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July 15-18, 2002

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Edited by
T. G. Scarbrough
Division of Engineering
Office of Nuclear Reactor Regulation
U.S. Nuclear Regulatory Commission
Washington, DC 20555-0001

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Board on Nuclear Codes and Standards of
the American Society of Mechanical Engineers
Three Park Avenue
New York, NY 10016-5990

Abstract

The 2002 Symposium on Valve and Pump Testing, jointly sponsored by the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the U.S. Nuclear Regulatory Commission, provides a forum for exchanging information on technical and regulatory issues associated with the testing of valves and pumps used in nuclear power plants. The symposium provides an opportunity to discuss the need to improve that testing to help ensure the reliable performance of valves and pumps. The participation of industry representatives, regulatory personnel, and consultants ensures the discussion of a broad spectrum of ideas and perspectives regarding the improvement of testing programs and methods at nuclear power plants.

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Participation by international presenters and attendees provide a broad perspective to the issues currently under consideration in the United States. We sincerely appreciate the excellent work of Ms. Linda McKenzie and other members of the publications and graphics staff of the U.S. Nuclear Regulatory Commission in preparing the proceedings for the symposium. Finally, gratitude is expressed to all attendees and their sponsoring organizations for a successful and informative symposium

Disclaimer and Editorial Comment

Statements and opinions advanced in the papers presented at the Seventh NRC/ASME Symposium on Valve and Pump Testing are to be understood as individual expressions of the authors and not those of either the American Society of Mechanical Engineers or the U.S. Nuclear Regulatory Commission.

The papers have been copy edited and recast into a standard format. By consensus, English units have been used as an expression of current industry practice with metric units also indicated where possible.

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Chairman
U.S. NRC

Susan H. Skemp
President
ASME International 2002-2003

Jack R. Strosnider, Deputy Director
Office of Nuclear Regulatory Research
U.S. NRC

C. Wesley Rowley
Vice President
Nuclear Codes and Standards
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C. Richard Favreau, Chairman
ASME Committee on Operation and
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Session 1(a)

Pumps I

Session Chair

William A. Gates
Southern Company

An Overview of Age-Related Failures in Primary Coolant Pumps and Motors

H. L. Hassenpflug, Ph.D.
Framatome ANP

Abstract

The majority of U.S. nuclear power plants are now twenty or more years old. Issues related to the initial design of the plant, including equipment such as the main coolant pumps and motors have generally been resolved. However, conditions which have resulted for protracted periods of successful operation are now emerging, and are in some cases causing equipment failures and limiting power production. This paper examines case studies in which the author has been involved during the past five years.

Introduction

Five failure mechanisms are presented. For each, its background is discussed, then each is examined with regard to its potential to become a generic problem for plants (applicability), its nuclear and industrial safety implications (if any), the potential costs associated with the repair, executed preventively, on an emergent basis, including the loss of power production. Methods of detecting these failure mechanisms including vibration analysis are discussed for each.

The failure mechanisms considered are shaft cracking, motor flywheel looseness, impeller looseness, impeller cavitation damage, and restraint seizure

Discussion

1. Shaft Cracking

Historical Background

In the late 1980s and early 1990s, several plants experienced pump shaft cracks or complete failures.

The majority of these were pumps manufactured by Byron Jackson and KSB. However, there were some isolated instances of failure in Westinghouse pumps. At that time, the utility industry spent considerable money both in the development of methods for detecting shaft cracks, and in understanding and correcting the root causes. The failures which occurred in that time frame were generally considered to be premature, and the result of design and operational considerations. By the mid-1990s, corrections to the design and operation of the pumps in question had largely eliminated the failures in the Byron Jackson pumps, and the KSB failures were limited to one site. Hence, the underlying drive to develop new tools for early detection diminished.

Some of the technologies which were being developed for specialized application to shaft crack detection such as ultrasonic testing, modal analysis, and torsional analysis were largely abandoned due to lack of demand.

The primary means of detecting shaft cracking has remained the established method of vibration trending. This method of detection has proven to be a sufficiently reliable detector that plants can shut down prior to shaft fracture, and thereby avoid nuclear safety issues. However it has been shown to have serious limitations, in that in many cases a shaft crack will not begin to alter the vibrational performance in a way that is clearly identifiable until failure is imminent (days to weeks away). In recent years, there has been renewed interest in the issues associated with shaft cracking since several Westinghouse pumps have begun to have cracks.

Unlike the failures of a decade ago, the pumps currently failing have operated successfully for twenty or more years. A clear consensus has not yet emerged as to whether these failures represent a trend in aging reactor coolant pumps.

Significance of Failure

Nuclear Safety

A complete shaft failure in an operating pump is, of course, most undesirable in that there is the potential for a loss of primary coolant. Although current vibration monitoring technologies do not typically give the advance warning desired from an economic point of view, a cracked shaft can virtually always be detected before a complete fracture occurs. It should further be pointed out, that in two pumps which underwent complete shaft fractures, there was not a significant loss of primary system coolant. Because of these considerations, nuclear safety is rarely the limiting factor which governs the need to address shaft cracks.

Economic Consequences

A shaft failure at operation has substantial economic consequences. Typically, a plant which is well-prepared to replace a pump, and does so in a 'dedicated' forced outage might expect a two- to three-week outage duration.

This assumes that the shutdown was timely and therefore no other components were damaged as a result of the shaft failure. In previous cases where the pump shaft has failed completely, there have been additional economic costs: The shaft fractures have not been perpendicular to the shaft axis. As a result, continued rotation of the motor against the fracture surface of the fully severed shaft has caused a cam-like action, forcing the motor rotor upward and damaging the upper thrust bearing and thrust runner.

Detection

Insights to the relationship between phase and shaft crack propagation

For several decades, it has been established that one of the early signs of shaft cracking is shifting of phase angle in the synchronous (1X) and twice synchronous (2X) vibration responses. Usually one or both of these vibration components sees amplitude growth as well. The extent to which each of the components which characterize a failure appear, can vary substantially from one type of pump to the next. For example, it has been observed in at least one shaft failure (75% depth), the remaining portion of the failed ligament remained approximately circular, even though that remaining portion was not concentric with the shaft centerline. In that case, the mechanism which has historically caused a 2X vibratory response in equipment with a failing shaft did not develop.

The development of the 2X response depends on the continuous loading of a shaft with a non-symmetric cross-section. Even though the remaining ligament was not concentric to the shaft centerline, it remained axially symmetric. Hence, the development of the 2X vibration was substantially lower than is commonly seen in such an extensive failure.

The observed trends may also differ depending on the axial location of the failure. That is, knowledge of vibration behavior during previous failures, even in similar pump designs, does not ensure the ability to predict a new wave of failures. It is quite possible, and even likely, that failures which develop after a long period with no failures, are the result of a previously unobserved mechanism. Crack detection in older equipment is further complicated by the presence of other vibration sources. Shaft cracking may be overlooked because of high vibration responses resulting from other issues which may or may be otherwise innocuous. For example, in one plant with Westinghouse pumps, a pump was removed from service because of very high vibrations. The overwhelming cause of the vibration was found to be impeller damage due to cavitation. However, when the rotating components were examined destructively, a substantial crack was found in the shaft as well. Whether that crack would have been detectable in the absence of the cavitation damage is speculation. It quite clearly escaped detection as it was. Hence, the capacity of vibration monitoring to detect shaft cracking is limited in all types of equipment because of variability in the types of cracks which occur. It is further limited in older equipment by the presence of other vibrational issues. Still shaft crack propagation has vibrational characteristics which allow it to be distinguished from other failure mechanisms. The characteristic that distinguishes shaft cracking from most other failure mechanisms is that it results in continuous trending, the rate of which is exponential as failure becomes imminent. It will almost always involve changes in phase, at least in the 1X vibration component.

2. Flywheel Looseness

Background

This 'feature' has shown increasingly in some aging reactor coolant pump (RCP) motors of Westinghouse and Jeumont design. Some, but not all, of the design considerations which cause this phenomenon are present in motors of competitor designs as well.

It also may be expected to occur in other types of motors. The flywheels on most Westinghouse and Jeumont RCP motors are relatively thin and of a very large diameter. The objective of this design feature is to achieve the maximum polar moment of inertia with minimal mass. One effect of this design is that the flywheel bore expands substantially (0.05-0.10 mm) at operating speed.

To ensure reasonable flywheel-to-shaft clearances at operating speed, the vendors designed the fit as a light interference at installation. On most, but not all Westinghouse and Jeumont RCP motors, it is necessary to remove the flywheel to access the upper bearing for service. Removal of the flywheel frequently causes deterioration of the flywheel bore fit. Even where the bore fit does not deteriorate, there is typically some fretting wear of the flywheel bore since there is a clearance at operating speed. Hence, after several cycles of flywheel removal, it is common to see galled areas in the flywheel bore, and for the bore fit to have clearance at installation. The resulting clearance at operation may be twice the design value.

Significance of Failure

Nuclear Safety

While the flywheel is usually considered as safety-related, there is no evidence that loosening of the bore fit diminishes structural integrity or its ability to control the pump coastdown time. Hence there is no apparent reason that the loosening of the flywheel bore should affect nuclear safety.

Potential for damage

The primary result of flywheel looseness (in terms of potential equipment damage) is the damage caused by the vibration itself, typically high bearing wear or the masking of other vibration issues. This is no history of or significant potential for major equipment damage as a direct result of operation with a loose flywheel. No plant of which the author is aware has had a forced shutdown as a result of flywheel looseness alone. A consideration, perhaps of greater significance from the perspective of safe plant operation, is that vibration changes due to flywheel looseness may be confused with other vibration issues.

Cost of Repair

The repair is typically addressed during an offline refurbishment, so there is typically not a cost associated with lost operating time due to flywheel looseness. The direct costs of repair can vary substantially, however. The repair methods to date have typically included the use of oversized keys. The cost of re-keying is minimal compared to the overall cost of a motor refurbishment.

For more extreme cases, a special replacement thrust runner can be fabricated which interlocks to the flywheel, wherein the function of centering the flywheel is transferred to the bore fit of the thrust runner. (NOTE: This design upgrade has been also been implemented in some newer motors at manufacture.)

Detection

A loose flywheel will show up in the vibration data, typically as a discrete shift in the 1X vibration level. This will be most noticeable at the upper motor (if multiple shaft probes are available). The shifts will frequently occur following an electrical transient event which imparts a 'torsional impulse' load into the motor.

The other key characteristic of flywheel looseness is that the 1X vibration level will often change following shutdown of the machine. It may suddenly 'jump' back to previous levels following re-start or it may not change until influenced by some external perturbations.

3. Pump Impeller Looseness (Compared To Flywheel Looseness)

There have been several cases of impeller loosening reported in Westinghouse pumps. The design of the impeller attachment is similar in Bingham, Jeumont and KSB pumps as well. Therefore, it seems reasonable that this failure mechanism has equal likelihood in those designs as well. Because of radiological considerations, these cases are less well-substantiated than the cases of motor flywheel looseness. While, at first glance, impeller looseness would seem to be very similar to flywheel looseness, the designs of the two interfaces are quite different.

Their failures are manifested somewhat differently in operating behavior, including vibrations. The two interfaces differ in the following ways:

- The impeller is typically installed on a taper using a heavy interference fit, whereas the flywheel uses a cylindrical bore and a very light interference fit.

- The impeller transmits a large amount of torque at steady state conditions which stabilizes its position, even when loose. Conversely, the flywheel only transmits inertia torque, which is non-zero only during transient events which influence the rotational speed.

- The impeller has little centrifugal expansion compared to that of the flywheel. The impeller is virtually never removed for routine service.

- The flywheel is much more massive (typically five times) than the impeller for comparably sized equipment. Owing largely to the other differences already mentioned, impeller loosening is a much rarer occurrence than flywheel loosening. The impeller is typically installed with two keys, 180 degrees apart, whereas the flywheel typically uses three keys.

Detection

These two phenomena are manifested similarly in vibration data in that the 1X vibration component undergoes sudden changes. On impellers with two keys, slippage is necessarily along the plane of the keys. The resulting vibration shifts caused by the movement of loose two-key impeller will always have the same phase (or 180 degrees out). Vibration shifts caused by a 3-key loose flywheel can be at various phase angles. The vibration shifts caused by a loose impeller are, of course, more obvious in the pump vibration signal than in the motor vibration data. The opposite is true for a loose flywheel. These distinctions provide a basis on which one can test for flywheel looseness.

Significance of Failure

Potential for Damage

Impeller looseness, like flywheel looseness, is only likely to cause damage in the long term because of the vibration levels themselves. Even when loose, the impeller is held in place by the high torsional loads it transmits. Further, impeller looseness occurs without any loosening of the impeller capscrew.

Economic Considerations

The repair of a loose impeller requires the removal of the pump internals, and the costs associated with it. Typically, the internals are replaced, and any refurbishment performed off-line.

4. Impeller Cavitation Damage

Background

Impeller cavitation and the damage it causes can occur in nearly any pump. It is the result of operation at conditions where, at least locally, the absolute pressure of the pumped fluid is zero, and where there is a discontinuity in the pressure profile as a result. That is, for the pressure profile to be continuous, the local pressure would have to be negative over some portions of the surface. This is, of course, not possible.

This condition will occur wherever the combination of dynamic and static pressure would be less than zero. For pumps such as reactor coolant pumps, the potential for cavitation may be driven by low system (static) pressure and/or high runout conditions (excessive flow and resultant low dynamic pressure). Several plants have performed inspections of their reactor coolant pump impellers. Some pumps have been inspected in-situ using a specially-designed video camera which enters the impeller through the suction piping. Pumps removed from service have generally been examined directly. In one plant which had Westinghouse 93A pumps, damage was observed on the top side of the vanes (looking at the pump in cross-section), 1-2 centimeters from the leading edge of the vane. The damage resulted in the removal of material from the vane surface, forming a 'dished' region on the 'back' side of each vane. A key feature of this damage as related to vibration characteristics is that, in some pumps, the extent of damage varied widely from among vanes.

Applicability

To date, severe impeller damage has been observed in the U.S. in some plants designed by Babcock & Wilcox. Others may however have similar vulnerabilities. The B&W-designed primary system uses two reactor coolant pumps per steam generator. During the plant startup and shutdown, the reactor may operate with only one pump per steam generator. The most extreme condition occurs at initial operation where the system pressure is low, and there is only one pump running in either loop. During a normal startup or shutdown sequence, these conditions would be expected to persist only for minutes to a few hours. However, over decades of operation, the accumulated time in single pump operation has, in some cases, been sufficient to result in severe mechanical damage. This may be compounded by the fact that many plants (of all designs) follow a specific startup sequence, so that a particular pump will endure most of the damage. Plants with one pump per steam generator have lower reverse flow at off-design conditions, and therefore do not exceed the design flow rates to the extent seen in the B&W-type loops.

Some B&W plants have modified operating procedures to minimize operation at off-design conditions. This has usually been done to resolve concerns other than cavitation. These plants are less likely to see cavitation damage than others who have not implemented such a modification.

Significance of Failure

Nuclear and Safety Considerations

Until cavitation damage is severe, there is little change in the operation of the pump except for that caused by deteriorating vibration performance. This is often compensated by trim balancing at the pump coupling. The hydraulic deterioration of pump performance is minimal, even though there may be through holes in impeller vanes. In the most extreme cases, a large piece of the impeller vane may fracture and break free, traveling through the discharge piping until it comes to rest (probably in the lower plenum of the reactor).

Economic Considerations

Cavitation damage can become sufficiently severe as to render the pump inoperative. The costs associated with such a failure may involve a forced outage, and will certainly involve pump replacement or refurbishment. Further, if a plant has found one or more pumps with severe cavitation damage, it is prudent to consider modifying operating procedures to eliminate the conditions which cause it.

Detection

The attack of cavitation on impeller vanes varies widely, even in pumps of the same design in nearly identical applications. In some cases, vibration data can provide indications. Generally, other techniques are needed to provide a clear diagnosis of cavitation damage.

Operational Assessment

Successful identification of cavitation damage is best achieved by review of operating history for a pump which is suspect. If the pump has endured substantial amounts of off-design operation, or if the pump has been physically observed in cavitation on a frequent basis, then it is a more likely candidate for this failure mechanism.

Video Inspection

Several vendors have remote video inspection techniques available for the visual inspection of the impeller surface, including the backs of the vanes. This technique provides a good level of confidence in the impeller's condition, but is, of course, an off-line technique.

Vibration Analysis

The key feature which may make impeller cavitation damage detectable in vibrations is that the individual vanes are not necessarily attacked to the same extent. Since cavitation causes the loss of mass from the attacked vane(s), it can alter the mechanical balance of the impeller, and hence the 1X vibratory response. In pumps where the attack has been observed to be fairly uniform among all vanes, and there has been little change in the vibratory response. However, where the attack is preferential on a particular vane, it remains so as the impeller vane deteriorates. Hence, for a pump with 'preferential' cavitation damage, the balance quality of the pump will deteriorate over a period of years. Because the physical damage continues at the same vane, the phase of the increased vibration will have the same phase. Over a period of years, one would likely find the need to install ever-increasing amounts of balance weight, always at the same location.

Cavitation damage may further be distinguished from other phenomena: Unlike vibration caused looseness, the vibration due to cavitation damage does not cause discrete change under perturbation. Unlike shaft cracking, there is no phase shift involved, even to the extent that a large portion of a vane breaks free. The vibration signature due to preferential cavitation damage is virtually all 1X vibration, typical for any mechanism which causes deterioration of mechanical balance.

5. Support Seizure/ Forced Misalignment

Background

Under normal conditions, seismic restraints and vertical pump supports impose negligible (horizontal) structural loads on a pump. However, the very design function of these components implies that they are capable of carrying extremely large loads, and imparting those loads to attached structures. In some plant designs, the pumps are supported vertically by skirts. Other plants rely on the primary piping for vertical support, but will have some very large restraints designed to limit the horizontal movement of the pump during a seismic event. Both of these types of restraint systems are designed to accommodate the thermal growth/contraction of the primary piping during plant heatup/cooldown. If either type of restraint seizes, there is potential for the development of enormous loading.

Skirt supports have reportedly seized at several plants. Such seizures may cause spikes to appear in the vibration data during plant heatup and cooldown, and may damage the sliding mechanism, as well. However, for skirt supports, both the support and the loads due to piping expansion are applied to the pump casing, so that the entire region is at or below the lowest bearing in the shaft assembly. Therefore, the alignment of the bearings remains unaffected by seizure in a skirt supports. Comparatively, pumps which are piping-supported may have seismic restraints which are typically applied at points removed from the primary piping.

In a case familiar to the author, the restraints were located at the top of the motor stand, and extended horizontally parallel to the primary piping toward the reactor. This axial separation between the piping attachment and the restraint reaction allows potential for axial bending of the structure of the pump-motor assembly. In this scenario, the pump case and the motor stand subjected to bending loads. The motor stand typically has a much lower stiffness (in bending about the vertical axis) than does the pump casing.

Therefore, the seizure of a seismic restraint at the top of the motor stand can cause bending of the motor stand. Since the motor stand controls the motor-to-pump alignment in an RCP, seizure can result in an externally forced misalignment.

In one instance, a seismic restraint remained seized for the balance of the fuel cycle, approximately nine months.

Significance of Failure

Potential for Damage

The potential for damage from this type of mechanism is very high. Because it induces severe misalignment in the motor and pump, it has the potential to induce fatigue failures in the rotating components of motor and pump, possibly causing shaft cracking. Bearing failure is highly likely. Severe misalignment is known to cause seal damage, typically damaging the shaft O-rings. This type of damage has been observed in a pump which operated with a seized restraint.

Economic Considerations

For this failure mechanism, the cost of prevention is usually only the cost of ensuring the proper operation of the restraints. The cost of failure can be extremely high because it has the potential to induce numerous other failure mechanisms.

Nuclear Safety

Similar to the economic considerations, the potential nuclear safety concerns from this problem are the result of the other failure mechanisms which it can induce.

Detection

Detection of forced misalignment has proven more challenging than expected. In the case where data was most available, the forced misalignment caused the vibration probes at the lower motor bearing to move out of range. This led to the erroneous conclusion that the probes had failed. There were other minor changes in vibration levels, but these were (erroneously) attributed to flywheel looseness.

In retrospect, a change in the centerline position of the shaft occurred. This was overlooked because it occurred in a probe which was thought to be inoperative. In the case in question, oil analysis provided the first indication of a significant problem. Inspection revealed that the lower motor bearing had been severely overloaded. While it had not failed catastrophically, it had worn severely from being overloaded. For this failure mechanism, prevention is easier than detection.

Conclusions

Shaft Cracking

1. Shaft cracking remains a substantial concern in the operation of reactor coolant pumps.

2. Detection of shaft cracking still depends largely on vibration monitoring.

3. Vibration monitoring provides adequate advance warning of shaft failures to ensure nuclear safety.

4. Vibration monitoring gives less than the desired advance warning for economic considerations.

5. The detection of shaft cracks in aging pumps is complicated by the presence of other vibration sources.

Flywheel Looseness

1. Flywheel looseness is a common occurrence in aging reactor coolant pump motors.

2. The primary damage due to flywheel looseness is wear due to high vibrational loads.

3. Flywheel looseness can obscure other vibration data indicative of more severe problems.

Impeller Looseness

1. Impeller looseness has occurred much less frequently than flywheel looseness.

2. The vibration characteristics are similar to, but distinguishable from flywheel looseness or impeller cavitation damage.

3. Impeller looseness does not usually result in forced repair.

Impeller Cavitation Damage

1. Impeller cavitation damage is a problem which develops over long periods of service, particularly off-design operation.

2. In its early stages, there are few operational consequences to the damage.

3. In extreme cases, cavitation damage may require pump replacement.

4. Cavitation damage may or may not appear in vibration signatures.

5. Where it does appear in the vibration signature, cavitation damage is identifiable.

6. Methods in addition to vibration analysis should be used to verify cavitation damage.

Support Seizure

1. For pumps with supports at locations away from the primary piping, a seized support may cause a forced misalignment of the machine.

2. Forced misalignment can cause severe damage to numerous components, including bearings, shafts and seals.

3. Forced misalignment may be difficult to detect. The key change in vibration monitoring is shift in the shaft DC centerline position (or gap voltage).

Analytical Modeling Used to Solve Troublesome Synchronous Vibration Problem in a Steam Generator Feed Pump

Dr. Maurice Adams, Machinery Vibration, Inc.
William A. Gates, Southern Company
Phillip W. Garrett, Southern Company

Abstract

At a U.S. utility, a troublesome synchronous vibration problem in the steam generator feed pumps was identified after installing proximity probes. The pumps had previously been instrumented with only accelerometers mounted on the bearing housings. One of the 4 installed pumps had experienced a turbine drive shaft break with internal pump damage, and on a separate occasion wiped journal tilting pad bearings without the accelerometers indicating any problems or concerns. The utility decided to add proximity probes to monitor direct shaft motion relative to the pump bearing housings. During startup after installation of the proximity probes, step changes in pump shaft motion and phase were noted. The shaft vibration was predominately synchronous, and reached a peak amplitude of 5 mils. The bearing clearance in the pump had been documented as 5.5-6 mils during the outage. Field troubleshooting identified that the vibration amplitude was cyclic. Daily changes in seal water flow/temperature caused changes in the shaft vibration amplitudes, but it was not clear if seal water flow or temperature was driving the vibration changes. Several modifications had been made to the pumps based on conventional wisdom. Improvements were made, but the root cause of the cyclic vibration changes remained a mystery. It was decided to use analytical modeling techniques to analyze the problem. It was determined that differential thermal expansion of the seal sleeves against the pump impeller could be causing the cyclic changes in pump vibration. A modification was proposed and test plan developed. The test plan was performed at the repair facility on the dynamic balancing machine. The tests were conclusive and validated the findings from the analytical model. The modifications have been successfully implemented in the field and the cyclic vibration eliminated.

This paper discusses the findings of the field troubleshooting, the analytical model and its findings, the modifications performed, the shop verification testing and ultimately the final results obtained at the plant.

Introduction

Vogtle Electric Generating Station (VEGS) is a facility south of Augusta, GA, licensed for operation to Southern Nuclear Operating Company (SNC). The plant has 2 units producing nominally 1200 megawatts electrical. Each unit has two, 50% capacity, steam generator feed pumps with turbine drives. The steam generator feed pumps are Pacific model 20x17 HVF, single stage, double suction. During startup of VGES Unit 2, after the fall 1996 refueling outage, the "2B" steam generator feed pump (SGFP) experienced unstable, high relative shaft vibration. At one point, the pump was tripped due to high vibration. The "2A" SGFP also experienced higher than desirable shaft vibration at certain conditions. The "2A" SGFP turbine had been overhauled during the outage. However, the only work performed on the pumps during the outage had been the installation of dual, radial (90 degree separation) proximity probes at the inboard and outboard journal bearings.

In January 1991, the "2A" SGFP turbine shaft had broken while in service. The break had occurred at the turbine shaft coupling. The shaft was repaired using a stub shaft to restore the shaft length geometry. During the investigation of the turbine shaft failure issues were raised regarding the geometry of the pump internals and the journal bearings. Modifications were made to the Unit 2 SGFP in the spring of 1992. Issues continued when journal pad damage was found on the "2A" SGFP in the fall 1993. The pumps had previously been instrumented with vibration probes on the bearing housings and it was decided to install dual proximity probes to enable the examination of the pump shaft vibration. Table 1 summarizes the history of the modifications performed and problems encountered previous to the 1996 refueling outage.

Initial Investigation

During startup of the Unit 2 SGFPs in the fall of 1996, the vibration in all cases was predominantly at the pump (1X) operating speed. The alarm level was originally set at 4.5 mils. The "2B" pump vibration was at acceptable levels until the pump speed increased to about 5300 rpm. At this speed, the relative shaft vibration exceeded the alarm level original alarm limit and it was decided to increase the limit to 5.0 mils peak to peak (p-p). There was a concern with the validity of the relative shaft vibration data. The reasons for this concern were:

- The pump bearing housing vibration showed very acceptable vibration at less the 0.12 in/sec peak. Thus, without the proximity probe data, the pump vibration appeared normal and there would not have been any concern with the pump.

Table 1: History of Modifications and Problems

Outage	Date	2B	2A
2R1	Fall 90	No Work	No Work
	Jan-91		Shaft Broke
2R2	Spr 92	A/B Gap Mod	A/B Gap Mod
		New Bearings	New Bearings
		Pump Reworked	
		Bearings Pinched	Bearings Pinched
		Reduced Brg Clearance	Reduced Brg Clearance
2R3	Fall 93	Inspected Brgs - OK	Pump Brgs Found Damaged
		SCS installed temporary proximity probes- Baseline Test	
2R4	Spr 95	No Work	No Work
2R5	Fall 96	Install Proximity Probes	
			Turbine Reworked

- The 2A and 2B turbine vibration were always at acceptable levels (less than 2 mils) during the startup monitoring.
- The calibration of the pump's proximity probes was in question since the pump shaft was stainless steel. Stainless steel requires a different calibration factor from the typical calibration factor of 200 mv/mil used for carbon steel shafts such as the turbine shaft.
- The pump shaft had been chrome plated at the journals. The eddy current (proximity) probe can see through a certain thickness of chrome to the fuse line. Since the fuse line is generally rough, the probe senses this irregular surface as runout. This runout is called electrical runout. Normally, the area seen by the probes is burnished to remove the chrome and fuse line, or the chrome is placed with sufficient thickness to prevent the probe from seeing through to the fuse line. Shop records showed no indication that either option was used on the 2A and 2B pump shafts.

The 2B outboard horizontal probe was replaced with the pump operating. In the process of replacing the probe, the replacement probe came in contact with the shaft and increased the shaft runout. This increase in shaft runout caused the outboard relative shaft overall vibration to exceed the 5 mils alarm and the pump was tripped by the control room.

With the 2B SGFP out of service, the pump bearings were inspected and shaft runout measurements made. A representative from the original equipment manufacturer (OEM) of the pump was on site to witness the inspection. The following conclusions were reached from this inspection.

- Bearings showed no damage with the tilt radial and thrust pads.
- Journal surfaces were clean with no obvious damage.
- Runout of the shaft in the vicinity of the proximity probes showed a mechanical runout of less than 0.5 mils p-p.
- The outboard proximity probes showed

Table 2: Summary of Events

	2B			2A		
Date	**Speed**	**Max Vib**	**Action**	**Speed**	**Max Vib**	**Action**
12-Oct	4330	1.5	Initial Startup			
15-Oct				3550	1.0	Initial Startup
18-Oct	5280	4.4	97% Power	5425	2.3	97% Power
			Increased Alarm to 5 mils			
21-Oct	5280	5.2	Replaced Prox Probe	5425	2.3	
			Vibration increased			
			Tripped Pump			
23-Oct			Installed Temp Prox Probes			
23-Oct	Unit Tripped due to 2B SGFPT blowout diagram - Loss of Vacuum					
25-Oct	5240	1.5		5300	4.0	Increased Alarm to 5 mils
25-Oct	5370	5.0	Step Change			
			Increased Alarm to 5.5 mils			
	5000	4.0	Reduced speed	6000	2.5	Increased speed
26-Oct	5100	5.0	Varied Oil Temp	5600	3.9	
			Increased Alarm to 6 mils			
6-Nov	5180	3.8	Reduced Seal Water Temp	5600	2.6	Reduced Seal Water Temp

a runout of about 1.5 mils p-p, indicating that most of the runout was electrical.

- The shaft to bearing clearance was measured at 6 mils.

To help resolve the concerns with the proximity probe instrumentation, it was agreed to perform the following prior to the restart of the 2B pump.

1. Install temporary proximity probes at the inboard and outboard bearing in the same manner as done in the baseline testing of fall 1993. The purposes of these temporary probes were two fold.
   - Provide a direct comparison of the present proximity probe measurements with the baseline measurements of fall 1993.
   - Provide an entirely independent set of proximity probe instrumentation for comparison with the permanent proximity probe instrumentation.

4. Perform insitu calibration of both the temporary and permanent pump proximity probe instrumentation.
   - The insitu calibration determined that the pump's permanent probes had the correct sensitivity (200 mv/mil).
   - The temporary probes were determine to have a sensitivity of about 270 mv/mil.

The Unit was restarted on October 25 and both 2A and 2B pumps were brought up to about 5300 rpm. At this time, the 2A pump had high vibration while the 2B pump had low vibration. While holding speed at about 5370 rpm, the "2B" SGFP took a step increase in vibration. The speed was quickly reduced to 5000 rpm with no immediate effect on the vibration level. After several minutes however, the vibration reduced below alarm levels. Figure 1 shows the step increase of the "2B" SGFP outboard horizontal proximity probe. It can be concluded from Figure 1 that the vibration was not related to the pump speed. The other probes (temporary and permanent) showed similar step increases in vibration. Figure 2 shows a time plot of the same event.

Tables 3 and 4 are summaries of the "2A" & "2B" SGFP 1X proximity probe data. Included in these tables are the results of the baseline testing in fall 93. The slowroll (SR) or runout

Table 3: Comparison of Vibration - 2B SGFP

			1993 Baseline				10/18/96				10/25/96				10/27/96	
			SR		5300 rpm		SR		5300 rpm		SR		5200 rpm		5200 rpm	
Bearing	Dir	Probe	1X	Ph	1X	Ph	1X	Ph	1X	Ph	1X	Ph	1X	Ph	1X	Ph
Inbrd	Horiz	Perm	na	na	na	na	1.2	287	1.6	44	1.2	270	1.5	320	1.7	50
	Vert	Perm	na	na	na	na	0.7	23	2.2	130	0.8	0	1.5	45	1.5	125
Outbrd	Horiz	Perm	na	na	na	na	0.8	327	4.1	343	1.0	325	1.9	315	4.5	345
	Vert	Perm	na	na	na	na	0.8	45	3.0	73	1.0	45	1.6	50	3.6	70
Inbrd	Horiz	Temp	0.4	350	0.4	90	na	na	na	na	1.3	285	1.3	215	na	na
	Vert	Temp	0.5	230	0.4	215	na	na	na	na	1.2	50	1.4	340	4.1	15
Outbrd	Horiz	Temp	0.2	135	1.0	40	na	na	na	na	0.7	5	1.3	310	4.7	290
	Vert	Temp	0.2	25	1.2	140	na	na	na	na	0.8	190	0.5	50	3.5	50

Using 278 mv/mil for Inbrd temp probes
Using 270 mv/mil for Outbrd temp probes

Table 4: Comparison of Vibration - 2A SGFP

			1993 Baseline				10/13/96				10/25/96		10/27/96	
			SR		5900 rpm		SR		5300 rpm		5300 rpm		5600 rpm	
Bearing	Dir	Probe	1X	Ph	1X	Ph	1X	Ph	1X	Ph	1X	Ph	1X	Ph
Inbrd	Horiz	Perm	0.1	310	2.3	80	0.7	220	1.2	160	1.6	140	2.0	205
	Vert	Perm	0.2	75	2.1	180	0.7	310	1.1	245	1.3	215	1.8	290
Outbrd	Horiz	Perm	0.9	88	2.5	80	0.5	325	2.0	65	4.0	50	3.3	145
	Vert	Perm	0.8	180	2.1	180	0.4	40	1.2	130	2.6	110	2.1	215

is also included in the tables. Note that the inboard horizontal temporary proximity probe mounting became loose during the testing and has been shown as 'na' on the 10/27/96 summary. Several conclusions can be reached from these comparisons:

- The 2B SGFP vibration is significantly increased since the baseline testing.
- The temporary and permanent probes compared very favorably on the 2B outboard bearing.
- The 2A SGFP outboard horizontal probe vibration has increased since the baseline testing.

To provide conclusive evidence that the permanent proximity probes were providing accurate data and that the 2B vibration was excessive, absolute shaft measurements were made. An accelerometer was mounted on a wood dowel. The wood dowel was held against the exposed shaft of the 2B pump at the inboard and outboard bearing. The results are shown in Table 5.

The absolute shaft measurements were higher than the relative shaft measurements as would be expected. Thus, the 2B pump shaft is vibrating excessively and the permanent proximity probes appear to be providing reasonable measurements. The temporary proximity probes were removed.

Investigating Seal Flow

With the validity of the permanent proximity probes established, investigation of the high vibration amplitudes was continued. It was observed while reviewing the vibration data that the amplitude was cyclic. The vibration sample rate was changed from 5 minute intervals to 15 second intervals. A repeatable cyclic pattern was noted, with a periodicity of approximately 12 minutes. Figure 3 shows the

Table 5: Comparison of Absolute and Relative Shaft Measurement

	Absolute Shaft 1X Vibration		Horiz Proximity Probe 1X Disp
Bearing	Velocity (in/sec p)	(mils p-p)	(mils p-p)
Inboard	0.95	3.5	2.1
Outboard	1.35	5.0	4.9

1X trend of the 2B pump outboard horizontal probe vibration during this sampling. The other probes showed the same pattern.

It was immediately suspected that the automatic control system for the seal flow was causing this fluctuation in vibration. This parameter was not included in the process parameters monitored in the plant computer and previously not examined. The "2B" SGFP seal flow controls were changed from automatic to manual and the vibration amplitude became steady. The initial conclusion was that the periodic pattern in the vibration was due to the cycling of the seal flow automatic controls. However, with the 2B seal flow in manual control, the seal water temperature was reduced from 150 to 125 degrees and a sudden drop in vibration amplitude occurred (see Figure 4). Thus the only conclusion at this point is that the vibration is very sensitive to the seal flow.

Journal Bearing Investigation

The Vogtle SGFP has tilt pad type journal bearings. A properly designed, manufactured and installed tilt pad bearing offers superior rotor dynamic stiffness and hydrodynamic stability, particularly in high speed, lightly loaded applications such as the SGFP. The Vogtle SGFP also has fixed bushing seals with seal water injection, and as such the seals may act as hydrostatic bearings. Changing the seal flow affects seal chamber pressure, which affects the rotor dynamic radial stiffness of the seals and thus the response to the excitation. A lack of proper journal bearing stiffness could allow the fixed bushing seals to be the dominant rotor dynamic component and explain why the SGFP shaft vibration is sensitive to seal flow-by.

This scenario was supported by the pump shaft centerline plots, Figures 5 and 6. During the step changes in vibration, the shaft position within the bearing clearance moves significantly for both bearings. This amount of movement is unusual and indicates a stiffness change in the rotor dynamic system. Also, the seal water flow (pressure) was varied for both pumps which changed the seal water outlet temperature. The trend plots, Figures 7 and 8, show how the vibration varied with the changes in seal water outlet temperature or seal water flow (pressure).

Additionally, spare pump bearings in the plant warehouse were inspected. Two issues were identified with the spare bearings:

1. The tilt pads were not machined to provide bearing pre-load which sets the bearing radial stiffness, and
2. The thickness of the babbit was not uniform, which would result in a non-uniform heat transfer across the babbit.

A very important design parameter for a tilting-pad bearing is the pad preload. The oil film developed by a positively preloaded tilting-pad bearing provides the increased stiffness. A tilting pad bearing with a positive preload causes the oil wedge between the shaft and the bearing pad to become a converging/diverging gap. This creates a hydrodynamic force (load) on the journal in addition to the rotor weight. Also, with the individual pads equally spaced within the bearing, the converging/diverging oil wedge will always be present. This improves rotor stability, regardless of the direction and magnitude of the excitation force.

It was reasoned at this point that the first step in resolving this step change in vibration phenomena was to stabilize the rotor centerline position. It was recognized that a loose bearing as well as a cocked bearing could also affect the performance of the tilt pad journal bearings. It was decided to modify the bearing pre-load and change from a spherical seat bearing shell to a cylindrical seat with a .0005" to .001" interference with the bearing housing, in addition to increasing the bearing tilt pad pre-load. A spare pump rotating assembly was refurbished at the OEM repair facility and the rotating assembly was precision balanced. Great care was taken to mechanically eliminate any factor that could affect a synchronous vibration component. Confident this had been accomplished, the rotating element with modified bearings was installed during the May 1998 Unit 2 refueling outage. The modifications were successful in stabilizing the rotor centerline, however, shortly after return to 100% power the inboard bearing (IB) took a 2 mil step change. Four days later the IB bearing took another 1 mil step change. The vibration remained cyclic and appeared to be related to changes in the inlet seal water temperature as small as 10°F. It was decided to perform an analytical analysis of the rotor dynamic system.

Rotor Dynamic Analysis

The purpose of the rotor dynamic analysis was to identify and explain the primary cause(s) of cyclic rotor vibration excursions that were identified to be time synchronized with the cyclic changes in shaft seal injection water inlet temperature. The cyclic rotor vibration excursion typified in Figure 3, displays a generic behavior particularly pronounced on the Vogtle 2-B feed water pump. Two types of analysis modeling were employed: (1) Rotor unbalance vibration response, and (2) Rotor bowing from differential thermal expansion of shaft sleeves.

The initial step is to build the rotor dynamic model. A cross-sectional layout of the Vogtle feed water pump configuration is shown in Figure 9. Superimposed on this layout are the finite-element model station numbers (1 through 17) which delineate the end points of the 16 beam bar elements into which the rotor is sectioned for the rotor unbalance vibration analyses employing the "Rotor Dynamics Analysis" (RDA) computer code developed by Machinery Vibration, Inc., and provided with the purchase of the referenced text book [1]. The RDA model accounts for stiffness and damping characteristics of both journal bearings lubricating films and both wear-ring radial-clearance water annuluses. Initial unbalance analyses with trial weights were performed to first determine if there might be a critical speed near the operating speed. No critical speed was found at any speed below 6000 rpm.

Additionally, rotor unbalance vibration responses were computed to determine the maximum incremental unbalance vibration possible from bowed shaft sleeves. Maximum possible bowing of the shaft sleeves is 2.5 mils based on the sleeve-to-shaft radial clearance of 0.001 to 0.0025 inch from manufacturing tolerances for shaft outer diameter (OD) and sleeve inner diameter (ID). This gives 0.0025/3 in. as the radial offset of shaft-sleeve center-of-gravity, and is insufficient to significantly affect unbalance vibration.

Rotor bowing caused by shaft sleeve differential expansion was then considered. A hollow cylinder geometry was used to approximate the combination of two sleeves on one side of the impeller with individual nominal lengths of 9¾ and 7⅞ inches, for a combined axial length of 17¾ inches. Differential thermal expansion computations based on a 10 °F differential temperature swing between sleeves and shaft, corresponding to 10 °F seal-injection water drain temperature changes, for the geometry in steel was performed. The axial compressive force (F) necessary to prevent the differential thermal expansion was then computed. Since the shaft has several times the cross-sectional area of the sleeves, it was assumed as a reasonable approximation, that the shaft completely restrains the sleeves axially without any resulting differential shaft axial growth. Under perfect manufacturing and assembly conditions (i.e., no tolerances), the compressive restraining force (F) would be co-axial with the cylinder centerline (i.e., "best case" scenario). Under a "worst case" scenario, the force (F) would be centered at the outer radius of the cylinder (R ≅ 7½"/2). For the calculations in this analysis, a reasonable intermediate value of R = 3½"/2 is used. Since the answers so computed are linear to this parameter, results for any other assumed radial offsets for F can be directly proportioned.

Differential Thermal Expansion:

$$\Delta L = \alpha(1/\mathrm{F}^{\circ})\,\Delta\mathrm{T}(\mathrm{F}^{\circ})\,\mathrm{L(in)} = 7\mathrm{x}10^{-6}\,\mathrm{x}\;10\;\mathrm{x}\;17.75 = \underline{0.00125\text{ in.}}$$

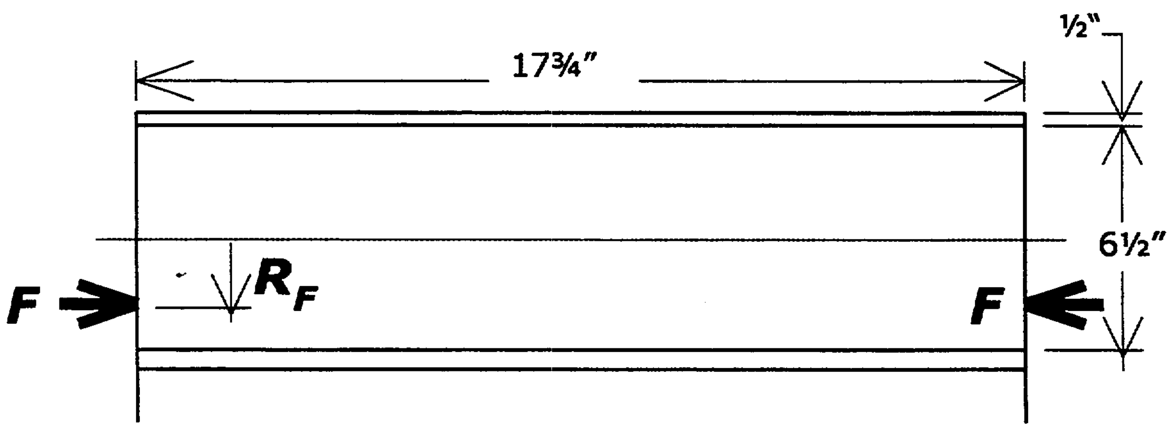
Compressive Force:

$$F = \sigma A = E\varepsilon A = E(\Delta L / L)A$$

$$= 30\text{ x }10^{6}(\text{psi})\text{ x }\left(\frac{0.00125\text{ in.}}{17.75\text{ in.}}\right)\frac{\pi}{4}(7.5^2 - 6.5^2)\text{in}^2 = \underline{23{,}230\text{ lbs.}}$$

Bending Moment:

$$M = F\text{ x }R = 23{,}230\text{ lbs. x }3.5\text{ in.}/2 = \underline{40{,}652\text{ in - lb}}$$



Bowing (Bending) Deflection Over Sleeved Section Of Shaft:

Bending area moment-of-inertia of shaft at sleeve locations is as follows.

$$I = \frac{\pi R^4}{4} = \frac{\pi}{4}3.25^4 = \underline{87.6\text{ in}^4}$$

$$y_{\max} = \frac{ML^2}{8EI} = \frac{40{,}652\text{ in - lb x }(37.5\text{ in.})^2}{8\text{ x }30\text{ x}10^6\text{ psi x }87.6\text{ in}^4} = \underline{0.0027\text{ in.}}$$

At impeller; both sleeve pairs in compression

For only one sleeve pair in compression, the maximum deflection over the 37.5 inch sleeved section of the shaft is determined by using the bending deflection of the $L = 17\ ¾$" bent length and the straight-line sloping deflection of the other 17 ¾" length section not in bending. Relative to a straight line joining the two axial end points of the 37.5 inch sleeved section of the shaft, the maximum radial deflection is also at the impeller, and is given by the deflection slope at the impeller times the $L = 17\ ¾$", as follows.

$$y_{\max} = \theta L = \frac{ML^2}{2EI} = \frac{40{,}652 \text{ in - lb x } (17.75 \text{ in.})^2}{2 \text{ x } 30 \text{ x} 10^6 \text{psi x } 87.6 \text{ in}^4} = \underline{0.0024 \text{ in.}} \left.\vphantom{\frac{A}{B}}\right] \text{At impeller; one sleeve pair in compression}$$

Based on the results of these analyses, it was concluded that the most likely cause of the cyclic rotor vibration excursions was shaft bowing caused by differential thermal expansion of the shaft sleeves. A positive 10 °F differential temperature of sleeves-to-rotor could produce a shaft bow in the range of 1 to 3 mils (radial). Unabated by bearing and seal straightening forces, this would translate into 2 to 6 mils Total Indicator Runout (TIR) (i.e., 2 to 6 mils peak-to-peak vibration). A worst case scenario is more than two times these numbers. This conclusion suggests the need to redesign both shaft-sleeve retaining nuts so that when the nuts are tightened, the shaft sleeves are not put into "stiff" compression.

The next step was to develop tests to challenge the basic conclusion that the cyclic variation of shaft-to-shaft sleeve differential temperature manifests as the cyclic increases and decreases in the Vogtle SGFP rotor vibration, particularly pump 2B. These tests were conducted in the Ingersoll-Dresser Pump (IDP, Charlotte, NC) repair shop using a spare Vogtle SGFP rotor mounted in a Schenck balancing machine. The essential unique feature of the test setup was the application of electric resistance heating elements arranged in close proximity to the shaft sleeves and housed in split cylindrical steel pieces that circumscribed the shaft sleeves on both axial sides of the impeller (see Figure 10). These heating elements were used to simulate the differential sleeve heating in actual pump operation. Furthermore, a pair of displacement proximity probes were installed (at 45° and 135° relative to horizontal) to target an impeller shroud OD cylindrical surface, to detect any significant rotor mid-span radial vibration response to the shaft sleeve heating. All tests were run below 1000 rpm, which is a significant factor since the pump operating speed is typically around 5000 rpm. However, the rotor first critical speed was well over 6000 rpm, and thus the low speed tests were considered reasonable approximations of the rotor response to unbalance or shaft deflection at the field operating speeds.

The initial test plan consisted of three scenarios: (1) both lock nuts securely tightened, (2) both lock nuts loosened, and (3) one lock nut re-tightened. To summarize, the tests showed a significant change in vibration amplitudes and phase angles as the shaft sleeves are heated by 10-15 °F, a temperature change commensurate with shaft sleeve differential heating during operational cyclic seal injection flow changes. These responsive vibration changes were primarily detected by the Schenck balancing machine real time readouts of indicated balance correction magnitudes and phase angles for both balancing planes. In general, the vibration changes were primarily "dynamic" as opposed to "static" unbalance changes, consistent with the rotor mid-span proximity probe measurements, which indicated only small radial vibration changes (approximately 0.2 mil p-p). The dynamic character of the vibration changes would be particular to the specific pump rotor shaft sleeve assembly tested. For other pump rotor shaft sleeve assemblies, the resulting unbalance change indications might be more static and less dynamic, and then would probably show a more significant radial vibration change at the rotor mid-span (impeller) axial location. As further confirmation of the effect of the shaft-sleeve differential heating, subsequent tests with both shaft-sleeve lock nuts loosened, did not show vibration changes comparable to the prior tests with both lock nuts shop tightened. Furthermore, subsequent tests with one lock nut re-tightened yielded vibration changes comparable to the first set of tests with both shaft-sleeve lock nuts tightened. It was concluded that the series of tests supported the findings of the analytical analysis that the cyclic rotor journal vibration excursions, often observed on the Vogtle SGFPs, was shaft distortion caused by differential temperature between the shaft sleeves and the shaft.

IDP proposed a design retrofit (Figure 11) which incorporated a lock nut configuration that inserts axially into an ID recess machined in the outboard end of each shaft sleeve. Between the end of the lock nut and recess machined into the sleeve, Grafoil packing was inserted. The lock nut tightens against the Grafoil packing instead of direct against the shaft sleeve. The packing provides a sufficiently soft axial clamping force so that shaft distortions are not caused by differential temperature between the shaft sleeves and shaft. The proposed design modification was implemented on the spare Vogtle SGFP rotor and tested as before. The rotor with the modified lock nut and sleeve responded identically to the tests where the lock nut had not been tightened. The rotor did not respond to the temperature differentials between the shaft sleeves and shaft. Based on these test results, it was decided to proceed with installation at the plant.

The shaft sleeve and locking nut modification has been installed on two Vogtle SGFPs. The vibration amplitudes were less than 1 mil on the first pump (1B) & less than 2 mils on the second pump (2A). The vibration amplitudes have remained steady on both pumps, being unaffected by fluctuations in seal water temperature.

Conclusions

The VEGS SGFP synchronous vibration problem provides an example where analytical modeling was an effective troubleshooting tool. The analytical model helped to identify and resolve a recurring component reliability issue at VEGS. However, before the results of analytical model could truly be conclusive several additional pieces of the puzzle had to be gathered or established, such as

1) Vibration data,
2) Pump mechanical condition (i.e., fits and tolerances),
3) Impeller and diffuser geometry (i.e., A-gap, B-gap, Overlap),
4) Bearing performance (pre-load),
5) Detailed rotor analysis, and
6) Shop testing of the conclusions.

With this information, it can be concluded that the ever-present manufacturing tolerances mean that the clamping force on the shaft sleeves probably have a radial offset in any such assembly. Therefore, unless the shaft sleeves are "softly" clamped, differential thermal expansion between sleeves and shaft are likely to impose some temperature sensitive shaft bowing. This conclusion was supported by the shop testing and ultimately with the successful implementation of a design modification to give "soft" clamping of the shaft sleeves.

These pumps routinely had challenged unit reliability at VEGP. The pumps required extra operational, management, and maintenance attention over many years. The use of analytical modeling made it possible to identify the "root cause" of the problem, and in turn resolve the long standing operational issues with the Vogtle feed pumps.

References

M.L. Adams, "ROTATING MACHINERY VIBRATION—From Analysis to Troubleshooting," Marcel Dekker, 2001, New York.

