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ABSTRACT

The 1990 Symposium on Inservice Testing of Pumps and Valves, jointly sponsored by the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the Nuclear Regulatory Commission, provided a forum for the discussion of current programs and methods for inservice testing at nuclear power plants. The symposium also provided an opportunity to discuss the need to improve inservice testing in order to ensure the reliable performance of pumps and valves. The participation of industry representatives, regulators, and consultants resulted in the discussion of a broad spectrum of ideas and perspectives regarding the improvement of inservice testing of pumps and valves at nuclear power plants.

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ACKNOWLEDGMENTS

The editors acknowledge the dedication of the authors and panel members for their invaluable contribution to the success of the symposium. Special thanks is extended to Tad Marsh, the symposium moderator; to Robert Dick, Ed Jordan and Larry Chockie who made the symposium opening addresses; and to Commissioner Kenneth Rodgers who gave the keynote address. Appreciation is expressed to the symposium organizing committee: Kevin Ennis, John Zudans, Robert Masterson, Ted Sullivan, Horace Shaw, and Joel Page. Gratitude is expressed to Gail Marcus and Tom Scarbrough for their special assistance in tasks crucial to the success of the symposium.

DISCLAIMER AND EDITORIAL COMMENT

Statements and opinions advanced in papers presented at the ASME/NRC Symposium on Inservice Testing of Pumps and Valves are to be understood as individual expressions of their authors and not those of the American Society of Mechanical Engineers nor the U.S. Nuclear Regulatory Commission.

The papers have been unaltered other than to put them into a standard format and to resize and upgrade the legibility of some graphics. The transcription of the question and answer periods following each paper presentation, and the panel discussions following each presentation section, have been edited for somewhat better economy of flow, and the names of specific panel members and attendees (and references to such) have been dropped from the questions and answers.

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PRESESSION ADDRESSES

PROCEEDINGS OF THE SYMPOSIUM ON INSERVICE TESTING OF PUMPS AND VALVES HELD AUGUST 1–3, 1989, AT THE HYATT REGENCY HOTEL IN WASHINGTON D.C.

GREETINGS

TAD MARSH, SYMPOSIUM MODERATOR
CHIEF, MECHANICAL ENGINEERING BRANCH
U.S. NUCLEAR REGULATORY COMMISSION

Good morning. My name is Tad Marsh. I am the Chief of the Mechanical Engineering branch in the NRC and I am the symposium moderator. It is good to see so many of us here. I think it is an indication of the importance of the issue and it is also an indication of the need for a symposium of this sort.

The NRC is trying different ways of communication with the industry and this is just one different way, and if it is successful I think we should think about continuing this type of endeavor. It appears to be successful. We welcome your suggestions. If there are any things that we can do better, or differently, please let me or the ASME know at any point either during or after the conference.

I have got a few administrative details I would like to get to. If you wish to receive calls, or if you need to let your office know where you are, the phone number for the hotel is 737- 1234, and ask them to contact you at the NRC/ASME symposium at the Regency desk, and that will get you right out here.

Some of you are wondering, I am sure, what the little red apple means on your little name tags. The red apple means that you're okay for dinner tonight. Please be aware there's a cash bar this evening at 5:30 before the dinner.

If there's a problem, if you think that you have paid but you don't have a red apple, please talk to the desk and we will see if we can get that squared away. There aren't very many places available, if any, but if you do wish to go, contact the Regency desk.

You should all have cards—3 x 5 or slightly larger cards, various colors. No color-coding. Just make sure you have cards. The purpose of those cards is for you to write down your questions of either the speakers individually or of the panel sessions following each one of the speaker's presentations. Please put down on the cards your name, your company and your question. When you have a question, when you have your card filled out, raise your hand up and somebody from the ASME staff will pick up the card and bring it up to the front podium where the moderator for that session will hold the card until it's time for it to be answered. Please put down your name and affiliation. That's important, because we want to know from whom the questions came so if we need to get back to you, we can.

We also are taking minutes, as you can tell. The purpose for taking the minutes is we are going to publish a NUREG at the end of this symposium. The NUREG will have in it the papers, the questions and answers. It will have the discussion sessions. It will have summaries by the panel members at the end of the discussion.

The NUREG will include a list of attendees. I understand many have been asking for a list of attendees because there are so many, widely interested people. That will come out with the NUREG at the end of the symposium. We look for the NUREG to be published no later than six months after the symposium.

We are also going to have in the NUREG the speeches that are being given by us and also, this evening, by Commissioner Rogers.

You can tell by your flier the plan for the closing was to have John Basile, the Vice Chairman of the Subcommittee on Performance Testing and O&M, and myself give closing comments at the end of the symposium. We felt we wanted to change that. We wanted to bring in the various sessions in more detail, so we're going to ask that the session moderators participate in that closing session itself. So, rather than there being two people, there'll be six people up here at the end, each one of which will give their own perspective. So we wanted to bring the session perspectives in themselves, rather than just two people.

I want to spend just a minute talking about the papers, authors and the Organizing Committee. The Organizing Committee did a super job for this symposium. They had a tough job because they had to, on very short order, choose topics that were known to be of high interest to the industry, to the NRC, to the ASME, then choose authors that are known to be expert in those areas, then contact the authors, ask for abstracts, get the papers and turn it all around quickly.

Normally this type of symposium could take two years to get this kind of thing organized by going open paper discussion, submitting them, review and that type of thing. But there was a need — a pressing need to have a symposium of this sort within a year.

So we chose a way, and the certain way was to choose papers, authors and get the papers in quickly, but we realize that there are other papers that are just as interesting and just as technically important, and that's why we have an open session being offered tomorrow afternoon.

The views that are being expressed by the authors in these papers don't represent the views of the Organizing Committee, nor the ASME, nor of the NRC. It's important to realize that. These are topics that have been of known interest, but they haven't received any blessing, per se, from the various organizations.

Why are we having this symposium? Just briefly, I would like to characterize the past with respect to inservice testing as having too little communication between the industry, the NRC, the ASME and utilities.

The NRC on its part didn't clarify the IST requirements. We didn't give out generic guidance on what programs were to have in them nor on relief requests. We didn't give out guidance on program implementation. There was not enough dialogue between the industry itself, various successful ways of accomplishing the code requirements. In general, there just wasn't enough importance given to what we think now is an extremely important area.

We hope things are changing. They are changing, as evidenced by a couple of things. First, we have new code standards — ASME, OM6 and OM10. You're going to hear about those two code standards today.

Also, there is new generic communications. Generic Letter 89–04, Generic Letter 89–10, both of which relate to inservice testing. It's generic guidance that has been needed for a long time. The workshops themselves that we held on Letters 89–04 and that we will hold on Letters 89–10 are further evidence of the need to communicate generically what the NRC wants, needs, and expects—what the code requirements are all about.

We recognize that this symposium is further evidence, we think, of the need for generic communication and the need for expressing opinions and experiences to further put IST on a better footing.

In short, we want to understand the problems you've had in those areas so that new directions we may take are done properly through more efficient and effective inservice testing.

So, the overall purpose of this symposium is to provide a forum for the exchange of technical, policy, and regulatory information in a free and open atmosphere.

I would like to also — and this is a large crowd and there isn't a whole lot of time allowed for discussion after each paper, but to the extent possible, share your views, your experiences, your opinions, because when those become known, that will make a successful symposium, when we hear as much feedback as possible.

WELCOMING ADDRESS
ASME O&M STANDARD AND BPVC SECTION XI INTERFACE

ROBERT L. DICK, CHAIRMAN
BOARD OF NUCLEAR CODES AND STANDARD
(DUKE POWER COMPANY)

Good morning. On behalf of the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers, I am certainly pleased to welcome you to the ASME/NRC Symposium on the Inservice Testing of Pumps and Valves.

Also, I want to extend a special welcome to those of you who are here from the international community. We have with us today representatives from Canada, from England, from Germany, from Japan, and from Sweden. And for this symposium we have gathered about 315 attendees and 31 authors. These individuals represent utilities, service organizations, manufacturers and insurance and inspection agencies, and, of course, the regulatory authorities.

With the number of attendees and the diversity of interests that is represented, we have all of the ingredients that should be necessary to make this symposium a success.

About a year ago, the NRC asked our Board on Nuclear Codes and Standards to co-sponsor a symposium to discuss industry concerns on inservice testing. The ASME also viewed the symposium as an opportunity to present the new O&M Parts 6 and 10, which replaced Subsections IWP and IWV of Section XI.

Well, it took a lot of work from both the ASME and the NRC, but the result is 31 technical papers to be presented over the next few days.

The Symposium Committee consisted of NRC, ASME, and industry personnel. They picked 16 topics that they felt would be of interest to almost everyone in the nuclear industry, and this series of papers and the four panel discussions have been grouped into what is being called the Planned Session.

During the Planned Session a series of four papers will be presented on each of four general topics. These four topics are generic considerations for pump inservice testing programs, pump testing techniques and limitations, generic considerations for valve inservice testing programs, and valve testing techniques and limitations.

The planning committee chose the titles and authors to strike a balance between the regulatory and the industry perspectives. The planning committee realized that many topics of interest to large segments of the nuclear industry were not covered in the Planned Session and therefore issued a call for papers.

The result of that is the Open Session, and during the Open Session speakers will present technical papers on topics that are related to inservice testing. These papers were solicited from across the entire nuclear industry. The 15 papers to be presented are the response to the committee's call for papers.

The Open Session covers a variety of topics, and I encourage you to stop by the session on Wednesday to listen to any of the presentations that may be of interest to you.

All the papers being presented over the next few days stem at least in part from regulatory and code requirements. I am sure you all know that the code requirements for inservice testing of pumps and valves have changed over the past 19 years.

ASME's activity in this area started with Section XI of the boiler and pressure vessel code. Section XI, originally titled Rules for Inservice Inspection of Nuclear Reactor Cooling Systems, was first published in 1970.

In an effort to respond to the needs of the nuclear industry, a number of new articles were added to Section XI in the 1971 edition. Two of those new articles, IWP and IWV, contain requirements for the functional testing of pumps and valves.

Also during the early 1970s, we recognized the need to develop standards for more than just inservice inspection. So, in 1975 the Board on Nuclear Codes and Standards established a new main committee to develop a series of standards to help ensure the safe operation of nuclear plants.

That committee was the Operation and Maintenance (O&M) Committee. Its charter was to develop codes and standards for the safe and efficient operation and maintenance of nuclear power plants. These documents would focus on structural and functional adequacy.

Then in the 1980s, the Board on Nuclear Codes and Standards began an effort to simplify the codes and standards process, to consolidate related codes and standards and to address the growing needs of the nuclear industry.

The board directed its O&M Committee to develop inservice testing requirements for pumps and valves based on the existing requirements in Subsections IWP and IWV of Section XI.

After several years of work, the O&M Committee's working group on pumps and valves and Section XI's working group on pumps and valves agreed to a proposal and sent that through the ASME approval process.

The proposal added Part 6 and 10 to the O&M standard and replaced the detailed requirements in Subsections IWP and IWV with a simple reference to the appropriate part of the O&M standard.

The revision to the O&M standard was accomplished in the OMA 1988 and OMB 1989 Addenda to that standard. The revision to Section XI was included in the 1988 Addenda to that section of the boiler code. These revisions completed the first step toward separating the inservice testing requirements from the inservice inspection requirements.

With the latest revisions to the O&M standard and Section XI, the inservice testing requirements were split into two parts. Specific technical requirements for the scope of the testing program, test requirements, and methods and records were transferred to the O&M standard. Section XI retained the programmatic requirements.

Programmatic requirements include placing pump and valve inservice testing and Section XI inservice inspection plan, and include the duties and responsibilities of the third–party inspector.

To address all the code requirements, IST personnel must use two documents: Section XI of the boiler and pressure vessel code and Parts 6 and 10 of the O&M standard. Having these requirements in two distinct documents poses some difficulty for the user of ASME codes and standards, and eventually we will address that situation.

To consolidate the two documents, the Subcommittee on Nuclear Inservice Inspection and the Operation and Maintenance Main Committee have set up an O&M Section XI joint transition task group.

The joint task group has been charged with developing a proposal to address a directive from the Board on Nuclear Codes and Standards. This directive calls for us to "recognize that the Operation and Maintenance Committee is the appropriate committee to establish inservice testing requirements and to proceed with making the O&M standard stand on its own, with the objective of eventual deletion of IST from Section XI."

Currently, the goal of the joint task group is to publish a two–part document consisting of an inservice testing code and a standard guidebook. The target date for completing this effort is July 1990. Further, it is intended that the inservice testing code will eventually have a direct reference in the Code of Federal Regulations.

The test group is under the able leadership of Ed Williamson, a retired senior vice president of design, Southern Company Services and a past–chairman of the O&M Committee.

Another on–going task for the Operation and Maintenance Main Committee is the maintenance of the published inservice testing requirements. This maintenance is accomplished by revising published requirements to reflect technological developments and user feedback.

User feedback comes from people in industry who use the standard and then write to ASME with questions and suggested revisions.

All users of ASME Codes and Standards should be aware that all ASME committees meet regularly to consider written requests for interpretation of the published rules and suggestions for possible revisions, and all of you are encouraged to submit proposed changes that you feel would enhance our national standards.

User feedback is the prime input used by all ASME committees to revise their codes and standards. The best way for committee members to understand the problems and concerns of the industry is to interact directly with people in the industry.

Also, the best way to ensure that codes and standards are interpreted correctly is to have committee members explain the requirements directly to the users.

One way to encourage the exchange of ideas through user feedback is through a symposium such as this, and over the next few days you will be listening to a number of presentations, some of which will be given by O&M Committee members.

Also, one responsibility that we placed on the O&M Committee members who are present is to bring back an understanding of what should be done to improve their documents. This symposium is intended to create an environment in which members of the nuclear power industry can focus their thinking and discussion of pump and valve inservice testing.

Those making presentations will be highlighting current problems, accomplishments, and concerns regarding inservice testing of pumps and valves, and we in the Codes and Standards community invite your comments and suggestions, which we will use to write more effective and useful standards for the industry.

During our three days here in Washington we should accomplish two things. First, you should learn about the new O&M requirements, solutions to inservice testing problems and, hopefully, problems to avoid.

Second, the Codes and Standards community should learn where improvements are needed in our standards. With these two goals in mind, please remember that we invite your comments and suggestions on how the O&M standards can be made more useful and effective.

Finally, on behalf of Jerrold Dewease, Chairman of the Operation and Maintenance Main Committee, and the Board on Nuclear Codes and Standards, I hope you enjoy the symposium and that you find it professionally rewarding.

Thank you very much.

OPENING REMARKS
ED JORDAN, DIRECTOR
AEOD, U.S. NUCLEAR REGULATORY COMMISSION

It is a pleasure to be here. Tad did not warn me there might be questions afterward. I would like to ask some questions and then get answers back out of this session.

It really is a pleasure to welcome you to the ASME–NRC Symposium on Inservice Testing of Pumps and Valves. This symposium continues the open dialogue between the NRC and the industry to discuss technical issues.

I am here in place of Jim Taylor, the Acting Executive Director for Operations. Jim asked me to convey his regrets that he is unable to be here because of a conflict with a commission meeting. I think there is an optimistic note. The commission meeting is on the status of the EPRI design requirements for advanced light water reactors.

I share Mr. Taylor's interest and support of IST programs and welcome the opportunity to discuss my views with you today. I have a perspective that has placed me with a strong view that the sum of regulatory and industry efforts in the area of inservice testing has not measured up, either in implementation or program content and quality, to the needs of the 1980s, and there is evidence it will lag the needs of the 1990s.

Normally, when a regulator speaks to you, he either provides praise and strokes, or he challenges you. I think at this point you know where I am.

With a continuing lag between the transfer of lessons from this technology into industry codes and standards, regulatory actions such as bulletins and generic letters will likely continue at their current pace.

What can both the regulator and industry do to improve the situation? Wouldn't it be better if industry standards could replace many of the NRC actions? That is the theme of my discussion today. The fact that you are here indicates you are interested in effecting improvements.

I have examined the available preprints of papers for this symposium, and I am heartened by their technical quality, and I recognize and appreciate the industry actions they were drawn from.

Industry and regulators must find ways to substantially improve the responsiveness of inservice testing guides and standards development to meet industry needs, and to satisfy the regulatory concerns for implementation.

The basis for my views follows from my eight years of experience as an inspector in an NRC regional office, four years as a member of the ASME Operations and Maintenance Committee, as a charter member and current chairman of the Committee to Review Generic Requirements, and currently as director of the Office for Analysis and Evaluation of Operational Data.

As an inspector, I observed the lack of guidance and standards to aid development of IST programs. As a member of the Operations and Maintenance Committee, I experienced the frustrations of trying to get good practices incorporated into proposed standards, as well as the seemingly endless reviews, noodles, and delays in issuance and implementation.

I have seen OM 6 and 10 in many drafts, and I regret that I participated in some of the delays. I think it is important that we generate ways to get these things through the process much more rapidly.

As a member of CRGR, I review backfit considerations of NRC staff efforts to address current generic technical issues that in many cases should have been resolved by more timely industry action.

As a director of AEOD, I get telephone calls from the NRC operations center in the middle of the night about pumps and valves that don't function properly, and I subsequently review analyses of events that were compounded by problems with valves and pumps.

Would you believe I keep a tabulation of pump and valve figures by my phone? Well, anyway, my staff does, and they provide me with infinite varieties of failures, and they initiate recommendations that oftentimes find their way to you in some form regarding corrective actions that should be taken.

These problems are real, and yet, from my oversight of the NRC program for performance indicators, I acknowledge that since 1984 safety performance of U.S. nuclear power plants has improved significantly. Here are the strokes.

The evidence includes reduced scram rate, fewer significant events, fewer safety system actuations, fewer safety system failures, and in 1988 some decline in forced outage rates. Industry action is doing this. However, I don't attribute much of it to the industry initiatives in IST. Do you?

Analysis of the data so far in 1989 indicates the improving trend may have flattened out. For instance, for scram reductions many of the easy things have been accomplished at your plants. When you look at the major contributors by nuclear steam supplier, you can see that systems like feed water control (which then contributed a large fraction of scrams in 1984) are no longer dominant.

Now, the causes are distributed more widely. Therefore, improvements in safety performance are more likely through broad programs rather than narrow system-oriented fixes.

I am certain that a strong IST program aimed at ensuring reliable operation of important pumps and valves can add to the level of safety at your plants and improve overall plant availability. Such a program, combined with an effective maintenance program and an aggressive root cause and corrective action activity ensure continued success.

If you will indulge me, I will briefly substantiate my concerns about IST implementation and program deficiencies from information collected from certain of my office's activities.

NRC has included IST program review and diagnostic team evaluations since its outset about two years ago. Examples of certain safety system valves listed in IST programs were not tested. Safety-related pump tests that were outside acceptance criteria limits specified in the test procedures were not recognized as nonconforming. Deviation reports were not prepared. Programs for tracking and trending test results did not exist. And reverse flow testing of check valves was inadequate. And these are picked from various diagnostics.

The NRC shares the blame because of a backlog of review and approval of submitted IST programs that led to uncertainties about the status of the programs. One of the purposes of the diagnostic is to look also at the NRC's programs and identify where we have contributed to the problem. This finding provided an impetus for issuance of Generic Letter 89-04, Guidance on Developing Acceptable Inservice Testing Programs, in April of this year to eliminate the obstacles to IST program implementation.

I believe the NRC you see today is both tough and demanding and responsive to industry concerns. For example, the NRC is receptive to changes in test frequencies where excessive test reduce availability and may be hard on equipment.

Utilities must have a program they believe will truly improve equipment reliability, and the NRC must have assurance that the program will be implemented, the tests are meaningful, and equipment will be restored to full operability.

The generic lessons are apparent and must be communicated. This requires not only good guidance and procedures, but support at the highest levels of the organization, and clear assignments of responsibility to highly trained and motivated professionals. Your operating staff must be provided with equipment that will work when called upon.

Too often, we find instances in which the operating staffs have learned to live with accumulating equipment deficiencies. They should not accept it, and they should not have to.

Industry and regulators do continue to find new failure modes or weaknesses in test programs through experience. Two such examples are the determination in 1985 that motor-operated valve stroke testing was ineffective in ensuring valves will operate under accident conditions, and the recognition in 1986 that centrifugal pumps may be degraded by extended or accumulated operation under low-flow conditions that results in impeller suction recirculation.

The concern is that although much good work has been done, these issues are not yet fully resolved, and the technical lessons are not yet codified.

Industry actions and response to motor-operated valves and check valve programs illustrate both the strengths and shortcomings of improvement efforts. The motor-operated valve effort through NUMARC, INPO, and NUMAC have utilized experts in small groups to develop guidance covering both detailed procedures and broad elements for comprehensive motor-operated valve programs. I have reviewed this guidance, and it is excellent.

Also, developments of diagnostic signature tracing equipment have been highly beneficial. These actions were taken following Bulletin 85-03 that required reexamination and testing of motor-operated valves in high-pressure applications.

Unfortunately, industry has been reluctant to provide a mechanism for implementation of appropriate elements of this new guidance, and NRC recently issued Generic Letter 89–10 that requires and extends the testing to all safety–related valves.

The check valve efforts by INPO and EPRI were significant advances but, again, the general approach appears insufficient to ensure implementation of good programs across the country. Perhaps this is the friction point between industry and the regulator.

Industry is content when technical guidance is developed in any form, preferably not a regulation. The regulator is uneasy unless there is a high likelihood every plant will abide by the new solution, hence, a regulatory requirement. If we do less will industry do more? If we do more, will industry do less?

There is another activity underway that should help the regulator and industry focus attention on systems, components, and activities that have the greatest safety impact. Over the next three years, an individual plant examination will be performed at each plant, which amounts to a PRA. This program will result in identification of the most risk–significant event sequences on the most important systems, components, and activities.

Rest assured, these system and component lists will include pumps and valves as the active components. Failure rates for those active components must be fed into the PRA. For years we have used values from generic tables because of insufficient plant data. Well, with over 1,000 reactor years of experience, real–world data is in vogue.

How does your plant's component failure rate look with respect to tabular values often used? How does it look with respect to other plants using same or similar components? What contribution is IST and your NPRDS information making to assess, maintain and improve that component reliability? Are you satisfied?

If a component's performance appears lower than many of your associates', have you asked what their recipe is? If your performance appears better, have you shared your recipe for success through participation in a code committee?

There can be no secret recipes for this industry to continue to improve safety performance. IST can help recognize problems if the tests are valid and sufficient with respect to operability.

I also believe upgrades in maintenance programs will contribute. Improvements in maintenance are expected to improve component reliability and reduce recurring problems. These go hand in hand.

I would like to summarize and restate some questions for your consideration during this symposium. Although plant safety performance has shown improvement since 1984, I seriously question whether the contribution of IST to those improvements is as large as it should be and could be.

When did you last feel industry codes and standards development had been thoroughly responsive to new technical issues related to IST? Are we doing the best possible job of IST for equipment and operating plants? Will another event in an operating plant related to IST deficiencies cause further public loss of confidence in the nuclear power option? Assuming a continued improvement in operating performance of existing plants, will industry codes and standards, particularly IST, be ready for another generation of nuclear plants if economics and environmental considerations combine to favor this alternative?

And, finally, I want to leave the message that NRC considers it essential that an effective IST program be in place at each operating plant. IST must be an integrated program and a changing program as new knowledge and events dictate.

Our objective should be to find ways to expedite development and sharing of experience and expertise, and the IST guidance should be codified in a more timely fashion.

I believe it is preferable to have an industry solution rather than an NRC solution. IST is not a regulatory exercise. An effective IST program should result in increased safety as a result of improved component and system reliability plus economic savings from improved availability at each plant.

I applaud the actions taken to separate O&M from Section XI and cause it to stand alone. We should look forward for further ways to streamline the process to meet the challenges of the 1980s and the 1990s, and beyond. Thank you.

HISTORY OF VALVE AND PUMP TESTING AND SECTION XI

LARRY J. CHOCKIE, PAST CHAIRMAN
ASME SECTION XI COMMITTEE, CHOCKIE CONSULTANTS

This morning I have chosen a topic that would allow us to start on the history of pump and valve testing and Section XI. Now, as you can see, I am somewhat older than the average of the group in this crowd. I have also retired as Chairman of the Section XI committee. And I have also retired as Chairman of the Board on Nuclear Codes and Standards.

I have, however, written a couple of chapters for a book recently published on the history of the development of the U.S. Nuclear Codes and Standards, which is published by Elsevier, and compares our nuclear codes and standards with those developed in Germany, Great Britain, France, Japan, China, and a couple of other countries as well.

I think it is important that this symposium begin with the understanding of what Section XI attempted to accomplish by putting pump and valve testing into the Code and build upon that, because if you attempt to use the Code for purposes for which it was not written, you can oftentimes get into difficulty.

Now, before I get into the history of Section XI, I would like to make a point—and I'll use the example of using the Codes for construction of nuclear power plants, particularly the BWR where I worked for General Electric. We had about 650 engineers who, from my perspective, wanted to use the Section III Code for everything they did. That justified their design, their choice of materials, etc., and, therefore, if they followed Section III, no one could criticize them.

However, when you start using Section III for such things as designing a float gauge to measure the level of fluid in a tank, it won't float. If you use Section III to make a rotameter to measure low flows such as in systems transferring fluid from one tank to another, the rotameter can not send the magnetic pulses through the thick wall of the pipe. I had 54 examples of these kinds of problems to try to keep the engineers on track. The message is don't use the code where it isn't supposed to be used. Use it for its purpose.

During the discussions during the open forum here, I intend to insert a few remarks where some of the uses of the Code are not as the Section XI Committee intended, particularly as the rules for pump and valve testing were developed. So let me get back to the history.

Back in the late sixties when several nuclear plants were in operation, the technical specifications included a section addressing inservice inspection and inservice testing for those plants. The problem, however, was that no two were alike, even for plants that were being built by the same company or even those which were supposedly replicate plants on the same site, the technical specifications did not provide similar testing programs.

As such, the industry and the Atomic Energy Commission (later named the Nuclear Regulatory Commission) decided we ought to do something about it, and the simplest solution would be to put an industry group together under ANSI N–45 sponsorship and a similar committee under the AEC. We would see what we could make out of the proposed testing programs.

Well, the industry group met for six months and the AEC (NRC) group met for six months, and then we combined the two groups and determined what was the best route to take. It turned out the two groups' proposals were very, very similar, in the proposed content of the suggested inservice inspection and testing programs. Actually, there were only four points of difference between the groups that had to be negotiated.

The purpose behind Section XI at that time was to provide a program that would validate the provisions of the Atomic Energy Act of 194[illegible] mainly the provision that the failure of the nuclear reactor pressure vessel would be an incredible event—by definition. In other words, the vessel would not be permitted to fail, and that provision was used in this country by the designers of the reactor pressure vessels and the plant. Also, it was used throughout the world, and still is.

Supporting that provision, i.e. the vessel would not fail, led to quite a few things. First, the designers had to use the very best materials to build the pressure vessel, and the design must be by analysis so the designers could postulate every possible operating condition and accident scenario and design the reactor vessel such that it could accommodate each and every condition.

One of the accommodations, too, was a very good program of inspecting and testing the vessel as it was being manufactured—inspecting it, reinspecting it, nondestructively and physically—and finally continue inspecting and testing it throughout its service life with 100 percent testing 100 percent of the time.

You will find then, that in the construction Code, Section III, and in the Inservice Inspection Code, Section XI, both address the nuclear reactor vessel as a superior element in the nuclear plant—demanding our undivided attention.

Section XI, for example, in addressing the reactor vessel requires 100 percent of all the tests, 100 percent of all the inspections, 100 percent of the time, and repeated over and over throughout the service life of the vessel, whereas everything else in the plant gets simply a sampling program. The vessel is the target of Section XI.

So, when it came to the need for the designers to protect the integrity of the reactor pressure vessel they put in a number of so–called emergency safety systems, or engineered safety systems, some called them engineered safeguard systems. These systems had one purpose only, and that was to protect the reactor pressure vessel during any perturbation of the operating conditions or during accident events. As such, these systems generally stand idle during the operation of the plant and are to bring it down safely or to mitigate the consequences of an accident. The designers also chose to provide triple redundancy for these systems; in other words, there were three systems available to do the job, even though only one system would be necessary.

Now, actually, there were not always three identical systems, but there were other systems which could be called upon and provide triple redundancy to perform the required function of shutting down the reactor safely.

The Section XI committee mandated all kinds of nondestructive and testing programs for the reactor vessel, and some sampling programs of inservice examinations and tests for the pressure–retaining piping systems, pumps and valves, etc. But when it came to these emergency safeguard systems, they sat idle all the time, no fluid, no flow, no pressure. They had to be functional when called upon.

How do we ensure that the systems are going to operate when necessary? The committee chose to address the active components in those systems, which happened to be the pumps and valves, and test them periodically to provide a high degree of assurance that those systems would operate when called upon.

The words in the initial editions of Section XI, and they are there still today, state that the purpose is to shut the reactor down safely and to mitigate the consequences of an accident.

Now, as things evolve over time, words sometimes change and result in different meanings. We had some difficulty way back when in trying to define such words as "safety–related," "safety–related function," "essential to safety." These words meant different things to different people, and they still do, unfortunately. I see Bob Bosnak smiling, because he and I are both on Section III and that committee has never yet been able to even define a "piping system."

Well, Section XI had its share of difficulties too, because when we started we were under the ANSI N–45, yet we determined "Handy Hannah Hints" as the ANSI standards were regarded at the time. To make the document mandatory, we needed to move the whole operation under the auspices of the ASME Boiler and Pressure Vessel Code, because the jurisdictions—that is, all of the states—already had laws on their books that adopted the Code as the requirements for the construction and the continued testing and maintenance of pressure–retaining equipment in their jurisdictions. It was assumed in the early sixties the states would take an active role in periodically examining and testing the components in a nuclear plant. It has not turned out that way though. The NRC is the main enforcer of the Section XI Code.

The point I now make is that to help ensure the integrity of the pressure–retaining ability of the nuclear reactor pressure vessel requires the functional testing or performance testing of pumps and valves in the emergency safeguard systems.

Research programs were coming along at the time, the late sixties and early seventies, such as the heavy section steel program, HSST, addressing that one important subject—the quality of that pressure vessel. How possible is failure? How sure are we that failure of that vessel would be an incredible event? We had the Rasmussen studies and the fault–tree analysis.

Yes, we began to put numbers on the potential probability of a failure and what kind of a failure the vessel failure would be. And considering nuclear vessels around the world, we are now talking in hundred of thousands of vessel–years of operation, and we know the numbers for the probability of failure and what kind of failure it would be.

The reactor vessel failures will not be cataclysmic or catastrophic. They will simply be leak–before–break–type failures. We are all pretty convinced of that. But that does not mitigate the necessity of continuing to ensure that those safety features in the plant and the safety functions of those engineered safeguard systems are working; for their function, or failure to function, was also considered in the analysis of potential vessel failures. And so, we had the necessity to include the performance testing of pumps and valves in the engineered safeguard systems in Section XI, to help ensure the maintenance of integrity of the reactor pressure vessel.

Again, the Section XI committee found themselves in a dilemma when it joined ASME to make the rules mandatory through automatic adoption by the jurisdictions and adoption by the NRC. The ASME Boiler and Pressure Vessel Code did not address the operational aspects of pressure–retaining equipment, only the pressure–retaining ability, and we had to go to the various boards and councils in the structure of the ASME to get permission to include pump and valve testing in Section XI. Approval was obtained in 1972, and the next published addenda, the Summer of 1973, included the two new subsections on pump testing and valve testing.

About the time the two new subsections on pumps and valves were included in Section XI, a new ASME committee was formed to provide standards for the operability of other equipment in a nuclear plant. This new committee was named Operation and Maintenance, and the jurisdiction for the operability of the pumps and valves included in Section XI was transferred to the O&M committee, with the understanding that improved rules for pump and valve testing would be developed by O&M and referenced in Section XI for mandatory enforcement. Again, the targets were those pumps and valves required to shut the reactor down safely and mitigate the consequences of an accident.

Additionally, the O&M Committee was to address standards for other equipment, such as air and gas treatment, operation of nuclear cranes, etc.

Well, I've spoken pretty much without looking at my notes. I did prepare a paper. It was provided to you in your handout, and my comments, along with the paper should establish that this conference should not overlook the importance of the testing of the pumps and valves necessary to shut the reactor down safely and to mitigate the consequences of an accident.

As the Section XI rules have been implemented since the early seventies, the real questions have become which are the pumps and which are the valves to be included in the pump and valve testing program. Selecting the pumps did not appear to be too difficult, since it is those pumps connected to the emergency bus. The main coolant pumps are not connected to the emergency bus, for example, and they are not needed to shut down the reactor either, so obviously you can leave those out.

When it comes to valves, the situation and the solution is not so simple, because out of a population that normally exceeds 11,000 valves in any one plant, how do you select those that are necessary to shut the plant down safely?

Again, such words as "important to safety," the "Q List," "safety–related," etc., simply do not mean the same thing to all people. Our committee has even had to define such terms as "active" and "passive" as they pertain to valves, and include the definitions in the Code itself. Helping to define the purpose the valves are to accomplish, and decisions as to what parameters are to be measured, are subjects this symposium can well address.

What are we really addressing? Do we need to go further than simply shutting the reactor down safely, mitigate the consequences of an accident? Are we looking at operations, such as the NRC's new generic letter suggests—the testing of pumps and valves to keep the plant operating safely? Because if pumps and valves necessary to keep the plant operating safely is the subject, then we have an entirely different population of pumps and valves than those addressed by Section XI, those required to shut the reactor down.

So, I wish you luck, I will be here throughout the symposium. I will insert my comments where I think they are appropriate, and I see today that we are turning this whole industry over to a younger group, a younger generation, and some of the policies and philosophies that us old–timers have included in the codes and standards are subject to revision. But remember, Section XI was not written to include the testing of such valves as those in the heating and ventilating ducts to the control room, nor diesel fuel transfer pumps. Don't attempt to use the Code for purposes for which it was not written. Thank you kindly.

KEYNOTE ADDRESS
INSERVICE TESTING – A CAUTION AND A CHALLENGE

KENNETH C. ROGERS, COMMISSIONER
U. S. NUCLEAR REGULATORY COMMISSION

Good evening, ladies and gentlemen. I'm delighted to be here tonight to offer you some of my thoughts and observations on the important subject of inservice testing (IST).

First, I want to commend you on this joint NRC–ASME conference and on the topics you are covering in it. Given the breadth and depth of the papers being presented, and the number of experts on this subject here tonight, I will not presume to give you an in-depth technical dissertation on IST. Rather, I want to offer you a caution and a challenge. I will include just enough technical details to validate my points, but I will leave it to you to consider these in greater detail sometime during your three days here.

Before I turn to my specific thoughts on IST, I would like first to provide you with my observations on the overall relationships among various inspection and testing activities at nuclear power plants, as well as their relationships to other activities. As you are fully aware, a large amount of inspection and testing is required to be conducted at nuclear power plants. Among these requirements are those associated with inservice inspection, inservice testing, and Technical Specification surveillance. The focus of each of these efforts is quite distinct. For example, inservice inspection is intended to provide assurance of the structural integrity of important components within the plant; inservice testing provides assurance of the operability of pumps and valves that are important to the safety of the plant; and Technical Specification surveillance is directed at the overall operability of safety systems.

Each inspection and testing effort should be important for the safe operation of the nuclear power plant and one effort can not substitute for another. Further, these efforts often require plant personnel with different expertise, but they cannot be conducted in isolation. All of them impact directly on the maintenance and operations activities of a nuclear power plant, and to some extent, on each other. It strikes me that the whole area of inservice inspection and testing is therefore a fruitful one for application of a systems approach to operations.

First, given the fact that several groups are involved with inspection and testing at the plant, it is imperative that effective lines of communication are established between these groups. Secondly, communication must extend to other groups, particularly corporate personnel and the maintenance staff at the plant. Communication must start with the planning for testing and inspection activities, and extend through communication of the results, feedback on impact of the results, and continual updating on changes in equipment or procedures of one activity that may have an effect on other activities. The importance of testing of pumps and valves to the planning of preventive maintenance programs and the scheduling of corrective maintenance activities is particularly noteworthy, and requires continuing attention to successful communication.

By establishing effective communication lines, it should be possible to conduct inspection and testing in a coordinated manner that provides assurance that structures, systems, and components important to the safe operation of the plant are capable of performing their safety functions, both individually and collectively. In the case of inservice testing, there must additionally be the assurance that there is adequate operational redundancy to assure plant safety during the testing.

I believe there is no question that IST can be a valuable and important tool in diagnosing the condition of selected nuclear power plant components during plant operation. It is essential that when components must be tested under operating conditions, the IST program permits monitoring of component status without interrupting the operation of the plant. IST certainly has proven its worth in a number of applications. Furthermore, there is a significant reservoir of new and largely untapped methods of non–invasive testing that could considerably expand the scope and value of IST.

On the other hand, inservice testing has not been without some problems and shortcomings. For example, certain IST methods currently in use have limitations which, if not fully understood and corrected, can result in testing which is largely without value, and which can lead to a false sense of security, testing which can degrade equipment, or even testing which can put the plant at risk. There have, unfortunately, been cases in the past where IST methods having such shortcomings have been used. As these limitations have come to light, appropriate actions have been instituted to correct them. Nevertheless, case histories emphasize a need to understand potential negative effects of IST on components. In addition, organizational and procedural weaknesses have been found in IST programs that have caused concern for the safety of the plant.

The caution then is to avoid inadequate or detrimental IST resulting from (1) the misuse of testing methods; or (2) ineffective implementation of IST programs; or (3) testing that puts the plant at risk. The challenge is to seek and select testing methods that will accomplish the intended objectives of IST.

Let me discuss some of my specific concerns, review some of the on–going developments in this area, and then consider some possible areas for expansion and further development.

WEAKNESSES OF CURRENT IST PROGRAMS

As I see it, current IST programs are limited by the fact that they do not address problems from a systems point of view, but rather focus on individual pumps and valves in isolation from the rest of the plant. This has resulted in both general problems in making the greatest possible use of IST, and in specific deficiencies in the testing of certain components. I will discuss both areas.

First, however, let me say a few words about my view of the importance of a systems approach at nuclear power plants. A nuclear power plant is a very large and complex system. It involves extensive hardware, software, and people. The hardware itself is a complex array of components, such as pipes, and mechanical and electrical systems. Major components such as pumps and valves are systems in themselves, as well as being parts of larger systems. By the nature of our training in particular engineering disciplines, we have become accustomed to viewing individual pieces of hardware or software in isolation (and to largely neglect the people component altogether). We have discovered that this insular view is seriously deficient, and a number of the more significant problems experienced at nuclear power plants are a direct consequence of someone having ignored the fact that taking action on one component of a power plant can sometimes affect a totally different component or system.

In the first place, this means that care must be taken that testing in itself does not put the plant at risk. Whenever testing interferes with the functionality of a component, one must examine the vulnerability of the plant to the system being tested. Probabilistic risk assessments may be helpful in establishing the necessary confidence that the plant will remain in a safe condition.

Further, this means that IST activities must be more carefully coordinated with all the other major activities in the plant, particularly with operations and maintenance. Planning for IST activity, as well as inspection activity, should take account of expected operations and maintenance activities. Inservice testing and maintenance have a common goal of providing assurance of the operability of plant components; therefore, the licensee organizations responsible for these systems activities must be closely coordinated. Further, inservice testing can provide valuable information to the maintenance organization. Maintenance planning and scheduling should, therefore, reflect information on component status derived from IST. Feedback on the uses of IST results can help improve IST programs. Changes, such as changes in IST procedures that might affect maintenance activities, or changes in maintenance procedures that might affect IST activities, must be promptly communicated.

Now, turning to some of the deficiencies that have been identified for specific tests, I want to highlight problems involving inservice testing of pumps in mini–flow conditions, testing of motor–operated valves (MOVs), and testing of check valves. While these problems are being corrected through changes to requirements, they are useful to review because they provide strong evidence of the consequences of having developed IST requirements without sufficient recognition of the systems aspects of the situation.

The Pump Mini–Flow Problem

First, let us consider the testing of pumps used in safety applications. Such testing is required by Section XI of the ASME Boiler and Pressure Vessel Code and plant Technical Specifications on a schedule, for the most part, of once every 3 months. For some pumps (such as those in the auxiliary feedwater system) this testing is performed through use of small recirculation lines that have a capacity of 5 to 10 percent of full pump flow. Recently, some examples have been identified where such operation resulted in damage to safety–related equipment. Therefore, in May of 1988 NRC issued Bulletin No. 88–04, "Potential Safety Related Pump Loss." In this bulletin, the NRC identified two potential miniflow concerns. The first concern involves the potential for the dead–heading of one or more pumps in safety–related systems that have a mini–flow line common to them or other piping configurations that do not preclude pump–to–pump interaction during mini–flow operation. A second concern is whether or not the installed mini–flow capacity is adequate for even a single pump in operation. Bulletin No. 88–04 requests licensees to evaluate their systems for potential mini–flow problems and to identify necessary modifications and an appropriate implementation schedule.

Valve Problems

There are thousands of valves in a nuclear power plant. Probably several hundred of them are crucial to the integrity of the defense barrier for the isolation of leaks and the prevention of release of radiation to the public. To manage a system of such complexity, the technical procedures for testing valves should be based on fundamental principles of mechanical and electrical engineering. Unfortunately, this does not seem to be the case. In particular, valve settings have generally been established in isolation. As a result, the valves may not always be able to perform safety functions as intended, and the IST performed may be misleading. Sound engineering principles suggest to me that valve diagnosis must treat the valve as a system in itself as well as a component of a larger system, and valve settings must reflect the full range of conditions the valve may experience.

For valves, as opposed to pumps, the issue is not so much one of rapid damage as it is of cumulative wear and tear from testing. When that wear and tear cannot be related to an identifiable benefit from such testing, the need for improved testing methods is, in my opinion, compelling. For MOVs, for example, some studies have shown that the required stroke–time test is of limited value, and that the more frequent operation required by the testing process results in accumulated wear to the valve seats. These concerns were recently documented for MOVs in NRC Generic Letter 89–10, "Safety–Related Motor–Operated Valve Testing and Surveillance."

In addition to concerns for damage to hardware caused by improper inservice testing, the implementation and procedural aspects of inservice testing at the plants have been a source of problems. For example, incorrect setting of operating switches in MOVs has led to instances where the operators were unable to open or close those valves electrically. One such instance involved a complete loss of feedwater in 1985 at the Davis–Besse nuclear power plant when MOVs in the auxiliary feedwater system could not be reopened electrically after their inadvertent closure. On the other hand, at Millstone Unit 3 in February of this year, due to incorrect torque switch settings an MOV in the safety injection line could not be closed electrically under full line flow following an inadvertent safety injection actuation. Generic Letter 89–10 will ensure that the inservice testing conducted at the plants provides assurance, through testing or acceptable alternative means, that safety–related MOVs will operate under such conditions.

With respect to check valves, experience has also revealed the need to improve inservice testing at the plants. Following several check valve failures, the Commission began a series of inspections to evaluate the effectiveness of inservice testing of check valves and to monitor the progress of the industry in resolving the concern for check valve reliability. Two particular problems found during these inspections are (1) the omission from the IST program of many check valves important to the safety of the plant and (2) the fact that testing was not always conducted in a manner allowing verification of the ability of the check valves to perform their safety functions. The NRC issued an Information Notice (No. 88–70) in August 1988 that discussed these findings.

NRC and Industry Efforts to Improve IST Programs

The NRC is working to improve the guidance provided to the licensees on the proper establishment of IST programs. One example of this effort is Generic Letter 89–04, "Guidance on Developing Acceptable Inservice Testing Programs," issued on April 3 of this year. Generic Letter 89–10, which I mentioned earlier, is also part of this effort. Additional guidance is being planned.

Industry groups have also recognized the need to improve inservice testing. The Institute of Nuclear Power Operations (INPO) has increased its emphasis on inservice testing. For example, on October 15, 1986 INPO issued Significant Operating Experience Report (SOER) 86–3, "Check Valve Failures or Degradation."

That SOER provides a summary of the methods available to detect check valve failures and degradation, and recommends that preventive maintenance procedures be established in a test and inspection program for check valves in several selected systems.

With respect to MOVs, INPO has been performing inspections that also include inservice testing of these valves at plant sites. Further, INPO prepared a summary description of the key elements needed for a comprehensive MOV program, highlighting aspects of inservice testing such as procedures, training, the need for assuring operability under design basis conditions, and use of diagnostic equipment.

The Electric Power Research Institute (EPRI) has also been involved with the industry efforts to improve inservice testing. For example, a report identified as EPRI NP–5479, "Application Guidelines for Check Valves in Nuclear Power Plants," was issued in January 1988. That report included information on methods to evaluate check valve degradation. EPRI is preparing similar application guidelines for MOVs which are expected to be issued later this year.

In an effort to improve the standards for inservice testing, the industry, with NRC participation, has developed ASME Operation and Maintenance standards OM–6 for pumps and OM–10 for valves as a replacement for those requirements in Section XI. A number of you here tonight have been associated with the development of these new standards, you should be proud of your efforts. While there are some problems in these standards that the NRC and you are working together to resolve, I believe the industry's use of the new standards will improve the quality and overall usefulness of IST.

Possible Innovations in IST

There are a number of newer, non–invasive testing and surveillance techniques that can provide improved diagnostic information and can avoid some of the shortcomings of testing that are caused by testing requirements to turn equipment on and off or disturb the operating system in some other way. These advanced techniques include monitoring of vibrations and/or acoustic emissions and other such "signals" from operating equipment. Both NRC and industry are engaged in a variety of efforts to develop and validate such techniques as replacements or add–ons to IST programs at nuclear power plants. Let me cite a few of these initiatives.

The industry has been investigating new methods to improve inservice testing. For example, MOV signature tracing techniques have been developed that provide methods of analyzing the condition and operability of a valve. These techniques can be used to predict possible future MOV problems. Many plants have employed them with varying degrees of success. NRC's Office of Nuclear Regulatory Research, through its Valve Performance Program, is currently validating the use of motor signature monitoring. Preliminary results indicate that motor signature instrumentation provides valid information for the periodic readjustment of MOVs to ensure their operability.

With respect to check valves, a variety of techniques have been used by the industry in the past to determine valve condition, including visual, acoustic, and radiographic methods. Now, the industry is evaluating additional new techniques. One such is Checkmate by MOVATS which provides an ultrasonic signature of valve disc movement. Another is a portable high energy radiography technique (MINAC–6) by Schonberg Radiation Corporation. Finally, radioactive tracing of the valve hinge and disc is a newly identified technique which merits further investigation.

NRC, through its Nuclear Plant Aging Research Program, is also evaluating advanced monitoring methods. Current areas of effort include acoustic signature analysis, ultrasonics, and radiography. Acoustic emission monitoring is being investigated for valve leakage as well as valve condition. Pressure noise and magnetic flux are also being studied. A draft final report on these areas, including recommendations for action by the NRC staff and industry, is scheduled for October 1989. NRC is also evaluating the use of visual inservice inspection techniques to detect check valve degradation, such as boroscopic inspection of valve internals.

In April 1989, nuclear utility representatives formed a Nuclear Industry Check Valve Group (NIC) for the exchange of technical information relating to the application, maintenance, and testing of check valves. Twenty–seven nuclear utilities are currently NIC members. An objective of NIC is to provide a vehicle for utility communication on check valve issues to INPO, NUMARC, ASME, and a mechanism for making recommendations for the resolution of generic issues. Plans are for NIC to monitor regulatory and industry requirements for check valves and the actual practices by its members on application, maintenance, and testing. NIC is also intended to provide a forum for the identification, evaluation, and development of guidelines for the use of non–intrusive examination methods.

CONCLUSION

I want to encourage the further development of inservice testing and surveillance techniques by NRC and our licensees, both in improving existing techniques and in developing new ones. The correction of deficiencies in existing techniques is important to assuring that IST is able to realize the positive contributions to safety it was designed for, without inadvertently degrading safety or operability, and without putting the plant at risk during the testing procedure. The development of new techniques to replace or enhance existing IST methods is important to achieving the full potential of IST in providing a capability to determine the status of critical pumps and valves without shutting down an operating plant.

Taken together, the improvement of existing techniques and the development of new ones have the potential to benefit the industry significantly in improved safety, reliability and efficiency of operations. The safety concern is, of course, particularly important. As I have discussed, there is a potential benefit to safety in reducing the stresses to safety systems that some existing IST techniques have caused. In the future, we need to learn from our mistakes and try to do a better job of examining proposed new methods and their uses before they are put into effect so as to assure that they do not have any inadvertent negative impacts on safety, either by impairing the ability of critical components to perform their safety functions during the test, or by introducing damage or wear to components that could cause them to fail later. To this end, the work being done by the NRC and industry is most encouraging. I, therefore, urge you to support the efforts underway by the various industry groups to develop and validate new IST methods and to prepare industry guidelines that address the known shortcomings in current IST methods. There is an opportunity for strong industry leadership in this area, as well as a need for continued NRC efforts which address regulatory issues.

However, no method, new or old, can be truly effective unless it is implemented fully and properly. Nuclear power plants managements must be aggressive in implementing the guidelines and recommendations from INPO and EPRI. It is important that individual licensees review their IST programs to determine where improvements can be made, such as by developing better engineered test conditions and by incorporating new testing methods for monitoring component condition. Further, licensees need to assure that their IST is effectively integrated with other key activities in their plants. In particular, IST can provide important information for the effective implementation of maintenance programs. Therefore, IST activities must be closely coordinated and effectively integrated with maintenance activities, both for long–term planning of maintenance programs, and for short–term scheduling of specific maintenance actions. IST offers the best potential for focusing maintenance toward the most critical areas by identifying degradation of important equipment early enough to be able to program the maintenance into the next planned outage. This can help assure safety without having to routinely perform maintenance on equipment where it is not yet necessary, or worse, having equipment fail, thus necessitating a costly unplanned plant shutdown and an urgent need for corrective maintenance.

In conclusion, I hope that, during the course of this symposium, you will be addressing some of these topics, and that, in your continued technical work you will be able to move toward an improved and expanded use of IST.

PLANNED SESSION PAPERS

PART 1: GENERIC CONSIDERATIONS FOR PUMP IST PROGRAMS

INTRODUCTION TO ASME/ANSI OMA–1989A PART 6 – "INSERVICE TESTING OF PUMPS IN LIGHT–WATER REACTOR POWER PLANTS" AND TECHNICAL DIFFERENCES BETWEEN PART 6 AND ASME SECTION VI, SUBSECTION IWP

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ABSTRACT

ASME/ANSI OMa–1988 Part 6, developed over the last ten years has recently been adopted by the Boiler and Pressure Vessel Code, Section XI. This new standard provides inservice testing requirements for certain Class 1, 2, and 3 pumps utilized in nuclear power plants and replaces ASME Section XI, Subsection IWP. This paper provides background information on Part 6, highlights the significant changes to pump testing requirements, and provides the philosophy associated with the new requirements.

ASME O&M STANDARDS ON PUMP AND VALVE TESTING

Background

Nuclear power plants are required by Federal regulation [10 CFR 50.55a(g)] to test certain ASME Code Class 1, 2 and 3 pumps and valves in accordance with ASME Section XI of the Boiler and Pressure Vessel Code. The Board of Nuclear Codes and Standards decided that nuclear pump and valve testing should be performed in accordance with an Operation and Maintenance (O&M) Standard. With that in mind a working group was formed, approximately 10 years ago, under the Operations and Maintenance Committee, with the objective to develop new and separate standards for the inservice testing of pumps and valves.

The new O&M pump and valve standards (Part 6 and Part 10) have just been published as the 1989a addendum of ASME/ANSI OM–1987 Operation and Maintenance of Nuclear Power Plants. These new standards are referenced in the 1988 addendum of Section XI and replace Subsection IWP and IWV of that code in their entirety.

Part 6 and 10 Goals and Objectives

The O&M Working Group on Pumps and Valves was directed to review ASME Section XI IWP and IWV, and use the lessons learned from past implementation, to develop new O&M standards. One goal of the Working Group was to develop standards which could be more easily followed and yet enhanced the most important aspects of the current standards. Any changes that were proposed were approved on sound technical, documented basis.

Most nuclear plant technical specifications require utilities to obtain specific written approval from the Nuclear Regulatory Commission (NRC) for any deviations from the Code. This has been very difficult for both the nuclear plant operators and the NRC, who must review and approve each deviation. For example, if a utility wants to determine flow rate of a pump by measuring the change in level of a tank divided by time instead of measuring flow directly, then the utility must request relief from Code requirements. This situation arises when there is no installed flow instruments—usually for small pumps. Whenever possible these standards have been revised to reflect current testing practices and the requirements have been changed to reduce requests for relief, keeping in mind regulatory approval precedent.

At first glance, one would think OM–6 and OM–10 are quite similar to the IWP and IWV sections that they replace. However, on closer inspection, an inservice test (IST) engineer will note many significant differences. The remainder of this article will look at the differences between IWP and Part 6. Specific details regarding all changes made are contained in Attachment 1.

An Overview of Changes in Part 6. Part 6, like its predecessor IWP, specified that pumps which perform a safety–related function in shutting down the reactor or in mitigating the consequences of an accident shall be tested. However, Part 6 scope does not limit testing to only ASME Class 1, 2 and 3 pumps, as does IWP. Part 6 requires all pumps which perform a safety function to be tested. It is not expected that this change will add many new pumps to IST programs. The testing ensures that pumps function correctly with regard to both the hydraulic and mechanical conditions. For the hydraulic conditions, the pumps' differential pressure and flow rate are compared to a reference value. Any deviations over certain limits must be reviewed, and if the deviations are not within the acceptance limits of the standard, corrective action must be taken. Similarly, the mechanical condition of the pumps are determined by measuring vibration levels while the pump is running and comparing these levels to a reference value. A key element of this standard is to set reference values when the pumps are known to be operating acceptably.

Mode of Vibration Measurement. Section XI IWP specified vibration levels to be measured in the displacement mode and compared to its reference value. Part 6 is similar in that the vibration levels must be compared to previously established reference values. However, Part 6 includes acceptance criteria based on vibration velocity as well as vibration displacement. Velocity measurements are usually superior to displacement measurements in determining severity of vibration and are widely used by the nuclear industry. Vibration severity is determined by the amplitude of the vibration and the frequency at which it occurs. For example, a pump might vibrate with one mil "peak to peak" at 1800 rpm and experience no problems. However, if the same pump were vibrating one mil at a frequency of 18,000 rpm, 10x the operating speed, the pump would be expected to fail. Velocity is a function of both amplitude and frequency, and thus a good indication of severity. Displacement is only dependent on amplitude. If one measures vibration displacement and assumes the amplitude is at rotating speed, a gross underestimation of the severity of the reading will occur if the amplitude is really at a high frequency. These changes to Part 6 will allow the use of velocity without having to request relief from the NRC.

For very slow speed pumps, the vibration acceptance criteria is given in mils displacement. At these low speeds, displacement is a better indicator of degradation than velocity. The displacement may be very large while the velocity is quite low, and thus equipment damage could be occurring problems, while vertical line shaft pump bearings are normally inaccessible. For these reasons, the hydraulic acceptance criteria for these pumps has been made more conservative by increasing the alert and required action ranges from 0.93 to 0.95 and 0.90 to 0.93 times the reference value, respectively.

Deleted Requirements. Several parameters are no longer required to be measured by Part 6. Bearing temperature, inlet pressure and lube oil level or pressure are no longer required. Bearing temperature, if monitored continuously, is an excellent indicator of bearing degradation. However, yearly readings have not proven beneficial. The other two parameters were dropped because they did not provide useful, trendable test data. The parameters were included originally as a help to the operators to ensure the pump was not damaged during testing.

Dry Sump Pumps. Part 6 specifically addresses testing of pumps which are installed in dry sumps. In the past, some utilities have been "bumping" these pumps quarterly. That is, they start the pumps dry. This pump test is detrimental to the pump and serves no useful purpose. These pumps must now be tested at least every two years. The owner will have to make provisions for supplying the necessary fluid inventory for the test.

