td></td>
<td>Mr. A. KOSSILOV</td>
<td>International Atomic Energy Agency Wagramestrasse 5 P.O. Box 100 A-1400 Vienna Austria</td>
<td>Tel. +43 1 2360 6074</td>
</tr>
<tr>
<td>OECD/NEA</td>
<td>Mr. J.P. CLAUSNER</td>
<td>OECD Nuclear Energy Agency &quot;Le Seine St. Germain&quot; 12, Blvd. des Iles 92130 Issy-les-Moulineaux France</td>
<td>Tel. +33 1 45 24 10 54</td>
</tr>
</tbody>
</table>