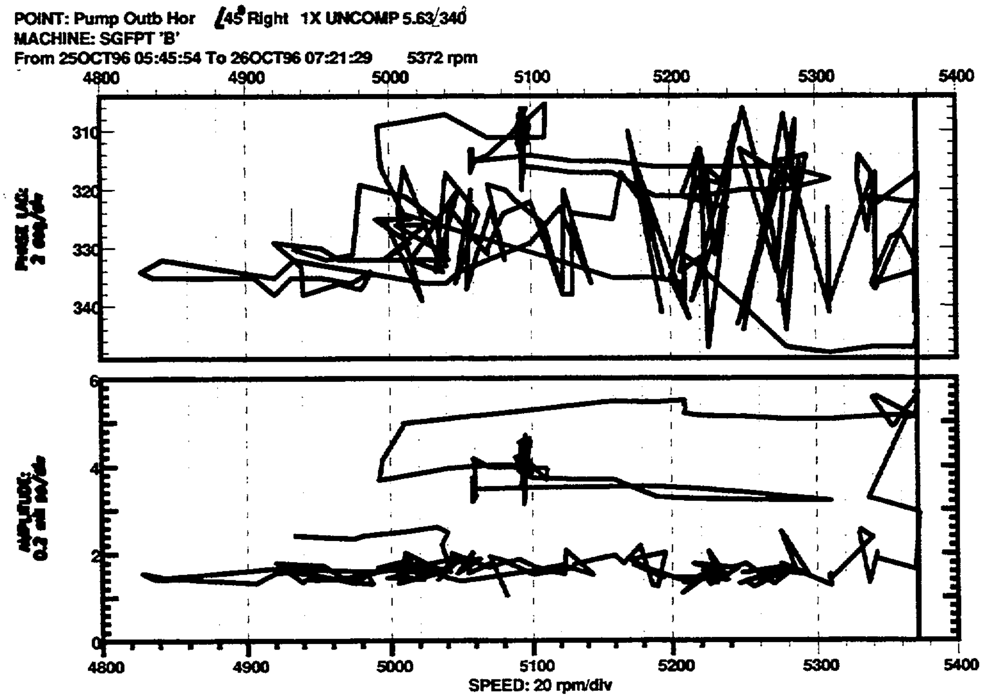


Figure 1. Speed Plot of Step Increase - 2B SGFP Outboard Horizontal Probe

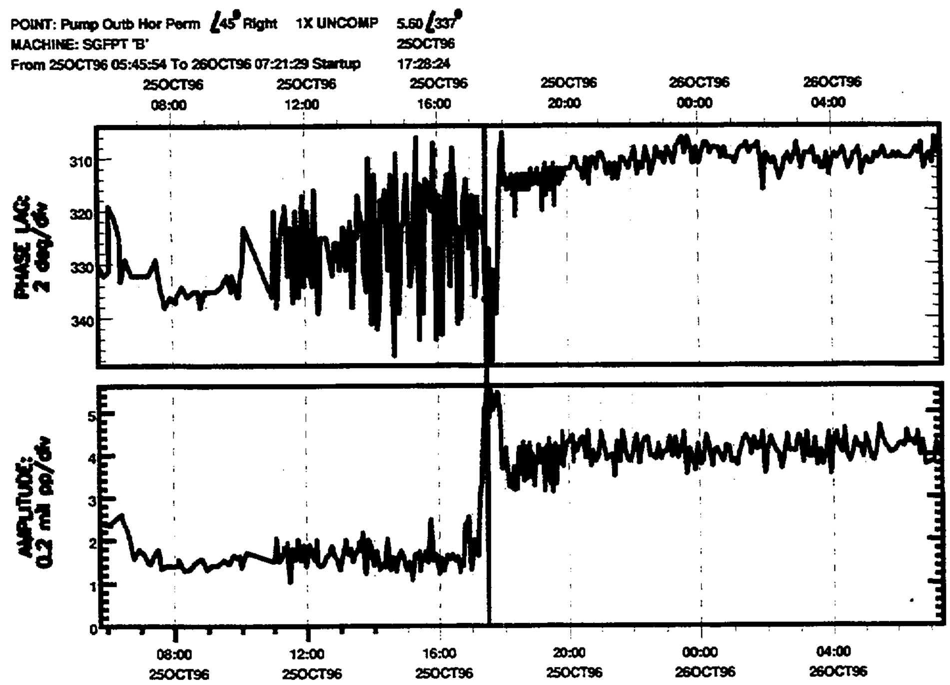


Figure 2. Time Plot of Step Increase - 2B SGFP Outboard Horizontal Probe

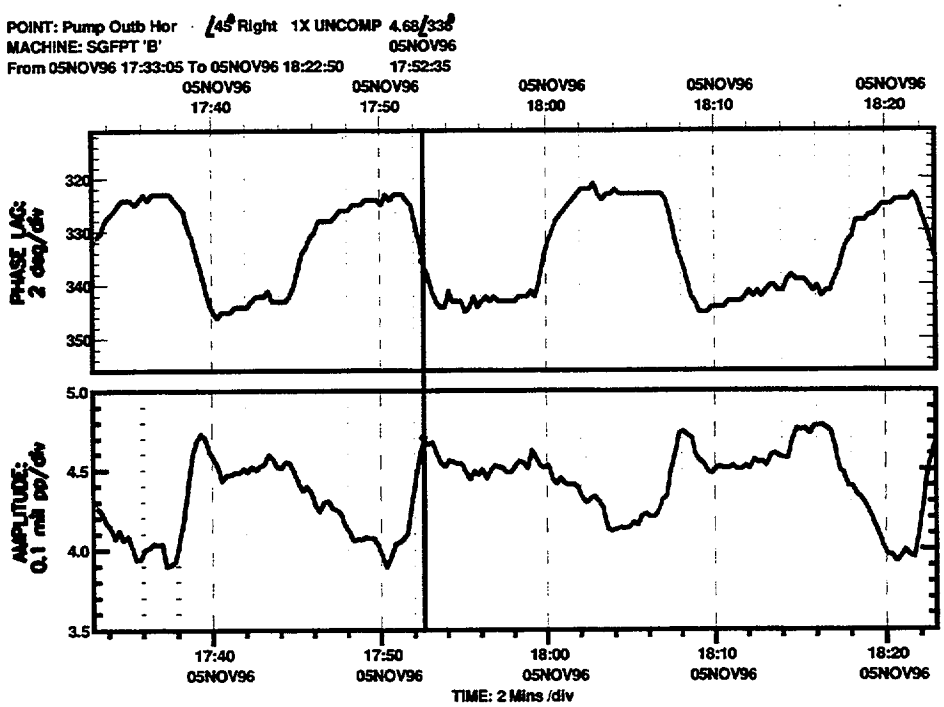


Figure 3. 1X Trend at 15 sec. Sampling Intervals

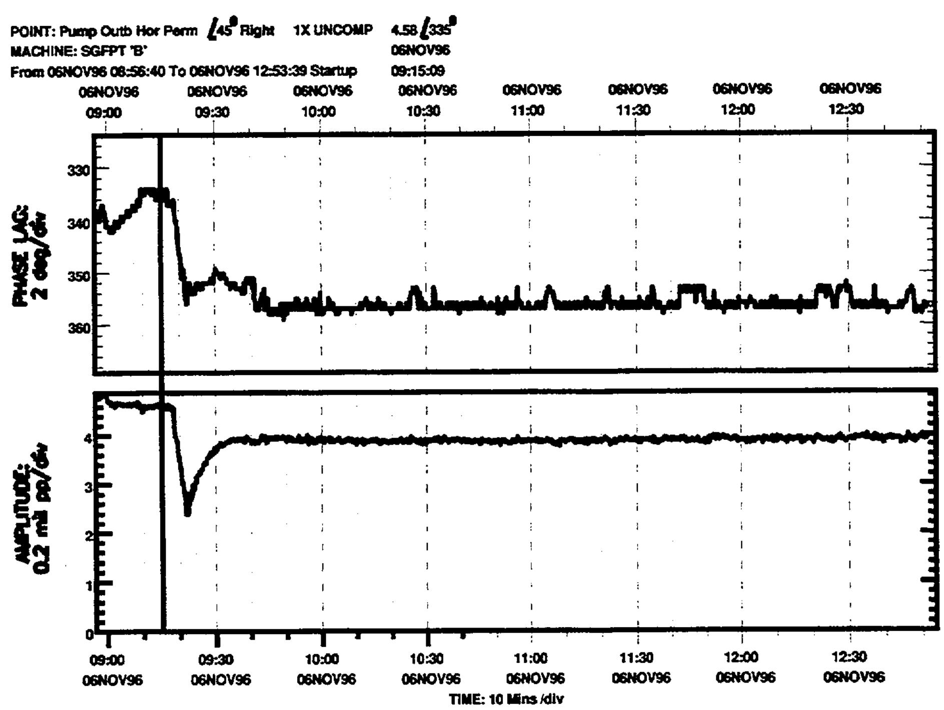


Figure 4. Reducing 2B SGFP Seal Water Temperature

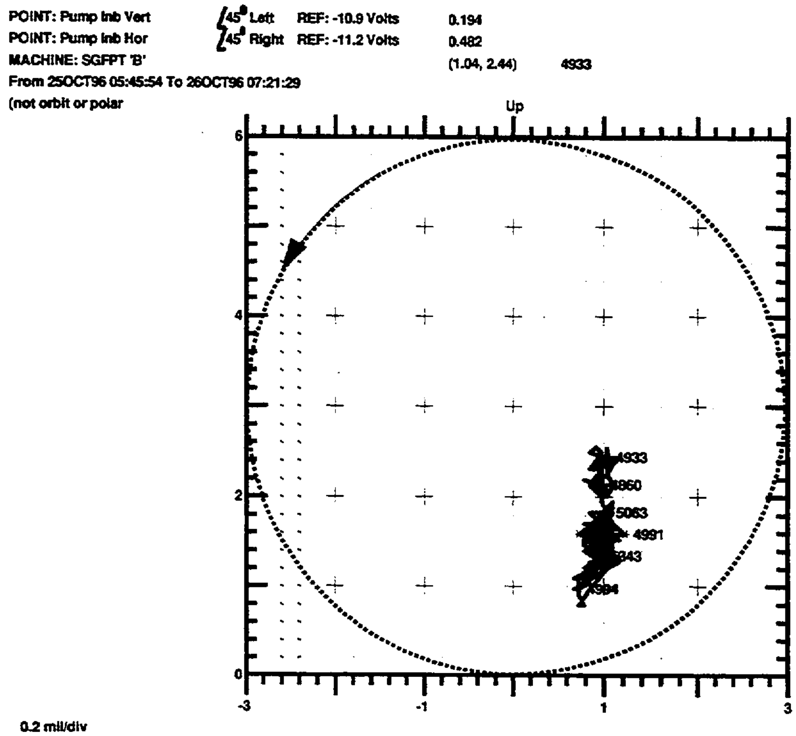


Figure 5. Shaft Centerline Plot During Step Change in Vibration - 2B Inboard Bearing

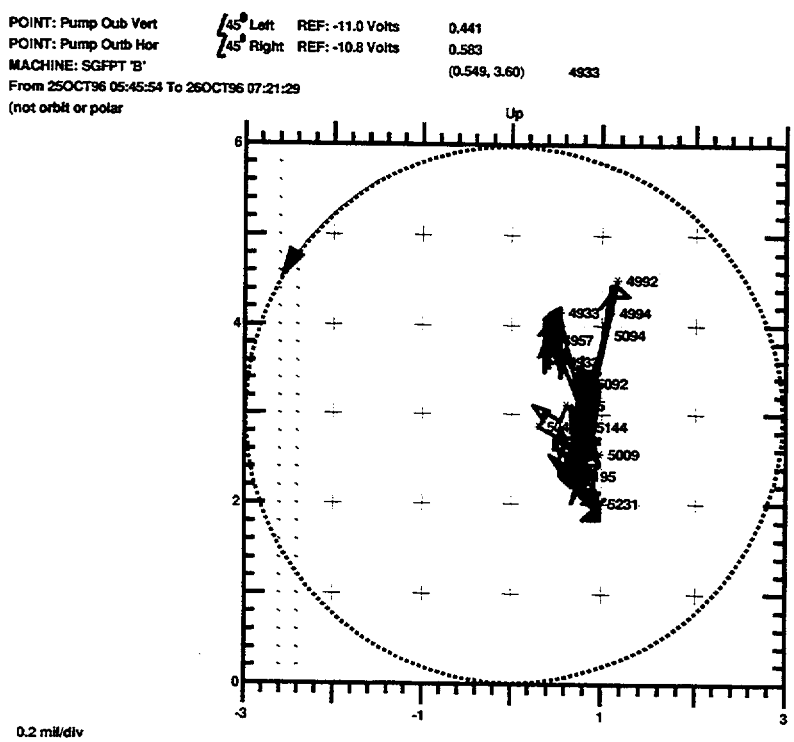


Figure 6. Shaft Centerline Plot During Step Change in Vibration - 2B Outboard Bearing

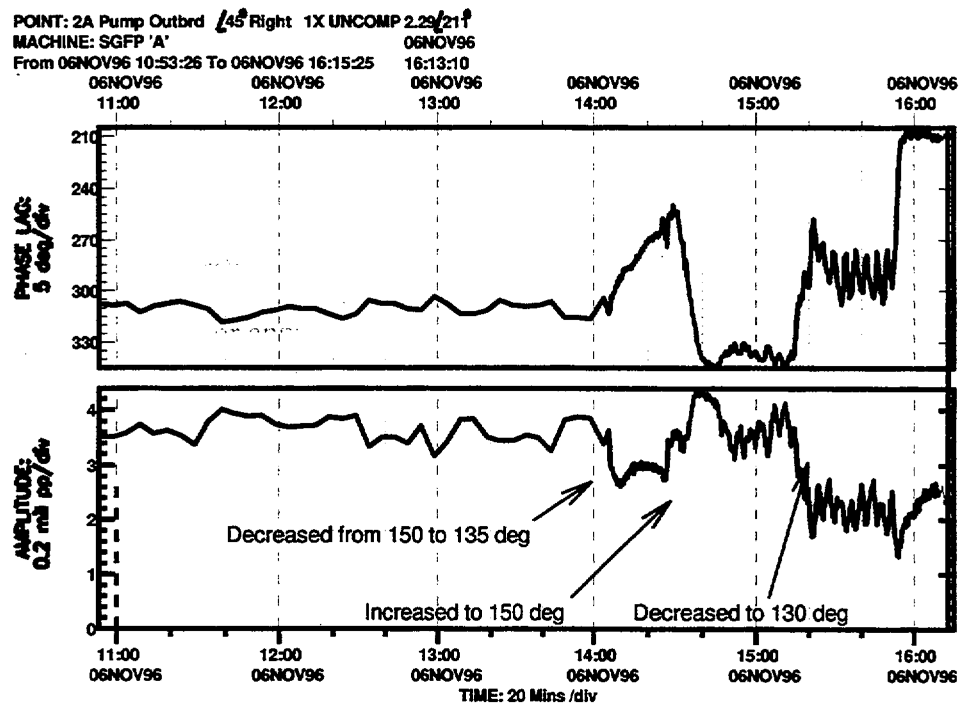


Figure 7. Varying Seal Water Temperature - 2A SGFP

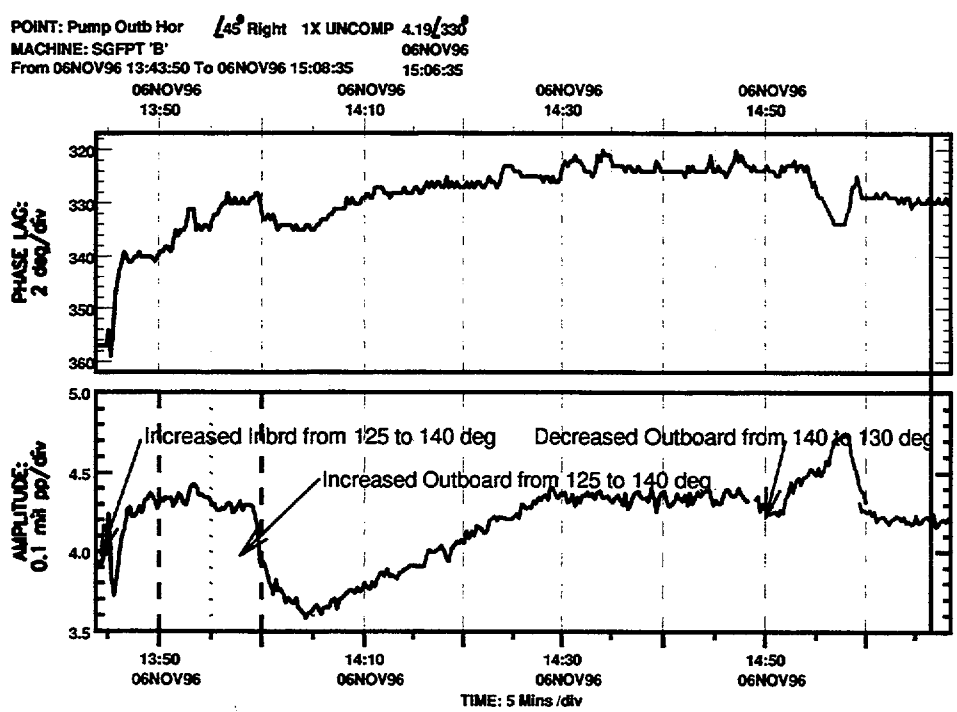


Figure 8. Varying Seal Water Temperature - 2B SGFP

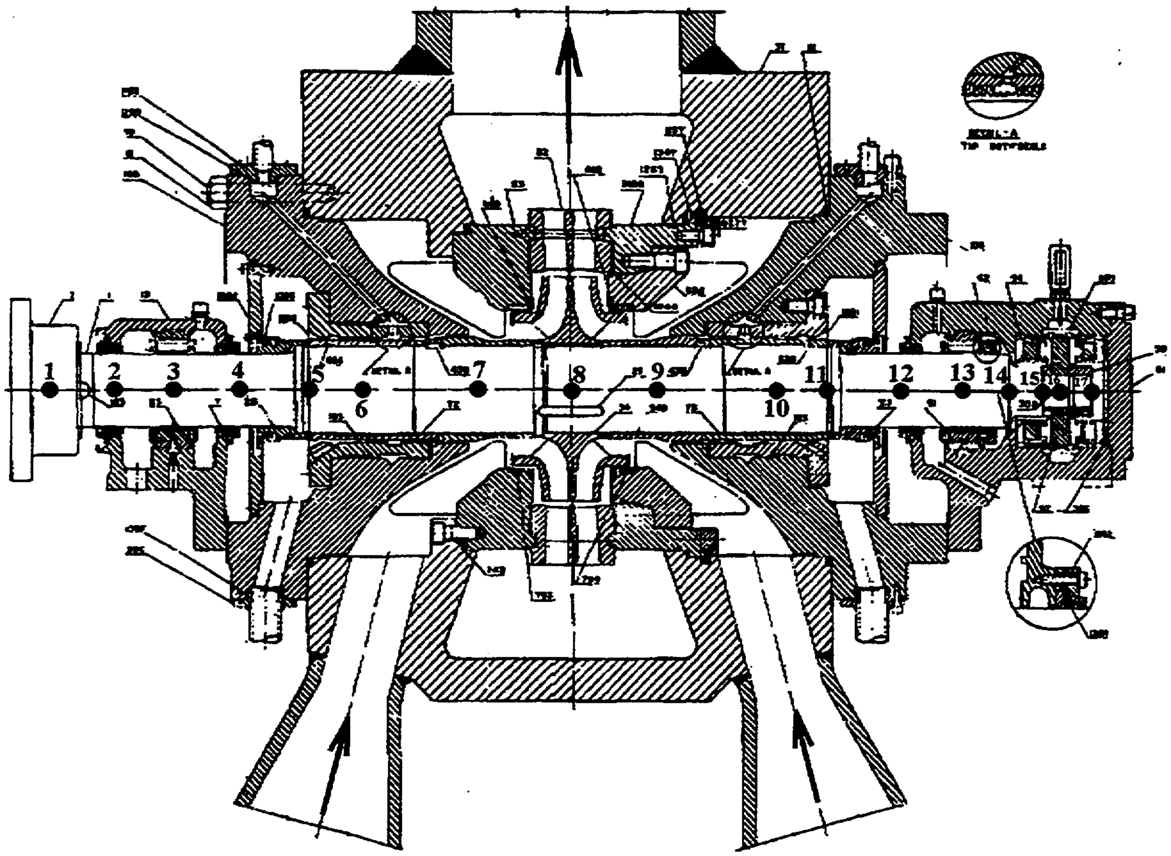


Figure 9. Vogtle feed water pump configuration

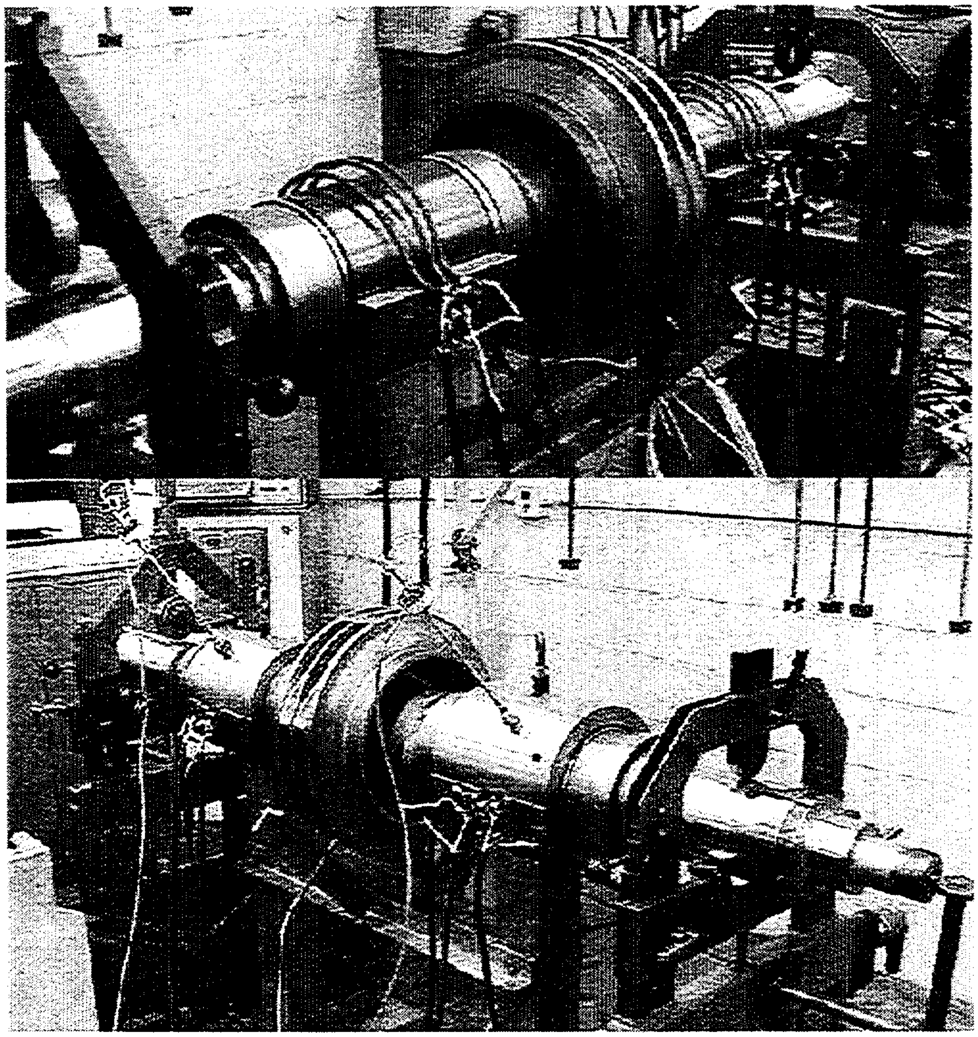


Figure 10. Test Stand Setup with Heating Coils, Temperature Probes, and Proximity Probes on Schenck Balance Stand

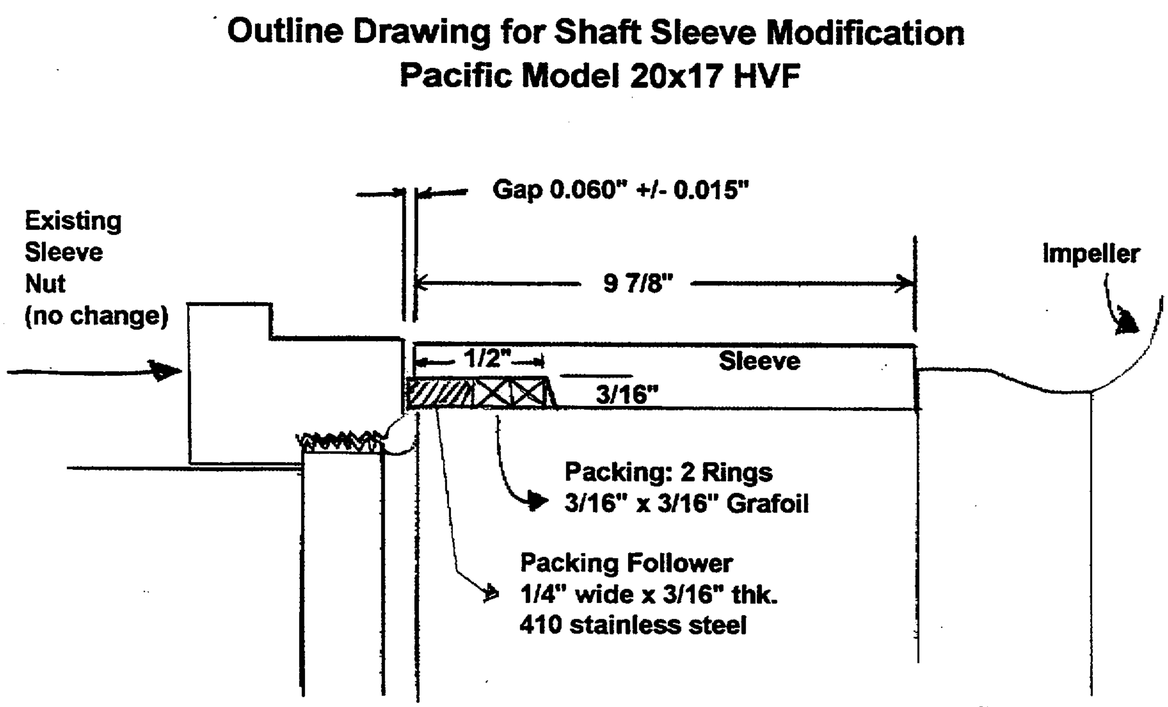


Figure 11. Proposed Modification for Shaft Sleeve and Sleeve Lock Nut

Field Balancing Reduces Excessive Synchronous Vibration on Centrifugal Charging/High Head Safety Injection Pumps

Thomas B. Lantrip and William A. Gates
Southern Company; and
Kevin L. Glandon
Southern Nuclear Operating Company

Abstract

Southern Nuclear Operating Company (SNC) purchased a total of three new centrifugal charging/high head safety injection (HHSI) pump rotating assemblies for two plant sites. The rotating assemblies were purchased from the original equipment manufacturer (OEM) as direct replacements of the installed rotating assemblies. The shaft material had been upgraded from Type 414 stainless steel to Nitronics 50, based on recommendations from the OEM and the Westinghouse Owners Group (WOG). The shaft material upgrade was recommended to address a problem of shaft cracking in this application. Even though the rotating assemblies had been dynamically balanced in accordance with OEM factory procedures, a potential synchronous vibration problem was identified during factory testing. Synchronous vibration refers to the vibration frequency related to the rotating speed of the pump. In situ balancing at the outboard seal sleeve collar was developed as an effective technique to reduce the identified synchronous vibration amplitude.

The centrifugal charging pumps are Model 2.5 RLIJ, 11 stage, horizontal, barrel type pumps. The pumps have different hydraulic performance (head/flow) requirements at the two plants; however, they are rotor dynamically identical. The first assembly was installed at the Farley Nuclear Plant (FNP) and required the addition of balance weights at the outboard seal sleeve collar to achieve acceptable vibration amplitudes. Subsequently, a second of the new rotating assemblies was installed at the Vogtle Electric Generating Station (VEGS). Again, the addition of balance weights at the outboard seal sleeve collar was required to achieve acceptable vibration amplitudes. The third rotating assembly was returned to the OEM to evaluate the dynamic balance of the rotating assembly. It was determined that the rotating assembly and individual components grossly exceeded the reported dynamic balance criteria. The third rotating assembly was properly balanced, then returned to the plant for installation. This assembly was installed and put in service without the need for field balancing due to acceptable vibration amplitudes.

Introduction

There are approximately 85 Pacific Model 2.5" RLIJ pumps currently in service at nuclear power plants in the U.S. They are used as the reactor chemical and volume control (charging) pump and/or as the high head safety injection (HHSI) pump. For some plants, the same pump serves both functions.

The pump is an eleven stage, horizontal, barrel case design (Figure 1) and, over the years, it has experienced many problems in this application. Problems experienced include shaft cracking, high vibration, bearing failures, and mechanical seal leakage. However, the dominant issue has been the problem of shaft cracking that dates back to the 1970s.

Eight charging pump shaft failures had been recorded in the nuclear industry by 1980. Five shaft failures had occurred in the threads for the pressure reducing sleeve lock nut, and three in the split ring grooves which axially located the impellers (twice at the 11th stage impeller location and once at the 4th stage impeller location). From 1982 to August 1999, nineteen additional charging pump shaft failures were reported. Nine failures since 1990 all occurred in the pressure reducing sleeve lock nut threads. Westinghouse and the pump OEM had recommended a series of shaft modifications in an effort to prevent shaft breakage. Prior to the change of material recommendation in 1997, the last modification had been in 1982.

Rev. 0 Original Shaft Material was ASTM A276, hardened, air cooled & tempered @ 1000°F

Rev. 1 Original Material with temper changed to 1150°F (10% endurance limit increase)

Rev. 2 Original Material oil-quenched instead of air cooled & tempered at 1150°F to 1200°F, August 1977

Rev. 3 Rev. 2 Shaft Material with shaft geometry & manufacturing changes (early 1978):

- Increased radius of split ring grooves (10% decrease in stress)
- Used formed tools to cut split ring groove radii
- Increased pressure reducing sleeve lock nut thread root radius (17% decrease in stress)

Rev. 4 Rev. 3 Shaft Design with lock nut design changes:

- Changed from one-piece design to two-piece design to more evenly distribute shaft thread loads (after April 1979)

Rev. 5 Rev. 4 Shaft & Lock nut Design with shaft geometry & manufacturing changes (November 1979 - Shaft drawing # D18844):

- Changed thread machining process from tool cutting operation to thread roll process (50% increase in fatigue endurance limit)
- Maximized thread root radius (15% decrease in stress)

Rev. 6 Rev. 5 Shaft (1982):

- Changed two-piece design lock nut back to a one-piece design with further enhanced shaft thread loading

Rev. 7 Rev. 6 Shaft (1997):

- Changed material to Nitronics 50, high strength or Custom Aged 625

It is interesting to note that all the shaft failures occurring since 1990 have been in the pressure reducing sleeve lock nut threads and that all the shafts were revision 4 or earlier designs. Therefore, it was prudent to upgrade rotating assemblies with revision 4 or earlier shaft designs with the upgraded shaft and material design. It was also decided to upgrade the shaft material during the normal course of maintenance replacement of the charging pump rotating assemblies. The new rotating assemblies purchased for FNP and VEGS had been upgraded to the revision 7 shaft geometry and Nitronics 50 material. Prior to the purchase of the new assemblies with upgraded shafts, both plants had previously installed new and refurbished rotating assemblies without problems of high vibration immediately upon installation.

In the fall of 1998, the first new rotating assembly with the Nitronics 50, shaft material was installed at FNP. When the pump was placed in service, the outboard bearing housing vibration in the horizontal direction measured .58 in/sec, exceeding In-Service Test program vibration requirements. Troubleshooting of the high vibration was performed and the basic conclusion was that the high vibration indicated rotor unbalance, as shown in Figure 2. Further examinations were performed to evaluate the integrity of the pump casing and foundation bolted connections, and the natural frequencies of the pump case and bearing housing. The information presented in Figure 3 identifies an apparent foundation looseness, which could have been attenuating the rotor unbalance response of the bearing housing. Natural frequency tests using impact test methods showed a natural frequency of the outboard bearing housing in the horizontal direction at 5040 rpm (see Figure 4) or within 5% of the pump running speed (4830 rpm). This natural frequency was the rocking mode of the pump/support structure. Attempts were made to reduce the outboard bearing housing horizontal vibration amplitudes by re-torque of the pump foundation and housing bolts, but were unsuccessful. Modifications to shift the bearing housing natural frequency further away of the pump operating speed were evaluated. Bracing the bearing housing and adding mass were possible, but considered a last resort due to time and cost.

During performance testing at the factory, the potential for a synchronous vibration problem was identified and it was determined that the pump synchronous vibration could be successfully reduced by balancing at the outboard seal sleeve lock nut collar. The lock nut collar has six set screws along its radial periphery. Balance corrections were successfully made at the factory by changing the weight (length) of the screws, which changed the weight distribution around the lock nut collar. With the concurrence of the pump OEM, it was decided to perform field balance corrections at the seal sleeve lock nut collar of the rotating assembly installed at FNP. By changing the weight of the set screws, the outboard bearing housing horizontal vibration amplitude was reduced to less than 0.10 in/sec, satisfying the In-Service Test program vibration requirements.

In the spring of 1999, a second of the new rotating assemblies was installed at VEGS. During functional testing of the pump at flow rates of 510, 170 and 85 gallons per minute (gpm), high vibration was measured on the outboard bearing housing in the horizontal plane. Reviewing the initial data, it appeared that the vibration could be flow dependent (see Figure 5). Subsequent testing indicated, however, that flow was not the primary source of the high synchronous (1X) vibration. With constant flow, the synchronous vibration would slowly increase during the initial 30 minutes of operation.

Impact tests identified a yawing natural frequency at 5202 rpm (86.7 Hz) or less than 8% above the operating speed of the pump (see Figure 6). The horizontal vibration was being influenced by this natural frequency. But as with FNP just months earlier, the vibration indicated rotor unbalance and that balancing at the outboard seal sleeve lock nut could be successful at reducing the vibration to acceptable levels.

It was decided to add an initial trail weight of 13 grams at the outboard seal sleeve lock nut. The pump was run with flow through the minimum flow line only, or approximately 60 gpm. The trial weight initially improved the vibration, but within a short period the vibration amplitudes began to rise. Analysis of the vibration data indicated a potential rub condition associated with the thrust bearing. Disassembly and inspection of the outboard thrust bearing was performed. It was determined that the holder for the thrust shoes had been cocked during installation. The condition was corrected by rounding the corners of the anti-rotation pin for the holders, such that the pin fit without binding in its slot in the bearing housing.

The pump was again placed in service, without making any changes to the trial weight. The pump vibration signatures continued to be characteristic of a rub condition. A second disassembly was performed which included the inboard and outboard bearings and mechanical seal assemblies. During the inspection, it was found that the inboard seal sleeve access plugs had lightly contacted the inboard seal housing (see Figure 7). It was noted that the outboard seal sleeve access plugs were flush with the seal face while the inboard plugs protruded from the housing approximately 1/4 inch (see Figure 8). The threaded holes for the seal sleeve access plugs were threaded deeper to allow the plugs to be set flush with the seal face and the pump was assembled. Following assembly, a third functional test was performed and the rub condition no longer existed. It was now believed possible to reduce the vibration by performing field balancing. Four balance moves were made, and the synchronous vibration amplitude was reduced to less than 0.10 inch/sec (see Figure 9). The overall bearing housing horizontal vibration was reduced from 0.50 inch/sec to 0.22 inch/sec, which satisfied the In-Service Test program vibration criteria.

At this point, two of three new rotating assemblies with the Nitronics 50 shaft material had synchronous vibration problems upon installation, and both had been indicative of an unbalanced condition. It was decided to send the third rotating assembly to an OEM repair facility, independent of the factory, for an evaluation of the rotor residual unbalance. The pump rotating assembly was disassembled and the rotating components (i.e., impellers, pressure reducing sleeve with lock nut, thrust collar with lock nut) were installed on the shaft. The assembled rotor was placed in the balance machine and the dynamic balance was checked. The left plane (first stage impeller) residual unbalance was 44.45 gm-in or 29.6 W/N and the right plane (eleventh stage impeller) was 50.75 gm-in or 33.8 W/N. The balance criteria for the factory balance had been 1 W/N. The thrust collar and lock nut were removed and the rotor balance checked. The left plane residual unbalance was 43.4 gm-in or 28.9 W/N and the right plane was 61.95 gm-in or 41 W/N. The pressure reducing sleeve and lock nut were removed leaving only the impellers on the shaft, and the rotor balance was checked. The left plane residual unbalance was 42.35 gm-in or 28.2 W/N and the right plane was 51.45 gm-in or 34.3 W/N. It was concluded that the thrust collar and pressure reducing sleeve had no appreciable influence on the rotor residual unbalance. It was decided to remove the impellers and check the residual unbalance of the individual impellers.

The results of the individual impeller balance checks are provided in Table 1. It can be concluded from Table 1 that the individual impellers were grossly out of balance. The impellers were individually balanced at the repair facility (see Table 2). Note an individual balance was not performed on the first and eleventh stage impellers, since these impellers are the balance planes during the assembled rotor dynamic balance.

The impellers were installed on the pump shaft and returned to the balance machine. The balance results are shown in Table 3.

The pressure reducing sleeve was installed on the pump shaft with impellers. Balance corrections were made in the pressure reducing sleeve only. The balance results are shown in Table 4.

The thrust collar was installed on the pump shaft with the impellers and pressure reducing sleeve. Balance corrections were made in the thrust collar only. The balance results are shown in Table 5.

At this point it was decided to accept the rotor as balanced. However, one question remained

Table 1: "As Found" Impeller Residual Unbalance

Impeller	**Plane 1**			**Plane 2**			**Static**		
Stage #	Unbalance (oz-in)	angle	W/N	Unbalance (oz-in)	angle	W/N	Unbalance (oz-in)	Angle	W/N
2	0.486	309	211	0.647	119	281	0.188	-88	81.9
3	0.827	175	360	0.732	322	318	0.452	57	196.5
4	0.249	164	108	0.25	347	109	0.013	71	5.7
5	0.528	54	230	0.816	220	355	0.329	17	143.2
6	0.464	280	202	0.37	95	161	0.101	-61	43.8
7	0.405	190	176	0.546	4	237	0.149	-12	64.9
8	0.633	30	275	0.674	218	293	0.100	-80	43.4
9	0.466	240	203	0.581	68	253	0.136	-84	59.1
10	0.757	147	351	0.705	351	307	0.308	78	134.0

Table 2: "As Left" Impeller Residual Unbalance

Impeller	Plane 1			Plane 2			Static		
Stage #	Unbalance (oz-in)	angle	W/N	Unbalance (oz-in)	angle	W/N	Unbalance (oz-in)	Angle	W/N
2	0.003	347	1.3	0.004	173	1.7	0.001	10	0.5
3	0.003	349	1.3	0.004	174	1.7	0.001	8	0.5
4	0.005	34	2.2	0.005	217	2.2	0.000	-55	0.1
5	0.003	325	1.3	0.004	149	1.7	0.001	-19	0.4
6	0.004	7	1.7	0.004	188	1.7	0.000	-82	0.0
7	0.004	355	1.7	0.004	183	1.7	0.001	89	0.2
8	0.002	351	0.9	0.003	180	1.3	0.001	17	0.5
9	0.004	313	1.7	0.004	141	1.7	0.001	47	0.2
10	0.002	25	0.9	0.002	212	0.9	0.000	-62	0.1

Table 3: Residual unbalance with impellers only

Condition	Plane 1			Plane 2		
	Unbalance (oz-in)	angle	W/N	Unbalance (oz-in)	angle	W/N
"As Found"	1.723	162	33	2.193	164	42
After Balance	0.034	133	0.64	.056	148	1.1

Table 4: Residual unbalance with impellers and pressure reducing sleeve

Condition	Plane 1			Plane 2		
	Unbalance (oz-in)	angle	W/N	Unbalance (oz-in)	angle	W/N
"As Found"	0.137	27	2.6	0.170	22	3.2
After Balance	2.297	54	1.5	0.036	161	0.7

Table 5: Residual unbalance with impellers, pressure reducing sleeve and thrust collar

Condition	Plane 1			Plane 2		
	Unbalance (oz-in)	angle	W/N	Unbalance (oz-in)	angle	W/N
"As Found"	0.068	79	1.3	0.108	259	2.1
After Balance	0.083	91	1.6	0.060	252	1.1

regarding the effect of rotor disassembly and assembly on the residual unbalance. The design of the pump requires that the rotor be disassembled after balance, so it can be stack assembled with the non-rotating components (e.g., diffusers, suction spacer, etc.) to complete the element assembly. Could this step adversely affect the residual unbalance of the rotating components? Therefore, the balanced rotor was disassembled, assembled and returned to the balance stand.

Comparing the data in Table 5 to Table 6, it was concluded that disassembly and subsequent assembly of the rotor did not have a significant effect on the residual unbalance of the rotor. The charging pump element was assembled and shipped to the plant for installation.

The third rotating element was installed at VEGS in the spring of 2001 without any vibration issues.

Conclusions

The first two replacement shafts installed and operated at FNP and VEGS exhibited high synchronous (1X) vibration. Subsequent vibration analysis showed that the unacceptable vibration was due to excessive rotor unbalance.

Natural frequencies and foundation issues were identified during the evaluation of the high synchronous vibration. Although these issues did contribute to the high synchronous vibration, rotor unbalance was considered the primary cause of the unacceptable amplitudes, and field balancing was performed to reduce

Table 6: Residual unbalance rotor after disassembly and assembly

Condition	Plane 1			Plane 2		
	Unbalance (oz-in)	angle	W/N	Unbalance (oz-in)	angle	W/N
#1-No Witness	1.632	321	1.1	1.727	275	1.2
#2 – Witnessed	2.604	265	1.7	2.562	266	1.7
#3 - Witnessed	2.196	265	1.5	2.345	265	1.6

the vibration to acceptable levels. The field balancing was performed at the outboard seal sleeve.

Evaluation of a third rotating assembly showed that the rotating assembly and individual components grossly exceeded the reported dynamic balance criteria from the OEM factory. The third rotating assembly was properly balanced, then returned to the plant for installation. This assembly was installed and put in service with acceptable vibration amplitudes without the need for field balancing.

When encountering high vibration amplitudes being influenced by a structural natural frequency, possible solutions include moving the natural frequency or reducing the forcing function. In this case, field balancing at the outboard seal sleeve, lock nut has proven to be very effective at reducing the forcing function caused by unbalance. However, it must be stated that the true culprit in the event was the failure to properly balance the individual impellers. Failure to individually balance the impellers can introduce coupled unbalance and possibly static unbalance following disassembly and assembly operations.

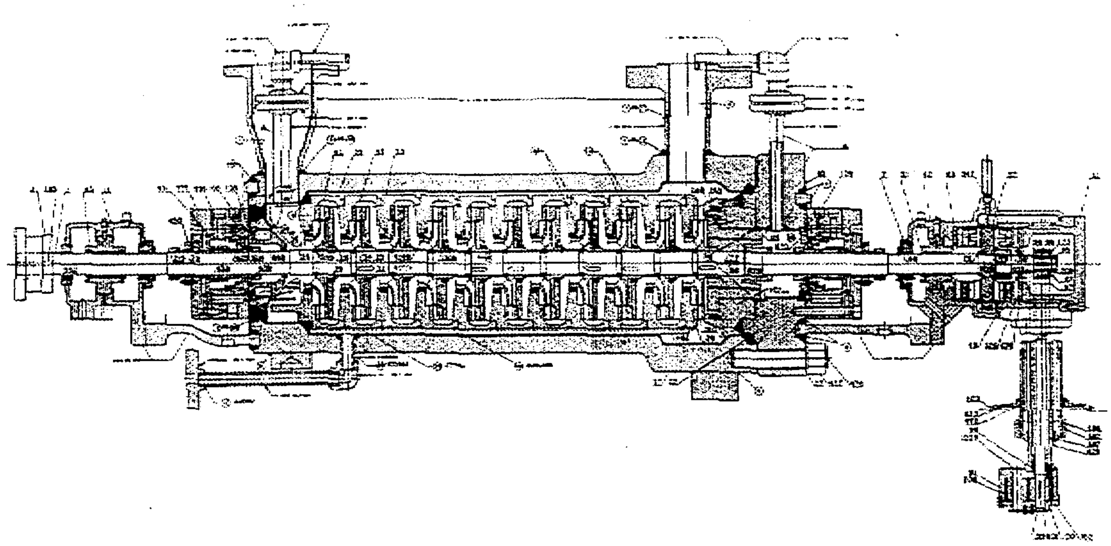


Figure 1. Cross Section of Typical Pacific Model 2.5" RLIJ , 11 Stage Pump in Charging/HHSI Applications

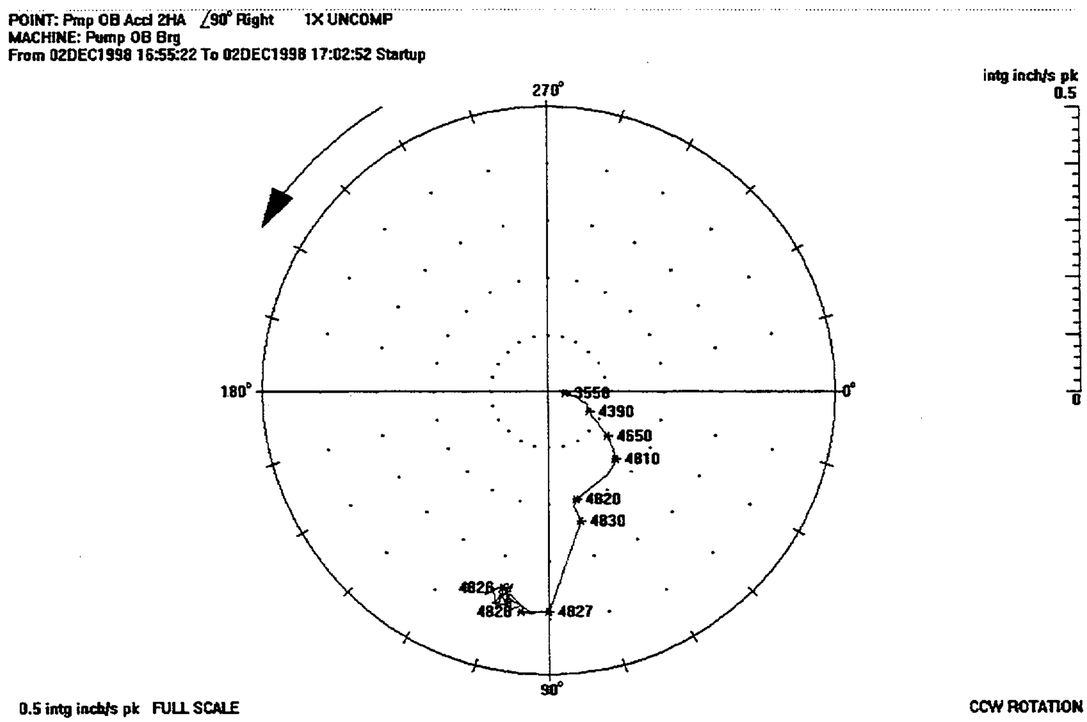


Figure 2. FNP charging pump outboard bearing housing vibration indicating unbalance response

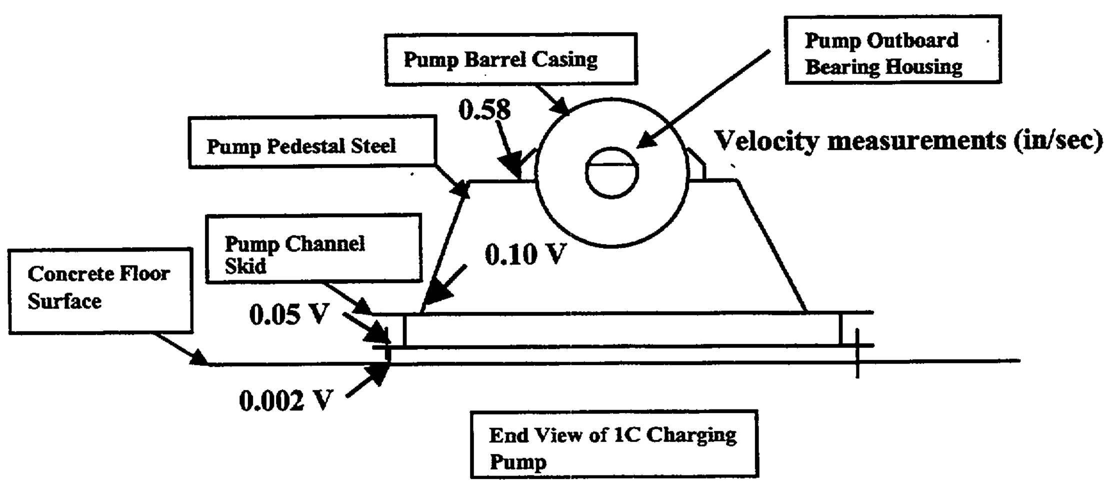


Figure 3. Vibration at FNP CCP Bolted Connections

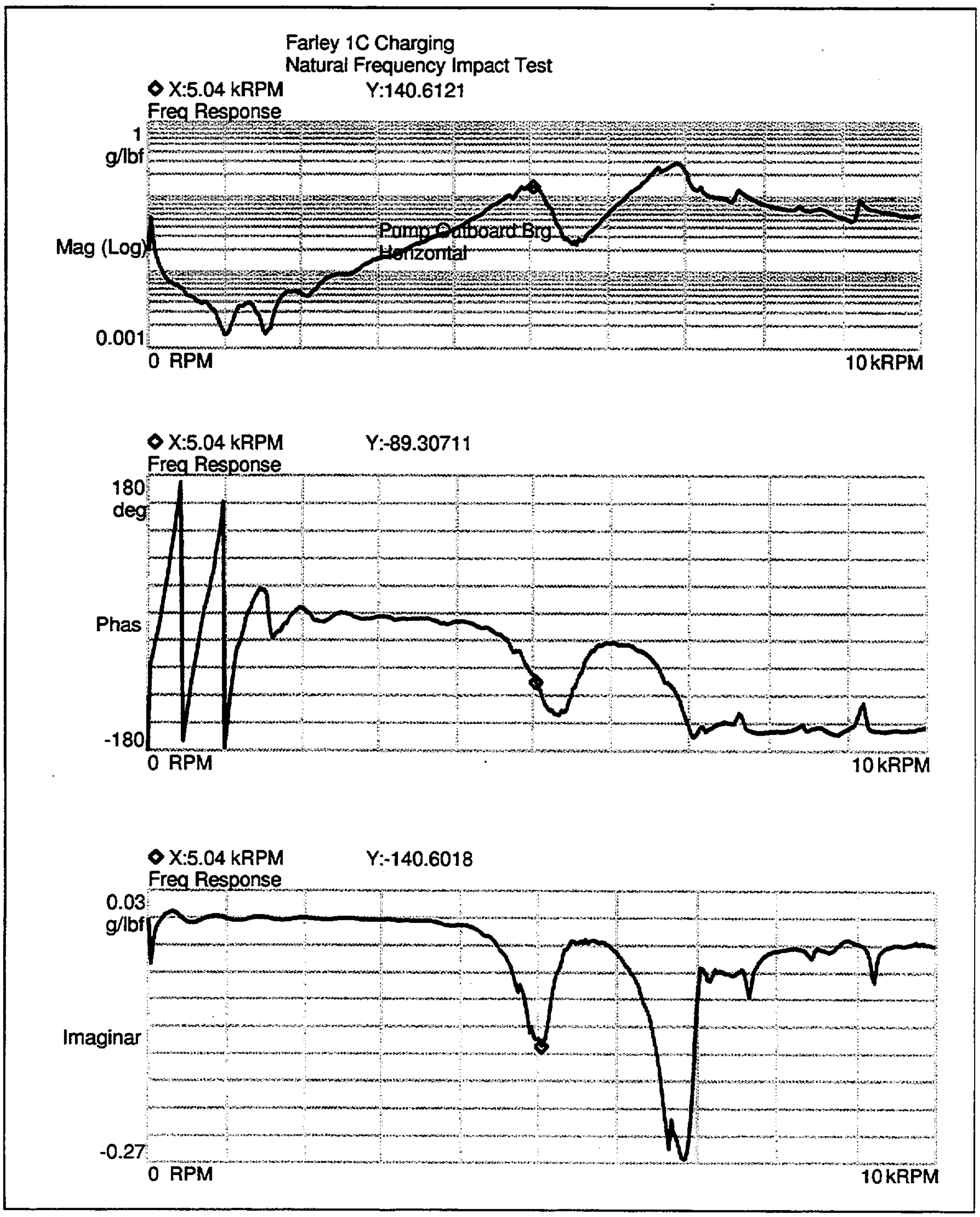


Figure 4. FNP CCP Outboard Natural Frequency Test Results

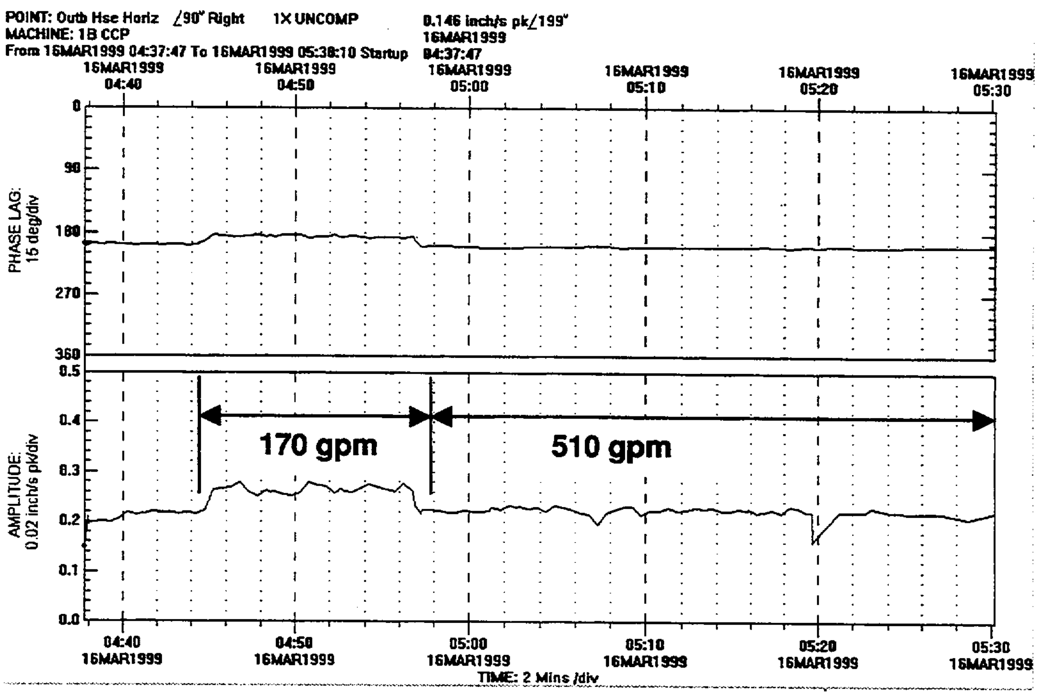


Figure 5. VEGS Initial High Outboard Bearing Housing Vibration

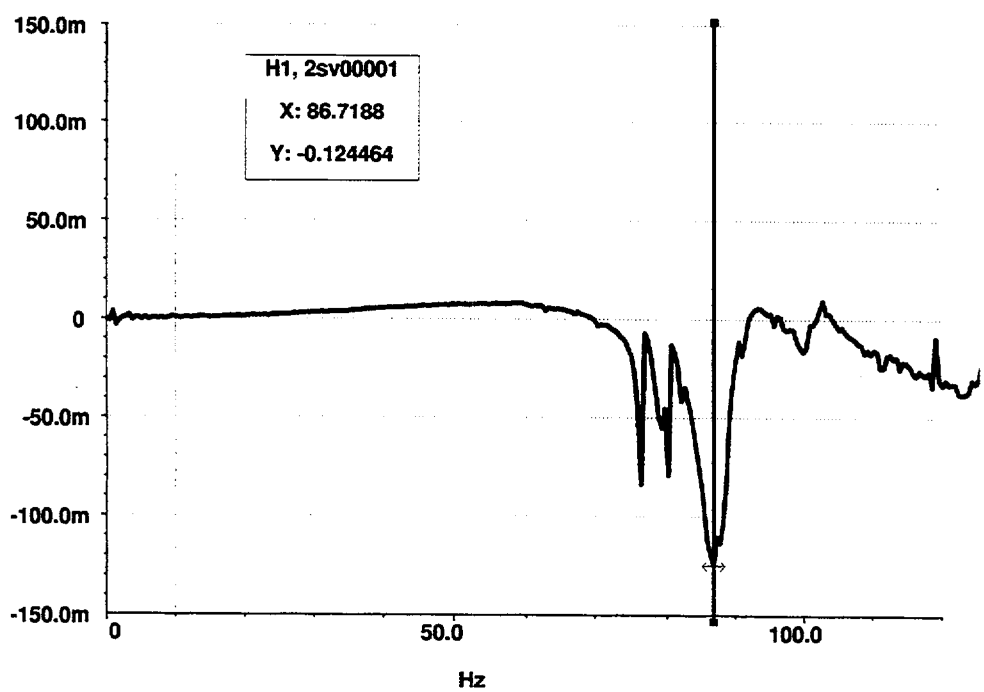


Figure 6. VEGS Outboard Bearing Housing Impact Test Results

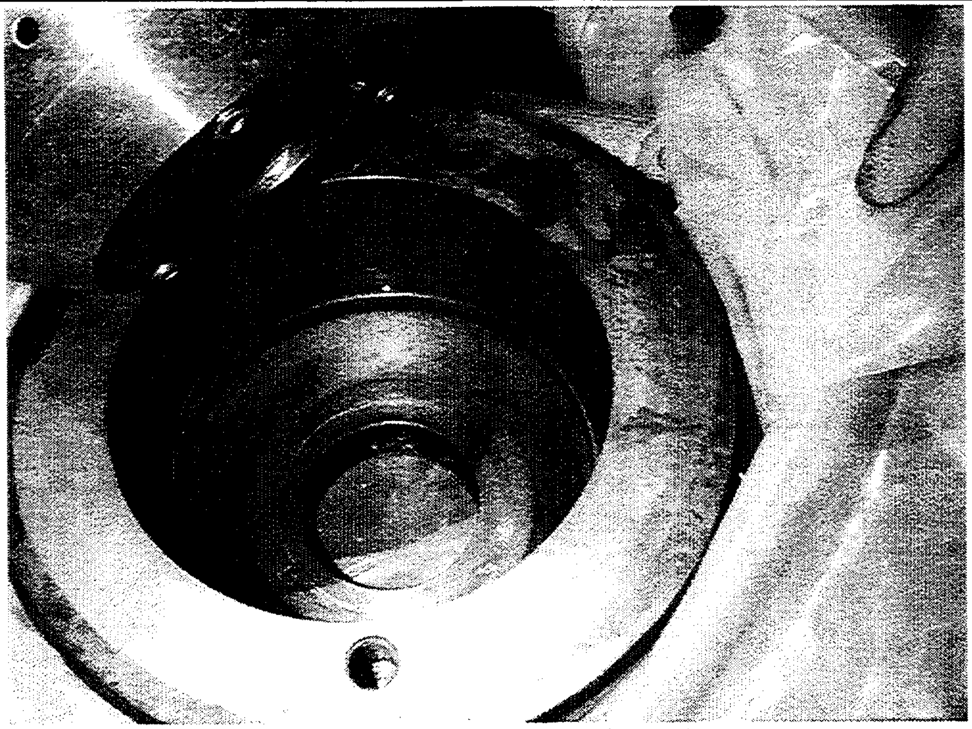


Figure 7. VEGS Inboard Seal Housing Rub

Figure 8. VEGS Protruding Inboard Seal Sleeve Access Plugs

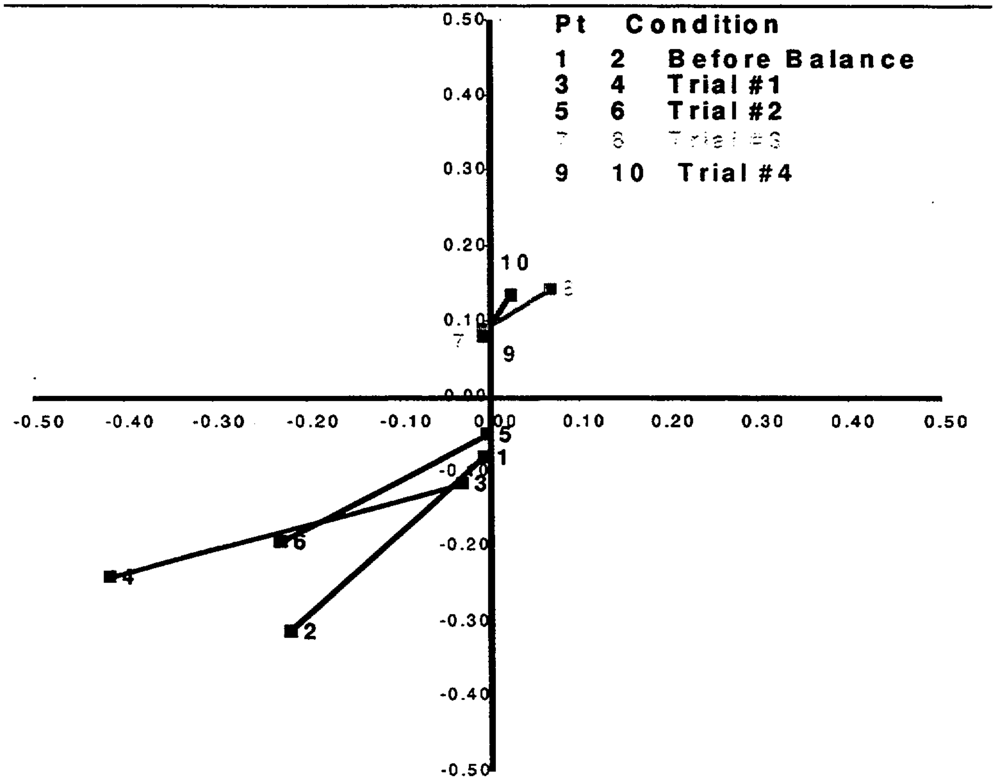


Figure 9. Affect of Trail Balance Weights on Synchronous Vibration

CVCS Charging Pump System: Positive Displacement Pump Replaced by a 3600-RPM Centrifugal Pump

Rick Koch
Flowserve Corporation

Background and Introduction

Combustion Engineering and Westinghouse Pressurized Water Reactors (PWRs) utilize positive displacement pumps (PDPs) in the Chemical and Volume Control System (CVCS). The CVCS system provides several functions including injection of boric acid into the reactor coolant system as well as seal injection for the Reactor Coolant Pumps (RCPs).

The existing positive displacement pumps have experienced a history of high maintenance activity, and unacceptable reliability and operational readiness—as a result of numerous malfunctions including high packing leakage, abnormally short plunger lives and broken pump cylinder blocks (e.g., working barrels), cracked piping, pipe support breakage, valve/valve spring breakage and shaft breakage. Suction stabilizers and discharge dampeners have been installed to mitigate the low frequency pressure pulsation energy. This did not eliminate the problem. In many of the plants, packing life has been only 2 months with severe leakage causing high radioactivity in the pumping room. In fact, these positive displacement pumps have been identified as one of the highest sources of radioactive water and noble gas leakage in many nuclear plants. Because these pumps are critical to plant operation, block and plunger failure have caused forced outages of entire plants.