Future Changes. At the February 1989 ASME Board on Nuclear Codes and Standards meeting, a motion was passed to recognize that the Operations and Maintenance Committee is the appropriate committee to establish inservice testing requirements. The O&M is to proceed with making OM – 1987 stand on its own, with the objective of eventual deletion of inservice testing from Section XI. An O & M/Section XI Joint Transition Task Group has been organized to implement this change.

The O&M working group on pump and valve testing is actively considering further improvements to these standards. The areas for improvement have been recommended by industry as well as regulatory bodies. Task groups and action plans have been established for the higher priority issues and results should be forthcoming. Some of the pump issues being considered include; improvements in mini–flow acceptance criteria, hydraulic acceptance criteria, trending, and analysis time limits – 96 hour criteria.

Comparison
IWP To Part 6

	IWP	PART 6
IWP-1100	Scope	All safety related; Class 1,2 and 3 deleted. Includes preservice testing.
IWP-1300	Owners responsibility	Requires owner to make design provisions.
IWP-1400	Bypass loops	Adds words to alert owner to minimum flow in bypass loops.
IWP-1500	Detection of change	Places more emphasis on reference valves.
IWA-9000	Definitions	Instrument accuracy Instrument loop Preservice test period Reference valves
	Preservice testing	New
IWP-3100	Inservice test	5. B & C allow "Determination" of P or Flow Rate in lieu of measurement – should reduce relief requirements.
IWP-3100-1		Delete lubricant level or pressure. Deletes bearing temperatures. Deletes inlet pressure. Discharge pressure only required for positive displacement pumps.
IWP-3110	Reference values	Adds emphasis on establishing reference values "Only when the pump is known to be operating acceptably".
IWP-3111	Effect of pump Replacement, repair and routine servicing on reference values	Following Maintenance, a test must be run prior to declaring the pump operable in lieu of within 96 hours.
IWP-3100-2	Allowable ranges of test quantities	Vibration:

	Alert	Action
<600 RPM	>2.5 VR to 6 VR	>6 VR
>10.5 mils	>22 mils	
>600 RPM	>2.5 VR to 6 VR	>6 VR
3.25 in/s	>.7 in/s	
Hydraulic:		
P (Positive Displ.)	.93 to 1.1 PR	
P (Vert. line shft)	.95 to 1.1 PR	
P (Centrifugal)	.9 to 1.1 PR	
Q (Centrifugal)	.9 to 1.1 QR	
Q (Others)	.95 to 1.1 QR	

Comparison
IWP To Part 6 (continued)

	IWP	PART 6
IWP–3210	Allowable ranges of inservice test Quantities	Part–6 no longer allows the owner to set his own ranges if those specified cannot be met.
IWP–3220	Time allowed for analysis of test	No change, however there were many negative ballots of this subject, and this area will get more review with possible changes.
IWP–3400	Frequency of inservice tests	No change, however there is an INPO concern on this subject which will be addressed in the first addenda to Part 6.
	Pumps lacing fluid inventory	Part 6 addressed pumps in normally dry sumps – must be tested every two years – new requirement.
IWP–3500	Duration of tests	Part 6 reduces minimum run time from fiv minutes to two minutes.
IWP–4120	Range	Adds provisions for digital measurements.
IWP–4130	Instrument location, transmitters and computers	Eliminates transmitters and computers – primary concern is response not transmission of signal.
IWP–4150	Fluctuations	Part 6 deletes 2% requirement on fluctuations.
	Frequency response range	Part 6 changes requirement for frequency response range of vibration instruments to be 1/3 minimum pump speed to 1000 Hertz.
IWP–4220	Pressure tap construction	Deletes requirement.
IWP–4230	Pressure tap location	Paragraph rewritten – additional documentation required – requirements for where tap is located have been deleted.
IWP–4300	Temperature measurement	Not required by Part 6.
IWP–4500	Vibration	Part 6 Imposes additional requirements; two Orthogonal directions on each accessible pump bearing. On vertical line shaft pumps, the top motor bearing shall be measured in all three directions.
IWP–4520	Instruments to measure amplitude	Type of instrument no longer specified.
IWP–6210	Summary listing	Part 6 deletes this requirement.
IWP–6220	Pump records	Expanded to include pump manufacturer's operating limits.

Comparison
IWP To Part 6 (continued)

	IWP	PART 6
IWP–6230	Inservice test plans	Requirements expanded to required documentation of method used to “determine” reference values.
IWP–6240	Record of tests	Expanded to require pump Identification reason for test evaluation/justification for changes to reference values.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
IWP–1100 Scope	1.1 Scope	Change: Adds "preservice testing".
This Subsection provides the rules and requirements for inservice testing of Class 1, 2, and 3 centrifugal and displacement type pumps that are installed in light–water cooled nuclear power plants, that are required to perform a specific function in shutting down a reactor or in mitigating the consequences of an accident, and that are provided with an emergency power source. The results of these tests are to be used in assessing operational readiness of the pumps during their service life.	This Part establishes the requirements for preservice and inservice testing to assess the operational readiness of certain centri–fugal and positive displacement pumps used in nuclear power plants. The pumps covered are those, provided with an emergency power source, which are required in shutting down a reactor to the cold shutdown condition, maintaining the cold shutdown condition or mitigating the consequences of an accident. This Part establishes test intervals, parameters to be measured and evaluated, acceptance criteria, corrective actions, and records requirements.	Basis: OM–6 has specifically added preservice testing as a requirement. The primary purpose of this standard is to detect change which may indicate a degraded condition. Pre–service testing will ensure that the basic testing program is in place and the pumps are tested when they should be in good working condition. Change: Deletes Class 1, 2, and 3. Basis: This standard should not be limited to ASME Class 1, 2, and 3 components. In newer plants, most of the pumps intended by the working group to be covered by this standard would be. However, there could be some pumps which are not ASME Class 1, 2, and 3 which do perform a function as described in the scope. The working group wants to stress the point that function is more important than the code class. Change: "Nuclear Power Plants: versus" Light Water Cooled Nuclear Power Plants. Basis: The working group's review revealed that there was no particular need to limit OM–6 to light water cooled nuclear power plants. The standard, as written, is broad based. Therefore, OM–6 has been broadened to include all nuclear power plants. Change: Addition of "Test intervals, parameters to be measured and evaluated, acceptance criteria, corrective actions, and records requirements."

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
		Basis: These statements better define the contents and scope of OM–6 and are consistent with other O&M standards. Change: Added "Maintaining the cold shutdown condition". Basis: To be consistent with OM–10 and IWV which include this phrase.
IWP–1200 Exclusion (a) Drivers are excluded from the requirements of this Subsection, except where the pump and driver form an integral unit and the pump bearings are in the driver. (b) Class 1, 2, and 3 pumps that are supplied with emergency power solely for operating convenience are excluded.	1.2 Exclusions The following are excluded from this Part: (a) drivers, except where the pump and driver form an integral unit and the pump bearings are in the driver; (b) pumps that are supplied with emergency power solely for operating convenience.	Change: Editorial and deletion of reference to Class 1, 2, and 3. Basis: The reference to Class 1, 2, and 3 is not necessary due to the change in scope of this standard as compared to Section XI.
IWP–1300 Owner Responsibility Water Cooled Nuclear Power Plants". In addition to the requirements of IWA–1400, the Owner shall have included (as part of the pump and plant design) valves, instrumentation, and other provisions needed to comply with this Subsection.	3.1 Owner's Responsibility (a) It is the Owner's responsibility to include in both the pump and plant design all necessary valving, instrumentation, test loops, required fluid inventory, or other provisions which are required to fully comply with the rules of this Part. (b) Each pump to be tested in accordance with the rules of this Part shall be identified by the Owner and listed in the plant records (see Section 7).	Change: Addition of requirement for test loops and fluid inventory to provisions needed to comply with the rules of the standard. Basis: Some pumps cannot be tested in their normal systems, these test loops are required. An adequate test fluid inventory is needed, in the test loop systems and/or sumps, to prevent dry running of pumps. Change: Addition of requirement to identify each pump to be tested and to list them in the plant records.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
		Basis: Record and report requirements, as detailed in Section 7000, are included in this paragraph for complete outline of Owners Responsibility.
IWP–1400 Bypass Loops Where a pump cannot practically be tested in its regular circuit, a bypass loop may be used.	3.2 Bypass Loops A bypass test loop may be used, provided the bypass is designed to recognize the pump manufacturer's operating conditions for minimum flow operation.	Change: Addition of provisions to consider limitations of minimum flow operation as stated by pump manufacturer. Basis: Extended operation at flows lower indicated as minimum flow by the pump manufacturer, may be damaging to the pump.
IWP–1500 Detection of Change The hydraulic and mechanical condition of a pump relative to a previous condition, can be determined by attempting to duplicate, by test, a set of a basic reference parameters. Deviations detected are symptoms of changes and, depending upon the degree of deviation, indicate need for further tests or corrective action.	2.1 Detection of Change The hydraulic and mechanical condition of a pump relative to a previous condition can be determined by attempting to duplicate by test a set of reference values. Deviations detected are symptoms of changes and, depen–ding upon the degree of deviation, indicate need for further tests or corrective action.	Change: Parameters to values. Basis: OM–6 outlines the establishment of specific reference values, rather than parameters. Text in Para. 2100 was changed accordingly.
IWA–9000 (IWP–2000 Deleted in 83W85) hot standby – see plant technical specifications inservice test – a test to determine the operational readiness of a component or system normal plant operating conditions – the operating conditions during reactor startup, operation at power, hot standby, and reactor	1.3 Terminology The following are provided to ensure a understanding of select terms used in this Part. inservice test – a test to determine the operational readiness of a pump instrument accuracy – the allowable inaccuracy of an instrument loop based on the square root of the sum of the squares of the inaccuracies of each instrument or component in the loop	Change: Article IWP–2000 was deleted and a glossary was added by IWA–9000 in the 83W85. OM–6 1300 a, d, h, and i exist in the glossary. OM–6 1300 b, c, d, and f exist only in OM–6. Definitions for "Inservice Life" and "normal plant operation" that were in IWP–2000 have been deleted although they exist in IWA–9000.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
cooldown to cold shutdown conditions. Test conditions are excluded.		Basis: For OM–6 1300 a, d, h, and i, the definitions are similar with the major difference being the example provided for routine servicing and the clarifier "that does not require disassembly or replacement of parts".
operating convenience – a provision to facilitate plant operation but not required to perform a specific function in shutting down a reactor to cold shutdown conditions or in mitigating the consequences of an accident	instrument loop – two or more instruments or components working together to provide a single output (e.g., a vibration probe and its associated signal conditioning and readout devices)	OM–6 1300 b, c, d, f, and g are added to clarify their use in the standard. "Inservice Life" and "normal plant operation" were deleted as they were not longer used.
operational readiness – the ability of a component or system to perform its intended function when required	operational readiness – the ability of a pump to perform its intended function	
passive valve – a valve that does not perform a mechanical motion during the course of accomplishing a system safety function	preservice test period – the period of time following completion of construction activities related to the pump, and prior to first electrical generation by nuclear heat, in which component and system testing takes place	
routine servicing (of a pump) – the performance of planned, preventive maintenance that does not require disassembly or replacement of parts	pump – a mechanical device used to move liquid	
system resistance – the hydraulic resistance to flow in a system	reference values – one or more values of test parameters measured or determined when the equipment is known to be operating acceptably	
test – a procedure to obtain information through measurement or observation to determine the operational readiness of a component or system while under controlled conditions	routine servicing – the performance of planned, preventive maintenance (e.g., replacing or adjusting valves in reciprocating pumps, changing oil, flushing the cooling system, adjusting packaging, adding packing rings or mechanical seal maintenance or replacement)	
	system resistance – the hydraulic resistance to flow in a system	

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
	4.1 Preservice Testing Each pump shall be tested during the preservice test period as required by this Part. This testing shall be conducted under conditions as near as practicable to those expected during subsequent inservice testing. Only one preservice test of each pump is required, except that the requirements of para. 4.4 shall be met.	Change: Addition of preservice testing requirement. Basis: Preservice testing is an integral part of preparations for inservice testing, as such it was felt that this standard would fall short without its specification.
	4.2 Inservice Testing Inservice testing in accordance with this Part shall commence when the pump(s) is required to be operable (see para. 1.1).	Change: New paragraph. Basis: This clarifies when inservice testing is to begin. It requires pump testing during all periods of time when the pumps are relied upon to perform a function as described in the scope.
IWP–3100 Inservice Test Procedure An inservice test shall be conducted with the pump operating at nominal motor name–plate speed for constant speed drives, and at a speed adjusted to the reference speed for variable speed drives. The resistance of the system shall be varied until either the measured differential pressure or the measured flow rate equals the corresponding reference value. The test quantities shown in Table IWP–3100–1 shall then be measured or observed and recorded as directed in this Subsection. Each measured test quantity shall then be compared with the reference value of the same quantity. Any deviations determined shall be compared with the limits given in Table IWP–3100–2 and the specified corrective action taken.	5.2 Test Procedure An inservice test shall be conducted with the pump operating at specified test reference conditions. The test parameters shown in Table 2 shall be determined and recorded as directed in this paragraph. The test shall be conducted as follows. (a) The pump shall be operated at nominal motor speed for constant speed drives and at a speed adjusted to the reference speed for variable speed drives. (b) The resistance of the system shall be varied until the flow rate equals the reference value. The pressure shall then be determined and compared to its reference value. Alternatively, the flow rate can be	Change: 5200 Preamble. Test procedure – format changed to put in a procedure format. No technical changes. Basis: Clarification of requirements. Change 5200 a. Pump speed – minor word changes. No technical changes. Basis: Format change only. Change: 5200 b. Setting system resistance is the intent of OM–6 and IWP, but OM–6 is more definitive. Basis: To expand the methodology for establishing system resistance.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
	varied until the pressure equals the reference value and the flow rate shall be determined and compared to the reference flow rate value.	Change 5200 c. Fixed resistance system – specific setting discussion for fixed resistance systems. Basis: Not covered by Section XI, IWP. An enhancement to current standard.
	(c) Where system resistance cannot be varied, flow rate and pressure shall be determined and compared to their respective reference values.	NOTE: 5200 b and c allow determination of differential pressure or flow rate whereas IWP implies these quantities must be measured. The reason for this change is to allow determination when instrumentation has not been installed or is impractical to install.
	(d) Pressure, flow rate, and vibration (displacement or velocity) shall be determined and compared with corresponding reference values. All deviations from the reference values shall be compared with the limits given in Table 3 and corrective action taken as specified in para. 6.1.	Change: 5200 d, first paragraph; comparison of test values to reference values and format changes. Basis: No technical changes.
	Vibration measurements are to be broad band (unfiltered). If velocity measurements are used, they shall be peak. If displacement amplitudes are used, they shall be peak–to–peak.	Change: 5200 d, second paragraph; new paragraph added. Basis: Since velocity measurements are now allowed by this standard it is necessary to specify that reading shall be peak. The peak–to–peak for displacement reading was previously in IWP–4510. Change: OM–6 requires reading to be broad band (unfiltered). Basis: The purpose of the vibration measurement is to determine change and to assess operational readiness. Filtered readings, if used could easily mask vibrations at frequencies other than the measured frequency.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis

IWP Paragraph No.

Table IWP–3100–1
Inservice Test Quantities

Quantity	Measures	Observe
Speed N (if variable speed)	X	
Inlet pressure P_i	X(1)	
Differential pressure ΔP	X	
Flow rate Q	X	
Vibration amplitude V	X	
Proper lubricant level or pressure		X
Bearing temperature T_b	X	

NOTE:
(1) Measure before pump startup and during test.

Corresponding OM–6 Paragraph No.

Table 2
Inservice Test Parameters

Quantity	Remarks
Speed: N	If variable speed
Differential Pressure: ΔP	Centrifugal Pumps, including vertical line shaft pumps
Discharge Pressure: P	Positive Displacement Pumps
Flow Rate: Q	
Vibration:	
Displacement, V_d	Peak–to–peak
Velocity, V_d	Peak

Change and Basis

Speed

Change: None.

Basis: NA

Inlet Pressure

Change: Not required by OM–6.

Basis: This parameter was deleted since there are no acceptance criteria. The only reason it exists in IWP is to "help" the owner set up his test and recognize that adequate suction pressure should be specified. OM–6 recognized that the owner is responsible to address testing limitations and that those limitations will be written into his procedures.

Differential Pressure

Change: None for centrifugal pumps.

Basis: NA

Pressure – (positive displacement pumps)

Change: Discharge pressure only, is required for positive displacement pumps (PD).

Basis: IWP requires differential pressure for both centrifugal and positive displacement pumps. Since discharge pressure is independent of inlet pressure for PD pumps, the requirement has been changed to require discharge pressure only as being the indicator of pump degradation.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
		Flow Rate
		Change: None.
		Basis: NA
		Vibration
		Change: Addition of velocity to the standard.
		Basis: IWP requires vibration amplitudes to be measured in terms of displacement, whereas OM–6 allows vibration amplitude in terms of displacement or velocity. Each can provide indication of pump degradation and should be allowed. Additionally velocity is the preferred parameter since it will generally provide the most information regarding pump operability.
		Proper Lubricant Level or Pressure
		Change: No longer required.
		Basis: This parameter is no longer required by OM–6. It was felt that it should be observed as part of regular maintenance practice, but that is has little meaning when associated with quarterly pump testing.
		Bearing Temperature
		Change: OM–6 does not require the quantity to be measured.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
		Basis: Bearing temperature increases rapidly until bearing failure, but only for those bearings outside of the pumped fluid flow flow path (bearings in separate housings). The main reason for deleting this requirement is that it is unlikely that bearing failure would be detected by a yearly test. The parameter is only indicative of pending pump bearing failure when it is continuously monitored which is not applicable for standby pumps.
IWP–3110 Reference Values Reference values are defined as one or more fixed sets of values of the quantities shown in Table IWP–3100–1, as measured or observed when the equipment is known to be operating acceptably. All subsequent test results shall be compared to these reference values or to new reference values established in accordance IWP–3111 and IWP 3112. Reference values shall be determined from the results of an inservice test, which may be run during preoperational testing, or from the results of the first inservice test run during power operation. Reference values shall be at points of operation readily duplicated during subsequent inservice testing.	4.3 Reference Values Reference values shall be determined from the results of preservice testing or from the results of inservice test. Reference values shall be at points of operation readily duplicated during subsequent tests. All subsequent test results shall be compared to these initial reference values or to new reference values established in accordance with paras. 4.4 and 4.5. Reference values shall only be established when the pump is known to be operating acceptably. If the particular parameter being measured or determined can be significantly influenced by other related conditions, then these conditions shall be analyzed.[a] a. Vibration measurements of pumps may be found–ation, driver, and piping dependent. Therefore, if initial vibration readings and high and have no obvious relationship to the pump, then vibration measurements should be taken at the driver, at the foundation, and on the piping and analyzed to ensure that the reference vibration measurements are representative of the pump and that the measured vibration levels will not prevent the pump from fulfilling its function.	Change: Editorial changes; deleted definition; added statement and footnote emphasizing the need to ensure that vibration readings represent pump characteristics and not those of the system or structural attachments. Basis: The definition of "reference value" is provided elsewhere in the standard. Pump vibration readings can be representative of other conditions unrelated to the mechanical condition of the pump. Change: Establish reference values from the results of preservice testing or the first inservice test. Basis: The intent is to allow more than one preservice test to be used in establishing reference values.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
IWP–3111 Effect of Pump Replacement, Repair and Routine Servicing on Reference Values After a pump has been replaced, a new set or sets of reference values shall be determined from the results of the first inservice test run after the pump is put into service. When a reference value or set of values may have been affected by repair or routine servicing of the pump, a new reference value or set of values shall be determined or the previous value reconfirmed by an inservice test run prior to, or within 96 hr after, return of the pump to normal service. Deviations between the previous and new set of reference values shall be identified, and verification that the new values represent acceptable pump operation shall be placed in the record of tests (IWP–6000).	4.4 Effect of Pump Replacement, Repair, and Maintenance of Reference Values. When a reference value or set of values may have been affected by a repair, replacement, or routine servicing of a pump, a new reference value or set of values shall be determined or the previous value reconfirmed by an inservice test run prior to declaring the pump operable. Deviations between the previous and new set of reference values shall be identified, and verification that the new values represent acceptable pump operation shall be placed in the record of tests (see Section 7).	Change: Merged the first two sentences; permits confirmation of existing reference values for pump replacement; requires testing of replacement pump identical to that required for repair. Basis: Editorial improvement: Makes testing requirements consistent. Change: A test, confirming reference values, must be run prior to returning a pump to service as compared to the Section XI requirement of running the test within 96 hours of returning the pump to service. Basis: This test is the basis of declaring the pump operable and, therefore, must be run prior to returning the pump to service.
IWP–3112 Establishing an Additional Set of Reference Values If it is necessary or desirable for some reason other than stated in IWP–3111, to establish an additional set of reference values, an inservice test shall first be run at the conditions of an existing set of reference values and the results analyzed. If operation is satisfactory, a second test run at the new reference conditions shall follow as soon as practical. The results of this test shall establish the additional set of reference values. Whenever an additional set of reference values is established, the reasons for so doing shall be justified and documented in the record of tests (IWP–6000).	4.5 To Establish an Additional Set of Reference Values If it is necessary or desirable, for some reason other than stated in para. 4.4, to establish an additional set of reference values, an inservice test shall first be run at the conditions of an existing set of reference values and the results analyzed. If operation is acceptable per para. [illegible], a second test run at the new reference conditions shall follow as soon as practicable. The results of this test shall tablish the additional set of reference values. Whenever an additional set of reference values is established, the reasons for so doing shall be justified and documented in the record of tests (see Section 7). The requirements of para. 4.3 apply.	Change: Minor editorial changes including reference to Paragraph 4400; added reference to basic requirements of reference values. Basis: Editorial improvement and clarification.

IWP Paragraph No.

Table IWP–3100–2 Allowable Ranges of Test Quantities

Test Quantity	Acceptable Range	Alert Range [Note (1)]		Required Action Range Note (1)]	
		Low Values	High Values	Low Values	High Values
P_i	[Note (2)]	[Note (2)]	[Note (2)]	[Note (2)]	[Note (2)]
ΔP	$0.93–1.02\Delta P_i$	$0.90–0.93\Delta P_i$	$1.02–1.03\Delta P_i$	$<0.90\Delta P_i$	$>1.03\Delta P_i$
Q	$0.94–1.02Q_r$	$0.90–0.94Q_r$	$1.02–1.03Q_r$	$<0.90Q_r$	$>1.03Q_r$
V when $0 < V_r < 0.5$ mils	0–1 mil	None	1–1.5 mils	None	>1.5 mils
V when 0.5 mils $< V_r < 2.0$ mils	$0–2V_r$ mils	None	$2V_r–3V_r$ mils	None	$>3V_r$ mils
V when 2.0 mils $< V_r \le 5.0$ mils	$0–(2 + V_r)$ mils	None	$(2 + V_r)–(4 + V_r)$ mils	None	$>(4 + V_r)$ mils
V when $V_r > 5.0$ mils	$0–1.4V_r$ mils	None	$1.4V_r – 1.8V_r$ mils	None	$>1.8V_r$ mils
T_b	[Note (3)]	[Note (3)]	[Note (3)]	[Note (3)]	[Note (3)]

NOTES:
(1) See IWP–3230.
(2) P_i shall be within the limits specified by the Owner in the record of tests (IWP–6000).
(3) T_b shall be within the limits specified by the Owner in the record of tests (IWP–6000).

Corresponding OM–6 Paragraph No.

Table 3a[1]

Pump Type	Pump Speed	Test Parameter	Acceptable Range	Alert Range	Required Action Range
Centrifugal and vertical line shaft [Note (2)]	<600 rpm	V_d or V_v	$\le 2.5\ V_r$	$>2.5\ V_r$ to $6\ V_r$ or >10.5 mils	$>6\ V_r$ or >22 mils
Centrifugal and vertical line shaft [Note (2)]	≥600 rpm	V_v or V_d	$\le 2.5\ V_r$	$>2.5\ V_r$ to $6\ V_r$ or >0.325 in./sec	$>6\ V_r$ or >0.70in./sec
Reciprocating		V_d or V_v	$\le 2.5\ V_r$	$>2.5\ V_r$ to $6\ V_r$	$>6\ V_r$

NOTES:
(1) Vibration parameter per Table 2. V_r is vibration reference value in the selected units.
(2) Refer to Figure 1 to establish displacement limits for pumps with speeds ≥600 rpm or velocity limits for pumps with speeds <600 rpm.

Change and Basis

Overview

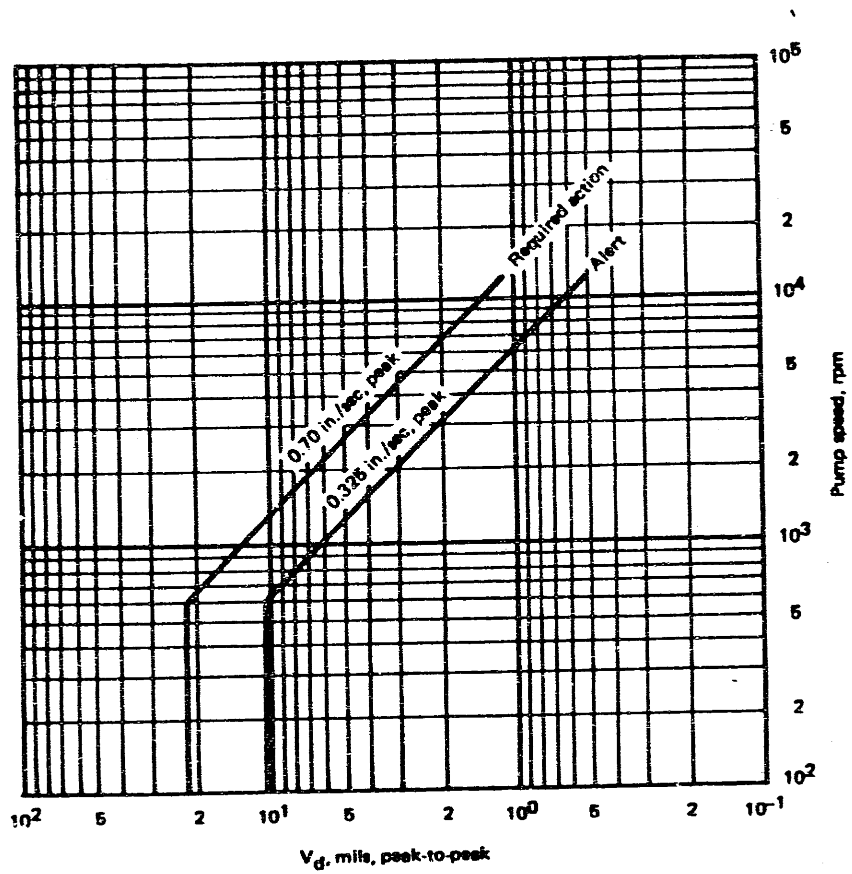
Early in the development of this standard, it was recognized there were many problems with detecting change in pump performances based on hydraulic parameters. Additionally, there were concerns raised about causing undue wear because of excessive testing. This is of particular concern for those pumps which have to be tested on a mini flow line. As a result, a change in emphasis was made towards using vibration as the primary indicator of pump degradation. Vibration testing is more inductive pump mechanical condition than hydraulic testing. It is expected there will be fewer pumps requiring increased testing or corrective action based on erroneous test results. A careful investigation showed that all pumps could not be treated the same. Thus Table 6100–1 differentiates between the three types of pumps; positive displacements, vertical line shaft, and centrifugal pumps.

While vibration is to be relied upon more heavily, it is also recognized that positive displacement and vertical line shaft pump degradation could go undetected with the typical instrumentation being used in most plants. Therefore, more stingent hydraulic acceptance criteria are still used for these pumps.

For all other centrifugal pumps, the hydraulic acceptance criteria have been relaxed. The extent of the change is to allow the equipment to

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
		be run in a "window "hydraulically and then to evaluate pump condition more closely with vibration. The purpose of the window is two fold. First, it ensures the pump is performing its primary function, that is, pumping liquid. Second, that it is operated in a specified narrow band where the vibration data will be comparable.
		Vibration Parameters
		Change: The table is divided into three groups: Centrifugal and vertical line shaft pumps less than 600 rpms; centrifugal and vertical line shaft pumps greater than or equal to 600 rpm, and reciprocating pumps.
		Basis: At slow speeds, less than 600 rpm, displacement is the better parameter to use and is recommended by specifying units in mils. At higher speeds, velocity is the best indicator and thus its units are specified in in/sec. Reciprocating pumps are in a class of their own due to the difficulties associated with vibration testing of this type of equipment.
		Change and Basis
		Change: Added fixed alert and required action values. Changed the multipliers for alert and required action based on reference values.
		Basis: The former Section XI testing philosophy, as required in IWP, sets alert and required action levels based on a reference value when the pump is known to be performing acceptably. This means of testing has the advantage of comparing the equipment to itself and thus telling

TABLE 3
RANGES FOR TEST PARAMETERS



degradation by change. It also has some disadvantages. For a very smooth running pump, it sets alert and required action levels restrictively low whereas for a rough running pump, it may set these values at such a high level that pump damage could occur before corrective action was required. OM–6 rectifies this situation by (1) setting upper levels of vibration such that increased testing or required action is required before damage will occur, and (2) by increasing values for alert and required action based on reference values. This increase will reduce the likelihood of premature maintenance.

The fixed values (0.325 and 0.7 in/sec) of vibration were determined by a task group established for this purpose. The task group reviewed behavior regimes (alert and action ranges) of international and Canadian government standards, and those of the major U.S. consultants and vendors.

Other non–mandatory standards such as OM–14 "requirements for Vibration Monitoring of Rotating Equipment" should be consulted for more details on vibration testing. These standards can be used to establish effective vibration monitoring programs.

Change: Figure 1 was added.

Basis: This conversion chart was provided to enable users to determine velocity acceptance limits for pump speeds less than 600 rpm and displacement acceptance limits for pump speeds greater than 600 rpm.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis

Table 3b

Test Parameter	Acceptable Range	Alert Range		Required Action Range	
		Low	High	Low	High
P (positive displacement pumps)	0.93 to $1.10P_r$	0.90 to $<.93P_r$	—	$<0.90P_r$	$>1.10P_r$
ΔP (vertical line shaft pumps)	0.95 to $1.10\Delta P_r$	0.93 to $<0.95\Delta P_r$	—	$<0.93\Delta P_r$	$>1.10\Delta P$
Q (positive displacement vertical line shaft pumps)	0.95 to $1.10Q_r$	0.93 to $<.95Q_r$	—	$<0.93Q_r$	$>1.10Q_r$
ΔP (centrifugal pumps)	0.90 to $1.10\Delta P_r$	—	—	$<0.90\Delta P_r$	$>1.10\Delta P$
Q (centrifugal pumps)	0.90 to $1.10Q_r$	—	—	$<0.90Q_r$	$>1.10Q_r$

GENERAL NOTE: The subscript r denote reference value.

Hydraulic Parameters

Change: "High" Alert values for hydraulic parameters were deleted.

Basis: The general consensus was that test failures that resulted from "higher than reference value" hydraulic measurements were caused by instrument fluctuations.

Change: "High" Required Action values were increased from 1.03 times reference to 1.10 time reference.

Basis: Although pump hydraulic performance is not expected to improve, it was decided to maintain a high value for Required Action. This assures that test repeatability is reasonable, thereby maintaining the quality of the vibration testing.

Change: "Low" Alert values for hydraulic parameters of centrifugal pumps were deleted.

Basis: As stated in the overview, these pumps are tested in a window of differential pressure and flow. This window insures that the vibration measurements will be comparable. Since the main emphasis of pump operability is based on vibration measurement, only that parameter has an alert range.

Change: "Low" Alert Range for flow rate measurements of Vertical Line Shaft Pumps and Positive Displacement Pumps was changed from (0.90 to 0.94) Q_r to (0.93 to 0.95) Q_r, respectively.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
		Basis: More stringent hydraulic acceptance criteria is necessary for these pumps. There are inherent deficiencies in vibration testing and degradation will be identified sooner through changes in hydraulic parameters.
		Change: "Low" Required Action values for flow rate measurements of Vertical Line Shaft Pumps were changed from 0.90 Q_r to 0.93 Q_r.
		Basis: Same as above.
		Change: "Low" Alert Range for differential pressure measurements of Vertical Line Shaft Pumps was changed from (0.90 to 0.93) P_r to (0.93 to 0.95) P_r.
		Basis: Same as above.
		Change: The measurement of discharge pressure was substituted for differential pressure for Positive Displacement Pumps.
		Basis: Discharge pressure is independent of inlet pressure. Therefore, variations in inlet pressure do not affect the discharge pressure. Discharge pressure is considered to be a more appropriate test parameter.
		Change: Note 1, regarding inlet pressure limits, was deleted.
		Basis: Differential pressure is considered to be a more meaningful hydraulic parameter for determining hydraulic performance degradation and

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
		5200 now allows "determination" of this parameter; therefore, a requirement to measure the inlet pressure and assure that it is within established limits is not appropriate.
		Change: The requirement to measure bearing temperature (T_b) and Note 2, regarding specified limits for bearing temperature measurements, were deleted.
		Basis: See discussion on bearing temperature (Table 2)
		Change: Note 3, a reference to IWP–3230 (Corrective Action), was deleted.
		Basis: This note did not provide any meaningful help.
IWP–3200 Analysis of Results	6 Analyses and Evaluation	Change: First sentence deleted.
IWP–3210 Allowable Ranges of Inservice Test Quantities	6.1 Acceptance Criteria	Basis: There is no need to state the purpose of the table which contains acceptance criteria (6100–1) as this is self explanatory.
The allowable ranges of inservice test quantities in relation to the reference values are tabulated in Table IWP–3100–2. If these ranges cannot be met, the Owner shall specify in the record of tests (IWP–6000) the reduced range limits allow the pump to fulfill its function, and those limits shall be used in lieu of the ranges given in Table IWP–3100–2.	If deviations fall within the alert range of Table 3, the frequency of testing specified in para. 5.1 shall be doubled until the cause of the deviation is determined and the condition corrected. If deviations fall within the required action range of Table 3, the pump shall be declared inoperable until the cause of the deviation has been determined and the condition corrected.	Change: Second sentence deleted.
	When a test shows deviation outside of the acceptable range of Table 3, the instruments involved may be recalibrated and the test rerun.	Basis: The working group could not endorse the current code's philosophy in letting the owner specify any acceptance criteria deemed appropriate when the limits of the applicable table could not be met. OM–6 requires the acceptance criteria to be met. There are provisions for the owner to review his testing results and, if justified, establish new reference values (4500).

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
IWP–3220 Time Allowed for Analysis of Tests	6.2 Time Allowed for Analysis of Test	
All test data shall be analyzed within 96 hr after completion of a test.	All test data shall be analyzed within 96 hr after completion of a test.	
IWP–3230 Corrective Action	6.1 (Same as Above)	Change: Deleted "and not returned to service" from Required Action Paragraph (b), and deleted entire Paragraph (c) dealing with type of corrective action (replacement repair, or analysis). Basis: IST requirements for returning an inoperable pump to service are covered in OM–6 5400, The type of corrective action (replacement, repairs, or analysis) and their effect on the reference values are covered in OM–6 4300, 4400, and 4500.
(a) If deviations fall within the Alert Range of Table IWP–3100–2, the frequency of testing specified in IWP–3400 shall be doubled until the cause of the deviation is determined and the condition corrected. (b) If deviations fall within the Required Action Range of Table IWP–3100–2, the pump shall be declared inoperative and not returned to service until the cause of the deviation has been determined and the condition corrected. (c) Correction shall be either replacement or repair per IWP–3111, or shall be an analysis to demonstrate that the condition does not impair pump operability and that the pump will still fulfill its function. A new set of reference values shall be established after such analysis. (d) When test shows deviations greater than allowed (see Table IWP–3100–2), the instruments involved may be recalibrated and the test rerun.		
IWP–3300 Scope of Tests	No equivalent paragraph	Change: New format.
Each inservice test shall include the measurement and observation of all quantities in Table IWP–3100–1 except bearing temperatures, which shall be measured during at least one inservice test each year.		Basis: Similar requirements (except bearing temperature) are covered under OM–6 5200. See discussion on bearing temperature (Table 5200–1).

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
IWP–3400 Frequency of Inservice Tests	5. Testing Methods	Change: Deleted (b), rewording no technical changes, coverage of all elements included in other paragraphs.
(a) An inservice test shall be run on each pump nominally every 3 months during normal plant operation. It is recommended that this test frequency be maintained during shutdown accomplished, although this is not mandatory. If it is not tested during plant shutdown, the pump shall be tested within 1 week after the plant is returned to normal operation. (b) Pumps that are operated more frequently than every 3 months need not be run or stopped for a special test, provided the plant log shows each such pump was operated at least once every 3 months at the reference conditions, and the quantities specified were measured, observed, recorded, and analyzed.	5.1 Frequency of Inservice Tests An inservice test shall be run on each pump, nominally every 3 months, except as provided in periods if this can reasonably be paras. 5.3, 5.4, and 5.5. 5.3 Pumps in Regular Use Pumps that are operated more frequently than every 3 months need not be run or stopped for a special test provided the plant records show each such pump was operated at least once every 3 months at the reference conditions, and the quantities specified were Change: Deleted (b), rewording no technical Section 6. 5.4 Pumps in Systems Out of Service sentence of (a) covered in 5400. or not required to be operable, the test Change: Recommendation in IWP–3400a 3 months prior to placing the system in an operable status, the pump shall be tested and the test schedule followed in accordance with the requirements of this Part. Pumps which can only be tested during plant operation shall be tested within 1 week following plant startup.	Basis: (b) is covered in 5300, third Change: Recommendation in IWP–3400a concerning test frequency has been deleted. Added first sentence of 5400. Basis: Generally, recommended good practices have been deleted from this standard. This is not to imply the recommendation is not valid but to place more emphasis on the mandatory requirements.
No Corresponding IWP Paragraph	5.5 Pumps Lacking Required Fluid Inventory Pumps lacking required fluid inventory (e.g., pumps in dry sumps) need not be tested in accordance with this Part every 3 months.	Change: Paragraph added to address testing of pumps that are normally in dry sumps. Basis: These pumps are not adequately tested by IWP.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
	These pumps shall be tested at least once every 2 years except as provided in para. 5.4. The required fluid inventory shall be provided during this test.	
IWP–3500 Duration of Test	5.6 Duration of Tests	Change: Removed reference to bearing temperature and bearing temperature measurement (OM–6 5200).
(a) When measurement of bearing temperature is not required, each pump shall be run at least 5 min under conditions as stable as the system permits. At the end of this time at least one measurement or observation of each of the quantities specified shall be made and recorded. (b) When measurement of bearing temperature is required, each pump shall be run until the bearing temperatures (IWP–4310) stabilize, measured or observed and recorded. A bearing temperature shall be considered stable when three successive readings taken at 10 min. intervals do not vary by more than 3%.	After pump conditions are as stable as the system permits, each pump shall be run at least 2 min. At the end of this time at least one measurement or observation of each of the quantities required shall be made and recorded.	Basis: This parameter is no longer measured otherwise, this paragraph has not changed. Change: Minimum pump run time changed from five minutes to two minutes. Basis: Two minute run time, after system stabilization, is considered adequate.
IWP–4100 Instruments	4.6 Instrumentation	Change: Elimination of temperature from Table 4110–1 (4600–1).
IWP–4110 Quality	4.6.1 General	
Instrument accuracy shall be within the limits of Table IWP–4110–1. Station instruments meeting these requirements shall be acceptable.	4.6.1.1 Quality. Instrument accuracy shall be within the limits of Table 1. Station instruments meeting these requirements are acceptable.	Basis: Temperature is no longer required to be measured by this standard. See Table 5200–1.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
Table IWP–4110–1	Table 1	
Acceptable Instrument Accuracy	Acceptable Instrument Accuracy	
Pressure ±2% of full scale Differential pressure ±2% of full scale Flow rate ±2% of full scale Speed ±2% of full scale Temperature ±5% of full scale Vibration amplitude ±5% of full scale	Percent Quantity [Note(1)] Pressure ±2 Flow rate ±2 Speed ±2 Vibration ±5 Differential pressure ±2 NOTE: (1) Percent of full scale for individual analog instruments, percent of total loop accuracy for a combination of instruments, or over the calibrated range for digital instruments.	
IWP–4120 Range The full–scale range of each instrument shall be three times the reference value or less	4.6.1.2 Range (a) The full–scale range of each analog instrument shall be not greater than three times the reference value. (b) Digital instruments shall be selected such that the reference value shall not exceed 70% of the calibrated range of instrument. (c) Vibration instruments are excluded from transmitters, if used" and added "and above.	Change: "Shall not be greater than three times". Basis: This change was made for clarity only. The intent of maintaining a proper instrument range remains the same.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
IWP–4130 Instrument Location Transmitters, Computers Instruments shall be located at their input source and read directly or via video transmission, or transmitters may be used. Instrument outputs may be fed directly into a computer for processing and indication or digital printout.	No Corresponding Paragraph	Change: Elimination of "Transmitters, Computers". Basis: This standard is concerned primarily with sensor/transmission response versus transmission. Generally recommended good practices have been deleted from this standard. It is the owner's responsibility to determine instrument location, transmitters, and computers. See OM–6 4613/IWP 4230 for instrument location requirements.
IWP–4140 Calibration Instruments, together with their transmitters if used, shall be calibrated either prior to the establishment of reference quantities or on a regular basis as established by the owner. New or repaired instruments shall be calibrated prior to test use. A system of calibration records shall be used to identify each instrument and its date of calibration or, alternatively, each instrument may contain an attached tag or sticker that records the date of last calibration.	4.6.1.4 Calibration. Instrument and instrument loops shall be calibrated in accordance with the owner's quality assurance program. New or repaired instruments shall be calibrated prior to test use.	Change: Specify Owner's QA Program Basis: All licensed nuclear power plants must address quality assurance per 10 CFR 50, Appendix B. Therefore, referencing the owner's quality assurance program: a: Ensures a control program the owner can easily implement. b: Is consistent with federal law. c: Makes this portion of the standard easy to interpret Change: Eliminate "together with their trainsmitters, if used: and added :and Instrument Loops". Basis: Use of Instrument Loops as defined in 1300 encompasses the instrument transmitters and any other component which makes up the loop. This clarifies the intent of the standard.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
IWP–4150 Fluctuations Symmetrical damping devices or averaging techniques may be used to reduce instrument fluctuations to within 2% of the observed reading. Hydraulic instruments may be damped by using gage snubbers or by throttling small valves in instrument lines. If throttling small valves in instrument lines. If throttling of small valves is used, the operator shall alternately open and close the valve several times to verify unobstructed pressure communication while observing the instrument reading.	4.6.5.1 Fluctuations. Symmetrical damping devices or averaging techniques may be used to reduce instrument fluctuations. Hydraulic instruments may be damped by using gauge snubbers or by throttling small valves in	Change: Elimination of "2 percent" and "If throttling of small valves is used, the operator shall alternately open and close the valve several times to verify unobstructed pressure communication while observing the the instrument reading". Basis: Pump testing focuses on repeatable steady state results. The use of damping devices assists the operator in determining these results. Damping can be useful in reading an instrument that is dynamically fluctuating but is in steady state. The working group's review revealed that there was no need to restrict the use of damping to within 2 percent of the observed reading. Pressure communication is established when instrument response is observed. It is expected that the owner will be use good technical practices and will be able to determine when an instrument line is totally obstructed/damped. The above instructions do not need to be incorporated into code.
IWP–4160 Position Sensitive Instruments Instrument that are position sensitive shall be either permanently mounted, or provision shall be made to duplicate the position for each position for each test.	4.6.1.3 Instrument Location. The sensor location shall be established by the owner, documented in the plant records (see Section 7), and shall be appropriate for the parameter being measured. The same location shall be used for subsequent tests. Instrument that are position sensitive shall be either permanently mounted or provision shall be made to duplicate their position during each test.	Change: Last sentence, none. Basis: NA. NOTE: The first two sentences of OM 4613 are applicable to IWP–4130. See that section for changes and basis.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
IWP–4200 Pressure Measurement	4.6.2 Pressure Measurement	Change: None.
IWP–4210 Gage Lines	4.6.2.1 Gage Lines. If the presence or absence of liquid in a gage line could produce a difference of more than 0.25% in the indicated value of the measured pressure, means shall be provided to assure or determine the presence or absence of liquid as required for the static correction used.	Basis: NA.
If the presence or absence of liquid in a gage line could produce a difference of more than 1/4% in the indicated value of the measured pressure, means shall be provided to ensure or determine the presence or absence of liquid as required for the static correction used.		
IWP–4220 Pressure Tap Construction	No Corresponding Paragraph	Change: Paragraph deleted.
Pressure tap shall be flush with, and normal to, the wall of the liquid passage.		Basis: This standard should not delineate construction details. Paragraph 4600, OM–6 adequately describes instrumentation requirements applicable to this standard.
IWP–4230 Pressure Tap Location	4.6.1.3 Instrument Location. The sensor location shall be established by the owner, documented in the plant records (see Section 7), and shall be appropriate for the parameter being measured. The same location shall be used for subsequent tests. Instrument that are position sensitive shall be either permanently mounted or provision shall be made to duplicate their position during each test.	Change: Paragraph rewritten.
Pressure taps shall be located in a section of the flow path that is expected to have reasonably stable flow as close as practical to the pump. Any line valves between inlet and discharge pressure taps shall be in an open position during the inservice test.		Basis: The working group feels that Article 4613 adequately covers instrument requirements. It is not the intent of a testing code to specify design requirements.
		Change: Additional documentation.
		Basis: This paragraph better defines quality control for instrumentation. The additional requirements help ensure repeatable pump test results by documenting instrument location and instrument type.

IWP Paragraph No.	Corresponding OM-6 Paragraph No.	Change and Basis
IWP-4240 Differential Pressure The differential pressure across a pump shall be determined by use either of a differential pressure gage or a differential pressure transmitter that provides direct measurement of pressure difference, or by taking the difference between the pressure at a point in the inlet pipe and the pressure at a point in the discharge pipe.	4.6.2.2. Differential Pressure. When determining differential pressure across a pump, a differential pressure gauge, a differential pressure transmitter that provides direct measurement of pressure difference or the difference between the pressure at a point in the inlet pipe and the pressure at a point in the discharge pipe, may be used.	Change: None. Basis: NA.
IWP-4300 Temperature Measurement IWP-4310 Bearings The temperature of all centrifugal pump bearings outside the main flow path and of the main shaft bearings of reciprocating pumps shall be measured at points selected to be responsive to changes in the temperature of the bearing. Lubricant temperature, when measured after passing through the bearing, and prior to entering a cooler, shall be considered the bearing temperature.	No Corresponding Paragraph	Change: Paragraph deleted. Basis: Bearing temperature is no longer required to be measured in Table 5200-1.
IWP-4400 Rotative Speed Measurement For all pumps directly coupled to motor drivers of either the synchronous or the induction type, the rotative shaft speed need not be measured. When any other type of driver or variable speed coupling is used, the rotative speed of the pump shaft shall be determined by measurement.	4.6.3 Rotational Speed Measurements. Rotational speed measurements of variable speed pumps shall be taken by a method which meets the requirements of para. 4.6.1.	Change: The paragraph has been made specifically applicable to variable speed pumps. No reference is made to pumps with variable speed couplings. Basis: The working group does not feel it is necessary to treat pumps with variable speed couplings differently from other variable speed pumps. Paragraph 5200a of OM-6 covers pumps with constant speed drivers.

IWP Paragraph No.	Corresponding OM-6 Paragraph No.	Change and Basis
IWP-4500 Vibration IWP-4510 Vibration Amplitude At least one displacement vibration amplitude (peak-to-peak composite) shall be read during each inservice test. The direction of displacement shall be measured in a plane approximately perpendicular to the rotating shaft, and in the horizontal or vertical direction that has the largest deflection for the particular pump installation. The location shall generally be on a bearing housing or its structural support, provided it is not separated from the pump by any resilient mounting. On a pump coupled to the driver, the measurement shall be taken on the bearing housing near the coupling; on close-coupled pumps, the measurement point shall be as close as possible to the inboard bearing. On reciprocating pumps, the location shall be on the bearing housing of the main pump drive shaft, approximately perpendicular to both the shaft and the line of plunger travel.	4.6.4 Vibration Measurements (a) On centrifugal pumps, measurements shall be taken in a plane approximately perpendicular to the rotating shaft in two orthogonal directions on each accessible pump bearing housing. Measurement also shall be taken in the axial direction, on each accessible pump thrust bearing housing. (b) On vertical line shaft pumps, the measurements shall be taken on the upper motor bearing housing in three orthogonal directions, one of which is the axial direction. (c) On reciprocating pumps, the location shall be on the bearing housing of the crankshaft, approximately perpendicular to both the crankshaft and the line of plunger travel. (d) If a portable vibration indicator is used, the reference points must be clearly identified on the pump to permit subsequent duplication in both location and plane.	Change: The paragraph has been restructed so that each type of pump is discussed in a separate subparagraph. Basis: This clarifies and highlights the requirements for each type of pump. Change: Subparagraph "a" requires vibration monitoring in two orthogonal directions on each accessible pump bearing (versus one direction on one bearing) plus in the axial direction on each accessible pump thrust bearing, which was not previously required. Basis: As more emphasis is being placed on vibration as the primary indicator of pump condition, the working group wanted to make sure that the information was the best available. Measurement in two orthogonal directions is considered necessary as a change in one direction may not be reflected in the other direction, and thus a significant change may be missed if only one direction is measured. This is also true for axial measurements, but it is only necessary to do axial measurements on thrust bearings. The same logic is behind the requirement to get the best information on pump condition within the limits of practicality. Change: Subparagraph "b" is new and addresses vertical line shaft pumps. Basis: Vertical line shaft pumps are now addressed as a separate class of pumps. This recognizes their special characteristics.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
		The only accessible bearing for this type of pump is the top motor bearing, as the pump bearings are generally under water. The top motor bearing is also the thrust bearing for the pump, thus the requirement for axial measurement. Change: Subparagraph "c" is essentially unchanged. Change: Subparagraph "d" has been relocated from IWP–4520 (a)(2). Basis: The working group feels that this is more logical place to put this requirement as the rest of the requirements from IWP–4520 have been deleted or moved to 4610.
IWP–4520 Instruments to Measure Amplitude (a) One of the following types of instrument shall be used: (1) a seismic transducer with transmission to a remote readout location; (2) a portable vibration indicator that clearly identifies the probe or measurement reference point to permit subsequent duplication in both location and plane; (3) an appropriately calibrated proximity measuring instrument that is designed for detecting the radial deflection of the rotating shaft or coupling.	No Corresponding Paragraph (EXCEPT IWP–4520b)	Change: The type of instruments to be used for vibration measurement is no longer specified. IWP–4520 (a)(2) has been relocated to 4640 d. Basis: The working group feels that there is no need to specify the type of instrument to be used as long as it meets the requirements of the 4600. The owner is responsible for instrument selection to meet the requirements of the standard. Change: Frequency range changed from "1/2 minimum speed to at least maximum pump shaft rotational speed" to "1/3 minimum pump shaft rotational speed to at least 1000 Hertz".
(b) The frequency response range of the readout system shall be from one–half minimum speed to at least maximum pump shaft rotational speed.	4.6.1.6 Frequency Response Range. The frequency response range of the vibration	Basis: This change corresponds with B&K publication on measuring vibration for

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
	measuring transducers and their readout system shall be from one–third minimum pump shaft rotation speed to at least 1000 Hz.	"velocity" testing further, this range of frequencies will more adequately envelop all potential noise contributors.
IWP–4600 Flow Measurement Flow rate shall be measured using a rate or quantity meter installed in the pump test circuit. The meter may be in any class that provides an overall readout repeatability within the accuracy limits of Table IWP–4110–1. Where the meter does not indicate the flow rate directly, the record shall include the method used to reduce the data.	4.6.5 Flow Rate Measurement. When measuring flow rate, use a rate or quantity meter installed in the pump test circuit. If a meter does not indicate the flow rate directly, the record shall include the method used to reduce the data.	Change: Reference to table of acceptable instrument accuracy requirements removed. Basis: This change is not substantive since the requirements are still contained in 4611 and still apply to the instrument.
IWP–6200 Requirements IWP–6210 Summary Listing A list of pumps shall be maintained to record the current status of the test program. When the quantities measured during a test are within the acceptable range of Table IWP–3100–2, only the date of the successful test is required to be listed.	No Corresponding Paragraph	Change: Deleted paragraph "Summary Listing:. Basis: Summary listing is not necessary as the information required under this paragraph is contained in other required records.
IWP–6220 Pump Records The owner shall maintain a record, which shall include the following, for each pump covered by this Subsection: (a) the manufacturer and the manufacturer's model and serial or other identification number; (b) a copy or summary of the manufacturer's acceptance test report, if available.	7.1 Pump Records The Owner shall maintain a record which shall include the following for each pump covered by this Part: (a) the manufacturer and the manufacturer's model and serial or other identification number; (b) a copy or summary of the manufacturer's acceptance test report if available;	Change: Added Paragraph 7100 (c) requiring a copy of the pump manufacturer's operating limits. Basis: It is necessary to know the manufacturer's operating limits in case they are more restrictive than the alert ranges or required actions ranges contained in this standard

IWP Paragraph No.	Corresponding OM-6 Paragraph No.	Change and Basis
	(c) a copy of the pump manufacturer's operating limits.	
IWP-6230 Inservice Test Plans The test plans shall include the following: (a) the hydraulic circuit to be used; (b) the location and type of measurement for each of the required test quantities; (c) the reference values (Table IWP-3100-1), limits of P_i and T_b (Table IWP-3100-2), and any other values required by this Subsection.	7.2 Inservice Test Plans The owner shall maintain a record of test plans and procedures which shall include the following: (a) the hydraulic circuit to be used; (b) the location and type of measurement for the required test parameters; (c) the reference values; (d) the method of determining reference values which are not directly measured by instrumentation.	Change: 7200 d was added to require documentation in the test plans of any method used to determine reference values. Basis: 4300 now allows the "determination" of reference values, as opposed to measurement or observation; therefore, it is necessary to provide the details of the method used for those that are determined.
IWP-6240 Record of Tests The record shall include the following: (a) date of test (b) measured and observed quantities (c) identification of instruments used (d) comparisons with allowable ranges of test values and analysis of deviations (e) requirements for corrective action (f) signature of the person or persons responsible for conducting and analyzing the test	7.3 Record of Tests The Owner shall maintain a record of each test which shall include the following: (a) pump identification; (b) date of test; (c) reason for test (e.g., post-maintenance, routine service test, establishing reference values); (d) values of measured parameters; (e) identification of instruments used; (f) comparisons with allowable ranges of test values and analysis of deviations; (g) requirement for corrective action; (h) evaluation and justification for changes to reference values; (i) signature of the person or persons responsible for conducting and analyzing the test.	Change: 7300 requirements were expanded to include pump identification (7300 a), reason for test (7300 c) and evaluation/justification for changes to reference values (7300 h). Basis: These were added to improve continuity from test to test and improve record keeping in general.

IWP Paragraph No.	Corresponding OM–6 Paragraph No.	Change and Basis
IWP–6250 Record or Corrective Action The record shall include a summary of the corrections made, the subsequent inservice test, confirmation of operational adequacy (IWP–3111), and the signature of the and verification of results.	7.4 Record of Corrective Action The owner shall maintain records of corrective action which shall include a summary of the corrections made, the subsequent inservice tests and confirmation of operational adequacy (see para. 4.4), and the signature of the individual responsible for corrective action and verification of results.	Change: None.

INTRODUCTION TO ASME/ANSI OMA–1988, PART 6 BASIS OF THE NEW VIBRATION MEASUREMENT CRITERIA AND REQUIREMENTS OF PART 6

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INTRODUCTION

An ideal pump would produce no vibration because all the energy would go to pumping liquid. In practice, vibration occurs when the pump's elements react against each other and energy is dissipated.

As a pump wears, changes in its dynamic properties begin to occur. These changes are reflected in an increase in vibration energy. This increased vibration energy may excite resonances and put extra dynamic forces on bearings causing more wear. Cause and effect reinforce each other, and lead ultimately to breakdown. By measuring the vibration at the pump elements which restrain the dynamic forces, such as bearing housings, the health of the pump can be inferred. This is the basis for using vibration for pump condition monitoring.

This paper describes the basis of the new vibration measurement criteria and requirements in Part 6, "Inservice Testing of Pumps in Light–Water Reactor Power Plants." The topics covered are: the background of why this change was necessary, classification of pumps typically used in nuclear power plants, a survey of vibration criteria, and the rationale for the new vibration measurement criteria and requirements in Part 6. A list of references and a bibliography are also included.

BACKGROUND

When the 0 & M Working Group on Pumps and Valves first started writing Part 6 as a replacement for Subsection IWP of Section XI, a decision was made that the new standard should, to the greatest extent possible, have a sound technical basis. This was particularly important in the area of vibration measurement. The vibration measurement criteria and requirements of IWP have not changed since IWP was added to Section XI in 1974. The vibration criteria contained in IWP were derived from experience, and were not supported by a technical basis.

When the Working Group started reviewing IWP, it became apparent that the vibration criteria needed revision. Industry consensus was that, in most cases, vibration velocity gave the best overall indication of machine condition. Many utilities were using vibration velocity in preventative and predictive maintenance programs. IWP required (and still requires) vibration to be measured as displacement. This required utilities to take an additional, and in their view, useless measurement. This was the greatest impetus for change in the vibration measurement criteria of IWP.

A Task Group on Vibration Monitoring was formed. The Task Group was asked to make recommendations to the Working Group on Pumps and Valves on vibration measurement criteria and requirements. The recommendations of that Task Group have formed the basis for the vibration measurement criteria and requirements of Part 6. This paper draws heavily on the work of that Task Group.

CLASSIFICATIONS OF PUMPS USED IN NUCLEAR POWER PLANTS

As the first step in preparation of the new vibration measurement criteria and requirements of Part 6, the types of pumps encountered in nuclear power plants were analyzed. Typical applications and types of pumps were reviewed. The results of this review served a two-fold purpose; it provided a basis for categorizing pumps and assigning appropriate classifications, and it gave an indication of the scope of coverage of the Standard.

The data collected is summarized in Table 1. Table 2 was derived from the maximum variances in Table 1. Table 2 lists ranges of data for each type of pump and classifies each type of pump in accordance with the guidance given in ISO 2372, Annex A.[1] It can be seen that all the centrifugal pumps fall into Class III. Reciprocating pumps are Class V. This result indicated that it would be possible to specify one set of vibration criteria for all centrifugal pumps.

VIBRATION CRITERIA

Next, various vibration standards in current use were reviewed. Standards for evaluating machine condition based on vibration levels take two forms. Some standards evaluate equipment based on a set of absolute criteria, usually in the form of a vibration severity chart. Other standards evaluate equipment based on

Table 1. Nuclear power plant pumps

Type	Class	Speed (rpm)	Horse Power	Flow (gpm)	Discharge Pressure (psig)		Description of Device
Vertical line shaft	III	300–900	300–750	To 8000		BWR service water	
		1800/3600	275	3215	100	PWR recirculation Spray inside Containment	
		1800/3600	293	3000	100	PWR recirculation spray Outside containment	
		1800	500	11500	55	PWR service water	
Centrifugal							
Vertical	III	1800/3600	500–1000	7700	182	PWR residual heat removal	
		1800/3600	500–1000	4270	290	BWR core spray	
		1800/3600	500–1000	150	2735	PWR high head safety injection	
		1800/3600	500–1000	3000	300	PWR low head safety Injection	
Horizontal (motor or turbine driven)	III	1800/3600	500–1300	1500	1000	Service not specified	
		1800/4500	460–4600	to 4500	1000–2200		Service not specified
		1800/3600	223	3000	100	PWR Containment Spray	
		1800/4260		340	1255–1430		PWR Auxiliary Feed Water
		3600	800	4000	250	BWR RHR service water	
Reciprocating	V	200	200	130	5000	PWR charging	
		520	75	50	1150	BWR standby liquid control	

Table 2. Range of parameters for nuclear power plant pumps

Type	Class	Speed (rpm)	Horse Power	Capacity Flow (gpm)	Discharge Pressure (psig)
Vertical line shaft	III	300–3600	275–750	To 8000	To 100
Centrifugal					
Vertical	III	1800/3600	500–1000	150–7700	182–2735
Horizontal	III	1800–4500	2230–4600	340–4500	100–2200
Reciprocating	V	200–520	75–200	50–130	1150–500

changes in vibration from a base or reference condition. Both alternatives were investigated.

Copies of vibration severity charts from three sources are included. They are representative of many of the absolute criteria reviewed, and illustrate typical details of presentation.

The vibration severity chart proposed by Energy Research and Consultants Corporation,[2] Figure 1, is based on data from pump experiments and was derived on the basis of extensive troubleshooting experience. It should be noted that the majority of the experience used to generate this chart was derived from turbine driven feed pumps.

The vibration severity chart proposed by J. S. Mitchell,[3] Figure 2, is designed for measurements taken on the casing of machines with casing–rotor weight ratios on the order of 5:1. Typical machines to which this severity chart can be applied include electric motors, pumps, fans, steam turbines and horizontally–split centrifugal compressors with external bearings supported through approximately 180°. Use of this chart is limited to operating speeds from 900–6000 rpm and bandwidth of the overall unfiltered measurements of approximately 1,000 Hz.

The vibration severity chart proposed by IRD Mechanalysis,[4] Figure 3, is in close agreement with Figure 2. The limitations of use for this chart are not specified.

A number of other sources were also reviewed.[5, 6, 7, 8, 9, 10, 11] Figure 4 plots the vibration criteria from several sources against each other. Note that all criteria have been converted to velocity in in./s, peak. This figure is illustrative of the amount of variability that exists among the various sources. It should also be noted that the ISO[1] and VDI[5] standards apply to machines with operating speeds from 600 to 12,000 rpm, and measurements limited to a frequency range of 10 to 1,000 Hz. Other sources did not state specific limitations or the basis of the criteria.

While the use of absolute criteria may, at first, seem to be attractive, it must be remembered that any generally applied vibration severity chart is, of necessity, a compromise. Such a chart must cover many types of machines operated in many different environments. If we consider only pumps in nuclear power plants, we find that they are not always typical of the type of machines on which the vibration severity charts are based. In addition, pumps in nuclear power plants are often tested at off–normal conditions (e.g., with the pump on a miniflow). This makes the application of a single set of criteria less than ideal. In addition to these limitations, absolute criteria can neglect significant changes in a pump which was originally running smoothly, as the change may leave the pump in the acceptable range. Although such a change may be below some arbitrary limit, it may, nonetheless, indicate a significant mechanical change in the pump.

Recognizing the limitations of absolute criteria, several sources recommend the use of relative change as a much more reliable indicator of actual machine condition. Bruel & Kjaer[12] recommends that an increase of a factor of 2.5 be considered a significant change warranting investigation and an increase of a factor of 10 is serious and may warrant the machine's shutdown.

VIBRATION MEASUREMENT CRITERIA AND REQUIREMENTS OF PART 6

Based on the analysis of vibration criteria given above, the Committee decided that relative change was

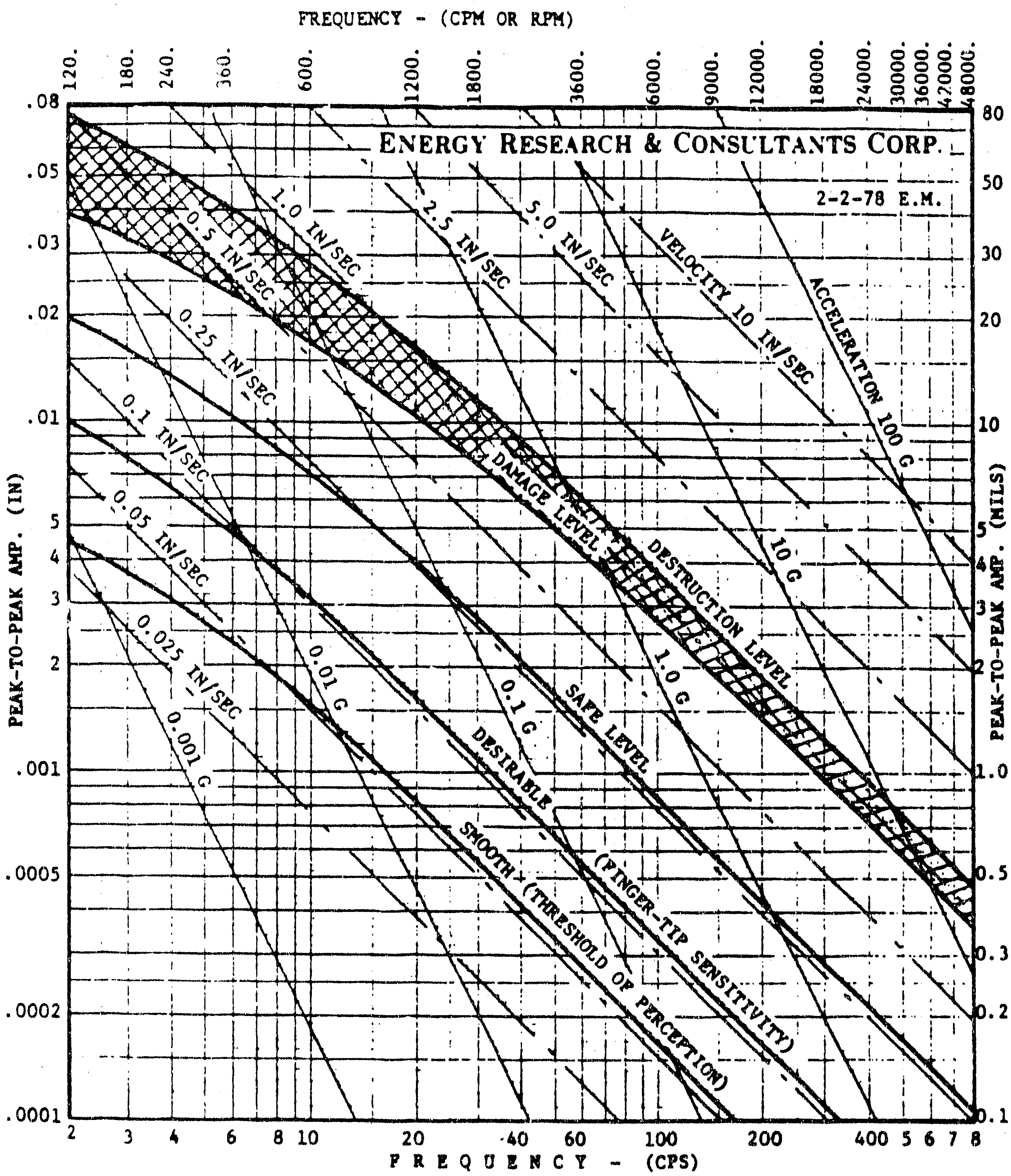

Figure 1. Allowable rotor vibration levels measured relative to the bearing cap (values shown are filtered readings at that particular RPM or frequency; experimental standards).

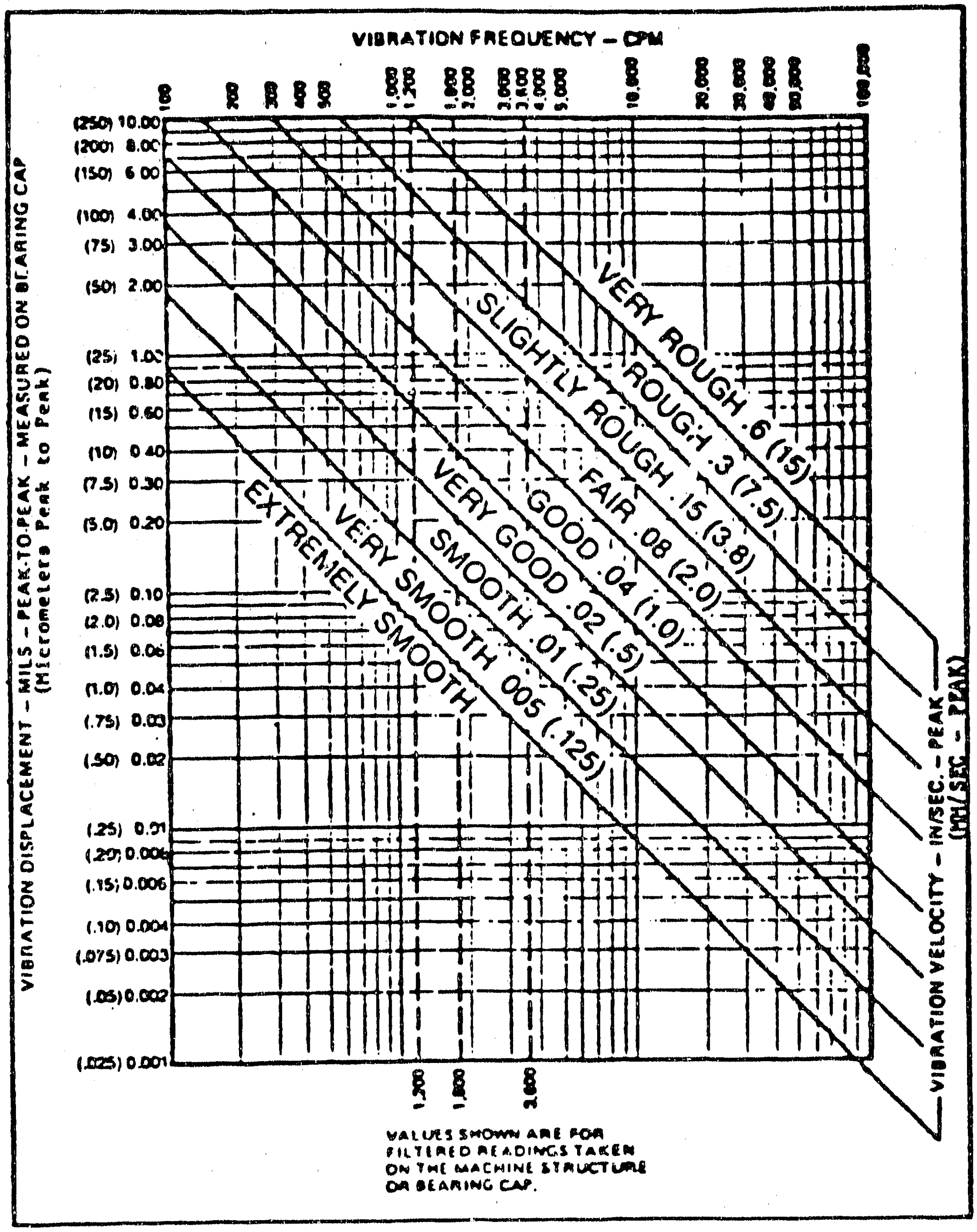

Figure 2. Casing (bearing cap) vibration severity chart. Source: An introduction to machinery analysis and monitoring, J. S. Mitchell, PennWell Books (Tulsa), 1981.

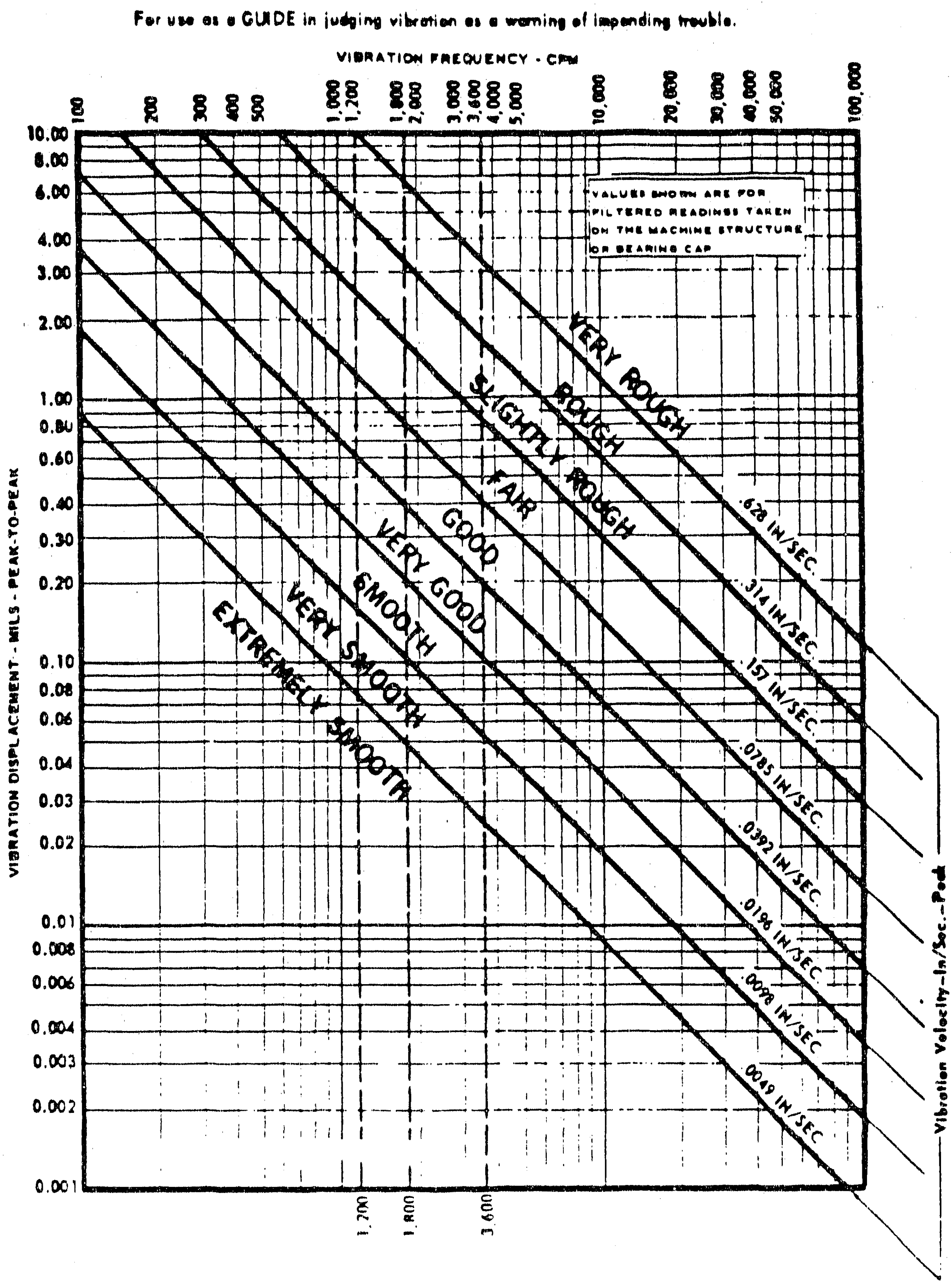

Figure 3. General machinery vibration severity chart.

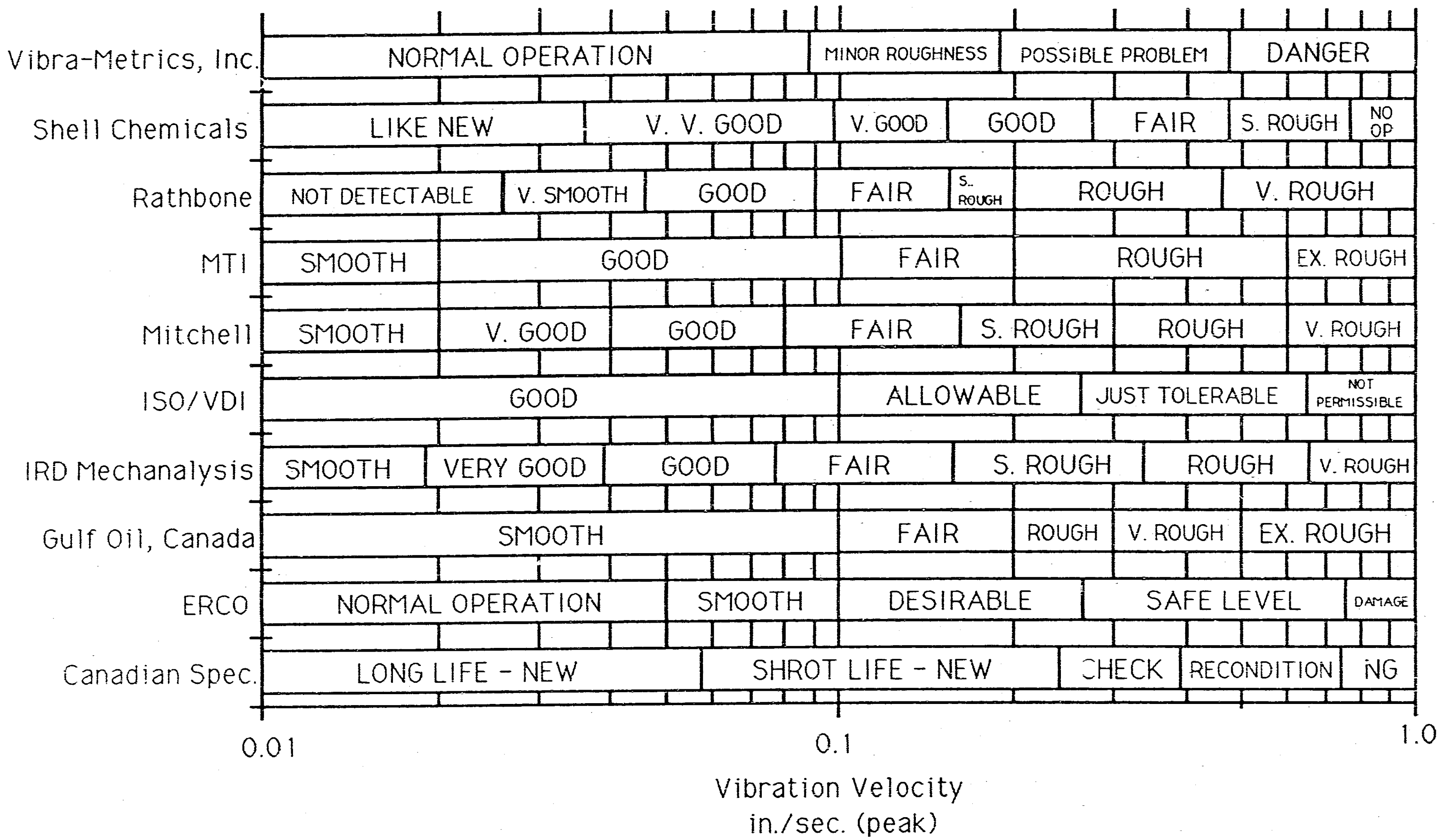

Figure 4. Vibration criteria.

the best indicator of actual pump health for nuclear power plant pumps. Examination of the ISO 2372[1] and VDI 2056[5] standards show that a change of one quality class (e.g., A to B) represents an increase in vibration by a factor of 2.5 and a change of two quantity classes (e.g., A to C) represents an increase in vibration by a factor of 6.3. Such a change could take a pump from "Allowable" to "Not Permissible".

The factors of change used in Part 6 were set conservatively at 2.5 times the reference value for "Alert" and 6 times the reference value for "Required Action". This is reflected in Table 3 of Part 6. One other advantage of using a relative change criteria is that it is applicable, with no modifications, to vibration measurements made in any units which use a linear scale.

The Committee also felt that there are vibration levels above which no pump should be operated. When the Committee looked at all the available sources it found some difficulty in translating the various terminologies used into the Code format of "Alert" and "Required Action" ranges. It also became obvious that there was some general agreement on the maximum permissible vibration. Table 3 is an attempt to derive "Alert" and "Required Action" ranges from the data of Figure 4. Figure 5 is the same as Figure 4, except that the "Alert" and "Required Action" ranges of Table 3 are plotted and the vibration limits of Part 6 are included. While many of the sources were in general agreement, there was not agreement on specific numbers.

For centrifugal pumps, limits were developed based on machines classified as Class III according to the ISO standard, and a consensus of the other standards reviewed. The limits of Part 6 were chosen as 0.325 in./s for "Alert" and 0.70 in./s for "Required Action". These limits were set somewhat liberally in recognition of the fact that pumps in nuclear power plants are not always tested under normal operating (i.e., full flow) conditions and that relative criteria were also being applied. Because of the unique nature of reciprocating pumps, no limit was set for the maximum permissible vibration. This may eventually change as additional data becomes available.

Table 3. Vibration criteria "Alert" and "Required Action" ranges

	Source	Behavior Regimes[a]	
		Alert Range	Action Required Range
1.	Canadian Government Specification (pumps > 5 hp)[11]	0.394 <= V <= 0.71	V > 0.71
2.	Energy Research and Consultants Corp. (ERCO)[2]	0.25 <= V <= 0.75	V > 0.75
3.	Gulf Oil Canada, Ltd.[8]	0.3 <= V <= 0.5	V > 0.5
4.	IRD Mechanalysis[4]	0.314 <= V <= 0.628	V > 0.628
5.	ISO 2372[1], VDI 2506[5]	0.25 <= V <= 0.628	V > 0.628
6.	J. S. Mitchell[3]	0.3 <= V <= 0.6	V > 0.6
7.	Materials Testing Institute (MTI)	0.2 <= V <= 0.6	V > 0.6
8.	Rathbone[7]	0.2 <= V <= 0.45	V > 0.45
9.	Shell Chemicals, U.K.[7]	0.47 <= V <= 0.75	V > 0.75
10.	Vibra–Metrics, Inc.[9]	0.19 <= V <= 0.48	V > 0.48

a. Units are velocity in in./s, peak.

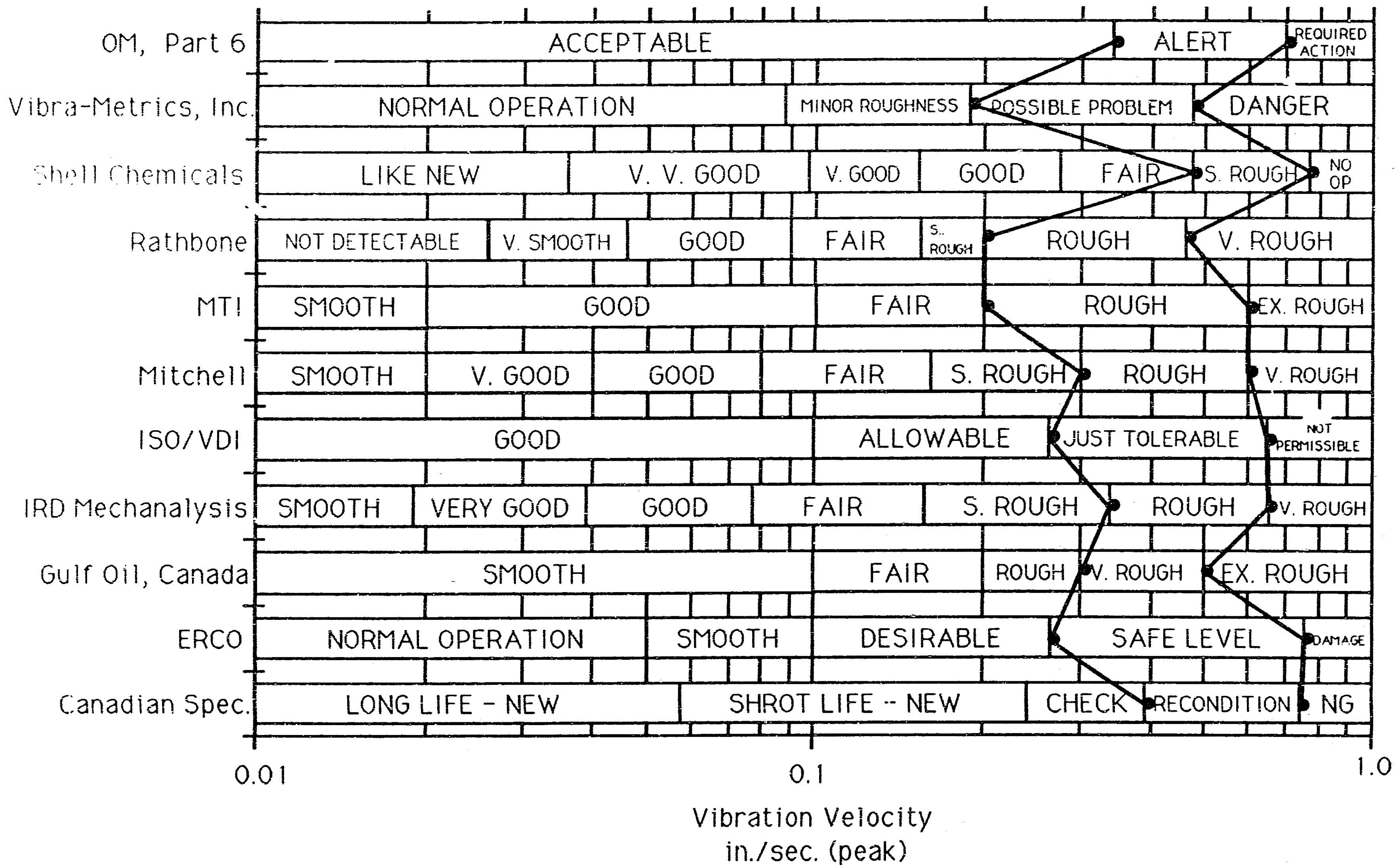

Figure 5. Vibration criteria "Alert" and "Required Action" ranges.

Although vibration velocity is generally considered the best measure of pump health, displacement is a better indicator of pump health for pumps which operate at low speed. The Committee chose to use 600 rpm, the lower limit of use for the ISO standard as the transition. The vibration limits for displacement are set at 10.5 mils for "Alert" and 22 mils for "Required Action", the equivalent of 0.325 in./s and 0.70 in./s at 600 Hz. It should be noted that Part 6 does not require the use of displacement for pumps which normally operate below 600 rpm and velocity for pumps which normally operate above 600 rpm. Part 6 indicates that displacement is preferred for pumps which normally operate below 600 rpm and velocity is preferred for pumps which normally operate 600 rpm.

The number and location of the required measurements has also been changed. On centrifugal pumps, vibration is now required to be measured in two orthoginal directions perpendicular to the rotating shaft on all accessible pump bearings. A measurement is also required in the axial direction on pump thrust bearings. For vertical line shaft pumps, the point of measurement has been defined as the upper motor bearing and measurement is now required in all three directions. These changes are in agreement with the literature and experience. No change has been made in the vibration measurement requirements for reciprocating pumps.

Another change in Part 6 is a clear definition of the type of measurement to be taken. Velocity is measured as peak (rather than rms) and displacement measured as peak–to–peak (rather than peak, average or rms). This is the general practice in the United States and Canada. In Europe, rms is generally used. The use of peak and peak–to–peak measurements tends to place more emphasis on transitory changes in vibration. The frequency range of these measurements is now from 1/3 pump operating speed to at least 1000 Hz.

CONCLUSION

This paper has attempted to show that the new vibration measurement criteria and requirements contained in Part 6 have a technically sound basis. The use of the new criteria will increase our level of confidence that pumps which are monitored in accordance with Part 6 are truly ready to perform their intended function when needed. In addition, the new requirements will decrease the burden on utilities which currently use a vibration velocity based predictive maintenance program. Changes in the measurement of hydraulic parameters made possible by the new vibration criteria should eliminate or decrease the number of "false alarms". The incorporation of Part 6 into plant surveillance programs will mark a significant increase in pump operational readiness.

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ATTENDEE – PANEL QUESTION AND ANSWER PERIOD

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

In regard to instrument loop accuracies, what components are to be included in the overall accuracy, plus or minus two percent. For example, a flow loop is required to have a plus or minus two percent accuracy. Does this include flow indicator, flow transmitter, square root converter, and flow element? Please consider that the flow element is usually not very accurate.

Response 1

In the past, the Code always considered the instrument accuracy in the ranges that are in the standard. I do recognize that there are some inaccuracies in the wiring and in the whole system of instruments that you need to consider to get the total accuracy.

I would add that, perhaps more than absolute accuracy, what we're interested in is repeatability, and in that case things like your flow element are not as much of a problem.

Question 2

Has the NRC/ASME resolved their differences in hydraulic pump parameters; i.e., 1.1 times reference value for delta P and Q being an acceptable range?

Response 2

We still have some differences of opinion in that area. We had a detailed two–day meeting with the NRC in Washington about two months ago where it was decided that, on the part of the Code folks on the working group, we would be looking at a task group that would try to see how we can best serve this concern. Larry Sage is the chairman of that task group.

We are currently asking if an alternative is to test pumps during refueling outages with a very detailed test setup and then do the same quarterly test in addition, with less emphasis on operability but more emphasis on the fact that the pump does start and pump fluid.

I think the important thing for everyone to know is that the NRC and the ASME have sat down and discussed this. There is a lot of communication about this item.

Question 3

When 0.70 in. per second or the required action range is not achievable during baseline testing, what are the alternatives?

Response 3

I think the only alternative is to go to the NRC with a relief request if you have a pump that is normally running at greater than 0.70 in. per second.

Question 4

4.6.12(b) states, "Digital instruments shall be selected such that the reference value shall not exceed 70 percent of the calibrated range of the instrument." Where did the 70 percent come from and why doesn't the Code specify an accuracy in terms of percent of reading rather than restricting the calibration range of the instrument?

Response 4

I think two things were operative in our decisions. The 70 percent essentially comes from the way Section XI handled digital instrumentation when they brought that in for pressure testing, and the only idea is you don't want to overrange the instrument if you happen to come up with something you are not expecting.

The reason we state things for other instruments as a percent of full scale is that is the way you usually buy an instrument—percent of full scale. That, perhaps, would not be appropriate in digital instruments. I think that is something we probably should take a look at.

Question 5

For a given pump flow rate test how can one account for a response that is better than the baseline test?

Response 5

I don't believe that an increase in pump performance a should be interpreted as implying there is anything wrong with the pump, first of all. You may be asking why people were concerned when we had the 1.03 action range before.

It's true that, for many reasons, you could not get the meter reference value, and you could go above the reference value during the test. I believe that response difference could be indicative of the test setup.

There may be something wrong with the way the test has been arranged in such case. Examples of that would be a misalignment in the loop. You change the resistance; you can get a different result.

We in the code believe that when you have a higher–than–what–you–expect type of value, you should rely on a parameter that is more indicative of conditioning, such as vibration, rather than the other parameter to declare a piece of equipment inoperable or have to go into a limiting condition of operation.

Question 6

It was stated in effect that OM 6 should not be used to determine operability until that operability has been initially established. Please discuss the intent of this statement and how the working group intends to establish the initial satisfactory operation.

Response 6

The reason I make that statement is that I have a fervent feeling that Part 6 in the way it is written right now cannot necessarily establish operability. We are not asking people to test to the safety limits or the safety parameter limits, such as of flow or pressure, that would be encountered if they had to mitigate the consequences of an accident.

At the same time, I think that a standard that would establish operability would require more than what an operational readiness standard would provide, which is what Part 6 is.

Question 7

A small reference value for vibration yields a small operable range for the pump. For example, VR equals .01 in. per second gives an alert range of 0.025 in. per second. The example shows that a smoothly running pump can be subject to corrective action. Has the committee addressed this?

Response 7

Yes, we have. It is our belief that, even though it may be a small level on an absolute scale, a change of an order of magnitude of six or something (which would get you into the required action range) is very significant.

Whether or not it is significant on an absolute scale, that was one of the things we didn't like about the absolute scale. You could have significant changes in mechanical condition without needing to go and find out why you have that change.

DIAGNOSTICS USED ON POWER GENERATION PUMPS

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BIOGRAPHY

Kevin R. Guy is a Senior Production Engineer at the Gibson Generating Plant of Public Service Indiana, where he is supervisor of the Predictive Maintenance Department. Guy has more than ten years of experience in vibration analysis. He holds a B.S. degree in Mechanical Engineering Technology from Purdue University. A member of ASME and the Vibration Institute, Guy also serves on the Board of Editors of the Shock and Vibration Digest.

ABSTRACT

Setting up a periodic vibrational monitoring program by the use of electronic data loggers. Acquired data will be analyzed and evaluated to determine pump condition. Measuring frequency, reporting procedures and conditions of mechanical components will be discussed in detail based on an actual case study.

INTRODUCTION

The majority of people think of predictive maintenance as a glorified vibration program. While they are correct in the assumption that vibration data is that backbone of the program, there is much more to predictive maintenance. Predictive maintenance is: vibration data collection, vibration analysis, testing, engineering, maintenance and quality control all wrapped up into one program. The goal of this program is to increase the unit availability and reliability while lowering the maintenance costs. This paper will discuss the procedures necessary to start a successful Predictive Maintenance program for power plant pumps.

The one item everyone needs to be aware of is that Predictive Maintenance Programs don't happen overnight; they occur over time. The normal time period is three to five years. The first several years is going to be filled with eliminating all the little equipment problems no one has taken the time to solve. During this time period, a vast amount of learning will take place. The learning stage will impact the engineer assigned to head the program, all people in contact with the program, and the plant itself.

The initial stages of the program will also be costly with the purchase of equipment and the increase in maintenance costs. The misnomer encountered with the birth of a Predictive Maintenance Program is it will save money immediately. This is totally false. The program will initially increase the cost of plant maintenance while you eliminate problems found during the early stages of your program. Management people are usually shocked by this, and it usually results in someone saying, "This program doesn't work," or "We never had this problem until we started Predictive Maintenance." The best advice is to keep management personnel informed as to all the possible consequences involved with starting programs.

INVESTIGATION

The first priority is to determine which pumps are going to be set up on a Predictive Maintenance Program. I recommend the person in charge of the Predictive Maintenance Program look over your yearly availability reports and see what pumps have led to your unit derates. If the trend holds true, you will find that boiler feed pumps and pump drives will be the greatest cause of your plant derates. Once you determine what your target pumps will be, investigation must begin into what has been the major cause of the problems.

I recommend the engineer responsible for predictive maintenance review the maintenance history on boiler feed pumps and their drivers for a period of five years. This will allow for a large enough time period to see any trends in equipment problems. Looking five years will give the engineer enough data to see if there are any repeat problems. These problems should be logged down and used in goal setting for the program which will be discussed later.

Once the investigation into which pumps will be targeted for predictive maintenance is complete and the past maintenance history of these pumps has been researched, you are now able to move on to the setup of the program. One thing never to forget while performing predictive maintenance: Don't expect the diagnostic equipment to find all your problems or solve them for you. The diagnostic equipment will help you keep tabs on your equipment health; but, you still need to understand how the equipment works and what its function is. The initial problems will be equipment related. However, later in the program many vibration problems are operational or system related. A lack of understanding in this area will lead to many unsolved problems.

STEPS TO PROGRAM SETUP

Equipment File Packages

The first item of business in setting up a program is to learn as much about the equipment as you can. This will entail putting together an Equipment File Package. The equipment file package will be the first step in the learning process on how the equipment operates ar. : what its function is in the system. Appendix 1 contains a generic Equipment Information Sheet. This sheet will contain all the pertinent information required for analysis, balancing and alignment. The information required for this sheet will not be readily available in the early stages, so you will be constantly adding to it as information becomes available. Examples of this would be the critical speed and balancing information that will become available after balancing is performed on the equipment and bode plots are generated. Alignment tolerances may also fall into this category if the equipment OEM is trying to sell his services. All other information required to fill these sheets out should be done while setting up the program.

The equipment file package should also contain prints of the equipment layout (Figures 1 & Figure 2), driver and driven, along with foundation information. These prints should be reduced down into a basic scheme as in Figure 3. Also needed is a pump curve (Figure 4). This should compile all the basic information required to do a basic analysis if a vibration problem comes up.

Equipment Category Breakdown

The next priority is to divide your pumps up into categories corresponding to their importance to the plant operation. Group #1 will be any capability reducing pumps (i.e. boiler feed pumps). These pumps should be monitored every two to four weeks. Group #2 will be any pumps that have a redundant backup (i.e. main condensate pumps). These pumps should be monitored every four to eight weeks. Finally, Group #3 is any miscellaneous pumps that don't fall into the first two groups (i.e. LPSW backwash pumps). These pumps should be monitored every eight to twelve weeks.

While breaking down the pumps into their individual groups, you need to also break them down into informal groups on how they are designed and mounted. Figures 5 to 8 show various examples of horizontally and vertically mounted designs. Knowing how equipment is mounted will give some insight into possible problems encountered when troubleshooting pumps. It Will also allow you to start a case history file on pumping problems encountered with different types of mounting schemes. Case histories will be discussed later in this paper.

Monitoring Equipment

This can be a very difficult part of your program, or it can be very easy. If you have done your job when filling out the equipment file package, you should have developed a feel as to what the past pump problems have been. Knowing what the past problems have been will allow you to look for problems previously experienced. Armed with this information, you should now be able to look into different types of monitoring programs.

If this is the first Predictive Maintenance Program you or your plant has been involved in, be very careful when choosing the type of equipment you will utilize. I do not recommend building an on–line monitoring system to do the job unless You have had several years of vibration data to design your system against. I recommend that data loggers or tape recorders, in conjunction with FFT analyzers, be utilized.

Data loggers are excellent for taking periodic data, and the software that accompanies the loggers allow you to trend vibration data and other equipment parameters. They will allow you to take signatures, along with the overall readings, if this is what you require.

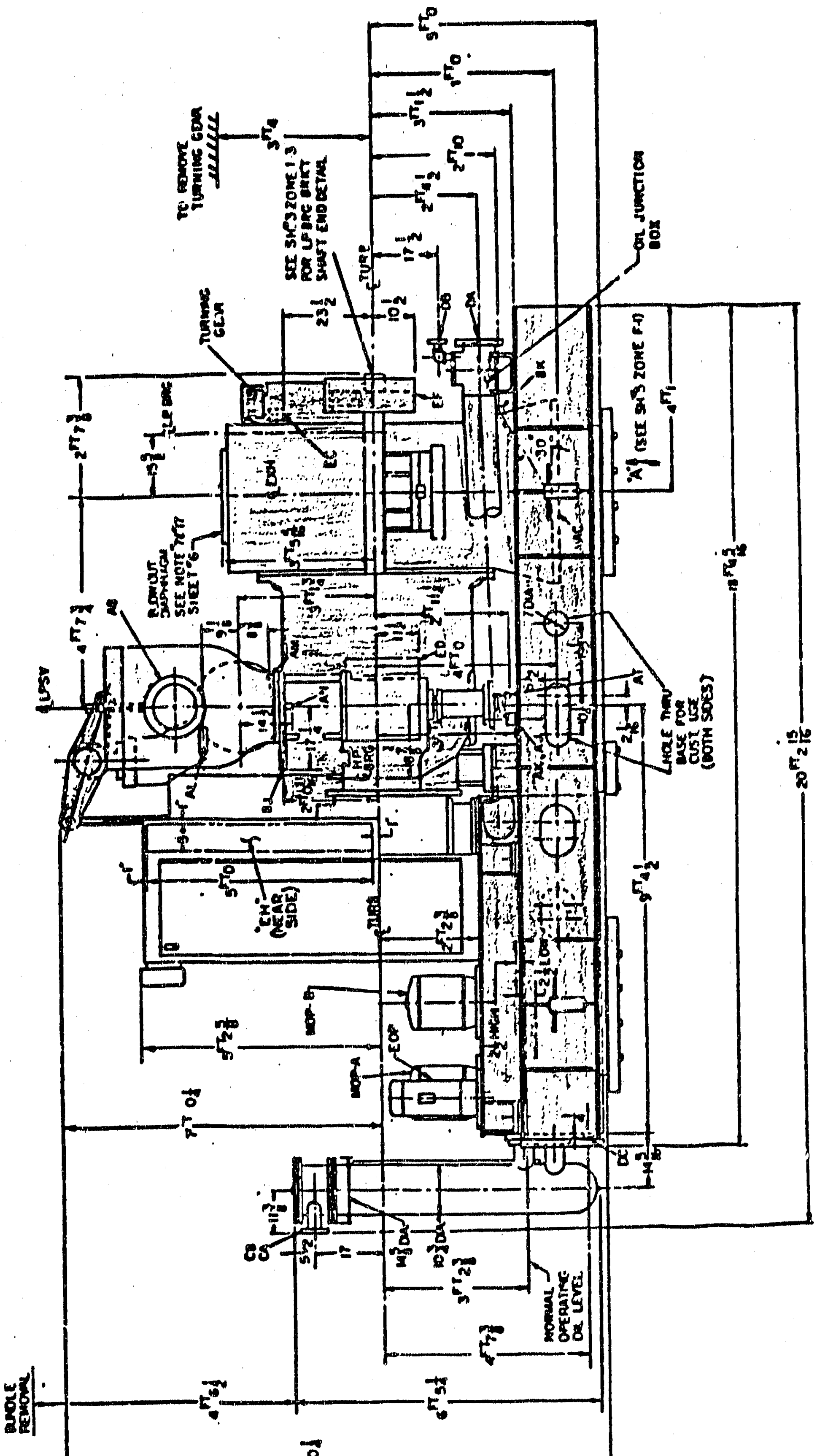
BUNDLE REMOVAL
TO REMOVE TURNING GEAR
TURNING GEAR
OIL JUNCTION BOX
HOLE THRU BASE FOR CUST USE (BOTH SIDES)
NORMAL OPERATING OIL LEVEL

Figure 1.

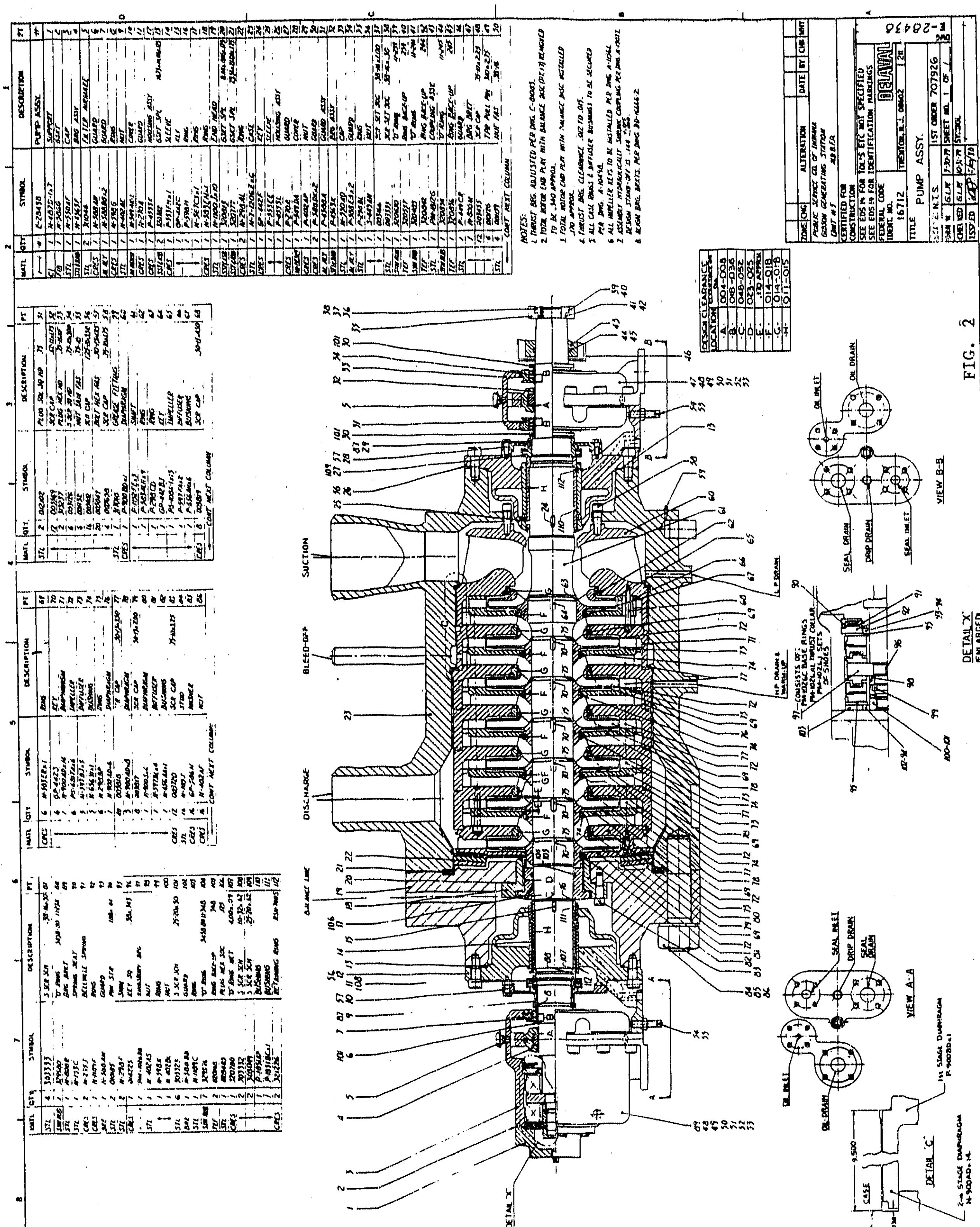
SUCTION
BLEED-OFF
DISCHARGE
VIEW A-A
VIEW B-B
DETAIL 'X' ENLARGED
DETAIL 'C'
DESIGN CLEARANCE
PUMP ASSY.
FIG. 2

Figure 2.

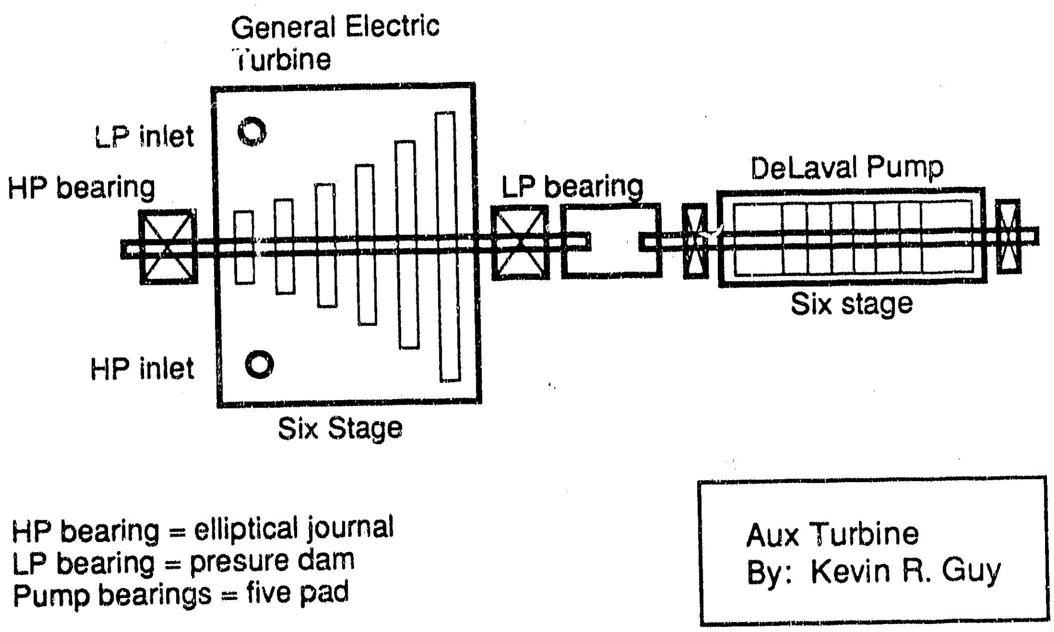
General Electric
Turbine
LP inlet
HP bearing
LP bearing
DeLaval Pump
Six stage
HP inlet
Six Stage
HP bearing = elliptical journal
LP bearing = presure dam
Pump bearings = five pad
Aux Turbine
By: Kevin R. Guy

Figure 3.

PUBLIC SERVICE CO. OF INDIANA
GIBSON GENERATING STATION - UNIT 5
S & L SPEC. NO. [illegible]-4749
P.O. NO. 5139-89
ITEM D- CLOSED COOLING WATER PUMPS
I-R ORDER NO. 002-32789

CAP. = 1500 GPM
MIN. RECIRC. FLOW = 750 GPM
TDH = 287 FT.
IMP. DIA. = 17.562"

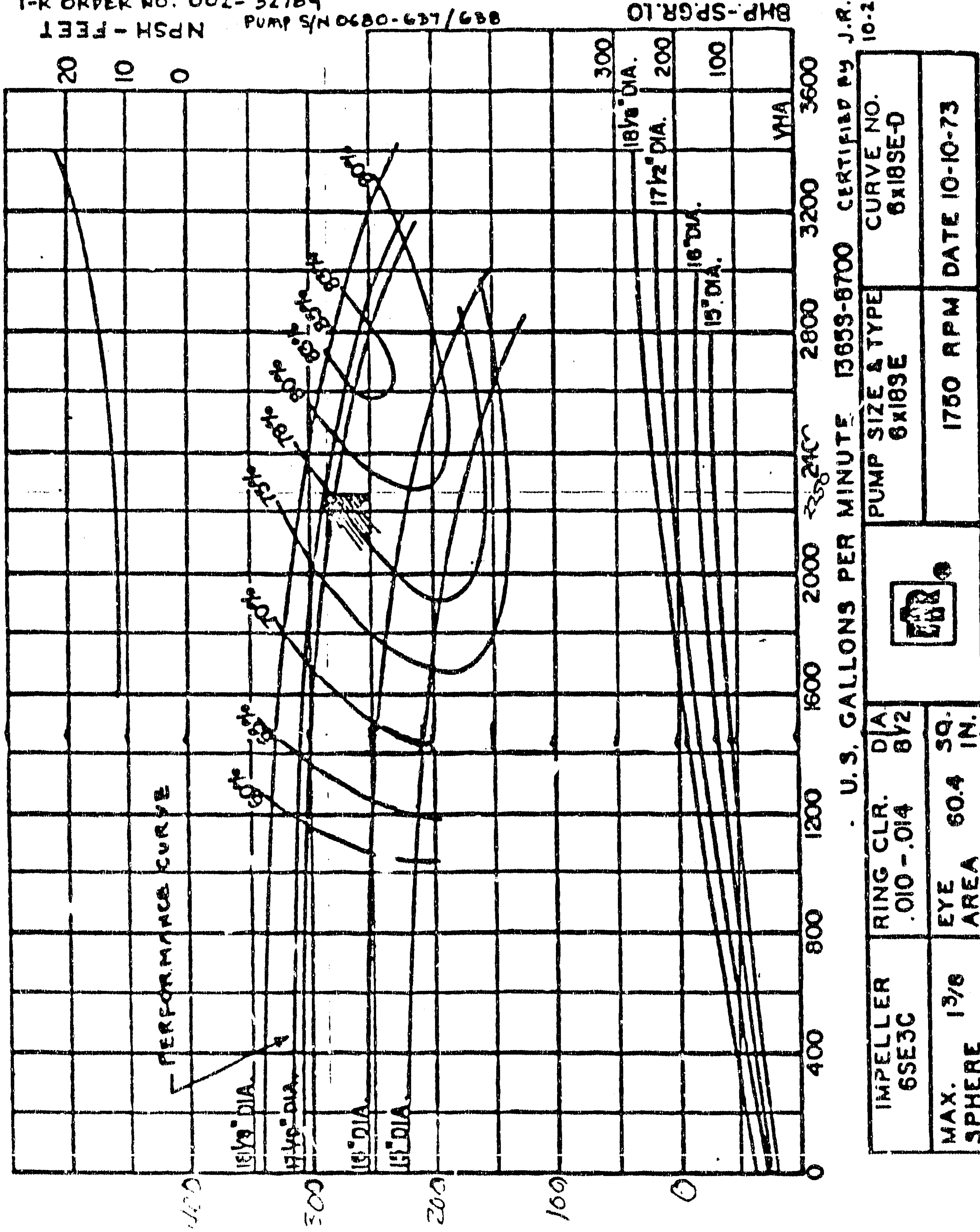

Figure 4.

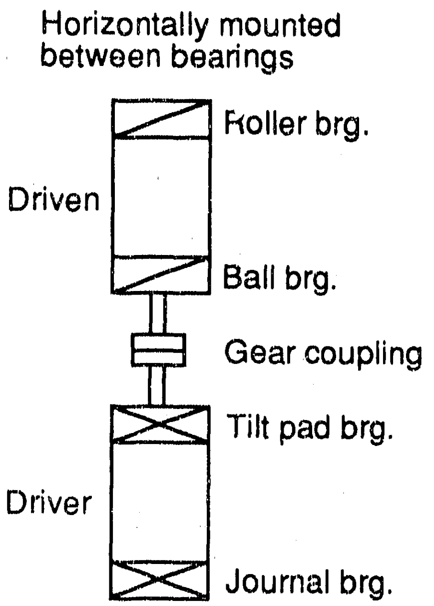

Figure 5.