Noble gas leakage and hydrogen cover gases stripped from solution have been recurring problems. The radioactive gas leakage into Radiological Controlled Areas (RCA) was a problem relating to personnel access to these areas for critical maintenance needs. The situation is a risk factor relative to the plant's ALARA (as low as reasonably achievable) goals.

A hydrogen gas cover is placed in the Volume Control Tank (VCT) to control oxygen and potential corrosion in the reactor coolant system (RCS). It was concluded that during the suction stroke of the PDP, reducing localized pressure below fluid vapor pressure, that the hydrogen is stripped from the fluid—during fluid acceleration and resulting cavitation. The high temperature and pressure of the gas, during the compression/pressure stroke is a contributor to fluid cylinder fatigue cracking. The existence of the hydrogen gas is hypothesized to be a contributor to the hydrogen embrittlement and material failure of the stainless steel blocks.

The CVCS system varies from plant to plant with the following pump arrangements utilizing positive displacement pumps:

Westinghouse Design–Category 1 - Three Variable Speed Positive Displacement Pumps

Westinghouse Design–Category 2 - One Positive Displacement Pump with Two Large Multistage Centrifugal Pumps

Combustion Engineering Design - Three Constant Speed Positive Displacement Pumps

Most utilities with the Westinghouse Design–Category 2 have had to address the problem with the positive displacement pumps by using the much higher flow (4850 rpm or 6000 rpm, motor-gear driven) centrifugal charge pumps (CCP) for all system operating conditions. Refer to the Figure 1 graphic of a typical Westinghouse PWR CCP (Flowserve pump model 25RLIJ-11). However, these pumps have also suffered major maintenance problems because of high vibration while running at much lower flows than that originally designed. This alternate back-up pump to the CVCS system is the High Pressure Safety (Emergency Core Cooling System—ECCS) Injection multistage (typical models 25RLIJ-11 or 3DVMX-9) centrifugal pump which is designed for higher operating flows to mitigate a small break loss of coolant accident (LOCA). Refer to the Figure 2 Performance Curve of a 900 horsepower CCP. Note that the Best Efficiency Point (BEP) is approximately 500 gallons per minute (gpm). The total flow of the operating CCP would be approximately 160 gpm, including miniflow recirculation, normal charge and injection flow to the RCP seals. Note that this results in the "low" (approximately 33% BEP) flow rate. Low flow operation can often be damaging to a centrifugal pump.

Some plants also have a restrictive operation mode, with the PD pump being limited to letdown flows of 75gpm to the RCS for chemistry control. A flow rate of 120 gpm is occasionally required. During such system demands, the Safety Related CCPs must be started.

Flowserve (Ingersoll-Dresser Pump Company) has solved these problems at several nuclear plants with a pump that uses a low flow high head, low Specific Speed pump with patented design impellers. The pump has been designated by the plants as Normal Charging Pumps (NCP). The pump is built to ASME Section III Class 2 Seismic Category I and is "safety related" for pressure boundary but not for a safety function related to safe reactor shutdown. This design (see Figure 3–Pump Sectional Assembly) has eliminated the positive displacement pumps and their inherent operational problems, as well as the low flow requirements of the larger ECCS pumps. The pump is designed such that the flow ranges demanded by the system span the NCP BEP flow—refer to Figure 4–Performance Curve. The first pump has been in successful operation since the Callaway Nuclear Plant start-up in 1994. Installations now include:

- Callaway – Units #1 and #2
- Wolf Creek
- Plant Vogtle, Southern Nuclear – Units #1 and #2
- Korea Electric Nuclear Power Corporation –Yonggwang Units #5 (qnt.2) and #6 (qnt.2)

Design and Analysis

An intense design development program in the late 1980s, utilized Computer Aided Design Analysis tools, along with actual laboratory testing. The following is a summary of critical design and reliability features that were analyzed:

- Computer aided finite element (FEA) stress analysis of the forged casing (barrel) and discharge head to optimize pressure boundary wall thickness and casing shape.

- Finite element stress analysis of the channel rings to analyze section thickness and deflection.
- FEA rotor dynamics to analyze pump rotor and shaft deflection, stress, endurance limits and critical speeds (natural frequencies).
- Laboratory tested with induced rotor unbalance, at variable speeds, to evaluate and quantify rotor critical speeds.
- Laboratory testing was conducted to evaluate effects of entrained gas. There was no effect up to 6% entrainment at design flow. Normal centrifugal pumps will lose performance when subjected to only 2% to 3% air/gas by volume rate.

  Air was introduced into the pump at volume rates up to 15%. The pump experienced less than 3% deterioration in developed pressure, at design flow. Normal centrifugal pumps will fail to perform under these conditions. At low flows, up to 20% entrained air resulted in just over 3% pressure degradation.
- Axial and radial thrust loads were measured on the test stand across the flow spectrum. Measured loads were below expectation. Resulting bearing life far exceeded predicted life reliability.
- Pressure pulsations (both suction and discharge) were measured across the operating flow range. Results were again better than expected.
- At one end user plant, the pump was subjected to operation with no suction flow for a period of time due to logic problems with the suction isolation valve. Follow-up pump testing and condition analysis vibration and hydraulic performance testing indicated no pump degradation.

Installation Experience

A typical installation process is discussed here; as it relates to one of the initial installations in the Callaway Nuclear Plant (see Figure 5 and Reference 1). The removal of the PDP and installation of the NCP was accomplished in two phases. The piping, wiring and foundation modifications were conducted initially during a refueling outage. The final removal of the PDP and installation of the NCP were performed during normal plant operation. The NCP was mounted on the existing PDP foundation. The NCP does not require the auxiliary equipment required by the PDP. The following support auxiliary systems were removed:

- Component Cooling Water (CCW)
- Demineralized water
- Compressed air
- Floor and equipment drains
- Suction pipe stabilizers
- Discharge pipe dampener

The NCP was started and has operated successfully since September 1994.

Pump Features

The successful technology employed at Callaway, Wolf Creek, Vogtle and KEPCO nuclear power stations replaces the positive displacement pumps with a multi-staged direct drive 3600 rpm barrel style pump. This pump offers many unique features:

- 3600 rpm pump delivers over 2600 psi at 87 to 160 gpm. Different impeller sizes and number of impeller stages are utilized to meet the specific plant hydraulic Conditions of Service (COS).
- Compact design will fit into existing space

constraints.

- Low speed provides flow stability over a wide range of flows without surging and pulsations.
- Ball-Ball bearing construction eliminates the need for a forced feed external lube system. There is less oil to discard, which will reduce radioactive waste for the plant.
- The 2X10CAM pump utilizes an internal flanged axial thrust-balancing device.
- The shaft material is 410SS material furnished in accordance with Flowserve's proprietary "super-straight" specification. This unique production and heat treating process results in essentially no shaft forging residual stress. Some plants may request Nitronics-50 and CA625 shaft material.
- Ball Bearings are "off the shelf" that are easily replaced.
- A proven design.
- The forged casing (or barrel) is designed to accept bolted flanged nozzle connections, to the plant piping.
- No external seal injection piping.
- The pump utilizes a patented modular impeller. See Figures 6 and 7. This unique "open vane" impeller is the "heart" of the pump design. It features precision-machined radial vanes, specifically shaped and oriented for optimum performance. The patented holes (shape, number and orientation) in the impeller shroud are configured to optimize hydraulic performance, head rise, and hydraulic stability and minimize axial thrust. The radial vane configuration results in a higher head coefficient than normal centrifugal pumps. What this higher coefficient means to the end user is a pump that is more forgiving when the environment may have entrained gas.
- The 2x10CAM meets all operating conditions that the present PD pumps perform at all Westinghouse and Combustion Engineering design plants.
- Assembly and disassembly is modularized and conducted without the use of a torch, thereby optimizing ALARA goals and pump serviceability.

Pump Qualifications

The CAM pump Nuclear RCP mechanical seals are manufactured to ASME Section III Class 1, 2 & 3 with or without an "N" stamp. Design and manufacturing process specifications meet the requirements of 10CFR50 Appendix B, Quality Assurance Criteria for Nuclear Power Plants, as well as 10CFR21, Reporting of Defects and Noncompliance.

Depending on plant requirements, an "N" stamp will be applied to the pumps jurisdictional pressure boundaries. Non-pressure boundary areas will be considered standard commercial construction and will be manufactured via the standard quality control program. Pedestals of the baseplate are manufactured as "NF" quality, while the remainder of the baseplate is considered standard commercial quality.

Reliability and Maintainability

The charging service with its many starts and stops, and low flow-high head operating conditions makes reliability and maintainability an important consideration. The 2x10CAM is designed with features specific to this application and its unique conditions.

Some of these important features include:

- Ball bearings are standard stock items. Bearings are easy to change with minimal downtime for the equipment.
- Mechanical seals are cartridge design making removal and reinstallation simple and quick. Exposure to radioactive fluid is kept to a minimum.
- No external seal piping is required. A pumping ring feature provides internal recirculation and fluid exchange in the seal cavity to maintain required temperature levels.
- Many pumps of this type have impellers that are shrunk fit to the shaft. The 2x10CAM will have slightly loose fit impellers and stage pieces, which will not require heat for removal. The channel rings also have a slightly loose fit. This optimizes time to assemble and disassemble, and eliminates the need for a heating torch. The heating torch also implies airborne contamination and not achieving desired ALARA goals.

  Some plants may prefer a slight interference fit between the stator stage piece channel rings, and between the impellers and shaft. This design and tolerance fit-up can be accommodated. The plant must recognize that a heating torch will be required for disassembly and assembly. A heating oven may be utilized for assembly purposes. The Instruction Book must be modified accordingly. The interference fit design does not optimize one of the original design concept goals—to achieve desired ALARA goals.
- Channel rings will utilize jack bolts to assure ease of assembly and disassembly in the field.
- The impeller open radial vane design, along with the shroud recirculation holes desensitize the CAM pump to gas ingestion. The PD pump was plagued with gas stripping and ingestion. The hydraulic performance and rotor dynamics stability, for most centrifugal pumps, are negatively affected by 3% to 5% gas volume. The CAM pump performed its task at levels 10% to 15% gas, by volume.
- Inconel external bolting will be used to prevent corrosion from the boric acid solution.
- Bearing housings are a machined fit to the barrel, no doweling is required.

Scope of Supply

2X10CAM - 12 Stage Horizontal Charge Pump Including:

- Cartridge mechanical seals.
- Ball bearing construction with ring oil lube.
- 316 stainless steel forged barrel construction.
- 12% chrome impellers, diffusers, and channel rings.
- Carbon steel drip rim baseplate.
- Inconel studs used for external bolting.
- Stainless steel flexible disc coupling.
- Seismic analysis on pump included.
- Provisions for bearing RTD's and vibration probe.
- ASME Section III Class 2 (see "Pump Qualifications" section for description).

Motor size, enclosure and horsepower rating depending on pump requirements and exact hydraulic conditions.

Summary and Conclusion

This success-proven pump design and "normal charge" system have resulted in improved pump and nuclear safety system reliability and reduced maintenance costs. The pump design and serviceability facilitate optimized ALARA features. With a financial analysis and resulting Payback Analysis conducted, justification to replace the PD pump with the Flowserve CAM model NC pump can be obtained.

References

1. Kochert, S. and Rauch, G. – Callaway Nuclear Plant: "Replacement of Positive Displacement Charging Pump," Nuclear Plant Journal, May–June 1997.

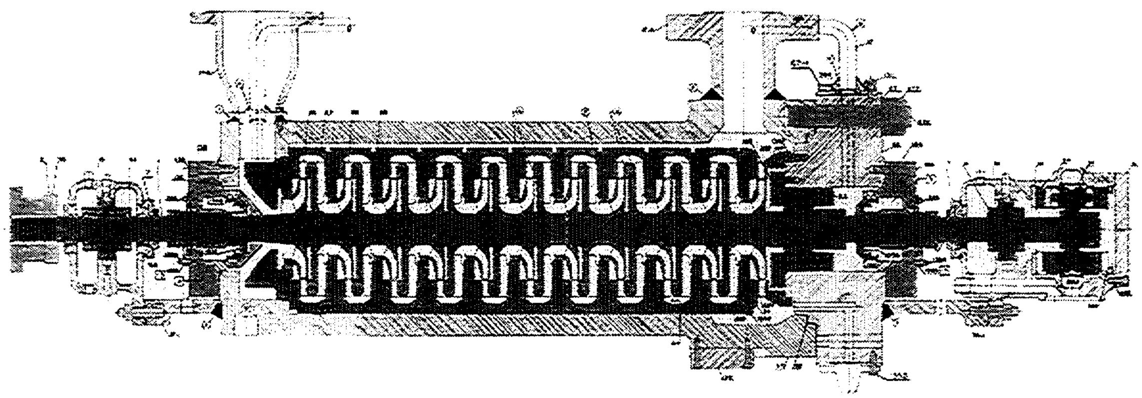


Figure 1. Cross Sectional drawing of Centrifugal Charge Pump (CCP), model 25RLIJ-11

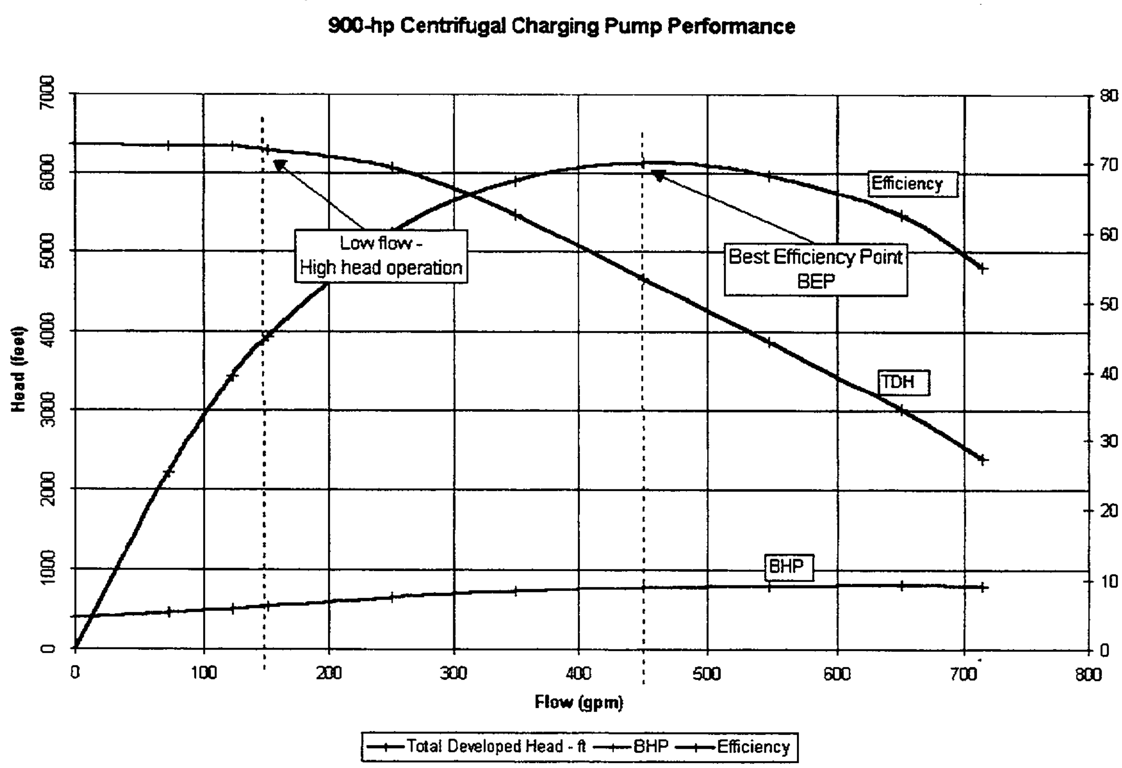


Figure 2. Typical Performance Curve for 900 horsepower Centrifugal Charge Pump, model 25RLIJ-11.

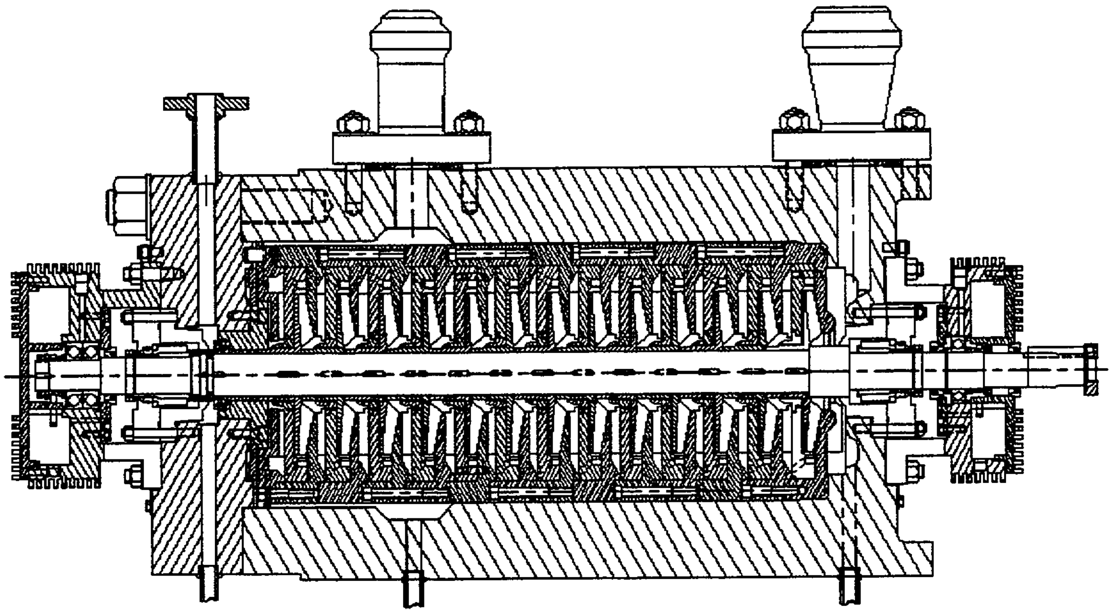


Figure 3. Cross Sectional drawing of Normal Charge Pump (NCP), model 2X10CAM-12.

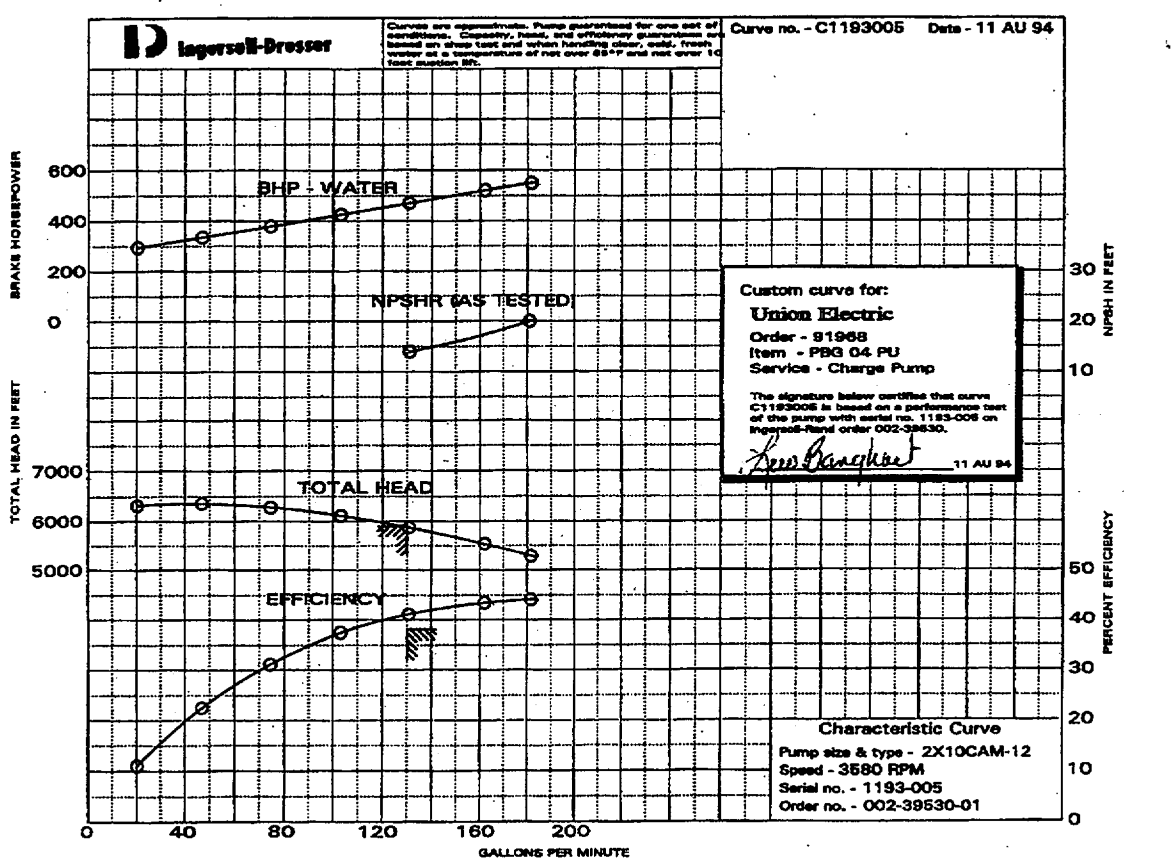


Figure 4. Typical Performance Curve for Normal Charge Pump, model 2X10CAM-12.

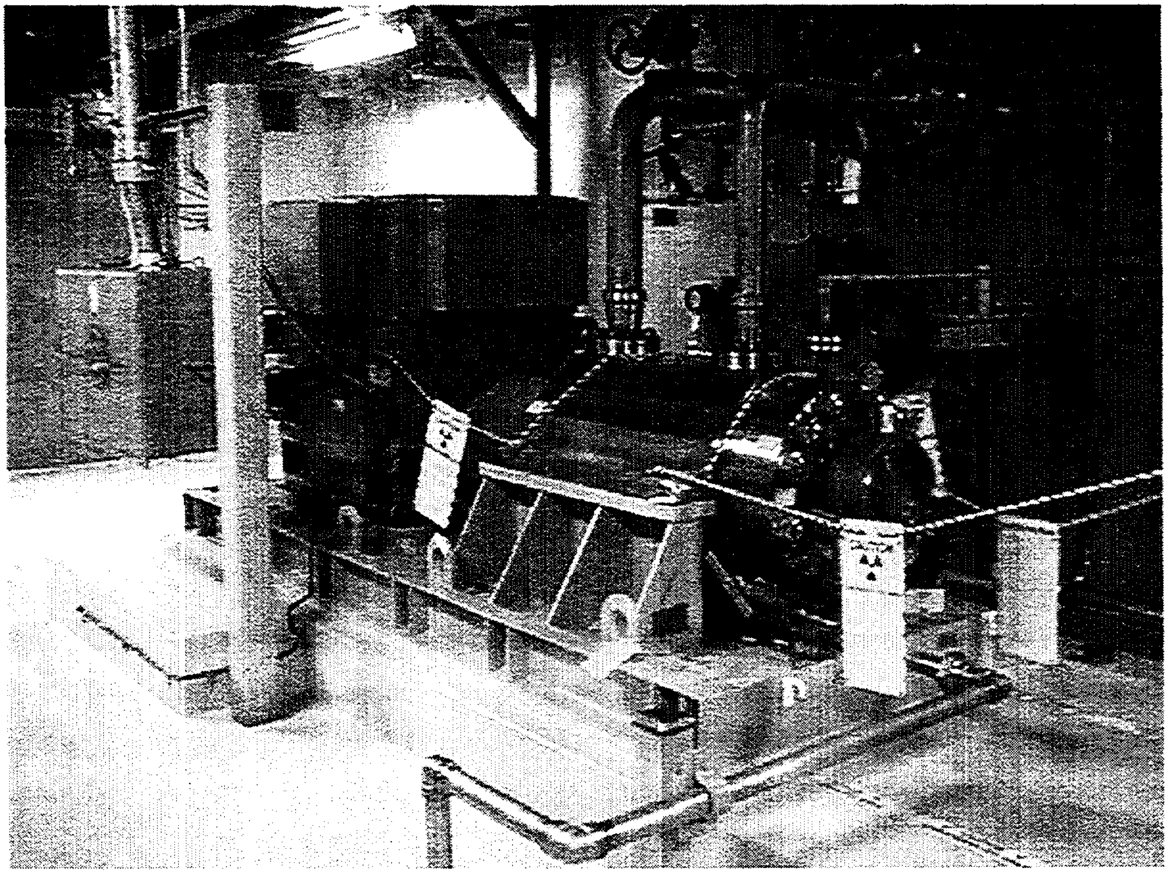


Figure 5. Photograph of Normal Charge Pump (CAM model), Plant Installation (Callaway Nuclear)

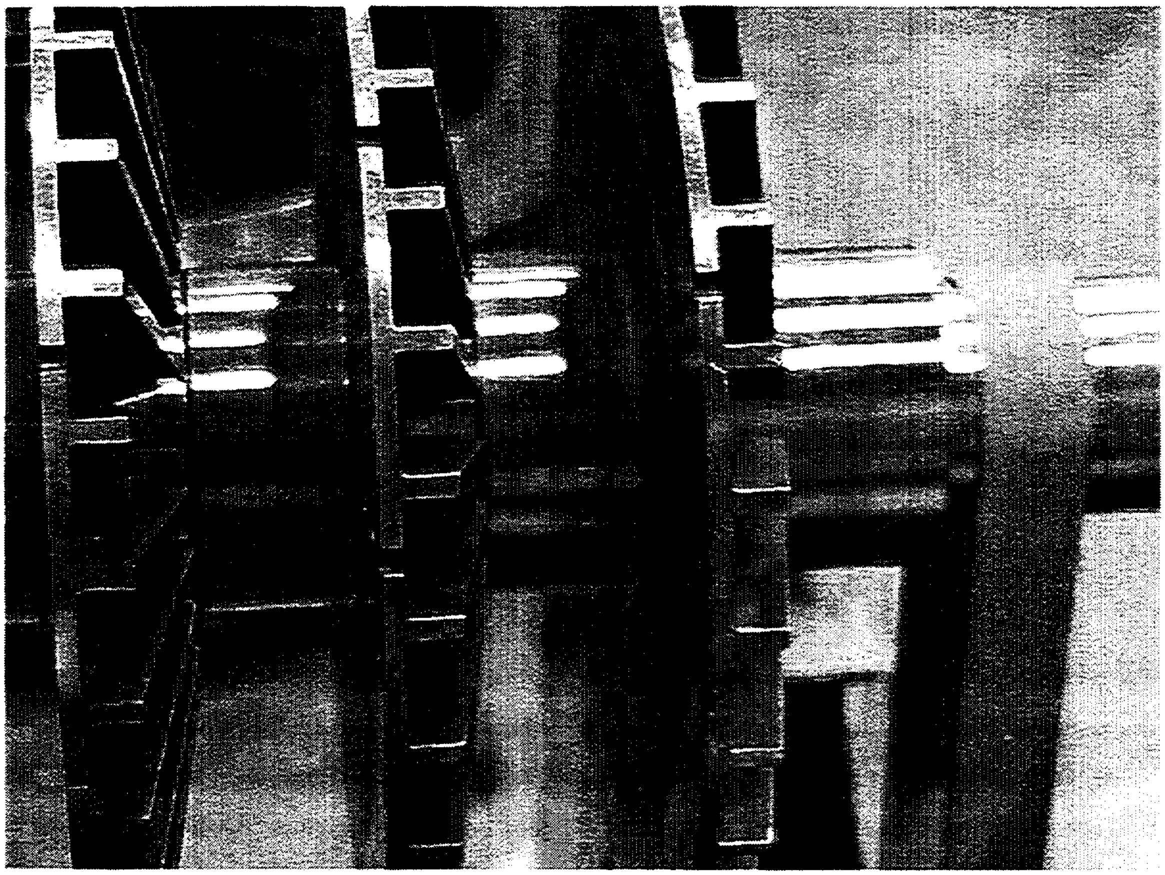


Figure 6. Patented design NCP Impeller Construction, radial vanes.

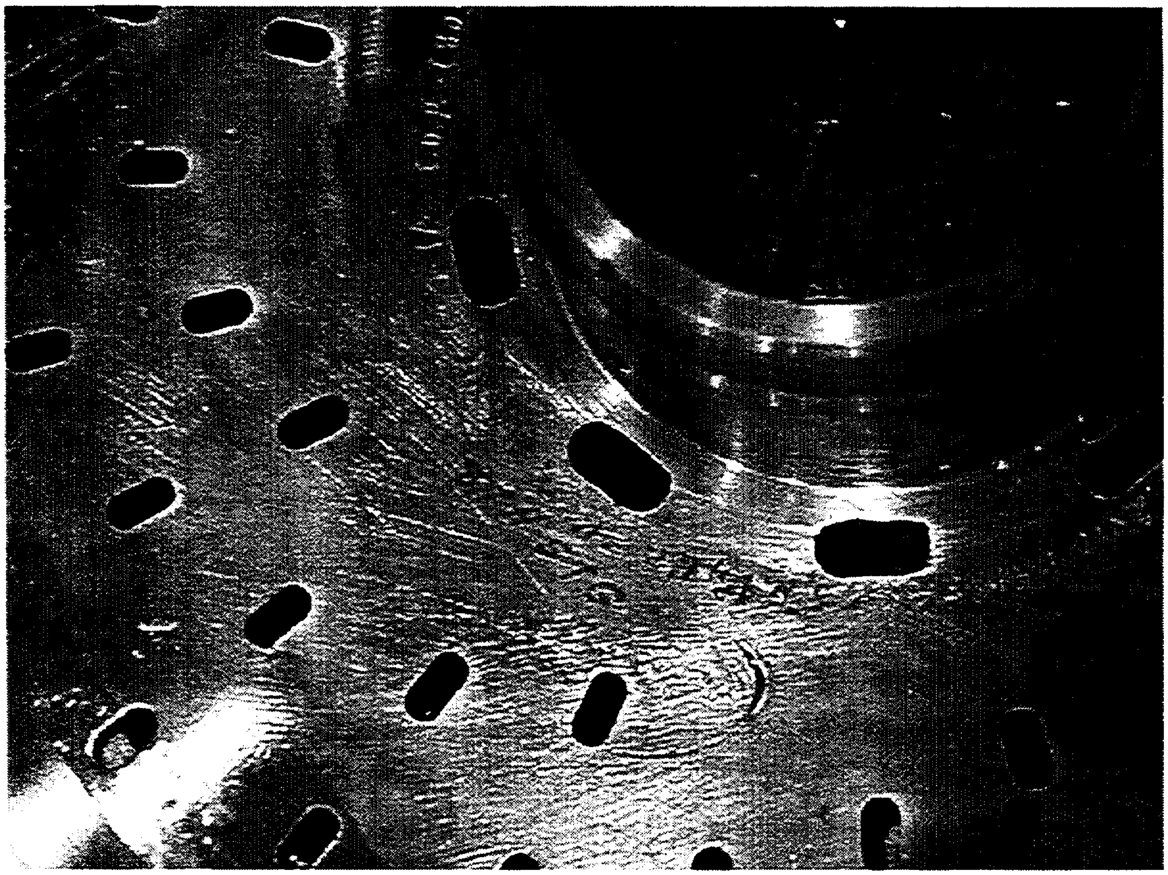


Figure 7. Patented design NCP Impeller Construction, unique balance hole construction.

In-Service Pump Testing by the Thermodynamic Method

Maurice A. Yates, and Mike Mancini,
Hydro-Yates, Inc.

Abstract

The paper introduces the concept of Thermodynamic Testing, a technique which is easily applied to in-service testing of the pumping plant and gives highly accurate and repeatable results on a wide range of pumps. The thermodynamic method of pump testing, which has been known since 1914, has over the last twenty years been given a boost by modern electronics and the computer.

The technique, which relies on measuring the very small increases in the pumped fluid temperature as it passes through the pump, has been perfected in the United Kingdom (UK) and is now available to monitor the performance of a wide range of pumps.

The technique has many advantages over the conventional test methods, which rely on the accurate measurement of flow. These advantages include ease of preparation, limited pipe-work constraints, highly accurate and repeatable results with little or no interference in the day-to-day operation of the pumps.

Legend

Cp Specific heat capacity
dT Temperature rise across the pump
g Acceleration due to gravity
H Head
P Pump shaft power
Pgr Input power
Q Flow rate
Ep Pump efficiency
Em Motor efficiency

Introduction

In the early 70s as Head of Technical Design for a UK Water Company, one of my responsibilities was to monitor the performance of the company's pumping plant. This actually proved to be quite a difficult task, as few pumping systems had adequate flow metering.

In 1983, an innovation grant from the UK Government led to the development of an automatic thermodynamic pump efficiency meter, which became known as the Yatesmeter. Since that time, the thermodynamic method has been used in testing pumps of all shapes and sizes, the largest being 70 MW and the smallest 0.5 kW.

In addition to pump testing, the technique is used to carry out in-service calibration of flow meters. This calibration is actually a complete one in that it calibrates from the fluid flow right up to the dial reading.

Discussion

The major difficulty in measuring pump performance is the measurement of flow through an individual pump. The use of state of the art transducers and electronics has lead to the development of a thermodynamic pump performance meter which gives quick and reliable data on a pump's performance.

In the in-service testing of pumps, we need to determine the following pump characteristics: Head vs. Flow, Power vs. Flow, and Efficiency vs. Flow. To date, these characteristics have been produced by measuring head, power and flow; and then calculating efficiency. The thermodynamic technique relies on the principle of conservation of energy, in that energy can neither be created nor destroyed, to develop the alternative approach of measuring efficiency, head and power; and then calculating flow.

If this works, we can dispense with the flow meter and its baggage, such as the number of straight pipe length required, pipe material, pipe thickness, lining material and thickness, and the shear difficulty of installing accurate flow meters into existing pipe-work. To consider the measurement of efficiency, we need to first consider what efficiency really is. In engineering terms, efficiency is the ratio of the work done to the effort required, usually expressed as a percentage.

Note that, to a plant operator, efficient operation may mean something entirely different such as: which pump starts easiest, which one requires the least effort and, above all, which pump keeps him in bed at night.

Taking the definition of efficiency we obtain the following:

Efficiency = (Work Done / Effort) or (Work Out / Work In)

Work In may be defined as (Work Out + Losses), therefore the pump efficiency is

$$Ep = Wo / (Wo + Losses)$$
$$Ep = 1/(1 + (Losses / Wo))$$

Therefore, if the term (Losses / Wo) can be determined then the pump efficiency may be calculated.

Considering energy that is lost, some of it simply does go to drain through gland water or bearing cooling water, but most of the energy losses are dissipated as heat into the pumped fluid. The energy imparted to the pump fluid may be expressed as:

$$Cp.Q.dT$$

And Work Out from a pump, excluding specific gravity is (g.Q.H), therefore

$$(Losses / Wo) = (Cp.Q.dT / g.Q.H).$$

The efficiency can thus be expressed,

$$Ep = 1/(1 + (Cp.dT / g.H)).$$

It is at this point that one can immediately see the advantages of this approach because we have now reached an expression for pump efficiency that is independent of flow. For clarity, the equation has been simplified. For instance, it excludes specific gravity and the compressibility of the fluid. However, while oversimplifying the situation, the above does show the principles of the thermodynamic test.

The great advantage of the method is its potential accuracy. The accuracy comes from the measuring of losses. Measurement inaccuracies associated with the small percentage of pumping losses makes the thermodynamic technique accuracy several times greater than the accuracy of the method of measurement. For example, assume a pump under test is 80% efficient and therefore has 20% losses. If the measurement uncertainty of these losses is +/-5% then the accuracy of the measurement of pump efficiency is:

Work Out + Losses = 100

No error:	80 + 20	=	100
5% error:	80 + 19	=	99

The measurement error is therefore +/-1%, which is 5 times greater than the actual accuracy of the method of measurement.

Having determined the efficiency of a pump we now need to know its flow rate. This can be determined from

Efficiency = (Work Out / Work In)

where,

$$Ep = g.Q.H / P$$

and,

$$Q = P.Ep / g.H$$

Since,

$$Q = (P / g.H) . (1/(1 + (Cp.dT / g.H)))$$

the equation for flow can be written as

$$Q = P / (g.H + Cp.dT).$$

In the case of an electric motor drive

$$P = Pgr.Em$$

and, the flow rate becomes,

$$Q = Pgr.Em / (g.H + Cp.dT).$$

Now we have obtained our measurements

Efficiency	-	Thermodynamically
Head	-	Pressure Transducers
Power	-	Electrical Input to System
Flow	-	by Calculation

This now allows us to draw the curves in conventional format, which are illustrated in Figure 1.

Advantages

The high accuracy of the technique is down to the basic fact that the fundamental measurement of temperature is used to determine the losses occurring in the pump and its associated system. As shown earlier, the measurement of the losses in a pump having an efficiency of 80% gave rise to a "five times" improvement when considering the actual efficiency measurement.

The following table takes this process further using 2% measurement accuracy as an example.

Pump Efficiency	% Error Loss	% Error Efficiency	Gain
90	2	.2	10
80	2	.4	5
70	2	.6	3.3
60	2	.8	2.5
50	2	1.0	2

This may be expressed as follows:

Overall Accuracy =
Measurement Accuracy X(1 – Ep)

Therefore, for highly efficiency pumps, the system has a great advantage over the conventional method. If we now consider the conventional method where Ep = g.Q.H / Pgr.Epm

In this situation each of the measured parameters, i.e., Flow, Head, Power and Drive Efficiency, have an equal effect on the overall accuracy and the method of solving the overall accuracy is:

$$dEp = (dQ^2+dH^2+dPgr^2+dEp^2)0.5$$

Typically in-service the following accuracies will apply:

Q	-	5%
H	-	0.5%
Pgr	-	0.5%
Epm	-	0.5%

In this case the overall accuracy would be

$$dEp = (5^2+0.5^2+0.5^2+0.5^2)0.5$$
$$dEp = 5.07\%$$

In the conventional efficiency measurement, the overall accuracy is essentially dependent on the actual reading whereas, in the thermodynamic technique, there is a reduced dependence upon the actual value of the temperature measurement which is determined by the actual efficiency of the pump.

Considering the simplified efficiency formula

$$Ep = (1 / 1+Cp.dT / g.H)$$
$$Ep+(Ep.Cp.dT) / g.H = 1$$
$$Ep.Cp.dT = g.H\ (1-Ep)$$
$$dT = g.H / Cp\ (1-Ep / Ep)$$

Assuming

$$g = 9.81 \qquad Cp = 4186$$

The equation for dT becomes:

$$dT = 9.81 / 4186\ H\ (1-Ep / Ep)K$$

or

$$dT = 2.34H\ (1-Ep / Ep)mK$$

Using this formula for a range of pumping heads, we obtain the following temperature rises in mK for various efficiencies:

Temperature Rise mK

Head m/Ep%	90	80	70	60	50
10	3	6	10	16	24
20	5	12	20	31	47
50	13	29	50	78	117
100	26	59	100	156	234
500	130	293	500	780	1170
1000	260	585	1000	1560	2340

We can then relate these figures to give an accuracy chart for a 1mK uncertainty in differential temperature measurement.

Efficiency Accuracy %

Head m/Ep%	90	80	70	60	50	Av
10	3.8	3.41	3.03	2.56	2.13	3
20	1.9	1.71	1.52	1.28	1.06	1.5
50	.76	.68	.61	.51	.43	.6
100	.38	.34	.30	.26	.21	.3
500	.076	.068	.061	.05	.04	.06
1000	.038	.034	.030	.026	.02	.03

From this table, we can see that the actual differential temperature becomes less important the higher the head measured.

If now we take the average figures for dT accuracy and add to them the effect of the head measurement accuracy, we obtain the following:

$$Ep = (dT^2+dH^2)0.5$$

% Accuracy			
Head			
m	dT	dH	Ep
10	3	0.5	3.04
20	1.5	0.5	1.58
50	.6	0.5	.78
100	.3	0.5	.58
500	.06	0.5	.5
1000	.03	0.5	.5

This relationship is shown in Figure 2.

One of the great attractions of the technique is its ease of use, particularly when considering existing installations. The technique requires the measurement of the fluid temperature and pressure before and after the pump coupled with the measurement of electrical power into the motor. The temperature measurement is taken at an insertion depth of 1/7th of the diameter of the pipe with a minimum insertion of 2". The measurements then can be taken online without disrupting the remaining pumping operation.

To install the equipment, taps are required in both the suction and discharge branches of the pump. These taps increase to 3/4" for pipework greater than 12". The measurement of electrical power is obtained by connecting a power meter in the motor control panel.

The thermodynamic technique's greatest advantage is that an installed flowmeter is not required. The meter's unique ability to measure flow without the use of a flowmeter avoids the necessity to install flowmeters to ascertain pump performance or indeed process flow.

Validation

Over the last twenty years, the thermodynamic technique has undergone rigorous testing by such UK organizations as Water Research Centre, National Engineering Laboratories, Central Electricity Generating Board (UK), Exeter University (UK) and Damstadt University in Germany. In addition, it has been used by many of the world's major pump manufacturers including Flowserve, Weir, KSB, Ebara and SPP. During this period in-service tests have been carried out on over 5,000 pumps and the results have identified five major areas where pumping costs may be reduced.

These are:

Pump Condition
Pump Operating Point
Operating Regime
Driver Selection
Installation Effects

Temperature Probe Location

Horizontal Pumps (Figure 3)

Suction Tap
Fitted in a straight length of pipe at least 2 pipe diameters away from the pump flange.

Discharge Tap
Fitted in a straight length of pipe at least 2 pipe diameters away from the pump flange before the non-return valve and the discharge valve.

System Tap
Fitted in a straight length of pipe at a common point on the discharge manifold.

Vertical Can Pumps (Figure 4)

Suction Tap
Fitted in the inlet branch to the pump can; here the distance from the can is not important as the tap will be out of the way of the influence of the pump impeller.

Discharge Tap
Fitted in the outlet branch from the pump can before the non-return valve and the discharge valve. As with the suction, its exact location is not critical.

System Tap
Fitted in a straight length of pipe at a common point in the discharge manifold.

Vertical Wet-Pit Pumps (Figure 5)

Suction Location
Here care must be taken in locating the submersible temperature transducer. Firstly, the probe must not be located in a dead flow area in the sump. It must be located approximately 3 feet from the pump bellmouth in the flow stream to the pump. In addition, the probe must be weighted such that it is stable in the flow.

Discharge Tap
Fitted in the discharge branch of the pump before the non-return valve and the discharge valve.

System Tap
Fitted in a straight length of pipe at a common point in the discharge manifold.

Contaminated Pumps (Figure 6)

For contaminated pumps or pumps pumping high temperature or high pressure fluids, the temperature probes cannot be inserted directly into the flow and thermal wells must be used.

The location of the thermal wells is the same as for the normal tap arrangement. The length of the thermal well should be such that the insertion depth is 1/7th of the pipe diameter with a minimum insertion of 2 inches.

Experience

The technique has been used in a variety of applications including nuclear and conventional power stations, hydroelectric and pump storage power stations, and water and waste water, mining, steel, petrochem and general water service pumps. In addition, it has been applied to air compressors, blowers and fans.

The technique is largely used to produce the pump performance curve (Figure 7). However, there is an increasing need to consider how the pump is operating in its particular application and, for this work, the system characteristic is required (Figure 8).

Examples

Sea Water Injection Pump

A 4 MW Sea Water injection pump was tested at the manufacturers' works. The test was witnessed by the purchaser and without knowledge of the conventional results which were carried out in parallel.

The test arrangement is shown in Figure 9. The results for this test are shown in Table 1, from which the curves are constructed and compared in Figure 10.

Waterworks Fixed Monitoring

Here, Yatesmeters are permanently connected to six large water pumps and the data is centrally analyzed to determine the most efficient running regime of the pumping plant. Since its installation some two years ago, the water company has reduced its pumping bill by over $100,000 (Figure 8).

Conclusion

The thermodynamic method has proven to be a very robust measuring technique for measuring in-service pump efficiency, system performance and flowmeter performance. It is easily applied and the tests are carried out with minimal interruption to the pump plant.