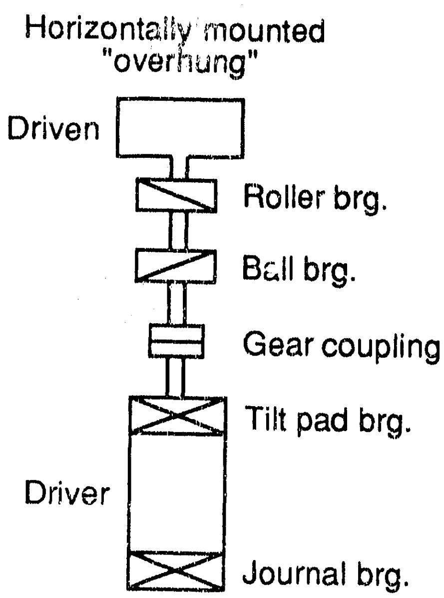

Figure 6.

The analysis capabilities of the data loggers, in my opinion, is minimal. Tape recorders will allow you to tape the data for later analysis, and from this you can make signatures for vibration trending. However, this method requires you to keep any external data (i.e. pump flows and pressures) in a separate area or you have to write it on the signatures. Some FFT's have the ability for the user to get overall readings from the spectral data; but, for those not familiar with FFT's, this could be difficult. My recommendation is, if you intend to take a lot of data each month (i.e. >500 points a month), use data collectors. Tape recorders can be difficult to operate and very time consuming. If you intend to keep information for later analysis (i.e. store real–time data), tape recorders are a must.

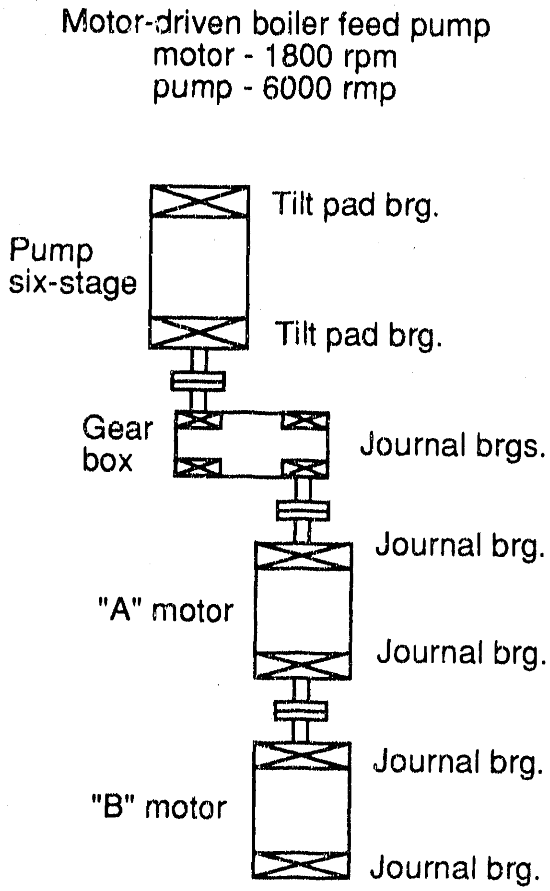

Figure 7.

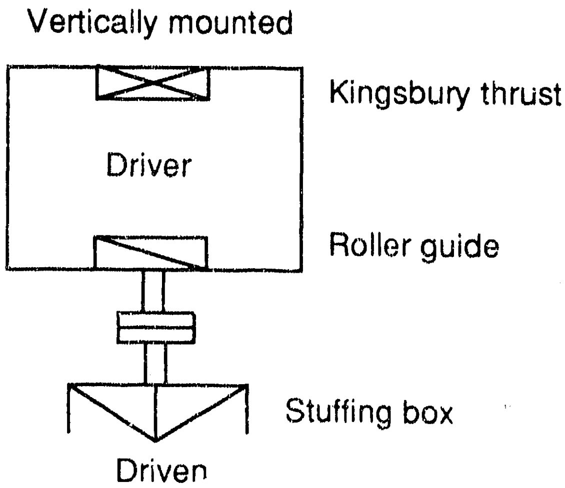

Figure 8.

Data Points

There are many vibration equipment manufacturers who say to only trend a few points on each equipment train. During the initial stages of any Predictive Maintenance Program, all available points should be taken. This data will show the trends, as to the points, which show the earliest occurrence of vibration problems. Once you learn which points will indicate your problems, then you can eliminate the less meaningful points on the equipment trains and take a minimal number of points. It should take approximately three to four years to get where a decision can be made to eliminate data points.

Turbine driven boiler feed pumps should be addressed as if they were turbine generators. Each month temperatures, pressures and steam flows, along with filtered and unfiltered vibration, should be taken. Phase angles will also be needed to trend vibration location so balance shots can be calculated. While I don't think signatures are needed, each month I recommend they be taken bi-monthly. Appendix 2 is an example of a vibration trend sheet for turbine generators that can be changed to fit your needs for turbine driven boiler feed pumps. Also, included in this appendix is external data that should also be trended with the vibration data.

Once the above steps are completed, you should be ready to start your vibration data collection for predictive maintenance trending. The other areas that need to be addressed for a complete predictive maintenance program are oil analysis and performance trending.

Oil Analysis

This is a service that can be added into a specification for your contracted oil supplier. Most suppliers provide some type of oil analysis to their customers. This service is normally atomic emission or atomic absorption. A program like this will allow for the trending of wear particles in your oil samples. Knowing this, along with what the material makeup of your equipment, will allow you to find where your wear particles are coming from. An alternative method of oil analysis would be ferrography. Ferrographic analysis will tell how the particles in the oil are being generated. Oil samples should be taken on the same schedule as the vibration periodic data. This will allow for correlation of oil data and vibration data. Past experience has shown that the oil reports will show problems three to six months before you see any vibration symptoms. Figure 9 is an example of what should be received from your oil lab.

Performance

The final area to set up for testing is performance parameters. This is an area that many people have different ideas on what should be done. Testing can range from full blown ACME pump tests with calibrated test instrumentation to pump tests involving plant instrumentation. If the plant instrumentation is calibrated and kept in good condition, there is nothing wrong with utilizing it. What you are trying to accomplish is to find what the normal operating parameters are and to find when the equipment falls outside this area (Figure 10). This same type of graph can be used for thermal performance parameters, if you are using a turbine to drive your pumps.

Start Condition Monitoring

Predictive maintenance, as stated earlier, does not happen over night. You don't just jump in and start predictive maintenance. Predictive maintenance is the culmination of three years work in condition monitoring. Condition monitoring will allow you to find the normal operation characteristics of your equipment. Without doing condition monitoring, you will have no baseline to judge your data against. Many predictive programs fail because people don't allow enough time to gather baseline data. During this condition monitoring period, many equipment problems will be found that need to be corrected before a reliable Predictive Maintenance Program can begin. The problems you will encounter are the three basic problems of: alignment, balance and mechanical looseness. These problems can be cured if addressed correctly. When troubleshooting for mechanical looseness, don't just think of looseness as loose bolting. The majority of the time, mechanical looseness is poor bearing fits and oversized bearing clearances. Balance and looseness will be the easiest to cure with alignment being the toughest. The three years you devote to condition monitoring will allow these problems to be addressed and corrected.

During the period of time condition monitoring is being performed, be aware of the impact you will have on other peoples' jobs. Mechanics will be out repairing equipment problems, and then you will come along with your vibration equipment, check the equipment over and report that there is still some type of basic problem. Many people will look upon this type of program as a way to check their work. All personnel in the plant should be made aware of what the program entails and how it will benefit them. Benefits to them will include less rework of jobs and less overhaul work.

PUBLIC SERVICE OF INDIANA
ATTN: P. MILLER/GIBSON STATION
P. O. BOX 1089
MT. CARMEL, IL 62863

ANALYSTS MAINTENANCE LABORATORIES, INC.

2450 HASSELL RD.
HOFFMAN ESTATES,
ILLINOIS 60195
(312) 884-7877

BOX 4002
SCHAUMBURG,
ILLINOIS 60194
TELEX: 258043

CUSTOMER NO. : #MO-333
UNIT REF. NO. : 35917
SENDER NAME : PUBLIC SERVICE OF INDIA
UNIT IDENT. : 1 TURBINE

MAKE / MODEL : GE
SHELL TURBO 32
REPORT DATE : 22Aug'88

STEAM TURBINE

AINTENANCE RECOMMENDATIONS FOR LAB NO. 0643
VALUES ACCEPTABLE FOR TESTS PERFORMED
RESAMPLE AT NORMAL INTERVAL

TALE PARTS PER MILLION BY WEIGHT — SPECTROCHEMICAL ANALYSIS — OPERATING DATA

LAB #	IRON	LEAD	COPPER	CHROMIUM	ALUMINUM	NICKEL	SILVER	TIN	SILICON	BORON	SODIUM	PHOS.	ZINC	CALCIUM	BARIUM	MAGNESIUM	TITANIUM	MOLYBDENUM	POTASSIUM	ANTIMONY	MI/HR UNIT	MI/HR OIL	OIL ADD GALS.
063[illegible]	[illegible]0.	000.	000.	000.	000.	000.	00.0	002.	000.	000.	000.	0010	0520	0000	0000	0000	000.	000.	000.	002.	0.0	0.0	0.00
4777	000.	000.	000.	000.	000.	000.	00.0	000.	000.	000.	000.	0020	0030	0000	0000	0000	000.	000.	000.	006.	0.0	0.0	0.00
8011	000.	000.	000.	000.	000.	000.	00.0	000.	000.	000.	000.	0020	0002	0000	0000	0000	000.	000.	000.	000.	0.0	0.0	0.00
0643	000.	000.	000.	000.	000.	000.	00.0	001.	000.	000.	000.	0020	0060	0000	0000	0000	000.	000.	000.	000.	0.0	0.0	0.00

PHYSICAL PROPERTIES TESTS

LAB #	SAMPLE DRAWN	W %VOL	V CS 40 C
0636	30Mar88	(0.05	34.9
4777	20Apr88	(0.05	28.8
8011	13Jun88	(0.05	32.2
0643	16Aug88	(0.05	29.4

UNDERLINED FIGURES INDICATE ABNORMAL VALUES. MAINTENANCE THAT MAY BE REQUIRED IS INDICATED ABOVE UNDER MAINTENANCE RECOMMENDATIONS AND SHOULD BE PERFORMED BY A QUALIFIED MECHANIC. PLEASE ADVISE US OF ANY MAINTENANCE PERFORMED ON THIS UNIT.
N/R = TEST NOT PERFORMED

ACCURACY OF RECOMMENDATIONS IS DEPENDENT ON REPRESENTATIVE OIL SAMPLE AND COMPLETE, CORRECT DATA ON BOTH UNIT AND OIL. THIS REPORT IS NOT AN ENDORSEMENT OR RECOMMENDATION OF ANY PRODUCT OR SYSTEM. ORIGINAL REPORT MAINTAINED IN ANALYSTS, INC. DATA FILES.

FOR LEGEND AND EXPLANATION OF PHYSICAL PROPERTIES TESTS PLEASE SEE REVERSE SIDE

FORM NO. 2004 (5/86)

Figure 9.

… ANALYSIS REPORT EXPLANATION

… ANALYSTS INC.

CUSTOMER NO
UNIT REF NO
SENDER NAME
UNIT IDENT

MAKE MODEL

REPORT DATE

MAINTENANCE RECOMMENDATIONS FOR LAB NO

LAB #	SPECTROCHEMICAL ANALYSIS — IRON	LEAD	COPPER	[illegible]	ALUMINUM	NICKEL	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	ZINC	[illegible]	BARIUM	MAGNESIUM	TITANIUM	MOLYBDENUM	POTASSIUM	ANTIMONY	OPERATING DATA — MI/HR UNIT	MI HR OIL	OIL ADD GALS

PHYSICAL PROPERTIES TESTS

LAB #	SAMPLE DRAWN	REPORTED PER DESIGNATED PACKAGES - SEE LEGEND BELOW

OIL ANALYSIS REPORT EXPLANATION

1. Mailing address for completed report
2. Location and contact information for the Analysts, Inc. Laboratory performing the service
3. Component identification and description
   3a. Unit Ref No. is a computer assigned number which is unique for each individual component. **Refer to this number on all correspondence or communication concerning a particular component.**
4. A number assigned each sample upon receipt for internal control
5. Twenty metallic elements which are identified and measured in parts per million. **These elements provide a monitor for wear particles, corrosion, debris, airborne contaminants, coolant additives and metallic oil additives. NOTE: values reported as 998 or 9980 represent concentrations beyond routine reporting limits**
6. Pertinent operating data supplied with the submitted sample
7. Physical property tests which may include, but are not limited to:
   A. Fuel Dilution - The amount of unburned fuel present in the lubricant of diesel and gasoline engines (also applied to some turbines)
   B. Total Solids - The total amount of solids contamination, both suspended and nonsuspended, present in the lubricant. This contamination consists of hard and soft carbons, soot, oxidation by-products, water, dirt and wear metals
   C. Water - Measurement for detectable levels of moisture present within the lubricant
   D. Viscosity - The measurement of a fluid's resistance to flow at a given temperature in relation to time. Most commonly understood as the "flow rate." This test may indicate the degree of dilution, shearing, oxidation (thickening) and/or product contamination
   E. SAE Viscosity Grade - The SAE grade equivalent of the viscosity @ 100° C
   F. Neutralization Number - A number expressed in milligrams of reagent required to neutralize one gram of lubricant
      1. Total Acid Number - The total amount of acidic products present in the lubricant
      2. Total Base Number - The level of reserve alkalinity remaining in the lubricant
   G. Particle Count - A numerical count of suspended particles present in a lubricant specifically measured within designated micron ranges

PHYSICAL PROPERTIES LEGEND

SAE WT - SAE Oil Grade
TAN - Total Acid Number
TBN - Total Base Number
PBN - Precursor Base Number
GLY - Glycol Test
SUG - Sugar Test
SULF - Sulfur % By Wt
P.C. - Particle Count
DIE-KV - Dielectric Strength
MFP-Mg - Millipore Filtration

Figure 9. (continued).

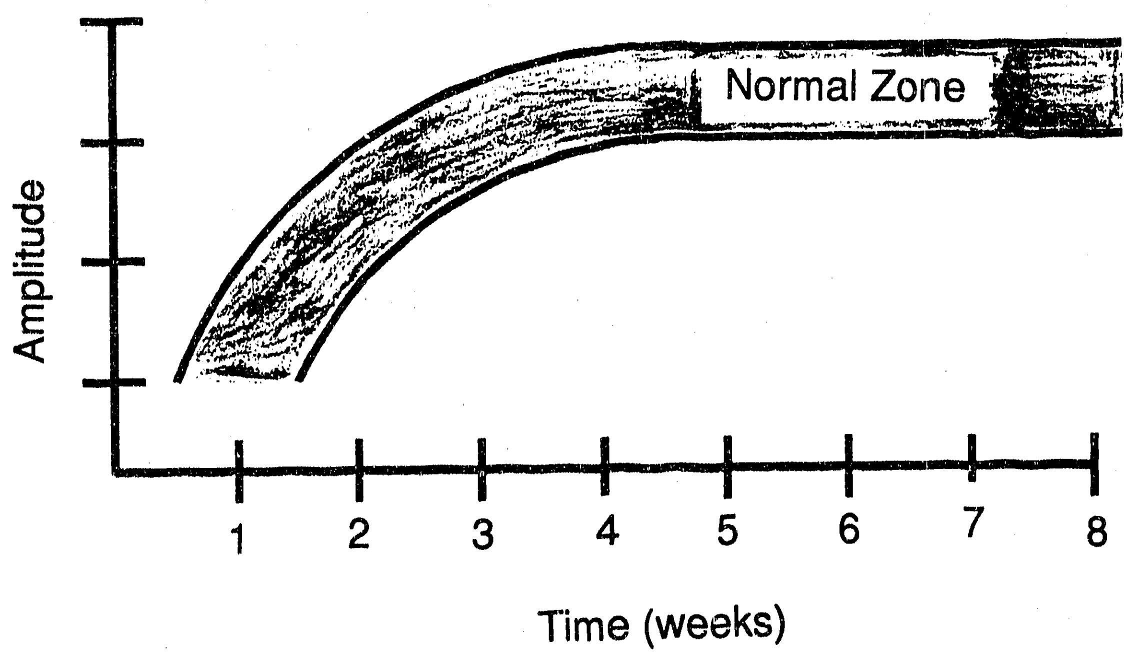

Figure 10.

Since the equipment is being monitored, you can perform overhauls on an as–needed basis instead of a scheduled basis. One thing you will still have to do is the routine preventive maintenance (i.e. oil changes and packing adjustments).

Case Histories

Documentation of equipment is essential for analysis of problems and records of past solved problems. These reports are also good for showing plant management what you have accomplished. During the initial period of your program, you will be highly visible; however, after a period of time when the initial problems are solved, y ır visibility will drop drastically. There is another more important factor to be considered; training. People don't tend to stay in one job all their life. They are promoted or leave for other reasons. Something is needed to train newer employees and assist the analyst when troubleshooting. The case history will fill this need. The case history should be divided into six parts:

1. Problem:

   Discuss how the problem was found and include a description of the equipment.

2. Symptoms:

   A description of the initial finding and any abnormal external visual findings.

3. Test Data and Observations:

   Tell what you are trying to accomplish with your test and what the priorities are.

4. Corrective Action:

   In detail, describe the findings along with the recommendations for repair of the problem.

5. Results:

   A brief discussion of the repair findings.

6. Conclusion:

   Describe how well the problem was diagnosed and what can be done to improve the diagnostics, and solution. Here is also the place to recommend improvements in analysis equipment and maintenance repair practices.

CONCLUSION

Setting Predictive Maintenance Programs up is not difficult if it is done in logical order. The key thing is to lay a firm foundation in which to build your program on. I recommend that you start slowly and be conservative. Don't try to grow too fast or get into analysis problems that are above your ability. The most important thing to ensure a successful program is to have dedicated people. Many managers believe this program can be run with people who devote only part time to it; it is a full time job. Part timers do not spend enough time on the job to become proficient in analysis. Also, don't skimp on training. Allow the people involved in the program to attend classes on vibration analysis. I recommend that equipment vendor training be avoided. While they train you on how to use their equipment, the training tends to be a sales pitch.

Predictive Maintenance Programs can be a very important cost saving addition to your plant maintenance department. Public Service Indiana's Gibson Generating Station has deferred over 125,000 manhours of equipment overhauls over the past four years. This is well over $2,000,000 dollars in labor cost savings to the plant budget. Looking at what it costs to run the program against what it saves equates to approximately 38 percent return on their investment each year the program has been in place (program is four years old).

APPENDIX 1

Date:________

Revision:______

PREDICTIVE MAINTENANCE INSTRUCTION SHEET

Equipment Name:

Number:

Classification:

Motor:

RPM:

Bearings:

Machine Flow:

Number Blades/Impellers:

Rotor Weight:

Shaft Critical Speeds:

Shutdown Limits:

Sensitivity:

Lag Angle:

Alignment Tolerances:

Desired Alignment Readings:

Special Instructions:

APPENDIX 2 (1 OF 3)

CUSTOMER ______________________

STATION & UNIT NO. ______________________

TURBINE NO. ______________________

CODE & LSB LENGTH ______________________

EQUIPMENT USED ______________________

EQUIPMENT PHASE ANGLE ______________________

ZERO REF. MARK COLOR ______________________

INLET OIL TEMP. ______________________

VIBRATION DATA SHEET

DATE, TIME LOAD/SPEED	FILTER	#1 BRG AMP	#1 BRG ∠	#2 BRG AMP	#2 BRG ∠	[illegible] BRG	[illegible] BRG ∠	#4 BRG AMP	#4 BRG ∠	#5 BRG AMP	#5 BRG ∠	#6 BRG AMP	#6 BRG ∠	#7 BRG AMP	#7 BRG ∠	#8 BRG AMP	#8 BRG ∠	MAIN GEN TE END AMP	MAIN GEN TE END ∠	BRUSHES GE END AMP	BRUSHES GE END ∠	#9 BRG AMP	#9 BRG ∠	#10 BRG AMP	#10 BRG ∠	STUB SHAFT
	OUT																									
	IN																									
	OUT																									
	IN																									
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	IN																									
	OUT																									
	IN																									

ROTOR CRITICAL SPEEDS ______________________

BALANCE SHOT LOG ______________________

OTHER DATA & REMARKS

APPENDIX 2 (2 OF 3)

Bearings	Vibration	Metal	Oil Drain
1.			
2.			
3.			
4.			
5.			
6.			
7.			
8.			
9.			
10.			

APPENDIX 2 (3 OF 3)

Unit #

Load:

1. Main Condensate Vacuum
2. Feed Water Flow–Economizer
3. Main Steam Pressure
4. Main Steam Temperature
5. Cold Reheat Pressure
6. Cold Reheat Temperature
7. Hot Reheat Pressure
8. Hot Reheat Temperature
9. Oil Temperature To Bearings

ATTENDEE – PANEL QUESTION AND ANSWER PERIOD

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Can you discuss the value of taking hydraulic data for determining pump degradation? Do you view vibration data as essentially sufficient for this purpose, or do you regard hydraulic data as a necessary complement to vibration data, and why?

Response 1

I think you really need the hydraulic data. If you have got equipment, pumps that have roller bearings in them, a lot of times if you start cavitating the pumps slightly, you will get slight roller bearing frequencies. If you go out and take vibration data, you'll think your bearings are going bad, whether you spike energy HFD or any of the other things these data collectors do. Or even if you look on an FFT, you'll start to see roller bearing frequencies, and immediately you'll think your bearings are bad, and you'll start to take your bearings out.

I think hydraulic data is needed, whether it is just pump suction and discharge pressures, or you actually go out and get flows. We take suction discharge pressures monthly, and then we try twice a year to do actual performance tests, just with flow meters, not necessarily a full–blown ASME test.

We had found that we were unnecessarily replacing a lot of bearings because we weren't looking at the performance of the pumps. And I think a lot of people make that mistake at the beginning of their vibration program. That is, they look at only the mechanical equipment itself, the integrity. They see a problem and automatically think it is bearings, and that is not necessarily correct. So I recommend hydraulic data, too.

PERSPECTIVES ON PUMP TESTING AND INSPECTION REQUIREMENTS IN NUCLEAR PLANTS

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PART ONE
PERSPECTIVES ON PUMP TESTING AND INSPECTION REQUIREMENTS IN NUCLEAR PLANTS

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ABSTRACT

This paper describes some of NRC's past experiences related to pump testing in accordance with Subsection IWP of the ASME Code, Section XI. Also discussed are NRC's role in the development of pump testing standards, insights gained through NRC evaluations of IST programs and a desirable direction for future pump testing.

This paper was prepared by an employee of the United States Nuclear Regulatory Commission. It presents information that does not currently represent an agreed–upon staff position. NRC has neither approved nor disapproved its technical content.

INTRODUCTION

It has long been the goal of most persons involved with pump testing in U. S. commercial nuclear plants to eliminate, or at least minimize, unnecessary surveillance testing; including the testing of pumps. Most NRC representatives agree that meaningful testing and exercising of pumps should be optimized. However, the supporting role of providing component exercise should not be underestimated. It minimizes problems associated with stagnant equipment, such as drying out of seals or packing, brinneling of bearing surfaces, accumulation of debris, etc.

NRC SUPPORT OF PUMP INSERVICE TEST CODE DEVELOPMENT

Inservice Testing (IST) requirements for certain pumps in light water cooled nuclear power facilities were incorporated into the Code of Federal Regulations, Title 10, Part 50 (i.e., "10 CFR 50" or "the Regulations") in 1976. Since that time the Regulations have referenced the ASME Code, Section XI, Subsection IWP ("the Code") for pump testing. It is expected that Part 6 of ASME/ANSI OM will soon replace Subsection IWP as the pump test code to be used.

The NRC has consistently participated actively in ASME industry consensus code development in the area of inservice pump testing.

MINIFLOW CONFUSION – THE VICIOUS CIRCLES

Circle One

As previously stated, pump testing in accordance with the Code was incorporated by reference into the Regulations in 1976. At that time the testing specified in the Code was required on a monthly basis, which was in agreement with the system–oriented pump test schedules required by "custom" plant technical specifications.

During the first three years of implementation, there was a growing sentiment within the industry that pumps were being overtested and that the monthly frequency of testing was resulting in unacceptably accelerated pump degradation, particularly for pumps that were being tested on miniflow. Although this perception was not supported by specific data, the ASME Section XI Code Committee proposed that this test frequency be changed to quarterly (i.e., testing at three month intervals in lieu of every month). The NRC concurred with the new proposed test interval by endorsing the Winter 1979 Addenda to the Code.

Later in the 1980s concerns began to arise with respect to the adequacy of miniflow recirculation lines to support various pump safety functions, including 10 CFR Part 21 notifications by some pump manufacturers, which indicated potential safety concerns. The NRC responded to these concerns by the issuance of several Information Notices[2,3,5] and Bulletins[1,4]. In general, these documents addressed various facets of "low–flow" or "no–flow" degradation of pumps. However, industry response to the NRC Bulletin 88–04 did not indicate that abnormal pump degradation was attributable to miniflow operation during inservice testing.

ASME/ANSI OM Part 6 is now a published document and the ASME O&M Working Group on Pumps and Valves has now isolated individual areas that are in need of further development. Therefore, it is now possible to assimilate and evaluate more specific data to support any needed changes. Future changes to pump test standards should, where possible, be based on "hard" data, rather than just perceptions.

Circle Two

Throughout the 1980s the NRC has consistently required that hydraulic instrumentation must be provided for IST, and only granted relief from this Code requirement on an interim basis until the necessary instrumentation could be installed. Over this same period most utilities have maintained the position that pump miniflow lines are "fixed resistance"; therefore, there is no technical need to install flow measuring instruments (i.e., the flowrate will always be the same for a specific differential pressure).

However, more recently it has been noted by at least one utility that since pump miniflow lines rarely have throttling capabilities, neither flow nor pressure can be fixed; and that this makes test repeatability difficult to accomplish due to the highly sensitive nature of the flow instrumentation from small changes in pressure. If miniflow test repeatability is indeed a problem which is due to the lack of throttling capabilities, it further supports the need to have both flowrate and differential pressure instruments installed in the miniflow line. This will enable the test personnel to develop a small segment of the pump hydraulic performance curve to which acceptance criteria can be applied. This approach is somewhat synonymous with that which is frequently applied to inservice tests performed in the "normal" flow paths of certain systems which cannot be fixed at certain flowrates due to their constantly varying demands. To further enhance the quality of hydraulic data, instruments installed to accommodate IST should have "range" capabilities only slightly higher than the reference values.

FUTURE DIRECTIONS

Better Tests vs. Increased Test Frequency

Both Subsection IWP of Section XI and Part 6 of ASME/ANSI OM require that the test frequency be doubled for a pump that has reached the alert range. Although this increased test frequency provides some measure of "on demand" reliability and also provides additional exercise, a better method of determining the reasons for the change in pump condition could be more meaningful. For instance, by taking good test data when performing baseline tests, which includes both filtered (frequency dependent) and unfiltered (frequency independent) vibration velocity data, a possibility exists to better determine why the pump has entered the alert range. From that an analysis can be performed to determine the need for corrective action, rather than simply keeping the pump on an increased frequency of testing.

Improved Hydraulic Measurements

In addition to improving the quality of vibration measurements, the addition of accurate and repeatable hydraulic instrumentation to pump test flow loops, particularly in miniflow test loops, will further enhance the quality of pump testing and minimize the data "scatter" which industry representatives have continually been plagued with. This is especially important for miniflow test loops, because the head–flow curves of some pumps are flat in the area related to low flow operation; therefore, the establishment of a reliable value for flowrate will enhance the entire test for hydraulic performance. In addition, as previously stated, flow gauges to be used in miniflow recirculation lines should have a range only slightly higher than expected flow rates to minimize fluctuations.

REFERENCES

1. USNRC Inspection and Enforcement Bulletin No. 86–01, "Minimum Flow Logic Problems That Could Disable RHR Pumps," May 23, 1986

2. USNRC Inspection and Enforcement Information Notice No. 86–39, "Failure of RHR Pump Motors and Pump Internals," May 20, 1986

3. USNRC Information Notice No. 87–59, "Potential RHR Pump Loss," November 17, 1987

4. USNRC Bulletin No. 88–04, "Potential Safety–Related Pump Loss," May 5, 1988

5. USNRC Information Notice No. 89–08, "Pump Damage Caused by Low Flow Operation," January 26, 1989

PART TWO
PUMP INSERVICE TESTING INSPECTION REQUIREMENTS

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ABSTRACT

This session will discuss NRC Inservice Testing (IST) inspection requirements delineated in 1 and E Inspection Module 73756, "Inservice Testing of Pumps and Valves." A brief discussion of IST inspection findings related to pump testing, including common problems with IST program implementation identified during IST inspections in Region III during the period 1983–1986, will also be included.

This paper was prepared by an employee of the United States Nuclear Regulatory Commission on her own time apart from her regular duties. NRC has neither approved nor disapproved its technical content.

INSPECTION HISTORY

In 1983, Region 111 began the systematic performance of Inservice Testing (IST) inspections by region–based inspectors. As a result, two inspectors from the Division of Reactor Safety were assigned to perform IST inspections and to determine the status of implementation of IST programs across the Region. This paper will focus primarily on experiences with the inservice testing of pumps.

Systematic inservice testing inspections in Region III began in early 1984. Region III inspectors also assisted with IST inspections in Regions II and V. By 1986 almost all the plants in Region III had been inspected at least once and follow–up inspections at selected sites were conducted. Follow–up inspections continue to be performed; however, this paper will only address those inspections in which the author participated.

After the first few inspections, it became clear that guidance regarding the inspection of IST program implementation was necessary to insure that all plants in each region were evaluated in a consistent manner. As a result, a draft inspection module was initially drafted in 1985. This draft considered inspection experiences gathered from Regions II and III over a two year period. Requirements of the ASME Code, Section XI, using editions from 1974 through 1980, and the appropriate addenda, were reviewed and translated into viable inspection line items. Once the module was drafted, it was validated during the next few IST inspections. It subsequently was submitted to NRC Headquarters for review and comment. Inspection Module 73756, "Inservice Testing of Pumps and Valves," was issued for use by the Commission on March 16, 1987.

Copies of the inspection module are available for your information and may be obtained from the NRC Public Document Room. It should be noted that the module references a number of interpretations of specific IST issues which have surfaced in the past. The referenced memoranda were attached to an inspection report for LaSalle County Station in 1985. This essentially placed the referenced material in the NRC Public Document Room. They are, therefore, available to the public.

PERFORMANCE OF IST INSPECTIONS

NRC IST inspections are generally unannounced; however, proposed inspection dates are discussed among the inspection staff, the appropriate Division of Reactor Project (DRP) section chief and the resident inspectors for the plant in question. Any specific concerns identified by the resident inspectors are added to the inspection scope. The inspector also informs the IST program reviewer in Mechanical Engineering Branch of the Office of Nuclear Reactor Regulation (EMEB) of the upcoming inspection. Occasionally, field observations can clarify concerns stemming from the EMEB review of the licensee's IST program.

If practical, the IST inspector(s) will obtain a copy of the latest IST program, including relief requests and copies of related correspondence between the licensee and EMEB, prior to performance of the inspection. Relief requests are then reviewed against Section XI requirements as well as other applicable regulations. At this point, specific areas of inspection focus are identified.

The inspection usually lasts a week. First, the administrative controls regarding test scheduling, test equipment and basic program implementation are assessed. Next, the conduct of testing is reviewed. Finally, actual testing is observed. Approximately 40% of an IST inspection is spent on the review of pump testing.

ASME Code interpretations and clarifications of Section XI requirements play an important role during IST inspections and NRC inspectors have discovered that obtaining such interpretations through the normal and appropriate ASME committee chain can be very time consuming. This can significantly delay resolution of an inspector's concern. In some cases, particularly where a given issue may have generic implications, the EMEB has provided technical assistance and support to regional inspectors.

INSPECTION EXPERIENCES

The goal of the inspection is to ascertain whether the licensee is conducting its IST program in accordance with the Code, within the boundaries of the Technical Specifications and in a conservative manner. Recognizing that IST programs are intended to identify equipment degradation prior to component failure, the inspector is charged with the review of at least three things:

1. Review of relief requests to determine whether the relief requests are valid, make sense and do not circumvent the Code requirements within the constraints of plant operations. The technical justification for selected relief requests from Code requirements is reviewed and, if questionable, is discussed with the resident inspectors, the appropriate EMEB reviewer and the licensee. The inspectors also look into the review status of selected relief requests which have been implemented prior to Commission approval. Past experience has shown that confusion regarding relief requests occurs roughly every second inspection.

2. If possible, inspectors observe at least one pump and one valve test. By observing test performance, the inspector can generally assess the effectiveness of IST. In cases where no testing is scheduled during the inspection, an in-depth review of test records and associated documentation is performed. If there are two IST inspectors, testing may be observed from the control room and at the pump simultaneously. In some cases correlation of the two activities have not effectively supported test conduct.

   An inspector can determine whether IST is understood and treated seriously by observing test activities. Generally, licensee personnel conducting the test know that IST testing is required, but they may not be aware that IST testing determines component operability and associated technical specifications apply. In several cases, test personnel have been unaware that test measuring equipment used for the inservice testing of pumps such as stopwatches, vibrometer and tachometers require periodic calibration to traceable standards.

3. Last, but not least, test records, including peripheral documents which interface with test documentation, are reviewed. These can include, but are not limited to, instrumentation calibration records for both permanently installed and portable test instrumentation, and unit start-up records. In most cases, the IST test is also used to determine component operability for returning the pump to service after an outage. A fair number of concerns were identified related to inconsistent or poor record keeping. At least two such concerns resulted in violations.

INSPECTION FINDINGS

During the period November 1983 through November 1986, IST inspections were performed at 18 of the 19 power plant sites located within Region III. Of these 18 sites, three plants experienced repeat inspections. Results of this three-year effort were: 23 violations and a fair number of unresolved and open items. Items related to the inservice testing of pumps fall into a few broad categories.

Of the 23 violations, 8 were issued for inadequate or nonexistent procedures governing the conduct and implementation of the IST program, 7 were issued for use of improperly calibrated or noncontrolled test equipment, 7 were issued for use of test methodology at direct variance with Code requirements and without an associated relief request, and 1 was issued for irretrievable test documentation.

Several unresolved items regarding documentation or specification of those tests which verify or establish new reference values and use of nonstandard pump testing acceptance criteria were also identified. The latter items were referred to EMEB for evaluation.

Since 1986, two violations have been issued in Region III for failure to maintain the points where pump vibration data is obtained. In both cases, the pumps in question were painted as part of the material condition improvement program, and the dots identifying vibration data points were painted over.

PUMP TESTING GOOD PRACTICES

Several good practices were identified during the 1983–1986 IST inspections. Briefly, these were: (1) obtaining vibration spectra for all pump bearings in all three orthogonal directions, (2) periodic rebuild of pumps regardless of test data trends, and (3) use of more than one set of reference values for certain pumps.

Generally, the vast majority of licensees are measuring vibration in velocity. Given the current available technology, instruments used to obtain vibration data for IST testing can simultaneously obtain velocity and displacement vibration spectra. In some cases, this data has been very useful in clarifying whether a pump has degraded despite that IST test data indicating that the pump was operating satisfactorily. Several licensees had designated an individual solely to vibration monitoring both within and beyond the Code requirements. Generally, this individual is a member of the technical staff who interfaces with both the IST coordinator and the maintenance department.

Some licensees schedule preventive maintenance for pumps based on inservice testing test data. Identified trends in any of the test parameters can trigger work requests to disassemble and inspect a specific pump. Several licensees periodically rebuild pumps regardless of whether or not IST data indicates any pump degradation.

In some cases, two sets of reference values are maintained for a given pump. This has been helpful in those cases where refueling outages run beyond schedule. Specifically, pumps which draw their suction from the suppression pool in a boiling water reactor may have two sets of valid reference values, one for normal operating suppression pool level and one for refueling pool level. This allows performance of inservice testing at either of the two pool levels and can alleviate delays in unit start–up resulting from the lack of valid pump tests.

CONCLUSION

Generally speaking, the administration and implementation of IST programs in Region III has improved greatly since the first round of inspections. As many plants are approaching or have begun their second ten–year inservice testing interval, programs are being upgraded and improved. This is evident from the decrease in violations issued for IST program deficiencies.

Inservice testing data can be used as evidence of pump reliability. Care must be taken to insure that other plant activities do not compromise the inservice testing program. Maintenance of thorough test records and associated documentation can be very helpful when evaluating whether a given component will continue to be operable and to identify component degradation as it ages. It is therefore important that IST programs, test procedures and associated instruments be periodically upgraded to insure that the reliability of pumps can be determined by the prudent inservice testing of pumps.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Do you have an example of good trending programs?

Response 1

It's pretty straightforward. You graph the data, you do a curve fit and a least squares analysis, and you keep an excruciating file. Not only do you keep the data of the test, but why the test was performed, and then you graph all the parameters.

It is basically common sense. But it is attention to detail, and it is done with an eye toward diagnosing what may be wrong with a given component. You know much better than I do what will determine what is wrong with a given component, and those are the parameters you should trend. But I have seen some "trending" where all they did was just keep the last page of the test procedure and stick it in a file. That is not a trend; that is a file.

Comment on Question 1

The working group has a task group that is looking into what is a good trending program. I think a good trending program has to involve statistical methods on how to analyze data that are continuous, measurable. And I think what we are doing there is to try to come up with a system in which you do plot points, but you calculate upper and lower control limits, you understand how data behaves around a mean, and you know when you have a problem—for example, when too many data are on the upper side or the lower side of the mean. And that kind of information, I think, is rightly developed in the working group that is interested in furthering the trending objectives.

There is some question whether just plotting is good enough, and I believe that it is not. But I think we need to work as an industry toward trending, and we need to use statistical methods that are available and used by many other industries to do this kind of work.

Question 2

Are vibration measurements of a pump during mini–flow testing, meaningful data for pump evaluation—especially in regard to calling a pump operable?

Response 2

You are quite familiar with the working group and you know we are still searching for that answer. But as I recalled earlier, I was discussing the industry responses to Bulletin 88–04, and there were several in there that actually included some vibration data, mini–flow and non–mini–flow, and there was absolutely no difference in the two.

So my guess is that it is a mixed bag; that you're going to find some where yes, it is indeed a problem, and even in the bag where it is not meaningful, I think you're going to find that splits into two also. You're going to find that in one of them what you are doing is actually masking your real vibration, and in the other you are actually causing it. In other words, what you are doing is actually causing so much damage that you are in fact reading something that is quite dangerous.

I'm saying that is what we are going to find, but we really need the data from people. I certainly have read the data, even in small mini–flows, where they said there is no difference at all between 90 percent best efficiency and 15 percent best efficiency. So it may be pump–specific.

Question 3

What is NRC's position on allowing a utility to change its tech spec pump testing frequency from monthly to quarterly to conform with Section XI and OM 6 required frequency? Does a tech spec change request the proper vehicle?

Response 3

Well, I think my answer would be to contact the licensing project manager.

Question 4

When changing a reference value, should it be done based on the first test after maintenance, or can it be developed using a cumulation of data points?

Response 4

Well, I personally believe that the first test after maintenance is very important, whether it be pumps or valves. Of course, we're talking pumps in this particular section. However, it isn't a requirement for OM 6.

I know we're certainly going to be looking into it. I believe we earmarked about 20 items to be looked at by the working group, and I believe that one of them is test after maintenance, asking how important is that first test.

And so all I can say right now is we haven't answered that. It is just my personal belief. Yes, it is a very, very important test to base it on. If you were to base it on a series, I just wouldn't feel good about it; and I can't even give you the technical reason. It's just a gut feeling.

Comment on Question 4

One thing, I've noticed, after we overhaul our pumps or are working with people after the overhaul, we always take an initial set of vibration data, usually spectrum, and then 24 hours later we would usually take another set. The second set of data is the one we go by. We just use the first set of data to find out if there is any major problem, and either shut the pump down or go back in.

But the second set of data we keep for trending purposes. We do keep the first set of data so we can have something to compare it to if the vibration 24 hours later goes up. But otherwise, we always use the 24–hour data. And we found that is a lot better. We usually get 30 to 40 percent lower amplitudes because of a break–in period.

Comment on Question 4

I just have one comment on that. I agree, but we have to remember that in the nuclear plants the majority of the pumps we are looking at are pumps that only operate when we are testing them. They are basically standby pumps, so we don't have that 24–hour break–in run to go back to.

MECHANICAL PREDICTIVE ANALYSIS METHODOLOGY VIRGINIA ELECTRIC AND POWER NUCLEAR STATIONS

VIRGINIA ELECTRIC AND POWER CORPORATION
JOE DEMARCO, STAFF ENGINEER

PRECISION MECHANICAL ANALYSIS, INC.
W. E. CLEVELAND, PRESIDENT

ABSTRACT

The purpose of this paper is to describe the unique, comprehensive Predictive Maintenance program implemented at Virginia Power's two nuclear plants. The program is unique because of it's thoroughness and because of the advanced tracking and trending techniques implemented.

INTRODUCTION

Today's manufacturing costs and continuously rising maintenance costs are requiring the Nuclear Industry to utilize all of the technologies available concerning predictive maintenance. At Virginia Power we have incorporated several state of the art technologies into one synergistic system enabling us to forecast a machine's present and future mechanical condition. To accomplish this goal, we have utilized the three primary indicators of potential mechanical problems, namely vibration analysis, lubrication analysis, and bearing temperature trending, in a truly effective, computerized predictive maintenance program. Additionally, we have a variable scheduling program with specialized features and a report-generation program with a built-in word processor. The system also has the ability to print reports, past and present, and to produce an individual machine's report history.

Predictive Maintenance efforts started with a pilot program in 1982 at the North Anna Power Station. The results of this pilot program were very successful and easily justified funding for further efforts. Starting late in 1983, a predictive maintenance laboratory was established and manned with two Senior Engineering Technicians. They not only performed the predictive analysis program for North Anna, but they also performed mechanical troubleshooting for both the North Anna and Surry Facilities. Presently both North Anna and Surry have predictive analysis technicians, equipment, and labs. In December of 1985, a decision was made to purchase the MECHANITRAK(tm) Predictive Maintenance System which was designed by a Florida based company, namely Precision Mechanical Analysis, Inc. or PMA. The MECHANITRAK(tm) system was implemented at all four nuclear units and is described herein.

We found that most Predictive Maintenance Programs incorporated a single method of nondestructive testing. The Predictive Maintenance Program at both Surry and North Anna Nuclear Plants is unique because it integrates the three best aforementioned indicators of potential mechanical problems into one computerized system. This is an important concept because some mechanical problems will not be detected as early when using only selected testing methods.

We have also incorporated an onsite contaminated oil analysis program. This service is also unique to PMA and is performed with their personnel on a regularly scheduled basis.

The Predictive Maintenance program at Virginia Power is still evolving. For instance, while currently utilizing some thermography, future plans call for it's incorporation into the overall program.

VIBRATION SPECTRUM ANALYSIS AND TRENDING

The two types of vibration analysis commonly used in Predictive Maintenance are Overall vibration readings (unfiltered) and Spectrum Analysis (filtered). Spectrum Analysis, the method utilized by the MECHANITRAK(tm) System, has many advantages over unfiltered readings. The major difference is that unfiltered readings will give only the machine's general operating condition, while Spectrum Analysis is more exact, pinpointing the specific problem. Unfiltered readings will show the existence of an irregularity, but without Spectrum Analysis exploratory maintenance may be required to uncover the exact problem.

It is necessary to establish a vibration baseline history for each machine to be incorporated in the Predictive Maintenance program. Vibration spectrums are established in three positions on each bearing on a given machine; specifically: horizontal, vertical, and axial positions.

Each accelerometer reading point is marked on each bearing by epoxying accelerometer bases. This insures that the readings are consistently taken in the same place on the bearing which allows proper data comparison.

Discrete frequency spectral analysis is performed on each bearing and position in the following manner. On the original baseline, the ten (10) highest amplitudes and associated frequencies are digitally stored by the computer for each measuring point. There is a vast amount of data gathered for the vibration aspect of the program alone and for that reason we utilize the computer software to point out the exceptions rather than the rules. If the machine's history is consistent with the unit's acceptable baseline (i.e. within predetermined tolerances) there is no need for further evaluation. This practice greatly reduces the amount of time required by the technician to review the data.

The following software functions are used as a basis to "flag" the exceptions rather than the rules and computer comparisons are performed in the following manner:

- Any new frequency that is non–existent in the original baselines.
- "Alert" ranges which are established in accordance with the baseline data for each predominant frequency. In setting these ranges, any previous baselines are compared for the entire history of the machine.
- "Alarm" ranges which are established in accordance with the baseline data for each predominant frequency. In setting these ranges, any previous baselines are compared for the entire history of the machine.
- "Low" ranges which are established in accordance with the baseline data for each predominant frequency. In setting these ranges, any previous baselines are compared for the entire history of the machine.
- The amplitude for each predominant frequency of the new reading is compared by the computer against a predetermined amplitude established by the system.
- The amplitude for each predominant frequency of the new reading is trended by the computer and "Out of Alert" or "Out of Alarm" projections are made. In order to forecast or predict the future amplitudes, three types of statistical regression analysis techniques are performed.
- During the establishment of the original baselines every amplitude of each predominant frequency is compared by the computer against a predetermined tolerance.
- If a problem frequency in the past vibration spectrum is now missing, the change is noted.
- The RMS value "overall" vibration amplitude is calculated for each vibration spectrum. The percent increase and/or decrease is determined, then compared against the baseline overall amplitude levels. Tolerances are automatically set by the computer software for the following types of exception reporting.
  - Alert level
  - Alarm level
  - Percent increase from the previous reading
  - Percent increase from the average reading.

LUBRICATION ANALYSIS AND TRENDING

In order to insure proper identification for each of the oil samples the computer program produces preprinted labels which are attached to the oil sample bottle for each scheduled oil sample. The computer program prints the following information on the labels:

Reservoir Description

- Reservoir Description
- Reservoir I.D. (lab identification)

- Blank space to write in the date the sample was taken

- Blank space to fill in number of hours lube has been in service

The following is a list of the most common lubrication tests that can be performed on the oil samples. It should be noted that not all of the tests listed in this section are performed on each reservoir sampled. The cost would be prohibitive and all tests listed are not necessary. A custom testing package was specified for each reservoir which was contingent on the sump capacity, criticality of the unit, type of system that was being lubricated, etc.

Direct reading emission spectrometric analysis

Direct–Reading (DR) ferrography

Particle counting

Micro–Ferrography (MF) or "Full" Ferrography

Macro–Contamination

Physical testing

Lube chemistry

Micropatch

Infrared absorption spectroscopy (IR)

Karl Fischer water test – measured in parts per million

Foaming tests

A more detailed list with the associated ASTM testing techniques is presented on the following page.

The most important tests can generally be divided into the following packages. From this list a starting point can be chosen for each reservoir being tested to which other needed tests can be added.

Test Packages:

Series 1:
- Spectrometric Metals
- Viscosity @ 40 degrees C
- Solids %
- Water %

Series 2:
- Spectrometric Metals
- Viscosity @ 40 degrees C
- Solids %
- Water %
- Total Acid Number (TAN)

Series 3:
- Spectrometric metals
- Viscosity @ 100 degrees C
- Solids %
- Water %
- Total Base Number (TBN)
- Fuel %

1PR:
Series 1
Particle count or DR ferrography
Infrared analysis

2PR:
- Series 2
- Particle count or DR ferrography
- Infrared analysis

3PR:
- Series 3
- Particle count or DR ferrography
- Infrared analysis

The science of lubricant analysis is becoming more and more complex, and for that reason we have listed each of the testing techniques and stated "Why This Test is Needed":

Metallo–Particulate Studies

Direct Reading Emission Spectroscopy. Monitoring metal deposition in lubricants at up to 10 microns (micrometers) size, essentially addresses "rubbing" wear regimen of equipment condition monitoring. Abnormal readings may trigger a request for Particle Counting, Analytical Ferrography or Micro–Patch Study to further resolve matters.

Contaminant metals (abrasives, coolant additives, etc.) may be detected, provided they are not above 10 microns in size.

Additive metals are detected, primarily to monitor consistency in product, as opposed to additive effectiveness or strength.

Particle Counting. Measures particulates from 5 to 40 microns in size, picking up where emission spectroscopy leaves off, and addressing the "fatigue" wear regimen of equipment condition monitoring. Particularly appropriate for units with on–board filtration. Abnormal readings may trigger a request for Analytical Ferrography or Micro–Patch Study. Particle Counting also measures non–metallic particulates in the 5–40 micron range.

Test Description	ASTM #
Ash %	D482
Base #	D2896
Cetane number (calculation only)	D976
Chloride	
Cloud point	D2500
Color test	D1500
Copper Corrosion	D130
Distillation	D86
Ferrography (analytical)	
Ferrography (direct reading)	
Fire point (flash point required)	D92
Flash point	D92
Foam test	D892
Freeze point	
Fuel %	D3524
Glycol (positive/negative)	
Gravimetric solids	
Gravity (API)	D287
Infrared (fixed points/single beam)	
Insolubles – coagulated pentane	D893
Insolubles – coagulated toluene	D893
Insolubles – uncoagulated pentane	D893
Insolubles – uncoagulated toluene	D893
Micro–organism (positive/negative)	
Micro–study	
Nitrate – parts per million	
Particle count (5 – 40 microns)	
Pensky–Martens	D93
Pour point (cloud point required)	D97
Silicates – parts per million	
Solids %	D91
Spectrometric metals (19 metals)	
Sugar (positive/negative)	
Sulfated ash	D874
Sulfur %	D1552
Total acid #	D664
Total base #	D664
Viscosity index (calculation only)	D2270
Viscosity @ 100 C	D445
Viscosity @ 40 C	D445
Water and sediment %	D1796
Water % by distillation	D95
Water % (estimate)	
Water – part per million (Karl Fischer)	D1744

Caution: Water contamination precludes a valid particle count; only size, not identification, is possible with particle counting.

Analytical Ferrography. A slide is made wherein magnetic materials (essentially iron–based) are systematically magnetized, ordered (large to small) and cured for viewing under microscope. This allows determination of the nature and shape of particulates which, in turn, often disclose the Casual effect (e.g., spiral cuttings usually denote abrasive wear).

Although magnets do not affect brass, aluminum and nonmetallic particulates, some quantity of these materials, if present in the lube, will precipitate on the slide simply from gravity's effect. The fact that they are random, and the ability to see their color or other characteristic features, allows further inferences to be drawn, rather than limiting one to only ferrous (iron–based) materials. Large abrasive chunks, for example, which would definitely escape emission spectroscopy, and which might exceed the particle counter's detection capability (if the particle count were rich in abrasives, but not too far off normal in total) can often be detected.

This test is highly recommended before making a decision to inspect (or not inspect) a piece of equipment, because it provides visual feedback to augment other tests which are inferential in nature.

Direct–Reading (DR) Ferrography. An abbreviated form of ferrography which ONLY addresses ferrous particles. Magnetic separation is again used to differentiate between "fatigue–oriented" wear (15+ microns) and "rubbing" wear (< 15 microns) reported as "L" and "S" (for Large and Small) respectively. As the ratio of L:S and the quantity of L + S changes, inferences are made. Abnormal readings may trigger a request for Analytical Ferrography or Micro–Patch Study.

Micro–Patch Study. The lube is filtered onto a filter patch and viewed through a stereo microscope. Particles of approximately 0.5 microns and greater are trapped on the filter. Test is best applied for identifying nonmetal contaminates, e.g., abrasives, fibers, rust, oxidation solids, etc. Abnormal findings may trigger a request for Analytical Ferrography.

Special Note: Rotary (non–reciprocating) equipment can show a tendency to develop abnormal levels of "fatigue–oriented" wear particles without an accumulation of "rubbing" wear particles. This strongly suggests that one should consider additional testing as a supplement to emission spectroscopy.

Summary: It is evident there are several approaches to detecting, monitoring and evaluating particulates and wear metals in lubes. It is important to recognize that no one test "does it all". This fact must be considered when selecting a "routine" testing package.

Lube Chemistry Tests

Infrared Absorption Spectroscopy (IR). This test addresses Organic (nonmetallic) aspects of the lube, which includes some additives, lube oxidation products, nitration products, coolant/water contamination, and differentiation between synthetic and mineral lubes.

Wave Functional LGTH Group	Typical Absorbance[a]
2.8 "HYD 2": water contamination (emulsified)	0.000–0.010
3.5 "C–H": hydrocarbon (mineral lube confirm)	<1.000
5.8 "OXY": oxidation (or synthetic confirm)	0.000–0.050
6.2 "NIT": nitration	0.000–0.050
9.6 "HYD": Glycol (or synthetic confirm)	0.020–0.100

a. Absorbance refers to specific wavelengths of light in the infrared spectral region which the functional group impedes. The greater the absorbance, the greater the concentration.

Water: As in any test for water, confirmation by infrared signifies a need for action, provided the sample was representative. Test is sensitive to as little as 0.1% water.

Hydrocarbon: Mineral lubes are basically carbon and hydrogen atoms linked together, hence the name "hydrocarbon". This value is expected to be high, verifying the presence of mineral lube. If the lube is a synthetic containing atoms/molecules in addition to hydrocarbons, this number will be appreciably lower than mineral lubes (perhaps 0.6 to 0.9). If synthetic lubes are in use and this number is more typical of a mineral lube, then contamination with mineral lube is implied.

Oxidation (OXY): When a lube oxidizes it gains oxygen. This oxygen attaches to the lube's basic molecular building blocks in a complex manner which is promoted (catalyzed) by heat and aeration (air entrapment). Its net effect is to thicken the lube and reduce its lubricity ("slipperiness"). Diagnostics such as air–fuel ratio checks, timing checks, vent point inspections, oil levels, emission control plumbing and load application will register a "high" number at this wavelength (0.6 to 0.9). This wavelength, therefore, helps serve as a confirming value for the presence of a synthetic, but it virtually precludes monitoring of oxidation.

Nitration: Nitration of a lube is the inclusion of nitrogen (basically in the form of oxides), and primarily occurs from non–ideal combustion cycles (either excessively lean fuel mixtures or improper spark timing). Gas engines are chiefly affected by this potential problem. The source on nitrogen itself is combustion air (of which nitrogen is the primary constituent). In the presence of moisture, nitric acid may form — this is a strong (corrosive) acid, capable of doing extensive damage to metallic components. It is imperative, therefore, that units be periodically checked for fuel mixture and general balancing, to suppress nitration potential.

Glycol: If a unit utilizes a glycol–based coolant, leakage of that coolant may be detected at this wavelength. Excessive glycol in the lube can readily ruin a piece of equipment; therefore it is important to monitor this potential contaminant (but also be aware that emission spectroscopy addresses potential coolant leaks differently, by detecting metallic additives, and this method is usually more sensitive than IR for coolant leak detection). There is also a tendency for this wavelength to be enhanced by oxidation or nitration.

Note: Some synthetics (particularly silicone–based) will register very high at this wavelength (0.8+), thereby confirming their presence, but precluding glycol monitoring.

Overall: It is important to have a new lube reference for IR. Most new lubes will register "background" (exhibit a value other than 0.00) for each wavelength above, this then becomes its reference point from which changes are monitored.

On the following page please note an example of a typical data sheet for oil analysis results.

Sample Oil Analysis Report

Unit ID: 02–EE–EG–2J/256			Generator Reservoir			Cap: .5 gal.
Lube Supplier: Gulf;	Type: Gulf Harmony;		Grade: 220			
Date samp'd	01/18/89	02/15/89	05/03/89	05/04/89	05/17/89	06/14/89
Hours	1157	1159	1195			
Wear metals (ppm):						
Iron	13	12	3	4	4	3
Chromium	0	0	0	0	0	0
Molybdenum	0	0	1	0	0	0
Aluminum	0	0	0	1	1	1
Copper	3	3	0	1	1	1
Lead	6	6	0	1	1	0
Tin	0	0	0	0	0	0
Silver	0	0	0	0	0	0
Nickel	0	0	0	0	0	0
Vanadium	0	0	0	0	0	0
Tatanium	0	0	0	0	0	0
Contaminant metals (ppm):						
Silicon	3	4	3	10	5	4
Sodium	1	1	0	2	0	0
Boron	0	0	0	0	1	0
Additive metals (ppm):						
Magnesium	0	0	0	0	0	0
Calcium	8	13	0	4	2	4

Sample Oil Analysis Report (continued)

Barium	7	7	0	0	0	0
Phosphorous	324	233	16	19	24	1
Zinc	15	18	4	6	11	8
Nonmetallic cont.						
Solids % vol.	0.2	0.1	<.1	<.1	0.2	<.1
Water % vol.	<.1	<.1	<.1	<.1	<.1	<.1
Lube data						
Vis.@ 40 deg. C	151	146	152	151	143	138
Total acid number	0.64	0.24	0.06	0.12	0.12	0.32
Other tests results:						
Infrared						
Water	0.000	0.000	0.002	0.003	0.003	0.001
Hydrocarbon	1.006	1.017	1.008	1.001	0.982	0.981
Oxidation	0.003	0.002	0.003	0.003	0.000	0.002
Nitration	0.006	0.004	0.005	0.003	0.000	0.000
Glycol	0.039	0.042	0.034	0.049	0.032	0.039
DR Ferrography						
Large particles	136.	134.	5.4	76.6	78.8	176.
Small particles	56.6	59.4	2.6	25.6	49.8	82.2

This lubricant is satisfactory for continued usage

Onsite Radioactive Lubricant Analysis

Changing the lubricants in radioactively contaminated systems is a very costly proposition in nuclear power plants. In addition to the basic expense of replacing the lubricant and the man–hours involved, there is also the ever increasing cost of low level waste storage and the ALARA considerations.

The used lubricant contains a wealth of information concerning the unit's mechanical condition and operating environment. Equipment maintenance, planned outages, and oil changes can be scheduled and performed based on the condition of the used lubricant.

The location of these units in high–rad areas, as well as their importance to plant production, make downtime for maintenance extremely expensive in terms of maintenance costs, lost production, and radiation exposure.

Therefore, the sensible alternative to routine oil changes and unexpected catastrophic equipment failures of contaminated systems is laboratory analysis of the lubricant to determine its condition and suitability for continued usage, as well as the mechanical condition of the unit itself.

There are very few laboratories which possess both the necessary licenses to transport and handle radioactive materials and the necessary equipment and expertise to determine the condition of the unit and lubricant. Those few laboratories who can provide this service usually have an extremely long turnaround time (weeks, even months) and the costs involved are usually prohibitive (usually in excess of $1000 per sample). Therefore, we analyze these oil samples onsite and obtain a complete analysis of the condition of the unit and lubricant within 24 hours.

BEARING TEMPERATURE ANALYSIS AND TRENDING

A bearing temperature baseline is established for each bearing on each machine that is incorporated in the Predictive Maintenance program. The bearing housing temperatures are taken and recorded at the same time the vibration spectrums are recorded and subsequently entered into the same MECHANITRAK(tm) program. The temperature data manipulation works on the same principle as the vibration and lubricant analysis, by flagging the exceptions rather than the rule. The temperatures are digitally stored in the computer's memory and compared in the following manner to determine the exceptions:

- The system compares the present reading with the previous reading and the average reading, then notes the percent change.

- The software compares the present reading with a preset tolerance and flags any reading that exceeds the tolerance.

DIGITAL DATA TRANSFER

We at Virginia Power are in the process of establishing a digital data transfer system that will allow data to be shared not only between plants but also with a corporate review system and with PMA. This allows data comparison from plant to plant and also allows us to trend the results of the program. Several people can review a critical problem over the phone, with all parties looking at the same data simultaneously. Figure 1 shows the data flow of this syst

REPORT GENERATION

Throughout this paper we have discussed the technical aspects of our program but one of the key elements to a truly successful Predictive Maintenance Program is reducing the data to a meaningful format for the maintenance personnel. We have to keep in mind that the purpose of our program is to make the maintenance personnel aware of three items:

1. Is there a problem with the unit?

2. If so, how serious is the problem?

3. What are the recommended corrective measures?

On the following page please note our report format which in addition to answering these three primary questions also provides the following key features:

- The severity of any amplitude is noted and can easily be reviewed.
- The probable source of the mechanical problems can be thoroughly described
- Recommended repairs and/or mechanical modifications shall be thoroughly described.
- Vibration amplitude can be reported in displacement, velocity, or acceleration.
- The software has the ability to provide a graphic representation of the numeric test results for each sample for trending purposes and shows the results from at least the six previous samplings.

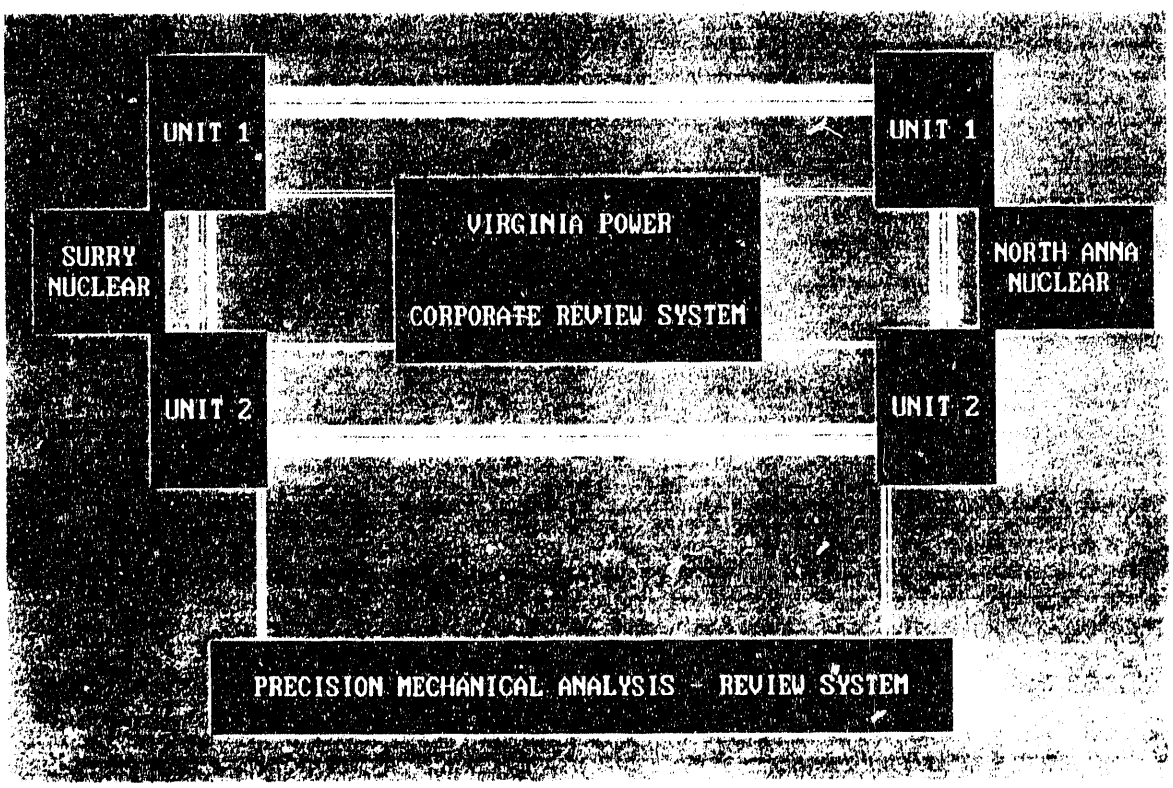

Figure 1. Illustration of data flow.

5.1 SAMPLE PREDICTIVE MAINTENANCE REPORT

PRECISION MECHANICAL ANALYSIS, INC.
5909-E HAMPTON OAKS PARKWAY
TAMPA, FLORIDA 33610
Telephone number (813)621-6463

****
* PREDICTIVE MAINTENANCE SEVERITY SUMMARY *
****

FEBRUARY 25, 1986 PAGE 1

CUSTOMER:VIRGINIA ELECTRIC
PLANT:SURRY STATION
PLANT AREA: ALL AREAS
RESPONSIBLE SUPERVISOR: JOHN HADDER

REPORT DISTRIBUTION: JIM OGREN
MIKE HADUCK

KEY

N=NORMAL
O=OBSERVATION
M=MODERATE
MH=MODERATELY HIGH
S=SEVERE
D=DELINQUENT
INVAL=INVALID SAMPLE
PA=PENDING ANALYSIS

MACHINE ID	MACHINE DESCRIPTION	INSP/DATE	VIB	OIL	TEMP
***** MAINTENANCE *****					
ARP-001A	CONDENSER VACUUM PUMP	02/11/86	N		N
ARP-001B	CONDENSER VACUUM PUMP				
	INBOARD DRIVER SLEEVE BRG.	02/05/86		M	
	(RES. ID: ARP1B/0392/175)				
CHHE-01A	CHILLER UNIT				
	MAIN OIL RESERVOIR	02/13/86		N	
	(RES. ID: CHHE1A/1864/117)				
CWP-001A	CIRCULATING WATER PUMP	02/20/86	N		N
CWP-001B	CIRCULATING WATER PUMP		D		D
EFP-0002	EMERG. FEEDWATER PUMP	02/26/86	M		MH
EGDG-001A	EMERG. DIESEL GENERATOR	02/12/86	M		N
EGDG-001B	EMERG. DIESEL GENERATOR	02/26/86	O		N
FWP-001A	FEEDWATER BOOSTER PUMP	02/17/86	O		N

PRECISION MECHANICAL ANALYSIS, INC.
5909-E HAMPTON OAKS PARKWAY
TAMPA, FLORIDA 33610
Telephone number (813)621-6463

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* PREDICTIVE MAINTENANCE SEVERITY REPORT *
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FEBRUARY 25, 1986 PAGE 1

CUSTOMER: VIRGINIA ELECTRIC
PLANT: SURRY STATION
PLANT AREA: ALL AREAS
RESPONSIBLE SUPERVISOR: JOHN HADDER

REPORT DISTRIBUTION: JIM OGREN
MIKE HADUCK

MACHINE ID MACHINE DESCRIPTION

***** MAINTENANCE *****

ARP-001B CONDENSER VACUUM PUMP----SEVERITY CODE: M
INBOARD DRIVER SLEEVE BRG.---- RES ID :ARP1B/0392/175

***************************MESSAGE***************************
The Tin and Lead content have increased during this sampling period. This indicates wear in the sleeve bearing. We suggest the bearing be inspected at the next possible opportunity. In the interim, however, this unit will be closely monitored.
****

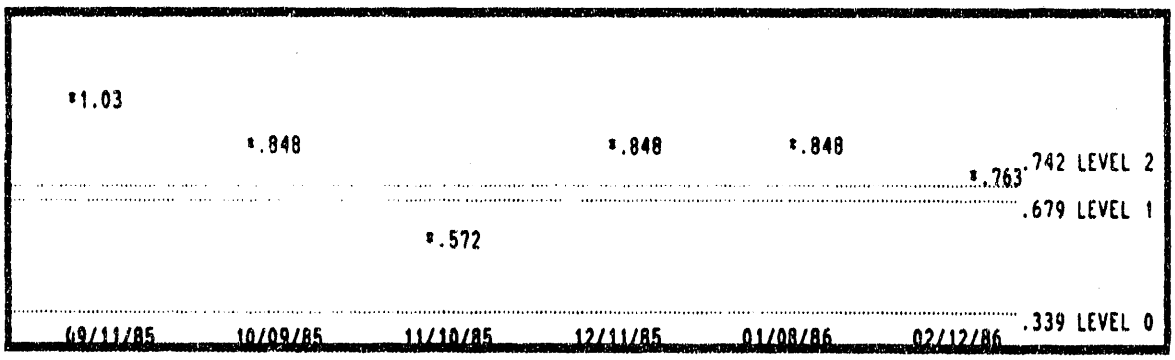
EFP-0002 EMERG. FEEDWATER PUMP----SEVERITY CODE: M

***************************MESSAGE***************************
There was a considerable increase in the temperature reading on the inboard turbine bearing from 120 deg. F to 156 deg. F. The vibration levels have also increased on this same bearing The unit is presently scheduled to be re-baselined three days prior to the upcoming outage and a detailed report will be issued at that time. The packing was also leaking on the inboard and outboard pump bearings.
****

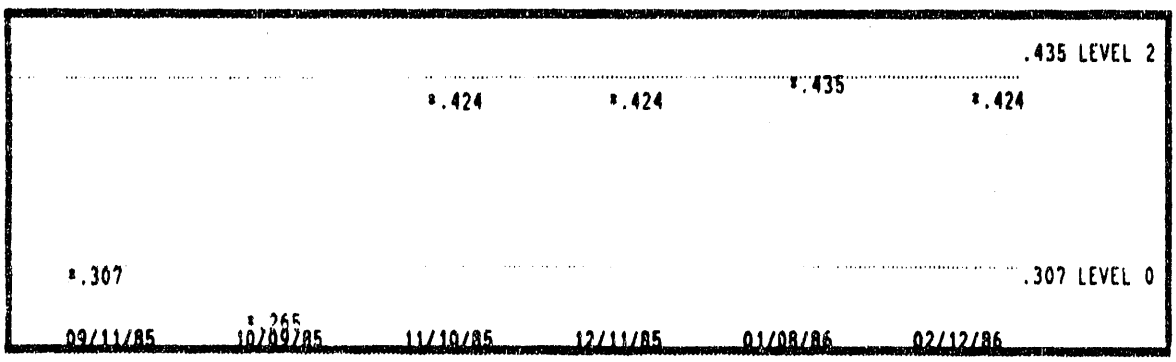
MACHINE ID	MACHINE DESCRIPTION
EGDG-001A	EMERG. DIESEL GENERATOR----SEVERITY CODE: M

**************************MESSAGE**************************
The vibration levels are basically stable, however, higher than desirable on this unit this month. The vibration levels indicate a possibility of fuel injection problems occuring. When time permits the fuel injection system should be inspected for correct operation. Please refer to the graphs in this section of the report.
****

MACHINE: EGDG-001A BEARING: DIESEL GEN-GENERATOR OB BRG POSITION: HORIZONTAL FREQUENCY: 9000 CPM
HARMONIC: 1 ANALYZED PROBLEM: GREATER THAN .28 in/sec AMPLITUDE UNITS: mils



MACHINE: EGDG-001A BEARING: DIESEL GEN-GENERATOR OB BRG POSITION: VERTICAL FREQUENCY: 18000 CPM
HARMONIC: 2 ANALYZED PROBLEM: GREATER THAN .28 in/sec AMPLITUDE UNITS: mils



MACHINE ID	MACHINE DESCRIPTION
EGDG-001B	EMERG. DIESEL GENERATOR----SEVERITY CODE: 0

*************************MESSAGE*************************

The levels on this unit are still not excessive on this unit; however, there are several new frequencies developing at multiples of rpm. The amplitudes of these frequencies are not excessive and the unit is listed in an observation state this month with no action required. This unit will be reinspected next month and you will be advised of any change.

****

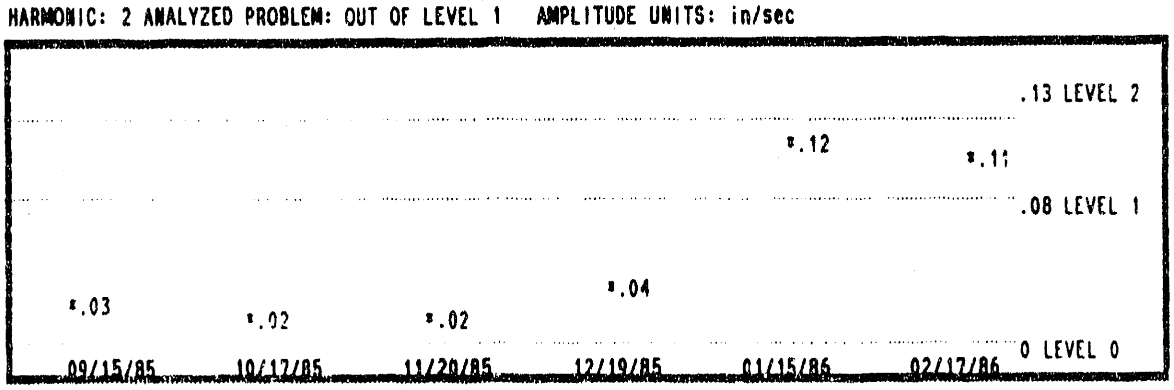
FWP-001A FEEDWATER BOOSTER PUMP----SEVERITY CODE: 0

*************************MESSAGE*************************

As indicated in the following charts there were considerable increases in the amplitudes at a frequency of two times the operating speed. The vibration levels have been like this since the outage. The vibration levels are not excessive but there is a very distinct difference in the levels at this frequency ever since the outage. There were no repairs made to the drive motor (to our knowledge); therefore, the most obvious conclusion for the source of the vibration is a difference in the drive coupling alignment.

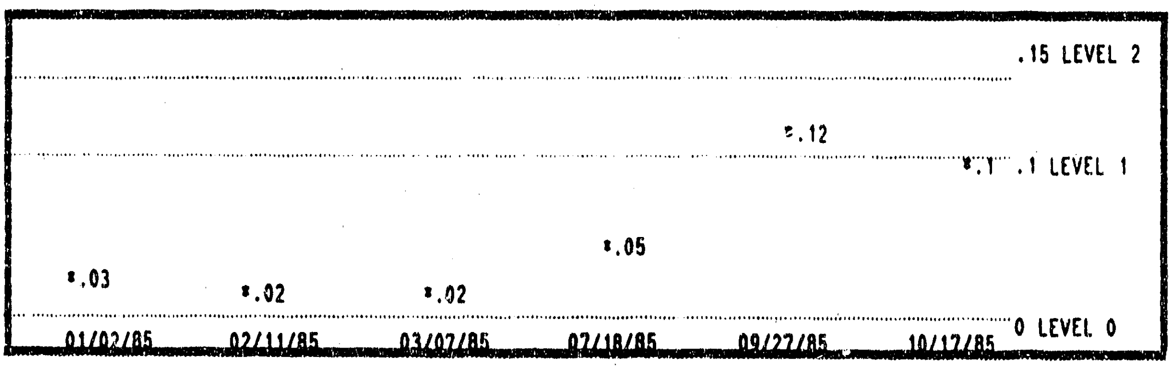
****

MACHINE: FWP-001A BEARING: DRIVER OUTBOARD BRG POSITION: HORIZONTAL FREQUENCY: 35700 CPM
HARMONIC: 2 ANALYZED PROBLEM: OUT OF LEVEL 1 AMPLITUDE UNITS: in/sec



MACHINE ID	MACHINE DESCRIPTION
FWP-001A	FEEDWATER BOOSTER PUMP

MACHINE: FWP-001A BEARING: DRIVER COUPLING/SHEAVE BRG POSITION: HORIZONTAL FREQUENCY: 35700 CPM
HARMONIC: 2 ANALYZED PROBLEM: PREVIOUS PROBLEM-NOW MISSING AMPLITUDE UNITS: in/sec



DELINQUENT REPORT

THE FOLLOWING ITEMS WERE NOT INSPECTED AS SCHEDULED DURING THIS INSPECTION PERIOD . PLEASE NOTE THE APPLICABLE REASON WHY EACH ITEM WAS NOT INSPECTED.

MACHINE ID	MACHINE DESCRIPTION
***** MAINTENANCE *****	
CWP-001B	CIRCULATING WATER PUMP

***************************MESSAGE***************************
This unit was out of service for maintenance. The motor was being disassembled at Tampa Armature Works for a pre-outage inspection.
****

EQUIPMENT LIST FOR SURRY STATION — VIBRATION

	Machine	Machine Description	Sampling Frequency
1.	1–BC–P–1A	1A–bearing cooling pump	Monthly
2.	1–BC–P–1B	1B–bearing cooling pump	Monthly
3.	1–CC–P–1A	1A–component cooling pump	Quarterly
4.	1–CC–P–1B	1B–component cooling pump	Quarterly
5.	1–CC–P–1C	1C–component cooling pump	Quarterly
6.	1–CC–P–1D	1D–component cooling pump	Quarterly
7.	1–CC–P–2A	Cooling wtr pump–charging mtr	Quarterly
8.	1–CC–P–2B	Cooling wtr pump–charging mtr	Quarterly
9.	1–CDP–1A	Chilled component cooling pump	Semiannual
10.	1–CDP–1B	Chilled component cooling pump	Semiannual
11.	1–CDP–1C	Chilled component cooling pump	Semiannual
12.	1–CDP–4A	Chilled water circ. pump	Semiannual
13.	1–CDP–4B	Chilled water circ. pump	Semiannual
14.	1–CDP–4C	Chilled water circ. pump	Semiannual
15.	1–CH–P–1A	1A–charging pump	Quarterly
16.	1–CH–P–1B	1B–charging pump	Quarterly
17.	1–CH–P–1C	1C–charging pump	Quarterly
18.	1–CH–P–2A	Boric acid transfer pump	Quarterly
19.	1–CH–P–2B	Boric acid transfer pump	Quarterly
20.	1–CH–P–2C	Boric acid transfer pump	Quarterly
21.	1–CH–P–2D	Boric acid transfer pump	Quarterly
22.	1–CN–P–1A	1A–condensate pump	Monthly
23.	1–CN–P–1B	1B–condensate pump	Monthly
24.	1–CN–P–1C	1C–condensate pump	Monthly
25.	1–CP–P–15A	Condensate polishing pump	Monthly
26.	1–CP–P–15B	Condensate polishing pump	Monthly
27.	1–CP–P–15C	Condensate polishing pump	Monthly
28.	1–CP–P–15D	Condensate polishing pump	Monthly
29.	1–CP–P–16A	Condensate polishing pump	Monthly
30.	1–CP–P–16B	Condensate polishing pump	Monthly
31.	1–CS–P–1A	1A–containment spray pump	Quarterly
32.	1–CS–P–1B	1B–containment spray pump	Quarterly
33.	1–CW–P–1A	1A–circulating water pump	Monthly
34.	1–CW–P–1B	1B–circulating water pump	Monthly
35.	1–CW–P–1C	1C–circulating water pump	Monthly
36.	1–CW–P–1D	1D–circulating water pump	Monthly
37.	1–EE–EDG–1	Emergency diesel #1	Quarterly
38.	1–EE–EDG–2	Emergency diesel #2	Quarterly
39.	1–EE–EDG–3	Emergency diesel #3	Quarterly
40.	1–FC–P–1A	Spent fuel pit cooling pump	Monthly
41.	1–FC–P–1B	Spent fuel pit cooling pump	Monthly
42.	1–FP–P–1	Motor driven fire services pmp	Quarterly
43.	1–FP–P–2	Diesel driven fire pump	Quarterly
44.	1–FW–P–1A	1A–steam gen feed water pump	Monthly
45.	1–FW–P–1B	1B–steam gen feed water pump	Monthly
46.	1–FW–P–2	Aux. feedwater Pump–2	Monthly
47.	1–FW–P–3A	3A–half size aux feed pump	Quarterly
48.	1–FW–P–3B	3B–half size aux feed pump	Quarterly
49.	1–FW–P–4A	Aux feed water booster pump	Quarterly
50.	1–FW–P–4B	Aux feed water booster pump	Quarterly
51.	1–IA–C–4A	Cont instrument air compressor	Monthly

Machine	Machine Description	Sampling Frequency
52. 1–IA–C–4B	Cont instrument air compressor	Monthly
53. 1–MG–1	Rod control–1	Monthly
54. 1–MG–2	Rod control–2	Monthly
55. 1–MS–T–1	Main turbine	Monthly
56. 1–RC–P–1A	1A–reactor coolant pump	Monthly
57. 1–RC–P–1B	1B–reactor coolant pump	Monthly
58. 1–RC–P–1C	1C–reactor coolant pump	Monthly
59. 1–RH–P–1A	1A–residual heat removal pump	Quarterly
60. 1–RH–P–1B	1B–residual heat removal pump	Quarterly
61. 1–RS–P–1A	1A–recirc spray pump inside	TBD
62. 1–RS–P–1B	1B–recirc spray pump inside	TBD
63. 1–RS–P–2A	2A–recirc spray pump outside	Monthly
64. 1–RS–P–2B	2B–recirc spray pump outside	Monthly
65. 1–SD–P–1A	1A–high pres heater drain pump	Monthly
66. 1–SD–P–1B	1B–high pres heater drain pump	Monthly
67. 1–SD–P–2A	2A–low pres heater drain pump	Monthly
68. 1–SD–P–2B	2B–low pres heater drain pump	Monthly
69. 1–SI–P–1A	1A–low head safety inject pump	Quarterly
70. 1–SI–P–1B	1B–low head safety inject pump	Quarterly
71. 1–SW–P–10A	Cooling wtr pump–charging pump	Quarterly
72. 1–SW–P–10B	Cooling wtr pump–charging pump	Quarterly
73. 1–SW–P–1AD	Emer svc wtr pmp 1A:diesel dn	Quarterly
74. 1–SW–P–1AM	Emer svc wtr pmp 1A:motor dn	Quarterly
75. 1–SW–P–1B	Emer svc wtr pmp 1B:diesel dn	Quarterly
76. 1–SW–P–1C	Emer svc wtr pmp 1C:diesel dn	Quarterly
77. 1–VP–P–1A	Station vacuum priming pump	Semiannual
78. 1–VP–P–1B	Station vacuum priming pump	Semiannual
79. 1–VP–P–1C	Station vacuum priming pump	Semiannual
80. 1–VP–P–2A	Intake vacuum priming pump	Semiannual
81. 1–VP–P–2B	Intake vacuum priming pump	Semiannual
82. 2–BC–P–1A	1A–bearing cooling pump	Monthly
83. 2–BC–P–1B	1B–bearing cooling pump	Monthly
84. 2–CC–P–2A	Cooling wtr pump–charging mtr	Quarterly
85. 2–CC–P–2B	Cooling wtr pump–charging mtr	Quarterly
86. 2–CDP–4A	Chilled water circ. pump	Semiannual
87. 2–CDP–4B	Chilled water circ. pump	Semiannual
88. 2–CH–P–1A	1A–charging pump	Quarterly
89. 2–CH–P–1B	1B–charging pump	Quarterly
90. 2–CH–P–1C	1C–charging pump	Quarterly
91. 2–CN–P–1A	1A–condensate pump	Monthly
92. 2–CN–P–1B	1B–condensate pump	Monthly
93. 2–CN–P–1C	1C–condensate pump	Monthly
94. 2–CP–P–15A	Condensate polishing pump	Monthly
95. 2–CP–P–15B	Condensate polishing pump	Monthly
96. 2–CP–P–15C	Condensate polishing pump	Monthly
97. 2–CP–P–15D	Condensate polishing pump	Monthly
98. 2–CP–P–16A	Condensate polishing pump	Monthly
99. 2–CP–P–16B	Condensate polishing pump	Monthly
100. 2–CS–P–1A	1A–containment spray pump	Quarterly
101. 2–CS–P–1B	1B–containment spray pump	Quarterly
102. 2–CW–P–1A	1A–circulating water pump	Monthly
103. 2–CW–P–1B	1B–circulating water pump	Monthly
104. 2–CW–P–1C	1C–circulating water pump	Monthly
105. 2–CW–P–1D	1D–circulating water pump	Monthly

Machine	Machine Description	Sampling Frequency
106. 2-FW-P-1A	1A-steam gen feed water pump	Monthly
107. 2-FW-P-1B	1B-steam gen feed water pump	Monthly
108. 2-FW-P-2	Aux. feedwater Pump-2	Monthly
109. 2-FW-P-3A	3A-half size aux feed pump	Quarterly
110. 2-FW-P-3B	3B-half size aux feed pump	Quarterly
111. 2-FW-P-4A	Aux feed water booster pump	Quarterly
112. 2-FW-P-4B	Aux feed water booster pump	Quarterly
113. 2-1A-C-4A	Cont instrument air compressor	Monthly
114. 2-1A-C-4B	Cont instrument air compressor	Monthly
115. 2-MG-1	Rod Control-1	Monthly
116. 2-MG-2	Rod Control-2	Monthly
117. 2-MS-T-1	Main turbine	Monthly
118. 2-RC-P-1A	1A-reactor coolant pump	Monthly
119. 2-RC-P-1B	1B-reactor coolant pump	Monthly
120. 2-RC-P-1C	1C-reactor coolant pump	Monthly
121. 2-RH-P-1A	1A-residual heat removal pump	Quarterly
122. 2-RH-P-1B	1B-residual heat removal pump	Quarterly
123. 2-RS-P-1A	1A-recirc spray pump inside	TBD
124. 2-RS-P-1B	1B-recirc spray pump inside	TBD
125. 2-RS-P-2A	2A-recirc spray pump outside	Monthly
126. 2-RS-P-2B	2B-recirc spray pump outside	Monthly
127. 2-SD-P-1A	1A-high pres heater drain pump	Monthly
128. 2-SD-P-1B	1B-high pres heater drain pump	Monthly
129. 2-SD-P-2A	2A-low pres heater drain pump	Monthly
130. 2-SD-P-2B	2B-low pres heater drain pump	Monthly
131. 2-SI-P-1A	1A-low head safety inject pump	Quarterly
132. 2-SI-P-1B	1B-low head safety inject pump	Quarterly
133. 2-SW-P-10A	Cooling wtr pump-charging pump	Quarterly
134. 2-SW-P-10B	Cooling wtr pump-charging pump	Quarterly
135. 2-VP-P-1A	Station vacuum priming pump	Semiannual
136. 2-VP-P-1B	Station vacuum priming pump	Semiannual
137. 2-VP-P-1C	Station vacuum priming pump	Semiannual

EQUIPMENT LIST FOR SURRY STATION – OIL RESERVOIRS

Customer: Virginia Power – Surry Station

Machine ID	Reservoir Description	Reservoir ID
1–BC–P–1A	1A–bearing cooling pump	1–BC–P–1A/368
1–BC–P–1A	1A–bearing cooling pump	1–BC–P–1A/369
1–BC–P–1A	1A–bearing cooling pump	1–BC–P–1A/370
1–BC–P–1A	1A–bearing cooling pump	1–BC–P–1A/502
1–BC–P–1B	1B–bearing cooling pump	1–BC–P–1B/371
1–BC–P–1B	1B–bearing cooling pump	1–BC–P–1B/372
1–BC–P–1B	1B–bearing cooling pump	1–BC–P–1B/373
1–BC–P–1B	1B–bearing cooling pump	1–BC–P–1B/503
2–BC–P–1A	1A–bearing cooling pump	2–BC–P–1A/374
2–BC–P–1A	1A–bearing cooling pump	2–BC–P–1A/375
2–BC–P–1A	1A–bearing cooling pump	2–BC–P–1A/376
2–BC–P–1A	1A–bearing cooling pump	2–BC–P–1A/529
2–BC–P–1B	1B–bearing cooling pump	2–BC–P–1B/377
2–BC–P–1B	1B–bearing cooling pump	2–BC–P–1B/378
2–BC–P–1B	1B–bearing cooling pump	2–BC–P–1B/379
2–BC–P–1B	1B–bearing cooling pump	2–BC–P–1B/805
1–CC–P–1A	1A–component cooling pump	1–CC–P–1A/380
1–CC–P–1A	1A–component cooling pump	1–CC–P–1A/381
1–CC–P–1A	1A–component cooling pump	1–CC–P–1A/382
1–CC–P–1A	1A–component cooling pump	1–CC–P–1A/806
1–CC–P–1B	1B–component cooling pump	1–CC–P–1B/383
1–CC–P–1B	1B–component cooling pump	1–CC–P–1B/384
1–CC–P–1B	1B–component cooling pump	1–CC–P–1B/385
1–CC–P–1B	1B–component cooling pump	1–CC–P–1B/807
1–CC–P–1C	1C–component cooling pump	1–CC–P–1C/386
1–CC–P–1C	1C–component cooling pump	1–CC–P–1C/387
1–CC–P–1C	1C–component cooling pump	1–CC–P–1C/388
1–CC–P–1C	1C–component cooling pump	1–CC–P–1C/808
1–CC–P–1D	1D–component cooling pump	1–CC–P–1D/389
1–CC–P–1D	1D–component cooling pump	1–CC–P–1D/390
1–CC–P–1D	1D–component cooling pump	1–CC–P–1D/391
1–CC–P–1D	1D–component cooling pump	1–CC–P–1D/809
1–CH–P–1A	1A–charging pump	1–CH–P–1A/392
1–CH–P–1A	1A–charging pump	1–CH–P–1A/393
1–CH–P–1A	1A–charging pump	1–CH–P–1A/810
1–CH–P–1B	1B–charging pump	1–CH–P–1B/394
1–CH–P–1B	1B–charging pump	1–CH–P–1B/395
1–CH–P–1B	1B–charging pump	1–CH–P–1B/811
1–CH–P–1C	1C–charging pump	1–CH–P–1C/831
1–CH–P–1C	1C–charging pump	1–CH–P–1C/832
1–CH–P–1C	1C–charging pump	1–CH–P–1C/833
2–CH–P–1A	1A–charging pump	2–CH–P–1A/398
2–CH–P–1A	1A–charging pump	2–CH–P–1A/399
2–CH–P–1A	1A–charging pump	2–CH–P–1A/813
2–CH–P–1B	1B–charging pump	2–CH–P–1B/400
2–CH–P–1B	1B–charging pump	2–CH–P–1B/401
2–CH–P–1B	1B–charging pump	2–CH–P–1B/814
2–CH–P–1C	1C–charging pump	2–CH–P–1C/815
2–CH–P–1C	1C–charging pump	2–CH–P–1C/834

Customer: Virginia Power – Surry Station

Machine ID	Reservoir Description	Reservoir ID
2–CH–P–1C	1C–charging pump	2–CH–P–1C/835
1–CN–P–1A	1A–condensate pump	1–CN–P–1A/404
1–CN–P–1A	1A–condensate pump	1–CN–P–1A/405
1–CN–P–1B	1B–condensate pump	1–CN–P–1B/406
1–CN–P–1B	1B–condensate pump	1–CN–P–1B/407
1–CN–P–1C	1C–condensate pump	1–CN–P–1C/408
1–CN–P–1C	1C–condensate pump	1–CN–P–1C/409
2–CN–P–1A	1A–condensate pump	2–CN–P–1A/410
2–CN–P–1A	1A–condensate pump	2–CN–P–1A/411
2–CN–P–1B	1B–condensate pump	2–CN–P–1B/412
2–CN–P–1B	1B–condensate pump	2–CN–P–1B/413
2–CN–P–1C	1C–condensate pump	2–CN–P–1C/414
2–CN–P–1C	1C–condensate pump	2–CN–P–1C/415
1–CS–P–1A	1A–containment spray pump	1–CS–P–1A/416
1–CS–P–1A	1A–containment spray pump	1–CS–P–1A/417
1–CS–P–1A	1A–containment spray pump	1–CS–P–1A/816
1–CS–P–1B	1B–containment spray pump	1–CS–P–1B/418
1–CS–P–1B	1B–containment spray pump	1–CS–P–1B/419
1–CS–P–1B	1B–containment spray pump	1–CS–P–1B/817
2–CS–P–1A	1A–containment spray pump	2–CS–P–1A/420
2–CS–P–1A	1A–containment spray pump	2–CS–P–1A/421
2–CS–P–1A	1A–containment spray pump	2–CS–P–1A/818
2–CS–P–1B	1B–containment spray pump	2–CS–P–1B/422
2–CS–P–1B	1B–containment spray pump	2–CS–P–1B/423
2–CS–P–1B	1B–containment spray pump	2–CS–P–1B/819
1–CW–P–1A	1A–circulating water pump	1–CW–P–1A/424
1–CW–P–1A	1A–circulating water pump	1–CW–P–1A/425
1–CW–P–1B	1B–circulating water pump	1–CW–P–1B/426
1–CW–P–1B	1B–circulating water pump	1–CW–P–1B/427
1–CW–P–1C	1C–circulating water pump	1–CW–P–1C/428
1–CW–P–1C	1C–circulating water pump	1–CW–P–1C/429
1–CW–P–1D	1D–circulating water pump	1–CW–P–1D/430
1–CW–P–1D	1D–circulating water pump	1–CW–P–1D/431
2–CW–P–1A	1A–circulating water pump	2–CW–P–1A/432
2–CW–P–1A	1A–circulating water pump	2–CW–P–1A/433
2–CW–P–1B	1B–circulating water pump	2–CW–P–1B/434
2–CW–P–1B	1B–circulating water pump	2–CW–P–1B/435
2–CW–P–1C	1C–circulating water pump	2–CW–P–1C/436
2–CW–P–1C	1C–circulating water pump	2–CW–P–1C/437
2–CW–P–1D	1D–circulating water pump	2–CW–P–1D/438
2–CW–P–1D	1D–circulating water pump	2–CW–P–1D/439
1–FW–P–1A	1A–steam gen feed water pump	1–FW–P–1A/440
1–FW–P–1B	1B–steam gen feed water pump	1–FW–P–1B/441
2–FW–P–1A	1A–steam gen feed water pump	2–FW–P–1A/446
2–FW–P–1B	1B–steam gen feed water pump	2–FW–P–1B/447
1–FW–P–3A	3A–half size aux feed pump	1–FW–P–3A/442
1–FW–P–3A	3A–half size aux feed pump	1–FW–P–3A/443
1–FW–P–3A	3A–half size aux feed pump	1–FW–P–3A/820
1–FW–P–3B	3B–half size aux feed pump	1–FW–P–3B/444
1–FW–P–3B	3B–half size aux feed pump	1–FW–P–3B/445
1–FW–P–3B	3B–half size aux feed pump	1–FW–P–3B/821
2–FW–P–3A	3A–half size aux feed pump	2–FW–P–3A/448

Customer: Virginia Power – Surry Station

Machine ID	Reservoir Description	Reservoir ID
2–FW–P–3A	3A–half size aux feed pump	2–FW–P–3A/449
2–FW–P–3A	3A–half size aux feed pump	2–FW–P–3A/822
2–FW–P–3B	3B–half size aux feed pump	2–FW–P–3B/450
2–FW–P–3B	3B–half size aux feed pump	2–FW–P–3B/451
2–FW–P–3B	3B–half size aux feed pump	2–FW–P–3B/823
1–RC–P–1A	1A–reactor coolant pump	1–RC–P–1A/454
1–RC–P–1A	1A–reactor coolant pump	1–RC–P–1A/455
1–RC–P–1B	1B–reactor coolant pump	1–RC–P–1B/456
1–RC–P–1B	1B–reactor coolant pump	1–RC–P–1B/457
1–RC–P–1C	1C–reactor coolant pump	1–RC–P–1C/458
1–RC–P–1C	1C–reactor coolant pump	1–RC–P–1C/459
2–RC–P–1A	1A–reactor coolant pump	2–RC–P–1A/460
2–RC–P–1A	1A–reactor coolant pump	2–RC–P–1A/461
2–RC–P–1B	1B–reactor coolant pump	2–RC–P–1B/462
2–RC–P–1B	1B–reactor coolant pump	2–RC–P–1B/463
2–RC–P–1C	1C–reactor coolant pump	2–RC–P–1C/464
2–RC–P–1C	1C–reactor coolant pump	2–RC–P–1C/465
1–RH–P–1A	1A–residual heat removal pump	1–RH–P–1A/466
1–RH–P–1A	1A–residual heat removal pump	1–RH–P–1A/824
1–RH–P–1B	1B–residual heat removal pump	1–RH–P–1B/467
1–RH–P–1B	1B–residual heat removal pump	1–RH–P–1B/825
2–RH–P–1A	1A–residual heat removal pump	2–RH–P–1A/468
2–RH–P–1B	1B–residual heat removal pump	2–RH–P–1B/469
1–RS–P–2A	2A–recirc spray pump outside	1–RS–P–2A/474
1–RS–P–2A	2A–recirc spray pump outside	1–RS–P–2A/475
1–RS–P–2B	2B–recirc spray pump outside	1–RS–P–2B/476
1–RS–P–2B	2B–recirc spray pump outside	1–RS–P–2B/477
2–RS–P–2A	2A–recirc spray pump outside	2–RS–P–2A/482
2–RS–P–2A	2A–recirc spray pump outside	2–RS–P–2A/483
2–RS–P–2B	2B–recirc spray pump outside	2–RS–P–2B/484
2–RS–P–2B	2B–recirc spray pump outside	2–RS–P–2B/485
1–SD–P–1A	1A–high pres heater drain pump	1–SD–P–1A/486
1–SD–P–1A	1A–high pres heater drain pump	1–SD–P–1A/487
1–SD–P–1B	1B–high pres heater drain pump	1–SD–P–1B/488
1–SD–P–1B	1B–high pres heater drain pump	1–SD–P–1B/489
2–SD–P–1A	1A–high pres heater drain pump	2–SD–P–1A/492
2–SD–P–1A	1A–high pres heater drain pump	2–SD–P–1A/493
2–SD–P–1B	1B–high pres heater drain pump	2–SD–P–1B/494
2–SD–P–1B	1B–high pres heater drain pump	2–SD–P–1B/495
1–SD–P–2A	2A–low pres heater drain pump	1–SD–P–2A/490
1–SD–P–2B	2B–low pres heater drain pump	1–SD–P–2B/491
2–SD–P–2A	2A–low pres heater drain pump	2–SD–P–2A/496
2–SD–P–2B	2B–low pres heater drain pump	2–SD–P–2B/497
1–SI–P–1A	1A–low head safety inject pump	1–SI–P–1A/498
1–SI–P–1B	1B–low head safety inject pump	1–SI–P–1B/499
1–SI–P–1B	1B–low head safety inject pump	1–SI–P–1B/887
2–SI–P–1A	1A–low head safety inject pump	2–SI–P–1A/500
2–SI–P–1B	1B–low head safety inject pump	2–SI–P–1B/501
1–FW–P–2	Aux. feedwater Pump–2	1–FW–P–2/827
1–FW–P–2	Aux. feedwater Pump–2	1–FW–P–2/828
2–FW–P–2	Aux. feedwater Pump–2	2–FW–P–2/829
1–MS–T–1	Main turbine	1–MS–T–1/452

Customer: Virginia Power – Surry Station

Machine ID	Reservoir Description	Reservoir ID
1–MS–T–1	Main turbine	1–MS–T–1/859
2–MS–T–1	Main turbine	2–MS–T–1 /453
2–MS–T–1	Main turbine	2–MS–T–1/860
1–EE–EDG–	Emergency diesel #1	1–EE–EDG–1/569
1–EE–EDG–	Emergency diesel #2	1–EE–EDG–2/570
1–EE–EDG–	Emergency diesel #3	1–EE–EDG–3/571
1–SW–P–1A	Emer svc wtr pmp 1A:diesel dn	1–SW–P–1AD/836
1–SW–P–1A	Emer svc wtr pmp 1A:diesel dn	1–SW–P–1AD/837
1–SW–P–1B	Emer svc wtr pmp 1B:diesel dn	1–SW–P–1B/838
1–SW–P–1B	Emer svc wtr pmp 1B:diesel dn	1–SW–P–1B/839
1–SW–P–1C	Emer svc wtr pmp 1C:diesel dn	1–SW–P–1C/840
1–SW–P–1C	Emer svc wtr pmp 1C:diesel dn	1–SW–P–1C/841
1–FP–P–1	Motor driven fire services pmp	1–FP–P–1/857
1–FP–P–1	Motor driven fire services pmp	1–FP–P–1/858
1–FW–P–4A	Aux feed water booster pump	1–FW–P–4A/842
1–FW–P–4B	Aux feed water booster pump	1–FW–P–4B/843
2–FW–P–4A	Aux feed water booster pump	2–FW–P–4A/844
2–FW–P–4B	Aux feed water booster pump	2–FW–P–4B/845
1–FP–P–2	Diesel driven fire pump	1–FP–P–2/846
1–CH–P–2A	Boric acid transfer pump	1–CH–P–2A/847
1–CH–P–2B	Boric acid transfer pump	1–CH–P–2B/848
1–CH–P–2C	Boric acid transfer pump	1–CH–P–2C/849
1–CH–P–2D	Boric acid transfer pump	1–CH–P–2D/850
1–FC–P–1A	Spent fuel pit cooling pump	1–FC–P–1A/851
1–FC–P–1A	Spent fuel pit cooling pump	1–FC–P–1A/855
1–FC–P–1B	Spent fuel pit cooling pump	1–FC–P–1B/852
1–FC–P–1B	Spent fuel pit cooling pump	1–FC–P–1B/856
New lube	New lube samples	New lube/830
New lube	New lube samples	New lube/854
New lube	New lube samples	New lube/861
New lube	New lube samples	New lube/868
New lube	New lube samples	New lube/869
New lube	New lube samples	New lube/870
New lube	New lube samples	New lube/871
New lube	New lube samples	New lube/872
New lube	New lube samples	New lube/873
New lube	New lube samples	New lube/874
New lube	New lube samples	New lube/875
New lube	New lube samples	New lube/876
New lube	New lube samples	New lube/888
New lube	New lube samples	New lube/889
New lube	New lube samples	New lube/890
New lube	New lube samples	New lube/890
2–EG–C–1	Emergency diesel air comp.	2–EG–C–1/001
2–EG–C–2	Emergency diesel air comp.	2–EG–C–2/002
00–EG–C–1	Emergency diesel air comp.	00–EG–C–1/003
00–EG–C–2	Emergency diesel air comp.	00–EG–C–2/004
1–EG–C–1	Emergency diesel air comp.	1–EG–C–1/005
1–EG–C–2	Emergency diesel air comp.	1–EG–C–2/006
1–CDP–1A	Chilled component cooling pump	1–CDP–1A–1B/891
1–CDP–1A	Chilled component cooling pump	1–CDP–1A–0B/892
1–CDP–1B	Chilled component cooling pump	1–CDP–1B–1B/893

Customer: Virginia Power – Surry Station

Machine ID	Reservoir Description	Reservoir ID
1–CDP–1B	Chilled component cooling pump	1–CDP–1B–0B/894
1–CDP–1C	Chilled component cooling pump	1–CDP–1C–1B/895
1–CDP–1C	Chilled component cooling pump	1–CDP–1C–0B/896

SUMMARY

Predictive maintenance is the most logical and cost–effective method of maintaining plant equipment, simply because maintenance and oil changes are performed on an as–needed basis only. In most cases the money saved from eliminating unneeded oil changes alone justifies a Predictive Maintenance Program.

There are many types of Predictive Maintenance Programs that can be established. To be effective, the use of computers is needed to handle the vast amounts of data quickly and with accuracy. This also enables the analyst to have all of the available data (vibration data, oil analysis, temperature readings, and past written recommendations) literally at his finger tips. The program needs to include more than one method of testing, such as combining a vibration analysis, oil analysis, and temperature analysis into one program. With these ingredients a Predictive Maintenance Program can be an indispensable managerial tool. Maintenance scheduling and ordering of replacement parts can be done well in advance. Instead of an unexpected forced shutdown of a critical piece of equipment, the maintenance personnel will be aware of any mechanical problems and can properly plan and schedule the repairs accordingly. At Virginia Power, Predictive Maintenance has enabled us to shift from a Reactive to a Proactive maintenance department.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Are the safety–related pumps in the plants covered by the program you describe? And, if not, why?

Response 1

Yes, the safety–related pumps are included. We have got predictive analysis groups at each station at Surry and North Anna, and they consist of two engineering techs per station.

We found early on that we had a lot of trend data on the balance–of–plant equipment, and not a whole lot on the safety–related equipment, primarily because that equipment was run on the back shift when operations could more easily schedule it.

With the advent of the digital data recorders and data takers, what we have done is trained our operators and revised our PTs so that the operators capture that data. They record on the PT the unfiltered readings that the code requires, and the data are then transferred over to the predictive analysis groups, so we have a corresponding spectrum to the unfiltered readings.

Question 2

How do you choose your discrete frequencies? How would you place OM 6 limits on a continuous, versus discrete, frequency and spectrum analyzer?

Response 2

The discrete frequencies are chosen by the system. It starts out with the highest 10 amplitudes on a particular scale in a frequency range. Then, if a new frequency comes into play, it starts automatically trending that, too.

The OM 6 limits how you would do that—I do not know that we would particularly want to trend equipment on a discrete frequency basis. You have several frequencies that will be oscillating, for example. You would have all of the equipment in the plant either in an alarm or alert stage continuously if you did apply OM 6 limits to a discrete frequency pattern.

Question 3

Have you had good results by doing temperature analysis?

Response 3

We have and we have not. That is a difficult one. By the program, we record bearing temperatures at the same point on the bearing cap each time vibration readings were taken, and that is on a monthly basis. There are a lot of variables for temperature trending. And I cannot say that we have detected a problem solely on the basis of bearing temperature trending. However, we have not discounted it yet, and we still continue to include it.

As I mentioned, we basically used pyrometers. At this point, we are looking at some infrared guns as well. But we are still continuing to include bearing temperature trending.

Question 4

Does your methodology employ monitoring of pump hydraulic parameters? If not, why not?

Response 4

I cannot speak for the IST portion of Virginia Power; I am speaking from a maintenance standpoint of predictive analysis. We look at a lot of things when we are out there recording data. We do record pump suction and discharge pressures as part of trouble–shooting. That is why we are there. It is not included in the trend program at this time.

In addition to that, normally when an inspection is performed for the predictive maintenance program, it is performed at the same time a performance test or a surveillance procedure is performed.

So, any predictive maintenance effort should be done simultaneously with the surveillance procedures so that you know you have that same consistency to track the data. In the earlier session, I heard a question about low flow, and that can definitely affect the pump. And it can really mess up the readings if you are not really careful.

So, we have to be concerned about the hydraulic parameters of the particular unit. And the best way to do that, from a predictive maintenance standpoint, is to ensure that the readings are taken concurrently with the performance test.

PANEL DISCUSSION: THE PURPOSE OF PUMP IST PROGRAMS – AVAILABILITY AND RELIABILITY

R. BAER (MODERATOR)
J. ZUDANS, L. SAGE, K. GUY, J. PAGE, P. ENG, J. DEMARCO, W. CLEVELAND

Question 1 (Pump Operability)

OM 6 and regulatory requirements effectively require that pumps be declared inoperable at a specific level of pump vibration. What criteria do you use to require pump shutdown? What is your assessment of the OM 6 vibration level limits?

Response 1

Panelist I personally am not familiar with the OM 6 standard. But, as for vibration levels, I do not just look at the overall vibration level, I look at the spectrum, and I also look at what the pump performance is. I think it is too difficult to come up with one generic level for vibration, whether it is in velocity, acceleration, or displacement.

I think it's something you have to look at overall. For ballpark equipment, if it is running at less than 600 rpm, we usually use 6 mils for shutdown limit. If it is running at over 1,000 rpm, our shutdown limit is usually 0.5 or 0.6 in. per second, overall. We do not necessarily shut down if the once per rev does not reach 0.6. It is the overall levels that we are more concerned about. In fact, we have shut pumps down that have been running down to 0.2 and 0.3 inches per second, depending on what the problem was in the spectrum, and depending on what the pump flow was.

Comment on Question 1

Panelist For the velocity anyway, that is fairly close to what we have used as the action level for OM 6—the required action. Again, we have to keep in mind that the pumps we are looking at are not run all the time; they run only, basically, when we are testing them, and often they are run on minimum flows.

So, you would expect to see slightly different vibration criteria for the overall spectrum of pumps than you would for pumps that are normally running and in service.

Question 2 (Deferred Maintenance Manhours)

How were the manhours that were "deferred" numbers determined at Gibson?

Response 2

Panelist To give a little background, the way we defer the manhours at Gibson, and they are starting to do this throughout the PSI system, is that every six weeks I receive a list—a preventive maintenance report that tells me six weeks in advance what equipment is to be scheduled to be overhauled.

We then pull out the past vibration trends, and the past oil trend, and also perform a performance test if it is a pump. And we assemble that into an information package with a cover letter that is routed through the plant.

It was felt that I was getting involved in too many areas I should not have been involved in. So, we decided we would let the supervisors and superintendents in their respective area have input into what equipment we deferred. And what we do is put this file package together. We tell them what the vibration overall readings are, whether we are in good shape, excellent shape, poor shape, what the oil reports indicate, and then what the performance levels are.

We make a recommendation, then, whether they should overhaul the equipment, or whether they should defer it for another year. And if they should not overhaul it and defer it, maybe we need to change a bearing, rework a coupling, or the like. Then we will route that through the plant.

Then what we do is on Wednesday of the week before the work is scheduled, we have a meeting in the afternoon at 2:00. And we sit down and go over each item that is scheduled for maintenance the following week. At that time we determine, based on what the superintendents and the people in the plant felt about my recommendation, whether they go along with me or they change and decide to overhaul it.

At the end of that, we determine how many manhours were determined for overhaul. Say we take a bottom ash pump. It takes about 160 manhours to overhaul that. If we decide that is not needed to be overhauled, we will take that 160 manhours and rededicate that to some area where we need the work. Or, if we do not need the work, then we will put them on preparing for outages or work that is coming up in the future.

So, essentially, it is a one for one. We do not really do away with the manhours; we just redirect them to where they are needed. It's a more effective use of our money.

Question 3 (Pump Operability)

IWP 3230(c) allowed analysis to be done to demonstrate that the—condition does not impair pump operability. Is this to mean the additional set of reference values of section 4500? If not, it does not seem clear why 3230(c) was dropped from OM 6.

Response 3

Panelist First of all, the members of the working group decided that if you are in the required action range after doing your test, and you have decided that you are in the required action range, there is no alternative but to declare the pump inoperable. That is the bottom line.

If you are in the required action range, but still meet safety system requirements, that is outside, again, of the scope of this code. The code is not there so you can meet the safety system requirements. They are to detect degradation and take corrective action on detecting degradation.

If you say that you are meeting safety system requirements but have a degraded pump, you nevertheless have said that the pump is inoperable.

Many pumps are put into systems and can deliver more than what the system requires to mitigate the consequence of the accident. Nevertheless, the testing is there to detect degradation. And once you identify that you are in a required action range, you must take corrective action. And that means you must declare it inoperable and you must go into your limiting conditions of operation for your plant until you can correct the problem.

The Code did not want to deal any more with an undefined specification that allowed lots of interpretation concerning when to declare something inoperable. This is not my personal feeling necessarily, but I think it is correct to say that if you are degraded to the point where you are in the required action range, you must take corrective action.

Question 4 (Identified Pump Failures Due to Vibration)

OM 6 equates an increase in vibrational energy with pump degradation. Have any pump failure modes been identified that are characterized by a decreased noise signature prior to failure? And, if so, how are such cases covered by OM 6?

Response 4

Panelist I think we determined there are some; I cannot give you specifics on what they are. But what I think we have come to realize is that anything that would not show up on vibration or show up as a decrease in vibration level would probably be severe enough to cause you to see a marked change in your hydraulic measurements also.

Question 5 (OM-6 Acceptance Criteria Range)

In your comparison of IWP and OM 6, you state that the owner can no longer set his own acceptance ranges. Question one, who now sets the new ranges? Question two, where does it (OM 6) say this?

Response 5

Panelist I think the way the new code reads, it omits the statement that the owner can set these new ranges if he cannot meet the acceptance criteria ranges specified in the code.

We deliberately took that sentence out because we believe it may not be appropriate to do that. However, if you want to set a new range, I assume that you could write a relief request for that. The code previously allowed you to go ahead and set new ranges without having to request relief or anything like that.

I think, again, this is probably a more conservative approach, because we believe the ranges are adequate as they are.

QUESTION 6 (Reference Points for Vertical Pumps)

What statistical data were used to justify more restrictive reference points for vertical and centrifugal pumps?

Response 6

Panelist Well, it is not more restrictive for centrifugal pumps. That is essentially the base case. It is more restrictive on the hydraulic measurements for vertical line shaft pumps. Those are the pumps with a long shaft suspended impeller type. And it has been our experience—very much my experience—that those sorts of pumps do not show degradation in vibration. So that is why the hydraulic parameters were set tighter there.

Response 6

Panelist Approximately in the 1974 to 1976 range, we identified, while I was working for the NRC, a problem with deep–draft pumps and the use of the vibration criteria in the code. The typical measuring point for vibration on deep–draft pumps was at the discharge housing for the pump, a very long distance in most cases from the actual impellers and bearings that are in the pump.