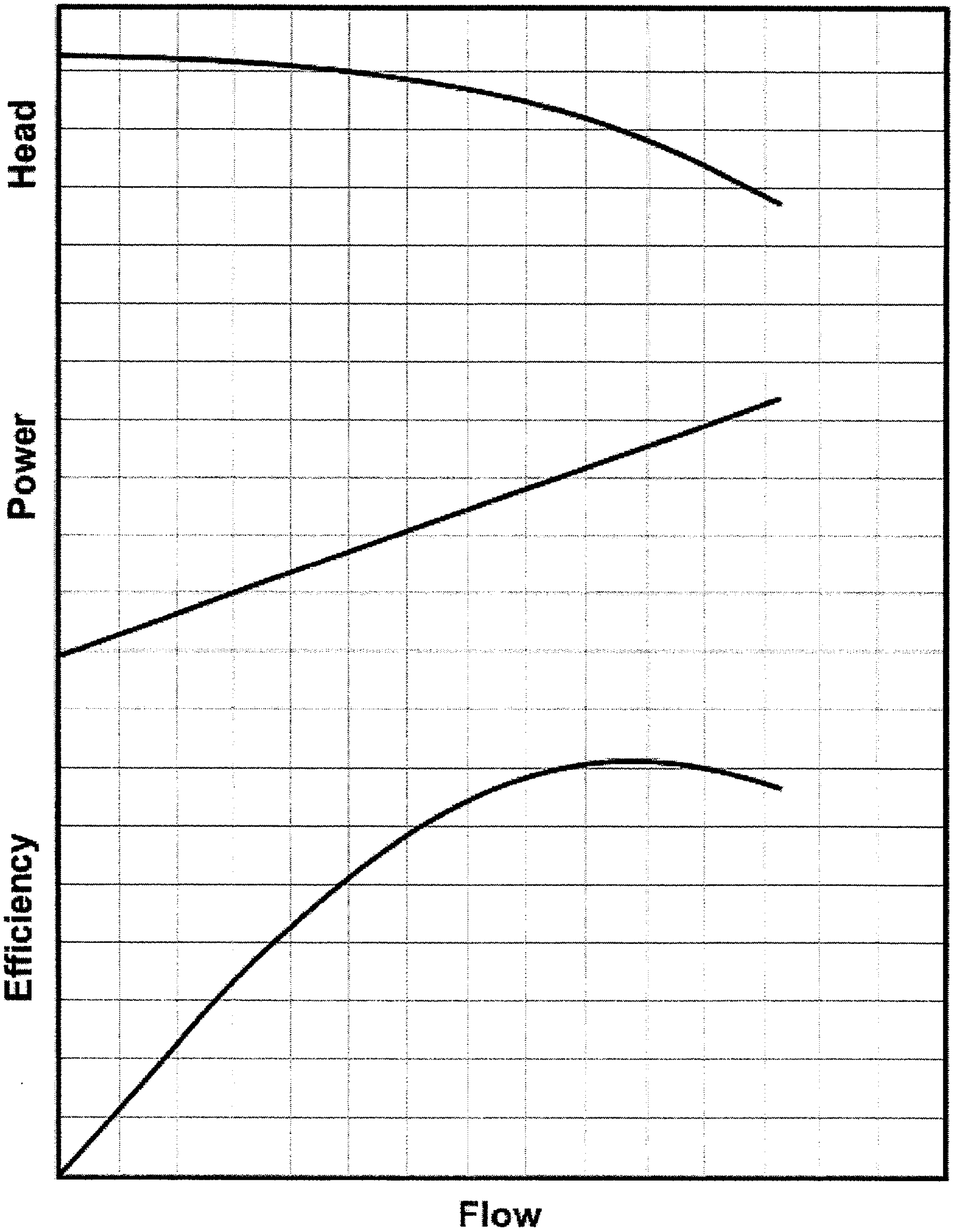


Figure 1 Pump Performance Curves

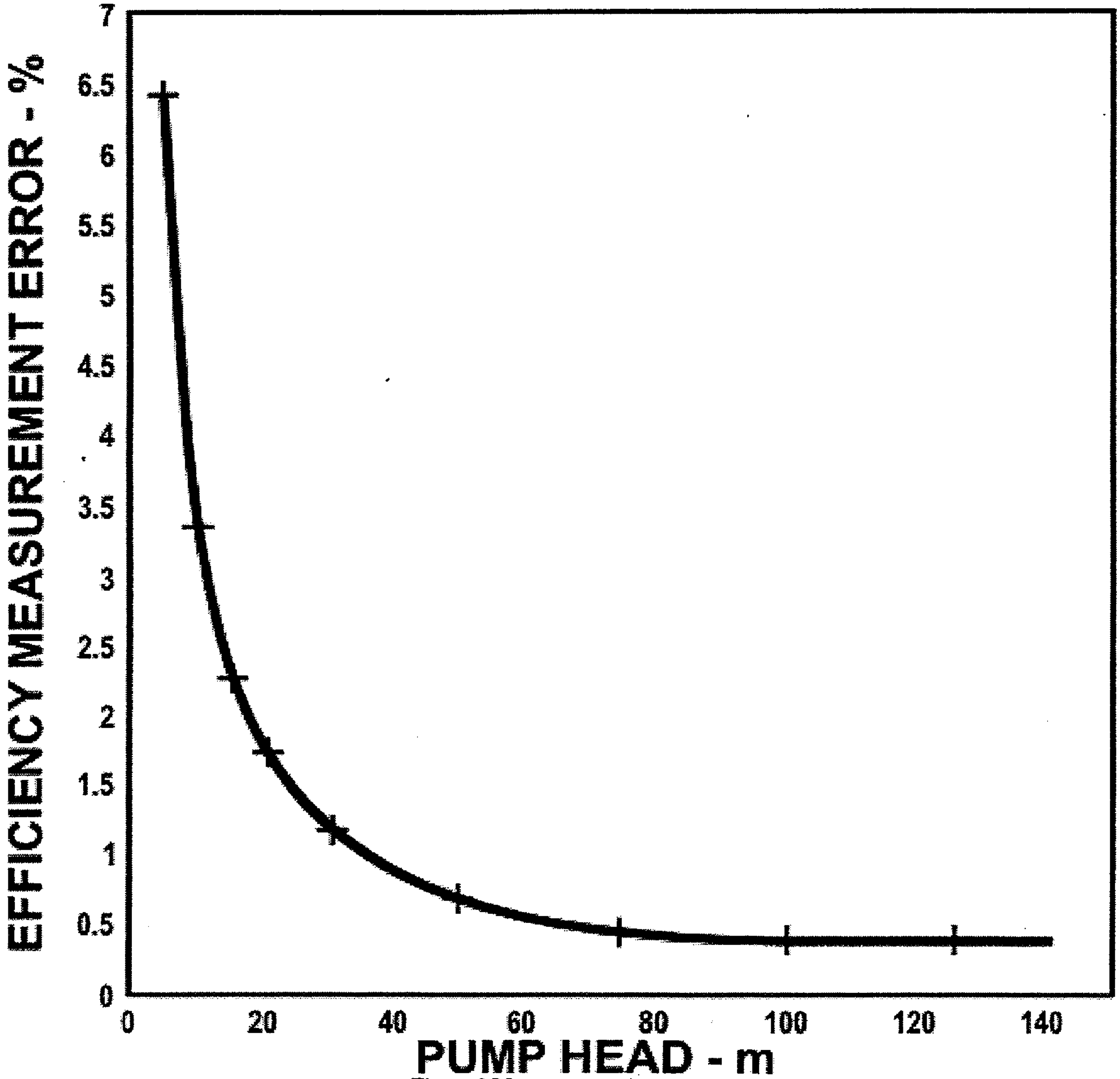


Figure 2 Measurement Accuracy

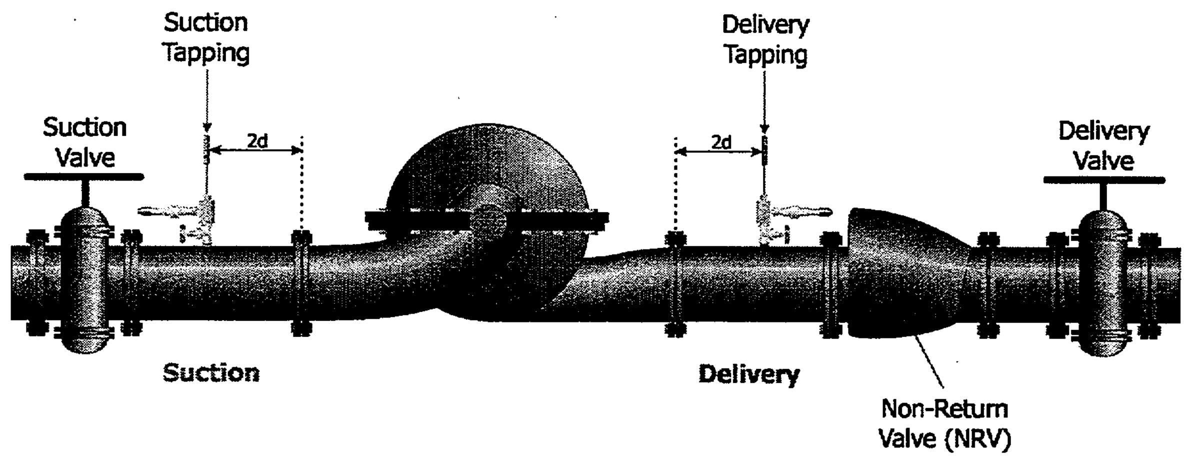


Figure 3 Horizontal Split Case Pump Tapping Arrangement

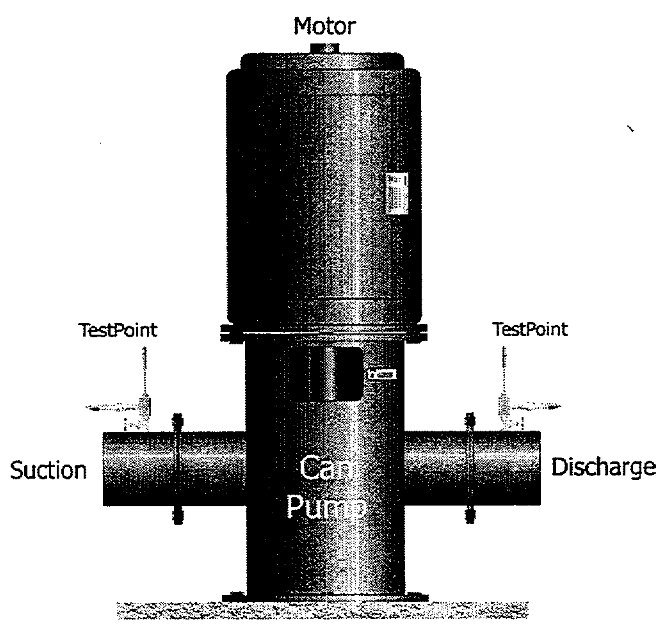


Figure 4 Vertical Can Pump Tapping Arrangement

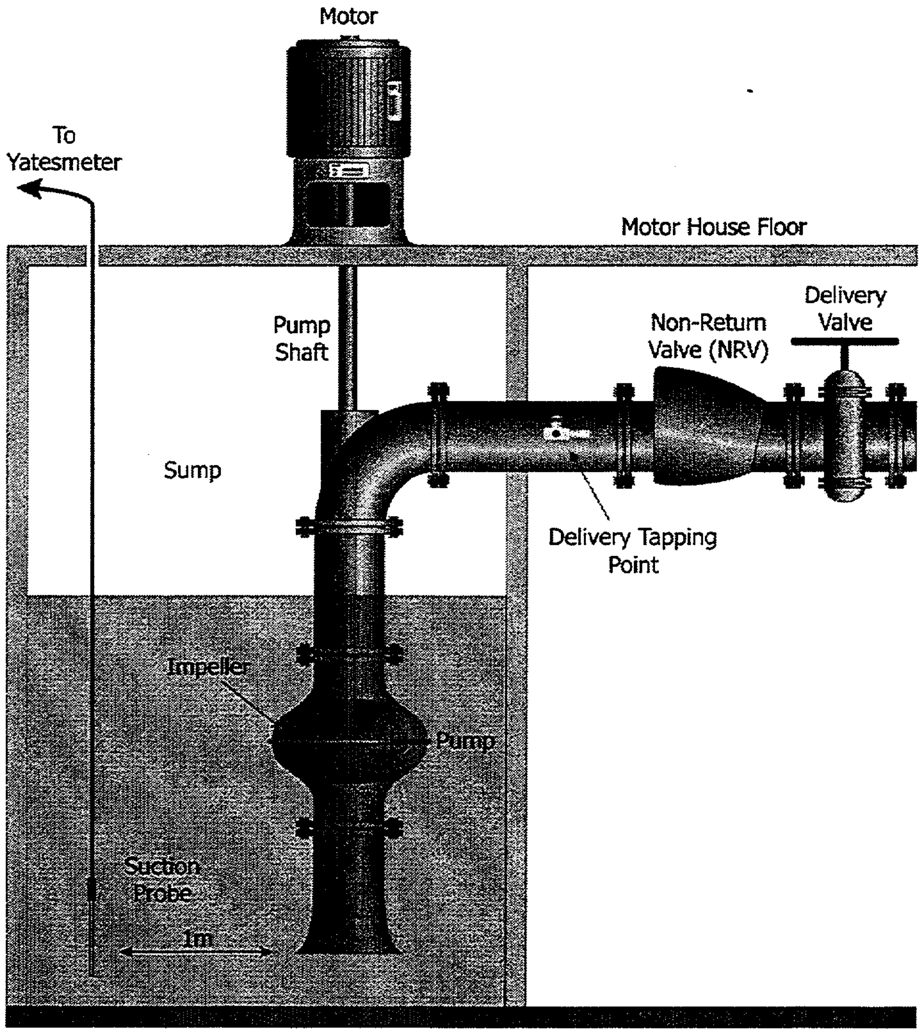


Figure 5 Vertical Wet Well Pump Tapping Arrangement

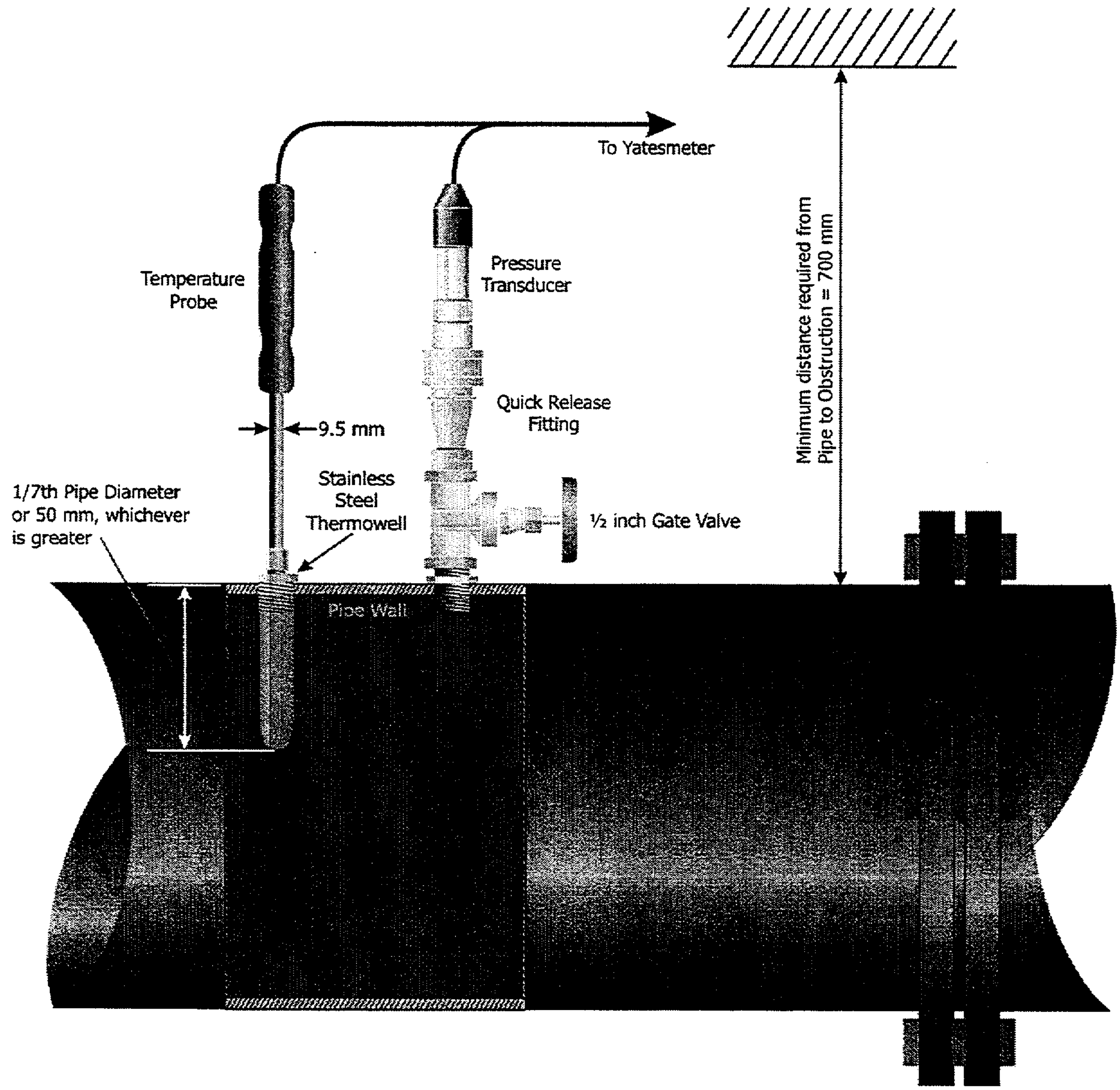


Figure 6 Contaminated Pumps Tapping Method

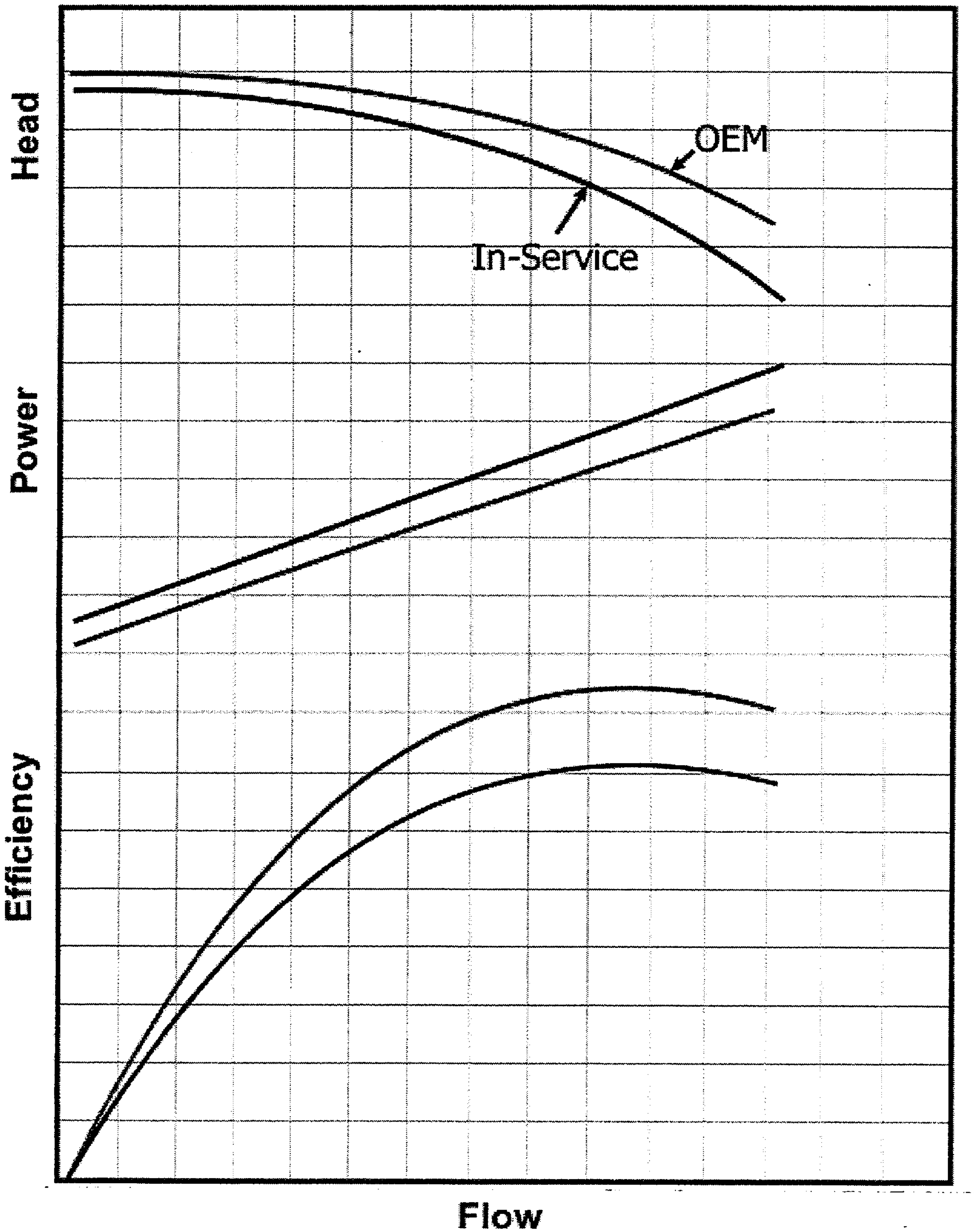


Figure 7 Typical In-Service Result

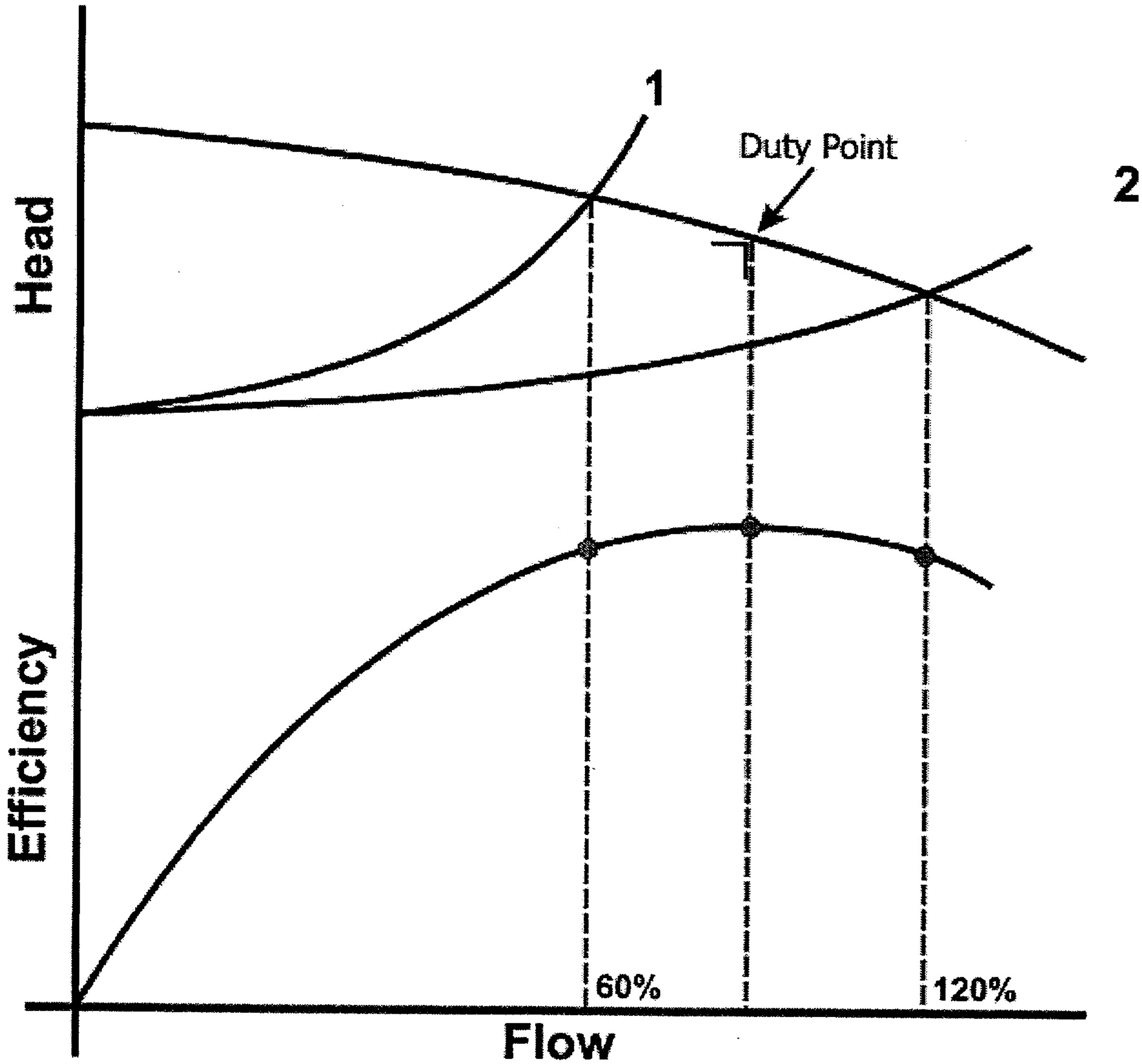


Figure 8 System Influence on Pump Performance

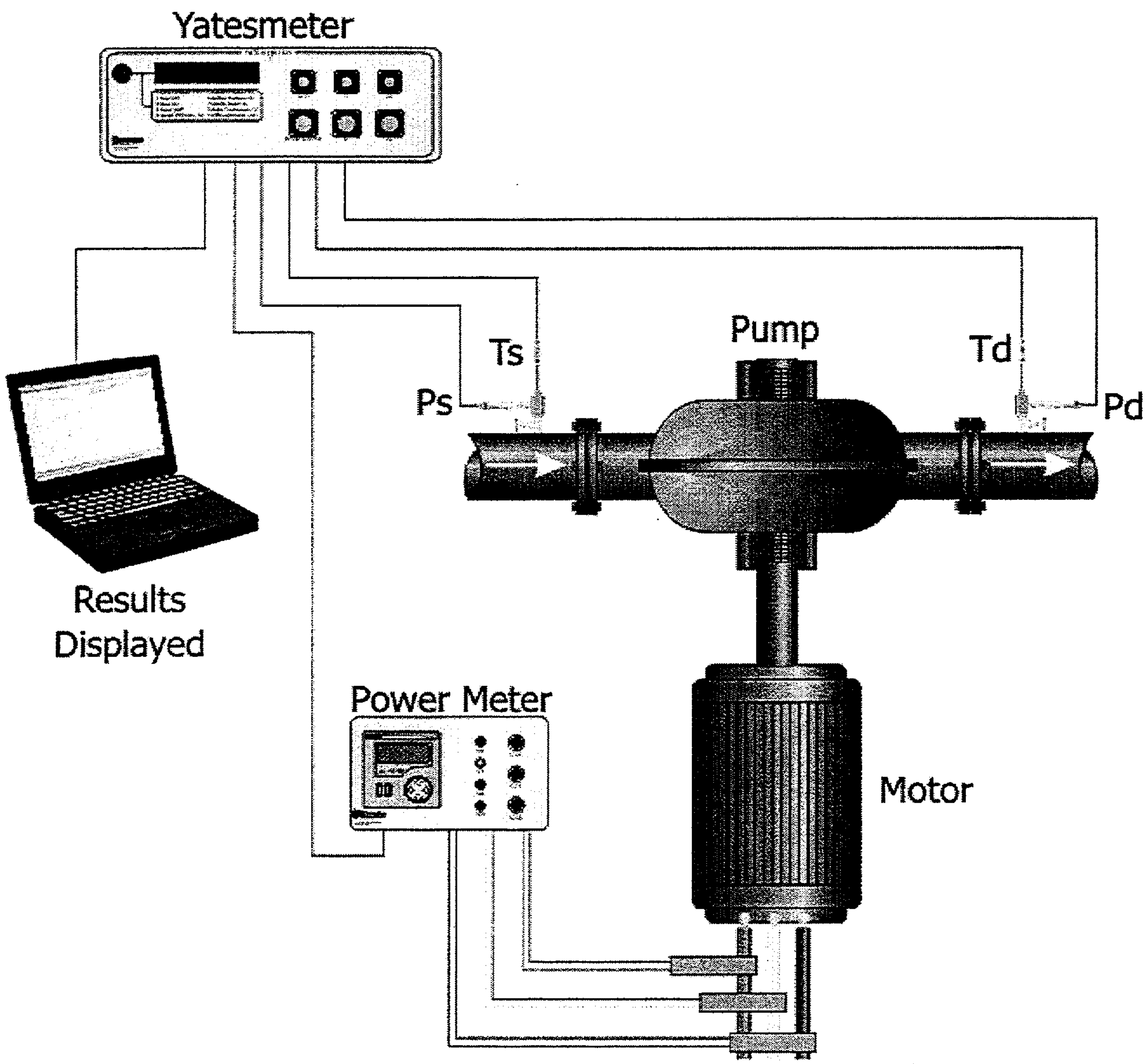


Figure 9 Portable Test Arrangement

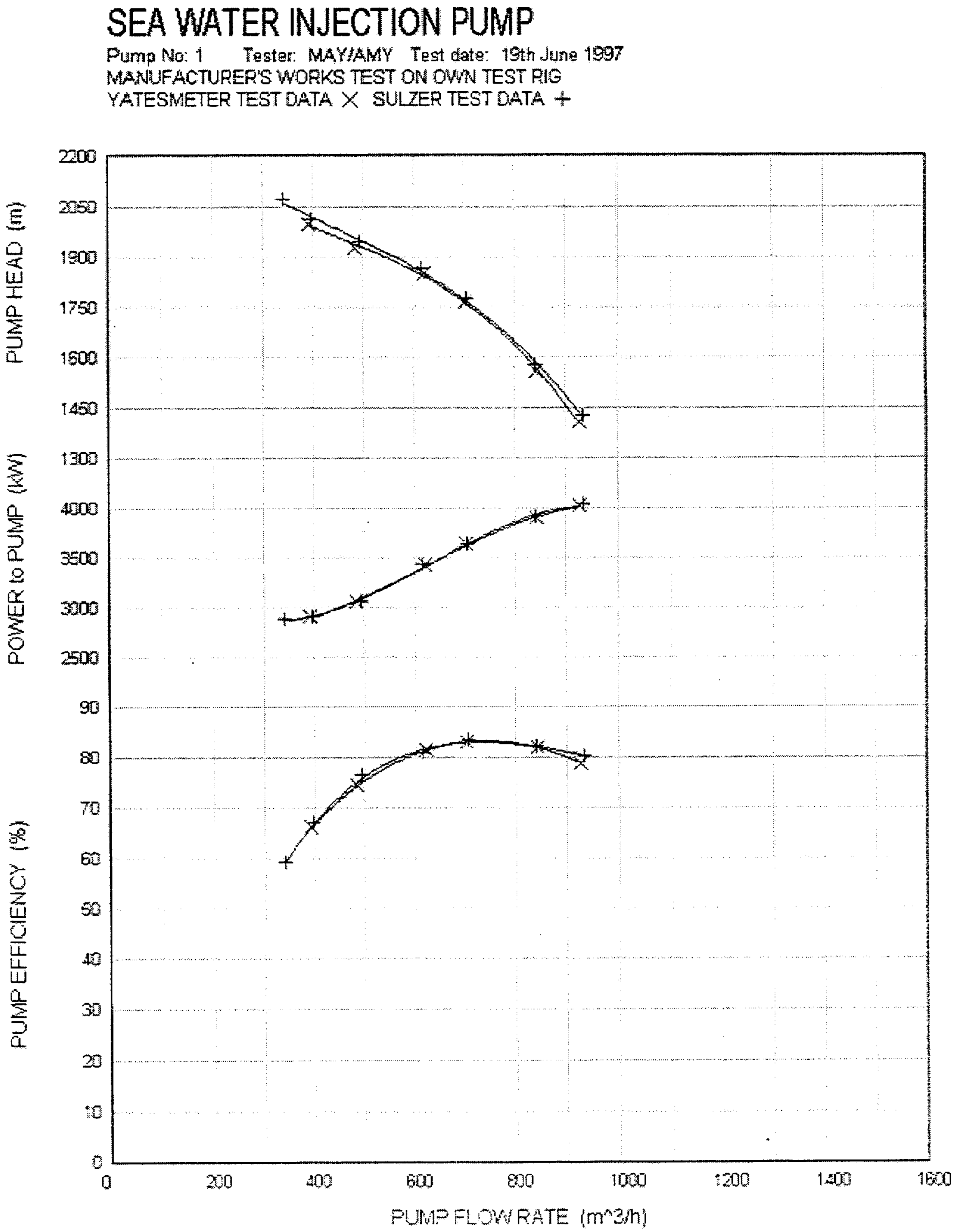


Figure 10. Sea Water Injection Pump

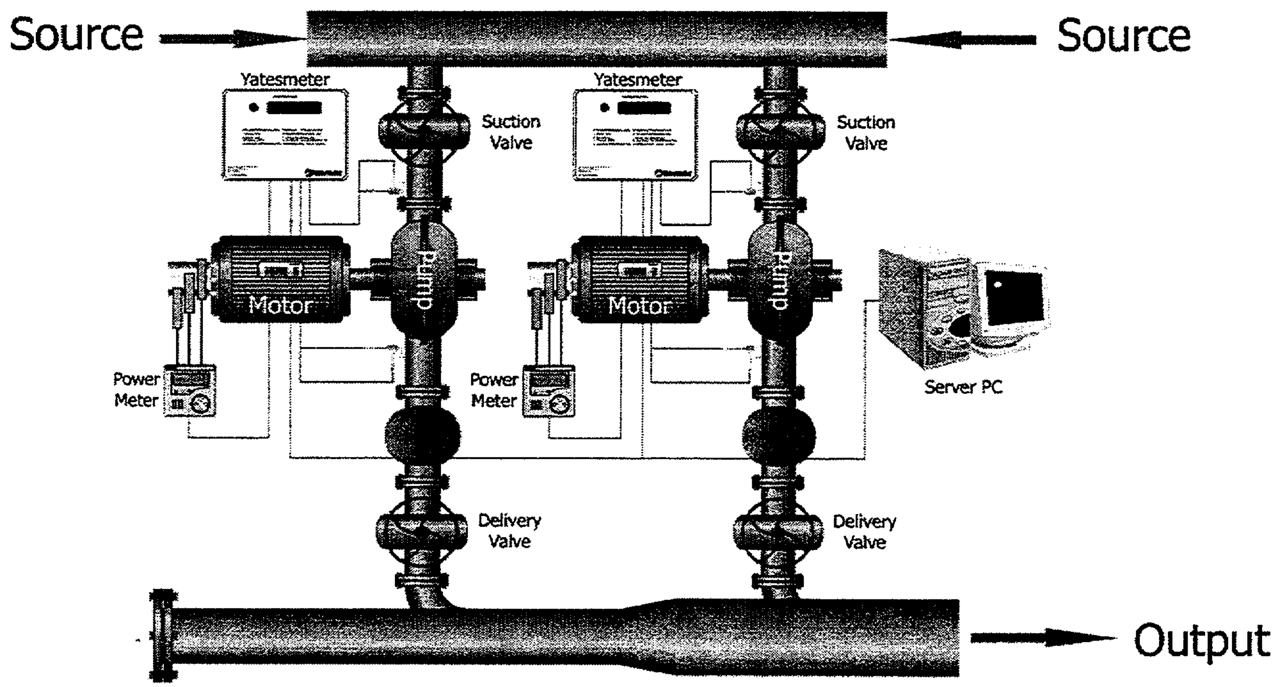


Figure 11 Typical Monitoring Arrangement

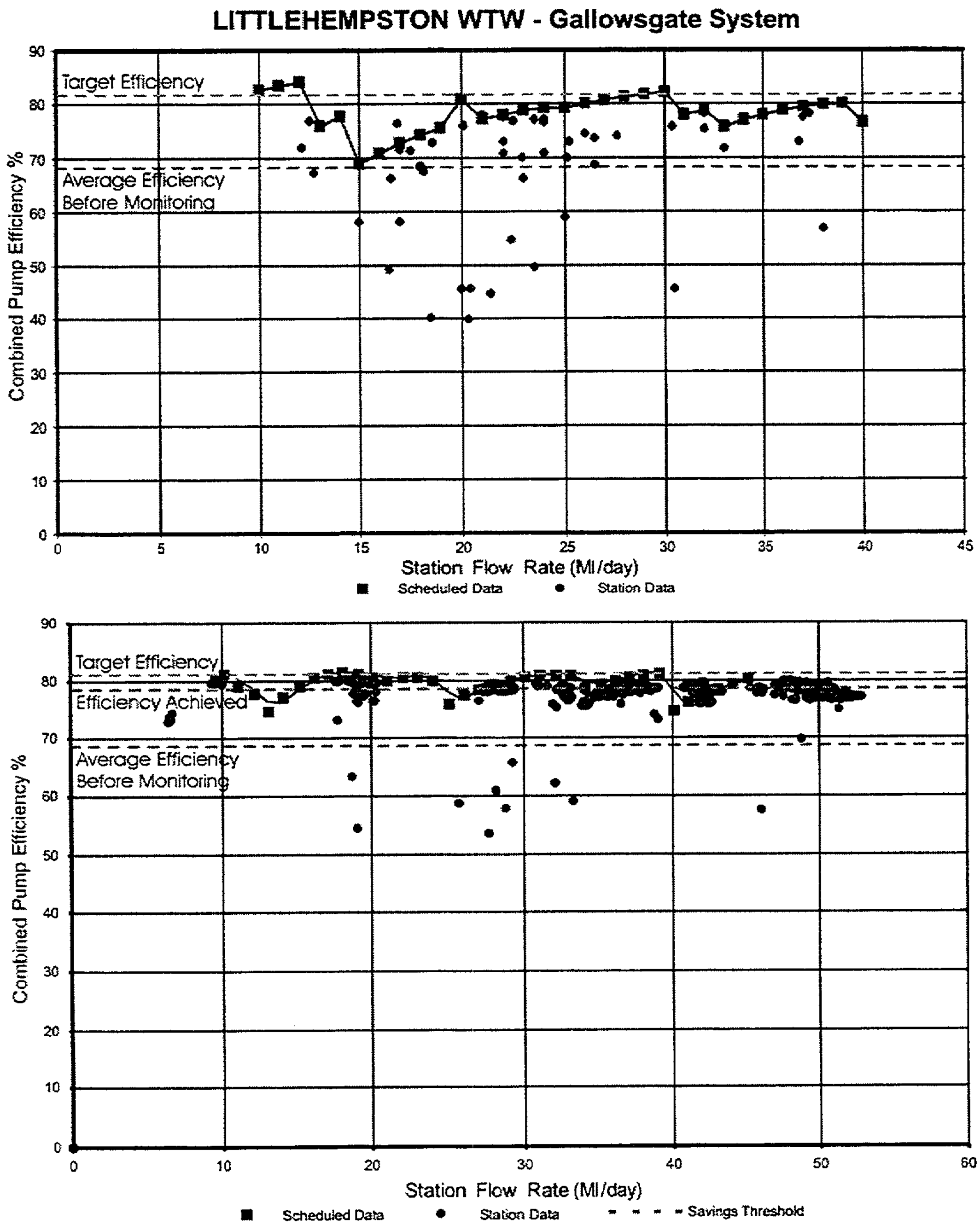


Figure 12 Pumping Operation Before And After Installation of Fixed Metering

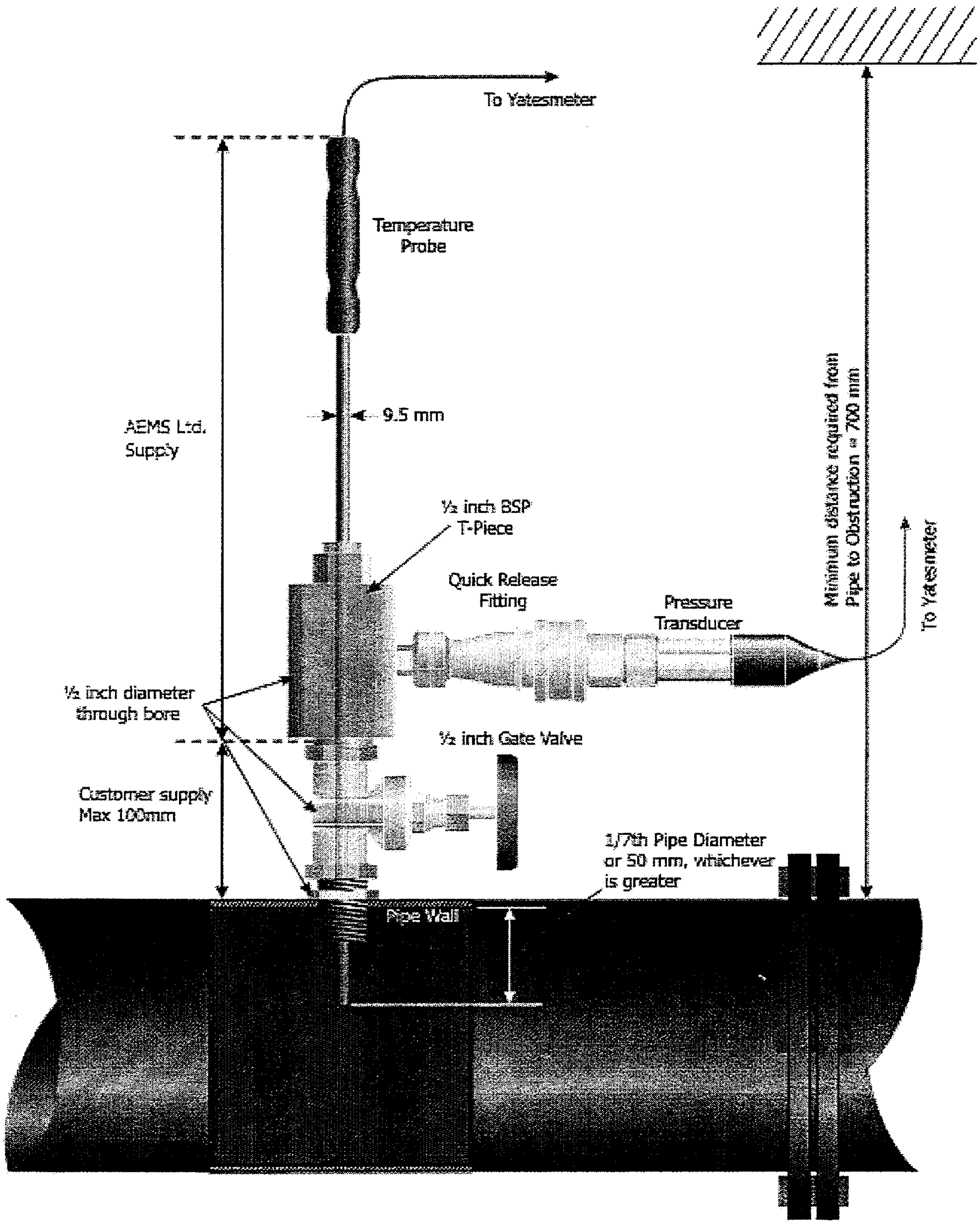

To Yatesmeter
Temperature Probe
AEMS Ltd. Supply
9.5 mm
½ inch BSP T-Piece
Quick Release Fitting
Pressure Transducer
Minimum distance required from Pipe to Obstruction = 700 mm
To Yatesmeter
½ inch diameter through bore
½ inch Gate Valve
Customer supply Max 100mm
1/7th Pipe Diameter or 50 mm, whichever is greater
Pipe Wall

Figure 13

APPENDIX

Thermodynamic Primary Test Data			
dT (mK)	Ts (°C)	ps (m)	pd (m)
1,243.92	29.01	48.84	1,885.55
1,120.69	30.30	48.22	1,793.58
1,093.51	32.24	47.29	1,594.04
1,170.44	33.42	53.44	1,453.65
1,777.90	28.94	47.59	1,980.10
2,517.23	27.10	48.09	2,054.11
3,234.81	25.96	48.15	2,106.03

Calculated Thermodynamic Data					
H (m)	Ep (%)	Pgr (kW)	Em (%)	Q (l/s)	N (rpm)
1,837.7	83.8	4154	91.79	177.3	3,456
1,746.6	85.1	4384	91.72	199.8	3,450
1,548.5	83.5	4707	92.05	238.3	3,453
1,402.3	80.1	4868	92.25	261.5	3,459
1,933.1	77.2	3765	91.87	140.8	3,473
2,006.4	69.3	3594	91.19	115.5	3,477
2,058.2	63.3	3527	91.40	101.1	3,460

Thermodynamic Test Data Corrected for Sp. Gr. and Referred to 3465 rpm			
H (m)	**Q (m^3/hr)**	**Ep (%)**	**P (kW)**
1849.1	621.2	81.4	3420.0
1761.8	704.5	83.0	3625.5
1557.7	844.7	81.9	3896.5
1405.8	927.9	78.7	4017.7
1926.1	486.8	74.4	3057.2
1996.6	394.6	66.2	2902.7

Sulzer Test Data Referred to 3465 rpm			
H (m)	**Q (m^3/hr)**	**Ep (%)**	**P (kW)**
2069.2	340.3	59.1	2882.2
2010.3	399.3	66.9	2903.8
1942.0	496.6	76.3	3058.9
1865.8	615.8	80.8	3437.5
1772.9	705.8	83.3	3631.7
1576.0	840.0	82.1	3902.4
1425.2	936.1	80.2	4025.3

Session 1(b)

Motor-Operated Valves I

Session Chair

Kevin G. DeWall
Idaho National Engineering
and Environmental Laboratory

Analysis of Required Actuator Torque for Rising Rotating Stem Globe Valve

Paul Swinburne
Entergy Nuclear Northeast

Abstract

This paper presents an analysis based on a first principles force balance for a rising rotating stem motor-operated valve (MOV). The rising rotating stem actuator drive design is commonly employed with small forged body globe valves. The examples presented include evaluation of thrust and torque requirements for a typical rising rotating valve application, considering both guide based and seat based differential pressure load cases.

Introduction

Rising rotating stem MOVs are a motor operation adaptation of a manual valve design. The handwheel is replaced with a spline drive adapter (see Figure 1 photograph) which turns and slides within a special internally splined actuator drive sleeve. A stationary Acme threaded yoke bushing converts the rotation of the stem into axial thrust. The yoke bushing is typically manganese bronze or similar material and often includes a grease fitting to reduce friction and facilitate lubrication. Because the spline adapter moves axially within the actuator, the length of travel is limited but, since the rising rotating configuration is only used on small valves, this restriction does not present any problem. Rising rotating configuration globe valves are primarily used for small forged body, high-pressure applications.

Analysis Approach

Thrust and torque analyses for rising rotating valves were typically performed with the same standard formulations for conventional (non-rotating) stem globe valves. However, MOV diagnostic testing has showed that rising rotating stem valves behave differently. Stem packing friction loads are significantly lower than the typical 1000 lb./in. of stem diameter. However, on closer examination the low apparent packing load is reasonable when you consider that the stem motion is primarily rotational. What was observed as the stem packing load is actually only the axial component of the packing friction force vector. Appendix 3 shows a typical MOV diagnostic test report for a rising rotating stem valve with the low apparent packing friction load. In the Figure 2 vector diagram, F_p is packing friction force resisting motion of the stem and acting in an opposite direction. This vector consists of a rotational component, F_{pr} and an axial component, F_{pa}. The packing friction torque, τ_{pf} is developed by the rotational packing friction, F_{pr} and the moment arm of the stem radius ($d_{stem}/2$). Thus:

$$\tau_{pf} = F_{pr} \cdot \frac{d_{stem}}{2} \qquad [1]$$

Also for the Figure 2 vector diagram, the axial component of packing friction, F_{pa} may be related to packing friction torque, τ_{pf} by the tangent of the thread lead angle such that:

$$F_{pa} := \frac{2 \cdot \tau_{pf}}{d_{stem}} \tan(\alpha) \qquad [2]$$

Appendix 1 provides a specific example of the relationship between axial packing friction and packing friction torque.

The problem noted with the much lower than expected packing load during MOV tests highlighted the need for a better analysis model for the rising rotating stem globe valve. Figure 3 shows free body diagrams of stem, disk and spline drive of a rising rotating valve for close (flow under seat) and open (flow over seat).

The primary difference with a conventional (rising non-rotating stem) globe valve is that the actuator only applies a torque load to the stem and thrust (F_{yb}) is developed by a reaction torque at the yoke bushing (τ_{yb}). Additionally there is a relatively small spline sliding friction force (F_{sp}). As previously discussed, the packing friction has both an axial force component (F_{pa}) and a torque component (τ_{pf}). At the valve disk, there may be an additional term for the disk-to-stem bearing friction torque (τ_{db}). This torque term should be included in the evaluation for the seat based case (differential pressure force based on seat area) when the disk is bearing on the seat with the stem still rotating relative to the disk. The remaining terms, F_{se} (stem end force), F_{dp} (differential pressure force) and F_{seal} (sealing force at seat) are the same as in a conventional (non-rotating) globe valve analysis.

Appendix 2 shows the analysis of the Figure 3 free body diagram model. For the case of close stroke with flow under the seat and guide area based DP load (Case 1) the required actuator torque, τ_{act} is:

$$\tau_{act} := \frac{\tau_{pf} + \left(F_{pa} + F_{se} + F_{dp}\right) SF}{1 - \dfrac{2\mu_{sp} \cdot SF}{d_{sp}}} \qquad [3]$$

For the case of close stroke with flow under the seat and seat area based DP load and sealing force (Case 2) the required actuator torque, τ_{act} is:

$$\tau_{act} := \frac{\tau_{pf} + \tau_{db} + \left(F_{pa} + F_{se} + F_{dp} + F_{seal}\right) SF}{1 - \dfrac{2\mu_{sp} \cdot SF}{d_{sp}}} \qquad [4]$$

And for the case of open stroke with flow over the seat with guide area based DP load (Case 3) the required actuator torque, τ_{act} is:

$$\tau_{act} := \frac{\tau_{pf} + \left(F_{pa} - F_{se} + F_{dp}\right) SF}{1 - \dfrac{2\mu_{sp} \cdot SF}{d_{sp}}} \qquad [5]$$

Where the terms in these formulas are:

- τ_{pf} Packing friction torque
- F_{pa} Packing axial friction force
- F_{se} Stem end (ejection) load
- F_{dp} Differential pressure (DP) load on disk
- SF Stem factor (ratio of torque to thrust force) for stem threads in yoke bushing
- μ_{sp} Drive spline friction coefficient
- d_{sp} Drive spline mean diameter
- τ_{db} Stem to disk bearing friction (seat based only)
- F_{seal} Seat sealing force (seat based only)

As noted previously the packing friction torque, τ_{pf} is related to the packing axial friction force, F_{pa} by the thread lead angle, α such that:

$$F_{pa} := \frac{2 \cdot \tau_{pf}}{d_{stem}} \tan(\alpha) \quad [2]$$

And:

$$\tau_{pf} := \frac{F_{pa}}{\tan(\alpha)} \cdot \frac{d_{stem}}{2} \quad [2a]$$

Application Considerations

The above formulas are based on a simple force balance with no 'valve factor' addition. As such they should not be used as written for blowdown flow conditions. As reported in the Electric Power Research Institute (EPRI) Performance Prediction Program (PPP) (Reference 1) tests on an unbalanced globe valve under compressible flow (hot water blowdown) conditions resulted in significantly higher thrusts than predicted. These higher thrust levels were attributed to side loading on the disk due to circumferential pressure variations.

The model equations [3], [4] and [5] (as developed in Appendix 2) do not include gravity load terms for stem and disk weight. As noted in the EPRI PPP, these loads are generally small and may usually be neglected.

The EPRI PPP Globe Valve Model (Reference 1, Table 6.1) notes that packing friction may be neglected for rising rotating stems. This is not explained in the EPRI PPP report but, as shown with the Appendix 2 example, the axial component of packing friction, F_{pa} is small. As discussed in the introduction, this phenomenon is noticed during MOV diagnostic testing. Since the EPRI PPP Globe Valve Model only considers thrust loads (and not torque loads) the effect of the rising rotating packing friction torque is not addressed.

The sealing force, F_{seal} in the Appendix 2 example (seat based case) is the stem force to develop a seat stress for leak-tight seal with a metal-to-metal contact seat. The method to calculate this sealing force comes from the EPRI Application Guide for MOVs in Nuclear Power Plants (Reference 2, section 5.1.5.10).

The rising rotating stem model equations developed in Appendix 2 contain several different friction terms. The EPRI PPP (Reference 1) provides a valuable source of friction data that may be applied to this model. The spline to actuator friction, μ_{sp} was assumed as 0.2. This term is similar to the torque reaction friction, μ_T used in the EPRI PPP, including the globe valve non-rotating stem case. EPRI recommends 0.5 for μ_T which may be reasonable for an external torque reaction arm. However, for a well-lubricated spline internal to the actuator, a lower friction coefficient (0.2) is more reasonable. The denominator in equations [3], [4] and [5] is very similar to the EPRI PPP Torque Reaction Factor (TRF). In the various EPRI PPP models, the TRF applies at the location where a reaction torque is applied to prevent stem rotation. This is inside the valve body for various PPP gate valve models or at a torque arm or anti-rotation key for non-rotating globe valves. The TRF in the EPRI PPP models is a factor that increases the required thrust from the torque and bearing friction to resist rotation. The equation [3], [4] and [5] denominator term increases required actuator torque due to sliding friction where the torque is applied to the valve rotating parts.

The conditions that cause the disk to stem bearing friction torque, τ_{db} are similar to friction conditions studied in the EPRI PPP Separate Effects Tests. The bearing conditions are probably most similar to the edge-on-flat or flat-on-flat cases. The Figure 3 model is based on the Edward Univalve® design (Reference 3). The disk to stem bearing is a Stellite wire inserted into a hole in the body guided disk. The wire is fed around adjacent circular grooves on the inside bore of the disk and outside diameter of the stem. Then the wire is welded to the disk at the hole securing the disk on the stem while allowing free rotation with a Stellite wire bearing. With this configuration, the stainless steel to Stellite coefficient of friction of 0.5 used in the Appendix 2 example is consistent with the EPRI PPP friction data (edge on flat) (Reference 1, Table 5-3).

Conclusion

This analysis might be interpreted to indicate that the rising rotating stem design is not particularly efficient. However, this is not the intent of the analysis. The rising rotating design is appropriate for the applications where it is used. As noted in the Appendix 2 conclusion, the packing friction torque load tends to predominate for most cases. Since this design is generally only used for small valves with small diameter stems, the torque requirements remain well within the torque capability of the smaller MOV actuators, in spite of the inefficiencies of the rotating stem friction. This analysis provides some insights into the performance of rising rotating stem globe valves that may help with the understanding of MOV diagnostic test results.

References

1. *EPRI MOV Performance Prediction Program: Topical Report, Revision 2,* EPRI, Palo Alto, CA, 1997. TR-103237-R2.

2. *Application Guide for Motor-Operated Valves in Nuclear Power Plants, Volume 1, Revision 1: Gate and Globe Valves,* EPRI, Palo Alto, CA, 1999. TR-106563-V1.

3. Edward Univalve® Globe Valve Catalog, Edward Vogt Valve Co., Raleigh, North Carolina (available at http://www.edwardvalves.com)

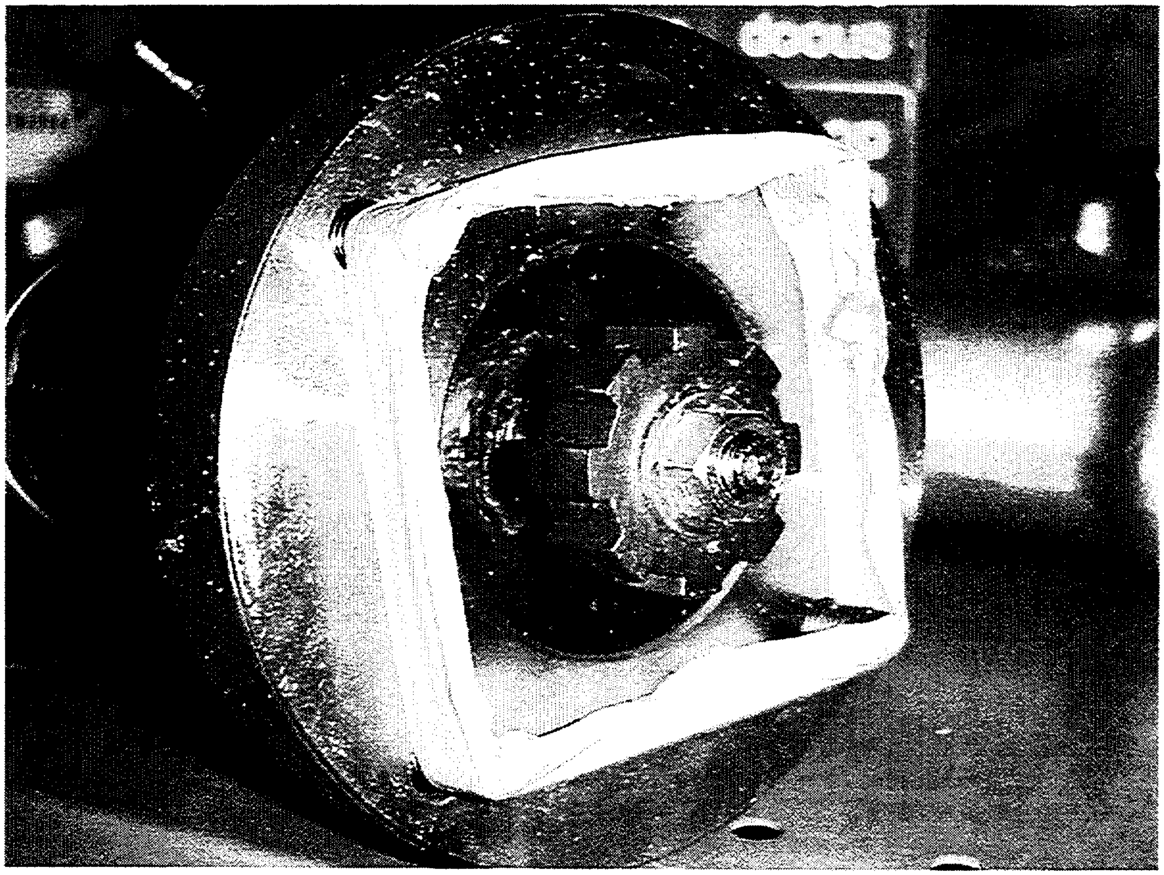


Figure 1

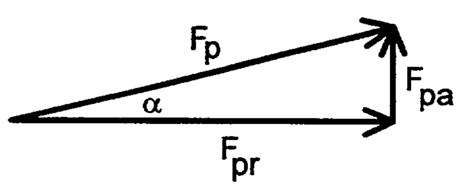


Figure 2

Where:

F_p is resultant packing friction vector
F_{pa} is axial component of packing friction
F_{pr} is rotational component of packing friction
α is the thread lead angle

Figure 2

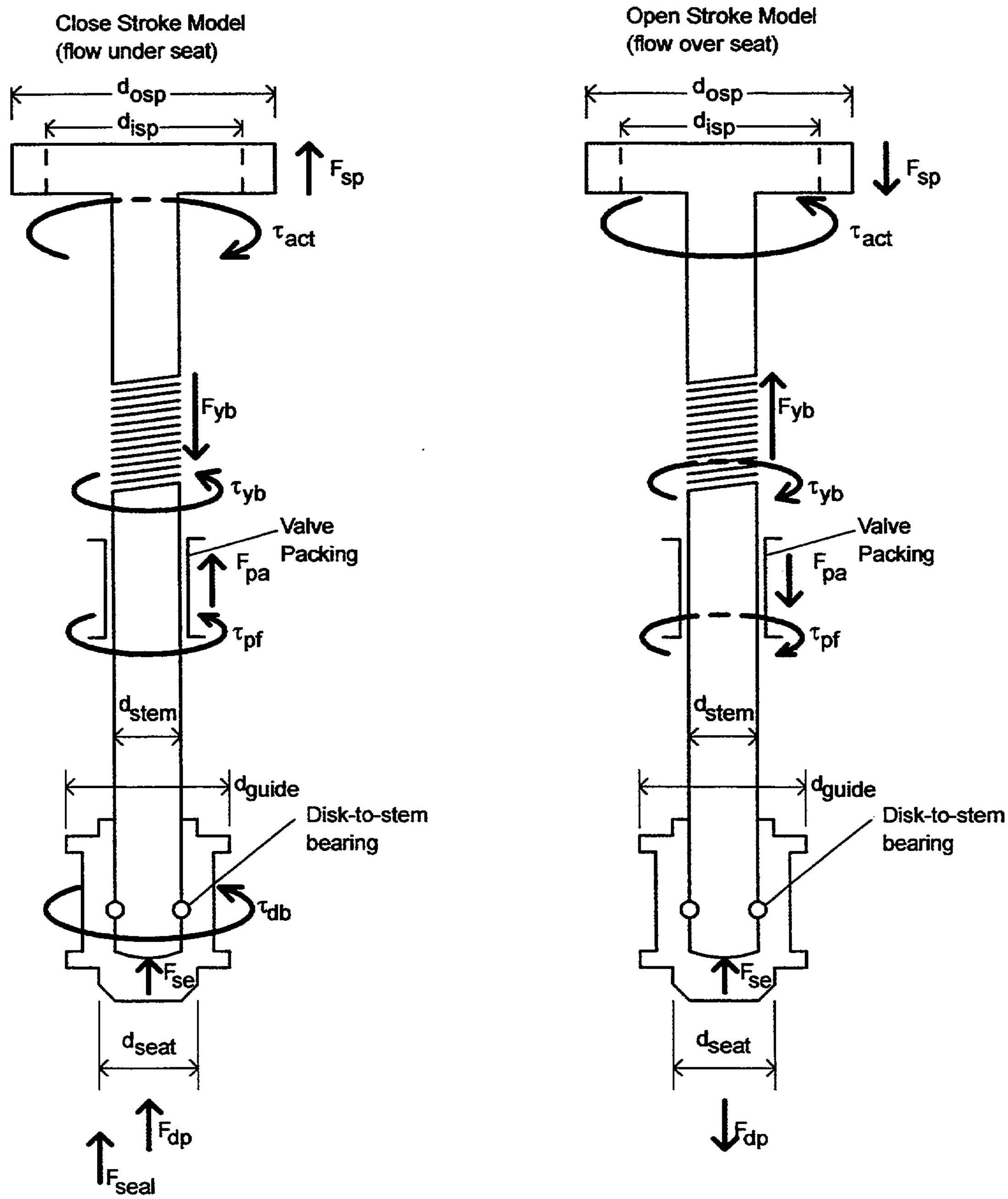

Close Stroke Model
(flow under seat)
Open Stroke Model
(flow over seat)
dosp
disp
Fsp
τact
Fyb
τyb
Valve
Packing
Fpa
τpf
dstem
dguide
Disk-to-stem
bearing
τdb
Fse
dseat
Fdp
Fseal

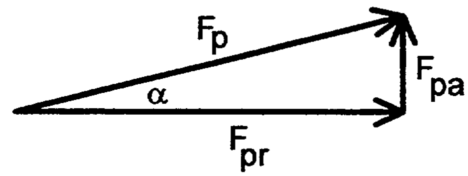
Figure 3

This appendix shows an example of a measured axial packing friction thrust measurement may be related to packing friction torque.

Stem data.. $d_{stem} := 0.75\text{in}$ $p := 0.125\text{in}$ (pitch) $\text{Lead} := 0.125\text{in}$

Thus the pitch diameter is.. $d_p := d_{stem} - 0.5p$ $d_p = 0.6875\ \text{in}$ $d_p = \frac{11}{16}\ \text{in}$

Vector diagram for packing friction..



From VOTES test data.. $F_{pa} := 25\text{lbf}$ (Typical of the "low" close direction axial packing friction noted in MOV testing.)

The thread lead angle is given by.. $\alpha := \text{atan}\left(\frac{\text{Lead}}{\pi d_p}\right)$ $\alpha = 3.312\ \text{deg}$

The packing rotational friction force is related to the packing friction torque by the moment arm which is the stem radius, thus the packing friction torque is..

$$\tau_{pf} = F_{pr} \cdot \frac{d_{stem}}{2}$$ [1] where F_{pr} is the packing rotational packing friction force

The vector diagram relates the axial and rotational packing friction forces by the thread lead angle such that: $\frac{F_{pa}}{F_{pr}} = \tan(\alpha)$

Thus.. $F_{pr} := \frac{F_{pa}}{\tan(\alpha)}$ [2] and evaluating.. $F_{pr} = 432.0\ \text{lbf}$

Converting from packing rotational force to torque (combining [1] and [2]) , we have.. $\tau_{pf} := \frac{F_{pa}}{\tan(\alpha)} \cdot \frac{d_{stem}}{2}$ $\tau_{pf} = 13.5\ \text{ft} \cdot \text{lbf}$

The above shows that a small "apparent" axial packing friction force corresponds to substantial (for this size stem) packing friction torque load.

ANALYSIS OF RISING ROTATING VALVE

Input values for both the guide and seat based analysis cases..

d_{guide}	Guide diameter	p	Thread pitch
d_{stem}	Stem diameter	L	Thread lead
d_{ost}	Seat outside diameter	μ_s	Stem thread friction
d_{ist}	Seat inside diameter	μ_{sp}	Spline to actuator friction
d_{isp}	Spline minor (inner) diameter	μ_{seat}	Seat to disc friction (for sealing force)
d_{osp}	Spline major (outer) diameter	μ_{db}	Disc to stem bearing friction (seat based case only)
d_{db}	Diameter for disc bearing friction	τ_{pf}	Torque for packing friction (assumed value)
P_{up}	Upstream pressure	P_r	Seat contact stress for sealing
P_{down}	Downstream pressure	θ	Seat angle (from stem axis)

Typical values for Edward size 1 Figure D36124ML Univalve. Application is a steam line drain valve (to condenser).

$d_{guide} := 1.07\text{in}$ $\quad p := 0.1\,\text{in}$ $\quad P_{up} := 1050\text{psi}$ (main steam pressure)

$d_{stem} := 0.625\text{in}$ $\quad L := 0.1\,\text{in}$ $\quad P_{down} := -13.5\text{psi}$ (condenser vacuum)

$d_{ist} := 0.68\text{in}$ $\quad \mu_s := 0.15$ (stem thread friction)

$d_{ost} := 0.775\text{in}$ $\quad \mu_{sp} := 0.2$ (assumed value, lubricated but similar materials)

$d_{isp} := 1\frac{5}{32}\text{in}$ $\quad \mu_{db} := 0.5$ (assumed value, stellite to stainless steel)

$d_{osp} := 1\frac{15}{32}\text{in}$ $\quad \mu_{seat} := 0.5$ (assumed value, stellite to stellite)

$d_{db} := d_{stem}$ $\quad \tau_{pf} := 9\text{ft}\cdot\text{lbf}$ (assumed, may provided by vendor and may be verified with testing)

$\theta := 45\text{deg}$ $\quad P_r := 6000\text{psi}$ (recommended for sealing 1000 psi line pressure)

Calculated values..

Mean spline diameter..	$d_{sp} := \frac{d_{isp} + d_{osp}}{2}$	$d_{sp} = 1.313\ in$
Differential pressure..	$DP := P_{up} - P_{down}$	$DP = 1063.5\ psi$
Sealing force (seat based only)..	Seat area.. $A_{seat} := \frac{\pi \cdot (d_{ost} + d_{ist}) \cdot (d_{ost} - d_{ist})}{4 \sin(\theta)}$	$A_{seat} = 0.154\ in^2$
Sealing contact force..	$R_r := P_r \cdot A_{seat}$	$R_r = 921.2\ lbf$
Stem sealing force (seat based only)..	$F_{seal} := R_r \cdot \sin(\theta) + \mu_{seat} \cdot R_r \cdot \cos(\theta)$	$F_{seal} = 977.1\ lbf$
Pitch diameter (ACME threads)..	$d_p := d_{stem} - 0.5p$	$d_p = 0.575\ in$
Stem factor from ACME screw thread formula..	$SF := \frac{\cos(14.5deg) \cdot \frac{L}{\pi} + d_p \cdot \mu_s}{2\left(\cos(14.5deg) - \frac{\mu_s \cdot L}{\pi d_p}\right)}$	$SF = 0.00508\ ft$
Thread lead angle..	$\alpha := \operatorname{atan}\left(\frac{L}{\pi d_p}\right)$	$\alpha = 3.169\ deg$

Definitions of torque and forces (see Figure 3)

<u>Torques</u>

τ_{act}	Actuator applied torque to spline
τ_{yb}	Reaction torque at yoke bushing
τ_{db}	Torque from disc to stem bearing friction (only applies to seat based case)

<u>Forces</u>

F_{se}	Stem end force, downstream pressure on stem area
F_{dp}	DP force on disc (guide or seat based depending on case)
F_{pa}	Axial component of packing friction
F_{yb}	Axial force on stem from yoke bushing
F_{sp}	Spline sliding friction force

Case 1 - Guide Based (close stroke flow under seat)

Force and torque balance for guide based case. Assume disc rotates with stem and neglect any disc to body rotating friction (or consider included with packing friction).

$\sum M = 0 \qquad \tau_{act} = \tau_{yb} + \tau_{pf}$ [1]

$\sum F = 0 \qquad F_{yb} = F_{sp} + F_{pa} + F_{se} + F_{dp}$ [2]

Stem factor relates torque and thrust force at yoke bushing such that.. $SF = \dfrac{\tau_{yb}}{F_{yb}}$ [3]

Spline axial force comes from applied torque and mean spline diameter.. $F_{sp} = \dfrac{2\cdot\tau_{act}}{d_{sp}}\cdot\mu_{sp}$ [4]

Axial component of packing force from packing friction torque and lead angle such that.. $F_{pa} := \dfrac{2\cdot\tau_{pf}}{d_{stem}}\tan(\alpha)$ $\quad F_{pa} = 19.1\,\text{lbf}$

Stem end load.. $F_{se} := P_{down}\cdot\dfrac{\pi\cdot d_{stem}^{\,2}}{4}$ $\quad F_{se} = -4.1\,\text{lbf}$ (under vacuum downstream)

DP load (no valve factor used).. $F_{dp} := DP\cdot\dfrac{\pi\cdot d_{guide}^{\,2}}{4}$ $\quad F_{dp} = 956\,\text{lbf}$

Rearranging [3] and substituting into [2].. $\dfrac{\tau_{yb}}{SF} = F_{sp} + F_{pa} + F_{se} + F_{dp}$ [5]

Substituting [1] and [4] into [5].. $\dfrac{\tau_{act} - \tau_{pf}}{SF} = \dfrac{2\cdot\tau_{act}}{d_{sp}}\cdot\mu_{sp} + F_{pa} + F_{se} + F_{dp}$ [6]

Rearranging [6].. $\dfrac{\tau_{act}}{SF} - \dfrac{2\cdot\tau_{act}}{d_{sp}}\cdot\mu_{sp} = \dfrac{\tau_{pf}}{SF} + F_{pa} + F_{se} + F_{dp}$ [7]

Solving [7] for actuator torque.. $\tau_{act} := \dfrac{\tau_{pf} + \left(F_{pa} + F_{se} + F_{dp}\right)\cdot SF}{1 - \dfrac{2\mu_{sp}\cdot SF}{d_{sp}}}$ $\quad \tau_{act} = 14.2\,\text{ft}\cdot\text{lbf}$

Evaluating forces..

$F_{sp} := \frac{2 \cdot \tau_{act}}{d_{sp}} \cdot \mu_{sp}$ $\quad F_{sp} = 51.9\,\text{lbf}$ (spline friction force)

$F_{yb} := F_{sp} + F_{pa} + F_{se} + F_{dp}$ $\quad F_{yb} = 1023.2\,\text{lbf}$ (yoke bushing force)

$F_{pa} = 19.1\,\text{lbf}$ $\quad F_{se} = -4.1\,\text{lbf}$ $\quad F_{dp} = 956.3\,\text{lbf}$

Evaluating torques..

$\tau_{yb} := F_{yb} \cdot SF$ $\quad \tau_{yb} = 5.2\,\text{ft} \cdot \text{lbf}$ $\quad \tau_{act} = 14.2\,\text{ft} \cdot \text{lbf}$ $\quad \tau_{pf} = 9\,\text{ft} \cdot \text{lbf}$

Case 2 - Seat Based (close stroke flow under seat)

Force and torque balance for seat based case with sealing force. Assume stem rotates on disc with sealing and DP force acting as bearing force.

$$\sum M = 0 \qquad \tau_{act} = \tau_{yb} + \tau_{pf} + \tau_{db} \qquad [8]$$

$$\sum F = 0 \qquad F_{yb} = F_{sp} + F_{pa} + F_{se} + F_{dp} + F_{seal} \qquad [9]$$

For seat based case the DP force is redefined in terms of outside seat diameter (per EPRI PPP globe valve model)

$F_{dp} := DP \cdot \frac{\pi \cdot d_{ost}^{2}}{4}$ $\quad F_{dp} = 501.7\,\text{lbf}$

Seal force was previously determined.. $\quad F_{seal} = 977\,\text{lbf}$

Torque for disc to stem bearing friction..

$\tau_{db} := \mu_{db} \cdot \left(F_{seal} + F_{dp}\right) \cdot \frac{d_{db}}{2}$ $\quad \tau_{db} = 19.3\,\text{ft} \cdot \text{lbf}$

Rearranging [3] and substituting into [9]..

$$\frac{\tau_{yb}}{SF} = F_{sp} + F_{pa} + F_{se} + F_{dp} + F_{seal} \qquad [10]$$

Substituting [8] and [4] into [10]..

$$\frac{\tau_{act} - \tau_{pf} - \tau_{db}}{SF} = \frac{2 \cdot \tau_{act}}{d_{sp}} \cdot \mu_{sp} + F_{pa} + F_{se} + F_{dp} + F_{seal} \qquad [11]$$

Rearranging [11].. $\frac{\tau_{act}}{SF} - \frac{2\cdot\tau_{act}}{d_{sp}}\cdot\mu_{sp} = \frac{\tau_{pf} + \tau_{db}}{SF} + F_{pa} + F_{se} + F_{dp} + F_{seal}$ [12]

Solving [11] for actuator torque.. $\tau_{act} := \dfrac{\tau_{pf} + \tau_{db} + \left(F_{pa} + F_{se} + F_{dp} + F_{seal}\right)\cdot SF}{1 - \dfrac{2\mu_{sp}\cdot SF}{d_{sp}}}$ $\tau_{act} = 36.5\,\text{ft}\cdot\text{lbf}$

The above cases show that the disc-to-stem bearing friction and seal force may have considerable impact on the required actuator torque.

Evaluating forces..

$F_{sp} := \frac{2\cdot\tau_{act}}{d_{sp}}\cdot\mu_{sp}$ $F_{sp} = 133.6\,\text{lbf}$ (spline friction force)

$F_{yb} := F_{sp} + F_{pa} + F_{se} + F_{dp} + F_{seal}$ $F_{yb} = 1627.3\,\text{lbf}$ (yoke bushing force)

$F_{pa} = 19.1\,\text{lbf}$ $F_{se} = -4.1\,\text{lbf}$ $F_{dp} = 501.7\,\text{lbf}$ $F_{seal} = 977.1\,\text{lbf}$

Evaluating torques..

$\tau_{yb} := F_{yb}\cdot SF$ $\tau_{yb} = 8.3\,\text{ft}\cdot\text{lbf}$

$\tau_{act} = 36.5\,\text{ft}\cdot\text{lbf}$ $\tau_{pf} = 9\,\text{ft}\cdot\text{lbf}$ $\tau_{db} = 19.3\,\text{ft}\cdot\text{lbf}$

Case 3 - Guide Based (open stroke flow over seat)

Force and torque balance for open stroke. For open stroke all for forces and torques are reversed (except for stem end load). Assume disc rotates with stem and neglect any disc to body rotating friction (or consider included with packing friction).

$\sum M = 0$ $\tau_{act} = \tau_{yb} + \tau_{pf}$ [1] (same as Case 1)

$\sum F = 0$ $F_{yb} = F_{sp} + F_{pa} - F_{se} + F_{dp}$ [13] (sign change for F_{se} from [2])

Stem factor relates torque and thrust force at yoke bushing such that.. $SF = \frac{\tau_{yb}}{F_{yb}}$ [3] (same as Case 1)

Spline axial force comes from applied torque and mean spline diameter.. $F_{sp} = \frac{2 \cdot \tau_{act}}{d_{sp}} \cdot \mu_{sp}$ [4] (same as Case 1)

Stem end load.. $F_{se} := P_{up} \cdot \frac{\pi \cdot d_{stem}^{\ 2}}{4}$ $F_{se} = 322.1\ \text{lbf}$ (for flow over disk, upstream pressure is pressure at stem end)

DP load (no valve factor used).. $F_{dp} := DP \cdot \frac{\pi \cdot d_{guide}^{\ 2}}{4}$ $F_{dp} = 956\ \text{lbf}$

Rearranging [3] and substituting into [13]..

$$\frac{\tau_{yb}}{SF} = F_{sp} + F_{pa} - F_{se} + F_{dp} \qquad [14]$$

Substituting [1] and [4] into [14]..

$$\frac{\tau_{act} - \tau_{pf}}{SF} = \frac{2 \cdot \tau_{act}}{d_{sp}} \cdot \mu_{sp} + F_{pa} - F_{se} + F_{dp} \qquad [15]$$

Rearranging [15]..

$$\frac{\tau_{act}}{SF} - \frac{2 \cdot \tau_{act}}{d_{sp}} \cdot \mu_{sp} = \frac{\tau_{pf}}{SF} + F_{pa} - F_{se} + F_{dp} \qquad [16]$$

Solving [16] for actuator torque..

$$\tau_{act} := \frac{\tau_{pf} + \left(F_{pa} - F_{se} + F_{dp}\right) \cdot SF}{1 - \frac{2\mu_{sp} \cdot SF}{d_{sp}}} \qquad \tau_{act} = 12.6\ \text{ft} \cdot \text{lbf}$$

The above result is similar to Case 1. The reduced actuator torque is due the stem end load with flow over seat (pressure load on stem end assists opening).

<u>Case 4 - Actuator Run Torque (static conditions)</u>

For the case of running torque under static conditions (no pressure or DP) simply eliminate the DP and stem end load terms. Thus:

$$\tau_{act\_run} := \frac{\tau_{pf} + F_{pa} \cdot SF}{1 - \frac{2\mu_{sp} \cdot SF}{d_{sp}}}$$

evaluating for the example conditions.. $\tau_{act\_run} = 9.27\ \text{ft} \cdot \text{lbf}$

The "expected" running packing friction thrust load based on running torque (as if this were a non-rotating stem) would be given by:

$$F_{p\_exp} := \frac{\tau_{act\_run}}{SF} \qquad F_{p\_exp} = 1824.0\,\text{lbf}$$

Ratio of "expected" (non-rotating) to actual axial packing friction for rising rotating..

$$\frac{F_{p\_exp}}{F_{pa}} = 95.34$$

The effect of drive spline friction may be seen by evaluating the denominator. Thus we have..

$$1 - \frac{2\mu_{sp} \cdot SF}{d_{sp}} = 0.981$$

CONCLUSION

These evaluations show us that the rotational torque loads predominate. The packing friction torque is significant in all cases. For the seat based case with sealing force, the disk to stem bearing friction torque is the largest factor contributing to actuator required torque. Because of the small stem diameter and small thread lead used on rising rotating stems, the stem factor, SF is small. This means that the actual torque at the yoke bushing that develops stem thrust is relatively small. The drive spline friction is another parasitic loss, however, evaluation of the denominator shows that this small (increase actuator torque about 2%).

Appendix 3

Evaluator:______________
Date Printed: 1/24/2002 15:53
TEST RESULTS

Tag Number: 29MOV-204A
Test Number: 12
Test Date: 12/1/98

Close			Open		
Stroke Time	19.754	seconds	Stroke Time	19.641	seconds
Bypass Time	1.375	seconds	Bypass Time	8.099	seconds
Max Running Force	-51	lbs	Max Running Force	62	lbs
Avg Running Force	-22	lbs	Avg Running Force	37	lbs
Thrust at CST	-3663	lbs	Disc Pullout Force		lbs
Maximum Thrust	-5033	lbs			
Available Thrust Margin	-3634	lbs			
Spring Pack Preload	-529	lbs			

Torque Sw. Setting O/C: 1.500/1.500

Calibration Range: 0 lbs to -5111 lbs

```
Date Printed:  1/24/2002      15:54
Torque Switch Setting Open/Close...............: 1.500/1.500
Limit Switch Rotor Adjustment (Y/N)............: N
Flow (gpm) Start/Finish........................:     0/     0
Upstream Pressure (psi) Start/Finish...........:     0/     0
Downstream Pressure (psi) Start/Finish.........:     0/     0
General Comments:
WR # 98-03120-03. AS LEFT TEST OF RECORD NEW TORQUE SW INSTALLED PACKING TIGHTEN
TOTAL OF 4 FLATS UP FROM 6 DOWN. REMOVED ADJUSTMENT COLLAR AND CLEANED THREADS
AND REINSTALLED. RISING ROTATING S/P GAP CHECKED AT O2/C2, 0.003 SEEN ON S/P
TRACE. TORQUE SWITCH OPENS AT H/B ON THE OPEN SIDE BUT IS COVERED BY THE BYPASS
MANUAL CAL CSB-32 (CASE 1)
```

Valve Information

Plant: 204A
Unit.: 29
Tag Number......: 29MOV-204A
Type............: GLOBE
Size............: 1"
Target Thrust...: 0 lbs
Orientation.....: VERTICAL
Location........: OVER TIP ROOM
Stem Material...: 416
Stem Diameter...: 0.750 inches
Threads per Inch: 8.00
Threads per Rev.: 1
E/Poisson Ratio.: 115.4 x 10E6 psi
VOTES Serial #..: A7542
BFSL Sensitivity -7.273E-0002 μv/v/lb
Spare Channel Offset: -0.18 in

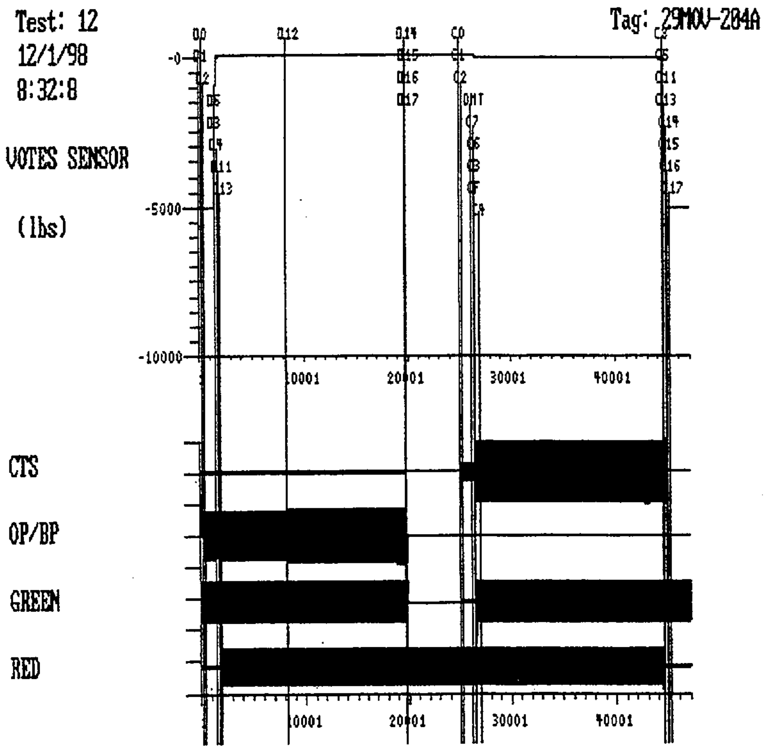
Valve Actuator

Actuator Type..: LIMITORQ
Size...........: SMB-000
Max Thrust Rate: 8000 lbs
Serial #.......: 256570
Order #........: 3B1908A
Worm Gear Teeth: 50
Gear Ratio.....: 100
Spring Pack #....0101-091

Actuator Motor

Voltage Type: AC
Volts.......: 575
Amp rating..: 3.00 amps
Nom. Speed..: 1700.00 rpm
Start torque: 2.00 ft-lb
Run Torque..: 0.40 ft-lb
Horse Power.: 0.10 h.p.

Signal Conditioner Calibration Due Date 11/13/99



Calibration Range: 0 lbs to -5111 lbs

```
Date Printed:  1/24/2002      15:53
Torque Switch Setting Open/Close...............: 1.500/1.500
Limit Switch Rotor Adjustment (Y/N)............: N
Flow (gpm) Start/Finish........................:     0/    0
Upstream Pressure (psi) Start/Finish...........:     0/    0
Downstream Pressure (psi) Start/Finish.........:     0/    0
General Comments:
WR # 98-03120-03. AS LEFT TEST OF RECORD NEW TORQUE SW INSTALLED PACKING TIGHTEN
TOTAL OF 4 FLATS UP FROM 6 DOWN. REMOVED ADJUSTMENT COLLAR AND CLEANED THREADS
AND REINSTALLED. RISING ROTATING S/P GAP CHECKED AT O2/C2, 0.003 SEEN ON S/P
TRACE. TORQUE SWITCH OPENS AT H/B ON THE OPEN SIDE BUT IS COVERED BY THE BYPASS
MANUAL CAL CSB-32 (CASE 1)
```

Valve Information	Valve Actuator	Actuator Motor
Plant: 204A	Actuator Type..: LIMITORQ	Voltage Type: AC
Unit.: 29	Size...........: SMB-000	Volts.......: 575
Tag Number......: 29MOV-204A	Max Thrust Rate: 8000 lbs	Amp rating..: 3.00 amps
Type............: GLOBE	Serial #.......: 256570	Nom. Speed..: 1700.00 rpm
Size............: 1"	Order #........: 3B1908A	Start torque: 2.00 ft-lb
Target Thrust...: 0 lbs	Worm Gear Teeth: 50	Run Torque..: 0.40 ft-lb
Orientation.....: VERTICAL	Gear Ratio.....: 100	Horse Power.: 0.10 h.p.
Location........: OVER TIP ROOM	Spring Pack #....0101-091	
Stem Material...: 416		
Stem Diameter...: 0.750 inches		
Threads per Inch: 8.00		
Threads per Rev.: 1		
E/Poisson Ratio.: 115.4 x 10E6 psi		
VOTES Serial #..: A7542	Signal Conditioner Calibration Due Date 11/13/99	
BFSL Sensitivity -7.273E-0002 μv/v/lb		
Spare Channel Offset: -0.18 in		

ATTACHMENT 3 - MOV TESTING EVALUATION

Page 7 of 8

32. List any other observations or comments not covered in this checklist or in general comments or any kind of disagreements with general comments.

This valve is a rising rotating. The Votes report indicates the average running loads to be approximately 30 lbs. Based on the calculated torque in the running region and dividing it by the Stem Factor we should be seeing approximately 300 lbs. Typically with rising rotating valve the running loads are not consistent with expectations. The COF/Stem Factor are above normal expectations at (.22/.0084) Based on the As left testing we have significant margin. The COF/Stem Factor is acceptable for rising rotating valves. A manual calibration was performed which is not unusual for a rising rotating valve. The calibration and trace was marked in accordance with ESB-32 Case 1. All set point criteria was met.

A new Torque Switch was installed. The old torque switch was identified during testing ~~to~~ as ~~Fatty~~ Faulty. At hammerblow on the open stroke the torque switch opens, at this time the bypass is in the circuit and stays in for 41% of the open stroke.
No operability concerns AWB 12-1-98

This valve meets the expectations set forth per acceptance criteria. Running loads are low however Rising Rotating valves indicate less than actual by a factor of 10x. Red light should be adjusted at next opportunity however this valve has an auto close and will close 100%. If this valve is not replaced during R-14 I recommend testing for trending @ R-14 for Torque Switch repeatability 2nd Party Complete
[signature] 12-1-98

Analyzer: [signature] Date: 12-1-98

MP-059.37 Rev. No. 07	ANALYSIS OF MOV DIAGNOSTIC TESTING USING LIBERTY TECHNOLOGIES "VOTES" SYSTEM*	Attachment 3 Page 82 of 88

Proper Setup and Verification of Limitorque® Position-Seated SB and SBD Actuators

Robert Cantwell
Crane Nuclear, Inc.

Abstract

The SB/SBD compensator is designed to allow the stem nut to float (move axially) within the drive-sleeve/compensator assembly during stem loading. Therefore, limit switch travel may occur without the corresponding stem / disk movement. This has lead to the improper set-up of some limit close valve actuators.

Introduction

This paper will address factors which should be considered to properly set up position-seated Limitorque® model SB and SBD Operators, provide an explanation of the dynamic effect on the operator, and provides a comparison of actual data from static and dynamic testing.

Limitorque® Corporation developed the SMB series multi-turn operator for use in a wide variety of applications. The units are typically utilized to control or isolate process flow through gate, globe, plug, ball, and butterfly valves. The SB and SBD model operators utilize a spring loaded stem nut to dampen seating shock and absorb thermally induced loads in high stem speed and high temperature applications. The SB model incorporates a single-compensating spring pack assembly to dampen or absorb valve-closing forces while the SBD model incorporates a double-compensating spring pack assembly to handle closing and opening forces.

Recent testing experience has shown that some position-seated Limitorque® model SB and SBD Operators may not have been set up to fully close under dynamic conditions. Model SB and SBD Operators that are position-seated should also meet or exceed the minimum calculated thrust requirement at control switch trip (CST) to ensure closure under dynamic conditions. The fact that the diagnostic signatures may indicate hard seat contact during static testing does not ensure hard seat contact during dynamic conditions.

Standard Limitorque® model SMB operators that are position-seated will normally travel to the same position during static or dynamic conditions. Therefore, it may not be necessary to achieve the minimum required thrust value at control switch trip.

Discussion

Limitorque® model SB and SBD Valve Operators set up to coast into the seat or set up with seating forces less than the minimum required thrust to overcome DP may not fully seat. On SB and SBD model operators, the stem nut is allowed to slide axially along splines within the drive sleeve/compensator assembly during stem loading; therefore, limit switch travel may occur without the corresponding stem/disk movement. The limit switch setting controls the number of drive sleeve revolutions needed for full valve travel on the SMB and SB/SBD model operators. But under dynamic conditions or periods of Load Sensitive Behavior, this relationship will become altered on the SB and SBD model operators. Valve operators that are not set up properly during static testing may require additional drive sleeve revolutions to achieve the same overall stem/disk position under dynamic conditions or load sensitive behavior. Position-seated SB and SBD Limitorque® Operators will travel to the sum of the compensator deflection and the actual stem travel.

NSSS Vendors supplied several plants with Limitorque® model SB and SBD operators. The NSSS Vendors normally supplied minimum Nut Deflection (ND) values to ensure seat closure during dynamic conditions. The ND or compensator deflection values were based on NSSS Vendor testing and compensator belleville spring design data. Specified compensator deflection values were assumed to be equivalent to a design basis stem thrust. Since the compensator spring assemblies are similar in design to the worm spring pack assemblies, they are also subject to the same uncertainties. During periods of prolonged compensator deflection with the valve fully closed, compensator belleville springs/washers are also susceptible to fatigue and may not have the same spring characteristics as new washers. The fatigue may induce spring pack gaps that cause the compensator assembly to indicate compensator deflection without actual spring compression. Improper maintenance activities can produce the same effect invalidating the original ND to stem thrust values as assumed by the NSSS Vendor. During testing, one should not assume that the compensator displacement values are correct, but measure the thrust and torque values using diagnostic equipment. The advantages of diagnostic equipment allows measurement of stem torque, stem thrust, spring pack displacement, stem position, compensator displacement, and various other actuator parameters.

Initially, there was a perceived benefit of limit close operators generating the same thrust during static and dynamic conditions and, therefore, Rate of Loading (ROL) uncertainties were not considered in the design basis evaluation. ROL or Load Sensitive Behavior (LSB) uncertainties need to be included in the design basis evaluation to account for the changes in torque that may be required to meet the design thrust requirement. The measured torque value during dynamic conditions will be higher due to LSB / ROL issue. The baseline static test must include error adjustments or allowances for increases in torque during dynamic conditions.

NRC Generic Letter 89-10, Supplement 6, Enclosure 1, Page 7 reads as follows:

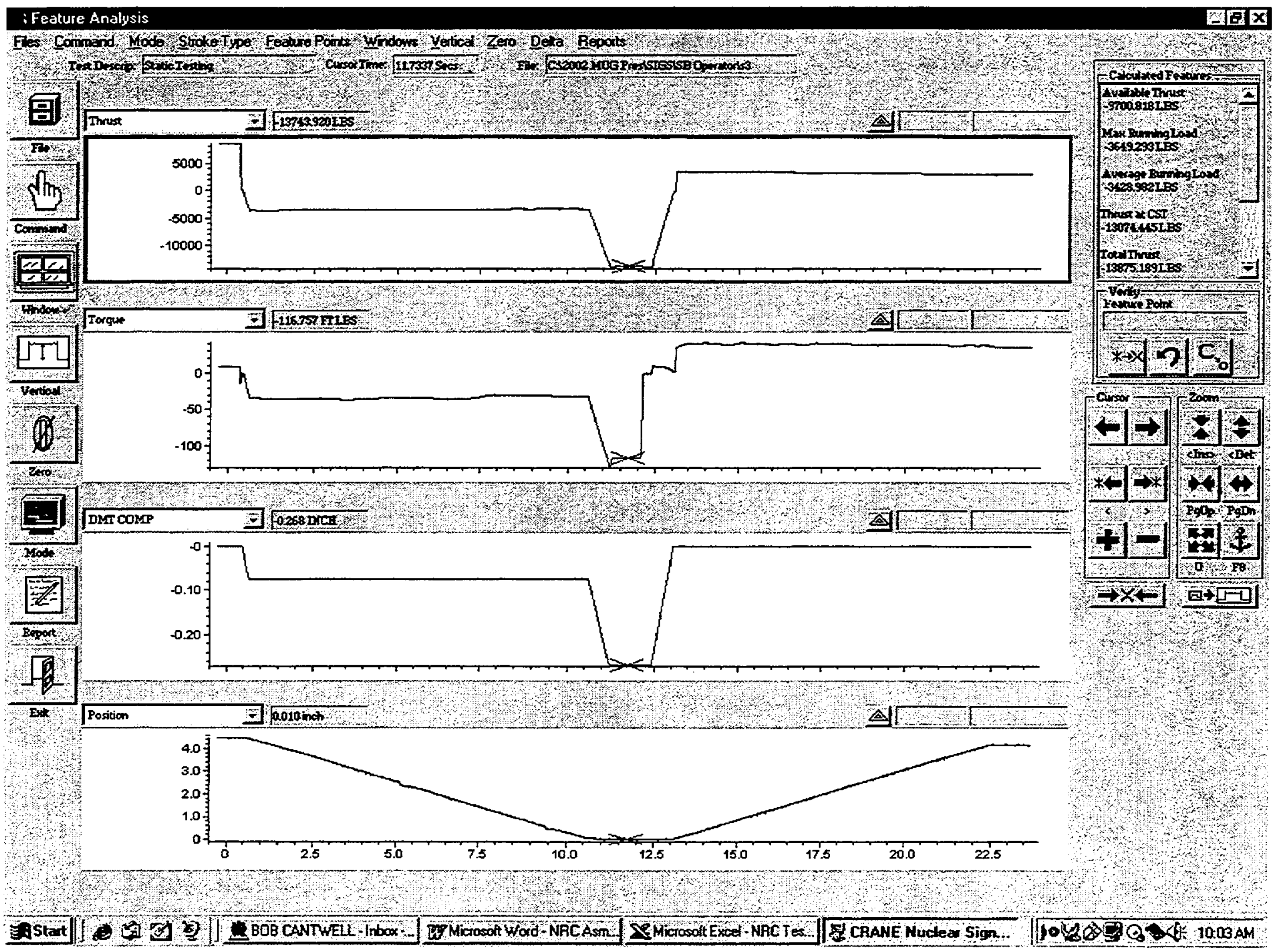
2. Differential Pressure Test Acceptance:

- The valve fully opens with appropriate torque switch bypass indication and fully closes with diagnostic indication of hard seat contact and control room indication.
- The control switch settings provide adequate thrust margin to overcome design-basis requirements, including consideration of diagnostic equipment inaccuracy, control switch repeatability, load sensitive behavior, and, margin for degradation until the next test.
- The motor output capability at degraded voltage is in excess of the control switch setting including consideration of diagnostic equipment inaccuracy, control switch repeatability, load sensitive behavior, and, margin for degradation until the next test.

- The maximum thrust and torque achieved by the MOV including diagnostic equipment inaccuracy and control switch repeatability do not exceed the allowable structural capability limits for the individual parts of the MOV.
- The diagnostic traces do not indicate any significant abnormalities or anomalies that might affect MOV operability.

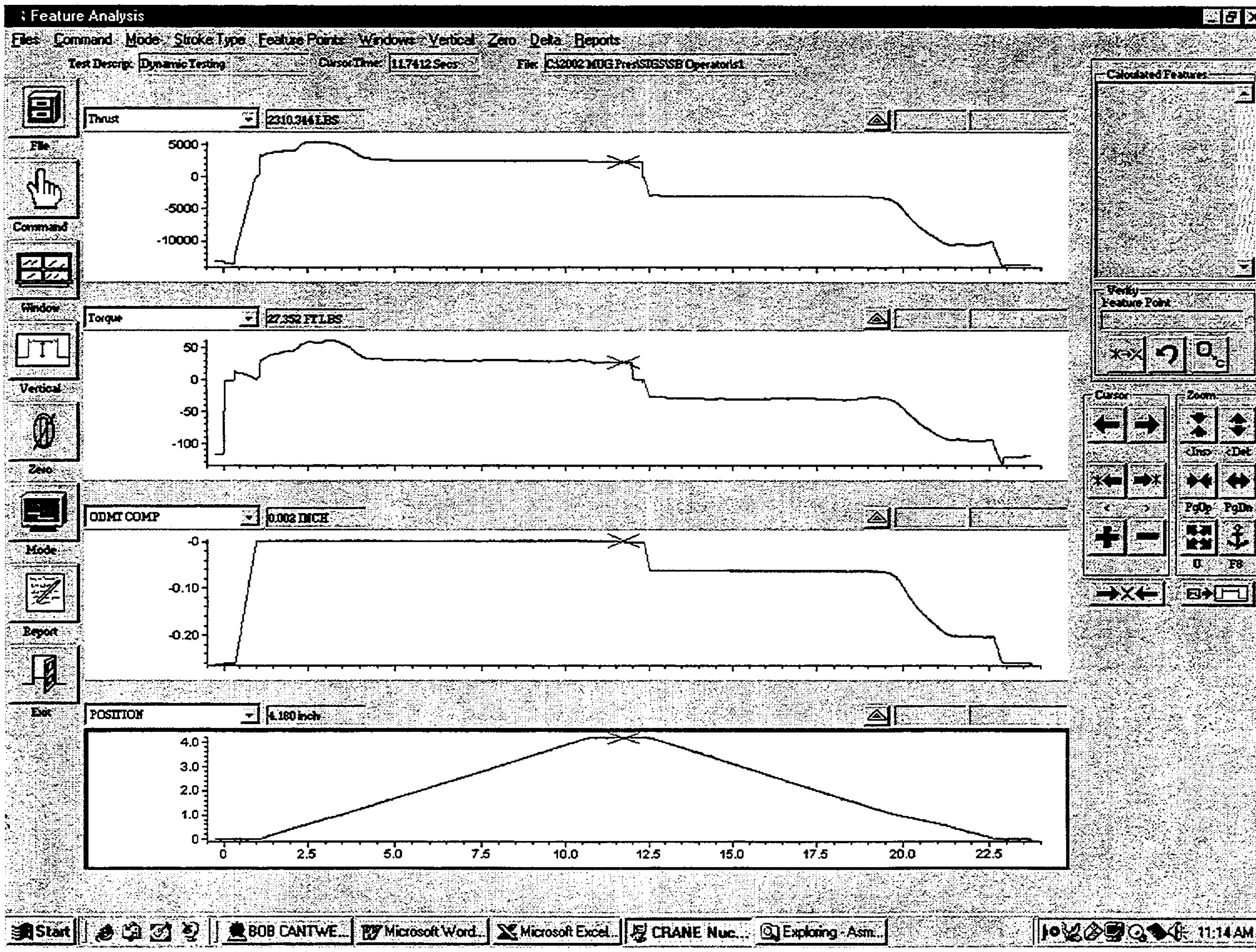
Static Test Results – SB 00 Actuator

The following plot is of a valve fully open, traveling closed and returning to the full open position under static conditions. The purpose of this plot is to illustrate the different characteristics observed during both static and dynamic diagnostic testing. The first window is Thrust, second window is Torque, third window is Compensator Displacement, and the last window is Stem Position.



Dynamic Test Results – SB 00 Actuator

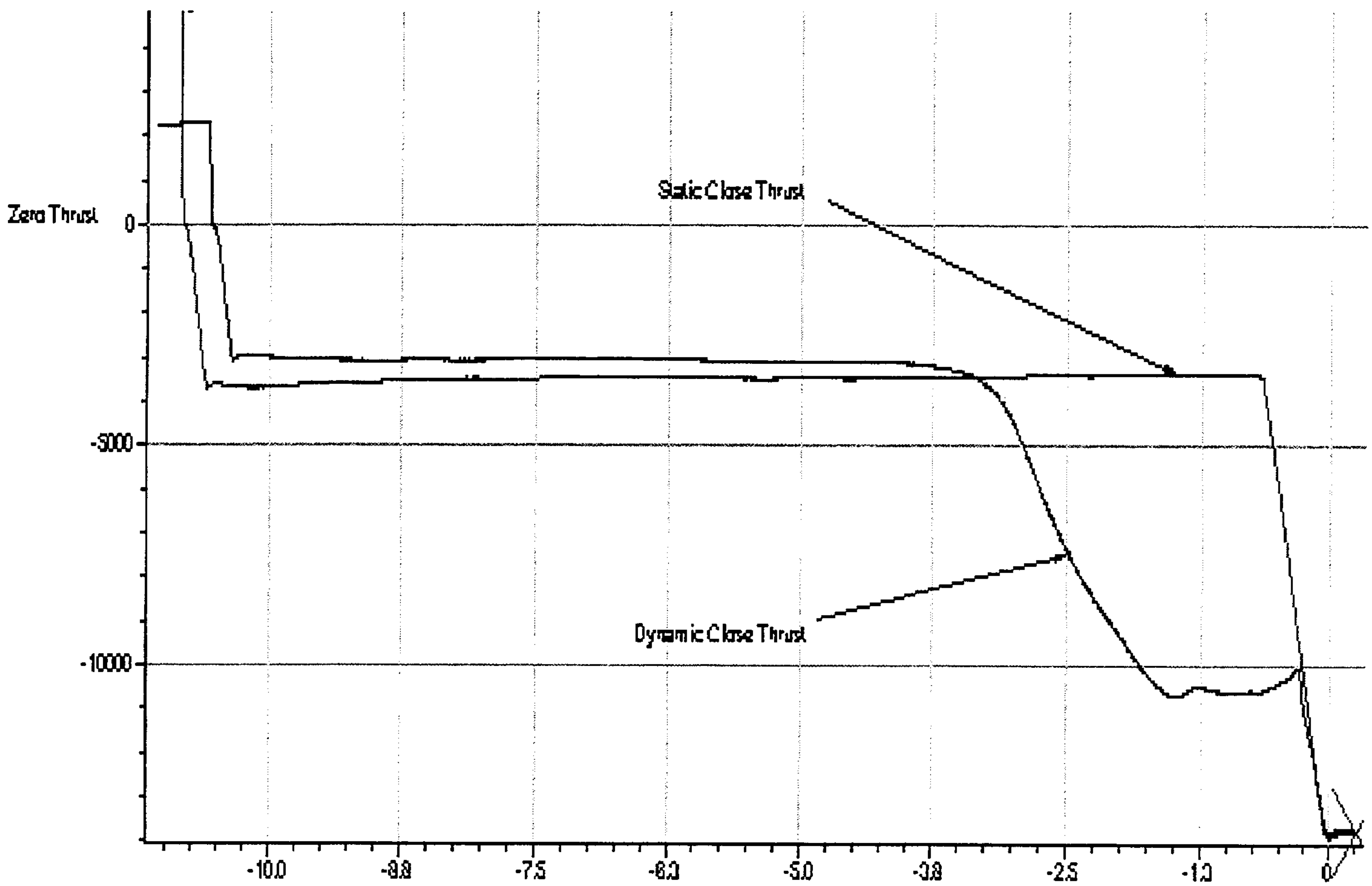
The following plot is of a valve fully closed, traveling open and returning to the fully closed position under dynamic conditions. The displayed data indicates changes in the signature profile due to the dynamic conditions. Again the first window is Thrust, second window is Torque, third window is Compensator Displacement, and the last window is Stem Position.



Stem Thrust Overlay (Static and Dynamic)

This plot is an overlay of stem thrust data from the dynamic and static testing. The data shows the valve producing very similar thrust data at control switch trip and total thrust values.

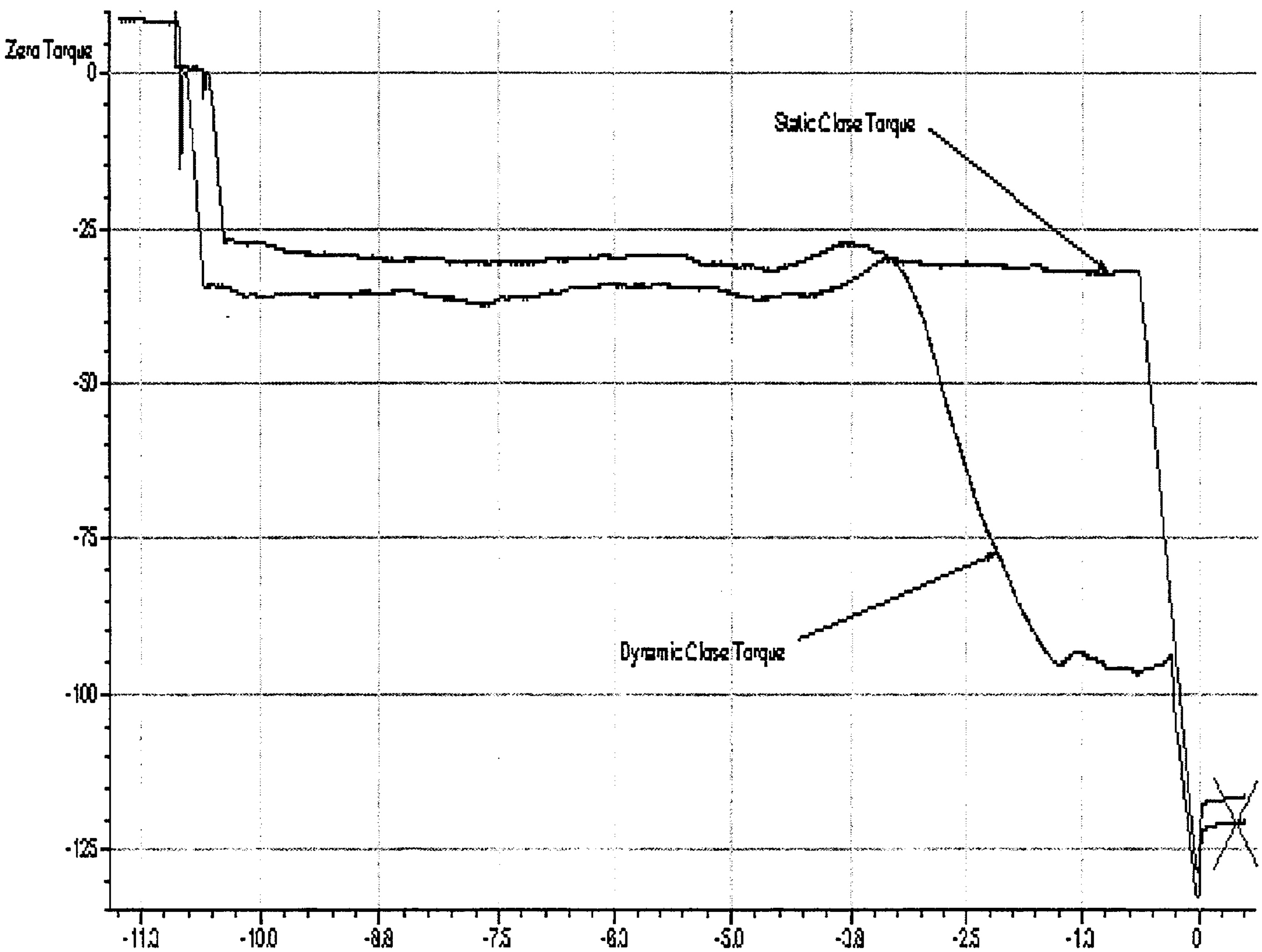
Closing Thrust (lbs)	Dynamic	Static
File Name:	S1	S4
Available Thrust:	10186.516	9700.818
Max Running Load:	3084.834	3649.293
Average Running Load:	4331.895	3428.982
Thrust at CST:	13271.35	13074.445
Total Thrust:	13848.936	13875.189
Max DP Effect:	10685.34	0
Hard Seat Contact:	9989.611	3373.6



Stem Torque Overlay (Static and Dynamic)

This plot is an overlay of stem torque data from the dynamic and static testing. The plot displays the complete valve stroke from fully open to the fully closed position.

Closing Thrust (lbs)	Dynamic	Static
File Name:	S1	S4
Available Torque:	99.833	88.893
Max Running Torque:	31.796	37.267
Average Running Torque:	40.599	34.282
Torque at CST:	127.185	121.202
Total Torque:	132.826	128.552
Max DP Effect:	96.927	0
Hard Seat Contact:	94.021	32.48

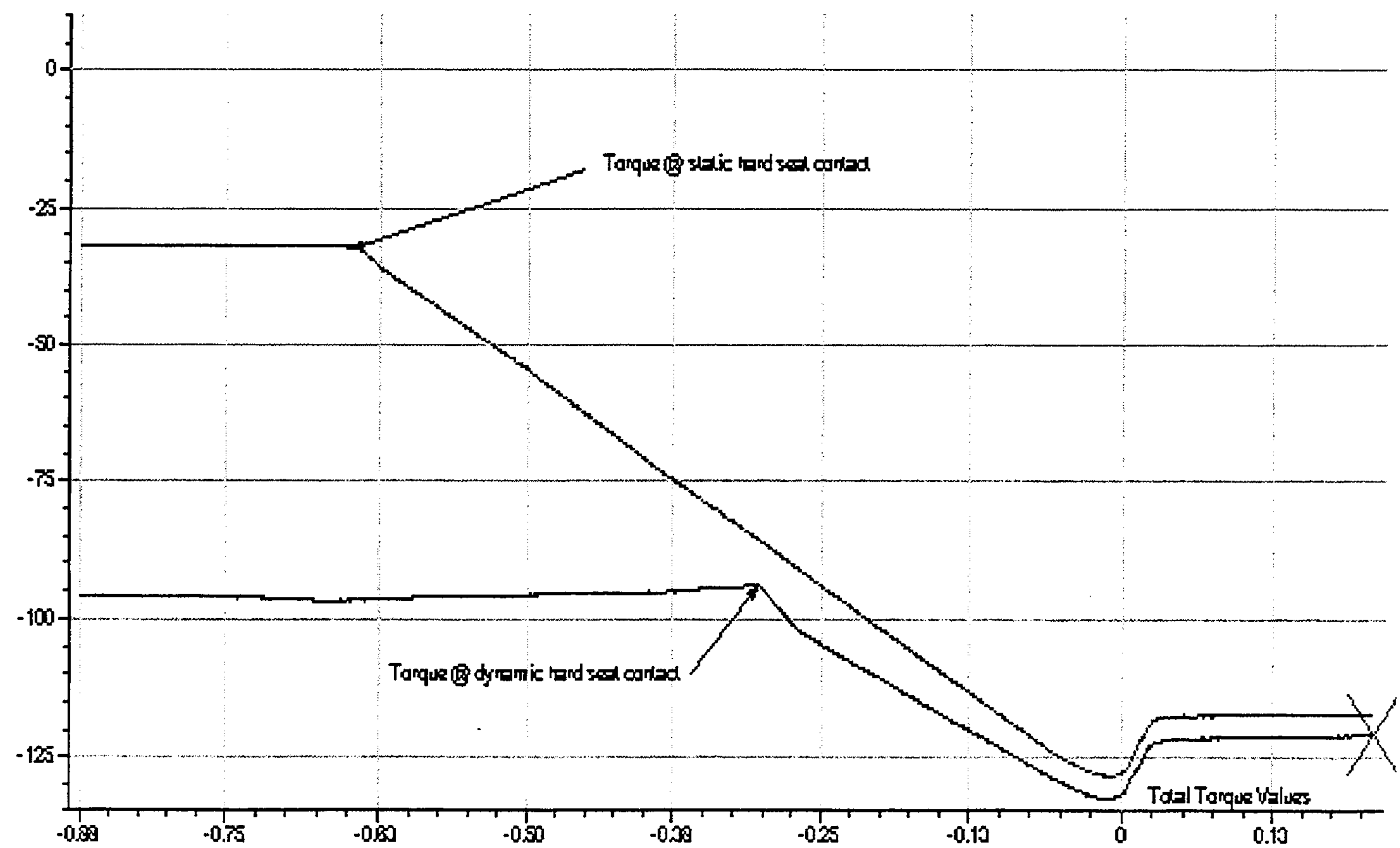


Stem Torque Overlay (Seating Area Only)

This plot is an overlay of stem torque data from the dynamic and static testing. The plot displays the portion of the trace during the dynamic effect and the seating. The valve stroked from fully open to the fully closed position. The actuator torque increase due to load sensitive behavior during dynamic loading conditions can be observed in the signature. Torque at Control Switch Trip (CST) increased from 121 ft.lbs. to 127 ft.lbs. and the total torque increased from 128.6 ft.lbs. to 132.8 ft.lbs.

This additional torque must be considered to ensure that none of the actuator torque limits are exceeded under dynamic conditions.

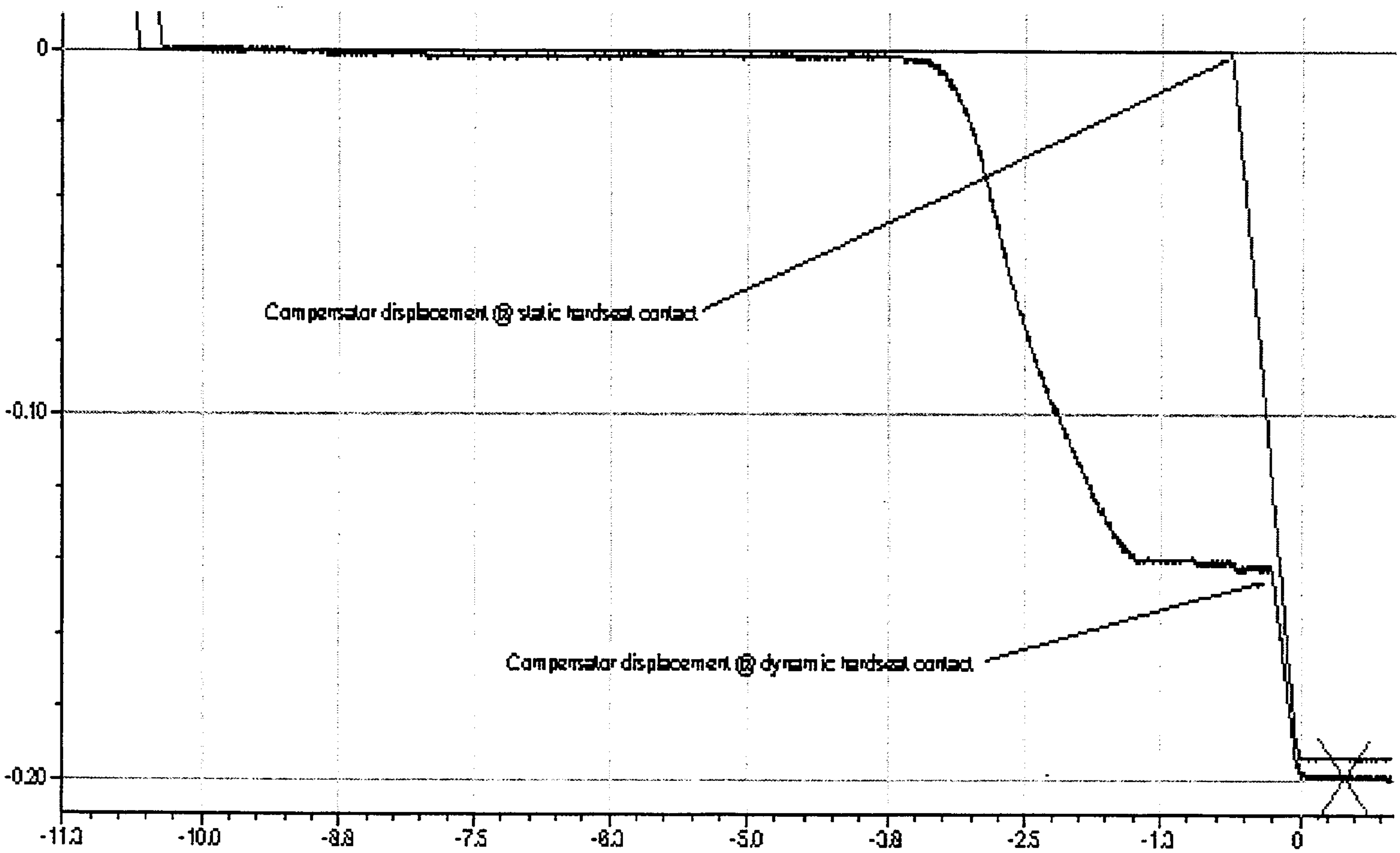
Closing Thrust (lbs)	Dynamic	Static
File Name:	S1	S4
Available Torque:	99.833	88.893
Max Running Torque:	31.796	37.267
Average Running Torque:	40.599	34.282
Torque at CST:	127.185	121.202
Total Torque:	132.826	128.552
Max DP Effect:	96.927	0
Hard Seat Contact:	94.021	32.48



Compensator Movement (Static and Dynamic)

This plot is an overlay of the compensator displacement data from the dynamic and static testing also. The plot displays the portion of the trace during the dynamic effect and the seating. The lines point to the hardseat contact during static and dynamic conditions. The static displacement at hardseat contact is 0.073 inch and the dynamic displacement is 0.203 inch. The valve stroked from fully open to the fully closed position.

Compensator Displacement (Inch)	Dynamic	Static
File Name:	S1	S4
Max Running Displacement:	0.064	0.074
Average Running Displacement:	0.063	0.073
Displacement at CST:	0.249	0.251
Total Displacement:	0.261	0.268
Max DP Effect:	0.205	0
Hard Seat Contact:	0.203	0.073

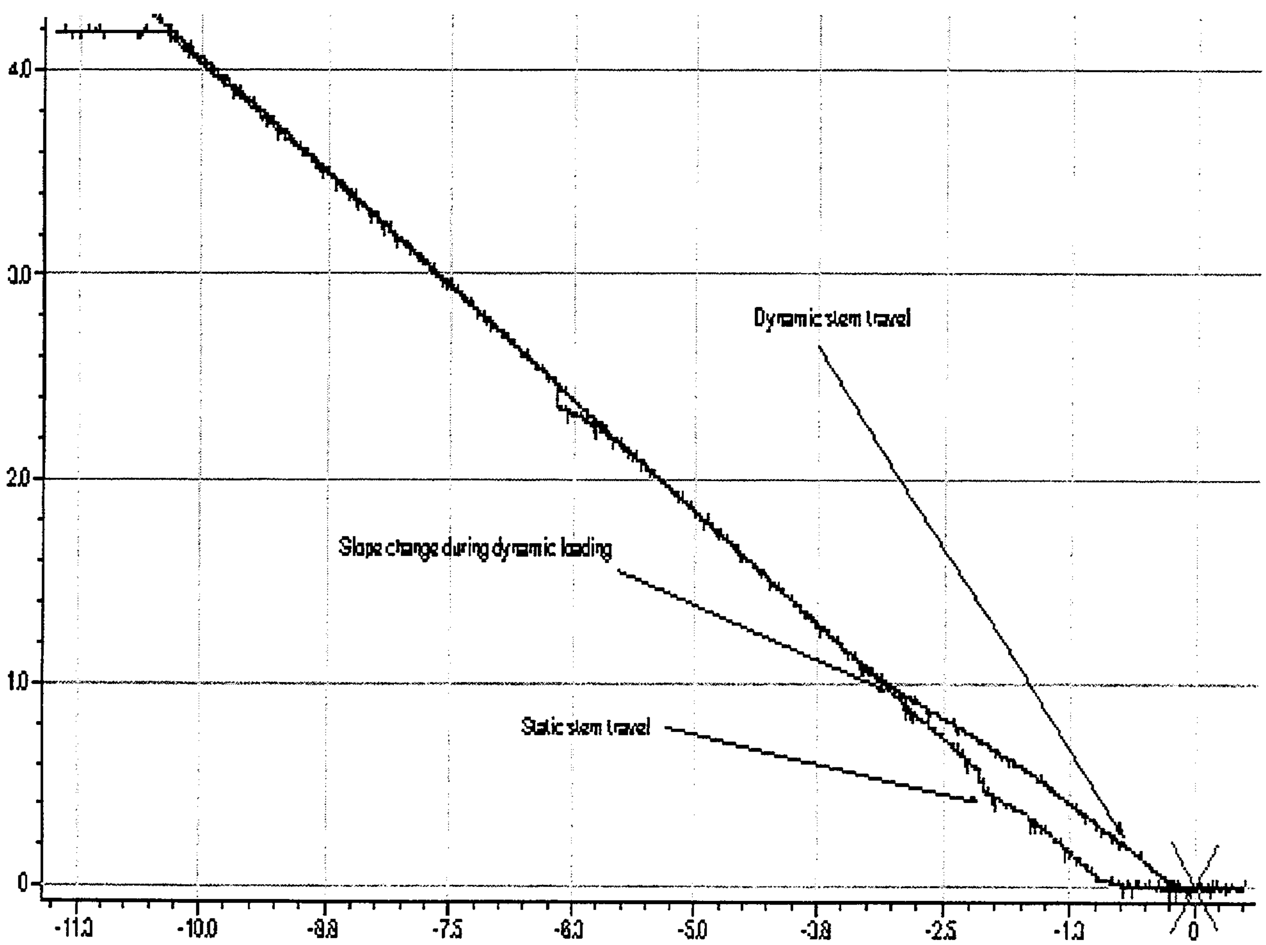


In this case, the total static compensator displacement exceeded the dynamic effect, indicating a proper static setup. Had the compensator movement caused by the dynamic effect been greater than the total static compensator movement, then the valve would not have fully closed.

Stem Position Overlay (Static and Dynamic)

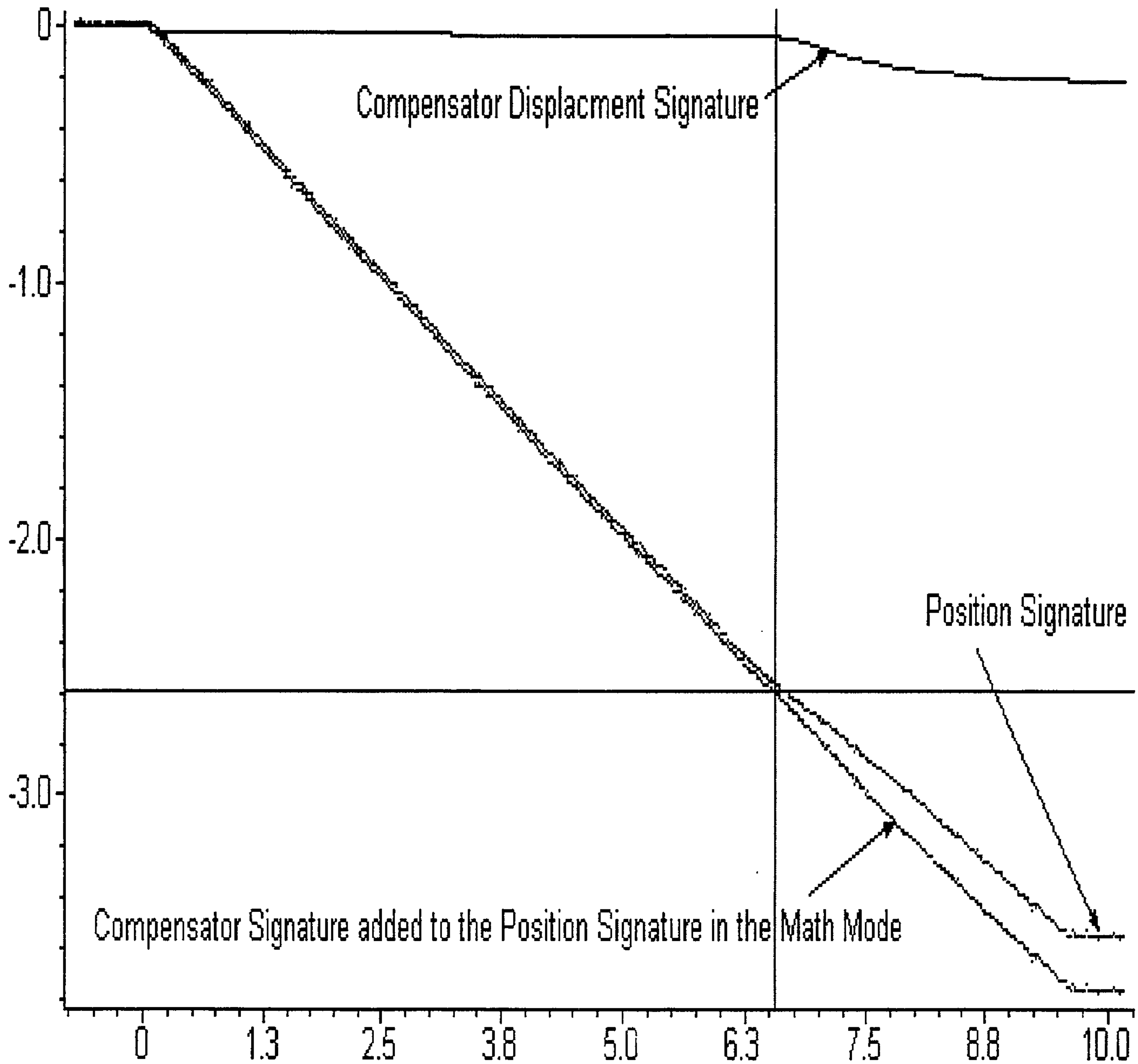
This plot is an overlay of the stem position trace from the dynamic and static testing. The plot displays the full travel from open to close during both static and dynamic conditions. The slope of the line changes during the dynamic trace due to the compensator moving prior to hard seat contact. The rate at which the valve is closing actually decreases during dynamic conditions due to the compensator movement. A percentage of the valve stem nut revolutions and limit switch travel is actually used compressing the spring compensator.