We felt that any vibration occurring in the bell housing of the pump near the impellers was being attenuated by the shaft and the column elements. And so you really were not doing a good vibration characterization for that particular pump.

In fact, some tests were performed at one utility where vibration looked fine; and upon disassembly of those particular pumps, there was severe degradation to the point where the pump had essentially destroyed itself.

At this point I was convinced, but had to convince others, that we might want to pay more attention to hydraulics. Nevertheless, since these pumps are normally always run in recirc., I am not sure that hydraulics are helping us that much. But we did want to go to slightly stricter hydraulic criteria in lieu of other techniques.

The only way that measurements could be made in the lower section of the pump or with vibration equipment that could withstand submersion and that kind of equipment at that time, and I think even now, is rather unreliable, in terms of life. So, that's kind of the history of where this came from.

The code people are putting out a lot of surveys. If you don't take them seriously, things are going to get more conservative. And if we don't get data for the industry, you have to realize things are going to get tighter and tighter out there.

Take surveys. I know they're a pain to fill out, but it's good information for us, and it helps us to write these codes and standards.

Panelist There is a very good EPRI report out. I don't have the number off the top of my head, but it was a research project that instrumented a number of the pumps of this type and showed that you can measure at the pumphouse, and you see absolutely nothing.

Voice I disagree that hydraulics is the resolution to the problem.

Panelist Propose an alternative. I mean, you have vibration, you have hydraulics. I don't know of anything else that we're looking at.

Panelist I will say one more time for the benefit of others that may not have heard my personal feeling; that is, that I have looked at the pumps that we have at St. Lucie and Turkey point, and I've looked at the ranges that are specified in OM 6, and I do not believe that I had one case where the pump actually fell into the required action range from past data that we've taken using the new acceptance criteria.

I completely endorse your recommendation that we get lots of data and can compare it. In the past we have not gotten that much data to support, but I think this is an attempt on the people and the code to try to prevent problems for the industry. And in lieu of something else, I personally would continue to support this position.

Panelist We did a detailed analysis on a fire safety pump at one particular utility, which I will keep to myself, but what did is we put underwater probes on the shaft segments between the bearing supports, and we also put accelerometers down at the bottom of the housing. And the vibration levels looked just fine up on the top of the motor, all over the top of the motor. Everything looked just fine, but what we found was that there was over 80 mils of deflection between the bearings, and there's a three–inch thick report that we issued that particular client.

And so I lost a lot of faith from that particular instance as far as measuring vibration on the top of the unit to determine the condition of what is going on 80 feet into the ground. That is a real–life thing that happened, and it's hard to believe that we would have 80 mils. worth of deflection on the shaft, and all the vibration levels up on the top were just fine. That really took place two or three years ago.

Panelist I had an experience with a vertical shaft. I had a 45–degree shaft, and I won't tell you the manufacturer, but I've since found out they've done this several times. I don't know if it's planned or not, but the utility kept snapping shafts, and we were taking vibration ratings up on the motor and getting acceptable readings, 600 rpm motor, vibrator readings at less than 2 mils.

We took a good pump out of the ground and did some bump testing on it, and suspected a natural frequency. Didn't find anything. Found the mode shapes. Put the pump back in the ground and ran it with instrumented probes, and we wiped out the probes. They were gapped out at 70 mils.

We took it back out, did some more bump testing, and did a modal analysis on it, and found out that the manufacturer on their cutlass bearings put every one at a node point, so, you know, at little or no support. I would have liked to have put accelerometers on the casing and seen what the casing was doing. But that's typical—I talked to other people and found out the same thing, that it will scare you if you put probes underwater.

Panelist We did the same thing on a unit that we tested, and we found the same thing, that they had the bearings on the node points as far as the modal analysis was concerned.

Question 7 (Range on Reference Values)

The way we're running the test now, you are supposed to set a given flow rate or a given delta p by adjusting the system valves or throttling something, and it says set it exactly at the reference. It doesn't give a band, plus or minus anything.

The problem arises both when you are trying to set a speed on, say, a turbine-driven pump to a reference speed, again with no band given. You have a digital tachometer. The spec might be 2000, and you can play with it all day long, and you can adjust anywhere from 1995 to 2005, but you never will peg it right on 2000.

If we don't give a spec out to the people doing the test, they will come up with their own, whether you want them to or not. And that same kind of problem happens when you try to peg flow, or delta p, or any other parameter. You always need, when you are working in the real world with a real system, a range of some sort.

Response 7

Panelist If I look at the tables for acceptable ranges, those are my ranges. If you look at the table, I guess it has ranges that you can fit within for flow, right?

Attendee You have a range for flow and a range for delta p. Maybe I'm being too strict on myself, but I thought I was supposed to put one or the other right on the reference value and then apply the other ranges to the parameter that was allowed to vary, either the flow or delta p, whichever one I didn't specify.

Panelist I think you are right. And my view is that you need to set your procedure the best that you can for your plant, and then if that deviates from the acceptable ranges in the standard, then you might have to ask for some relief.

But I think you are right. I think it's difficult to get it right on the money each time.

Panelist I think these ranges should be set in the procedure, set to X flow plus or minus something. I don't know what the NRC's comment on that would be, how they would view that, or whether they're going to insist upon you being right on. And I don't know if anyone here wants to comment on that.

Panelist It is the real world, and you can't get it right on any one number. And generally speaking, if there isn't a defined range, you take a look at the instrument accuracy, a plus or minus five percent. I'm not going to quibble over something like that.

Attendee I haven't had anyone quibble over it yet. We set our own spec for whatever we're trying to fix at plus or minus 0.5 percent, which is exceedingly tight for a lot of our instrumentation and systems, and no one has quibbled over it yet. But as long as it is not specified, you are always at the mercy of whatever inspector comes down. And some of them are reasonable, and some of them are not.

Panelist Well, I'm very glad you brought this point up, and hopefully the Code Committee will take care of that.

Question 8 (Accounting for Instrument Inaccuracies)

Is it acceptable to establish a low required action range for a pump at 90 percent of reference flow and/or pressure if there exists only a 10 percent design margin between the system design requirements and actual pump performance, or is it necessary to restrict the low required action range to account for the worst case of instrument accuracy of plus or minus two percent such that low required action range would be at 92 percent of reference flow or pressure?

Generally, how do we take into account instrument inaccuracies when establishing required action ranges? And maybe that is the broader question to ask.

Response 8

Panelist The answer to the second part is yes, do the second part. Adjust your acceptable action ranges in accordance with taking into the calculation the accuracy of your instruments, and if you come up with 0.92 as the limit your system needs to meet, you need to meet that. In other words, we're assuming that 0.9 will always meet system requirements for shutting down the plant or mitigating the consequence of accidents. If it does not meet that, you must up your required action range.

Panelist OM 6 or Section XI does not address the fact of what do you have to meet for, your tech spec or FSAR commitment for pump flow. And that is something that we're talking to the NRC about right now. And it may lead to changes in the code in the future, but for right now that is a whole separate test, so don't get the two tests confused.

Attendee I concur with that. I just had an inspection, and we had to put a tech spec reference in. And you really need to separate out your tech spec value and test from the others and just stick with an IST and an IST format. Otherwise, you can get in a little trouble.

Question 9 (Hydraulic Performance of Degraded Pumps)

During the experiments you just described, describing specific vertical line shaft pumps with very high vibration problems, was the degraded condition of these pumps also manifested as diminished hydraulic performance at the time of your experiment?

Response 9

Panelist Yes, we did note it. The pump flow seemed to be off quite a bit, and there was quite a bit of wear on the impellers. We determined that from putting some test gauges on and doing some poor man's pump calculation, and noticed the flow was down around 8,000 gpm. And so we pulled the pump out and repaired it. During the repair we saw some vibration problems where there had been some rubbing on the impeller, so we decided to put the probes on, and subsequently did the oil analysis and found out the bearings were put in the wrong place.

They've been pretty consistent about it. I know of other pumps that they've done the same thing with, so they are consistent. That's about all you can say.

Panelist It didn't happen that way on the one we worked on, because we were very meticulous. It was a brand-new pump that was assembled. Every shaft was measured for runout to make sure it was straight. Every bearing clearance was checked. The impeller clearances were okay.

The pump performed properly hydraulically, but we found that there was a large amount of deflection between the cutlass bearings on the shaft, and that was attributable to the reason why they had consistent shaft failure and bearing failure in the unit. Our report went back to the manufacturer, and they redesigned the unit as a result of the tests that we performed.

Panelist I will add, in the case that I observed, we had a natural frequency of the column of 900 rpm on an 1800 rpm pump. Every bearing carbon insert was missing after the test, and there was significant degradation of the lower journal bearings of the two-stage housing. And the pump was still pumping, and there was a quarter-inch cut in the shaft near the bottom of the housing. And all through this the vibration was just fine.

Question 10 (Fixed Flow Rate)

For mini-recirc flow testing with mini-flow recirc lines that are instrumented for flow, what constitutes "fixed flow" when there is no capability to adjust flow? In other words, what is the allowable deviation for flow rate from reference flow rate?

Response 10

Panelist As far as I know, it has already been answered. You have no adjustment. You may develop a smaller curve just based upon the fluctuations in that mini-flow recirc line. But you can not do without adjustments, because you don't have an adjustment. So you simply compare the two values against the allowable values.

Question 11 (IWP Reduced Range Limits)

Article IWP 3210 states that reduced range limits can be specified in lieu of Table IWP 3110–2 limits if the 3100–2 limits cannot be met. Is it the intent of this article to allow the establishment of less conservative limits such as a low required action range set at 85 percent of reference value instead of the code allowable 90 percent of reference value? If this is not the intent, please explain what the intent is.

Response 11

Panelist I guess the OM 6 people just said 90 percent was the minimum.

Question 12 (OM–6 Required Action Range)

A value of 0.01 in. per second, a value of greater than 6 VOR would only result in a vibration level of 0.06. However, corrective action would be required. Isn't it likely that any maintenance performed on the pump would probably result in making the vibration of the pump worse?

Response 12

Panelist I think it is our feeling that if you have that big of a change in your vibration, even though it is still at a fairly low level, something has changed significantly mechanically with the pump, and you should at least investigate it. And at that point, I would say personally you would want to do spectral analysis and maybe some other things to see why you are getting this increase. So you might not want to go ahead and just pull the pump apart, but something has changed that you really have to take a look at.

Attendee On what you said before, my question is, that based upon what the Panel has said about once you get to a limit, then you are in a required action range, and you have to do something to repair that pump. I don't know what option you have allowed by OM 6, other than doing the repair on that pump, to declare it inoperable, because no matter what you do, you would still be in the required action range.

Going along with that other response, you might be in a situation where if you do maintenance on the pump, you will end up worse off than when you started. Is there something you did like monitor it more closely to see if it is really a problem, or is it just something the pump's working in or whatever?

Panelist I would just say, and I have a bit of a disagreement on what should be done, that this is a change we are looking at right now. I believe there are situations where it is not appropriate to do a repair because you are in the action range, that it may be something that is not hurting the pump, but you still need to investigate it, and that is the point. And I don't know what's going to happen; that is to say, there are some differences of opinion.

Panelist If deviations fall within the required action range, Table 3, "the pump shall be declared inoperable until the cause of the deviation has been determined and the condition corrected." That doesn't mean to me that I have to take it apart unless I find that necessary. For example, if the mounting bolts came loose during the las. cycle of operation, I now find out that it's a mounting problem, and I retighten it, and I bring it back in. I don't have to go and do anything else about it. But the way the code is written right now—and I understand your point, I think, completely—you have to go into an inoperable status.

Attendee That was the question. Maybe when the task group is considering this position that is something we'll have to take into consideration for people who have pumps that are very smooth–running. Any time you state a position, there is always the "what if I have this condition," I just thought it was maybe something that you could consider up front by this task group.

Panelist It puts the inspector in an incredible bind the way that it stands. For example, if you've got a pump that has got 0.01, and then you come up with 0.06, I trot out to your plant armed with my OM 6 and say "oh, well, the pump's inoperable. If the pump's inoperable, why aren't you in your tech spec LCO," and off we go to the races.

I understand that similar situations for—I know we're supposed to be talking about pumps, but the only analogy I can come up with is for fast–acting valves, valves that stroke in one second, and maybe that day the guy has a sore thumb and so when he hits the button, he will hit it a little bit slower or a little bit fast, and all of a sudden wham, the valve is inoperable. And so that might be something that you can add to your lookout list.

Panelist I believe we would still end up with the pump being inoperable until you did some sort of analysis, say yes, I've looked at it, and I know why it's changed, and it's okay. But you still have to do that initial inoperability until you can make that decision.

Attendee I agree with you there. I agree, inoperable. But my point is that maintenance might not be the answer. And by the earlier discussion it felt like really that's your only option on the OM document, and I just think that needs to be addressed.

Panelist It is going to be addressed. It has been noted by Howard Maxwell very clearly, and actually he noted several possible changes to that very aspect.

I think he had three mean reference values for different reasons on pumps. It goes against the basic philosophy that the task group on vibration monitoring used when we set up that criteria, that we would use a change from reference as being the primary criteria and the absolute values as the backup criteria. However, after Mr. Maxwell put in his comments, our initial response was just that it is against our basic philosophy, and we think people can live with it. At least we think right now, the way it is written.

We are already thinking about reforming a task group on vibration, and who's going to be on it I'm not sure. However, that is one of the very first items that will be looked at.

Attendee That's one of the things that spurred my question. He wrote a paper a couple of years ago about vibration and absolute limits. And one of the things he did say, if it was a very smooth running pump, that if you do maintenance, unless you are in a factory condition, chances are you might make it worse if it's low already.

Question 13 (Vibrational Reference Data)

As stated by the Panel, it is better to take vibration reference data after the pump has run for approximately 24 hours. In nuclear plants many pumps are in a stand-by condition and are not run in prior to obtaining baseline data. By taking reference data prior to a 24–hour run–in aren't we putting ourselves at risk for taking data that may not be repeatable in future IST tests?

Comment on Question 13

If a typical test is only two minutes or five minutes, it is many years before you could even approach 24 hours, and in many cases I don't think the utility will want to run on mini flow which is what's required through the mini flow line for 24 hours.

Response 13

Panelist There's really nothing in the code that would prevent you from taking the reference when you initially run and then run in for 24 hours and take another set of reference values.

I believe the NRC would accept it if you are doing something like that. The problem comes if it is a pump that has to run on a recirc line, and you can't do a 24–hour run–in, then you are pretty well stuck.

Panelist Conversely, there is a requirement that you cannot stay at power unless you do another test after doing maintenance. You have to do something right after you complete your corrective action to demonstrate that the pump is, again, reading the same or an adequate reference value. And I agree that a test after establishing an initial reference value would be good to break in.

Now, I'm not a pump expert, but as I remember, it may not take 24 hours to stabilize some type of modifications to a pump. It may take much less time. And so maybe some of the experts could tell us about that.

Attendee We had to replace an impeller and bearings, and normally the auxiliary feed water pumps are only run on recirc. What we found is that after two or three of our routine tests the levels had dropped and we used that as a justification for doing another reference value test so that we reset the values down so we were getting meaningful alert and action values.

So, you don't necessarily have to do a 24–hour run. You can just wait. And if your trend shows a decrease from the original reference test and then levels out, then you can say my pump has worn in and justify running another reference test.

Panelist At some of the facilities we monitor the bearing temperatures on the unit after maintenance and after the bearing temperatures stabilize. That normally gives the pump a sufficient amount of time to run so that everything is normalized, rather than letting it run for a 24– hour period.

That ensures also that the temperatures are not excessive on the unit after a rebuild and that the thrust clearances are set properly and so forth, and so we use temperature as a criterion, and when the temperatures normalize for so many consecutive readings over a given period of time, then we know that the unit is stabilized and we use that as a criteria rather than time.

Panelist When I stated earlier about the 24 hours, that is, for several reasons, normal. I think in

95 percent of the items we check immediately after overhaul. As soon as you turn it on, you are going to know if there is a problem or not. The 24–hour period is so the guys who don't like to stay up until four o'clock, can come back in the morning and take the data again. That's probably realistic.

But if you don't know there's a problem when you turn it on—and I use temperature also. Once temperature is stabilized, your data is going to be pretty consistent for you.

Question 14 (Vertical Pump Hydraulic Parameters)

Do the more restrictive hydraulic parameters for vertical line shaft pumps apply for those pumps whose casings are not submerged and are accessible for vibration monitoring?

Response 14

Panelist I would say no. Those criteria do not apply to those pumps because you can get pretty close to the bearings there.

Most of the vibration measurements on other centrifugal pumps are made right on the casing, on the flange, and so if you can do essentially the same thing where the bearings are near that location, then I would say that that does not apply. But it would have to be based on a case–by–case type of evaluation like a close coupled volute pump. It's vertical, but it is not a deep–lined shaft pump, and so I would say that that does not apply.

Question 15 (IWP Reference Values)

The ASME Code allows that, following maintenance, the old reference values may be verified rather than setting new reference values. Does the old reference have to exactly match the test data to be considered verified, or is there an old reference that can be used?

Also, what documentation is necessary to record reconfirmation of old reference values?

Response 15

Panelist I would say—and this is purely personal opinion—if you are still in the acceptable range, that's probably good enough. But, again, that is just personal opinion.

Panelist Here's another one. If you are replacing your impeller, then I would say that that does not necessarily hold true, because an impeller can have a significant change in hydraulic performance and the pump will be just fine. But if you are just replacing bearings, then you should come pretty close to the old reference values with this, and I wouldn't apply any plus or minus 10 percent criteria.

I think that that would be an NRC item of scrutiny. Why did you go ahead and establish new criteria when you only replaced bearings or tightened up some bolts or prepared a seal or those kinds of examples.

But you could run into having to establish new reference values when dealing with parts that affect hydraulic performance.

Question 16 (Flow Measurability of Boric Acid Pumps)

Plants having boric acid pumps in their IST program typically have problems with flow measurability and repeatability. Has anyone found a viable solution to this problem?

Panelist The only time that I have ever looked at the inservice testing of a boric acid transfer pump, they were attempting to use—and I say attempting to use—ultrasonic flowmeters, and they had several problems. I know what the code committee's opinion is of ultrasonic flowmeters. I do not know what anybody else's opinion is of ultrasonic flowmeters, but they are very, very difficult to play with. They are very difficult to calibrate.

I found that the licensee basically said, they don't get emergency power and they pulled them out of the program because they do not mitigate the consequences of an accident; so they said.

Panelist We had lots of debates on changing the use of emergency power, and I think this is an example where the words we ended up with are a lot better. It talks about the function of those pumps.

If those pumps don't perform one of those functions of mitigating an accident or being needed to shut the reactor down, it doesn't have to be included in the IST program. And, I don't think anyone ought to be punished because they put a pump on the emergency power source, frankly.

Panelist Just looking at the IWP being on the emergency power source is not the only criteria for putting it in. You don't have to put in every pump that is on emergency power.

That is just like a screening criteria. If it isn't, then it's out. But because it is doesn't mean it's in.

Panelist We've seen cases where plants have had to add boric acid transfer pumps into their IST system. That's not unusual. If you need the shutdown margins, you have to have that system in.

But we run into the same kind of questions that you are giving us. How do you measure this thing?

In cases like that, we've said use tank volumes, time measurements. Use whatever other techniques you can to measure that kind of flow. We understand that type of problem, but there are other techniques you can use. Concentration changes, volume changes, and tanks.

Question 17 (Effectiveness of Pump IST Program)

Moderator Would one of the panel care to talk about whether the IST program, as a whole, has been helpful in cutting down the instances where pumps were declared inoperable for other reasons and, therefore, avoided outages, or is it just something extra to do and without much benefit?

Response 17

Panelist First of all, let me go back to the other comment that we didn't get to talk about, which is, should the IST be used for operability.

I think that question has to be considered with respect to how the tech specs are being used today to invoke the requirements of the IST programs. While I'm not going to answer that question, because I don't have the answer, I think it is a very important issue that needs to be considered, with regard to whether IST programs have reduced unplanned days off line, and have increased reliability of pumps.

I think that data are pretty scarce. Whether that is true, I think there are cases where IST has identified pumps that have been degraded and corrective action has been taken without having to shut the plant down.

Some of those cases are in multiple pump systems where you only need two or three, and one completely fails, which is kind of a gross type thing. But we have, I think, identified some cases where vibration has helped.

The intent, though—and this is my personal opinion—is that we need to marry many of the maintenance practices, preventive maintenance measures with IST in the future, because then we will really be doing prevention by prediction, which is a very favorite term that a lot of what the Code was there for.

Some of the parameters are hard, when you are doing mini recirc pump testing. It's hard to detect degradation in that mode. But, I think that we're going in the right direction with the new standards.

We may not be there yet, as evidenced by the number of tasks that we are facing on the working group right now trying to identify ways that we can improve the existing new standard.

Our eyes are open. We are very happy to receive all comments and we are very happy, I think, to work on them. But I can't say with 100 percent surety that I have any data that say that IST is effective in every case identifying pump problems.

Panelist I think the purpose of IST is something very different than plant availability and reliability. I think the sort of program that Virginia Power is doing is great, but that sort of program doesn't mean a lot when the pump is only run when you test it. That's fine for a pump that makes money for you, but not for a stand-by pump.

And our IST is really just for stand-by pumps, and it is to make sure that they will perform their safety function and not to make sure they will be enhancing plant availability or reliability.

Panelist Since 1983, the most notable change that I've seen in NRC Region III is a change in focus. I realize that electric power plants, nuclear power plants, are there to make money. I think we all know that.

But, there is a subtle change in plant personnel. Now, there is more concern; there's more cohesiveness about a given component rather than, let's make electricity.

Don't get me wrong. It's still let's make electricity, but the care and the forethought that are put into the writing of surveillance test procedures and whether or not they definitively identify component degradation prior to catastrophic failure is another question.

But the change in philosophy with the members of the plant's staff I see as very encouraging. It is something that supports the safe operation, and I am beginning to get a sense of the commitment to safe operations aside from just operations to make money. And so, that is very heartening, and that's just a comment.

Attendee I'm a member of the O&M 6 and 10 working group, and I'm also a member of some various task groups. One of the task groups is on stroke timing. I am also vice chairman now of a new special task group just formed. I'm vice chairman with chairman Larry Sage on the response to the April NRC meeting in regard to a lot of questions on various aspects and what we need to do in the future of IST.

There are some various task groups like stroke timing that need information from the utilities to aid in coming up with a new guideline on what specifics we're going to take for stroke timing.

I happen to have some forms along with us on a questionnaire, and I would like some of the utility members to please take them and fill them out. It will further aid the O&M committee to work on stroke timing and also on valve operability and valve testing.

And, also, in the future we're going to be issuing a pump parameter questionnaire to the utilities to further help in getting our information out.

Panelist I certainly endorse that statement. The committee truly attempts to come up with consensus standards. There's a lot of knowledge out in the industry, and most of the members of the working groups and subcommittees and the main committee are utility or architect/engineer people.

And the way to get your thoughts and your opinions factored in is, number one, volunteer to be on a working group. Someone will give you a yes almost assuredly. And the other alternative is what that gentleman just suggested, let your views be known. Fill out questionnaires and get your expertise factored into the process.

PLANNED SESSION PAPERS

PART 2: PUMP TESTING TECHNIQUES AND LIMITATIONS

ENHANCE PUMP RELIABILITY THROUGH IMPROVED INSERVICE TESTING

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INTRODUCTION

Periodic testing of pumps installed in nuclear power plant safety systems has long been used to verify the overall readiness of the pump, its driver, and its support systems to respond to safety demands, demonstrate mechanical soundness, and prove hydraulic capability. Although the pump testing programs used in most operating plants test some, or part of, the fluid system, including heat exchangers and various types of valves, clearly the major focus of such testing is on the performance of the pump. ASME Section XI, Subsection IWP, Inservice Testing of Pumps in Nuclear Power Plants, has been the basis for such test programs and describes various parameters that should be monitored. ANSI/ASME OM–6 has taken the place of Subsection IWP, and has changed a number of test ranges and shifted the focus of testing for centrifugal pumps from hydraulic criteria to an emphasis on changes in mechanical criteria, such as vibration levels. EPRI has undertaken a study to assess the effectiveness of existing testing programs to accurately monitor and predict performance changes before either pump performance degrades or an actual failure occurs. Anticipated changes in inservice testing techniques are directed towards enhancing the validity of test data, ensuring its repeatability, and avoiding deterioration of the pump assembly. There is a new–found interest in test programs of all types that has occurred, in part, because of an increase in reported pump degradation and pump failure.

Inservice testing of pumps, which has long been a basis for assuring operability, has apparently produced an opposite effect; namely, the appearance of a reduction in reliability.

Discussion

Recent experience has shown that an increasing number of safety–related pumps, many with little running time, are beginning to fail. Questions that are being raised include: are safety–related pumps less reliable than other types of pumps in power plants, are the testing programs themselves the cause of such problems, and, in the final analysis are the programs in place not sensitive to detecting actual performance capability or changes in vital parameters?

With the average age of nuclear plants increasing, the large number of nuclear plants in operation today, and the even greater number of safety system pumps installed and subject to periodic testing, problems related to performance failure are to be expected. There appears to be evidence that much of the testing that was meant to determine the operational readiness of the pumps may actually be causing excessive wear and damage to the equipment. The number of reports stating that pumps are degrading to performance levels with minimum margins is increasing. Questions are being raised about the type of tests being run, their frequency, the diagnostic tools used to assess the performance of the pumps, and the overall design of systems. It appears that existing guidelines are not adequate. Testing programs seem to produce results of varying accuracy and occasionally the testing itself results in equipment damage or failure, which is contrary to the objectives of the program.

The purpose of inservice testing of pumps, as defined by Subsection IWP–1100, is to produce results "to be used in assessing operational readiness of the pumps during their service life". Without further guidance, compliance with the above requirements has resulted in the design of fluid systems that makes it difficult to assess the performance and operational readiness of pumps not tested at, or near, their original design basis conditions. Many fluid systems were not designed to comply with the intent of test criteria. Results of some test programs produce data that make effective diagnosis difficult. Repeatability of data is a persistent problem. On occasions, some form of pump damage has occurred with little, if any, evidence of change in test parameters, either mechanical or hydraulic.

Perhaps there needs to be a clearer distinction between inservice hydraulic and mechanical testing. Methods should be directed towards producing the necessary evidence of capability without being unnecessarily intrusive to plant personnel nor damaging to the equipment.

To improve pump testing programs in nuclear service, we need to review some of the major characteristics of pumps that lend, or do not lend, themselves to inservice testing. Since inservice testing is intended to show the response of equipment to safety demands, as well as proof of performance, attention should be focused on the elements of system design that can enhance pump testing by improving the reliability of test results, minimizing the stress on the pump and its associated components, and produce a verifiable record of performance.

To ensure a clearer focus, let us look at the pump assembly itself, the pump/system, and discuss some of the problems, possible solutions, and a summary of the impact of these solutions.

The following pump characteristics need to be considered:

1. Pumps are designed to operate, not to remain idle. Infrequent operation at any flow, but especially flows other than design, often is more stressful than continuous operation or no operation at all.

2. Pumps are hydraulically designed usually to operate over a wide range of flows but perform best at, or near, their best efficiency point of flow.

3. Performance of "identical pumps" varies from pump to pump, even when they are built at the same time. Performance of the same pump can vary from test to test. Although manufactured to exacting tolerances, no two pumps are ever exactly the same; therefore, small performance variations should be expected.

4. Routine maintenance of the pump or its parts can, and does, alter some variables that are monitored in test programs. The design of the system and the acceptance criteria for testing should recognize these variations.

The goals of enhanced pump inservice testing must recognize the above pump characteristics.

The emphasis on pump capability (flow and head) is high because, for many systems, a pump's capacity and developed head are the only verifiable elements of a system's performance. The adequacy of a pump to perform initially, and over a long-term basis, in the event of a safety incident must be sustained and demonstrated over the life of the plant.

The need to improve a test program is also dependent on the following:

- The amount of margin, or excess capacity, that exists in either the pump or system. This margin is a direct measure of the existing pump's capability over the minimum required for safety.

- The arrangement for testing the equipment under conditions as close as possible to those required by safety analyses.

Temporary or test arrangements test only the pump and may not reflect the actual system demand.

To enhance pump reliability we need to expand on the following points which were mentioned earlier.

1. The frequency of pump testing and the duration of inservice tests, by themselves, should not result in degradation of either mechanical or hydraulic performance. If the pump, its driver, and support auxiliaries have been specified properly for the required design conditions, as well as the type of testing necessary to prove inservice capability, testing will not harm a pump. Because of time and manpower restraints, inservice tests occur on a regular schedule, but usually last for only a brief period of time. Let's consider the following points:

   Point 1 — Frequency Of Tests — pumps, as well as any other rotating piece of equipment not in service, tend to suffer from the problem of rotor sag. Rotor sag is particularly severe where bearing spans are great and, due to specific service conditions, shaft diameters are small and internal clearances are tight. Long periods of in operation affect the performance of a pump since inoperation often leads to premature internal wear and possible bearing wear or failure, possible gasket leaks, and the unavoidable issue of internal corrosion. Frequent tests minimize the degree of rotor sag and provide an earlier signal of pump problems. One solution to this problem would be to open up internal clearances. Although the efficiency would fall, the reliability of the pump would be measurably improved.

   Point 2 — Duration Of Testing — any bearing, after being left idle in standby or emergency service, requires time after starting to initiate proper oil flow and to achieve a stable running temperature. Shortened tests prevent the establishment of stable conditions, both mechanical and hydraulic, and can result in degraded mechanical and hydraulic performance.

2. Selecting an optimal test point where the pump should be tested on the pump's characteristic curve would result in improved test results. Unless limited by the Manufacturer, or restricted by limitations of Net Positive Suction Head (NPSH) or the rating of the pump's driver, a centrifugal pump can be expected to operate at any point on its curve; however, if a reliable point of operation for the purposes of inservice testing is needed, a different set of criteria must be used.

A typical centrifugal pump head–capacity curve is illustrated on Figure 1. Even though performance is normally plotted from zero flow, flows less than Point A or greater than Point C should be avoided since they are usually regions of internal hydraulic instability where changes in performance from test to test may not be indicative of a problem.

Often existing system configuration dictates that safety pumps be tested at or below the minimum continuous recirculation flows specified by the pump manufacturer. Testing flows at or near a minimum flow point specified by the manufacturer may be convenient but usually result in an undefined point. Many times these test flows are below the 25 percent flow shown as Point A in Figure 1. This can result in inservice tests that produce inconsistent test results that may or may not indicate that the pump has degraded.

Figure 2 illustrates the impact of wear on the measured performance of a pump. Experience has shown that both the new pump and worn pump characteristic curves share the same shutoff head if the impeller(so diameter(s) and speed remain the same. However, as a pump wears at its wear ring(s) and other close running fits, internal leakage back to the suction side occurs and the curve dips (illustrated by the dotted line). If inservice testing is performed in very low flow regions, the ability to differentiate between the new and worn condition approaches a level below instrument accuracy. This is particularly true if the head capacity curve is flat in the area

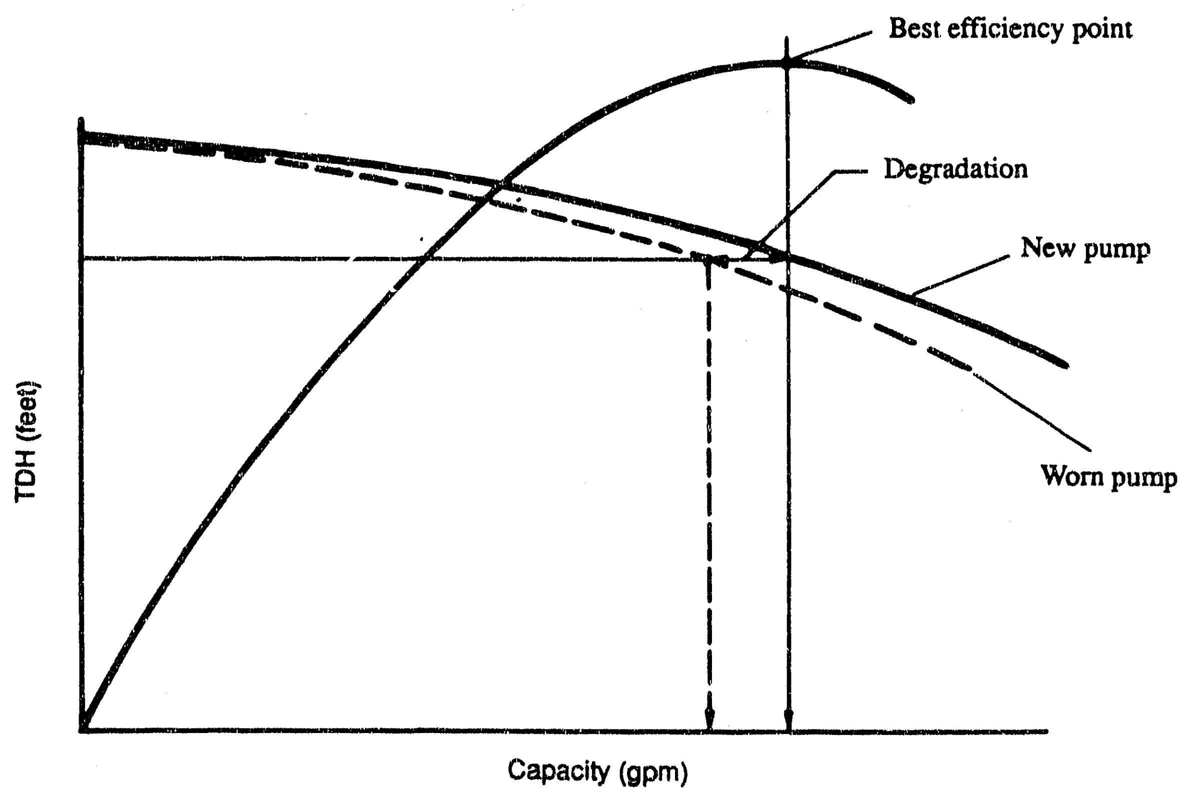

Figure 1.

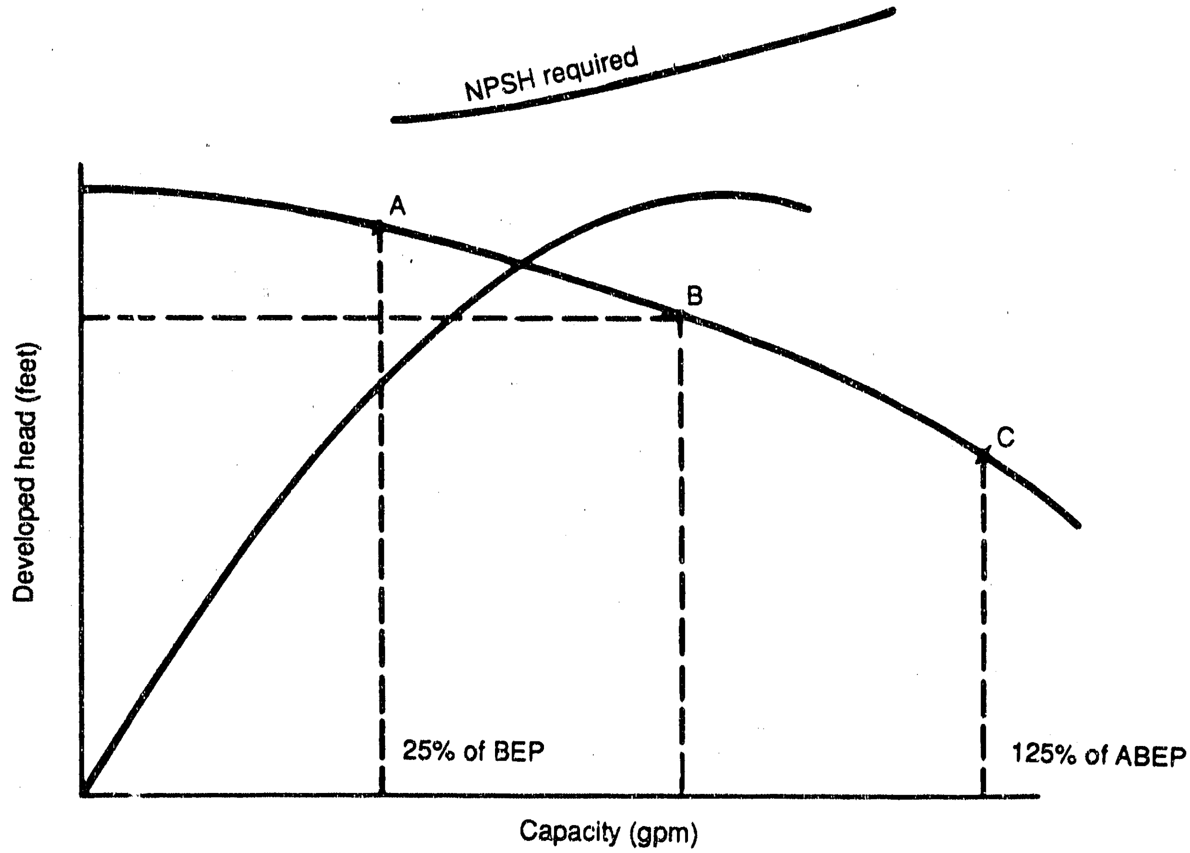

Figure 2.

where inservice test data is taken. Clearly, relying on testing in this region will not show the true degradation that is occurring further out on the pump curve. Furthermore, the characteristic of a worn pump condition can be attributed to flow–induced errors in measurement instrumentation and not degradation. Differentiation between flow errors and a degraded pump curve is difficult with a good set of shop test data and initial field data, but differentiation is more difficult if such comparisons are made after the equipment has been in operation for some time.

3. Variations in shop and field performance test readings, used as reference values, are a normal occurrence in testing centrifugal pumps. The specification for the pump and the selection of reference values must be made so as to avoid too narrow margins for test readings since variations will occur.

   Pumps are designed and manufactured with minor casting and machining differences. When taken collectively, the mechanical differences, even among identical pumps, can result in minor, although measurable, changes. For purposes of shop testing and eventual field testing qualification at the inservice test capacity, judgment should be exercised to ensure that a sufficient margin exists to permit both wear and normal variation in performance.

4. Perhaps a more overlooked element of variation in field test results is the problem of maintenance of the pump over the life of the plant, through preventative programs. Along with improving testing methods, controls over repair or upgrade of pump assemblies should be tightened when such work impacts on either the hydraulic or mechanical performance.

   Minor changes in clearances within the pump, the cleanup or finish of hydraulic surfaces, changes in metallurgy as well as routine adjustments in rotor setting, bearings, seals, and supports can have a significant influence on pumps and their performance.

An attempt has been made through the evolution of ANSI/ASME OM–6 to improve the credibility of inservice testing by employing a greater reliance on monitoring mechanical parameters, such as vibration levels, after a pump has been initially installed and baselined. Although it is true that a pump's noise level or vibration can change due to upset or distress, years of experience have shown that noise level or vibration monitoring is not a predictable signal of loss in performance or imminent failure. A heavy duty pump, or one over–constructed for the service, may show no measurable sign of distress due to damage or wear, either in initial testing or inservice testing if tests are performed at off–design points. It is far easier to establish nominal hydraulic performance levels for pumps that are predictable and repeatable than to quantify a level of mechanical ruggedness for evaluation that enables vibration or noise measurements alone to signal a change in performance.

To improve the quality of test results required from inservice testing of pumps, a new level of standards should be considered.

CONCLUSIONS

Enhancement of pump testing to assure safety functions should encompass program elements that address the needs of the pump and the system. Many components must respond properly and perform satisfactorily for a system to function.

In recognizing the mechanical and hydraulic characteristics of pumping equipment and systems, there should be a clearer distinction between performance and demand capability.

Performance

Routine testing of a pump at less than design conditions introduces both uncertainty in results and the potential for damage. Performance capability, including margins, should be installed at the time of initial design criteria and acceptance testing both in the shop and field. Inservice testing for performance in the field usually results in lost performance unless the pump is tested at its design conditions and properly maintained.

If a pump has been hydraulically designed with ample capacity margin and properly constructed for the range of fluid conditions when exposed to local environmental elements, the pump will not lose performance if run at its design conditions or not run at all.

Demand

Overall system response to a safety indication should be the focus for future testing. Measurement of performance margins in fluid systems for components other than pumps, such as piping, valves, and heat exchangers should be undertaken to demonstrate where additional system margins can be found.

Continued inservice testing should focus on the proper response of all the components within a system. Due to the complexity of controls and instrumentation, errors in circuitry pose a greater risk to safety system function than the small loss of pump capacity that could occur due to performance testing.

SUMMARY

To enhance pump testing programs, i.e., to reduce pump degradation and demonstrate overall performance margins, additional test/demonstration programs should be studied to determine the extent of margin in other system components. Determination of such margins could reduce the burden of pump tests.

For many plants, changing test line sizing and arrangement, relocation of instruments, and requalifying pump performance will be a major undertaking. Overall, however, enhanced testing procedures will lead to a significant reduction in deviations in pump performance, reduction in pump damage and failure, increased level of confidence in pump and system response and capability, and eventually a higher level of assurance in plant safety.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

I agree with everything you are saying up there basically, except my one question is that pumps are failing, circulating at a high rate. Why does the industry data support a high failure rate?

Response 1

We'll listen to a presentation later on today concerning failure rates. I think I'd like to put it into two categories. The failure of pumps is two–fold. It fails to produce either a certain required capacity or head, depending on either tech spec requirements, and then there's the failure of the pump mechanically.

I think we've all seen the documentation of a disturbing rate of failure of auxiliary feed water pumps at high head per stage in the split case or barrel type pumps that have been working at very low minimum flows. Again, the failure is qualitative.

I think EPRI, as well as other people perhaps, will be presenting the data later on that indicate the failure is there. I think we know it's there, and I think we suspect that unless we do something to recognize why these machines are on the ragged edge of demonstrating hydraulic performance or mechanical performance, we're going to see continued high failure rates.

There's a controversy still circulating around about what a minimum flow is. Manufacturers are wrestling with the problem only because of what this is going to mean to the population where presently it's installed.

Perhaps I see it a little bit more intimately because of my day–to–day involvement with the pumping industry as well as with the user industry. But without being able to give you numbers and cite statistics, I think we will all probably see the same reports as well as the areas that are coming in about why pumps are failing.

I think the bottom line is that pumps are failing because we're not operating them the way they were designed to operate, and we're not conveying back into the design of the systems and the design of the pumps what we expected them to do.

I don't think all hope is lost. I think we can retrofit these systems and vastly improve the performance and almost to the point without eliminating operator error get to the point where these pumps are as reliable as the balance of plant equipment that is run continuously.

MEASUREMENTS OF VIBRATIONAL PARAMETERS FOR PUMP TESTING

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ABSTRACT

Determining a pump's condition by vibration measurement requires an understanding of the generated forces, the equations of motion of the machine's structure, and the effect that instrumentation has on the measurements. Unbalance, misalignment, bearing damage, off design operation (miniflow, cavitation), and other machinery problems generate forces inside the machine which usually can be inferred by the motion of the outside of the machine. These forces are usually generated on the rotor and may cause damage to the rotor or bearings by wear, overload, or fatigue. These forces also cause motion of the structure of the machine which can be measured as displacement, velocity, or acceleration. Within certain limits, this vibration can be used in categorizing the severity of the machine's condition.

The measurement system including the transducer, conditioning electronics, calculation choices, and measurement electronics will significantly affect the results. Transducers, with some distortion, transform vibration into electronic signals. Various parameters may be calculated from the resulting wave forms, including four different kinds of amplitude. The waveform can be Fourier transformed, and the resultant spectra analyzed for amplitudes and frequencies.

VIBRATION BASICS

The forces in a pump which can lead to vibration come from either the driver through torque and moments on the coupling, or from other sources in the machine's environment, e.g. nearby pumps, pressure pulsations in the fluid, etc. The torque applied by the driver causes rotation of the rotor and is opposed by forces on the rotor from the fluid.

The rotation of the rotor produces centrifugal forces and forces at the coupling. The centrifugal force is proportional to the square of the speed, the mass of the rotor and the distance between the mass center and the rotational center. This "unbalance" force is perpendicular to the axis of spin and is resisted by the bearings and the structures that support the bearing. The force rotates with the rotor and so at a single point on the bearing appears as a sinusoid with a frequency equal to the speed of the machine. The frequencies of vibration are important parameters and are usually denoted as multiples or harmonics or the running speed of the machine. For example, the unbalance force is a 1xrpm frequency force. The forces on the rotor from the coupling are both radial and axial and can be once per revolution, twice per revolution, or higher multiples (i.e. 1 x rpm, 2 x rpm, etc). These forces are resisted by the radial and thrust bearings.

The forces on the rotor from the fluid are more complicated. The torque from the rotor is resisted by the mass of the fluid, and generates not only resisting torque but also radial and axial forces. These dynamic forces are not sinusoidal at a single frequency, as is the unbalance force, but are random. The frequency band of this random "noise" extends from zero Hertz (cycles per second) to a frequency dependent on the velocity of the fluid. The amplitude of the force is largest at the low end and decreases at higher frequencies. The upper frequency of any significant amplitude ranges from 20 or 30 Hz for a feed pump at miniflow to 100 Hz for a Reactor Coolant Pump at higher than design flow. Because the velocity and acceleration vectors on the fluid align with the pump internals at the design flow, the amplitude of the radial flow forces is minimum at design flow. At miniflow or lower there is extreme misalignment of the fluid forces to the pump internals which causes the radial flow forces to be large.

As the impeller vane approaches a diffuser or cutwater in the pump, a pressure pulse is generated. The frequency of these pulses is called the vane passing frequency, 1 x VP, and is the least common denominator of the number of the vanes and diffuser vanes times the rpm.

The random flow induced forces and the vane passing forces are resisted by the bearings, but they are also transmitted to the pump case, the piping, and the fluid up and down stream of the pump. Any of these mechanical or fluid structures can amplify the forces through the mechanism of a resonance discussed below, and then reintroduce large forces onto the rotor or pump case.

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The bearings also have some rotation. The lubricant in a sleeve bearing rotates at approximately 0.42 x rpm. Under certain circumstances large motions can be generated at this frequency. Roller element bearings have rollers and cages which rotate at speeds different from the rotor. Faults in these elements can generate their own frequencies or can modulate the rotor's vibration including multiples of 1xrpm. The bearings, especially sleeve bearings are non–linear. This means that they act as frequency mixers and can generate new frequencies which are multiples or sum and differences of the frequencies of the applied forces.

As an example, consider a simple involute–type pump operating at 1800 rpm, with ball–bearings, 5 impeller vanes operated at mini–flow. The following frequencies of forces would be expected:

- 0–20 Hz random
- 0.4 x rpm (approximately) bearing cage pass
- 1 x rpm unbalance and misalignment
- 2 x rpm misalignment or bearings
- 3 x rpm misalignment or bearings
- 3–6 x rpm ball pass (not an even multiple)
- 5 x rpm vane passing.

These forces are transmitted from the rotor through the bearings to the supporting structure, or through the fluid to the case to the supporting structure. These forces are dynamic and affect the structures through which they pass. If relative motion occurs between two structures (shaft and bearing for example) then wear will occur. The forces will also cause flexing. Moderate flexing will cause fatigue, and large motions will cause overload failure. The motion, or vibration, of a piece of the structure is what we measure to estimate the forces, wear, and fatigue of the various structures in the machine. Because the forces originate on the rotor, the motion of the rotor is usually the largest motion in the machine. As the forces move through the bearings and other structures of the machine, the motions are attenuated, sometimes amplified, and distorted.

EQUATIONS OF MOTION OF A STRUCTURE

The equation of motion for a single degree of freedom system is:

F = kx + cv + m (1)

k	=	spring stiffness [pounds/inch]
m	=	mass [pounds–sec^2/inch]
c	=	damping [pound–sec/inch]
x	=	displacement [inches]
v	=	velocity [inches/second]
a	=	acceleration [inches/second^2]
F	=	exciting force [pounds] .

If the force is a sinusoid, then the equation can be solved for x as:

F = f*sin(wt) (2)

x = f*sin(wt–P)/sqr[(c*w)^2+(k–m*w^2)^2] (3)

where:

P	=	Atn(c*w/(k–m*w^2)) in radians
w	=	frequency in radians per second
t	=	time in seconds .

Note that the displacement is a sine wave whose amplitude and phase change with frequency. Secondly, notice that if w=0 then A=f/k which is a simple spring motion. Also, when w=0, the phase (time lag between the force and the displacement) is 0. If the frequency is very high then m*w^2 is much larger than k, and (m*w^2)^2 is much larger than (c*w)^2 so the displacement becomes only a function of the frequency and the mass. Figure 1 shows a plot of displacement versus frequency.

Figure 1 shows the effect of frequency and damping on the amplitude of vibration. The phase also changes, getting larger as a proportion of the period as the frequency increases. At large frequencies, the time lag approaches 0.5 of the period or 180 degrees. The frequency where the maximum vibration occurs is called a natural frequency, a resonance, or in rotating equipment a critical speed. The damping in typical machinery ranges from 0.3 for journal bearings to .005 for welded structures.

Because velocity is the derivative of displacement and acceleration is the derivative of velocity, the motion of the structure can be measured and described by any of the three terms. The relationships are:

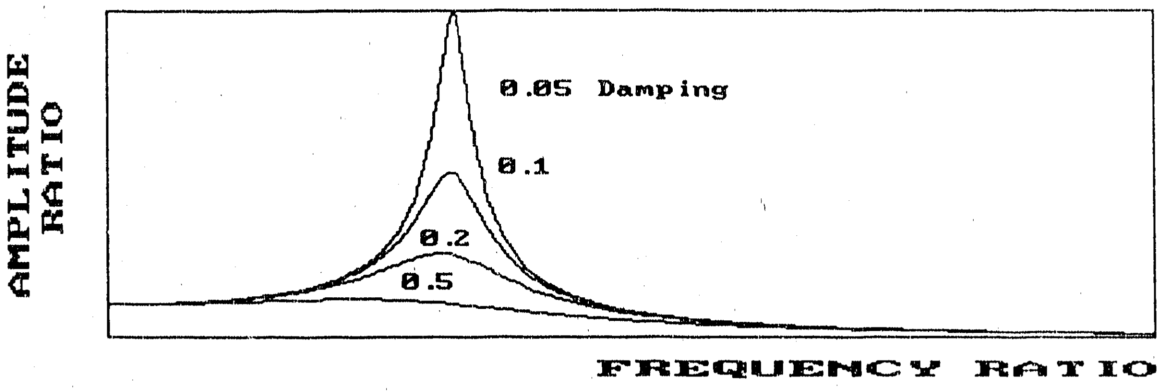


Figure 1.

$$d = A*\sin(wt+P) \quad (4)$$

$$v = A*w*\sin(wt+P+PI) \quad (5)$$

$$a = A*w^2*\sin(wt+P+2*PI) \quad (6)$$

Note that the velocity is shifted 90 degrees and multiplied by w, and that the acceleration is shifted by 180 degrees and multiplied by w^2.

The choice of the units to be used is dependent on the objective of the person making the measurement. In determining the condition of machinery there are four basic conditions we wish to detect: contact between rotating and stationary parts, overload, fatigue, and wear. Contact between the rotor and stator is obviously a function of the displacement of the rotor and the relative internal clearance. Overload is function of the force applied and, as seen above, is the sum of functions of displacement, velocity and acceleration. However, for very low frequencies the displacement term is very much larger than the other two. At very high frequencies the acceleration term is the highest, and the force as a percent of yield is much higher than the displacement as a percent of the internal clearances.

Between the "low" and "high" frequencies the displacement is small compared to internal clearances, and acceleration or inertial forces are small, so the main effects of the motion are wear and fatigue. Wear is related to energy which is proportional to velocity squared. In addition, fatigue (force times cycles) is proportional to velocity.

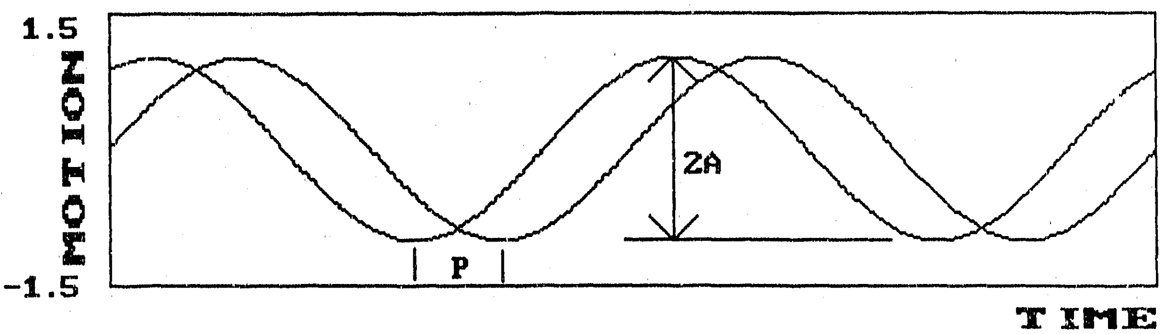
The definition of "very high" and "very low" is somewhat subjective, but machines which operate below 600 rpm are usually measured in displacement, and machines above 10,000 rpm are usually measured in acceleration. In the speed range between 600 and 10,000 rpm velocity is usually used to monitor for machine condition. So, for most power plant machines, and for most problems, velocity is the appropriate measure. There is one important exception to this rule. Sensors which directly monitor the shaft motion, such as shaft riders and proximity displacement sensors, are closer to the generated forces than bearing housing readings and suffer less distortion. They also measure a displacement which can be directly compared to the bearing clearance. These advantages outweighs the inconvenience of using a displacement measure and calculating alarms from the equivalent velocity.

SIGNAL ANALYSIS BASICS

The vibration transducer transforms the structure motion to a time varying signal. This signal is usually interpreted in the time domain, i.e. as a waveform. Waveforms that do not have any random components are analyzed as a combination of simple sine waves. Each sine wave has three parameters as seen in Figure 2: the frequency or period, the amplitude, and the time lag or phase.

A in the equation in Figure 2, is the amplitude zero to peak (0–p). This measure is usually used for acceleration and velocity signals. The double amplitude 2A, or peak to peak (p–p), is usually used for displacement signals. In Europe, and for signals with a random component, the Root–Mean–Squared measure is used. For a single sine wave the amplitude rms equals 0.707*A. Rarely is the average value of 0.5*A used. The relationship between the different measures given here are only for single sine waves. For the more usual combinations seen in vibration, the relationship becomes more complicated.

The term w in Figure 2, is the frequency in radians per second. It is usually converted into Hertz (Hz) or



x = A*sin(w*t)
y = A*sin(w*t+P)

Figure 2.

cycles per second, cycles per minute (cpm or rpm), or orders (cpm divided by the rpm of the machine).

The term P is the phase between the two waveforms. Phase angle is the only parameter that requires a reference, either another signal, or another frequency in a multi–frequency waveform. The phase is in radians in the equations, but is usually expressed in degrees. It is not a physical angle however, it is the time lag between two sine waves scaled as a proportion of the period.

For waveforms which are more complex, there are many other parameters which can be measured including crest factor, rise time, modulation period, etc.

The analysis of a simple sine wave can easily be done in the time domain. But in vibration analysis the waveforms are usually not single sine waves but combinations. The Fourier transform is used to analyze the signal in the frequency domain. A spectrum is a plot of amplitude versus frequency where the amplitude of each frequency in the signal is plotted as a bar graph. The phase of each frequency is also computed but is usually not displayed except when doing advanced analysis.

The spectrum shown in Figure 3 separates each frequency so that each can be analyzed separately. The rms value of the amplitudes is calculated by the analyzer. Often, but not always, this amplitude is converted to zero to peak or peak to peak. Each frequency is compared to the rotational speed of the machine and any other known sources (vane pass frequency, ball pass frequency, 60 Hz electrical noise, 120 Hz magnetic forces, etc.). Each amplitude is compared to overall amplitude limits and to the amplitude of the 1 x rpm peak. Peaks which are higher than 0.5 time the amplitude of the 1 x rpm peak and a significant fraction of the amplitude limit, could represent potential problems.