The test data indicates that the valve is setup properly. The stem position trace indicates full stem travel during both the dynamic and static conditions.



Overlay of Stem Position and Compensator Displacement

This plot is another way of examining the data previously collected. The plot is a display of the stem position trace and the compensator displacement trace combined with the resulting data when the two traces are added together. The resulting trace from the addition of the two signatures represents a more linear trace and therefore shows the actual rate of travel has not changed, only the rate at which the stem is traveling in the closed direction.



Summary

Rate of Loading or Load Sensitive Behavior uncertainties need to be included in the design basis evaluation for limit close valves to account for the changes in torque that may be required to meet the design thrust requirement. This allowance for ROL and LSB will ensure that the actuator motor or structural limits are not exceeded.

The limit switch set point controls the number of stem nut revolutions, which does not necessarily translate to relative stem travel.

The static Baseline Test must be set up above the design dynamic thrust requirement to account for compensator movement and ensure the valve reaches the fully closed position.

Acknowledgments

South Texas Project Nuclear Operating Company (STPNOC) / South Texas Project Nuclear Generating Station.

References

NRC Generic Letter 89-10, Supplement 6.

MOV Regulatory Experience in Korea

Walter Kim, Key-Yong Sung and O-Hyun Keum
Korea Institute of Nuclear Safety

Introduction

MOV (motor-operated valve) performance has been one of the major issues in nuclear society due to the fact that MOV design basis capability may not be ensured under some circumstances. The Korean regulatory body has followed foreign regulatory activities on MOVs and reviewed some MOV failure data for a few reactors in Korea. The Korean regulatory requirement on MOV design basis verification and power-operated gate valve (POGV) pressure locking and thermal binding (PLTB) was issued in June 1997, and the Korean utility established a program and initiated program implementation in 1999. The program is to be completed by 2004, approximately one year earlier than requested by the regulatory requirement.

The purpose of this paper is to (1) introduce the historical background and the contents of Korean MOV regulatory requirement, (2) describe how the MOV regulatory work is performed, and (3) introduce the current status of the Korean MOV program, and regulatory experience and concerns. Design basis capability for about half of all safety-related MOVs has been verified as of the end of 2001. Some regulatory concerns have been raised and the Korean utility and regulatory body is making an effort to resolve them appropriately.

Regulatory Requirement

Background

Based on operating experience, research results and regulatory inspection world widely, MOV performance has long been a safety issue because an MOV may need greater force under design basis conditions than under normal operating conditions when it is required to operate. The surveillance requirements of plant technical specifications and ASME Code inservice test requirements were identified to be far too short to assure MOV operability under design basis conditions.

In this regard, the U.S. Nuclear Regulatory Commission (NRC) issued Generic Letter (GL) 89-10 regarding design basis function verification of MOVs and GL 95-07 regarding pressure locking and thermal binding (PLTB) phenomena of power-operated gate valves (POGVs), respectively.

MOV failure data were reviewed in Korea for two 900 MWe Westinghouse (WH) reactors in the late 1980s. The review results are shown in Table 1. The failure data review was based on trouble reports for both units. The trouble report is issued when an abnormal condition related to valve operation is found, and when planned or unplanned maintenance is needed. As shown in Table 1, a total of 227 failures were reported during approximately three years for the two reactors. Failure modes are categorized as failure to open/close, stem or internal leak, trouble/disable light, torque switch failure, limit switch failure, motor failure, double indication, etc.

A total of 27 failures on Motor-Operated Butterfly Valves with EIM actuators were reported in a 600 MWe CANDU reactor for approximately four years since the plant began its commercial operation. Most failures were found to be related to torque switch, limit switch, and mechanical problems.

In addition, MOV failures at two Framatome design plants were also reviewed in 1994. During a surveillance test, two containment isolation valves of the component cooling system were identified as not fully closed.

To address the MOV failure experience in Korea, the Korean regulatory body issued the regulatory requirement on MOV and POGV on June 13, 1997 as follows. NRC regulatory concerns have been considered in making the Korean regulatory requirement

Administrative measure

The licensee is required to follow this regulatory recommendation to ensure design basis capability of MOVs and POGVs.

a. Safety-related MOVs should be reviewed on design bases, verified on switch settings, and tested under design-basis conditions where practicable to show that MOVs can perform their safety functions properly. If valve testing is impracticable, an alternative method should be considered to ensure their design-basis functions and the reasons should be justified with documents.

b. Safety-related POGVs susceptible to pressure locking and/or thermal binding should be reviewed and appropriate corrective actions, such as analysis and/or test, design change, and operation procedure modification, should be followed.

c. Design basis review plan and test plan should be submitted within two years and final safety evaluation should be submitted within eight years from the issue date of this administrative measure for operating reactors. An annual report should be submitted at the end of each year until program completion. Each valve's design basis review report should be submitted two months before the test date of the valve. Organization, training program, evaluation priority, and quality assurance program for this administrative measure should be addressed in the review and test plan.

d. PLTB review plan should be submitted within two years and final safety evaluation should be submitted within five years from the issue date of this administrative measure for operating reactors. The annual report should be submitted by year-end until program completion. Each valve's review report and corrective action plan should be submitted two months prior to the planned corrective action date.

e. Future planned reactors should reflect the regulatory recommendation at the time of plant design and submit documents showing MOV and POGV can perform their design-basis safety functions by the time of commercial operation license issuance.

f. If and when the licensee cannot meet the regulatory schedule specified above, justification should be made in written form and a revised schedule is to be submitted and approved.

Regulatory activity process

Regulatory activities can be categorized into three steps, review of design basis analysis report; on-site inspection for diagnostic test; and review of evaluation result report.

The first step of the regulation process is to review the design basis analysis report submitted by the utility two months prior to the date of refueling outage. Regulatory inspectors are to focus on the design basis differential pressure and operational margin calculation in order to be informed what valve needs to be followed in detail.

On-site regulatory inspection covers static and/or diagnostic test witness and interview with plant personnel. Care is paid to data acquisition system including sensors, test condition, especially development of differential and flow rate. Review for the design basis analysis report also can be performed during the on-site inspection process. For on-site inspection purposes, one or two business trips are planned during the given refueling outage.

If negative margin valves are found and cannot be corrected during the given refueling outage, the utility has to submit an interim safety assessment report for each valve with negative margin to the regulatory body, which contains the utility's interim corrective action plan to ensure valve operability until final corrective action is completed. The interim safety assessment report including corrective action plan is reviewed by the regulatory inspector, and the review result is discussed at the combined regulatory and utility meeting, which is routinely held on site when the reactor is ready to regenerate electricity after the refueling outage. The meeting is not only for MOV concerns but also for all regulatory concerns raised during the outage period. The regulatory inspection summary for MOVs is to be one item to discuss at the meeting if negative MOVs are found.

The last regulatory inspection process is a review of the evaluation report submitted by the utility approximately two months after the date of plant restart. The evaluation report contains all information about each evaluated valve including design basis review and diagnostic test results. Regulatory attention is paid to (1) operational margin, (2) abnormality of diagnostic signals and its physical meanings, (3) corrective actions taken before/during/after diagnostic test, (4) quality assurance, etc. The regulatory inspection report is made every refueling outage incorporating all three regulatory inspection results. Additional regulatory requirements may be imposed as necessary.

Current status of MOV program

Implementation status

The Korean utility, KHNP (Korea Hydraulic and Nuclear Power Corporation), which formally was KEPCO (Korea Electric Power Corporation), began the MOV design basis verification program in 1999. The utility has evaluated design basis capabilities for 774 valves out of approximately 1,800 MOVs as of the end of 2001. The utility is planning to complete the evaluation for all safety-related MOVs by 2004, earlier than requested by the regulatory body. Table 2 shows the total number of MOVs to be evaluated and the current evaluation status as of the end of 2001.

Revision of Government Notice on Inservice testing of pumps and valves

The Government Notice on inservice testing of safety-related pumps and valves is now in the process of revision and is to be issued in June 2002. The main purpose of revising the Notice is to include periodic verification requirement for the MOV and to address design basis verification and periodic verification of safety-related air-operated valves (AOVs). The new Notice will also reflect the revised ASME OM Code and 10 CFR 50 on pump and valve inservice testing, as appropriate.

Regulatory Experience

Design basis differential pressure calculation method

Differential pressure determination is a key task in the evaluation of MOV design basis function because the required force under design basis conditions is significantly affected by differential pressure. The simple Bernoulli equation and Electric Power Research Institute (EPRI) System Flow Model (SFM) were adopted for differential pressure (DP) calculations in the initial stage of the Korean MOV implementation program. However, the utility tried to find ways to reduce the DP through detailed modeling of the piping system for the valves with negative margin, and employed a new code, Flowmaster, for that purpose. The regulatory body requested justification on the use of the Flowmaster code, and how the calculation results from the code are drawn compared to the two methods already used. The regulatory body reviewed the document submitted by the utility and accepted the use of Flowmaster in case (1) fluid inertia is expected in high flow speed piping system, and (2) more detailed calculation is needed to simulate the practical configuration of the piping system in which the valve is installed for the purpose of margin improvement. The calculation procedure of the Flowmaster code has been set up as requested by the regulatory body. Recently, the utility is trying to use another commercial code in DP calculations for two phase flow system to lower the differential pressure. The same regulatory approach will be followed as done for the Flowmaster case.

Design basis capability of Non-Limitorque actuators

Eight different kinds of actuators are installed in Korean plants. Actuator suppliers include Limitorque, Rotork, Autotork, EIM, Hopkinsons, ITT, Jamesbury, and Joucomatic. Design basis capability for the Limitorque actuator is of little concern because lots of information and well-established calculation formula are provided. However, for the other actuators, so called non-Limitorque actuators, only limited or no technical information is supplied by the actuator manufacturers, and this creates difficulty for the utility to estimate actuator capability. The utility is now employing the Limitorque practice and formula in calculating the capability of non-Limitorque actuators. The non-Limitorque actuator capability based on Limitorque practice may or may not be reasonably conservative. Upon requested by the regulatory body, the utility is now trying to assure actuator capability determination through (1) continued effort to obtain necessary technical information from actuator vendors, or (2) testing the actuator itself on an actuator torque test stand, etc. For the actuator torque test stand, the utility has established a temporary test procedure to conform to the quality assurance program. The regulatory review will be done when the test result using the torque test stand is submitted.

Valves with negative operational margin

The MOV design basis verification program is being implemented in all 18 operating plants. Most valves evaluated so far under design basis conditions showed sufficient operational margin. But twenty some valves evaluated during the 2001 time frame showed negative operational margin. Table 3 describes the valves with negative operational margin. Westinghouse design reactors showed higher number of negative margin valves compared to other plant design reactors. The valves installed in chemical and volume control system and main steam system showed a general weakness in operational margin, and similar results are expected for the reactors with similar design, if evaluated. The diagnostic test signal is currently being scrutinized and design basis analysis is being reviewed in more detail by the utility to find ways to improve the margin for the negative margin valves. They are also trying to take measures to modify operational/abnormal/emergency procedure(s) even on a temporary basis, and to modify or replace the valve or actuator part/assembly. The number of valves with negative margin needs to be considered as a temporary number and can be changeable after reflecting re-evaluation results. The regulatory body asked the utility to pay special care to the negative and low margin valves, and special regulatory attention will be paid to those valves in the periodic verification regulatory process.

Worm gear degradation

The Hopkinsons actuators installed in a 25-year-old 600 MWe Westinghouse design reactor were found to have structural weaknesses in the worm gear. While checking diagnostic signals, the utility found severe degradation in worm gear teeth in five actuators enough to challenge valve operation. The degraded worm gears were replaced with new ones, and no abnormal signature trace was found with the replaced worm gear. The regulatory body considered this degradation of the worm gear to be a generic problem for the given actuator type and requested the utility to inspect all Hopkinsons actuators. The inspection result showed 2 to 3 mm wear for all actuators, but it was not considered to impair the proper operation of the valves. The root cause for the degradation of the worm gear was not yet identified, but two possibilities can be assumed: improper selection of worm gear material, or improper maintenance or operation practice of the valve. The degree of degradation for all Hopkinsons actuators will be monitored every refueling outage as requested.

Undersized thermal overload relay

The appropriateness for thermal overload relay size is now being evaluated in parallel with the MOV verification program, and the evaluation results are compared with three acceptance criteria: criterion A, criterion B, criterion C. respectively. Thermal overload relay satisfying criterion A means the relay is sized to assure motor operation and motor protection during the duty cycle. Criterion B means valve operation and motor protection can be ensured only during valve stroking. Criterion C is for ensuring motor operation to the minimum. A total of fifteen thermal overload relays in the safety injection system of CE (Combustion Engineering) design plants were found to be undersized such that even criterion C was not met. A CANDU reactor revealed to have 8 undersized thermal overload relays in the shutdown cooling system. The regulatory body requested the utility to replace the undersized relay with a proper sized one and to review the possibility of this problem in the same nuclear steam supply system (NSSS) design reactors.

PPM applicability for Deloro 40 on Deloro 50 material

A Ni-Cd alloy material, Deloro 40 on Deloro 50, is used in safety-related MOVs in some CANDU reactors in Korea. Deloro 40 on Deloro 50 means a valve adopted

Deloro 40 as a disc seat material and Deloro 50 as a body seat material, respectively. EPRI MOV Performance Prediction Methodology (PPM) was employed for a significant number of valves with Deloro 40 on Deloro 50, because of the high cost for dealing with heavy water and the difficulty of simulating design basis conditions in case they are evaluated by dynamic tests. Responding to the regulatory body's request for the applicability of PPM evaluation to the Deloro 40 Deloro 50 material, the utility submitted a document from the consulting company that participated in the PPM code development. However, the document did not disclose the applicability of PPM to Deloro 40 to Deloro 50 material. It only recommended that Deloro 50 on Deloro 50 coefficient of friction test result be employed to the Deloro 40 to Deloro 50 material because the two materials are assumed to have similar characteristics. The regulatory body is reviewing if the evaluation result based upon the assumption is reliable and conservative. Seven valves evaluated based on the assumption showed low margin (below 7 percent) and one valve showed a negative margin. All valves concerned are installed in the emergency core cooling system. They are 10-inch gate valves with SMB-2 actuators.

Development of Motor Control Center (MCC) test methodology and equipment

Diagnostic test is recognized to be essential in MOV design basis and periodic verification work. The utility is now currently employing foreign-made diagnostic equipment and analysis methods. The utility is considering using the equipment and analysis program attached to the equipment in the MOV periodic verification phase. However, a Korean university and a research institute recently developed a new diagnostic equipment and analysis software program. The developer did a detailed comparison between the newly developed method and commercially operating methods. They want to hear the regulatory position on the developed methodology.

Others

A few more concerns have been identified since the Korean MOV program was started. Design basis determination and its DP calculation issue in some safety systems for CANDU-600 plants is one of them. The design basis verification requirement for new valves, which are qualified according to the ASME qualification requirement, is another concern. Weak link analysis for the universal joint and shaft, which connect actuator and valve components in the Framatome design plants, became one of them.

Conclusions

The Korean MOV design basis verification requirement was issued in 1997 and the Korean utility has been performing its implementation program since 1999. About 43 percent out of the approximately 1,800 safety-related MOVs in 18 reactors has been evaluated as of the end of 2001. Most valves showed a sufficient operational margin but a few valves showed a negative margin. Not only the utility but also the regulatory body is paying special attention to the valves with negative and low operational margin. Regulatory activities in MOV regulation consist of review for design basis analysis reports, on-site inspection, and review of evaluation result reports. A regulatory inspection report is prepared on an every refueling basis incorporating every regulatory activity. The Government Notice for inservice testing of pumps and valves is being revised, mainly to include the MOV periodic verification requirement, and AOV design basis and periodic verification requirement, and is to be issued in June 2002.

The utility is trying to use alternative tools in calculating differential pressure to minimize over-conservatism and to improve the operational margin for negative margin valves. The Non-Limitorque actuator capability issue was drawn because actuator suppliers except Limitorque have not provided sufficient data. Hopkinsons actuator has shown a structural weakness in the worm gear teeth. Several thermal overload relays in CE-1000 and CANDU-600 reactors have been found to be undersized. The PPM applicability issue for Deloro 40 on Deloro 50 disk and seat material in CANDU reactors was identified. A university and an institute jointly developed a new diagnostic and analysis method at MCC.

References

U.S. NRC Generic Letter 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance."

U.S. NRC Generic Letter 95-07, "Pressure Locking and Thermal Binding of Safety-Related Power-Operated Gate Valves."

The first KINS MOV workshop (KINS/AR-654, 1999. 2.1).

The second KINS MOV workshop (KINS/PR-003, 2000. 2.1).

The third KINS MOV workshop (KINS/PR-016, 2001. 3.21).

Technical procedure for power-operated valve design basis capability verification (I), (II), (III) (Korea Hydraulic and Nuclear Corporation).

Summary report on Motor-Operated Valve NRC Information Notice (KINS/AR-727, 2000. 3).

Table 1. Failure Data for two WH-900 reactors (1987. 1 - 1990. 5)

Failure Modes	**# of failures reported**
Failure to Open/Close	43
Stem or Internal Leak	31
Trouble/Disable Light	31
Torque Switch Failure	18
Limit Switch Failure	9
Motor Failure	8
Double Indication Position	5
Others	91
Total	227

Table 2. Total number of MOVs to be evaluated and evaluation status as of the end of year 2001			
Reactor Design	**# of valves to be evaluated**	**# of valves evaluated**	
		Year 2000	**Year 2001**
WH-600	135	31	43
WH-900	444	64	57
CE-1000	772	177	200
CANDU-600	273	60	52
Framatome-900	176	17	73
Total	1,800	349	425

Table 3. Valves with negative operational margin (2000-2001)		
NSSS Design	**# of valve with negative margin**	**Evaluation method**
WH-900	6	Dynamic Test
	6	PPM
CE-1000	2	Dynamic Test
CANDU-600	1	Dynamic Group
	2	PPM
Framatome-900	2	Dynamic Test
	2	Dynamic Group
Total	21	

Quarter-Turn Motor Operated Valve Motor Power Monitor Testing at Catawba Nuclear Station —A Picture is Worth a Thousand Words

C. A. Helmers
Duke Energy

Abstract

In the early 1990s, Duke Power became aware of the potential benefits which motor control center (MCC) based diagnostic testing could offer and has since made extensive use of the Motor Power Monitor (MPM) system during motor operated valve (MOV) testing activities. Duke Power began collecting "baseline" motor power data on most NRC Generic Letter (GL) 89-10 MOVs in anticipation of quantitative MPM analysis techniques being developed that could eventually lead to a reduction in the need for traditional "at-the-valve" diagnostic testing. As baseline motor power data were collected and evaluated, the strong qualitative value of MPM data was soon realized.

This paper specifically looks at two Catawba Nuclear Station (CNS) case studies where the qualitative features of MPM testing proved to be quite valuable in helping to ensure that MOV "health" and proper operation had been restored.

Introduction

Regulatory driven quarter-turn MOV diagnostic testing at CNS is performed using an actuator torque test bench. The torque test bench allows the MOV actuator torque switches to be diagnostically set according to predetermined acceptance criteria. Additionally, actuator capability is then verified by demonstrating that the torque switches do indeed "trip" at reduced voltage conditions. The actuator is then installed back on the valve, the limit switches are set and an MPM "retest" is conducted to conclude the work.

As quarter-turn MOV MPM retests are typically run under static conditions, the resulting traces are reviewed for "smooth and typical" running loads and the seating/ unseating events. However, no "analysis" of the MPM data is performed as CNS believes further refinements in the quantitative approaches are needed. As such, and in this era of the increasing resource constraints, the idea of eliminating the quarter-turn MOV MPM retest was at one time briefly entertained.

However, recent experiences have proven that the MPM retest traces, as they are retrievable, can be an invaluable tool for ensuring MOV "operability" and functionality have been restored when operating problems are encountered and additional differential pressure testing appears to be impractical. This paper presents two such case studies. In the first case, when a large NRC GL 89-10 butterfly MOV was returned to service after extensive repairs, MPM retests were performed to help verify that MOV health had been restored. In the second case, another NRC GL 89-10 butterfly MOV was returned to service after being declared "inoperable" due to the close direction limit switch being out of adjustment, thereby resulting in excessive seat leakage. Again, MPM testing was used to ensure proper disc seating had been restored, thereby allowing the MOV to be returned to service without performing a system flow-balance retest and incurring the associated "unavailability" time.

Case Study 1

Background

CNS MOV 1RN003A is a 48" motor operated butterfly valve. The MOV isolates the Nuclear Service Water System pump pit "A" from the site's "assured" standby heat-sink pond. The MOV is normally closed and has a design basis function to open. The MOV is an NRC GL 89-10 "program" application and in probabilistic risk assessment and NRC GL 96-05 terminology, has been categorized as "medium-risk."

1RN003A is somewhat of an unusual MOV application (see Figure 1). The valve itself sits under approximately 30 feet of water. The actuator sits at ground level, mounted to a floor pedestal. A 60-foot reach-rod connects the valve with the actuator/floor stand combination.

Event

In July of 2000, CNS MOV 1RN003A was declared inoperable after a night of troubleshooting. Control room indication problems started the initial investigation. After several manual handwheel and MPM data acquisition strokes, Engineering concluded that a significant running load had materialized and that catastrophic failure would be likely if the MOV was stroked open against design basis conditions.

Figure 2 shows an opening stroke MPM data trace taken less than one year earlier. The MPM data trace shows expected butterfly MOV performance, with a typical unseating event and a smooth running load region. Figure 3 shows an opening stroke MPM data trace taken during the night of troubleshooting in July of 2000. When compared with Figure 2, it was apparent a substantial loading problem had developed somewhere within the MOV 1RN003A assembly. Consequently, MOV 1RN003A was declared inoperable, "tagged" in its design basis position (open), and subsequently, a long difficult troubleshooting process began.

Among other things, extensive discussions were held with supporting vendors, angle of twist observations from the initial night of troubleshooting were analyzed, a dive team was brought in to perform underwater inspections, the actuator was replaced, and the original actuator was tested and refurbished in hopes of finding a problem. Also, drawings were reviewed countless times, MPM traces were "broken-down" numerous times and bearings in the actuator floor stand were inspected with a boroscope. None of this yielded any results with respect to root cause determination.

Ultimately, the dive team was brought back on-site to help remove valve 1RN003A from the system so that it could be sent back to the vendor's manufacturing facility for disassembly, inspection and root cause analysis. At the factory, it was observed that the upper valve shaft (split shaft valve) had seized to the O-ring gland (packless valve with a double O-ring shaft seal design).

Detailed design drawings of the O-ring gland, the upper valve shaft and the shaft bushing were immediately reviewed by vendor and CNS personnel. It was discovered that the diametric clearance range for the O-ring gland/upper valve shaft interface was tighter than the diametric clearance range for the upper valve shaft/bushing interface. Consequently, the O-ring gland itself had been inadvertently designed to act as the upper valve shaft bushing. Thus, contact forces between the upper valve shaft and the relatively small O-ring gland resulted in significant contact stresses, then galling and eventually seizure of the two valve parts.

Valve 1RN003A was rebuilt at the vendor's manufacturing facility. Specifically, the upper valve shaft was replaced, the O-ring gland was cleaned up and its internal bore diameter was appropriately increased so that future contact with the upper valve shaft would not occur. The valve arrived back on-site at CNS and was returned to service in October of 2000.

The "Return to Service"

Valve 1RN003A was reinstalled into the system during a refueling outage in which both the Nuclear Service Water System pump pits were being drained for other maintenance reasons. This made the job much simpler as a dive team was not needed to perform underwater work. Subsequently, all required retests were successfully performed. These tests included an actuator "functional" stroke for proper control board light indications, an "inservice" (IST) retest to verify stroke time requirements were met and an MPM retest. Figure 4 shows the results of the MPM retest opening stroke. The MPM data trace demonstrated that, once again, MOV 1RN003A was performing as expected. In addition, the vendor had performed static baseline stem torque testing on the valve at their facility before returning it to CNS. The results indicated that the valve had been properly rebuilt, as the running loads had returned to normal levels. However, doubts still remained as to whether the health of the entire 1RN003A MOV assembly had been returned.

A design basis differential pressure test was suggested as one alternative method that would eliminate any lingering questions concerning the health of MOV 1RN003A. However, before committing to the resource intensive and demanding preparations of performing such a test on an emergent basis during a refueling outage, Engineering was asked if there was any other way to demonstrate that CNS could have confidence that all problems associated with the MOV 1RN003A event had been eliminated. At that point, Engineering turned to MPM test data, as that was the only test data which measured the load performance of the MOV as an entire assembly and for which there was also a historical record.

The individual MPM traces from October 1999, July 2000 and October 2000 were overlaid so that they could be viewed concurrently (see Figure 5). The three trace overlay provided an excellent visual representation of MOV 1RN003A performance under static system conditions at the three points over the previous year. It was this "picture" that eventually provided assurance that the running load problem had been corrected and that the health of MOV 1RN003A had been restored to pre-event levels. The overlay graphic was received very well by management in the event close-out presentation. The graphic was also the clinching piece of test data that finally satisfied the remaining concerns of one of the resident NRC inspectors during a quite thorough audit of the entire event and the actions taken to ensure MOV 1RN003A was fully "operable" before being returned to service. It is believed that it was the visual impact of the MPM overlay graphic which ultimately provided plant management and regulatory personnel with confidence that all concerns associated with the subject event had, indeed, been successfully resolved. The MPM overlay graphic essentially summarized the return to service testing results in a quick, simple and easy to understand format without a lot of "numerical" computations that may have only confused the issue with the non-MOV personnel.

Summary

CNS MOV 1RN003A was rebuilt and reinstalled after the valve's upper shaft and O-ring gland had seized due to a design deficiency. All required retests for returning the MOV to service were successfully completed, but plant management desired further assurance that all associated problems had been addressed. Design basis differential pressure testing was suggested as one alternative for accomplishing this. However, before committing to the resource intensive and demanding preparations associated with emergent MOV differential pressure testing during a refueling outage, Engineering was asked to investigate other avenues that could possibly be used as a basis for demonstrating that the health of MOV 1RN003A had, indeed, been restored.

Ultimately, it was a three trace time history MPM overlay graphic and not any kind of quantitative data analysis which convinced plant management and NRC regulatory personnel that MOV 1RN003A was ready to be returned to service. Note: Appendix A is a copy of the posting CNS made to the INPO "newsgroup" board regarding the MOV 1RN003A event.

Case Study 2

Background

CNS MOV 1KC050A is a 20" motor operated butterfly valve. The MOV serves as the non-essential header isolation valve for the Unit 1 "A" train Component Cooling System. The MOV is normally open and has a design basis function to close. The MOV is an NRC GL 89-10 "program" application and, in probabilistic risk assessment and NRC GL 96-05 terminology, has been categorized as "low-risk."

Event

In April of 2001, CNS MOV 1KC050A was stroked to the closed position to support testing activities associated with several other MOVs. Operations personnel in the immediate area of MOV 1KC050A noted audible leakage passing through the valve even though it had reached the fully closed position. The MOV was put in manual mode, and the handwheel was used to seat the disc to eliminate the leakage.

Two months earlier, in February of 2001, MOV 1KC050A had undergone actuator limit switch adjustments, "functional" testing, an IST retest and an MPM retest following actuator torque bench testing. The last Unit 1 "A" train Component Cooling System flow-balance test had been performed during the previous Unit 1 refueling outage approximately four months earlier in the fall of 2000. The leakage through MOV 1KC050A and into the non-essential header could not be immediately quantified; thus leaving questions as to whether the flow-balance was still valid. Consequently, MOV 1KC050A was declared "inoperable," logged into the Technical Specification Action Item List and was left in the manually adjusted closed position. The problem was sent to

Engineering for analysis and formulation of a satisfactory recovery plan.

After a review of surge tank level computer point changes, the Component Cooling System engineer determined that the leakage through MOV 1KC050A during the event was approximately 82 gpm. While this leak had a significant impact on the flow-balance dynamics of the system, the net-positive-suction-head available to the "A" train pumps still remained above the minimum limit required for operability. However, had this scenario taken place on CNS Unit 2, which has a number of elevation differences when compared against Unit 1, the minimum net-positive-suction-head limit for the pumps would have been violated, and this would have become an NRC "reportable" event.

Event Analysis and Corrective Actions

The problem simply boiled down to a less than adequate close direction limit switch adjustment made during the February 2001 actuator testing activities. Closing of MOV 1KC050A is controlled by actuator limit switch (versus actuator torque switch) and, in this case, the close direction limit switch had been set to cease operation before the valve disc was appropriately seated. It should be noted that it is not known how well MOV 1KC050A "seated" before the February 2001 actuator limit switch adjustments, only that the valve did not seat as well following the work.

Figure 6 shows the closing stroke data trace from the last MPM retest for MOV 1KC050A, which was performed in March 1996. While not largely significant, a small power increase can be observed during the seating event, indicating some amount of disc movement into the valve seat. When the closing stroke data trace from the February 2001 MPM retest is viewed (see Figure 7), it is apparent that there is little, if any, power increase associated with the seating event at the end of the stroke. Again, it should be remembered that the important thing is not how well the disc was seating at each of the two dates, but that the disc was not seating as well following the February 2001 actuator testing activities.

In May of 2001, Engineering and Maintenance teamed together in an attempt to make corrective adjustments to the close direction limit switch on MOV 1KC050A so that the valve could be returned to operable status. During this work, extensive use was made of the MPM data acquisition system to ensure optimum disc seating was obtained. A number of follow-up limit switch adjustments were made after viewing respective MPM data trace closing strokes at the MCC. Once it was felt that the close direction limit switch was set about as well as could be expected, the maintenance work was closed out and the final MPM retest stroke was taken. The results of the MPM retest stroke in the closing direction are shown in Figure 8. As can be observed, a significant power increase associated with disc seating was now evident. Consequently, it was felt that MOV 1KC050A was now seating better than it had been following the March 1996 actuator testing activities. Thus, MOV 1KC050A was declared operable and logged out of the Technical Specification Action Item List.

Event Close-Out

The CNS senior resident NRC inspector was fully aware of the implications surrounding this event with respect to the previously discussed system differences between Units 1 and 2. Thus, the senior inspector subsequently requested a meeting with Engineering to discuss the adequacy of the site's MOV actuator limit switch adjustment activities before another such incident did result in an NRC "reportable" event.

Of particular concern to CNS Engineering was a statement by the senior inspector that CNS may have to consider a system flow-balance retest following all such MOV limit switch adjustment activities, or at least in plant applications where an improper functioning MOV could put a system flow-balance at significant risk of invalidation. Certainly, CNS did not want such a suggestion to become reality unless absolutely necessary, as this would increase both the strain on site testing resources and the "unavailability" numbers of certain systems. Furthermore, Engineering felt such a change to the site's MOV retest approach was not warranted, as this was an isolated event with no trend of similar type failures.

Consequently, CNS Engineering took a number of steps to "tighten" the site's approach toward the setting of the close direction MOV actuator limit switch on butterfly valves such as 1KC050A. Among those steps were the following items. First, a procedural control was put in place that required a field review and sign-off of an MPM data trace showing clear evidence of a seat load "ramp" in the closing direction. Second, a numerical rule-of-thumb was also added to the procedure to "expect" an approximate 40%-60% total "real power" increase at close direction limit switch actuation during the seating event. This rule-of-thumb is not a requirement, it is only for guidance purposes, and it was estimated following a review of MPM data traces from several similar MOVs which showed satisfactory seating events. Finally, ten CNS butterfly MOVs which are controlled by the actuator limit switch in the closing direction and which are in plant applications where an improperly set close direction limit switch could put a system flow-balance in significant jeopardy were identified. A procedural caution statement was added to ensure the risk significance of the close direction limit switch setting on these ten MOVs was known to the end-user.

At the meeting with the senior NRC resident inspector, all of the procedural changes which were intended to strengthen the site's MOV actuator limit switch setting activities, along with the MPM overlay graphic which is shown in Figure 9, were reviewed thoroughly. By the conclusion of the meeting, the senior inspector was completely satisfied that MOV 1KC050A was fully operable, that CNS had taken the appropriate procedural corrective actions and, most importantly, that CNS had the ability and tools to ensure such an event did not happen again.

Summary

In April of 2001, CNS MOV 1KC050A was stroked to the closed position to support testing activities associated with several other MOVs. Operations personnel in the immediate area of MOV 1KC050A noted audible leakage passing through the valve. Subsequently, MOV 1KC050A was declared inoperable and was left in the manually adjusted closed position.

MOV 1KC050A had undergone actuator limit switch adjustments just two months previous. The CNS senior resident NRC inspector requested a meeting with Engineering to discuss the adequacy of the site's MOV actuator limit switch adjustment activities and potential ideas for preventing another such event.

Consequently, CNS Engineering took a number of steps to "tighten" the site's approach toward the setting of the close direction MOV actuator limit switch on butterfly valves such as 1KC050A, including stricter use of the MPM system. Of primary importance was the site's intention to begin

procedurally using the MPM system as an additional maintenance tool, with actual use requirements during actuator limit switch adjustment activities, and not just using the system in a formal "after-the-fact" retest fashion.

In the end, the senior inspector agreed with the site's position that this event was, at this time, isolated in nature and did not warrant the future addition of mandatory system flow-balance retests following similar actuator limit switch adjustment activities. As in Case Study 1, it is again believed that the MPM overlay graphic made a strong visual impact. Consequently, regulatory personnel felt confident that CNS could reliably and predictably set the close direction limit switch on MOVs such as 1KC050A.

Conclusion

While in the early stages of development, Duke Power saw the potential benefits that motor control center based MOV testing might eventually offer and quickly began collecting "baseline" MPM data during routine periodic diagnostic test activities. After several years, though, for reasons discussed previously in this text, CNS briefly considered eliminating the quarter-turn MOV MPM retest.

However, this paper has presented two case studies where the visual aspects, the overlapping features, and the retrievable nature of MPM test traces provided the necessary information that ultimately assured both station management personnel and NRC regulatory personnel that MOV "health" and proper operation had been restored before the return to service. Thus, while the industry continues to work on refining quantitative analysis methods, MPM testing has proven its worth and has carved itself an important niche as a valuable qualitative MOV retest and maintenance tool at Catawba Nuclear Station.