Let's review and consolidate what has been discussed so far. The spectrum in Figure 3 shows the forces of our example pump, with the problems listed earlier. These forces are transformed to motion by the structure which is represented in Figure 4 as a transfer function. When the force spectrum is multiplied by the transfer function the result is the spectrum of the vibration shown in Figure 5.

This concept of vibration as the product of a force spectrum and a structure's transfer function is basic to an understanding of vibration measurement and diagnostics. The vibration measured incorporates two constituents: the originating forces and the effect of the structure. Each has a profound effect on the resulting motion, but usually only the forces can cause damage. The vibration itself is usually not damaging to the machine but is just a faint echo of the forces which are causing damage deep inside the machine. There is, of course, an exception. When the frequency of a small non–damaging force is aligned with a resonance, destructive motions can occur.

MEASUREMENT SYSTEM BASICS

For hand–held vibration readings there are two types of sensors. The most common type is an accelerometer. To measure displacement, the signal from this sensor must be integrated twice. Since the noise is also integrated twice, low frequency noise can become quite large. This requires filters which limit the low frequency range of this sensor. The second type of sensor is a velocity transducer which utilizes a magnet and

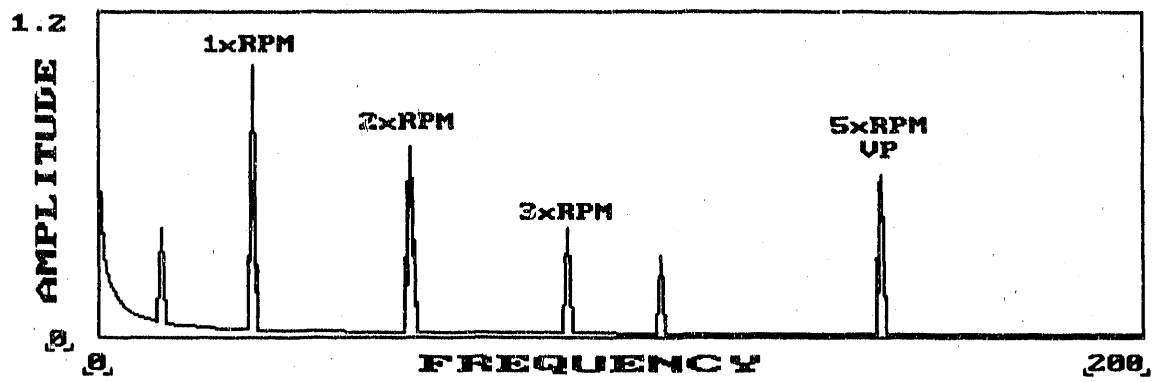


Figure 3.

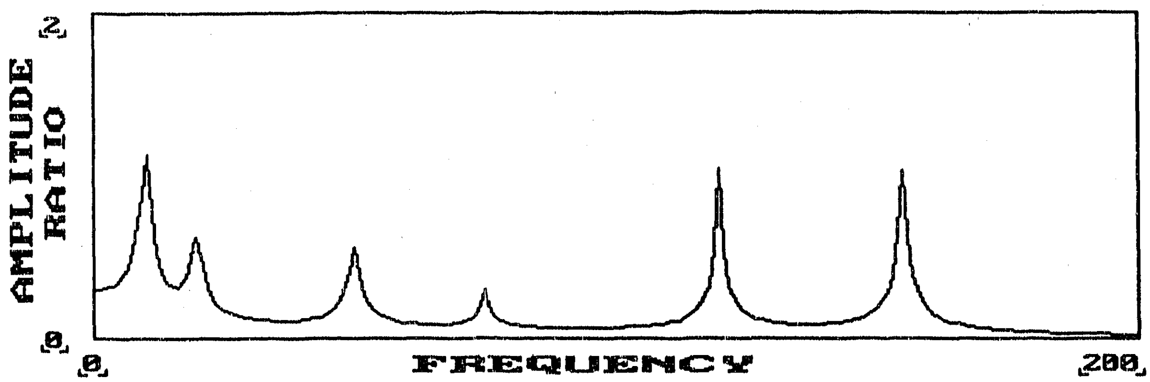


Figure 4.

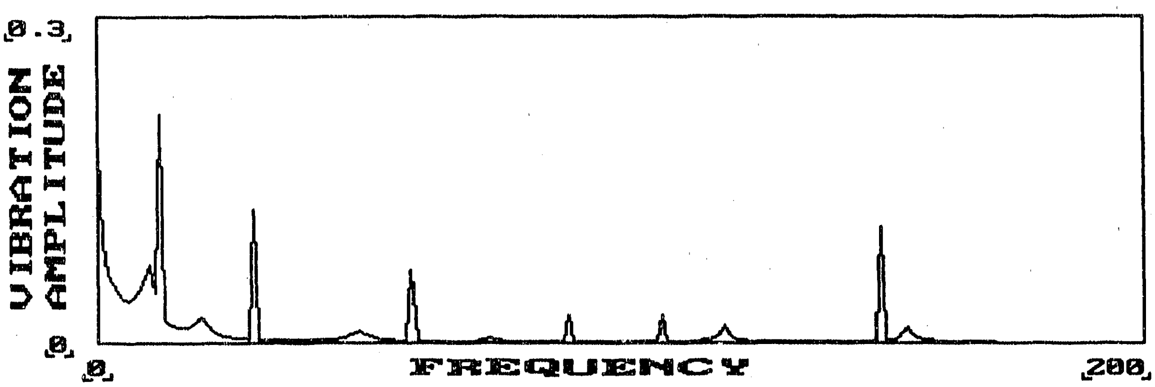


Figure 5.

a coil of wire. One of these is spring–mounted in a single degree of freedom system with a low resonant frequency. It is designed to be used well above its resonance which limits its low frequency response. It is also susceptible to magnetic fields.

The concept of a transfer function modifying a signal as it passes through applies to the system used to measure the vibration. The sensor which converts the motion to an electrical signal has a transfer function of its own. The instrument which receives the signal and displays the amplitude or other parameters also has a transfer function. These instruments also sense signals other than the machine vibration. Noise is any signal sensed that is not the vibration of the machine. Magnetic fields, temperature transients, motion induced into the sensor from outside the machine are all potential noise sources.

Take for example the traditional vibration measuring instrument shown in Figure 6. The probe (or "stinger"), the sensor, and the readout instrument can be represented by the block diagram shown in Figure 7. If the vibration spectrum from our example is applied to this set of transfer functions and additional inputs, the result is the spectrum shown in Figure 8. The measuring instrument will calculate and display the amplitude of the time domain representation of this spectrum.

There is a considerable chance of error even if the instrument is calibrated in accordance with the manufacturer,s specifications. A good vibration program attempts to reduce the chance of error by careful selection of instruments, proper training, and evaluation of readings. You may have thought that this was enough uncertainty, but there is one other consideration. The designs of the various instruments on the market provide different methods of measuring the amplitude of

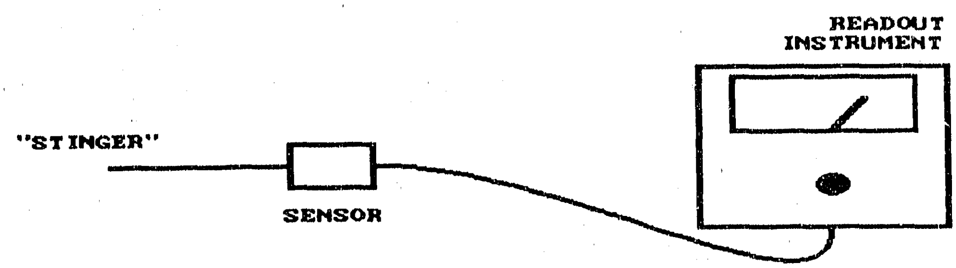


Figure 6.

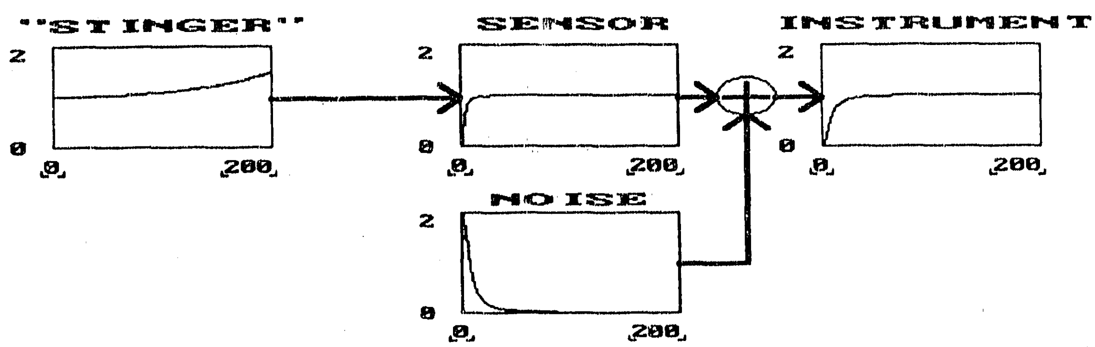


Figure 7.

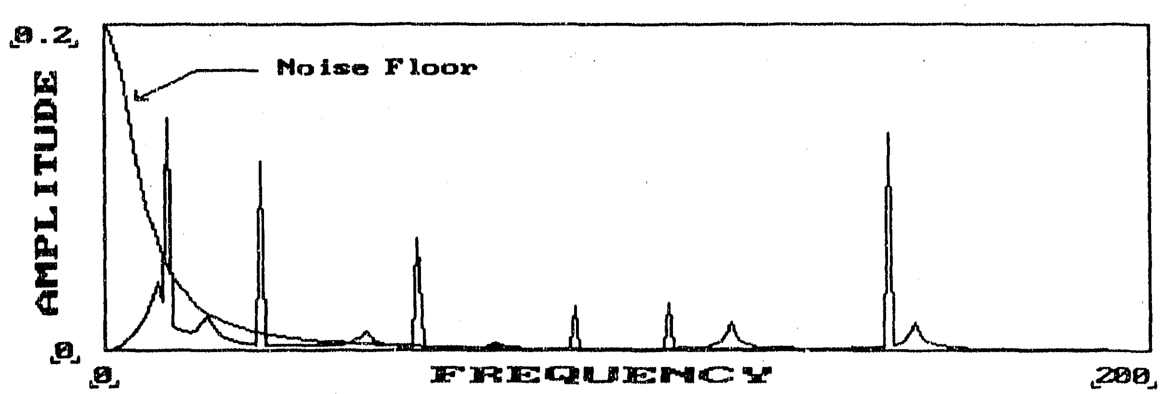


Figure 8.

the signal. As was discussed earlier there are four methods of calculating the amplitude of a waveform. All four methods are used in different manufacturer's instruments. They will usually be displayed in 0–p or p–p, but these are not true readings except for pure single sine waves. Two accurate instruments measuring the same vibration, with the same filters, one measuring rms and displaying 0–p, and the other measuring 0–p and displaying 0–p, may differ by 20%. In addition, all instruments use some form of averaging on the measured amplitude. Because the different manufacturers use different amounts and types of averaging, their instruments will read differently for varying signals.

GUIDELINES FOR VIBRATION MONITORING

In spite of the numerous potential problems discussed above, reliable estimates of machine condition can be obtained from vibration measurements. The following "rules–of–thumb" will provide some assurance that you are making an good estimate of your machine's condition.

1. Use displacement to determine vibration severity of machines which operate below 600 rpm.
2. Use velocity to determine vibration severity of machines which operate above 600 rpm and less than 10,000 rpm.
3. Vibration readings which contain frequencies below 10 Hz (600 cpm) will be inaccurate if taken with some common instruments. Check the specifications of the instrument you are using.
4. Vibration readings which contain frequencies below 2 HZ (120 cpm) are very difficult to measure accurately. Unless you have positive proof to the contrary, assume that the actual readings are more than 100% higher.
5. Centrifugal pumps operating at mini–flow almost always have vibrations at frequencies below 10 Hz and below 2 Hz. These frequencies will often cause large displacements hiding the true machine condition when measuring displacement. Velocity readings are less sensitive to these frequencies.
6. Two different instruments will almost always measure low frequencies differently, unless they are the same manufacturer and model.
7. Except for flow induced low frequency vibration, frequencies other than 1xrpm are more harmful than the 1xrpm vibration.
8. Do not use a "stinger" unless you know what you are doing.
9. Do not take hand–held displacement readings unless you know what you are doing, use a magnetic base, or a stud mount instead.
10. Do not use velocity pickups around aluminum frame motors without a magnetic shield.
11. If a vibration varies, the most reliable reading is the average.
12. Readings under 0.1 IPS O–p or under 0.1 mils p–p are unreliable unless stud or magnetically mounted. Amplitudes this low are usually not worth measuring accurately.
13. Use 0.3 IPS and 0.6 IPS (10 and 20 mils p–p) for screening limits. Above 0.3 IPS check the trends and frequencies. Above 0.4 or 0.5 IPS analyze the problem and make plans to repair. Depending on the problem and the trend, a machine may be able to operate for several months to a year above 0.6 IPS but it is usually economic to begin repairs. Stable unbalance, misalignment, flow induced, and resonance amplified vibrations indicate less damaging problems than bearing problems, rubs, or any quickly trending problem.
14. For vibrations at frequencies where instruments are inaccurate, or where the case is massive compared to the rotor such as explosion proof cases, etc., look for a doubling of amplitudes for a screening limit.
15. Measure both the velocity and the displacement. Calculate the velocity divided by the displacement times 19,000. If the result is not close to the rpm of the machine, then either the measurement is incorrect or there are frequencies present other than 1 x rpm.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

I'm having a little trouble figuring out how you can equate and use this specific frequency information that you were displaying and apply it to the OM–6 or IWP program. How do you sum all those pieces up and apply it to the simplistic specs that are in those programs?

Response 1

As I said when I started out, I'm not talking about any of the OM standards. This is just machinery practice.

I think that there's a problem with the current standards, in that they don't address the analysis or diagnostics of the spectrums. We never make any decisions on balance–of–plant equipment without looking at the spectrums to see if we can't figure out what's going on.

You run into instrument problems, human errors. It's a flow problem, it's not a vibration problem, and going with an overall amplitude is good for screening.

What we do in our balance–of–plant equipment is use 0.3 and 0.6 IPS as our screening limits, but we don't take any drastic decisions without looking at the spectrum.

Question 2

What are the best methods of attaching the stinger to the motor pump bearing? Some utilities use magnets to mount the accelerometer during the test. Other utilities use buttons.

Response 2

Well, the stinger rod is normally something like an eight–inch aluminum rod that screws into the sensor, and that is usually handheld. Stingers are very bad because of their resonant frequency.

There's no good answer to how to mount it. From a technical point of view, the best way is to stud mount it in accordance with an API 678 standard, but that is, except for permanent accelerometers, hard to do and it's hard to keep your surfaces clean. For general machinery monitoring there is an acceptable compromise, and that is to use handheld probes without the stinger. In other words, you put the accelerometer either with a very small stinger at less than half an inch or directly with the probe on the bearing housing.

For slow speed readings, where you are going to be measuring in mils, you can get significant low frequency errors because you can't hold your hand exactly still and you are taking very small readings. So, we use magnets, but magnets are tricky. You have to be real careful with them, make sure they don't rock. They are not usable above a certain vibration amplitude because they'll shake themselves off. So you may even need to hand hold the magnet on there if your amplitudes are a little higher.

Question 3

OM–6 states that the vibration spectrum shall be from one–third minimum pump speed to 1,000 Hz. Has the committee considered reducing the range to suit operating conditions at the discretion of the licensee?

Response 3

I guess, at the discretion of the licensee, no. That has not been considered yet.

Question 4

With the Inteletron system we use, we are able to use a spectrum range of anything we select, from zero to 400, zero to 1,000. The resolution of the equipment increases if we reduce that spectrum.

Because of our equipment limitations, is it acceptable to go ahead and use the zero to 400, or is there an 800 spectrum?

Response 4

From a technical point of view, I like taking an unfiltered overall value, and that would be up to 1,000 Hz. Some instruments provide that.

Some instruments are limited by the same frequency range you have for your spectrum. That's what it looks like for the overall instrument use. So, we need to get the benefit of both of those. It turns out to be a high frequency that's outside our normal analysis range.

We go for the overall. And that happens enough times that I think it's worthwhile taking an overall from a low frequency, up to about 1,000 Hz.

LOW–FLOW OPERATION AND TESTING OF PUMPS IN NUCLEAR PLANTS[a]

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ABSTRACT

Low–flow operation of centrifugal pumps introduces hydraulic instability and other factors that can cause damage to these machines. The resulting degradation has been studied and recorded for pumps in electric power plants. The objectives of this paper are to (1) describe the damage–producing phenomena, including their sources and consequences; (2) relate these observations to expectations for damage caused by low flow operation of pumps in nuclear power plants; and (3) assess the utility of low–flow testing.

Hydraulic behavior during low–flow operation is reviewed for a typical centrifugal pump stage, and the damage–producing mechanisms are described. Pump monitoring practices, in conjunction with pump performance characteristics, are considered; experience data are reviewed; and the effectiveness of low–flow surveillance monitoring is examined. Degradation caused by low–flow operation is shown to be an important factor, and low–flow surveillance testing is shown to be inadequate.

INTRODUCTION

This paper is an outgrowth of studies on auxiliary feedwater pumps that were conducted under the Nuclear Plant Aging Research Program sponsored by the U.S. Nuclear Regulatory Commission. These studies led to a review of low–flow operation and testing of centrifugal pumps and their consequences; the results are the basis for the discussions herein.

A centrifugal pump is designed for best performance at a specific combination of capacity, head, and speed, that is, the best efficiency point (BEP). At the design or BEP flow rate, the fluid motion is compatible with the physical contours of the hydraulic passages and is therefore well–behaved. However, careful consideration must be given to the operating range for a given application to minimize undesirable effects of off–design flows.

During operation and testing, flow rates can be on the order of 5 to 15% of BEP flow. These flow rates stem from system bypass lines being sized to limit the temperature rise of the pump without regard to hydraulic behavior effects. Deterioration resulting from low–flow hydraulic instability influences was not considered until relatively recently.

For reduced flow, the larger the percentage of BEP flow, the better it is for the pump. In well–engineered pumps, 25 to 35% is sufficient to avoid the dangerous range of off–design operating flow. However, for some pumps, any flow below 50% of BEP capacity may cause severe vibration.

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b. Oak Ridge National Laboratory, Oak Ridge, TN 37831–8063. Operated by Martin Marietta Energy Systems, Inc. for the U.S. Department of Energy.

In this paper, the effects of low–flow operation on hydraulic behavior are examined; damage–producing mechanisms and resulting pump degradation are discussed. Centrifugal pump operating experience is summarized, and important aspects of surveillance testing for determining machine health are reviewed.

Pump performance characteristics and the types of degradation incurred must be recognized when designing an effective surveillance program. In addition, hydraulic instability can complicate the data interpretation process. These factors preclude significant reliance on low–flow surveillance testing for operational readiness determinations.

LOW–FLOW DEGRADATION

Hydraulic instability is a term used to describe unsteady flow phenomena that become progressively more pronounced as a pump is operated farther away from the best efficiency point (BEP). The hydraulic instability is manifested by flow recirculation in both the suction and discharge regions of an impeller stage when operating below the design flow.[1–5] This recirculation, or hydraulic stall, is the result of disorganization of the internal flow field that occurs at the impeller eye and exit as well as outside the impeller shroud and hub. (See Figure 1 for nomenclature.[1]) These disorganized flows can be significant contributors to deterioration (i.e., aging and service wear) of pump components because of resultant cavitation, pressure pulsations, unbalanced forces, and vibration. The intensities of any pressure fluctuations accompanying such flows increase with pressure rise or energy level of the pump.

Hydraulic instability associated with low–flow operation can result in cavitation erosion; unstable head–flow characteristics; breakage of impellers, shafts, and cutwaters or diffuser vanes; failures of seals, thrust bearings, and axial thrust balancing devices; and vibration failures. Vibration failures can also result from such things as interactions between pump and driver, foundation, or piping; design deficiencies; or control valve deficiencies. In addition, vibration can be a symptom of bearing failure, internal rubbing, or incipient seal failure.

Cavitation can be caused by entrained gases or flow recirculation in various regions of the pump while operating at off–design flow rates. It can also result from insufficient net positive suction head (NPSH) or combined insufficient NPSH and hydraulic instability. Cavitation, regardless of the source, can cause serious damage to the pump impeller, diffuser or volute, and return vanes.

A properly designed pump stage should have a head–flow curve that is continuously rising as the flow is decreased below the BEP value during constant speed operation. This increase should be continuous with decreasing flow down to recirculation flow, as a minimum. The pump minimum flow should be higher than the onset of system instability however. When the head curve contains a droop or exhibits a flat midportion, the head–flow curve is unstable, which is symptomatic of hydraulic instability. Parallel pump operation in the unstable region of the head–flow curve can be difficult.

Impeller breakage is a frequent result of problems such as vibration and hydraulic instability; fluctuating axial forces on impellers contribute to seal, bearing, and axial–thrust balancing device failure. Hydraulic instability, such as rotating stall and cavitation, and high induced vibrations at high frequencies are especially destructive.[6] However, design deficiencies can also be the cause of failure.

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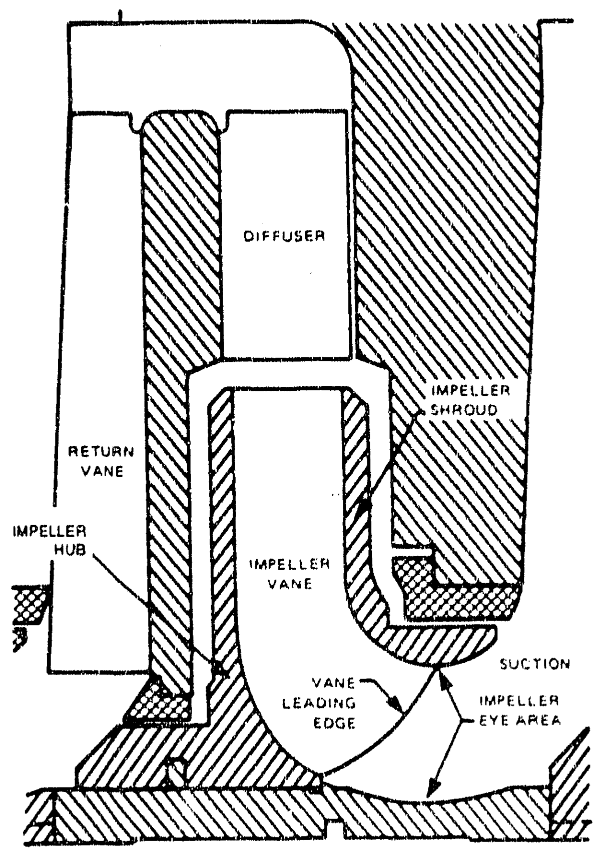

Figure 1. Pump stage terminology (diffuser–type discharge chamber).

To give added perspective on hydraulic instability and its influences, a brief examination of flow behaviors in a pump stage is instructive. A single stage is shown in Figure 1.

Several factors are associated with low–flow conditions that cause detrimental effects. At these conditions, all impellers develop flow instability in the form of flow reversals in the inlet and discharge regions; these reversals are called suction and discharge recirculation.[7] The attendant recirculation cells are depicted in the upper part of Figure 2. Part of the liquid flows out of the outer portion of the inlet eye with high rotational velocity and reverses direction to join the main flow into the eye, giving rise to vortex formation. The resultant vortex action induces pressure surges and pulsations that cause rapid deterioration by cavitation erosion of impeller metal in the entrance region.[6]

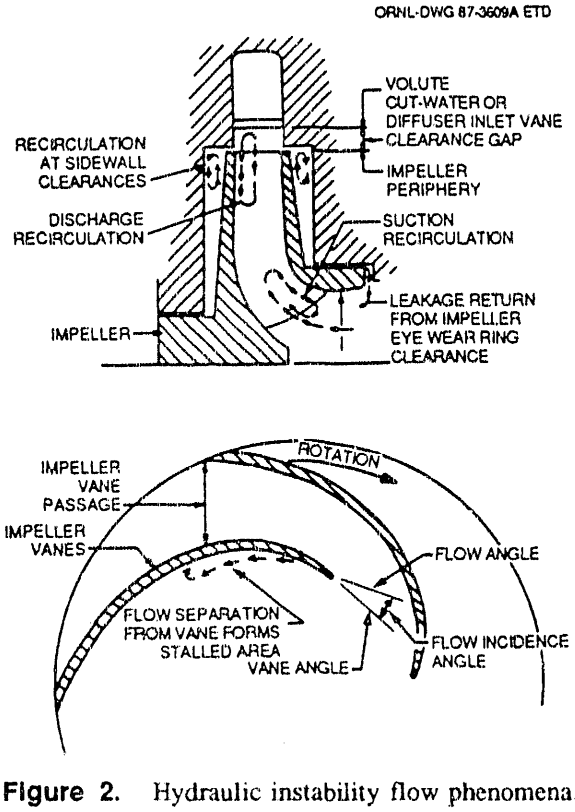

Figure 2. Hydraulic instability flow phenomena. Redrawn with permission after Reference 7.

Likewise, discharge recirculation creates surges and local deterioration by cavitation erosion at the impeller tips. Recirculation in the suction and discharge regions does not necessarily occur at the same flow rate.

Because pumps are designed for BEP flow, off–design conditions, such as low flow, cause a mismatch between the fluid flow angle and the physical angles built into the pump impeller and discharge diffuser or volute vanes. Low–frequency pressure pulsations therefore arise because of internal recirculation within the impeller as well as inside the diffuser or volute. This recirculation stems from flow separation from the metal surfaces and produces stalled areas that eventually wash out and again reform; the phenomena is termed "rotating stall" because of low frequency cyclic rotation from passage to passage. The lower portion of Figure 2 illustrates flow behavior inside an impeller.

The geometric relationship between the rotating impeller and the stationary components of the case influences: (1) the attenuation of discharge recirculation effects; (2) reduction of secondary flow circulation in the space between the impeller sidewalls and the casing; (3) the intensity of the vane–passing pulsations at the diffuser inlets or volute cutwaters; and (4) disturbance at the impeller inlet due to leakage return from the impeller eye wear ring clearance.[7] The gap between the impeller periphery and the diffuser vanes or volute cutwater is therefore very important.[5,8,9,10] When the clearance gap is too small, vortices caused by discharge recirculation will tend to collapse near the metal surfaces, causing cavitation.

The pressures acting on the impeller hub and shroud fluctuate; these fluctuations, in turn, give rise to fluctuating net axial forces on the impeller. The impeller will therefore suffer axial position instability, moving back and forth within limits imposed by the thrust bearing and the structure. These fluctuating forces contribute to failure, as noted previously.

Secondary flow circulation in the space between the impeller sidewalls and casing can be controlled by adjusting the clearance gap between the periphery of the impeller and the casing.[5,9] This reduces the unbalanced axial forces.

Cavitation surge can be produced by hydraulic instability, or low NPSH in combination with low–flow recirculation. The latter can be eliminated with sufficiently large NPSH. Cavitation surge also results in damaging pressure pulsations.

Because of increased recognition of the propensity for degradation as described above, operation at off–design conditions with emphasis on low–flow aspects is being given increased attention. Pump manufacturers have developed guidelines for establishing minimum flow limits on pump operation,[3,7] and studies are continuing. Electric Power Research Institute (EPRI) is currently conducting an extensive program on reliability and performance of multistage centrifugal pumps;[6] this program is expected to provide important information on low–flow operation. A study on the influence of surveillance testing at low flow on failure of emergency pumps in nuclear power plants was recently completed by EPRI.[11]

It was noted earlier that intensities of pressure fluctuations accompanying disorganized flows increase with pressure rise or energy level of the pump. Other factors that influence low–flow pump performance and minimum continuous stable flow (MCSF) are specific speed N_s and suction specific speed N_{ss}. The most critical factors are power intensity and N_{ss}.[3] The power intensity is a function of the brake horsepower (bhp) per stage; a power intensity factor F_{pi} can be defined by

$$Fpi = (bhp/stage)/(impeller\ diameter)^3.$$

Small pumps having low Fpi are least likely to suffer damage from hydraulic instability.

N_s is a reference number that describes the hydraulic features of a pump, whether of the radial, semiaxial, or propeller type. The N_s is related to the pump speed N (rpm). the full–capacity flow Q (gpm), and the head per stage H (ft), with each being the BEP value; thus, $N_s = N(Q^{0.5})/H^{0.75}$. N_s affects flow separation from vanes and backflow in the impeller. The backflow driving force increases with increased N_s.

N_{ss} is used to categorize impeller suction design and performance characteristics and is related to the net positive suction head required (NPSHR) rather than H as in the case of N_s. Thus, N_{ss} is given by the following expression:

$$N_{ss} = N(Q_e)^{0.5}/(NPSHR)^{0.75}.$$

The capacity Q_e is per impeller eye, and the quantities in the equation are BEP values. As N_{ss} increases, the impeller tends to become more susceptible to inlet vane flow separation.[3]

MONITORING

Pump characteristics. N_s not only influences low–flow pump performance, it is important in characterizing pumps;[12] Figure 1 demonstrates this fact. Shown are sets of characteristic curves, that is, head H, efficiency E, and power input P, vs flow rate; efficiency (with flow rate as a parameter); and impeller shape (radial to axial flow) as functions of N_s. Reference 13 indicates that pumps used in engineered safety feature systems of pressurized water reactors (PWRs) have an N_s range from 741 to 3080; an examination of high– and low–pressure safety injection, containment spray, and auxiliary feedwater (AFW) pump data for a single PWR plant gave a N_s range from 1000 to 1850. The range of estimated maximum N_s values for pumps in engineered safety feature and safety–related systems of boiling water reactors (BWRs)[14] range from 500 to 4000. Cooper et al[9] gave $N_s = 1503$ as a typical value for feed pumps discussed in Reference 15. Hence, for many centrifugal pumps in nuclear plants, it can be assumed that the characteristic curves will approximate those shown on the left and in the center of the upper portion of Figure 3.

Characteristic curves for an electric–motor–driven AFW pump are given in Figure 4. The value of N_s is 1174. Note that the characteristic curves tend to correspond to those shown on the left side of Figure 3, with the head–flow curve being stable.

Pump degradation results in head–flow curves such as those shown in Figure 4 being altered as shown schematically in Figure 5(a). The degraded performance curve is depicted by the solid line that is based on the assumption of constant leakage between stages.[12] This assumption also yields the comparison of degraded vs nondegraded performance for high N_s pumps shown in Figure 5(b). Compare the dashed curve of Figure 5(b) with the H curve in the upper right of Figure 3. The latter curve applies to predominantly mixed to axial flow pumps; the indications are that safety–related pumps with high N_s are fewer in number than those having intermediate N_s.

Monitoring Details

The results emphasize the importance of factoring details of centrifugal pump performance into the design of programs for detecting, tracking, and assessing aging and service–wear degradation. Measurements of pressure and flow at flow rates approaching shutoff do not allow meaningful assessment of degradation for pumps with performance curves like those shown in Figure 5(a). However, it is important that flow as well as pressure measurement be taken at low flows because pressure and flow are essentially independent in such cases. This means that a zero flow condition could otherwise go undetected.

Operation at flow rates above MCSF is required to provide margin for avoiding failure. Reliable operation below this level is not possible.[9] Determinations of the highest flow rates at which off–design recirculating flows occur have been derived from test data.[3] These curves provide relationships between N_{ss} and percent BEP flow for MCSF. Considering the pump

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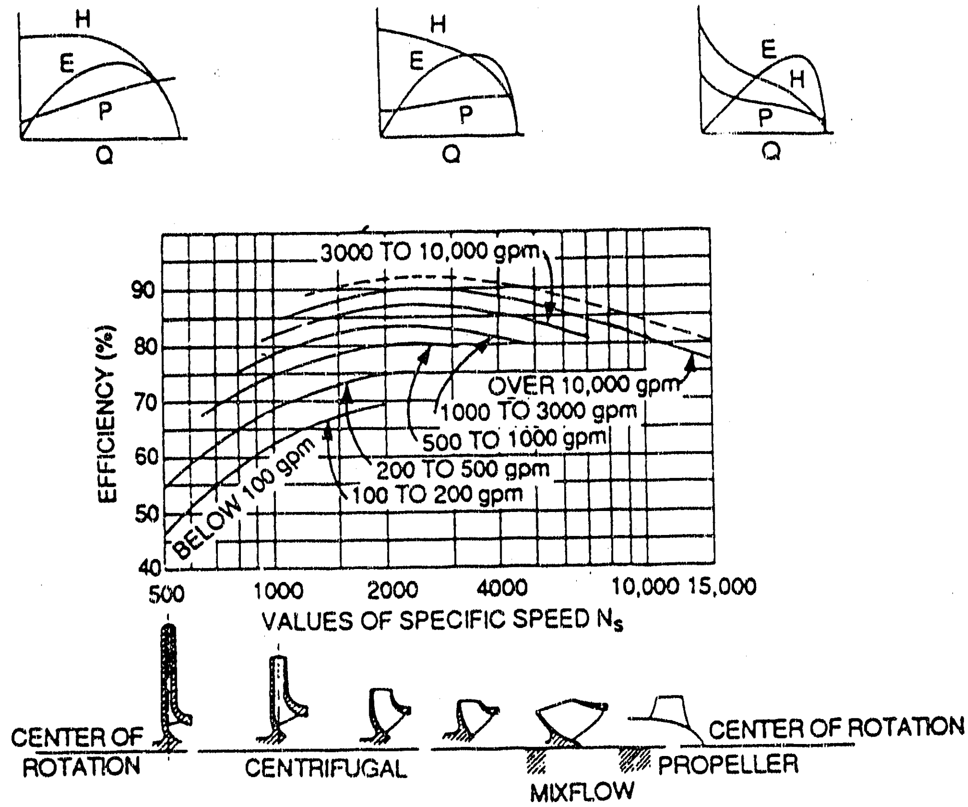

Figure 3. Approximate relative impeller shapes and performance characteristic variations with specific speed. Reprinted with permission from McGraw-Hill, Reference 12.

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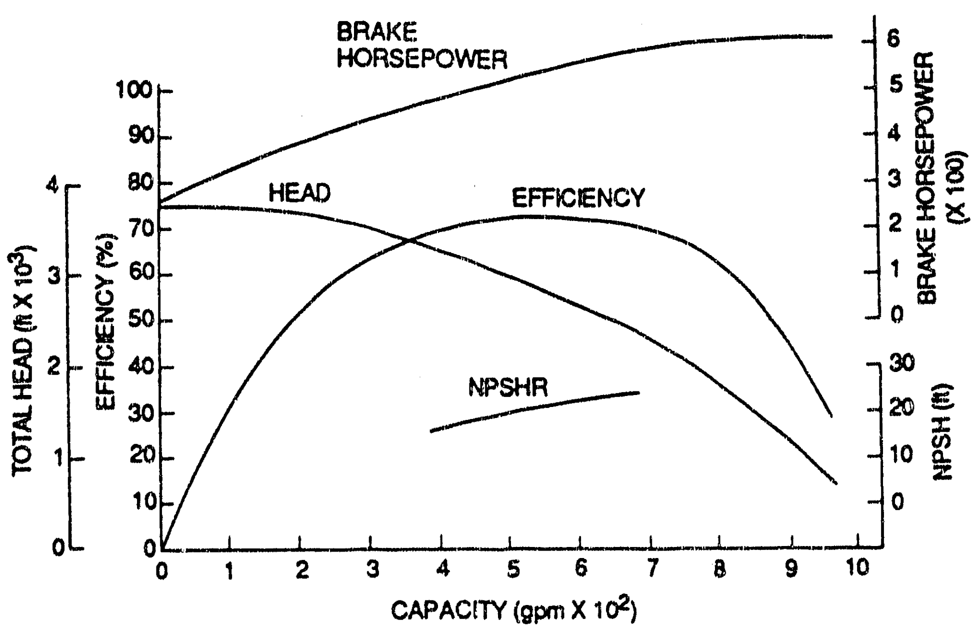

Figure 4. Characteristic curves for AFW pump.

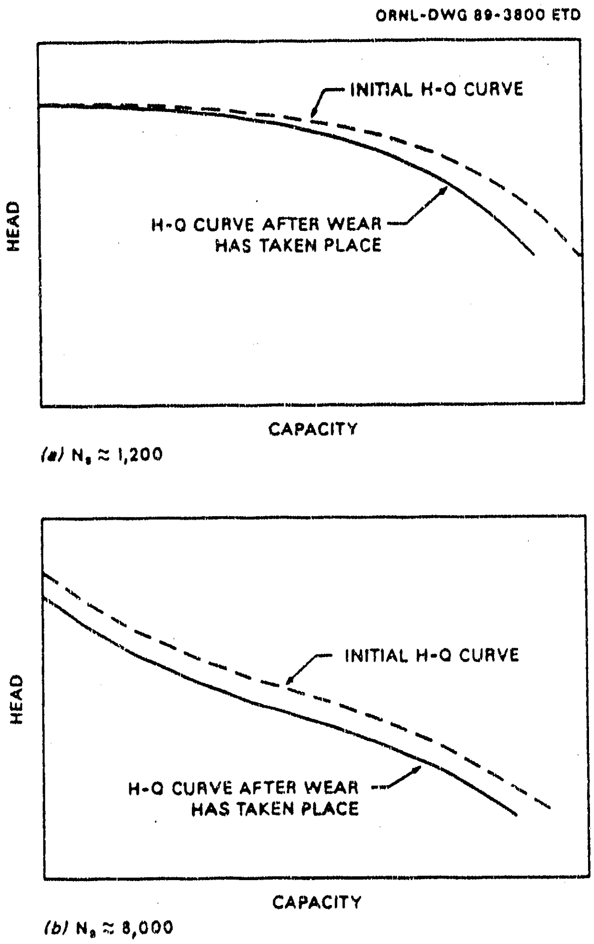

Figure 5. Effect of wear on head–capacity curves.

examples for one PWR plant cited previously, the MCSF is >30% in all cases. Generally, low–flow operation in nuclear plants is at rates much lower than 30%; flow rates $\leq$10% are often used.

Aging and service wear degradation can be difficult to detect and monitor on the basis of measurements taken during surveillance testing. In most plants, the bypass flow test provides neither the proper operating range of flow nor sufficient running time to comprehensively trend and assess vital signs.

Results of an extensive study on understanding and managing aging and service wear in AFW pumps are given in References 2 and 16. Inspection, surveillance, and monitoring methods for detecting and tracking aging and service wear are discussed in detail, and recommendations are made. The recommendations, which are also applicable to other centrifugal pumps, include periodic disassembly and inspection as an essential element.

POWER PLANT OPERATING EXPERIENCE

Through EPRI–sponsored projects, a significant amount of information on centrifugal pump performance has been published. This helps to compensate for lack of information on pumps used in nuclear power plants. The most significant summary of data, from the standpoint of this paper, is given in Reference 15, which gives results from a survey of feed pump outages.

This EPRI survey[15] covered centrifugal pumps used in both fossil and nuclear–fueled electric power plants. The survey of outages is based on input from 96 utilities covering 240 generating plants; a total of 1327 pumps was covered, with 1204 being feed pumps. The feed pump can be categorized as horizontal or vertical, single or multistage, and low or high head. Pump stages were either diffuser or volute type; the casings were cast or forged; and the pump drivers were electric motors and turbines. Pump types included in the survey are boiler feed, nuclear feed, and feedwater (FW) booster pumps (see Table 1).

The pump types investigated were reduced to five basic frame sizes based on reference pump flow produced at a reference speed of 3570 rpm. To achieve this reduction, the following relationships were used. The reference flow is given by $Q_R = 3570\ (Q/N)$, where Q = design flow and N = design speed. The reference head H_R is given in terms of the design head H by $H_R = H\ (3570/N)^2$. The reference power absorbed per stage (HP_R) is related to the design power by $HP_R = HP\ (3570/N)^3$.

The frame sizes were designated A through E (see Table 2). The A–frame–size boiler feed pumps are also used as AFW pumps for PWR service and as special purpose nuclear pumps. The survey was not extended to that portion of the A–frame pumps, however. Table 3 summarizes the data from the survey.

Data from Reference 15 were used along with estimates of average number of hours per outage to develop Table 4, which gives relative importance rankings of failures.[6] Cavitation, unstable head curve, impeller breakage, and vibration combine to provide the leading cause or symptom of failure. These elements generally result from hydraulic instability associated with operation at less than BEP flow rates. Hence, this table clearly illustrates the importance of detrimental effects resulting from low–flow operation. Further, since the pump population embraced by the survey includes a

Table 1. Pumps covered in survey

Type	Description
Boiler feed pumps	Horizontal, multistage, forged outer barrel, welded–on suction and discharge nozzles. Impellers can be in line or opposed (largest percentage in line). Diffusers and volutes equally favored for discharge chambers.
Nuclear feed pumps	Called reactor feed pumps in BWRs and steam generator feed pumps in PWRs. Usually single stage, double–suction design. Some multistage units, forged casing, diffuser or double volute discharge chamber.
Feedwater booster pumps	Almost all are horizontal, single–stage, double–suction, cast casing, single– or double–volute outer casing.

Table 2. Pump frame sizes[a]

Frame Size	Reference PumpFlow (gpm)
A	Up to – 2,200
B	2,000 – 4,400
C	4,000 – 9,000
D	8,000 – 16,000
E	15.000– up

a. Source: Taken from Reference 15.

Table 3. Failure rates of utility pumps

Pump Type No.	Pump Type	Failures Reported From: Total Stations	Failures Reported From: Total Generating Units	Number of Failed Pumps	Number of Failures	Total Pumps Surveyed	Number of Failures/ Failed Pumps	Failure Rate (%)
1	Boiler feed	150	203	362	763	1044	2.1	34.7
2	Nuclear feed	20	30	61	133	160	2.2	38.1
3	FW booster	28	40	123	155	123	1.3	100.0
	Subtotal	178	240	546	1051	1327		

Source: Taken from Reference 15.

Table 4. Pump failure rankings

Pump Failures: Components or Symptoms	Feedpump Outages	Average Hours Outage[a]	of Grand Hours	Percent Relative Total	Ranking
Cavitation, unstable head curve, impeller breakage, vibration	650	48	31,200	45	1
Seals	602	32	19,264	27.8	2
Wear rings	155	48	7,440	10.7	3
Axial balancing device	337	16	5,392	7.8	4
Shaft broken/damaged	77	48	3,696	5.3	5
Journal bearing	209	8	1,672	2.4	6
Thrust bearing	58	8	464	0.57	7
			69,128		

a. Estimated average hours per outage.

Source: Taken from Reference 6 and based on data from Reference 15.

significant representation of nuclear type pumps, the low-flow failure implications for these pumps are inescapable.

The EPRI study[11] on surveillance testing of standby pumps in operating nuclear power plants was to determine whether test-related failures were caused by some aspect of these tests and to identify corrective measures to minimize the occurrence of such failures. This study was instigated in response to concern by pump manufacturers that testing pumps at low flow — on the order of 10% of BEP flow — may lead to premature failure of packing, seals, and rotating element components as a result of higher vibration during low-flow testing. Instances of pump vibration during low-flow testing and of vibration-induced damage to pumps and valves have been reported at both BWR and PWR plants.

Both PWR AFW pumps and BWR residual heat removal service water (RHRSW) pumps were addressed. However, this study neither provides conclusive evidence against nor vindicates the use of low-flow testing practices. It does support, however, the expectation that low-flow test operation will lead to degradation and failure and concludes that prolonged operation of AFW pumps at very low flow (in the range of 10% BEP flow) can cause high vibration, which can manifest itself in bearing and wear-related failures.

Reports on failures of centrifugal pumps in nuclear power plants include classic examples of deterioration induced by low-flow hydraulic instability. In May 1986, Susquehanna Unit 1 experienced loss of the emergency service water (ESW) system as a result of low-flow cavitation damage to the pumps (LER 86-021-00, May 20, 1986). The suction bell of one pump was damaged so severely that it was severed from the body. The impeller vanes of this pump also were eroded through the thickness.[17] Similar, but less extensive, damage was found in the other three ESW pumps. A later inspection of the RHRSW pumps also revealed similar cavitation damage in each case.

Inspection of the residual heat removal (RHR) pumps in the Vermont Yankee Plant was prompted by NRC Information Notice 86-39.[18] Through-the-wall impeller cracks were found in two pumps, and evidence of low flow induced cavitation erosion was found in the impeller suction regions of all four pumps.[17]

A locked rotor condition occurred in an electric-motor-driven fire pump at the Haddam Neck Plant (LER-88-003-00, March 3, 1988). This condition resulted from damage to a brass bushing in a stuffing box. The damage was attributed to prolonged low-flow operation of the pump.

Low-flow degradation generally develops slowly, and, in the early stages, does not affect pump performance appreciably. Therefore, the damage produced is not easily detectable. Damage such as erosion, fatigue cracking, and nondisabling breakage in pump stages cannot be observed without disassembly of the pump. Surveillance examination of pumps, as usually practiced, thus leads to high probabilities that this type of degradation will go undetected until pump or system failure occurs and ability to perform the required safety function is lost.

CONCLUSIONS

Hydraulic instability from low–flow operation gives rise to degradation mechanisms that, if allowed to persist, will cause damage to the pump internals and eventually lead to failure. The intensities of the degradation mechanisms are functions of the energy per stage and the specific speed. Manifestations of hydraulic–instability–induced effects are cavitation erosion; unstable head–flow characteristics; breakage of impellers, shafts, and cutwaters or diffuser vanes; failures of seals, thrust bearings, and axial–thrust balancing devices; and vibration failures.

The deterioration that occurs develops slowly, and, in the early stages, changes in pump performance are not perceptible. Head and flow data are not reliable indicators of health because head vs capacity curves, in many instances, exhibit very little change at low flows (especially as shut–off flow is approached) because of degradation. Also, damage, such as erosion, cracking, and nondisabling breakage in pump stages, cannot be observed without pump disassembly and inspection. Finally, hydraulic instability can complicate the data interpretation process. These factors combine to preclude placing reliance on low–flow testing for operational readiness determinations and establish periodic disassembly and inspection as a necessary element in surveillance testing programs.

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ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

You had one graph or chart up there that showed for type A, B, and C different pumps. What's the source of that, and can we use that source as being credible? Is that a published document?

Response 1

That is in a published document. It's referenced in the paper. I don't have the number, but it was in a paper written by C. C. Herald and R. Paul Grave of Ingersol Rand. You may wonder how the curves were determined.

The minimum continuous stable flow determined in that diagram is the lowest flow at which a pump can operate without exceeding the noise and vibration limits imposed by API specification 61, sixth edition. It's a very credible source.

Question 2

On multiple stage pumps, is there one stage that is more susceptible to difficulty, damage, or problems than another stage due to operation at low flow, like the first stage or the last stage?

Response 2

No, there has not been a differentiation made between stages in a single pump.

CORRECTIVE MEASURES FOR UTILITY PUMP LOW FLOW HYDRAULIC INSTABILITY[a]

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ABSTRACT

Diagnostics and corrective actions are presented for various types of damages and failure modes mainly for nuclear safety related pumps. These pump types are:

- Nuclear Aux. and Nuclear Feedwater Pumps (Multi–stage, Horizontal)
- HPSI and Charging Pumps (Multi–stage, Horizontal)
- Reactor Feed (BWR), Steam Generator Feed (PWR), LPSI and Booster Pumps (Single Stage, Double Suction, Horizontal)
- RHR and Core Spray Pumps (Single Stage, Single and Double Suction, Vertical)
- Primary Coolant Pumps (PCP) and Circ. Pumps (Single Stage, Single Suction, Vertical)

Solutions to "Hydraulically" induced unstable forces are discussed in the flow operating mode. Some examples are given in the run–out mode (LOCA).

Damages discussed stem from generic engineering design (Designer under–estimated the Hydraulic force magnitudes), manufacturing errors, incorrect over-hauling, assembly and operating procedures. Cases presented are narrowed down to the following symptoms, components, or technologies:

- Shaft breakage at typical locations due to Hydraulic forces, design and manufacturing errors.
- Thrust bearing, Balancing device, Coupling and Structural failure due to axial vibration (Shuttling) of the rotor.
- Journal bearing damage due to large Hydraulic forces and incorrect bearing geometry.
- Impeller and Diffuser/Volute damages due to large Hydraulic forces and/or incorrect impeller to diffuser/volute geometry.
- Impeller eye (Inlet) caviation damage due to incorrect impeller inlet geometry or flow condition.
- Incorrect materials, hardness and heat–treatment, etc.

The cases discussed are shown before or during failures and are supported by measurements and performance evaluation after corrective actions. Actual design modifications are shown.

List of References are given for detailed evaluation of the corresponding technology background for the cases discussed.

POWER PLANT PUMP TROUBLE–SHOOTING STATE OF THE ART OVERVIEW

INTRODUCTION

The sizes of electric generating stations grew rapidly in the late 60's. Large nuclear plants were under construction, getting closer to start–up. The sizes of units being designed grew almost daily without the benefit of operating experience in the smaller sizes. The demand for very large size feed pumps (without

a. Presented at the ASME/NRC Symposium on "Inservice Testing of Pumps and Valves" on August 1–3, 1989 in Washington, D. C.

proven performance behavior) was the greatest ever. So was the competition among pump manufacturers. Low speed, multistage feed pumps were first offered with more or less known performance. Large, high energy input nuclear feed pumps were quickly designed by pump vendors to replace these low speed, multistage designs. This was not an engineering decision, it was a marketing move to remain competitive in the new lucrative nuclear business. Low speed (1200 to 1800 rpm) booster pump designs were quickly scaled up to over 5000 rpm speeds without proof of reliable operation. A new monster was created: "High Energy Input" single stage, double suction pumps for nuclear application. The damage was done and the utilities which owned these pumps were stuck and had to make them operate. A new era started for high head-per-stage designs which carried over into boiler feed pump designs, as well as, large injection pumps for the oil fields. The first large nuclear plant owners were committed to an accelerated pump research program by necessity. Ten large nuclear complexes are listed in Table 1 as an example for those who contributed heavily to the state of the art of the present high energy input feed pump designs through the numerous failures of the early designs.

Table 1. Nuclear power stations that initiated improving pump technology due to major failures in new pump designs. Also improved feed pumps for fossil-fuel applications

Number	Power Company	Nuclear Station	Year	MW[a]
1	NSP	Monticello	1970	540.
2	CECO	Dresden 2,3	1971	2 × 800
3	NUSCO	Millstone 1	1972	650
4	CECO	Quad Cities 1,2	1972	2 × 800
5	Y.A.	Maine Yankee	1972	900
6	NSP	Prairie Island 1,2	1973	2 × 650
7	P.S. WISC.	Kewaunee	1973	650
8	CECO	Zion 1,2	1973	2 × 1100
9	ROCH G&E	Ginna	1974	500
10	DECO	Enrico Fermi 2	1974	1300
11	PECO	Limerick 1,2	1986	2 × 1150[b]

a. Values modified by author.

b. Described in References 1 and 11 at the end of this paper.

Summary of where the problems could originate.

- Basic Hydraulic Research (academia, research institutes, OEM's research facilities, etc.)
- Basic engineering errors
- Manufacturing errors
- Assembly errors at the OEM
- Installation errors of the rotor at the plant
- Operational error.

In case of problems:

- Field service: lack of feed back to
- Design engineering of the OEM.

This is all one big loop. Any link of the chain broken, trouble starts.

Typical vibration frequencies in high speed, high energy input boiler feed pumps (BFP)

Figure 1 is from Reference 1.

1. Rotational (synchronous)
2. Two times rotational
3. Half frequency (in the vicinity of half rotational speed)

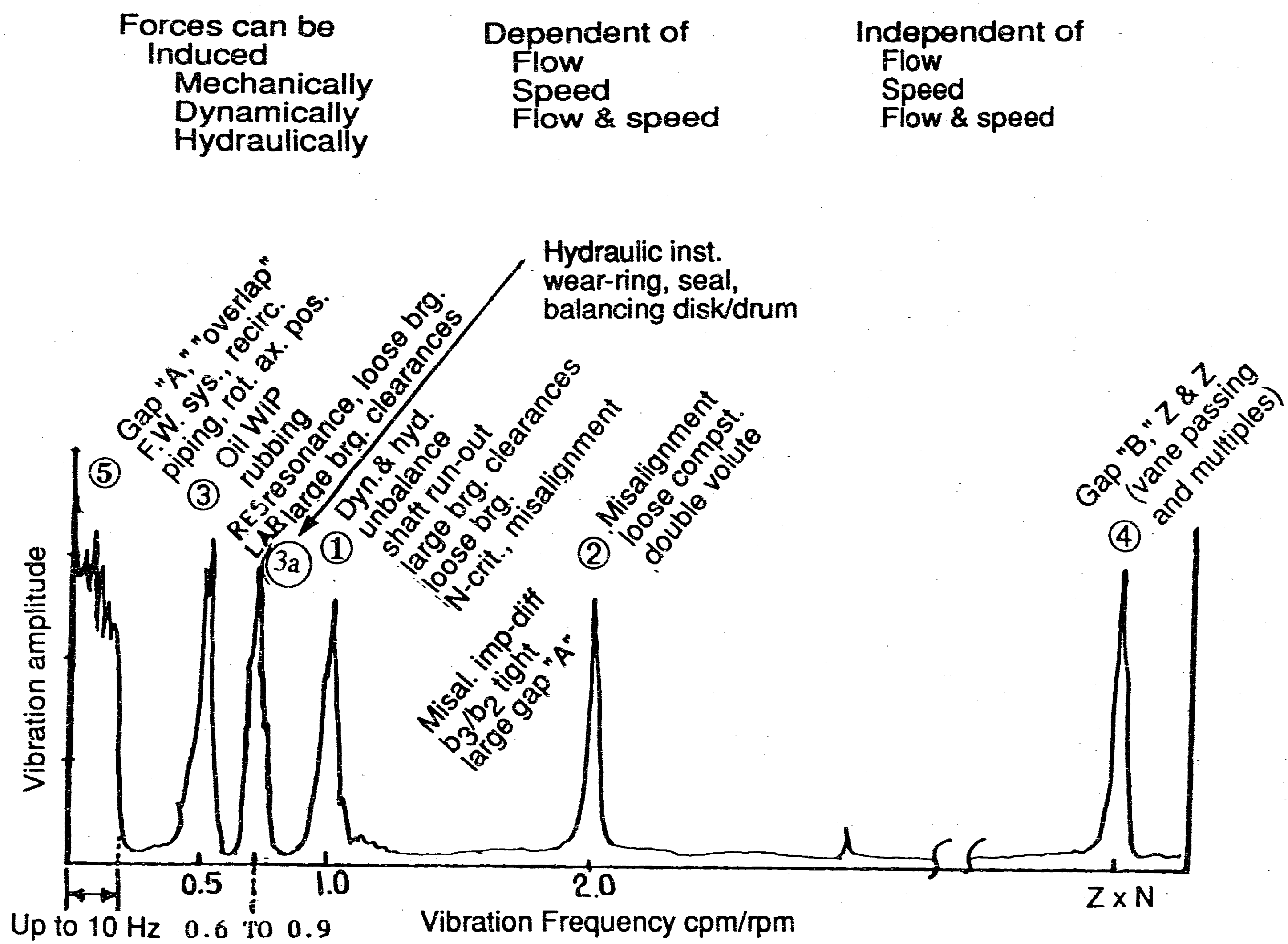

Figure 1. Major hydraulically induced vibration components (from References 1, 2, and 4).

3/a Subsynchronous, in the range of 0.6 to 0.9 x rpm

4. Vane passing (number of impeller vanes x rpm, and/or its multiples)

5. Low frequency (up to 10 Hz).

Give examples of each.

Examples where a combination of more than one frequency is present. How do you separate one from the other and how do you determine which one is the root cause of the problem?

Classical field examples are presented for each frequency range shown in Figure 1.

Frequency 1: APS–Palo Verde Nuclear auxiliary feed pump IST Alberta Power, Sheerness 1 and 2 (Figures 2 and 3).

- Start up boiler feed pump at 6000 rpm exhibiting only synchronous vibration component (Figure 4).
- Missing portion of the coupling key induced vibration to destruction level.

Dynamic balancing standards:

- ISO permits 15 W/N
- Milspec 167 4 W/N
- ERCO 1 W/N.