CASE STUDY 1
MOV 1RN003A
ASSEMBLY DRAWING

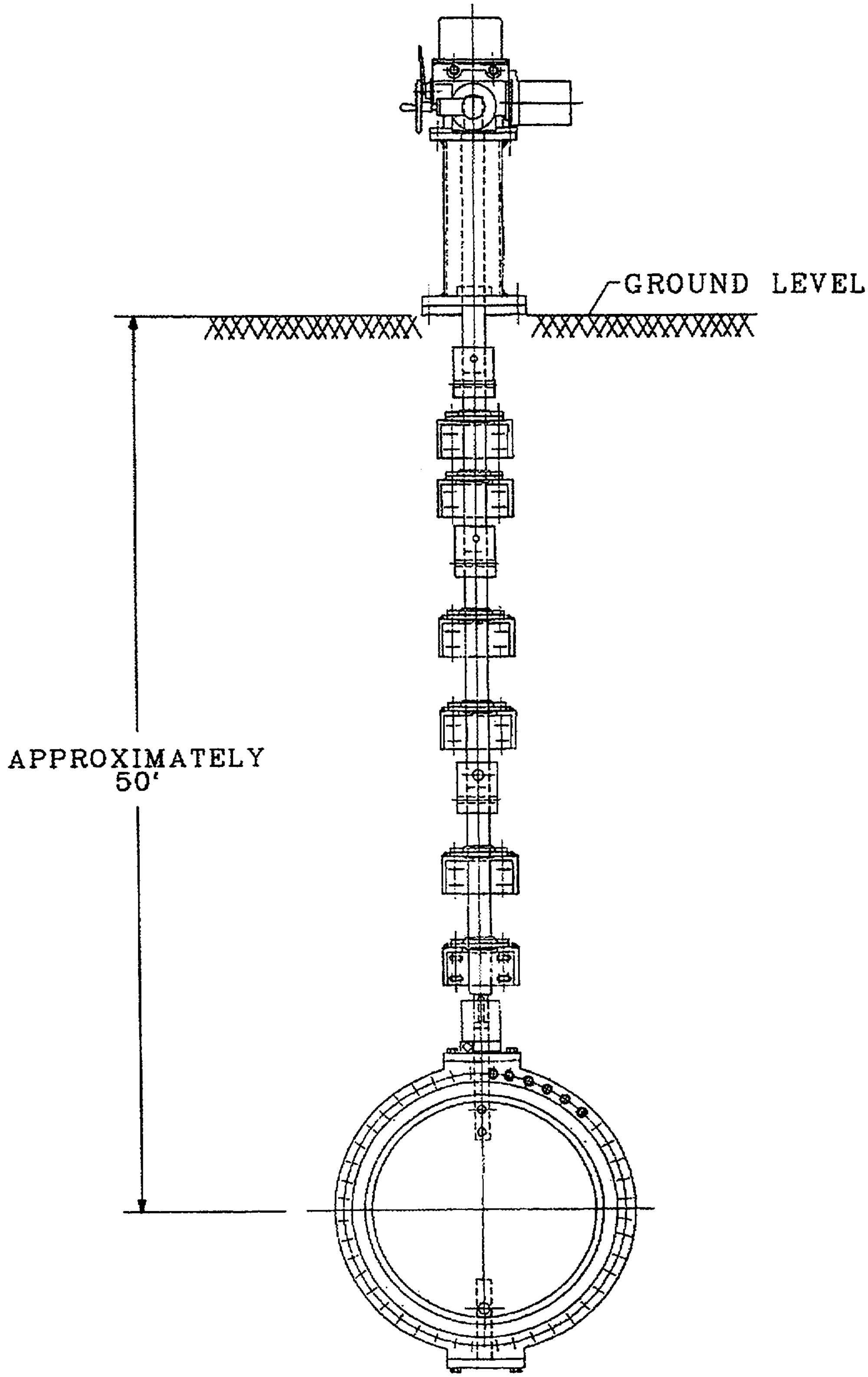


FIGURE 1

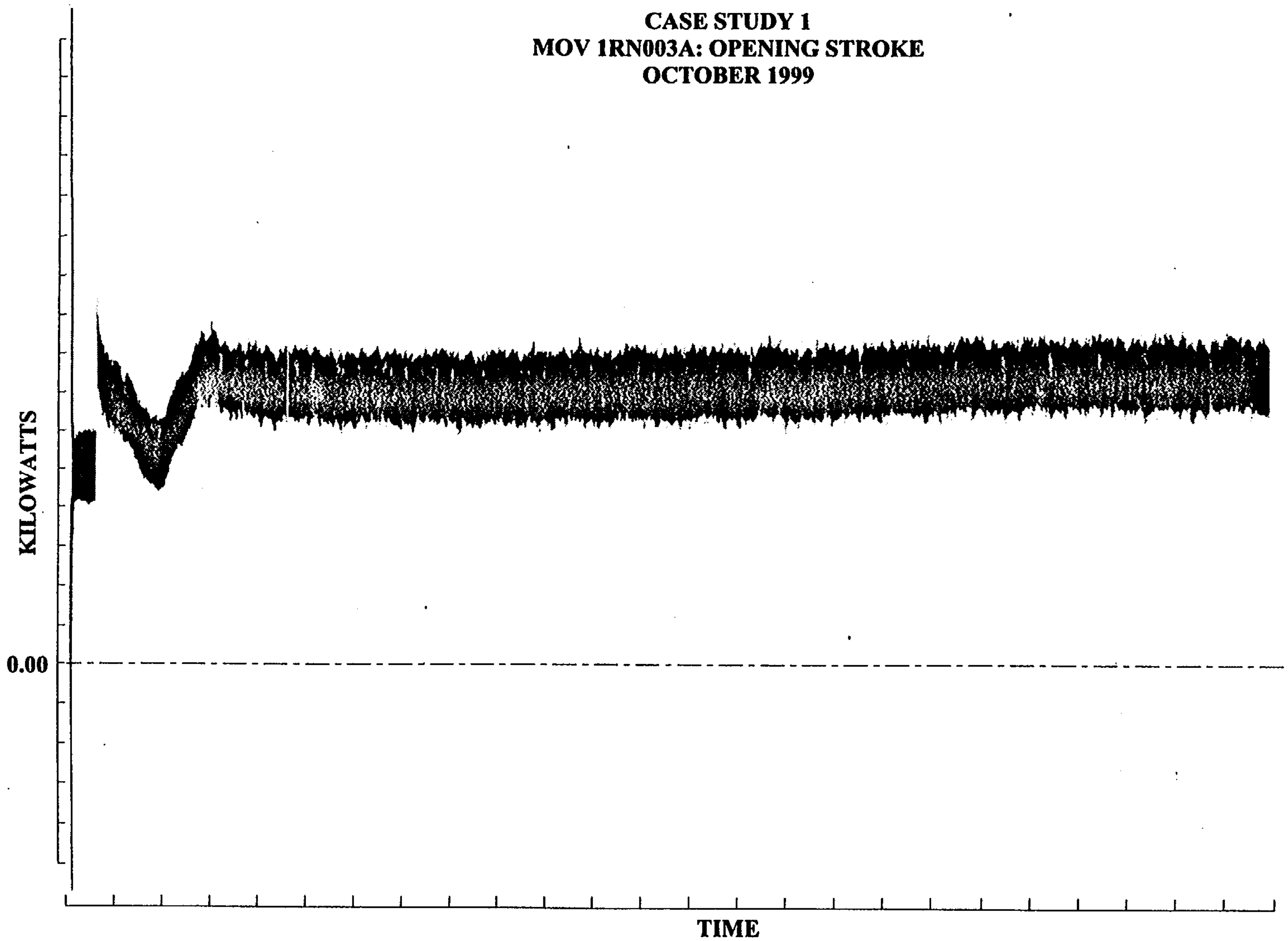

CASE STUDY 1
MOV 1RN003A: OPENING STROKE
OCTOBER 1999
KILOWATTS
0.00
TIME

FIGURE 2

CASE STUDY 1
MOV 1RN003A: OPENING STROKE
JULY 2000

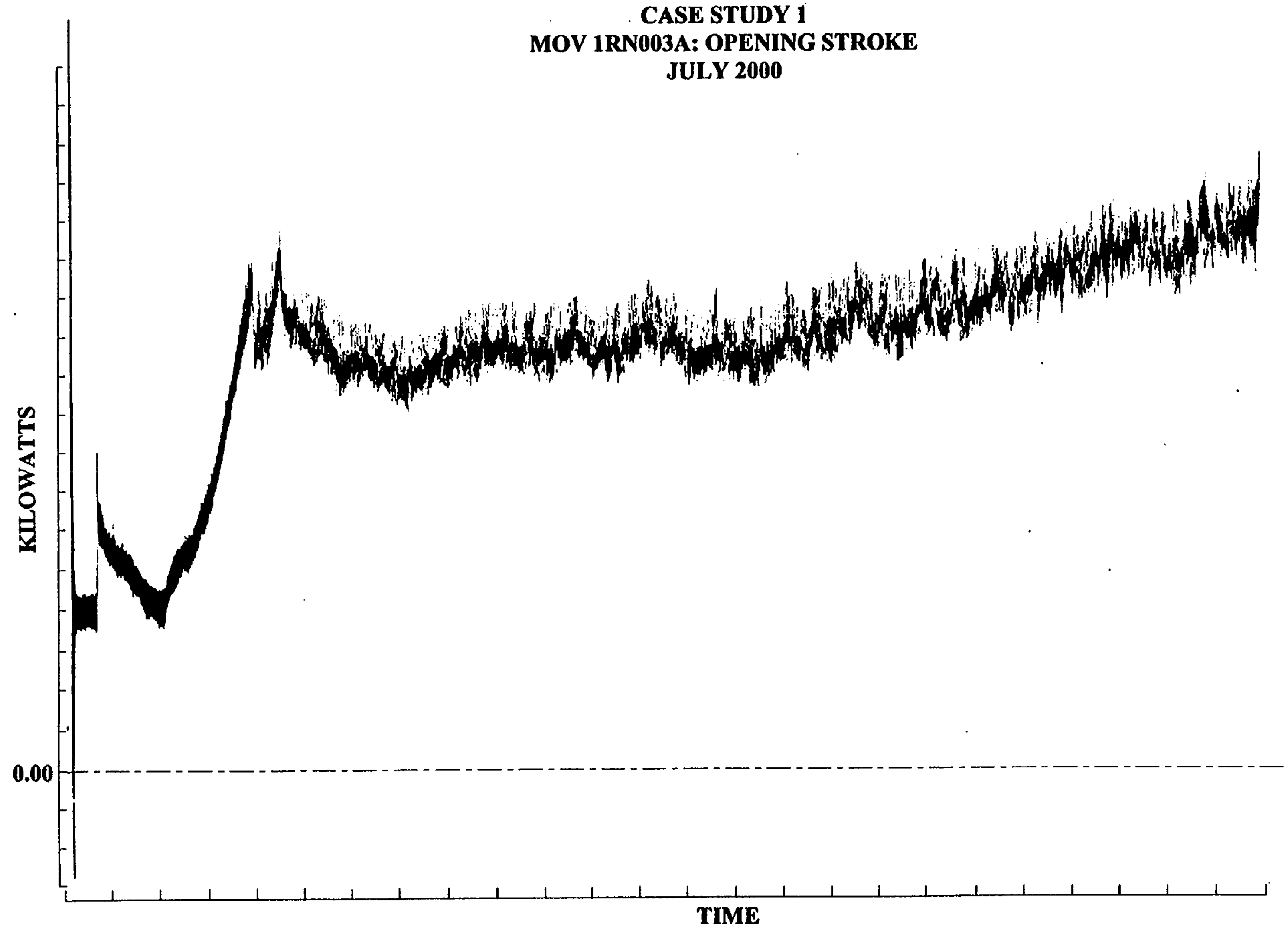


FIGURE 3

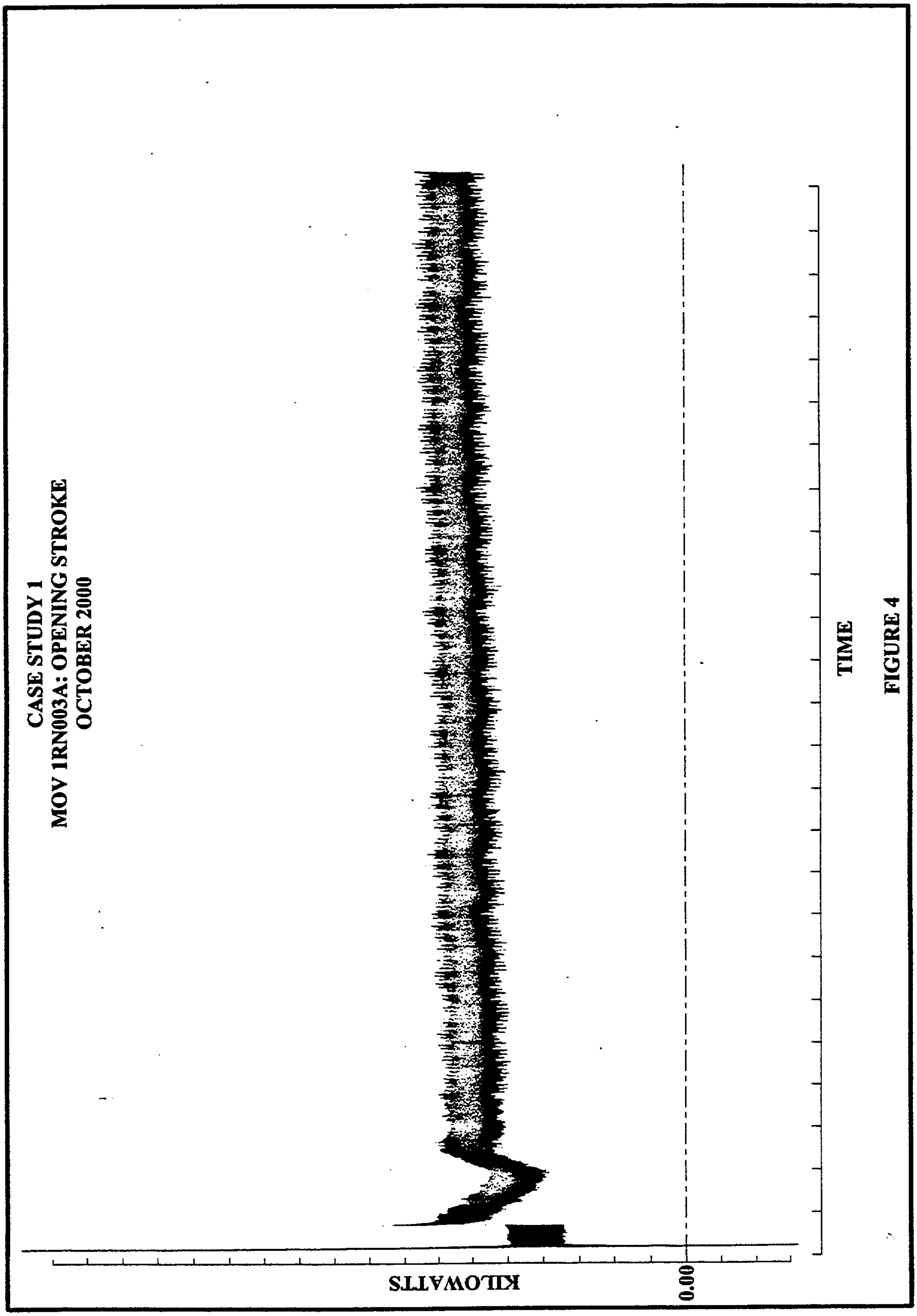

CASE STUDY 1
MOV 1RN003A: OPENING STROKE
OCTOBER 2000
KILOWATTS
0.00
TIME

FIGURE 4

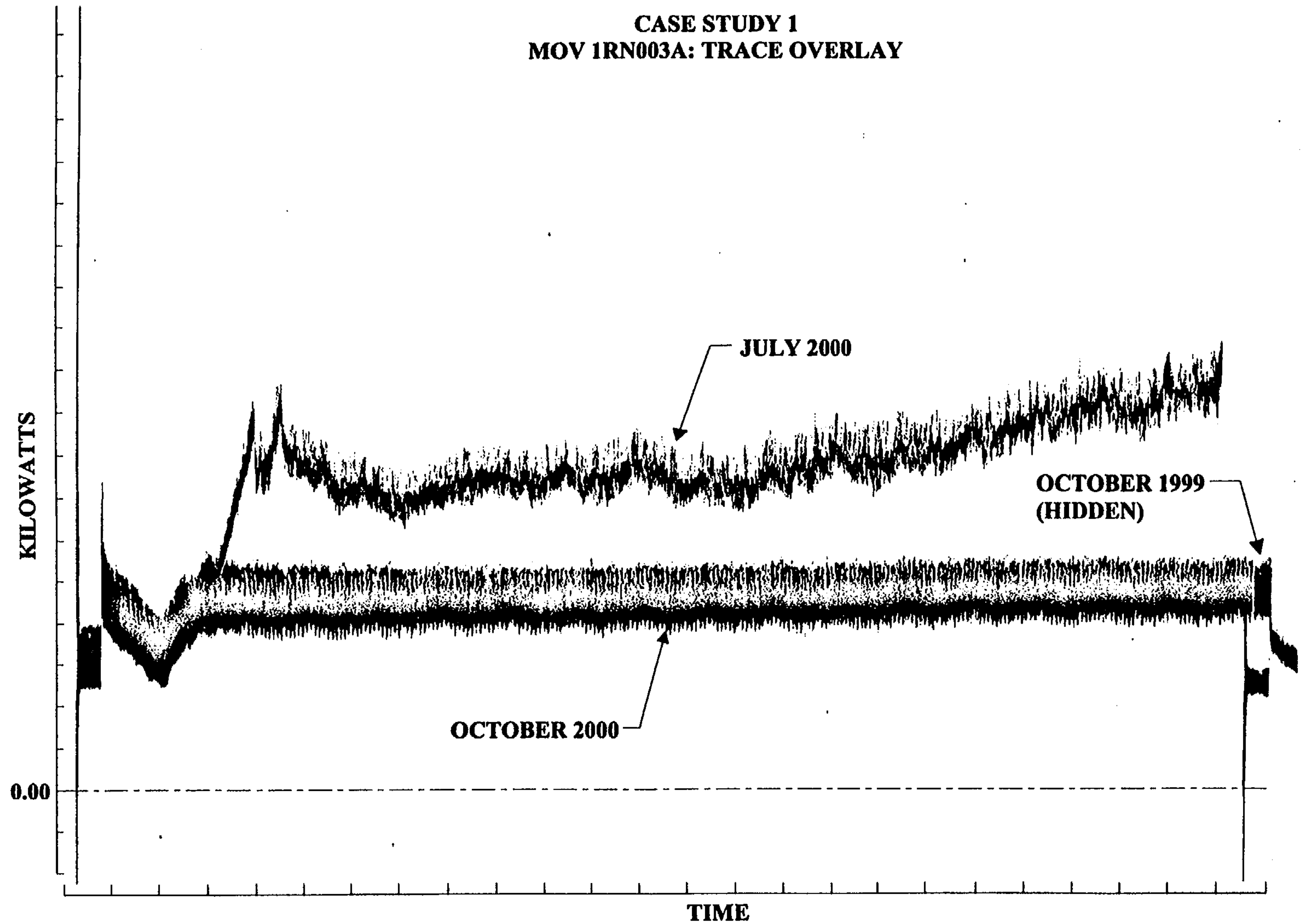

CASE STUDY 1
MOV 1RN003A: TRACE OVERLAY
KILOWATTS
0.00
JULY 2000
OCTOBER 1999
(HIDDEN)
OCTOBER 2000
TIME

FIGURE 5

CASE STUDY 2
MOV 1KC050A: CLOSING STROKE
MARCH 1996

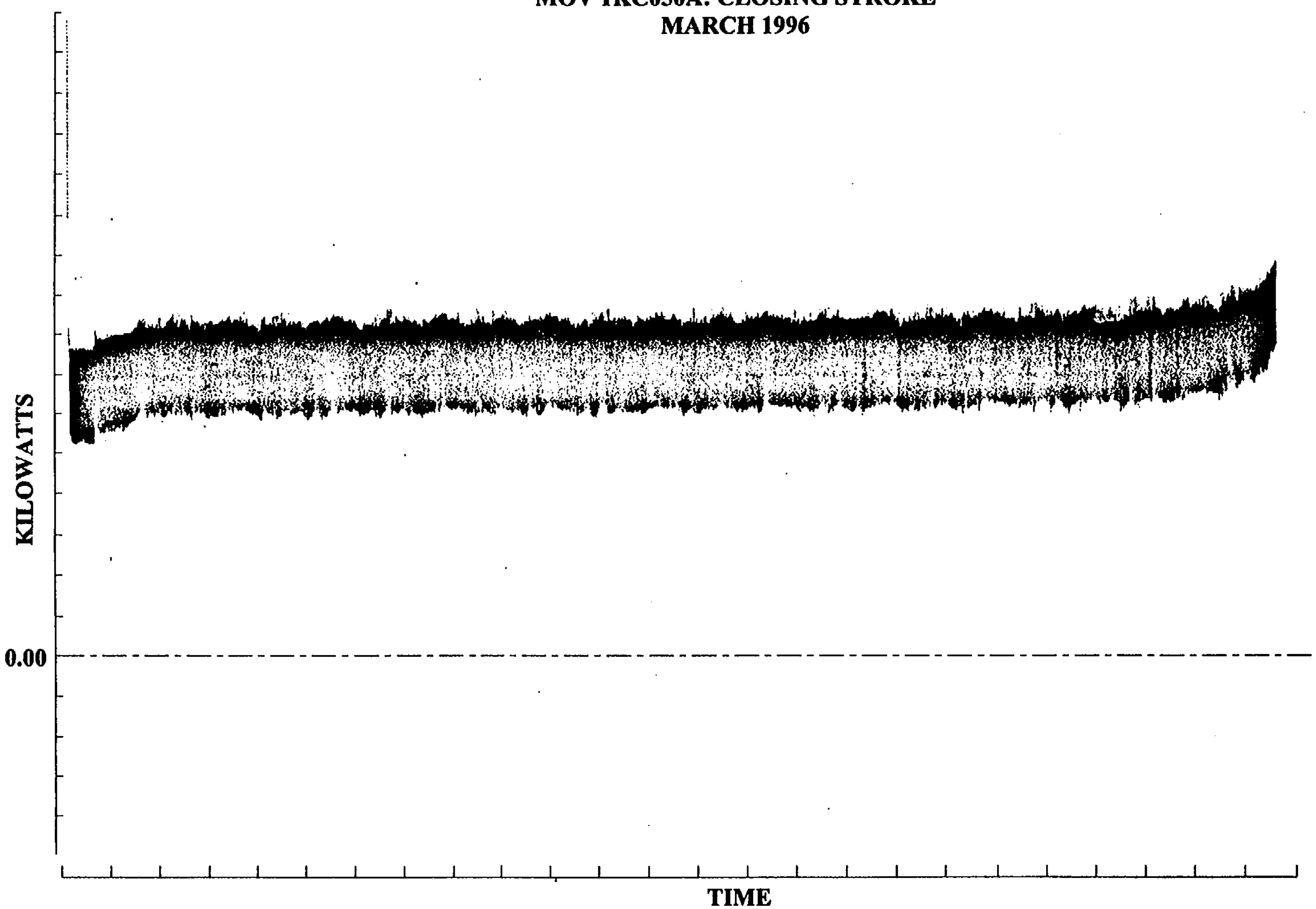


FIGURE 6

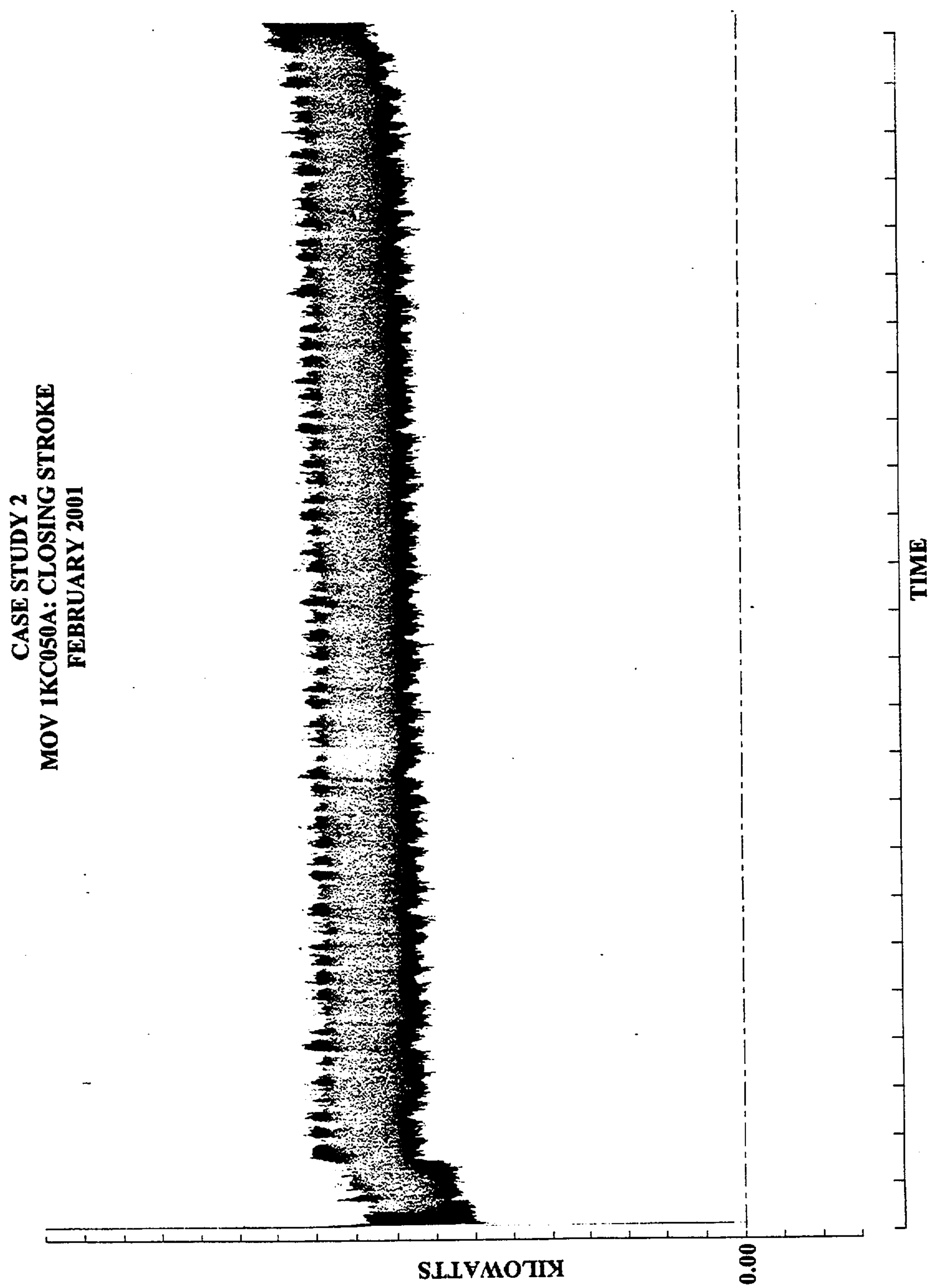

CASE STUDY 2
MOV 1KC050A: CLOSING STROKE
FEBRUARY 2001
KILOWATTS
0.00
TIME

FIGURE 7

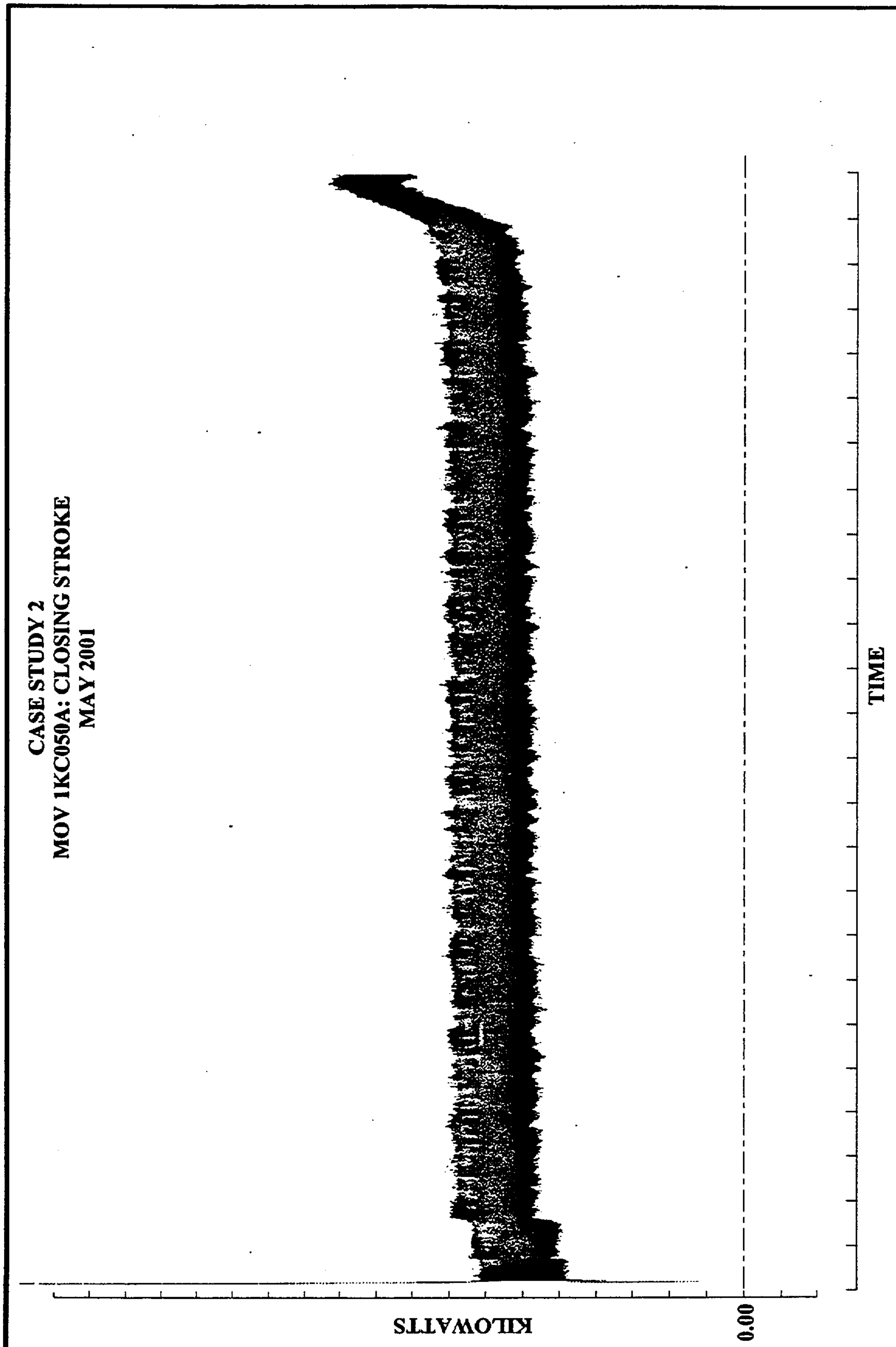

CASE STUDY 2
MOV 1KC050A: CLOSING STROKE
MAY 2001
KILOWATTS
0.00
TIME

FIGURE 8

CASE STUDY 2
MOV 1KC050A: TRACE OVERLAY

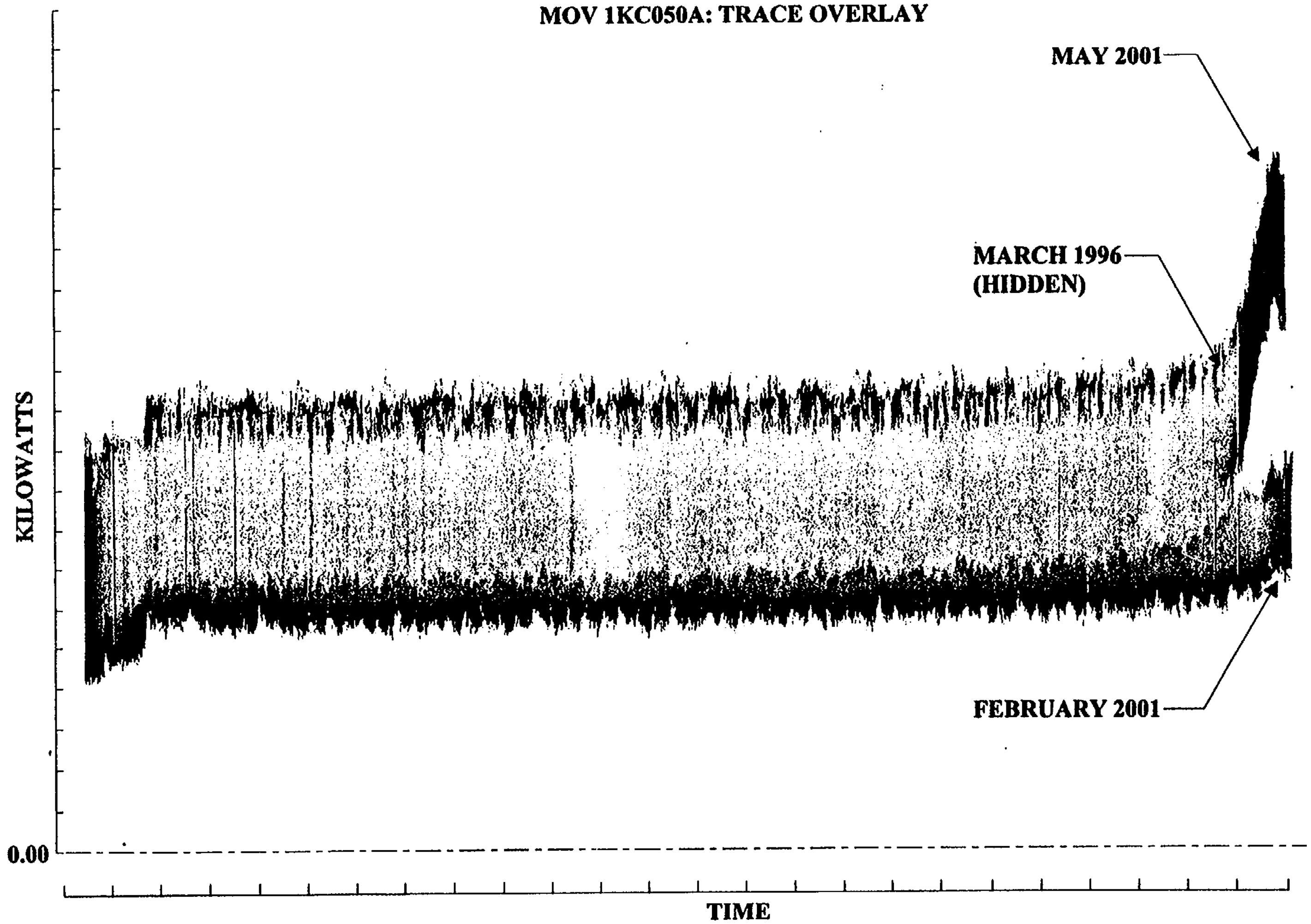


FIGURE 9

Appendix A
Posting to INPO Newsgroup (wnn.equipmentperformance.valves)

Plant:	Catawba Nuclear Station
System:	Nuclear Service Water
Valve:	1RN003A; 48" Motor Operated Anchor/Darling (Contromatics) Butterfly Valve, 150 lb. Pressure Class, with a Limitorque SMB-00 operator and an H3BC gearbox.
Contact:	Curt Helmers (803) 831-4160; email: cahelmer@duke-energy.com

Brief Summary

In July of 2000 the upper valve shaft seized to the O-ring gland (packless valve with a double O-ring shaft seal design). Upon inspection, it was determined that the internal diameter of the O-ring gland was smaller than the internal diameter of the upper valve shaft bushing. Consequently, the O-ring gland acted as the upper valve shaft bushing and eventually seized to the upper valve shaft. The manufacturer (now Flowserve) reviewed the design drawings and determined that the O-ring gland was improperly designed. As such, it was determined that this design problem could potentially affect similar valves at other nuclear plants.

Discovery

1RN003A isolates the Standby Nuclear Service Water Pond (SNSWP) from the A-train Nuclear Service Water System (RN) for both Catawba Nuclear Station (CNS) units. This valve was opened to facilitate routine RN system maintenance. The valve failed to reach the full open position and stopped at an intermediate position (between 85°–95° open, after the "open torque switch bypass" logic had dropped out of the control circuit).

Detailed Problem Description and Root Cause

Motor operated valve (MOV) 1RN003A is the SNSWP supply isolation for the RN system pump pit "A." The valve itself is located under approximately 30 feet of water. The motor operator sits on a floor stand at ground level in the RN pump house. An approximately sixty foot extension shaft/reach rod attaches the valve to the motor operator/floor stand. Consequently, troubleshooting was extremely difficult.

A temporary station modification was implemented to remove valve 1RN003A from the system. 1RN003A was removed and shipped to Flowserve Corporation, Williamsport, Pennsylvania. It should be noted that this valve is a "Contromatics" design that was produced at the Contromatics facility in Laconia, New Hampshire under the Anchor/Darling name in 1994. At the time, Anchor/Darling had just "acquired" Contromatics and was in the process of assimilating the company's operations into A/D Valve Company. Fairly soon after the CNS valves were shipped (the total "lot" size was 6 valves), the Laconia, New Hampshire facility was closed, and all of its valve operations were moved to the Anchor/Darling headquarters in Williamsport, Pennsylvania.

A1/3

Upon arrival, the valve was disassembled and inspected. Disassembly revealed that the O-ring gland (similar to a packing gland) had seized to the upper valve shaft due to excessive galling (the O-ring gland should turn freely on the shaft). Ultimately, the O-ring gland had to be removed from the upper valve shaft by means of a hydraulic press.

Detailed design drawings of the O-ring gland, the upper valve shaft and the shaft bushing were immediately reviewed by Flowserve and Duke Energy personnel. It was discovered that the diametric clearance range for the O-ring gland/upper valve shaft interface was tighter than the diametric clearance range for the upper valve shaft/bushing interface.

Flowserve feels that the definitive cause of the failure was the unusually tight diametric clearance range between the O-ring gland internal diameter and the upper valve shaft. CNS Engineering concurs. The tightness of the O-ring gland internal diameter on the upper valve shaft eliminated the natural "float" of the upper valve shaft, thus preventing it from moving against the bushing wall where "bearing" type loads due to differential pressure are intended to be absorbed and spread out over a relatively large and uniform area. In effect, the O-ring gland itself was inadvertently designed to act as the upper valve shaft bushing.

With the diametric clearance range between the O-ring gland and the upper valve shaft being as tight as it was, significant contact stresses resulted when differential pressure forced the upper valve shaft surface against the relatively small area of the O-ring gland internal diameter. These high contact forces led to galling between the two surfaces as the valve was stroked. Eventually the two surfaces seized together and rendered the valve "inoperable."

Resolution

Valve 1RN003A was rebuilt at the Flowserve facility in Williamsport, Pennsylvania (the thrust spacers, the upper valve shaft, the seat ring, the disc pins and both shaft bushings were all replaced). Specifically, the O-ring gland was cleaned up and its internal bore diameter was increased by approximately 0.025", which Flowserve felt was adequate to prevent any future contact with the upper valve shaft. However, the O-ring gland itself would not fit back into its intended valve body bore. Consequently, the outer bore of the O-ring gland was machined down to the low end of its original tolerance range and it subsequently fit back into place satisfactorily. Similar O-ring gland repairs were made to the five remaining identical CNS valves, none of which showed significant signs of similar galling. Valve 1RN003A arrived back on-site at CNS, was reinstalled into the RN system and was returned to service in October of 2000.

Final Disposition

Flowserve has participated in the development of this notification and has reviewed their butterfly valve designs for potential generic applicability issues. This review indicated that the CNS valves were the only ones affected by the design problem as described herein. Thus, Flowserve feels no further reporting is necessary.

Duke Energy is posting this item for awareness only. CNS has butterfly valves supplied by other manufacturers which utilize the same double O-ring shaft seal design, and no similar problems have been encountered. Awareness of this problem may help to ensure that future valves are correctly designed, regardless of the manufacturer.

A3/3

Implementation of Code Case OMN-1 for MOV Testing

Shawn Comstock
Wolf Creek Nuclear Operating Company

Abstract

ASME Code Case OMN-1, "Alternative Rules for Preservice and Inservice Testing of Certain Electric Motor-Operated Valve Assemblies in Light-Water Reactor Power Plants, OM Code-1995, Subsection ISTC," is an alternative to the Code requirements of stroke time and position indication testing for certain motor operated valves (MOVs). Wolf Creek implemented OMN-1 during the implementation of the Joint Owners' Group Program on MOV Periodic Verification—also known as the JOG program (JOG). The review of OMN-1 and JOG in 1996 determined that the two approaches were compatible and would yield increased economic benefits if implemented together. Wolf Creek was the first nuclear plant in the U.S. to obtain permission to use this Code Case as an alternative in 1997. The programmatic aspects of OMN-1 at Wolf Creek were completed in December 2000. Based upon the savings in terms of reduced outage critical path time, labor, equipment and radiological exposure, this approach is projected to save at least $2.8 million over the current licensed life of the plant.

This paper presents the transition approach and lessons learned during implementation at Wolf Creek, which may be of use to other licensees that wish to explore the use of ASME Code Case OMN-1. Also, the incorporation of ASME Code Case OMN-1 into the Inservice Testing (IST) code and its relationship to industry Generic Letter 96-05 MOV periodic verification programs will be discussed.

What is ASME Code Case OMN-1?

ASME Code Case OMN-1 is a methodology that can be used to diagnostically test an MOV under an IST Program. In order for an MOV to perform its safety function in an accident, it has to be able to reposition under the conditions that could potentially be present, which are documented in a plant's safety analysis.

For an MOV to successfully complete its mission, it has to generate enough torque or thrust to overcome the forces that are present under accident conditions. The minimum amount of torque or thrust that is needed for successful performance of an MOV is considered its design-basis condition. Simply put, ASME Code Case OMN-1 describes how to measure and assess an MOV's design-basis performance values.

Regulatory and Industry Influence

The Nuclear Regulatory Commission (NRC) issued a Bulletin and two Generic Letters that have influenced MOV testing programs in the United States. These are Bulletin 85-03, and Generic Letters 89-10 and 96-05. These regulatory documents recommended that licensees develop and implement a program to determine and periodically verify an MOV's design-basis capability. In order to address Generic Letter 96-05, the industry owner's groups formed a collaborative effort known as the "Joint Owners' Group Program on Motor Operated Valve Periodic Verification" (JOG). The JOG put together a document that describes how to develop and implement an MOV program that measures and uses an MOV's design-basis performance values.

The NRC endorsed both the JOG and ASME approach. Industry primarily chose to implement the JOG approach and a few plants chose to utilize their own MOV testing programs that were approved by the NRC. Not one plant in the U.S. chose to implement ASME Code Case OMN-1 by itself. Two licensees chose to use the JOG approach with ASME Code Case OMN-1, Wolf Creek and Comanche Peak. Comanche Peak was the first to use a risk-based approach for inservice testing and had to commit to the use of ASME Code Case OMN-1 for MOV testing under the terms of their regulatory exemption.
Wolf Creek chose to transition their MOV and IST programs to a combined JOG and OMN-1 approach to improve safety and reduce O&M costs. Since the implementation of the Wolf Creek and the Comanche Peak programs, at the time of this writing, other licensees that have reportedly chosen to transition to ASME Code Case OMN-1 include Palo Verde, San Onofre, Davis-Besse, and South Texas.

JOG and ASME Code Case OMN-1

Industry chose JOG over OMN-1 for a variety of reasons. Primarily, the JOG approach was just cheaper to implement as it described the approach and terminology that industry had already chosen to address the NRC generic letters. Also, a JOG program is outside of the IST program, which allows additional flexibility, less implementation effort and testing of fewer valves. Because no one had implemented ASME Code Case OMN-1 and MOV program engineers were not familiar with this approach, the fear of the unknown created an additional barrier.

While JOG and OMN-1 are not the same thing, they are compatible. Wolf Creek was fortunate in that the existing MOV program was very similar to JOG and OMN-1. A cost-benefit analysis revealed that implementing JOG/OMN-1 as a combined approach would result in a reduction in O&M, radiological exposure and an improved safety function capability. A comparison of the documents determined that the differences between the two documents could be reconciled. The two approaches provided some synergy in that the guidance from the two documents supplemented each other with additional detail.

Benefits of JOG/OMN-1

The implementation of OMN-1 with JOG resulted in a total of 1,666 fewer tests under the IST program over an 18-month fuel cycle. This has obvious benefits with respect to labor costs and "as low as reasonably achievable" (ALARA) radiation exposure efforts. Another benefit is a reduction in the normal wear rate for MOVs that were tested quarterly. Third, because an MOV is usually taken out of its initial safety position when testing is performed, its passive safety function capability is enhanced, thus improving its capability to protect the health and safety of the public in the event of an accident. Based upon the savings in terms of reduced outage critical path time, labor, equipment and radiological exposure dollars the combined approach is projected to save at least $2.8

million over the current licensed life of the plant.

Transition Approach

The first step in the implementation process was the identification of all existing procedures for MOV testing. The second phase was an analysis of compliance between the MOV procedures and JOG/ OMN-1. The areas where compliance was not achieved were identified, and revisions or new procedures were established to address compliance deficiencies. The third phase was the identification of all existing procedures (other than stroke time or position indication tests) that exercise MOVs and the development of an exercise verification procedure. The fourth phase was the development of program document revisions to describe the implementation of OMN-1. These changes comprised all of the aspects of implementation of JOG/OMN-1 and were completed in December 2000. The final phase was the development of procedure revisions to remove stroke time and position verification from surveillance procedures to reduce administrative overhead.

Implementation of the JOG approach with OMN-1 made the transition easier. The aspects of risk categorization, test frequency determination, appropriate mix or static/ dynamic testing and performance evaluation primarily came from the JOG document. The aspects of exercising, trending, documentation, and testing after maintenance came from ASME Code Case OMN-1.

Code Case OMN-1 indicates the use of torque for expression of acceptance criteria and trending. Paragraph 6.1.1 of OMN-1 allows flexibility in the expression of MOV torque margin. This paragraph was taken advantage of for rising stem MOVs where thrust is the primary concern for margin assessment and the parameter of friction co-efficient is conservatively assumed from industry guidance that has been accepted by the NRC. Conservatism is applied from both documents for all MOVs in that it is assumed degraded voltage, reduced motor efficiency and high temperature conditions are present with the potential error in measurement equipment factored in after data is acquired.

Lessons Learned

A self-assessment was performed in September 2001. The self-assessment concluded that Wolf Creek was compliant with OMN-1, but several weaknesses were identified. The performance of a self-assessment after the program has been in place for several months is critical to successful implementation.

The primary finding was that Wolf Creek was not very aggressive in taking advantage of the economic benefits offered by OMN-1. This finding was primarily due to the final phase of project implementation.

The final phase of implementation was the revision of surveillance procedures to eliminate unnecessary stroke and position indication tests. Few of these revisions have been implemented. Rather than approving and issuing these revisions, the procedures or procedure sections were simply no longer scheduled under the surveillance program where practical. The procedure revisions are still in place, but they are low priority mainly because these changes are not a requirement for compliance with OMN-1. The overhead associated with keeping the procedures in place and scheduling partial surveillance tests is an economic disadvantage.

Post-maintenance testing was another area of weakness and is directly related to the reason procedure revisions to remove stroke time and position indication tests are low priority. At Wolf Creek it is necessary to formally evaluate fluid transfer effects that could result from maintenance in the absence of a system clearance order. If a procedure exists it may be taken credit for in the fluid transfer evaluation. If not, a formal evaluation must be performed or a system clearance order has to be developed and established before maintenance can be performed. Although procedures were identified that reposition MOVs for periodic exercising, they were not evaluated for their use in post-maintenance testing. Over the years, surveillance procedures for stroke timing and position indication were revised so they could easily be used for post-maintenance testing. Until this problem is assessed for each MOV, it is not desirable to revise the existing surveillance procedure to eliminate its stroke time and position indication test.

The measurement of containment isolation valve stroke time is a separate Technical Specification from the IST program. Wolf Creek utilizes the improved Technical Specifications, which state that containment isolation valve stroke times are to be measured in accordance with the IST program. It was not recognized that the use of OMN-1 under the IST program indirectly verifies acceptability of these stroke times by measuring and verifying all of the parameters that must be present in order for a valve to stroke in the amount of time assumed by the safety analysis under design-basis conditions. In other words, if a valve passes its OMN-1 test, it will always stroke in the time required during an accident even though the time is not measured directly by a calibrated instrument. However, if a valve fails its OMN-1 test, it would not be capable of stroking in the time required during an accident, even though it could meet its stroke time requirement if it was capable of stroking at all. Because this was not recognized, some MOV stroke tests were still being performed quarterly.

Other areas of weakness dealt with a lack of procedure guidance. While activities required by OMN-1 were being performed such as margin assessment and trending, the guidance in procedures were viewed as minimal. To the outside observer, this made it difficult to follow how activities were being performed without additional explanation from the MOV engineer. Essentially, the MOV engineer's skill was being relied upon to complete these activities and it would be difficult for someone without this knowledge level to determine what was needed from procedural guidance.

A number of strengths also were identified in the self-assessment. The risk-ranking documentation was one example. Wolf Creek utilized three categories for the risk ranking; high, medium and low. The data provided to the expert panel and expert panel discussions that identify the logic for the final ranking are documented for each valve. The processes and procedures used for field testing also were identified as a strength. Probably the most important strength found was the high knowledge level and strong ownership demonstrated by the MOV engineer, which has significantly helped Wolf Creek successfully transition to the current MOV processes.

OMN-1's Future at ASME

ASME Code Case OMN-1 is being incorporated into the ASME Code. The use of this process will replace stroke time testing of MOVs. This change is already underway at ASME. The Code Case will be incorporated into the OM Code as Mandatory Appendix III. When the NRC approves this Code in

10 CFR 50.55a, all licensees will be required to use this approach under the IST Program.

OMN-1 and GL 96-05 Differences–an Industry Perspective

The primary difference from an industry perspective between the ASME approach and the approach industry chose to address GL 96-05 is in the scope of the two programs. At Wolf Creek, two valves were in the IST program that weren't in the MOV program and two valves were in the MOV program that weren't in the IST program. As it turned out, these four valves had been in both programs in the past, and the justifications for removal were valid for both programs. As a result, all four valves were removed from both programs using the same justifications. In looking at other plants that have scope differences, this type of consolidation is needed. Often the MOV and IST programs were developed at different times with different levels of conservatism. Primarily, it is the IST program that has been more conservative with respect to scope. In most cases to date, it will be possible to justify the exclusion of these valves from the IST program's scope.

Wolf Creek does not have any safety related ball or plug MOVs, so the exclusion allowed by Generic Letter 89-10 Supplement 1 was not a factor. However, this is a concern in industry for those plants that do have safety-related ball or plug valves that have been excluded from the MOV program. This will result in a scope increase unless the NRC approves a relief request for these valves.

Conclusion

ASME Code Case OMN-1 is effective at verifying an MOV's design-basis capability. It is approved by the regulator as a means to address Generic Letter 96-05. For the most part, the industry has chosen the process developed by JOG to address GL 96-05. Wolf Creek successfully implemented a combined JOG/OMN-1 program. This paper and the lessons learned at Wolf Creek may be used as a resource for others that desire to make this transition. Implementation of OMN-1 can provide an economic benefit. In the future, all licensees will be required to update their IST programs to the requirements of ASME Code Case OMN-1 when it is implemented in the OM IST Code as Mandatory Appendix III.

Session 1(c)

Risk-Informed IST of Valves & Pumps

Session Chair

Craig D. Sellers
ITS Corporation

Risk-Informing the Special Treatment Requirements of the NRC Regulations

Thomas G. Scarbrough and Michael C. Cheok
U.S. Nuclear Regulatory Commission

Abstract

In Title 10 of the *Code of Federal Regulations*, the U.S. Nuclear Regulatory Commission (NRC) has established special treatment requirements for structures, systems, and components (SSCs) that perform safety functions at U.S. commercial nuclear power plants. These requirements address such aspects of SSC functional capability as environmental and seismic qualification, quality assurance, and inservice inspection and testing, and are based principally on deterministic considerations. The NRC is developing an alternative regulatory framework that will allow the application of risk insights to determine appropriate treatment for plant SSCs in lieu of the current special treatment requirements. Implementation of this framework will enhance safety by focusing NRC and licensee resources in areas commensurate with their importance to health and safety. It will identify areas where additional requirements may be needed, and will provide flexibility in plant operation and design which can result in burden reduction without compromising safety.

I. Introduction

The regulations of the U.S. Nuclear Regulatory Commission (NRC) in Parts 21, 50, and 100 of Title 10 of the *Code of Federal Regulations* (10 CFR) contain special treatment requirements that impose controls to ensure the quality of SSCs that are within the scope of the regulations. Special treatment requirements are defined as those requirements that exceed normal commercial and industrial practices to provide a greater degree of confidence in the capability of SSCs to perform their safety functions under design-basis conditions throughout their service life. Special treatment requirements encompass such aspects as quality assurance, environmental and seismic qualification, inspection and testing, and performance monitoring.

II. Risk-Informed Regulation Initiative

The NRC has established an initiative to risk-inform the regulatory requirements for the treatment of SSCs used in nuclear power plants in the United States. As discussed in several Commission papers prepared by the NRC staff (e.g., SECY-99-256 and

SECY-00-0194), Option 2 of this initiative involves categorizing plant SSCs based on their safety significance, and specifying the treatment that would provide an appropriate level of confidence in the capability of those SSCs to perform their design functions in accordance with their risk categorization. Under Option 2 of the NRC's risk-informed regulation initiative, RISC (risk-informed safety class)-1 SSCs are defined as safety-related SSCs that have functions determined to be of high safety significance by a categorization process. RISC-2 SSCs are nonsafety-related SSCs of high safety significance. RISC-3 SSCs are safety-related SSCs with functions of low safety significance. RISC-4 SSCs are nonsafety-related SSCs of low safety significance. As described in SECY-98-300, the NRC staff expects there to be confidence that safety-related SSCs categorized as low risk-significant remain functional under design-basis conditions. Similarly, in SECY-00-194, the staff stated that nuclear power plant licensees will be required to maintain the functional capability of safety-related SSCs using existing or new programs.

On November 7, 2001, the NRC staff held a public workshop to discuss the boundary conditions and possible alternatives for the treatment of low-risk safety-related SSCs that would be required by the proposed rule (10 CFR 50.69) to be prepared under Option 2 of the risk-informed regulation initiative. The boundary conditions represent those attributes that the proposed rule must satisfy to be considered acceptable under Option 2 of the risk-informed regulation initiative. The NRC staff determined the boundary conditions for the treatment of low-risk safety-related SSCs under Option 2 to be as follows: (1) nuclear power plant licensees must maintain the design functions of safety-related SSCs with functions of low safety significance (referred to as RISC-3 SSCs) at the conditions under which the intended functions are required to be performed as described in the updated FSAR; (2) treatment must maintain functionality of RISC-3 SSCs consistent with the categorization process assumptions; and (3) the level of regulatory assurance for the treatment of RISC-3 SSCs needs to be consistent with the NRC's mission to ensure adequate protection of the health and safety of the public. At the public workshop, the NRC staff discussed various alternatives in satisfying the boundary conditions for Option 2 of the risk-informed regulation initiative.

On February 21, 2002, the NRC staff held a public meeting with stakeholders to discuss the consideration of 10 CFR 50.55a in the development of 10 CFR 50.69 as part of Option 2 of the NRC's effort to risk inform the regulations. The NRC regulations in 10 CFR 50.55a incorporate by reference Sections III and XI of the American Society of Mechanical Engineers (ASME) *Boiler and Pressure Vessel Code* (BPV Code) and the ASME *Code for Operation and Maintenance of Nuclear Power Plants* (OM Code), and also certain standards of the Institute of Electrical and Electronics Engineers (IEEE). At the meeting, the NRC staff presented an overview of the Option 2 effort and the status of the 50.69 rulemaking. ASME representatives discussed the ASME Code process and stated that, at this time, ASME does not have a single position on the consideration of 10 CFR 50.55a in the 50.69 rulemaking, because of the significant differences of opinion among its members on the issue in terms of safety and burden, and because the staff position regarding compliance with ASME risk-informed Code Cases had not been set. ASME indicated that it would consider the need to take a position on 50.69 following establishment of a staff position on the use of ASME risk-informed

Code Cases. ASME is continuing to prepare Code Cases to reduce burden on licensees for low-risk safety-related SSCs. NEI considers the requirements in 10 CFR 50.55a to represent an unnecessary burden for licensees and NRC. NEI believes the proposed requirements in the draft 50.69 rule provide sufficient regulatory assurance for low-risk safety-related SSCs.

The NRC staff is considering the feedback provided during the public workshop and meeting in preparing a proposed rule (10 CFR 50.69) for risk-informing the special treatment requirements of the NRC regulations. The staff will review the comments submitted following issuance of the proposed rule for public comment. Upon completion of the review of public comments, the NRC staff plans to prepare a final rule to risk-inform the special treatment requirements of the NRC regulations if determined to be appropriate.

III. Proof-of-Concept Effort

On July 13, 1999, STP Nuclear Operating Company (STPNOC), licensee of the South Texas Project Units 1 and 2 nuclear power station, submitted a request under 10 CFR 50.12 for exemptions from the special treatment requirements of 10 CFR Parts 21, 50, and 100 for SSCs categorized at STP as low safety-significant (LSS) or non-risk significant (NRS) that are within the scope of these regulations. The NRC staff conducted the review of the STPNOC exemption request as a proof-of-concept effort for Option 2 of the risk-informed regulation initiative. In its submittal, the licensee requested approval of the exemptions primarily based on its categorization process that would allow the treatment of SSCs at STP according to their risk significance. Although relying heavily on STPNOC's categorization process in reaching the conclusions regarding the individual exemption requests, the staff recognized that the functionality of SSCs must be maintained consistent with the Option 2 approach, and to support the implicit assumption in the categorization process that SSCs will remain capable of performing their safety functions under design-basis conditions. The staff did not consider it necessary to maintain the same level of confidence in the functionality of low-risk SSCs as provided by the special treatment requirements. In assessing functionality, the staff's review focused on whether the programmatic elements of the licensee's treatment processes, if effectively implemented, could be sufficient for the exempted SSCs to remain capable of performing their safety functions under design-basis conditions. The staff determined that it was not necessary to assess the details regarding how the licensee will implement its treatment processes for safety-related LSS and NRS SSCs. On August 3, 2001, the staff granted STPNOC's request for exemptions from many of the special treatment requirements in the NRC regulations for safety-related LSS and NRS SSCs in consideration of the categorization and treatment processes to be applied at STP.

With respect to its proposed treatment practices for SSCs, STPNOC stated that safety-related SSCs classified as high safety-significant (HSS) or medium safety-significant (MSS) at STP will continue to receive treatment required by the NRC regulations, and will be evaluated to identify any risk-significant functions not being treated under its current programs. STPNOC will evaluate nonsafety-related HSS and MSS SSCs to determine whether enhanced treatment is warranted for their safety-significant functions. Rather than implementing the special treatment requirements of the NRC regulations, STPNOC stated that it will apply alternative treatment processes to provide reasonable confidence that safety-related LSS and NRS SSCs will be capable of performing their safety functions, commensurate with their significance to safety. STPNOC did not request an exemption from the provisions of 10 CFR Parts 50 and 100 that specify design or functional requirements for SSCs. STPNOC also will not use the exemptions to change any design or functional requirements contained in its FSAR or plant technical specifications.

In its revised FSAR, STPNOC states that the purpose of treatment applied to safety-related HSS and MSS SSCs is to maintain compliance with the NRC regulations and the ability of these SSCs to perform their risk-significant functions consistent with the categorization process. The safety-related HSS and MSS SSCs will continue to receive the treatment required by the NRC regulations. Where STPNOC takes credit for safety-related SSCs performing functions that are beyond the design basis, the licensee will evaluate whether these risk-significant functions are adequately treated under its current programs.

The purpose of treatment applied to nonsafety-related HSS and MSS SSCs is to maintain their ability to perform risk-significant functions consistent with the categorization process. Nonsafety-related HSS and MSS SSCs will continue to receive any existing special treatment required by the NRC regulations. Additionally, STPNOC will consider the risk-significant functions of these SSCs for enhanced treatment. Nonsafety-related HSS and MSS SSCs may perform risk-significant functions that are not addressed by the special treatment requirements in the NRC regulations or STP's current treatment programs. When a nonsafety-related SSC is categorized as HSS or MSS, STPNOC documents the condition under its corrective action program, and determines whether enhanced treatment is warranted to enhance the SSC's reliability and availability.

The purpose of the treatment practices for safety-related LSS and NRS SSCs is to provide STPNOC with reasonable confidence that these SSCs will maintain their functionality under design-basis conditions. In its FSAR, STPNOC describes the processes for design control; procurement; installation; maintenance; inspection, test, and surveillance; corrective action, oversight; and configuration control, that will be applied to safety-related LSS and NRS SSCs. For example, standards required by the State of Texas and national consensus commercial standards will be used at STP in the treatment of safety-related LSS and NRS SSCs. Further, STPNOC will consider available recommendations for SSC treatment from the applicable vendors, or might use an alternative to those recommendations if there is a technical basis that supports the functionality of the safety-related LSS and NRS SSCs.

STP outlines other high-level aspects of the treatment processes for safety-related LSS and NRS SSCs in its FSAR. For example, STPNOC's design control program for safety-related LSS and NRS SSCs will continue to comply with 10 CFR Part 50, Appendix B, "Quality Assurance Criteria for Nuclear Power Plants and Fuel Reprocessing Plants." The purpose of the procurement process for safety-related LSS and NRS SSCs will be to procure replacement SSCs that satisfy the design inputs and assumptions to support STPNOC's determination that these SSCs will be capable of performing their safety functions under design-basis conditions. The technical requirements (including applicable design-basis environmental and seismic conditions) for items to be procured will include the design inputs and assumptions for the item. One or more of five methods (vendor documentation, equivalency evaluation, technical evaluation, technical analysis, and testing) described in the FSAR are said to provide a sufficient basis to determine that the procured item can perform its safety-related function under design-basis conditions, including applicable design-basis environmental (temperature and pressure, humidity, chemical effects, radiation, aging, submergence, and synergistic effects) and seismic (earthquake motion, as described in the design bases, including seismic inputs and design load combinations) conditions. The purpose of the installation process for safety-related LSS and NRS SSCs will be to achieve proper installation and testing of replacement SSCs to support STPNOC's determination that these SSCs will be capable of performing their safety-related functions under design-basis conditions. The purpose of the maintenance process for safety-related LSS and NRS SSCs will be to establish the scope, frequency, and detail of maintenance activities necessary to support STPNOC's determination that these SSCs will remain capable of performing their safety-related functions under design-basis conditions. For an SSC in service beyond its designed life, STPNOC will have a technical basis to determine that the SSC will remain capable of performing its safety-related function. The purpose of the inspection, test, and surveillance process for safety-related LSS and NRS SSCs will be to obtain data or information that allows evaluation of operating characteristics to support STPNOC's determination that these SSCs will remain capable of performing their safety-related functions under design-basis conditions throughout the service life. STPNOC's corrective action program for safety-related LSS and NRS SSCs will continue to comply with 10 CFR Part 50, Appendix B. The purpose of the management and oversight process for safety-related LSS and NRS SSCs will be to control the implementation, and to assess the effectiveness, of the treatment processes to support STPNOC's determination that these SSCs will remain capable of performing their safety-related functions under design-basis conditions. STPNOC will accomplish its management and oversight process through approved procedures and guidelines, including qualification, training, and certification of personnel. STP procedures will also specify requirements for documentation, reviews, and record retention related to completed work activities. STPNOC indicates that planned changes to, or elimination of, commitments described in the FSAR or other licensing bases documentation that address issues identified in documents such as NRC generic communications (e.g., generic letters or bulletins), NRC orders, and notices of violation, related to safety-related LSS and NRS SSCs will be evaluated in accordance with an NRC-endorsed commitment change process. STPNOC states that its design control process will ensure that the configuration of the Station is properly reflected in design documents and drawings.

STPNOC considered the treatment of nonsafety-related LSS and NRS SSCs not to be subject to regulatory control.

In response to the STPNOC exemption submittal, the NRC staff reviewed the licensee's proposal to apply (1) the current special treatment requirements to safety-related HSS and MSS SSCs, and evaluate their risk-significant functions to identify any functions not being adequately treated under its current programs; (2) any existing special treatment required by the NRC regulations to nonsafety-related HSS and MSS SSCs (such as equipment relied on to meet regulatory requirements in 10 CFR 50.62

associated with an anticipated transient without a reactor scram) and consider the risk-significant functions of these SSCs for enhanced treatment; and (3) alternative treatment processes to safety-related LSS and NRS SSCs in lieu of the NRC special treatment requirements. The licensee did not provide the categorization of specific SSCs and treatment to be applied to those SSCs, because the licensee intends to implement its alternative treatment processes in parallel with the implementation of the categorization process over the remaining licensed period of the facility. Nevertheless, the NRC staff recognizes the significance of the exemptions from the regulations in terms of the potential impact on the treatment applied to SSCs performing safety functions at STP. For example, the licensee indicated that approximately 77 percent of the safety-related SSCs at STP might be categorized as LSS or NRS.

The scope of the NRC staff's evaluation did not include an assessment of the licensee's procedures for implementing the treatment processes at STP. The staff did not consider the review of the details for implementation of the program to be necessary given the conclusion that SSCs identified as LSS or NRS by the categorization process do not contribute significantly to plant risk based on a sensitivity study conducted by the licensee. Effective implementation of the treatment processes, such that the design bases and functionality of the safety-related LSS and NRS SSCs are maintained, remains the responsibility of the licensee. The NRC staff's evaluation relied on the use of sound engineering judgment by the licensee in implementing the treatment processes.

The NRC staff did not characterize the treatment processes established by STPNOC as a performance-based alternative to the special treatment requirements of the NRC regulations. Performance-based processes monitor specified attributes indicative of operational performance of SSCs, evaluate the operational performance against specified acceptance criteria, and require corrective actions if the specified acceptance criteria for the SSC are not achieved. Typically, special treatment requirements (such as those contained in 10 CFR 50.49 for environmental qualification, and 10 CFR Part 100, Appendix A, Section VI, for seismic qualification) are not performance-based, because they only require a one-time qualification test or analysis, prior to placing the SSC in service, to verify that the SSC can perform its safety functions at design conditions. Also, with the possible exception of inservice inspection and test processes, STPNOC's treatment processes are not performance-based, because SSCs typically are not operated under design-basis conditions.

With respect to safety-related HSS and MSS SSCs at STP, the NRC staff recognized that the licensee will continue to apply the special treatment requirements of the NRC regulations. In addition, the staff reviewed STPNOC's high-level description of the process in the FSAR to validate assumptions credited in the risk assessment for safety-related HSS and MSS SSCs that support or perform risk-significant functions beyond the design basis of the plant. The staff's review focused on whether the treatment for safety-related HSS and MSS SSCs can provide an acceptable approach to maintain their functionality not only with respect to the design-basis functions addressed under the NRC regulations, but also regarding any risk-significant functions assumed in the categorization process that are beyond the design basis for those SSCs. The NRC staff found the process described in the FSAR to contain high-level elements and objectives that, if applied with sound engineering judgment, will allow STPNOC to evaluate the treatment applied to the safety-related HSS and MSS SSCs to ensure that the existing controls are sufficient to maintain the reliability and availability of these SSCs in a manner that is consistent with their categorization.

With respect to nonsafety-related HSS and MSS SSCs at STP, the NRC staff noted that the licensee will continue to implement applicable special treatment requirements (e.g., anticipated transient without reactor scram in 10 CFR 50.62, and station blackout in 10 CFR 50.63) of the NRC regulations. In addition, the staff reviewed STPNOC's high-level description of the process in the FSAR to validate assumptions credited in the risk assessment for nonsafety-related HSS and MSS SSCs that support or perform risk-significant functions. The staff's review focused on whether the treatment process for nonsafety-related HSS and MSS SSCs can provide an acceptable approach to maintain their functionality. The NRC staff found the process described in the FSAR to contain the elements and high-level objectives that, if effectively implemented, will allow STPNOC to evaluate the treatment applied to nonsafety-related HSS and MSS SSCs to ensure that the existing controls are sufficient to maintain the reliability and availability of these SSCs in a manner that is consistent with their categorization.

With respect to safety-related LSS and NRS SSCs at STP, the NRC staff evaluated the licensee's request for an exemption from the special treatment requirements in 10 CFR Parts 21, 50, and 100 of the NRC regulations. The licensee did not request an exemption from the provisions in 10 CFR Parts 21, 50, and 100 that specify design or functional requirements for safety-related LSS and NRS SSCs to perform their safety functions. Further, the licensee stated that the exemptions will not change any design or functional requirements in the FSAR or plant technical specifications. Based on STPNOC's robust categorization process, the NRC staff found that the treatment for safety-related SSCs determined to have a low impact on plant risk may be reduced from the level provided by the special treatment requirements of the NRC regulations. However, all safety-related LSS and NRS SSCs continue to be required to be capable of performing their safety functions under design-basis conditions (albeit at a lower level of confidence than for SSCs categorized as HSS or MSS). The NRC staff's evaluation of the treatment for safety-related LSS and NRS SSCs focused on the elements and high-level objectives of STPNOC's treatment processes described in the FSAR to determine whether STPNOC can maintain the design bases and functionality of safety-related LSS and NRS SSCs under design-basis conditions.

An NRC-sponsored study by the Idaho National Engineering and Environmental Laboratory described in NUREG/CR-6752 (January 2002) revealed that normal commercial and industrial practices vary widely between nuclear power plants, and apply a wide range of activities regarding the functionality of balance-of-plant SSCs at individual plants. For example, licensees might apply specific controls for design, installation, and monitoring of a balance-of-plant SSC that directly supports the generation of electric power, but might allow a balance-of-plant SSC that does not directly support power generation to degrade with repairs performed when the SSC is found to not be functional. As a result, the NRC staff has determined that reliance only on industrial and commercial practices may not provide an adequate basis for reaching a finding that the functionality of safety-related LSS and NRS SSCs will be maintained.

With respect to its categorization process, STPNOC modeled common-cause failure in multiple train systems (intrasystem effects) in the probabilistic risk assessment (PRA) system analyses for all active components within a system. However, STPNOC explicitly modeled potential common-mode failures in diverse systems in its PRA for only certain basic events. STPNOC stated that, for other types of equipment (such as motor-operated valves), potential changes in the basic event failure data were not carried across diverse systems (intersystem effects) because the licensee believed that the unique operating condition for diverse systems affects the failure rates for their applicable components. The NRC staff considers that the treatment processes could affect SSC reliability across multiple plant systems within the scope of the exemption. The sensitivity study performed by STPNOC increased the failure rates of modeled LSS SSCs and their common-cause relationship by a factor of 10. The staff considered STPNOC's assertion that the assumed increase in failure rate in the sensitivity study bounds the failure rate that might result from the reduction in treatment to be reasonable only if treatment processes for safety-related LSS and NRS SSCs described in the FSAR are effectively implemented such that SSC functionality is maintained. NUREG/CR-5485, "Guidelines on Modeling Common-Cause Failures in Probabilistic Risk Assessment," indicates that defense strategies for common-cause failures typically include design control; use of qualified equipment; testing and preventive maintenance programs; procedure review; personnel training; quality control; barriers; diversity (functional, staff, equipment); and staggered testing and maintenance. The NRC staff considers effective implementation of the treatment processes for safety-related LSS and NRS SSCs to be necessary to ensure that the potential for common-cause failures is minimized.

In evaluating the specific aspects of the STPNOC submittal, the NRC staff reviewed the FSAR to determine if it is consistent with the technical bases for the requested exemptions. The staff's review focused on whether the FSAR provides an understanding of the purpose of the treatment processes for SSCs at STP to maintain the design bases and functionality of these SSCs under all design-basis conditions. The staff also considered whether the treatment process would be able to maintain the capability of the SSCs to perform risk-significant functions beyond the design basis where credited in the categorization process.

The NRC staff concluded that the process described in the FSAR contains elements and high-level objectives that, if applied with sound engineering judgment, will allow STPNOC to evaluate the treatment applied to the safety-related and nonsafety-related HSS and MSS SSCs to ensure that the existing controls are sufficient to maintain the reliability and availability of these SSCs in a manner that is consistent with their categorization.

The NRC staff concluded that the treatment processes described in the FSAR contain elements and high-level objectives that, if effectively implemented, will provide reasonable confidence that safety-related LSS and NRS SSCs at STP are capable of performing their safety functions under design-basis conditions, including environmental and seismic conditions, throughout their service life. As part of its review, the staff identified initial approaches considered by STPNOC that could have led to ineffective implementation of the treatment processes for safety-related LSS and NRS SSCs. For example, STPNOC initially suggested it was unnecessary to perform engineering analysis, qualification testing, or other specialized efforts to provide empirical evidence or other justifications of an SSC's ability to function in adverse environments. However, STPNOC subsequently specified in its FSAR the methods that will provide a sufficient basis to determine that a procured item can perform its safety function under design-basis conditions, including applicable design-basis environmental and seismic conditions. The NRC staff emphasized that STPNOC is responsible for effective implementation of the treatment processes for safety-related LSS and NRS SSCs to ensure that these SSCs remain capable of performing their safety functions under design-basis conditions.

IV. Conclusions

The NRC regulations specify special treatment requirements for SSCs that perform safety functions at U.S. commercial nuclear power plants. These requirements address such aspects of SSC functional capability as environmental and seismic qualification, quality assurance, and inservice inspection and testing, and are based principally on deterministic considerations. The NRC is developing an alternative regulatory framework that will allow the application of risk insights to determine appropriate treatment for plant SSCs in lieu of the current special treatment requirements. Implementation of this framework will enhance safety by focusing NRC and licensee resources in areas commensurate with their importance to health and safety. It will identify areas where additional requirements may be needed, and will provide flexibility in plant operation and design which can result in burden reduction without compromising safety. Option 2 of the NRC's initiative to risk-inform the regulatory requirements for the treatment of SSCs used in nuclear power plants involves categorization of SSCs based on their safety significance, and specification of the treatment that would provide an appropriate level of confidence in the capability of SSCs to perform their design functions in accordance with their risk categorization. In a proof-of-concept effort, the NRC recently granted exemptions from the special treatment requirements for safety-related SSCs categorized as having low risk significance by the licensee of the South Texas Project, based on a review of the licensee's high-level objectives of the planned treatment for safety-related and high-risk nonsafety-related SSCs. The risk-informed regulation initiative and the STP exemption review reflect the NRC's ongoing efforts to incorporate risk insights into the regulation of nuclear power plants.

V. References

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Optimisation of Testing Parameters Using Risk/PSA Analyses

Gheorghe Florescu, Mihail Cojan
Institute for Nuclear Research, Pitesi, Romania

Abstract

Probabilistic Safety Assessment (PSA) methodology development created the opportunity for multiple nuclear power plant (NPP) model applications in order to improve operation, maintenance, testing and repairing of systems, structures, and components (SSC). An important task in risk/ PSA analyses is to optimize the operational parameters for Critical Systems, Structures, and Components (CSSC) in order to maintain the NPP availability and safety, to prevent the occurrence of accidents, especially severe accidents. The paper presents a brief description of the most important aspects of the methods used to optimize the testing parameters for CSSC taking into account the previous developed NPP PSA model and PSA modeling tools.

Systems, structures and components that influence decisively NPP reliability are considered as critical. Also, for accident conditions, the SSCs, which have a major influence on system availability/operability, are considered as critical. Risk analysis and PSA techniques are used as a basis for testing optimization.

Examples related to valve and pump testing optimization, specific for Cernavoda 1 NPP, are selected and presented in order to state the analytical methods. CPSE phase B+ (Cernavoda Probabilistic Safety Evaluation) model was used—and briefly stated in the paper—as a data/information base to perform the analyses. PSAMAN v.4.0 computer code, developed in Institute for Nuclear Research (INR), was used as an analytical tool to perform the PSA model processing.

1. Introduction

The events that appear in the power plant or installation operation are recorded. Then, a study is performed to classify and to group them in some categories. The events that have or could have an impact on the plant safety barriers or could affect the operational personnel or the public are selected for further analyses in order to identify the event root causes and to prevent the recurrence of such events.

Usually, for NPP operation, the maintenance department is made up of an electric maintenance section and a mechanical maintenance section, and consists of many persons. The significant events are those events that could appear during plants/ installation commissioning or during operation of such utilities and have impact on the safety barriers/margins/limits of them.

The analysis methodologies are based on probabilistic/deterministic techniques. The deterministic process of the evaluation of the installation safety is based on the design basis accident identification. These events could consist of extreme conditions (but possible) considered to be possible to appear in the power installation that are analyzed. Until now the probabilistic process of analyzing the significant events was not specifically considered as part of the significant events analysis methods.

In the analysis of one power plant system or installation/process the first step is familiarization with this installation/process. Significant events are considered the events that appear during a system operational state and affect the safety barriers leading to undesired consequences and impact to the environment. The main methods to analyze the significant events are based on deterministic and probabilistic approaches.

2. Analysis Methods

The further analysis of the significant events is performed in order to guarantee that in the future the possibility of appearance of such events will be as minimal as possible, and also, in order to identify the related events that could lead to similar events in the power plant.

Ranking of the operational events

The ranking of the operational events was conducted taking into account the possible applications of such events. One of the questions is whether ranking is the appropriate term to denominate such events processing activities and whether classification or screening, sorting, grouping are not more appropriate terms. The decision was dependent on the complexity of the ranking term. Processing of the operational events means: *selection of the event, sorting of the event, grouping of the event, ranking of the event.*

A preliminary judgement is necessary using the event description. Ranking method selection depends on some ranking criteria and application of the ranking results. The ranking criteria are both qualitative and quantitative and additionally refer to safety impact, management and human performance deficiencies, possible accident consequences, likelihood reasons, number and efficiency of available barriers to prevent degradation of plant state.

The operational events ranking criteria consider: *nuclear safety impact, significance of the normal operation perturbation due to operational event, significance of the transient induced by the operational event, common cause impact of the operational event, design deficiencies that were revealed by the operational event, departure of the estimated probability comparing target objective.*

Decision to perform a detailed system/unit analysis depends on the significance of the operational event looking to the ranking criteria. Performance indicators include plant availability, numbers of reactor trips, total number of the operational events, number of the events per year, reactor shutdown per year, number of human errors. The most known method to process the operational events is the method to obtain the failure rates of the components.

Processing of operational data to obtain SSC failure rates

The SSC reliability data are an essential component of any probabilistic safety evaluation study. The quality of data could have a major change on the quality of the overall PSA study. A complete PSA database must consist of: *components failure data, SSC test and maintenance data, initiating events data, common cause failures data, uncertainty data, human errors data, SSC aging data.*

The reliability data may make many estimations of the SSC failure rates. Development of a reliability database is done by means of elementary rules of mathematical statistics. The types of data are discrete or continuous. Bulk data are processed in order to give a shape that is appropriate to the subject of an analysis. By grouping data, part of the information is lost, but could obtain more clarity and adaptability by looking to the data processing operations.

SSC configuration management

During NPP operation, the SSC state may change due to failures, maintenance and testing. These changes lead to modifications in plant configuration that affect the plant. The management of SSC configuration means: identifying risk significant configuration and providing solutions to avoid them and to restore the configuration with an acceptable risk level, assurance of flexibility in NPP operation when the implications on risk are minimum, prevention of high-risk recurrence due to SSC aging generated by maintenance.

Modification/Optimisation of AOTs and STIs

The allowed outage times (AOTs) and the surveillance test intervals (STIs) risk measures can be used to categorize requirements. Component test requirements can be treated according to their contribution to plant safety.

By surveillance tests, the failures modes are identified, which significantly contribute to risk and the types of test, which simulate demands, experienced in risk-significant accident scenarios. When a component fails to perform its function, plant risk level increases due to loss of function or capability. The increased risk level depends on the importance of the component in defining plant risk. The increased risks existing during downtime are termed downtime risks.

By AOT is defined the time that could be allowed for out of operation of an SSC. The necessity to modify AOT appears: if the failures that appear during operation require a long time, if the component AOT is not clearly defined in technical specifications and lead to ambiguity between operation personnel and regulatory body, if it is necessary the monitoring of the risk.

The critical maintenance activities could be identified using some importance factors:

> "*Risk Achievement Worth*" (*RAW*) = contribution of a function to risk level
>
> "*Risk Reduction Worth*" (*RRW*) = contribution of a function to risk reduction level

The functions that have high *RAW* are important for the quality assurance programs and for inspection activities.

The functions that have high RRW are of interest for the efforts to reduce risk. In this paper is presented a case of association between operating events and CSSC. The main reference was the PSA study for Cernavoda NPP, Unit1 CPSE B+ .

AOT versus risk

The following steps should be considered to determine AOT: *identify all the SSCs for which AOT is assessed, verify if the PSA model contains these SSCs, determine the contributions in risk due to AOT, determine the acceptance criteria of AOT contribution, determine AOT that satisfy criteria.*

There are two contributions associated with AOT: *singular AOT contribution and AOT frequency contribution.* Singular AOT contribution is associated with a failure occurrence and AOT frequency contribution is the cumulated contribution due to several successive failures.

Calculation of singular AOT contribution:

F_1 = level of risk with the SSC failed;

F_0 = level of risk with the component available;

d = failure time;

$(F_1 - F_0) \cdot d$ = singular AOT contribution = r_{AOT}

Calculation of frequency AOT contribution:

R_{AOT} = frequency AOT contribution $= w \cdot d \cdot (F_1 - F_0) = w \cdot r_{AOT}$

w = failure rate.

Loss of the process control during system operation could lead to undesirable events and, as a result, to loss of operational safety.

The factors that generated the event

A graphical representation is used for means of which the events are caused. The diagrams use symbols for representation of the initiating event, normal process events, causal factors, root causes, failure/defense barriers, and basic events.

The possible causal factors are focused on the diagrams. Using this technique, the irrelevant causal factors often become evident. Such diagrams are useful for complex situations being more relevant than narrative description.

Using the established symbols is possible to represent actions, modification/deviations, barriers, causal factors and root causes.

The causes that generated the event

The causes could be grouped in causes related by: management, system/plant design, operation/test/maintenance procedure, activities planning/organization, operation of the plant/system/installation, etc.

The effects of the significant events on the plant operation and on the environment

These types of analyses are correlated with different analysis methods: probabilistic analyses, thermo-hydraulic analyses, physical calculation/analyses, others studies/analyses for plant/installation safety.

Significant events analyses types

In the analysis of the significant events, deterministic and probabilistic analyses are used. The deterministic analyses are based on the design basis accident identification. Such events include extreme conditions that are possible in the power installation. The design of the safety systems, which have the role to control and mitigate the consequences of the events, is performed based on them. The recommendations that result after utilization of the deterministic methodology are incorporated in the design, operation standards, procedures and rules. For criteria, standards and safety limits, establishment/ development research, standardization, design, operation activities are performed. The results of these efforts must lead to a high level of safety and a large field of rules and standards comparing to other fields. The deterministic methodology doesn't deny the "*probability*" or "*likelihood*" approach.

The method of assessment of significant events using PSA model/study

Defining the event significance

To define the event significance the following factors must be taken into account: event frequency, the possible consequences of the event, the uncertainty due to the event understanding and the assessment tools.

In this context, the significance of the event is obtained from the event frequency and the possible causes. The method analytical basis is derived from the importance data that are used in the PSA studies. From the point of view of the risk a numerical measurement, S_X, of the event could be obtained from the following relation:

$$S_x = \frac{PDF_x}{PDF_{BL}}$$

where: PDF_x - the updated plant damage frequency related to the observed event "x"

PDF_{BL} - the reference frequency of the plant damage.

The reference frequency of the plant damage results from the plant specific analysis or from the generic PSA results that are applied to a specific plant. The numerical significance of the observed event is designated by an updated fraction of the plant damage that implies the event. The analysis of the updated fraction must take into account the possible differences between the actual frequency and the impact of the observed event comparing with the data and the impact used in the PSA analyses.

The main purpose of this evaluation is to develop a numerical scale that allows event comparison and the analysis of the priorities in order to upgrade the plant.

The method for the event significance measurement, based on PSA model, does not substitute but completes the results obtained from a complementary method.

3. Test Optimisation Using PSA Results

The PSA analysis is capable of evaluating the risk and the safety and operation implications to the nuclear power plant. Risk analysis is the main way to assure the optimisation of the maintenance activity. Evaluation of test activities has to carry out the following objectives:

- Identification of the field of interest for personnel that perform test and maintenance activities;
- Identification of the fields where the regulation modifications are required, taking into account the plant safety;
- Optimisation of risk/benefits using planning and supervision tests;
- Detection of dependent failures and configurations of the plant with risk significant implications;
- Minimization of dependence between human errors in activities such as maintenance, test, repairs, calibration.

The process of optimisation can be done at the component level, the system level and the plant level. The risk-based application enables the test plan to be changed to reduce the risk if necessary or to perform needed tests with confidence that safety is adequate.

A quantitative PSA method is used to prioritize or identify risk-significant equipment. The method involves the use of PSA results to prioritize all equipment modeled in the PSA according to objective figures of merit, which merit measure risk-significance. Using importance evaluations, we developed a procedure for identifying risk-significant equipment, which consists of the following steps:

- The initial test program: includes initial list with data referring to components, equipment test activities;
- The list with important components:

obtained using risk analysis, important evaluation (*Risk Achievement Worth, Fussel-Vesely Importance*);

- The comparison between the list of the test programs with the initial list;
- The evaluation of comparison and implementation of the observations using a final form as a document.

The aim of the first step is to obtain the initial list with data referring to all the system components subjected to test/maintenance program. For the second step, risk indicators have to be calculated for each component from the initial list. The ranking of the components in terms of their importance to the system unavailability was done using:

- The Risk Reduction Worth (RRW) that determines the relative proportion of risk induced by the failure
- The Risk Achievement Worth (RAW) that determines the relative measurement of the potential loss of the system if the component is unavailable
- The Fussel-Vesely Importance (FV) that determines the fractional contribution of the component failure to the system unavailability.

To analyze the new values, PSA analysts must cooperate with maintenance and operation personnel. These can judge the predictable evolution of the reliability of equipment in accordance with the maintenance program, whereas safety experts can judge where this evolution is acceptable or not. For step 4, the final results and recommendations made by the PSA and maintenance experts should be assembled in a final form as a document.

Optimisation of Maintenance, Test and Repair

For the evaluation of failure probability of the SSC, it is necessary to add the component unavailability due to maintenance or test:

$$Q_m = \frac{m}{M} \tag{1}$$

$$Q_t = \frac{t}{T} \tag{2}$$

where:

m – the maintenance time duration,
M – time interval in which is performed the maintenance activity,
t – the test time duration,
T – time interval in which is performed the test activity.

The formula to calculate the failure probability for the irreparable component during the mission is:

$$P = \lambda \cdot T_m \tag{3}$$

where:

T_m is the mission time.

The calculation formula for the irreparable component's unavailability is identical with the above one:

$$\bar{A} = \lambda \cdot T_m \tag{4}$$

For a component that failure is monitored, in the dormant state, the unavailability calculation formula is:

$$\bar{A} = \frac{\lambda \cdot MTTR}{1 + \lambda \cdot MTTR} \tag{5}$$

where:

λ - the failure rate for the dormant state,
MTTR – mean time to repair with the

waiting/discovering time, so the time between the moment when the fault occurred and the moment of the fault detection, given in time units.

For a tested component (dormant state component, periodically tested), the unavailability is calculated using the formula:

$$\overline{A} = \frac{\lambda \cdot T}{2} + \lambda \cdot MTTR \qquad (6)$$

where:

T – test interval.

The model for which is performed the analysis could be for instance a system that consists of many valves, a group of heat exchangers for water cooling, or a group of two redundant pumps. Using these formulae and risk/PSA analyses the testing parameter should be optimized (see Figure 1).

System level analysis

The optimization of testing parameters using risk/PSA analyses was applied at a system level to two systems (one is a process system "End Shield Cooling System" and the other is a safety system, "Emergency Core Cooling System") and at the component level to analyze the test intervals and test durations influence on the process system reliability and failure probability.

It used the following formulae:

F1. $$P = \frac{\lambda \cdot T}{2} \qquad (7)$$

F2. $$P + Q = \frac{\lambda \cdot T}{2} + \lambda \cdot MTTR + \frac{t}{T} \qquad (8)$$

F3. $$Q = \frac{t}{T} \qquad (9)$$

Process system analyzed

The "End Shield Cooling System" (ESCS) is a process system that contains many valves, heat exchangers, pumps and other mechanical and electrical components.

Table 1 includes some test activities associated with this system. To the pump motor PM2 are associated 2 testing activities.

Table 2 presents the processed results for test optimization.

Figure 2 shows the failure probability variation (F1 and F2) with the test interval and test duration. The F1 and F2 curves are

Table 1. Test activities associated with the ESCS system

No.	Equipment	Activities	Duration [h]	Test interval [days]	Type of activity
1.	Pump P2	Measurement of the vibrations at the pump-motor bearings. Visual inspections at pump P2.	0.5	56	Test
2.	Pump motor PM2	Monthly check of nominal currents at pump motor PM2	1	56	Test
3.	Relief valve RV65	Verification on test desk of relief valve RV65	5	1820	Test
4.	Relief valve RV66	Verification on test desk of relief valve RV66	5	1820	Test
5.	Relief valve RV67	Verification on test desk of relief valve RV67	5	1820	Test

Table 2. The processed results for test optimization. ESCS System – Process system analysis.

No.	T [days]	F1	F2 [t = 20h]	F3 [t = 20h]	F2 [t = 4 h]	F3 [t = 4 h]	F2 [t = 2 h]	F3 [t = 2 h]
1	400	7.7981E-04	7.8671E-04	0.0021	7.8119E-04	0.00042	7.8050E-04	0.00021
2	350	7.6157E-04	7.6944E-04	0.0024	7.6315E-04	0.00048	7.6236E-04	0.00024
3	300	7.4332E-04	7.5252E-04	0.00278	7.4516E-04	0.00056	7.4424E-04	0.00028
4	250	7.2508E-04	7.3611E-04	0.0034	7.2729E-04	0.00067	7.2618E-04	0.00034
5	200	7.0684E-04	7.2062E-04	0.0042	7.0960E-04	0.00083	7.0822E-04	0.00042
6	150	6.8859E-04	7.0698E-04	0.0056	6.9227E-04	0.00112	6.9043E-04	0.00056
7	120	6.7765E-04	7.0063E-04	0.00695	6.8224E-04	0.0014	6.7995E-04	0.0007
8	100	6.7035E-04	6.9793E-04	0.0084	7.8065E-04	0.0017	6.7311E-04	0.00085
9	75	6.6123E-04	6.9799E-04	0.0112	6.6858E-04	0.0023	6.6491E-04	0.00115
10	56	6.5356E-04	7.1266E-04	0.015	6.6342E-04	0.003	6.5849E-04	0.0015
11	25	6.4298E-04	7.5329E-04	0.034	6.6505E-04	0.0067	6.5402E-04	0.00335
12	10	6.3751E-04	9.1326E-04	0.084	6.9266E-04	0.017	6.6509E-04	0.0085
13	1	6.3423E-04	3.3904E-03	0.84	1.1857E-03	0.17	9.0997E-04	0.084

closed when the test duration is 2 hours for the test intervals between 30 and 100 days. This method is an approximate test sensitivity analyses.

Figure 3 shows the failure probability variation (F1 and F3) with the test interval and test duration. The F1 and F3 curves are closed when the test duration are 2 and 4 hours for the test intervals between 70 and 100 days. Due to the low sensitivity of system reliability to test parameters this is an approximate test sensitivity analyses.

Component level analysis

The standby pump (P2) was considered that is a testable component for the ESCS. The processed results for test optimization are presented in Table 3.

Figure 4 shows the failure probability dependence of testing parameters using the formulae F1 and F2. It was observed that the intersection between F1 and F2 curves (for t = 2 h and 4 h) occurs at more than 190 days and with the curve F2 (for t = 20 h) at approximately 390 days.

Figure 5 shows the failure probability dependence of testing parameters, using the formulae F1 and F3, for the same component (pump P2). This figure is not so suggestive/ appropriate to be used for optimization of testing parameters.

However, the analysis at the component level indicates that the greatest sensitivity of failure probability with the testing parameters is more relevant than the analysis at the system level.

Such type of analyses constitutes the first step in optimization of the testing parameters.

Accident/Event sequence level analysis

In the CPSE B+ study the late core damage accident was analytically estimated as having an occurrence frequency of 7.4E-05 events/ year. From the accident sequences for 'Small LOCA' (occurrence frequency 6.23E-06 events/year), the dominant sequence has a contribution of 8.6%.

Optimizing the test/repair/maintenance and operation parameters could be accomplished

Table 3. The processed results for test optimization. ESCS System – Component level analysis (standby pump)

No.	T [days]	F1	F2 [t = 20 h]	F3 [t = 20h]	F2 [t = 4 h]	F3 [t = 4 h]	F2 [t = 2 h]	F3 [t = 2 h]
1	400	0.016685	0.01875	0.0021	0.0171	0.00042	0.0168	0.00021
2	350	0.01463	0.016995	0.0024	0.0151	0.00048	0.0076	0.00024
3	300	0.0119	0.01532	0.00278	0.0128	0.00056	0.0059	0.00028
4	250	0.0092	0.013885	0.0034	0.0102	0.0067	0.00052	0.00034
5	200	0.008466	0.01263	0.0042	0.0086	0.00083	0.0042	0.00042
6	150	0.06411	0.011975	0.0056	0.007524	0.00112	0.003762	0.00056

taking into account the effects on the accident/event sequence. So could taking a certain accident sequence, from an event tree, and discussing it related to the above parameters. The test/repair/maintenance and operation parameters are related at the system level. The analyses could be applied for a certain initiating event (LOCA 2) that has a certain occurrence frequency and a certain core damage failure contribution.

Safety system analyzed

The "*Emergency Core Cooling System*" (ECCS—a safety system) was considered for analysis. In the case of a small LOCA initiating event occurrence, the ECCS doesn't supply water make-up in Primary Heat Transport System (PHTS).

The processed results for test optimization are presented in Table 4.

Figure 6 shows the (F1) curves that give the failure probability with the test interval and test duration (F2). It could be observed the convergence of the F1 and F2 curves for test intervals between 10–30 days.

Table 4. The processed results for test optimization. ECCS System – Safety system analysis.

No.	T [days]	F1	F2 [t = 20 h]	F3 [t = 20 h]	F2 [t = 4 h]	F3 [t = 4 h]	F2 [t = 2 h]	F3 [t = 2 h]
1	400	0.426	0.4272	0.0021	0.4264	0.00042	0.4263	0.00021
2	350	0.382	0.3828	0.0024	0.3817	0.00048	0.3816	0.00024
3	300	0.3342	0.3358	0.00278	0.3345	0.00056	0.3343	0.00028
4	250	0.2844	0.2867	0.0034	0.285	0.00067	0.2846	0.00034
5	200	0.2324	0.2355	0.0042	0.233	0.00083	0.233	0.00042
6	150	0.1783	0.183	0.0056	0.18	0.00112	0.1787	0.00056
7	120	0.145	0.151	0.00695	0.15	0.0014	0.1454	0.0007
8	100	0.1222	0.1295	0.0084	0.124	0.0017	0.123	0.00085
9	75	0.0935	0.1036	0.0112	0.0956	0.0023	0.0945	0.00115
10	56	0.0714	0.0853	0.015	0.07413	0.003	0.07727	0.0015
11	25	0.03478	0.06758	0.034	0.04125	0.0067	0.038	0.00335
12	10	0.0169	0.0995	0.084	0.0336	0.017	0.02523	0.0085
13	1	0.0061	0.841	0.84	0.175	0.17	0.0896	0.084

In Figure 7 are represented the curves that give the variation of (F1) failure probability with test interval and test duration (F3). It could be observed the intersection of F1 and F3 curves for the test interval is between 7 and 25 days.

4. Conclusions

The paper presents elements of the methodology and techniques to be performed in order to establish and optimize the parameters for test, repair and operation of testable systems or components. The method tried to establish the appropriate type of analysis that is suitable for establishment of the optimum test interval and test duration. Many times the test interval is selected on design or manufacturer specifications. Supplementary to these specifications, the results of an analytical process of test optimization could help the activities to establish the test/repair parameters. The method is not exclusive; this method could be an important factor in decisions related to test activities.

The methodology for analysis of risk related optimization of the test activities/parameters is presented in the paper above. Sometimes the failure probability of the system/component and the unavailability of system/component due to testing could be compared with a target value. This target value is not always known and credible. The results presented above are designated to state the method of test optimization. Detailed analysis is necessary if there are requirements to exactly determine/ optimize the testing/repair/operation parameters.

The research activities will continue in order to establish the appropriate models and formulas for optimizing and assessment of the testing/repair/maintenance/operating parameters (activities). These activities will constitute the basis for cooperation with the Utility "CNE – Prod" in order to use the results for the Cernavoda–1 NPP overall power production activities. The final goal of such research activities will be to determine the SSCs for the plant and for these SSCs to increase the reliability/availability factors.

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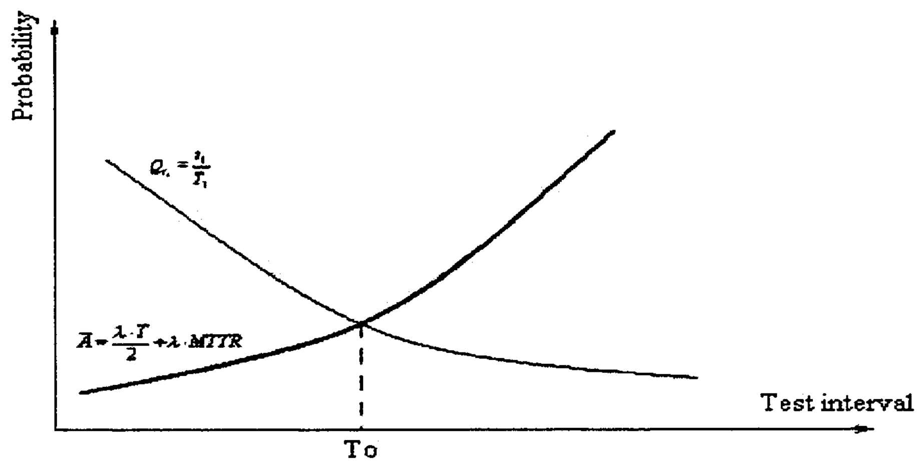

Figure 1. Optimization of test interval

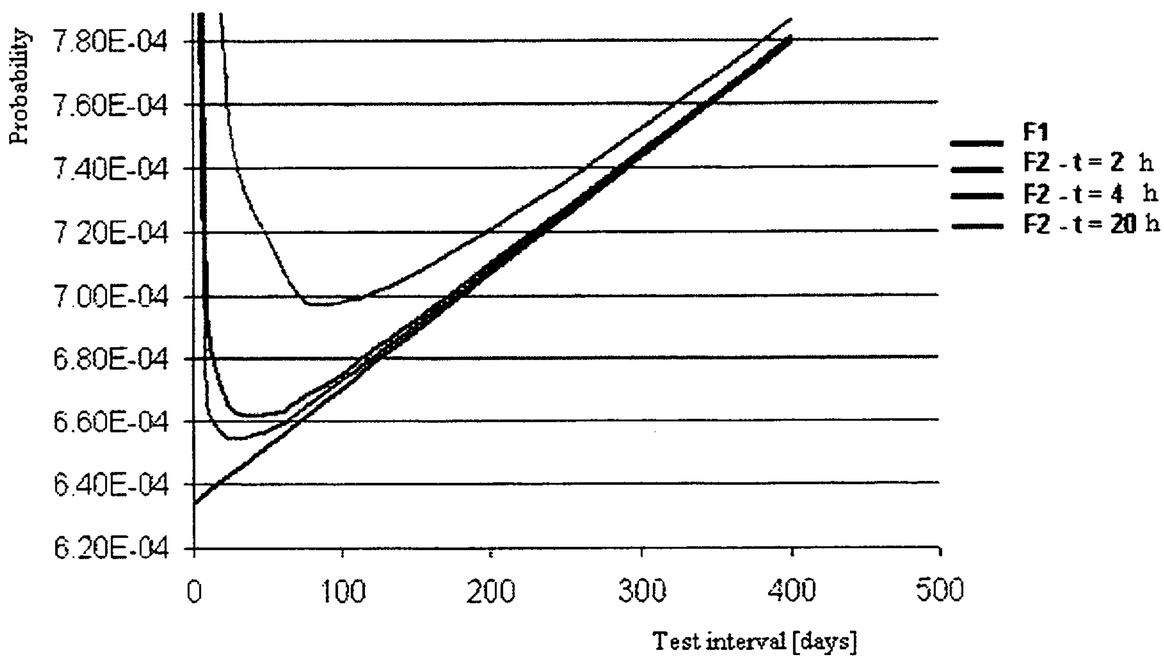

Figure 2. Testing parameters optimization. System ESCS level analysis. Formulae F1 – F2

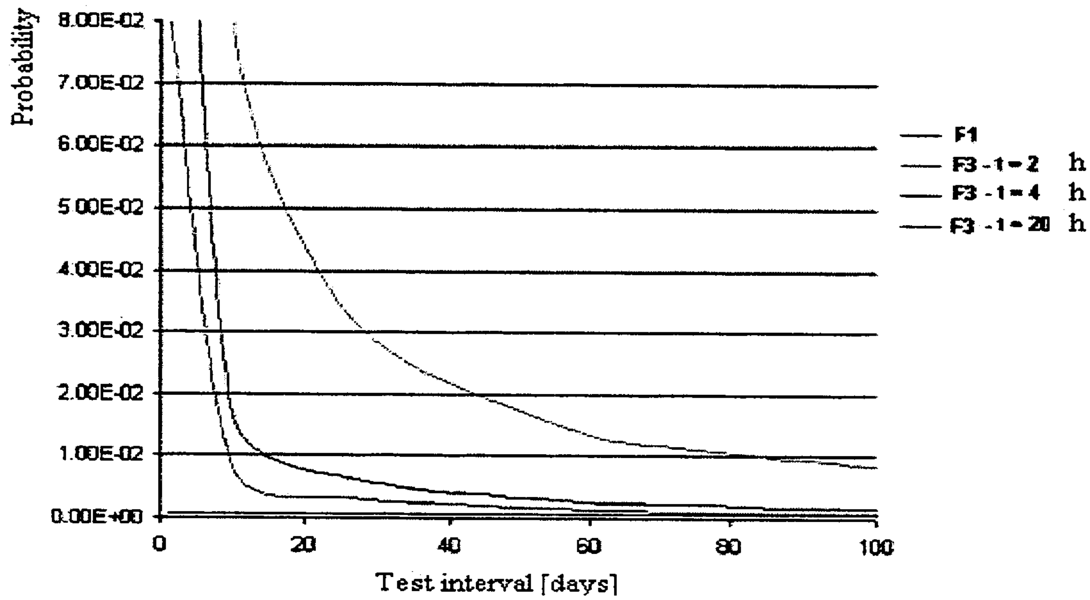

Figure 3. Testing parameters optimization. System ESCS level analysis. Formulae F1 – F3

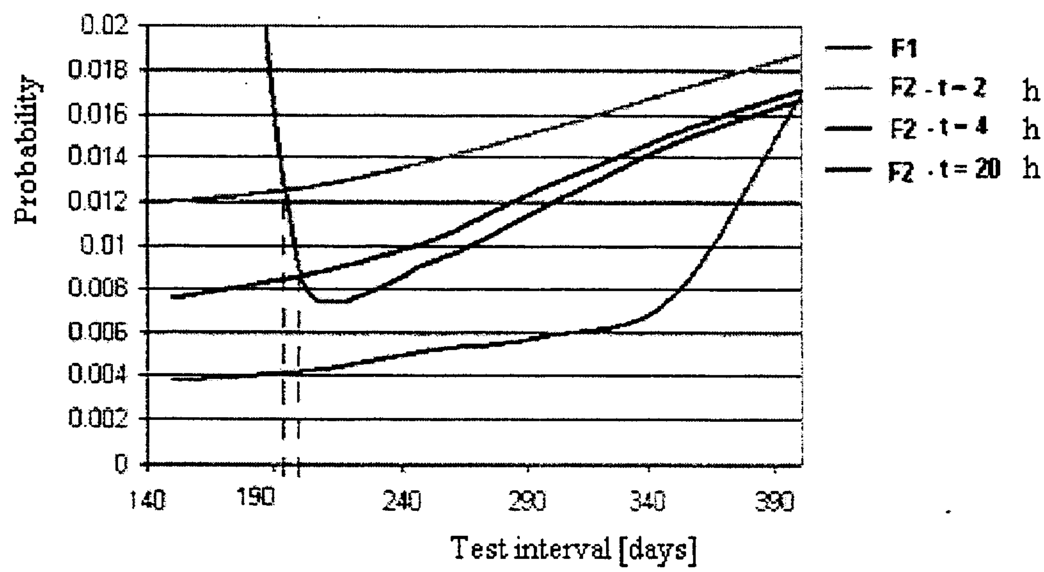

Figure 4. Testing parameters optimization. System ESCS component level analysis (formulae F1 - F2)

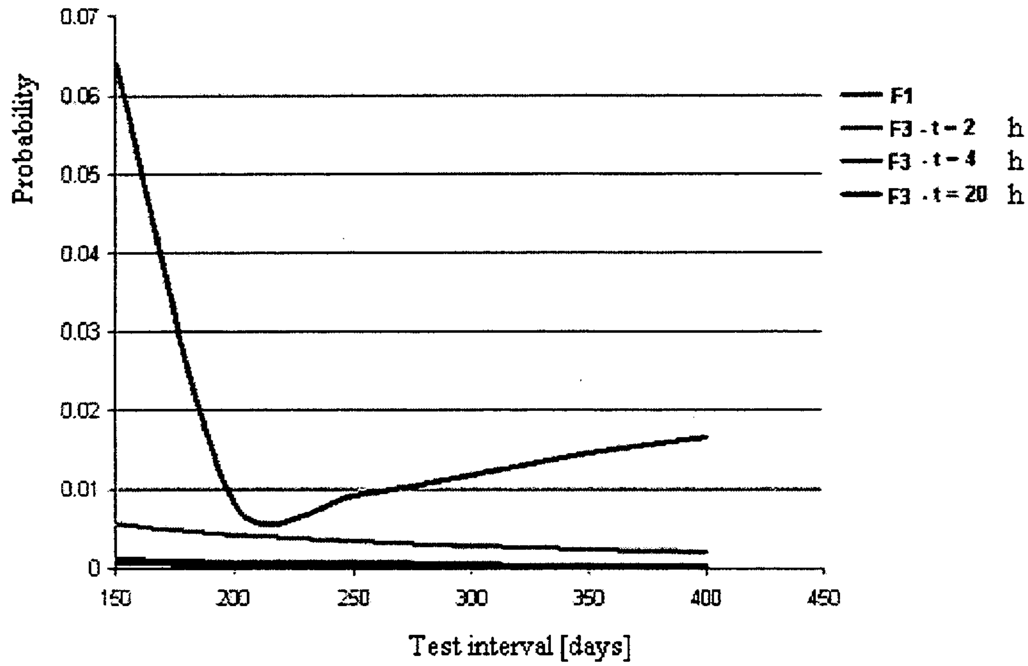

Figure 5. Testing parameters optimization. System ESCS component level analysis (formulae F1 – F3)

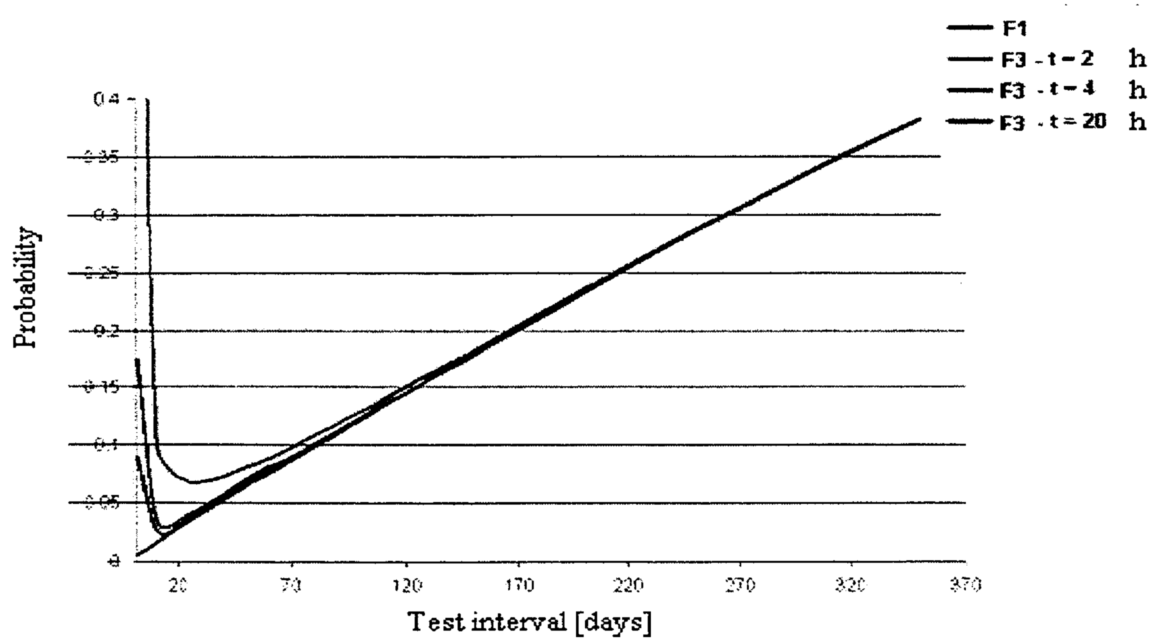

Figure 6. Testing parameters optimization. System ECCS level analysis. Formulae F1 – F2

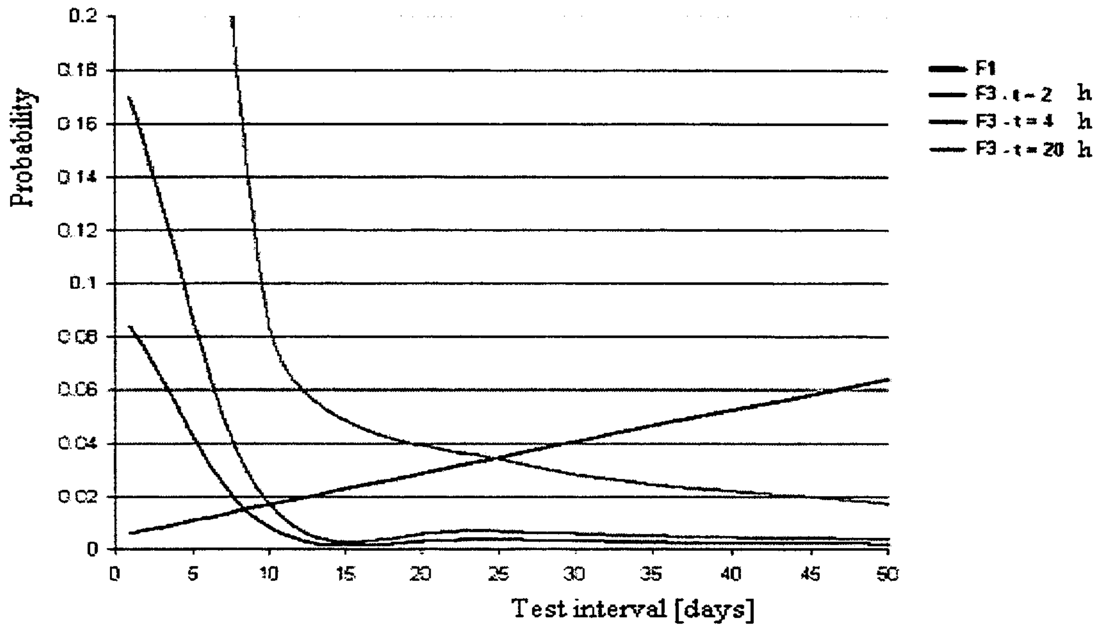

Figure 7. Testing parameters optimisation. System ECCS level analysis. Formulae F1 – F3

Risk-Informed In-Service Testing at San Onofre Nuclear Generating Station (SONGS)

Maureen K. Coveney and William J. Parkinson, Data Systems and Solutions
Darryl L. Barney, Southern California Edison

Abstract

The risk-informed inservice testing (RI-IST) implementation methodology used at the San Onofre Nuclear Generating Station (SONGS) has been developed into a standardized project and submittal adaptable to a variety of probabilistic risk assessment (PRA) types and IST program bases. It has been successfully applied to both pressurized water reactors (PWRs) and boiling water reactors (BWRs) in the U.S. and internationally, and has recently been adapted for use in a submittal performed in conjunction with a 10 CFR 50 Option 3 exempt components regulatory submittal, as well as a motor-operated valve (MOV) risk-informed testing submittal. The standard submittal facilitates regulatory review and, as evidenced by the safety evaluation reports (SERs) for Comanche Peak and SONGS, considers the issues identified in the Regulatory Guides and relevant NRC-approved ASME Code Cases. Equally important, the standard submittal and project have been demonstrated to reduce the costs of an RI-IST project and submittal, and of regulatory review time. The results of an RI-IST program would be expected to reduce plant risk, improve safety, improve overall pump and valve reliability, reduce the costs of implementing an RI-IST program, and reduce burdens in on-line and outage maintenance programs. The payback time for the program is estimated to be one or two refueling cycles. Finally, the lessons learned from RI-IST program implementations at Comanche Peak and at San Onofre are now becoming available to further improve the effectiveness of the approach and the resulting RI-IST program definitions and program implementations.

Introduction

In December of 1998, Southern California Edison (SCE) submitted a risk-informed Inservice Testing (RI-IST) program for SCE's San Onofre Nuclear Generating Station (SONGS) to the U.S. Nuclear Regulatory Commission (NRC) for their consideration. In March of 2000, the submittal was approved with minor changes.

The program outline conformed with the NRC-approved methods and Regulatory Guides.[1,2] This program bears close resemblance to that employed by the NRC-approved RI-IST pilot program at Comanche Peak, and further incorporates insights from their RI-IST Safety Evaluation Report (SER).[3] In addition, SCE has constructed the RI-IST program consistent with key elements of the NRC-approved ASME Code Cases, including:

- The performance of suggested risk assessment sensitivity studies and the incorporation of expert panel guidance, consistent with the component categorization Code Case, OMN-3, and

- The incorporation of staggered testing and implementation guidance, consistent with the component Code Case, OMN-1.

Given the reliance of risk-informed initiatives on insights derived from the Probabilistic Risk Assessment (PRA), the risk assessment satisfies industry standards associated with PRA. Enhancements to the PRA were made in support of the RI-IST and other plant applications, including use of the latest common cause failure (CCF) method and data, as well as enhanced modeling of external event initiators. The inclusion of IST program effects on cumulative plant risk was comprehensive, as a total quantitative estimate of risk was produced considering both average and dynamic plant models. This quantitative evaluation of key RI-IST program elements included the effects of compensatory measures, the influence of staggered testing on CCF, and the adverse effect of some ISTs on risk.

A key element of the RI-IST program was the Integrated Decision-making Process (IDP). SCE's IDP ensured that key safety principles such as defense-in-depth and safety margins were maintained. The process considered relevant component-specific information, including design basis safety functions, PRA risk importance, and a detailed analysis of component corrective maintenance history. Therefore, the Integrated Decision-making Process assured a detailed evaluation and Panel approval of component categorization results and supporting studies.

Further, insights from the Integrated Decision-making Process supported the conclusion that several safety enhancements to a plant IST program could be derived, both directly and indirectly, by implementing the results of the probabilistic and deterministic approach presented in SONGS' regulatory submittal. These safety benefits were treated both quantitatively and qualitatively, providing a reasonable and justifiable basis for implementing the RI-IST program.

Background

The intent of current inservice testing (IST) programs is to include all active, safety-related pumps and valves that are credited in the plant design basis safety analysis. In general, the IST equipment lists are developed by review of plant drawings showing ASME Code Class 1, 2, and 3 classification boundaries. All components within the boundaries are then reviewed to determine whether or not they have been credited with an active safety function under the plant licensing basis. The Updated Final Safety Analysis Report (UFSAR) analyses and other design basis documentation provide the primary bases for these determinations.

After the publication of its policy statement[4] on the use of PRA in nuclear regulatory activities, the Commission directed the NRC staff to develop regulatory guidance that incorporates risk insights. Concurrently, industry risk-informed pilot projects explored the process for supplementing traditional engineering approaches in reactor regulation with probabilistic information. This effort has culminated in several relevant and extremely significant regulatory advances in the area of risk-informed applications:

- Issuance of Regulatory Guide (RG) 1.174[1] and companion regulatory guidance (including RG 1.175[2]), which provide the regulatory framework to fashion an inservice testing program that focuses resources on risk significant pumps and valves,

- NRC acceptance of Texas Utilities' (TXU) Comanche Peak Steam Electric

Station (CPSES) relief request,[3] one of the industry risk-informed pilot projects,

- NRC acceptance of SCE's risk-informed Technical Specifications amendments requests, one of the industry risk-informed pilot projects,[5,6] and
- NRC acceptance of SCE's IST relief request.[7]

As has been demonstrated by both the CPSES pilot project and the SONGS RI-IST project, improvements to IST programs using a risk-informed approach can reduce operating costs while maintaining a high level of plant safety. Possible savings from improved IST programs include reduced costs associated with performing ISTs, such as:

- Time required to perform the tests and analyze results;
- Costs of specialized test equipment;
- Effects on critical path outage duration; and
- Radiation exposure.

For these reasons, it is advantageous for utilities to pursue IST program improvements. The impact of changes on plant safety is of primary interest and is the controlling factor in implementing such changes. However, changes that negligibly affect plant safety should not be ruled out, especially if such changes can lead to significant plant performance improvements in other areas.

Project Scope

The scope of this project was to build an RI-IST program for SONGS, one which optimizes safety benefits in ensuring pump and valve performance. The project applied a risk-informed approach for performing a comprehensive IST program review and for proposing program enhancements.

The principle results of the project were recommendations for adjustments to test frequency intervals for a large percentage of IST components. Thus, the scope of the effort did not aim to reduce the number of components within the scope of an IST program. Instead, this project focused on optimizing the overall component test schedule by applying resources commensurate with the component safety function, performance, and relative risk. In this study, all components within the scope of the IST program were examined. However, only those determined to be less safety significant were considered for Code relief. Component experts reviewed the more safety significant components to ensure that the appropriate tests were identified for their respective failure modes.

Project Approach

The SCE risk-informed IST project was developed and implemented by the SONGS Station Technical group with PRA support from the Nuclear Safety Group and was guided by a cross-functional plant Expert Panel as well as industry experts who participated in the TXU risk-informed pilot project. The SCE project employed a method that blended probabilistic and traditional engineering insights to identify opportunities to reduce those IST-related regulatory requirements and commitments that require significant resources to comply with and/or implement, but contribute insignificantly to safe and reliable operation. Using risk-informed technologies, the project determined the safety significance of IST components, as well as components not in the IST program. Then the project applied a combination of deterministic and risk-informed methods to determine testing intervals and compensatory measures that correspond to each component's safety significance. The results of this project provide the basis for this request to implement an alternate testing strategy to the NRC.

Overall project objectives and milestones were established by key risk-informed IST project members. The project was divided into the five major tasks listed below:

1. Component Function Evaluation
2. Component Corrective Maintenance Evaluation
3. Calculation of Risk Measures Using the SONGS PRA
4. Component Risk Categorization by Expert Panel
5. Cumulative Risk Evaluation Using the SONGS PRA

The component function evaluation established the design basis safety functions of IST components and related these functions to component failure modes modeled by the PRA. Modeling implications were also identified, including the component or system-level assumptions that affect the level of credit the PRA affords an IST component's safety function. The component corrective maintenance evaluation validated the basis for the PRA reliability assessment and demonstrated how it compared to generic and plant-specific experience. It also established a baseline for future monitoring that is needed to compensate for some of the components whose testing frequency requirements are reduced.

The PRA was then used in a variety of ways to evaluate the safety significance of components and their functions. Sensitivity studies demonstrated the robustness of the methods and the results. This process was followed by an Expert Panel review and validation of the risk categorization.

The Expert Panel considered and ultimately validated the results of all work activities and studies performed by the IST project members. The Expert Panel consisted of members with expertise in the areas of power plant operations, plant maintenance, PRA and nuclear safety analysis, reliability engineering, component and systems engineering, design engineering, and Inservice Testing (including ASME B&PV Code Section XI and ASME Code Cases). Representatives from licensing and engineering occasionally participated in meetings when requested by the Expert Panel. In addition to ensuring an integrated effort through active technology transfer, the Expert Panel served as the central point of decision-making for major technical issues and offered guidance to risk-informed IST project members in performing their work. Further, due to common membership of several members on the risk-informed IST Expert Panel and the Maintenance Rule Expert Panel, consistency in decision bases was assured. It was concluded that the strength of this risk-informed IST program and the integrity of its results lie both in the robustness of the methodology and in the quality and work of the Expert Panel. The Expert Panel process was implemented according to clear guidelines and operated directly from documentation produced in earlier tasks.

All project tasks were conducted with reproducibility and retrievability in mind. The project deliverables—including tables of IST functions, PRA functions, PRA risk measures, component ranking outcomes, component functional failures, Expert Panel decision bases, valve groups, test interval information, and monitoring requirements—are housed in a database from which the IST coordinator may administer the risk-informed IST program. In addition, all Expert Panel judgments have been transcribed and indexed to ensure that component information is traceable and retrievable.

Conformance with Key Safety Principles

The proposed RI-IST program meets all acceptance criteria and guidance specified in RG 1.174 and RG 1.175, including the four element approach to evaluating proposed changes in Section 2 of RG 1.174. These acceptance criteria include the five principles of integrated decision-making discussed in Figure 1 of RG 1.174, such as maintaining defense-in-depth and safety margins. In addition, upon implementation of the RI-IST program, several safety benefits to the plant IST program can be derived both directly and indirectly.

Direct Safety Enhancements

Possibly the most important safety benefit results from reducing the frequency of a few ISTs which place the plant at greater risk because of their current IST interval. The quarterly ISTs on the low pressure safety injection (LPSI) discharge valves increase the frequency of interfacing systems loss-of-coolant-accident (ISLOCA) conditions and thereby the frequency of a large early release. In relaxing the test interval for these components, SONGS realizes a significant and quantifiable safety benefit.

In general, relaxing IST intervals for many lower priority components allows SCE to focus greater attention and resources on high priority IST components. A resource reallocation of this nature could translate into many direct safety enhancements. Test requirements associated with the high priority group of IST components are expected to be more rigorous and demanding in nature than for the other groups. These requirements provide added assurance that any problems that may impact the functionality of the components will be identified and resolved expeditiously. Second, the resulting risk-informed IST program considers whether some risk-significant components that are outside the scope of ASME Code Classes 1, 2, and 3 should be added to the IST program to improve safety. Finally, because extensive testing can have adverse safety and operational consequences, reduction of testing may reduce component wear-out and operator burden. These changes are expected to improve safety.

Indirect Safety Enhancements

There are other indirect safety benefits to this approach that are as important. Risk-informed prioritization efforts identify the safety-significant IST components and the impact of their potential failures 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 component. Such information is valuable because it relates the performance of the IST component 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.

RI-IST Project Results

Component categorization of IST valves and pumps yielded the following results:

Risk Ranking	Percentage of Components (Units 2 and 3)
High safety-significant components (HSSCs)	15.8% (144 components)
Low safety-significant components (LSSCs)	83.1% (757 components)

According to the above table, 83.1% of the components ranked were eligible for interval extension.

Upon completion of the implementation effort, safety enhancements are expected from focusing resources on HSSCs and reducing the testing frequency on LSSCs, as discussed above. Because extensive testing on LSSCs may adversely impact safety, reduction of testing should reduce component wear-out, operator burden, system unavailability, cost of testing, and radiation exposure. Reduced testing could also achieve an optimum balance between the positive impacts of testing and the negative effects of removing equipment from service and entering a less than optimum plant configuration, that have the potential to result in valve misalignments. Focusing of resources on HSSCs includes improved testing of emergency chillers and enhanced testing of selected high importance check valves (use of nonintrusive check valve diagnostics, including trending) and pumps (including spectral analysis, and thermography) beyond code testing requirements. The cumulative effects from reduced testing of LSSCs and enhanced testing of selected HSSCs are tangible risk benefits that were not used in quantifying the risk impact of the risk-informed IST program.

Given the relaxation of test intervals, the addition of components to the program and the non-quantified tangible risk benefits, the impact of the proposed RI-IST program will be either risk beneficial or, at the very least, risk neutral.

Implementation

Several insights have resulted from the effort to implement the SONGS RI-IST program upon receipt of the NRC's SER. These insights could benefit other utilities seeking to implement an RI-IST program of this nature.

Early in the RI-IST evaluation process, project members tried to facilitate the NRC's review of the RI-IST by excluding SONGS Technical Specifications changes from the RI-IST process. In retrospect, this decision may have resulted in a reduction of program benefits. For instance, the following areas of SONGS Technical Specifications affect how the RI-IST program will be implemented:

- The post-accident monitoring specification requires calibration of position indication at a hard two-year interval. To achieve optimum benefits, the specification should read, "the frequency [shall be] per the IST program." This would affect 230 of 402 position indication tests at SONGS.
- Per specification, the Engineered Safety Feature time response testing on valves must occur at a two-year interval. As above, the specification should be "per the IST program," which would affect 225 of 668 timed stroke tests (open and /or closed).

At the time of the RI-IST submittal, motor-operated valve (MOV) diagnostic testing involved intrusive, "at-the-valve" testing. Subsequent to the RI-IST program approval by the NRC, SONGS began to explore remote condition monitoring using motor torque analysis derived from motor voltage and current (Crane MOVATS MC²). Testing MOVs from the breaker represents a significant resource benefit. However, tests performed at the breaker do not require either local observation or manual operation of the valve. Stated differently, where "at-the-valve" testing allows scheduling all required tests at one time, a transition to remote torque analysis significantly complicates test scheduling since manual strokes and position indication tests must be scheduled separately. In addition, this new technology impacts the SONGS at-power testing model. In short, this new technology effectively eliminates the requirement for physical testing, but it also results in scheduling and administration challenges.

Integrating the requirements of a stagger testing model with the SONGS 12-week site planning schedule introduces additional scheduling challenges. The SONGS IST stagger testing model is based on traditional Technical Specification definitions, which use a two-year fundamental interval for stagger testing. The two-year basis uses a 92-day quarter. The maintenance work planning department uses a 12-week (or 84-day) schedule. Currently, blending the two schemes is a manual process that involves comparing due dates and expiration dates to ensure continued compliance. As the site moves toward implementing the INPO AP-913 reliability process, SONGS will build or upgrade the tools associated with managing complex scheduling tasks, which will likely include development of a common automated scheduling tool to integrate activities from a broad spectrum of site departments.

From the standpoint of managing changes to SONGS procedures, implementation of RI-IST has had no appreciable impact. In the future, SONGS plans to alter procedures to accommodate valve grouping, as well as online-offline classifications.

Despite the program administration challenges discussed herein, SONGS has experienced a substantial benefit to implementation of RI-IST program, as far fewer inservice tests are performed, allowing personnel to focus resources on components with higher safety significance. For both units in a 24-month cycle, shutdown inservice tests dropped from approximately 1500 down to 800 with no significant effect on plant safety. The benefit experienced at SONGS is consistent with the initial experience in implementing RI-IST program at Comanche Peak, for which 1700 tests were reduced to 900 during one 18-month refueling cycle.

Cost Benefit

The costs are derived from actual implementation experience.

Expected Costs. SCE has spent or is spending approximately $466K to implement its RI-IST program.

A utility can expect to exploit lessons learned and past experience in the following areas:

- Performing a failure history analysis,
- Determining new program elements (determination of failure rates, selection of compensatory measures, identification and evaluation of monitoring requirements and the corrective action program),
- Generating Expert Panel work packages,
- Facilitating an Expert Panel,
- Recording and maintaining all RI-IST information in a RI-IST database, and
- Producing a regulatory submittal.

For these key areas, a utility can expect to realize cost reduction because methods are well-understood and procedures or methodologies similar to those developed at SCE will apply at a follow-on plant.

Conversely, a utility can expect to incur costs greater than those incurred by SCE depending on the quality of the PRA and the availability of information for the IST program.

Expected Savings. Annual cost savings expected from an RI-IST implementation at a typical utility is expected to be $230K, derived from the following tangible cost centers:

- Reduction in tests (due to rescheduling roughly 2000 tests, 10% of which are cold shutdown/refueling tests and the remaining 90% are quarterly tests)
- Reduced time required to set up for tests
- Pre-processing and post-processing of test results
- Reduced work planning time
- Reduced critical path time
- Reduced dose

In addition to the above cost areas, implementation of the RI-IST will afford cost reduction in the following areas, which are more difficult to quantify:

- Focusing a majority of efforts on highly safety significant components
- Facilitating a train-based outage strategy
- Potentially affecting near critical path items in the outage
- Reducing operator burden (i.e., reducing the opportunity for errors)
- Reducing the effect of testing on test equipment, waste produced, consumables required, etc.

Based on the one-time highest potential cost of $466K and the expected cost savings from tangible cost centers, the program will pay for itself within 1.3 years. After that time, a utility will realize approximately $200k in cost savings per year, in addition to dose reduction and decreased operator burden.

Conclusion

The RI-IST implementation methodology used at SONGS has been developed into a standardized project and submittal adaptable to a variety of PRA types and IST program bases. It has been successfully applied to both pressurized water reactors (PWRs) and boiling water reactors (BWRs) in the U.S. and internationally, and has recently been adapted for use in a submittal performed in conjunction with a 10 CFR 50 Option 3 exempt components regulatory submittal, as well as an MOV risk-informed testing submittal. The standard submittal facilitates regulatory review and, as evidenced by the SERs for Comanche Peak and SONGS, considers the issues identified in the Regulatory Guides and relevant NRC-approved ASME Code Cases. Equally important, the standard submittal and project have been demonstrated to reduce the costs of an RI-IST project and submittal and of regulatory review time.

The results of an RI-IST program would be expected to reduce plant risk, improve safety, improve overall pump and valve reliability, reduce the costs of implementing an RI-IST program, and reduce burdens in on-line and outage maintenance programs. The payback time for the program is estimated to be one or two refueling cycles.

Finally, the lessons learned from RI-IST program implementations at Comanche Peak and at San Onofre are now becoming available to further improve the effectiveness of the approach and the resulting RI-IST program definitions and program implementations.

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Special Treatment of Low Safety Significant (Exempt) Pumps and Valves at STP

Glen E. Schinzel
South Texas Nuclear Project

Abstract

The South Texas Project (STP), located on the Texas Gulf Coast about 85 miles southwest of Houston, is a twin-unit nuclear facility with Pressurized Water Reactors rated at 1250 MWe each. STP has been a leader in risk-informed applications since 1982. Recently, STP submitted a broad-based exemption request to the Nuclear Regulatory Commission (NRC) to remove the least important safety-related components from the scope of the special treatment requirements required by Title 10, Part 50 of the *Code of Federal Regulations* (10 CFR Part 50). The STP approach was termed a 'proto-type pilot' of the NRC's SECY 98-300, Option 2 approach. The exemption request's goal was to enhance overall nuclear safety while reducing costs associated with power generation. Upon grant of the exemption, this goal would be achieved by focusing resources and attention on the safety-significant components while reducing burdensome treatment on those components that were determined not to be safety significant.

The approach to the exemption request required component importance to be determined using both probabilistic and deterministic insights. To date, 50,000 components in 40 systems have been 'categorized' using this blended approach with only 10% of the components determined to be safety significant. Once the relative importance of the safety-related components is determined, the treatment for the least important components can then be adjusted based upon their safety significance.

The STP exemption request focused on the regulations in 10 CFR Parts 21, 50, and 100, including 10 CFR50.50.55a(f) addressing Inservice Testing (IST). STP had worked closely with the NRC since 1998 to gain approval of this important ground-breaking exemption. The approval of the exemption was ultimately granted on August 3, 2001.

Since approval of the exemption, STP has begun the cautious and deliberate implementation of the exemption allowances. Implementation of these allowances will occur over a number of years, with the most immediate IST focus occurring in the areas of Motor Operated Valve (MOV) stroke time testing, testing scope reductions for check valves and pumps, and in reduced relief valve testing per the 1987 O&M Code requirements. A significant benefit of the exemption approach (Option 2) is that the low safety significant components are removed from the scope of regulatory testing, and the regulated treatment is replaced by a commercial approach that provides sufficient confidence that the components will perform their safety

function under design-basis conditions.
The exemption approach (Option 2) is also different than the Risk-Informed IST approach in that with RI-IST, the components remain within the scope of a regulated treatment program.

Additional exemption implementation activities are also occurring at STP in the areas of Local Leak-Rate Testing, Parts Procurement, Maintenance Rule, and Maintenance-related activities.

Risk Informed-Inservice Testing at Comanche Peak Steam Electric Station

Gary McGee
TXU Energy

Abstract

The risk informed method has been endorsed by the NRC as a way for licensees to reduce burden without sacrificing safety. Risk informed-inservice testing (RI-IST) was implemented at Comanche Peak Steam Electric Station (CPSES) on December 4, 2000. TXU Energy at Comanche Peak became the first pilot program to fully implement an RI-IST program. This paper will cover three areas:

1) RI-IST project development,
2) RI-IST project implementation, and
3) RI-IST project performance for the past 18 months, including comparison to the old IST program.

The RI-IST project was started in 1998 as a continuation of the Maintenance Rule effort. The RI-IST program was designed to be consistent with the Maintenance Rule and other industry risk-based programs. Because this initiative was to reduce existing regulatory burden, the methodology applied risk measures in a manner intended to maintain or improve plant safety. The approach taken included four steps. First, risk importance was determined. This was based on the results of the Individual Plant Examination (IPE) and Individual Plan Examination External Events (IPEEE), as well as risk insights during other plant operating modes. The importance of components in the IPE or IPEEE models and in the IST Plan was then evaluated. The next step addressed the adequacy of these models through a number of sensitivity analyses. The third step evaluated the cumulative impact of low risk significant components on plant risk if their inservice test intervals were extended. Finally, the last step was to review the process and results with an expert panel that was knowledgeable concerning plant risk, design, operations, and performance. From the expert panel, components were segregated into high safety significant components (HSSC) and low safety significant components (LSSC).

Once all IST components were segregated into HSSC and LSSC components, then test frequencies were reassigned. Any non-IST component that was rated as HSSC was added to the IST Plan. For an HSSC, test frequencies are as specified by OM-1987, Part 6 and 10. For an LSSC, frequencies were reduced from quarterly to 18 month and to as much as six years. All LSSC were grouped and were scheduled to be tested on a staggered test basis.

The major problem areas with developing RI-IST were threefold:

1) When rewriting the IST Plan, the IST Engineer discovered how difficult it was to group the components logically and usefully.

2) Operations had to rewrite almost every test procedure due to new groupings of valves and new frequencies of tests.

3) The Work Control organization had to coordinate closely with Operations and the IST Engineer to rewrite nearly all IST surveillance work orders and ensure that no surveillance tests were missed.

When the RI-IST Program was implemented the transition was almost transparent. Nothing changed except that motor operated valves were no longer stroke time tested and much fewer inservice tests were performed.

1 Unit: 18-month cycle 1700 tests vs. 900 tests

Now that the program is implemented, it is proving to be far superior to the old IST Program. Initially, it was very hard to see the results of RI-IST since the savings were "soft-money" versus "hard-money"; i.e., you need the same amount of people and equipment to operate each program. The savings show up in other ways over time. The largest savings is an estimated four days of Refuel Outage Critical Path time at approximately $600,000 per day. And more importantly, this savings in time has made mode changes much easier to manage and schedule. Another example is a Steam Generator Blowdown Waterhammer Prevention Mod that was cancelled because it was no longer needed. The cause of the waterhammer was the opening of an air-operated valve during that valve's quarterly stroke time test. With RI-IST, that valve is tested every six years and can be tested in a mode of operation that will not cause water hammer. The savings on the modification is $250,000. Another example is that Comanche Peak will be updating to the 1998 ASME OM Code. One of the requirements of this Code is a requirement to exercise test check valves fully in both directions regardless of safety function position. This will require extensive rewrite of current procedures and much greater use of acoustic emissions (AE) testing. This requires Operations, QC, Maintenance and Engineering personnel and will be significantly time consuming. The greatest majority of LSSC valves in the IST Plan are check valves. The long test intervals will mean considerable savings on manpower and cost, yet the staggered test basis will mean that the data will continue to flow at a meaningful rate to judge the performance of plant components. This ultimately achieves what the NRC and the Industry set out to accomplish, a program that optimizes resources for plant safety.