Example:

LCRA–Ferguson Plant:

- ISO would permit 3.6 Oz–In
- Milspec 167 0.97 Oz–In
- ERCO 0.24 Oz–In

We balanced the rotor to 0.10 Oz–In

Frequency 2: Two times rotational. There are many examples that could be presented. However, when that component is present, all other vibration components also show up as doubles (like IS 0.75 is present, there will be a 1.5 x n also present, or if vane passing is present, twice vane passing also will appear).

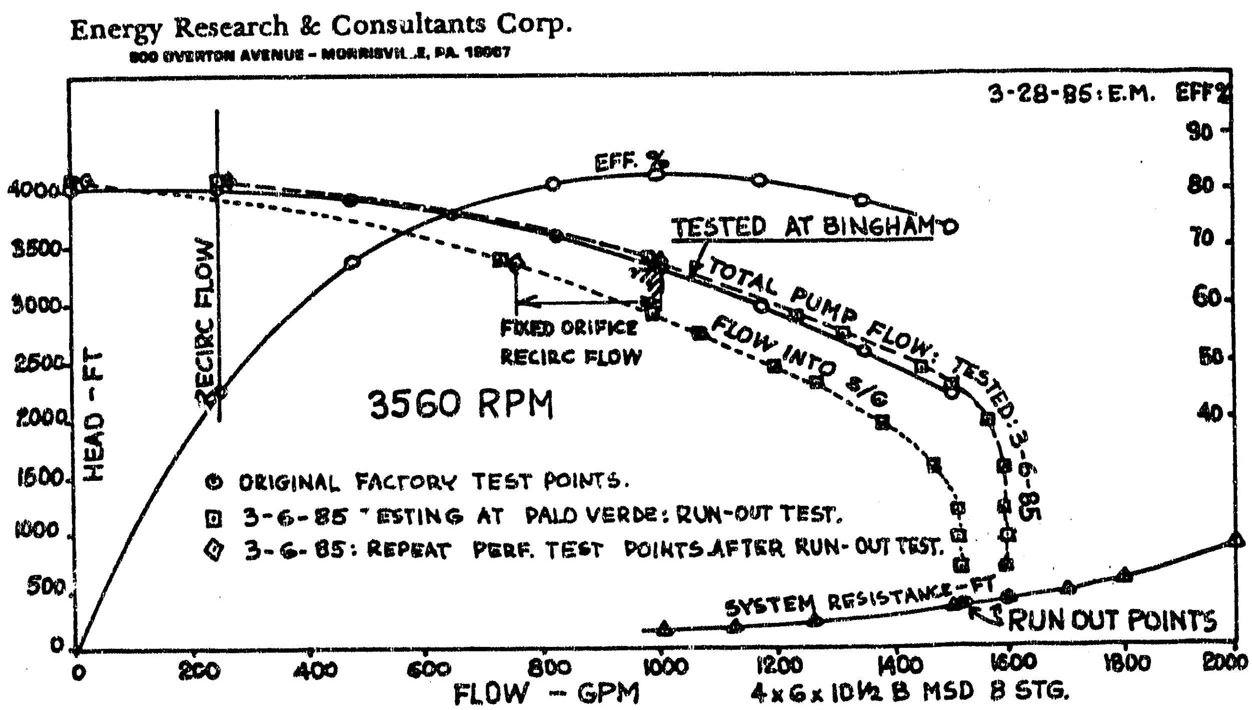

Figure 2. Bingham–made nuclear auxiliary feedwater pump testing at the Palo Verde Nuclear Generating Station, Unit 1.

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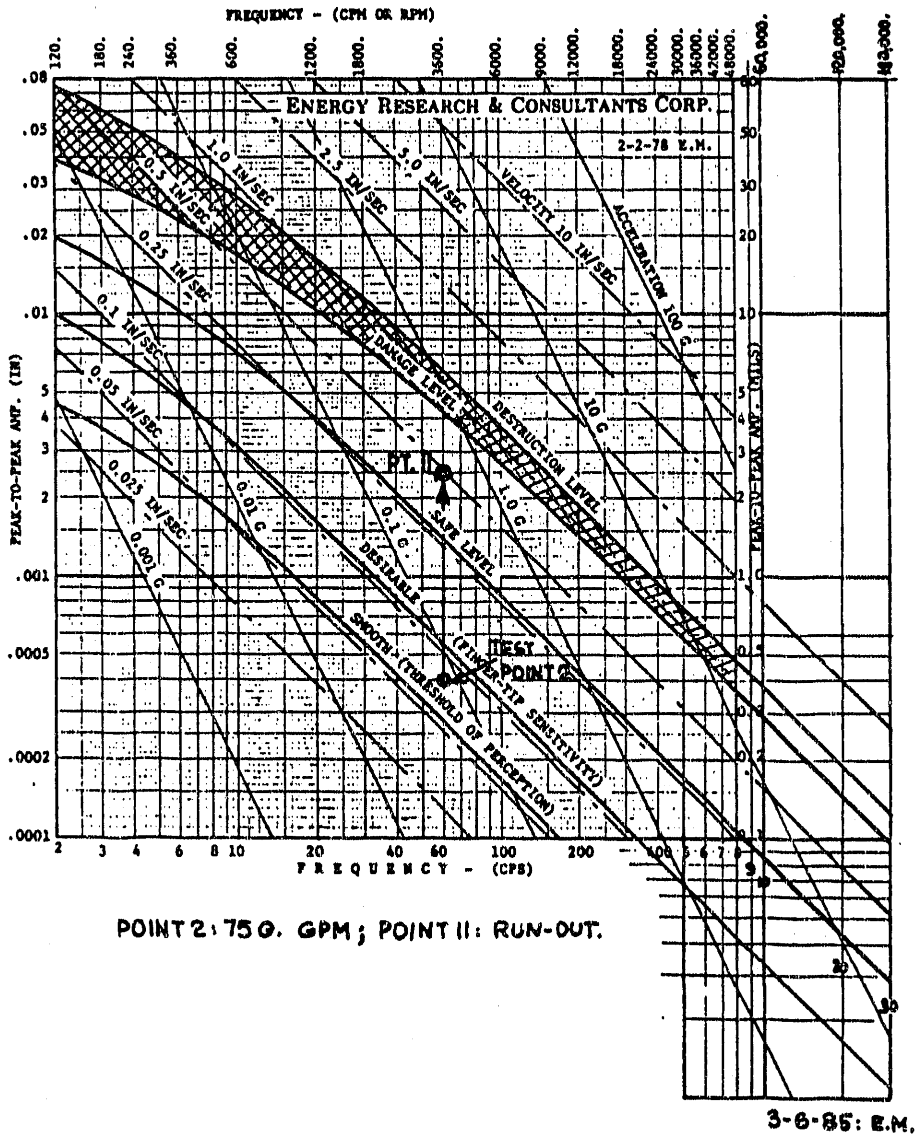

Figure 3. Allowable rotor vibration levels measured relative to the bearing cap (values shown are filtered readings at that particular rpm or frequency; experimental standards).

Frequency 3: In the vicinity of 1/2 x rpm rotor dynamic instability. Not permissible to any degree. Self excited vibration, destructive, will result in destruction of the rotor.

Frequency 3/A: Hydraulic instability. Four examples are given between frequency ratios of 0.6 and 0.92.

3/B: Borderline case at and around 0.6 this could be dynamic or hydraulic instability or the combination of both. This is one of the most difficult cases.

Frequency 4: Vane passing (no. of impeller vanes x rpm)

Frequency 5: Low frequency vibration below 10 Hz. Many things can excite this frequency such as foundation, piping, turbine drive governor, etc. But we are talking about pumps, hence give some pump examples how the pump can develop this frequency.

Combination of frequencies 1 to 5: Sheerness and Ferguson are excellent examples.

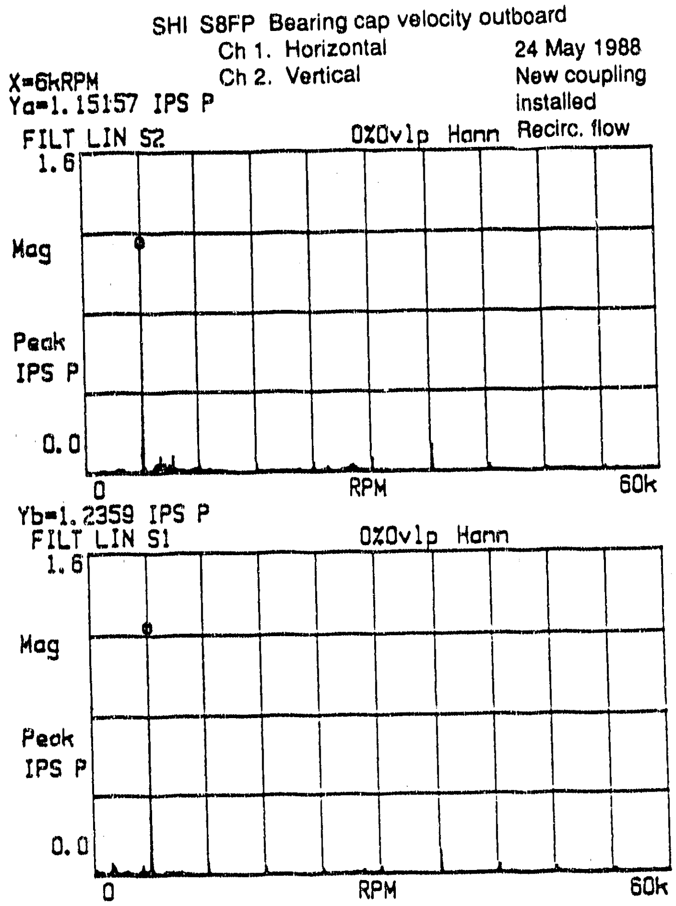

Figure 4. Rapidly increasing synchronous vibration component indicates possible high rotor unbalance. Rotor was properly balanced but during assembly incorrect coupling key was applied. Vibration increased to destruction level (motor driven, constant speed 6000 rpm start–up boiler feed pump). Alberta PW. LTD.

Figures 5 and 6 are presented to illustrate these concepts. Figure 7 is a graph of a rapid increase of the 1/2 x rpm vibration component. Figure 8 is and example of a sub–synchronous vibration that increased to damaging levels. For the final remedy, see Table 2.

Changing the tilting–pad type journal bearings did not eliminate the problem at Cane Run (Figure 9), indicating that the origin was hydraulic and not rotor–dynamic instability. Figures 10 and 11 provide additional examples.

Conclusions–Findings–Corrections:

1. Rotor was found out of balance by several times the permissible maximum. Corrected to proper level.

2. Bearing clearance was found more than two times the permissible maximum. Corrected to: 1 mil/in+1 mil std.

3. Looseness between bearing upper half and bearing housing was excessive. Corrected to zero by shimming.

After correcting the above three items, the feed pump vibration became less than one mil in any operating mode, resulting in complete absence of sub–synchronous vibration components.

IMPELLER–DIFFUSER INTERACTION–A REMEDY FOR INSTABILITY

All impeller and diffuser dimensions and geometry for a typical modern BFP are shown in Figure 12. Definitions of Gap A, Gap B, and a new dimension "Overlap" have not appeared previously in pump design technology, other than in publications by the author.

- *Gap A* is the radial clearance between the outer diameter of the impeller sideplates

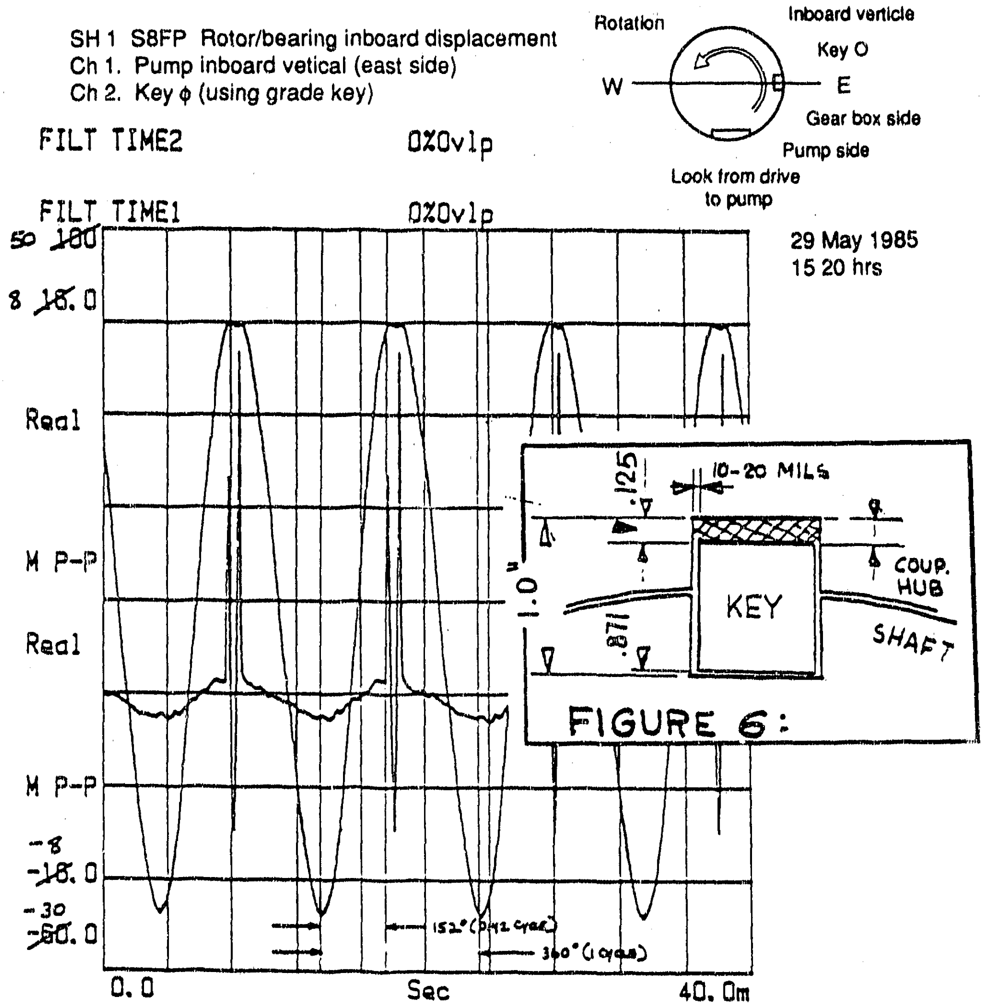

Figure 5. Rapidly increasing vibration amplitude is purely synchronous (1 x rpm) leading to destruction caused by a "minor" mistake of applying the wrong coupling key.

Figure 6. Detail of coupling key.

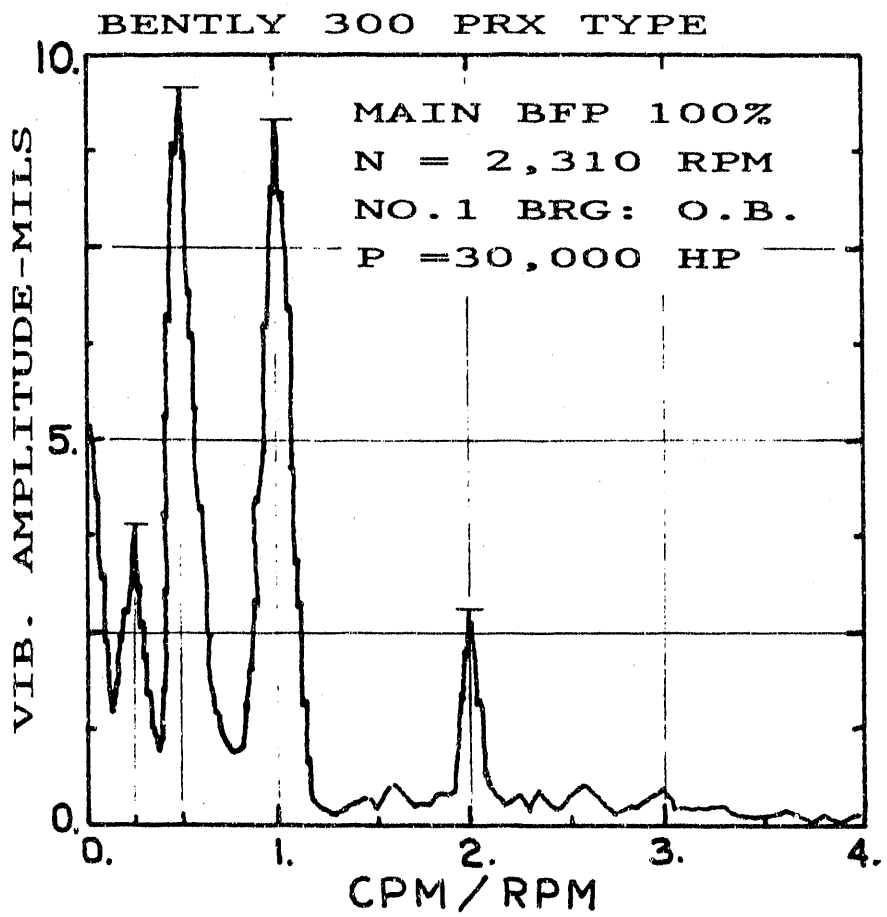

Figure 7. Rapid increase of the 1/2 x rpm vibration component measured at the main boiler feed pump O.B. The journal bearing did not permit the unit to go on–line. This component was already over 10 mils at 3,100 rpm. Operating speed at full load is over 5,000 rpm.

(hub and shroud) and the inner diameter of the diffuser channel sideplates, given in inches or more commonly in mils (e.g., 50 mil = 0.050 in.) Usually, this dimension does not exist in volute designs, proving that its importance was not previously recognized.

- *Overlap.* The axial "Overlap" of Gap A is the useful axial length of the small hydraulic passage between the impeller and diffuser/volute sideplates. The recognition of its major technological significance in pump hydraulic design is a major breakthrough. It has a very great controlling effect on both the static and dynamic components of the axial thrust damping developed by the impeller, damping of rotor motion, pump hydraulic stability, and feedwater system stability.

- *Rotor axial positioning* actually means centering the hydraulic channels of the impeller exit and diffuser inlet. Figure 13 shows various configurations, ranging from good to very poor.

- *Ratio (b3/b2) of the hydraulic widths* of diffuser inlet (b3) to impeller exit (b2) is very critical for rotor axial positioning A general rule in diffuser–type pump hydraulic design calls for this ratio to have a minimum value of not less than 1.15.

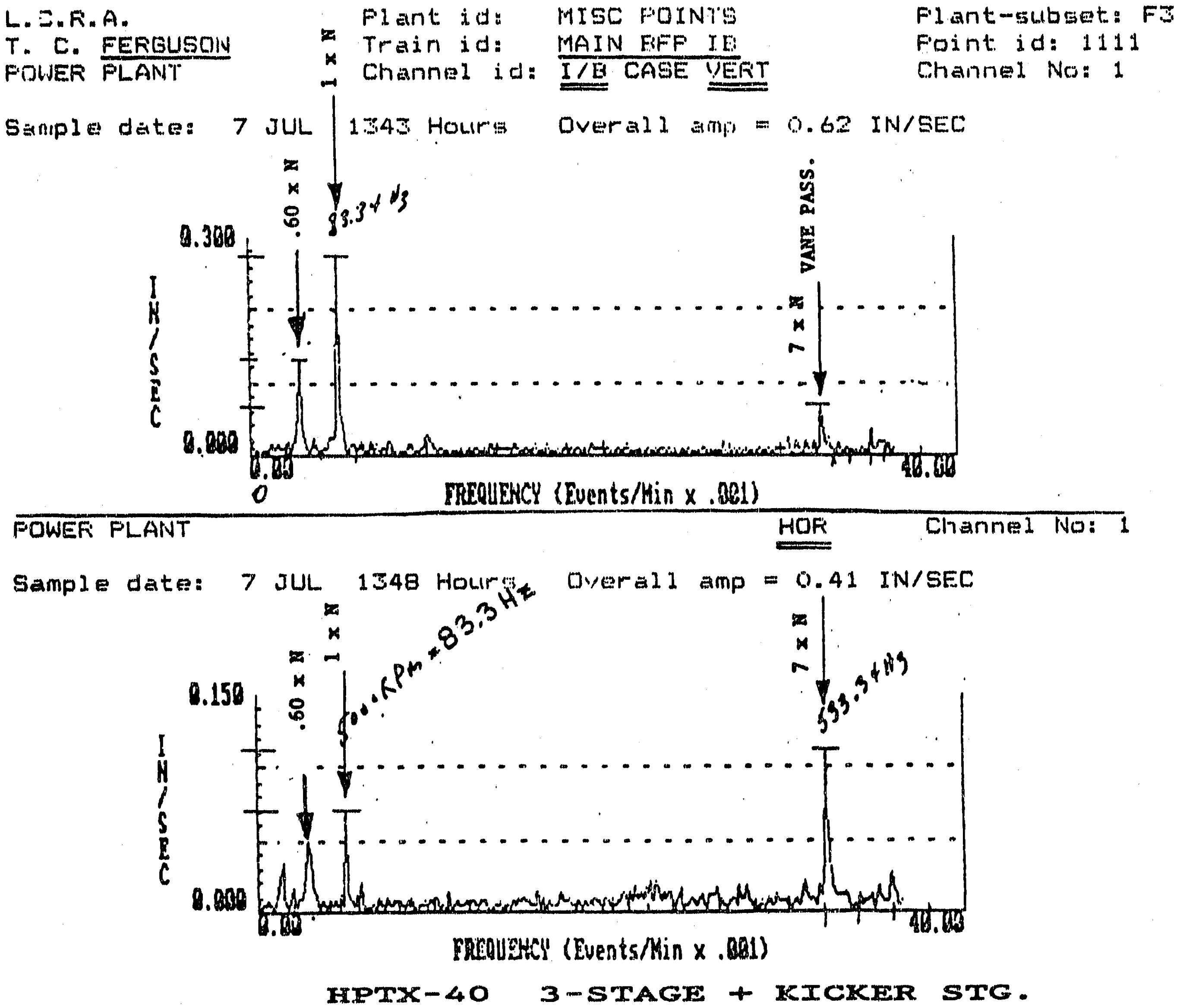

Figure 8. Sub-synchronous (0.6 x rpm) vibration component continuously increased to damaging levels. For final remedy, see Table 2.

- *Impeller sidewall thickness* ("S") has a major effect on all the above geometries.

- *Gap B* is the radial clearance between the impeller vanes at the outer diameter (D2), and the diffuser/volute vanes at the inner diameter (D3). It is always given in percent of D2 (Example: D2 = 16 in., D3 = 17 in., D3/D2 = 16/17 = 1.0625; hence the gap is 6.25%).

- *Axial space* behind the impeller hub and shroud can have a great effect on disk-friction related losses. If the space is too narrow, it permits the boundary layer to dominate the flow region.

Description of Gap A and the Overlap

Gap A and its overlap greatly influence fluid mechanical action at the impeller periphery, in the impeller channels, and the diffuser passages, in the gap itself, and in the space behind the impeller sideplates. It also acts as a low-frequency filter between the active hydraulic channels and the space behind the impeller. Its effect can be quite different for various pump types, designs (such as vertical or horizontal, double-suction or single-suction impeller, single stage or multistage,

Table 2. Summary of the boiler feed pump and booster pump modifications at LCRA–Ferguson during 1988:

- (February 28, 1989)
- Balanced 36 × better than the "ISO" (Sulzer recom.) standard.
- Changed both couplings (badly worn) and fine balanced.
- BFP coupling key wrong (resulted in 5.6 oz–in unbalance). Made correction.
- *Journal BRG*: Changed Sulzer BRG. to *tilt–pad* (Bec. the 1/2 freq. whirl). Half freq. whirl was exactly 1/2 × rpm.
- Put the "MAKAY" taper into the balancing disk, minor dimensional corrections.
- Redesign retaining procedure of bal. disk (remove copper shim).
- Change impeller wear–ring geometry.
- Change impeller wear–ring material and hardness to up–to–date standards.
- Found many cracks in seal housings, weld repaired, remachined.
- Found many cracks in impeller vanes at the eye. Welded up, hardness of impellers brought back to present standards.
- Barrel Vertical Faces (FITS) out of perpendicular as much as 50 mils. Rebored.
- Re–shaped all diffuser channel to proper geometry.
- Found kicker stage backwards (wrong rotation). Corrected it.
- Welded up diffuser side plates and impeller O.D.–s for proper *Gap A* & *overlap*.
- Opened up Gap "B" to proper dimensions.
- Introduce proper shrink–fit between impellers and shaft.
- Original shaft seals were changed to durametallic mechanical seals just prior to above changes. Having no back flow from seal to the balance disk L.O. chamber, now we can monitor L.O. flow.
- Final rotor assembly was made as good as possible.
- *Master* alignment at the plant (by Bill Saxton) between turbine–BPF–gear–booster.
- Pinch–checked all bearings (some were way out).
- Blued coupling bores to shaft.

Results:

- Vibration: Levels from recirc. flow to max load never exceeded 0.2 mil(=.05 mm) at any of the bearings.
- *Efficiency*: Excellent data were taken at 300 MW before and after. A clear *3.8%* improvement in efficiency was achieved.
- *Booster Pump*: The previously heavy axial vibration (shuttling) completely gone (Bec. of the change in gaps "A" & "B", and the "overlap ratio".

Future Action:

- Booster impeller has an unusually "Large Eye" geometry, resulting in heavy suction piping vibration. Gap "A" change eliminated the heavy axial shuttling of the rotor, but the booster pump needs either:
  - New impeller design, or
  - Dr. Paul Cooper's "backflow recirculator (Eye Catcher)
    (New impeller design is always a risk. The "Eye Catcher" works).

Table 2. (continued)

Conclusion:

The boiler feed pump (Sulzer type: HPTX–40–3) is now a most up–to–date machine as much as some design restrictions restrained us from complete up dating. The machine is more efficient than ever, and is good for the next 40 years of operation.

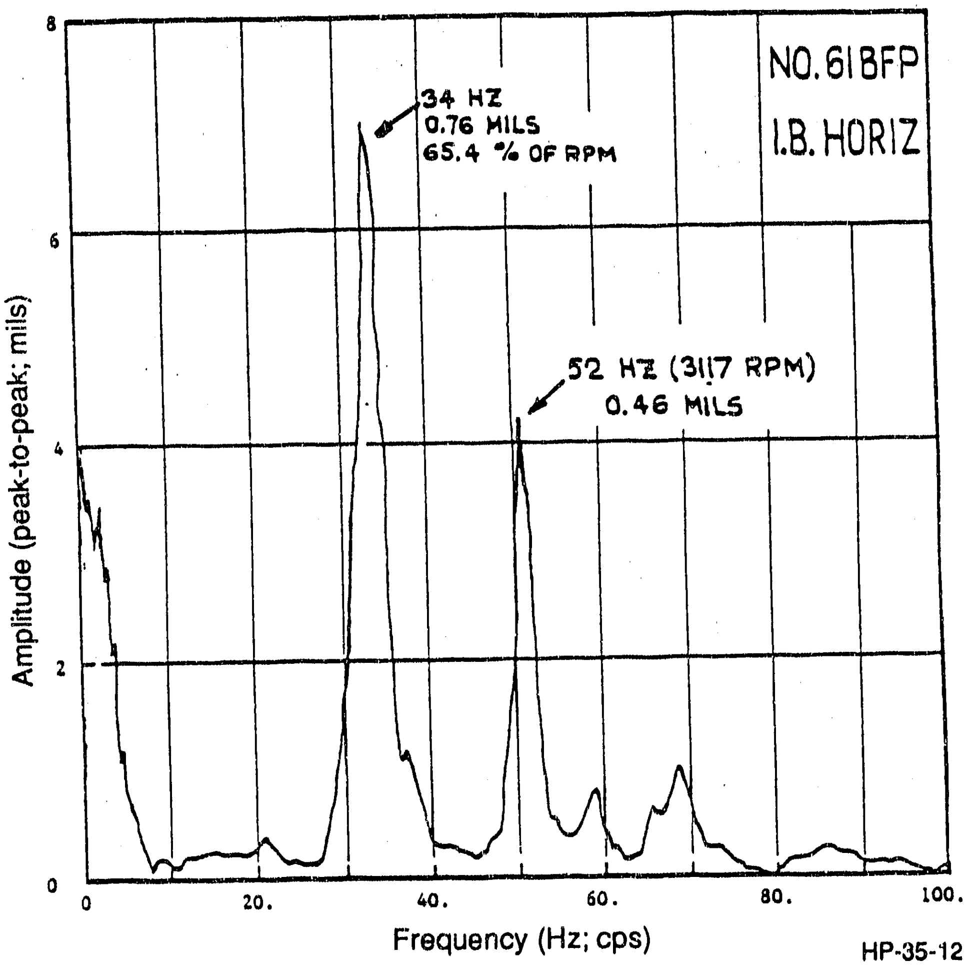

Figure 9. Onset of subharmonic resonance as detected in the I.B. horizontal component of the No. 61 boiler feed pump at Cane Run, owned by Louisville G&E Co. The magnitude of this subharmonic component rapidly increased to 7 mils, necessitating a unit shutdown.

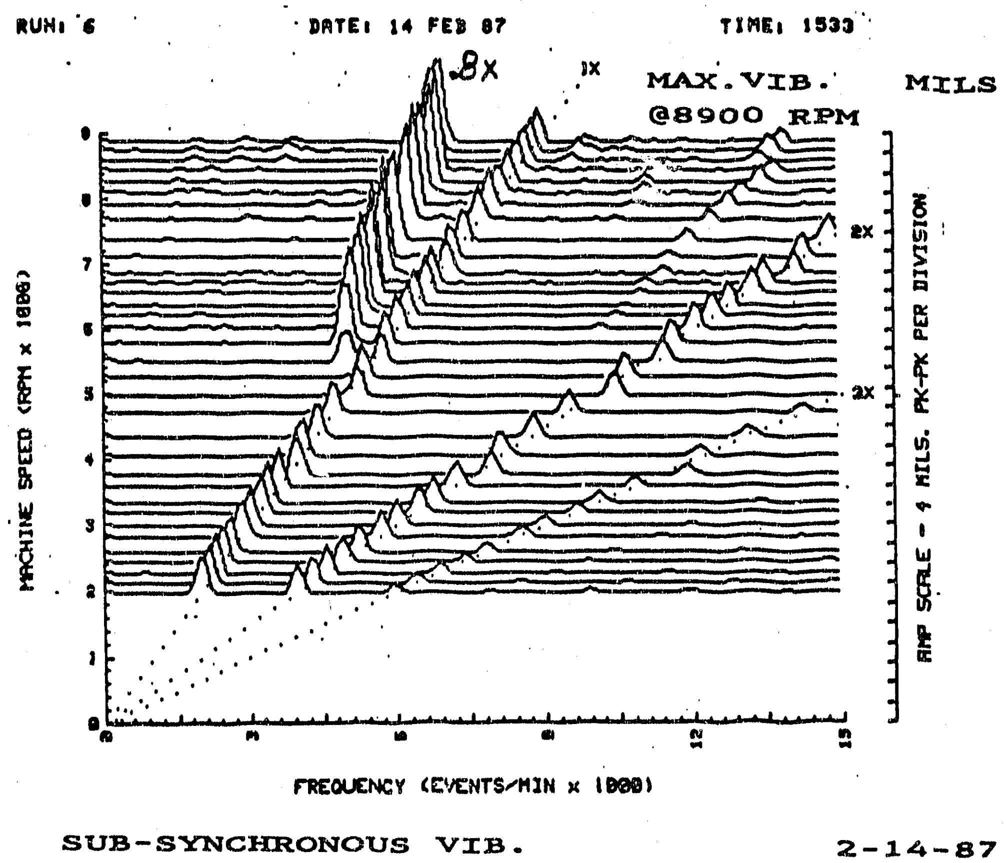

Figure 10. Frequency ratio 0.8 rapidly growing, leading to destruction of the rotor. Pumps at P.S. Oklahoma operate at 9,200 rpm.

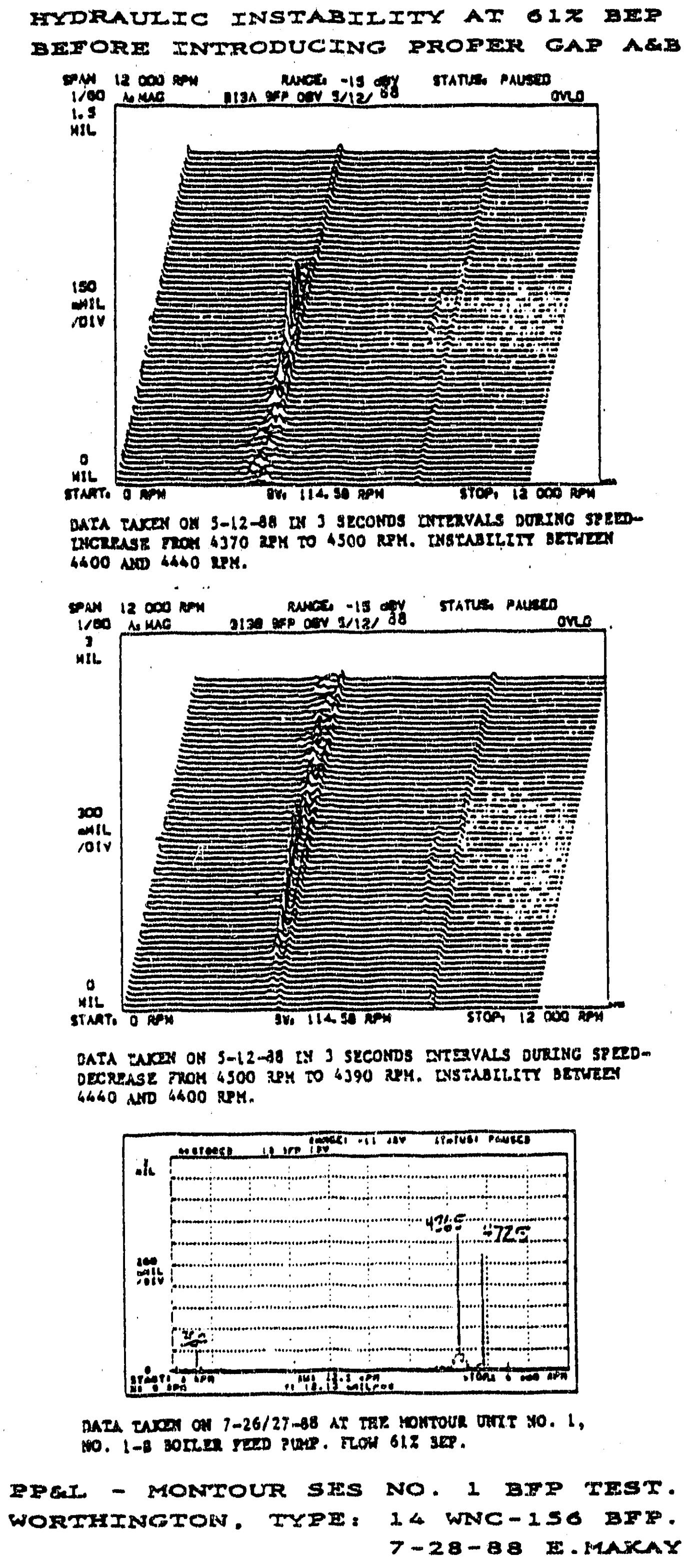

Figure 11. The highest sub–synchronous frequency ratio experienced in a BFP.

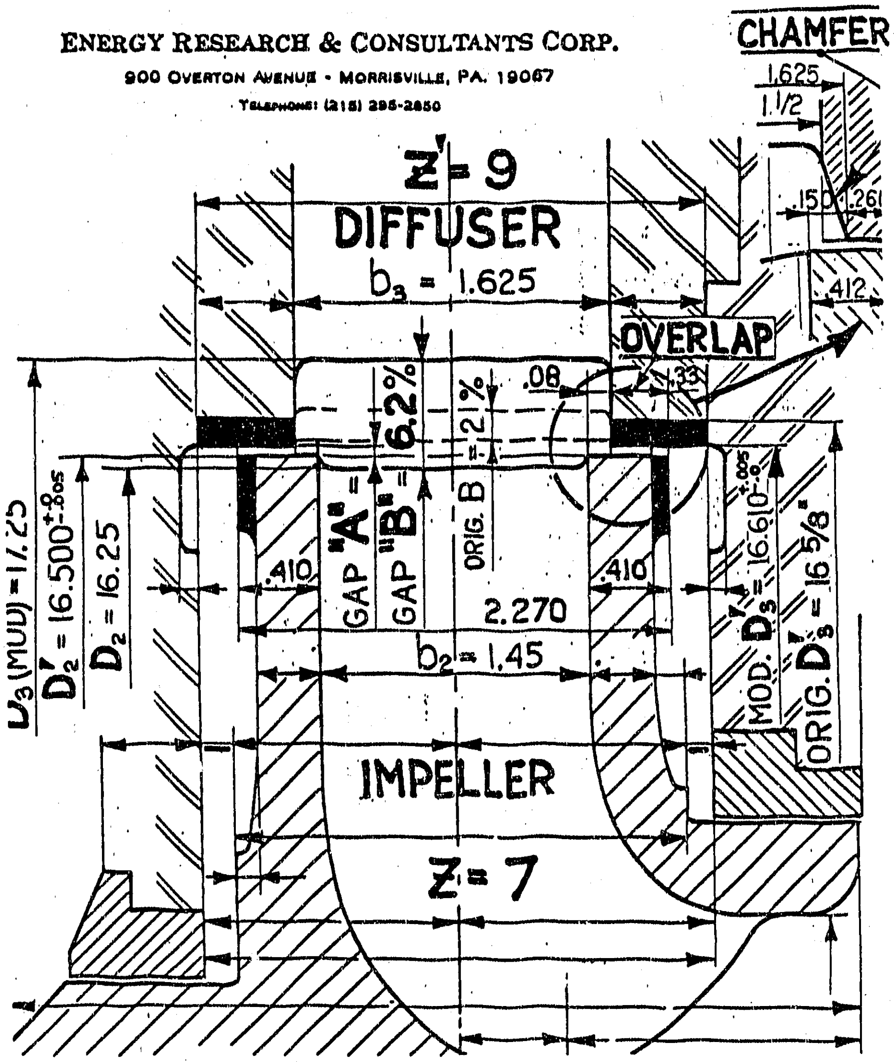

Figure 12. Correct rotor position, correct Gaps "A" and "B", correct overlap ratio, impeller and diffuser centerline to centerline, and axial thrust normal in all flow conditions.

light or heavy rotor configurations), N_s, S_s, and energy per stage. It is most significant for high–energy pumps such as BFP's, NFP's, and high–head feedwater booster pumps.

Gap A is an effective filter for high amplitude responses at very low frequencies (up to 10 Hz) for high–energy–input pumps. Its effect diminishes as S_s increases. The high energy concentration shifts toward the impeller eye at flows below BEP because of the dominating action of flow recirculation at the eye. When the S_s is very high, as in inducers, the Backflow Catcher of Cooper[13] is very effective for eliminating low–frequency flow oscillations and stabilizing the head curve at lower flows. There is a region between low flows and high flows where both the Overlap and the Backflow Catcher can be effective. Probably the combination of the two concepts would offer the most effective solution in eliminating hydraulic instability caused by low flow. Neither concept is recognized, understood, or accepted by most hydraulic designers. Current pump design philosophy is very conservative, especially for utility applications.

The comprehension and physical visualization of the Gap A and its overlap is not as simple and apparent as that of Gap B (Figure 12). The complexity of the interactions between flows at the impeller exit, diffuser flow reversal, boundary layer effect, creation of a new flow pattern in Gap A, and the influence of the radial gap and its overlap in the axial directions is beyond simple visualization. The interaction between the impeller exit and diffuser inlet geometry and their connection with the effect of Gap B is a complex phenomenon. Publications on this subject are very limited.

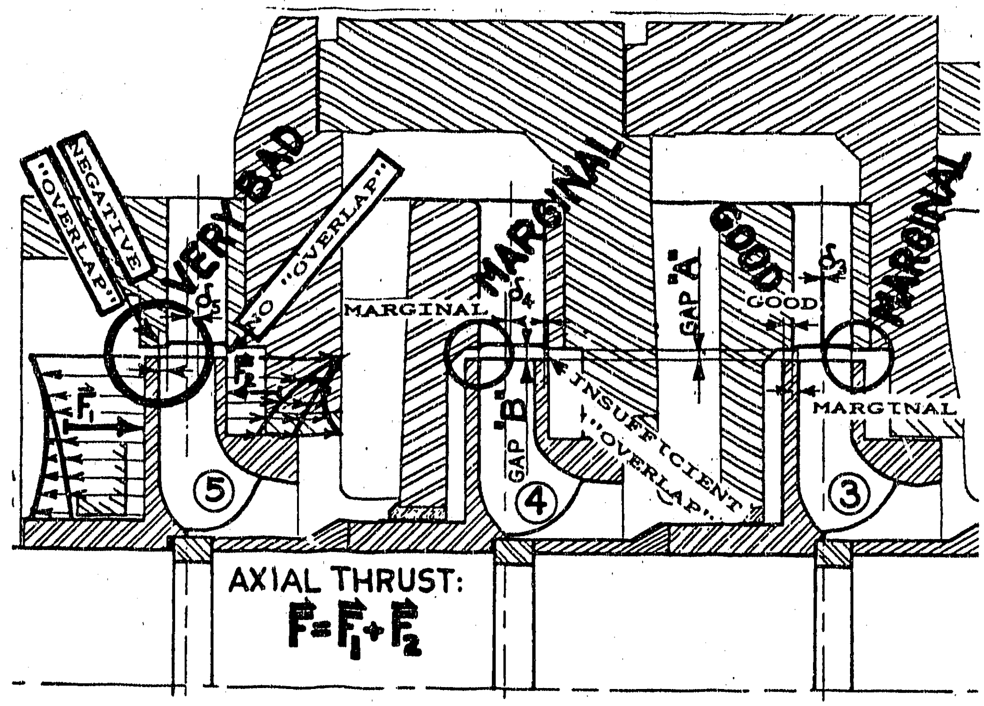

Figure 13. 5–Stage BFP. If Gaps "A" and "B" are incorrect, hydraulic instability may occur at low flow operation. With axial misalignment (as shown above for Stage 5), pump is unstable at all flows with or without proper "A" and "B".

Occasionally, the importance of this area is mentioned. However, the influence of Gap A and its Overlap on pump stability behavior at part load, on system instability, on the static and dynamic components of axial thrust, or the use of this geometry as a corrective device for the above items is found only in References 1, 2, 3, 5.

The multiple effects of Gap A geometry and its Overlap are strong and unique, especially when they are properly coupled with the Gap B geometry. Based on many field applications and measurements, they can be summarized as follows:

1. *Head–Capacity Curve Stabilization.* Introduction of the proper Gap A geometry in a pump, coupled with the proper Overlap and vane–to–vane radial distance between impeller and diffuser/volute (Gap B), improves the head curve stability at part load, as shown in Figures 2, 14, and 15. This improvement may eliminate or improve instabilities thought to be from pump, feedwater flow, feedwater control system, valve, and often BFP turbine–drive–governor instabilities. It can have a significant effect on piping vibration and on failures of major and minor components.

2. *Axial Thrust Stabilization.* Stabilizing the axial thrust eliminated abrupt changes in the static magnitude and dynamic fluctuations of axial thrust; hence, it can eliminate such effects as the following:

   - Thrust–bearing overheating or failure, or any axial–thrust related problem (if drum: thrust bearing; if disk: disk and thrust bearing failures.)
   - Part load instability of static and dynamic axial forces, especially for single–stage double–suction feed pump designs.
   - Thrust reversal problems such as reported in References 18 and 19.
   - Balancing disk/drum leak–off flow fluctuations (unless seal injection flow control is not properly set).
   - Unstable balancing disk/drum performance (usually results in catastrophic failure).

   It is shown in Figures 16 to 20 how the rotor axial thrust changes as the %BEP

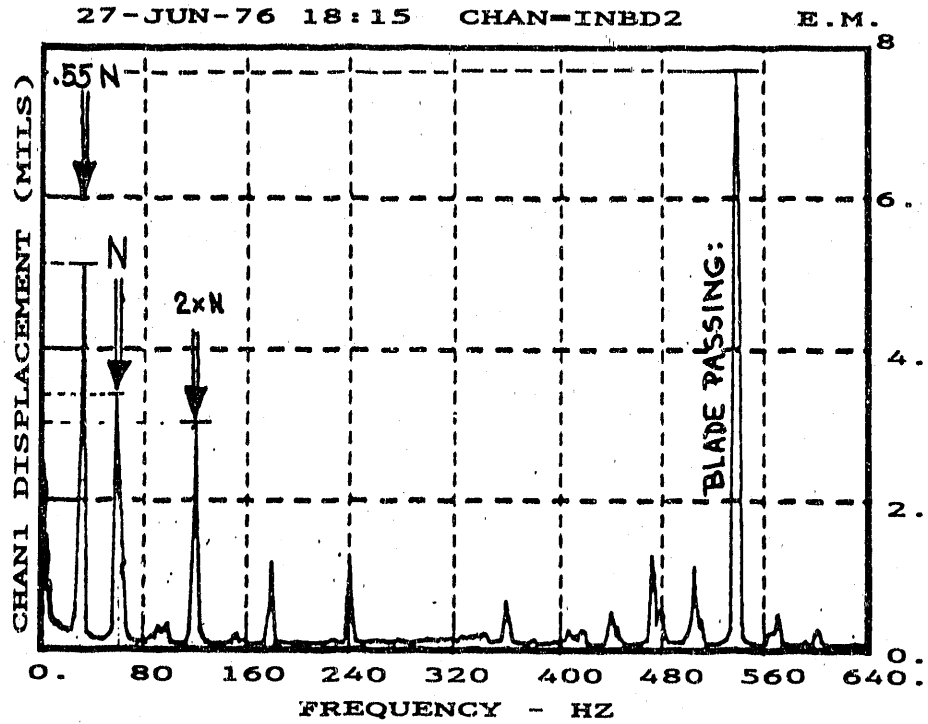

Figure 14. Growing vibration amplitude of an 8-stage boiler feed pump at vane-passing frequency is followed by rubbing induced partial frequency (0.55 x rpm) vibration component (Reference 1).

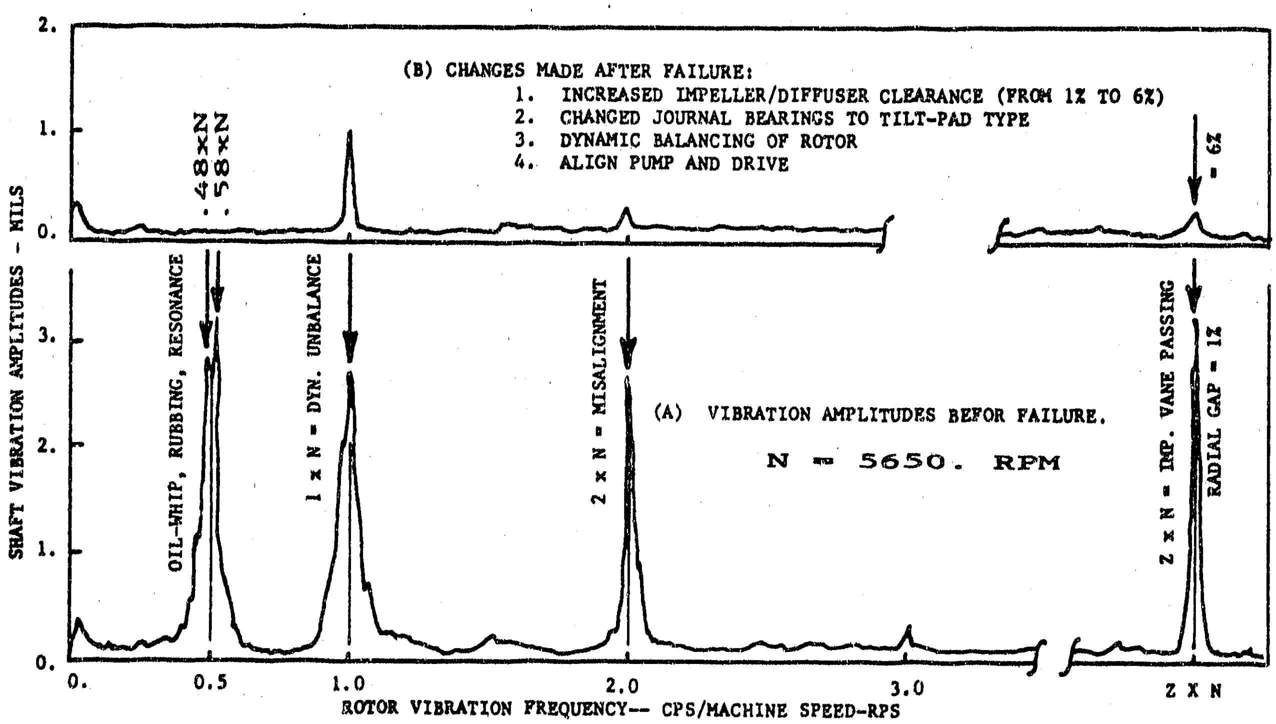

Figure 15. Multi-stage boiler feed pump shaft vibration history before failure and after proper modifications were introduced.

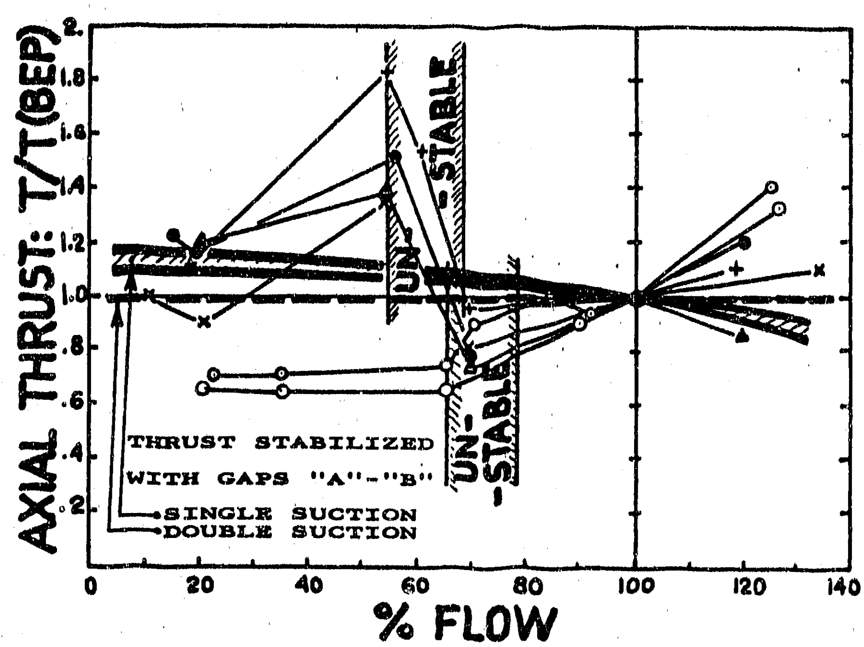

Figure 16. Axial thrust variation of several multi–stage BFP with non–optimum impeller–to–diffuser geometry. Also shown with optimum geometry both for single and double suction impeller designs.

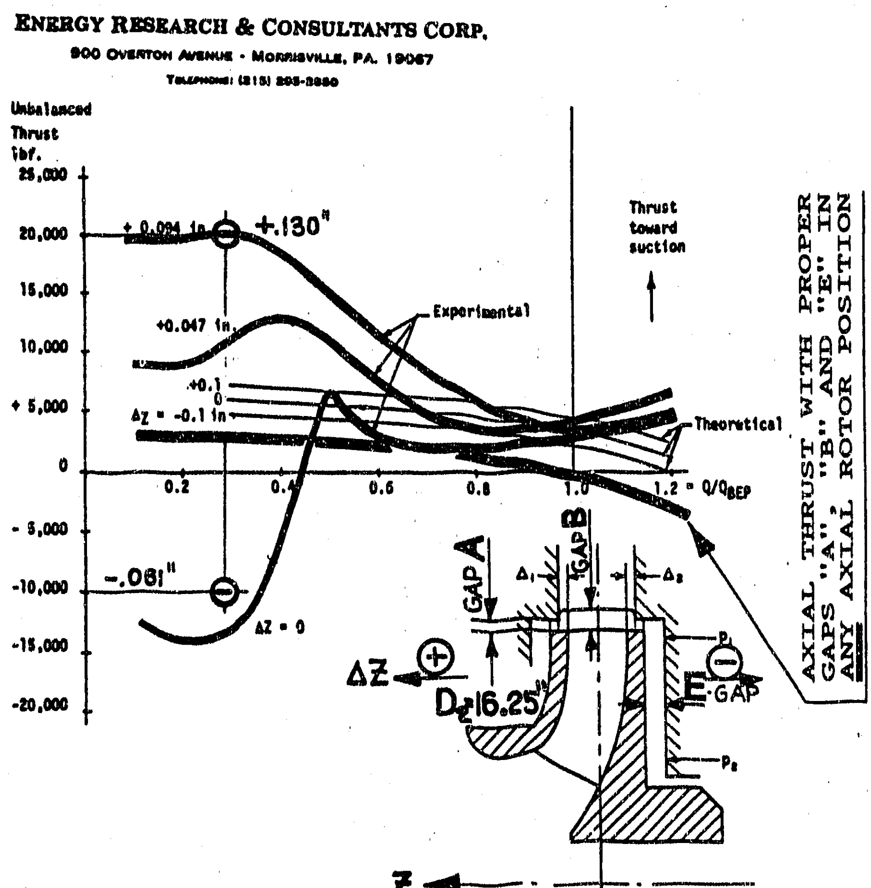

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Figure 17. Axial thrust variation with flow rate: effects of small axial shifts in impeller location on axial thrust for an 8–stage feed pump with balancing "drum" (Reference 13).

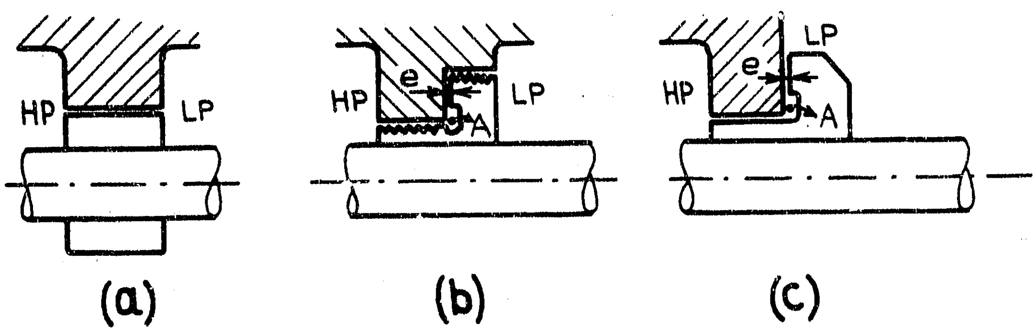

Figure 18. Type of balancing devices for boiler feed pumps: (a) drum; (b) and (c) disk.

flow varies, how axial misalignment of the rotor changes the thrust. The response of the axial balancing device is shown for the "balancing drum", the "balancing disk," without tapered face, and with tapered vertical face. It is clear that the disk with tapered face has superior performance.

3. *Rotordynamic Improvement.* Provides positive rotordynamic damping to stabilize motion, resulting in smoother running pumps, less rubbing and wear, less bearing and seal problems, and longer rotor life; therefore, the result is less need for maintenance. This was not known until many field modification showed repeated and consistent improvements of rotor behavior. The evidence is not based on theoretical calculations, but on a long list of field applications.

4. *Feedwater System Instability Improvement.* Elimination of feed pump instability reduces, or in many cases completely eliminates the system instability problem. This is probably the most significant contribution of the geometry of Gap A and its Overlap.

5. *Increased Pump Efficiency.* When the geometry modification is properly incorporated in the pump, the pump efficiency increases. This is a minor but significant contribution to development of pump technology.

The foregoing are critical from an engineering perspective; however, the bottom line for the power stations is:

- Increased plant availability
- Significant decrease in maintenance efforts
- Decrease in operating cost
- Increased reliability of operation and system behavior, which is particularly important for cyclic operation of very large units
- Noise reduction when properly combined with Gap B modification
- Increased pump efficiency.

Many cases have shown that the concept works equally well for both diffuser–type and volute–type pumps. The effect of Gap A was first demonstrated for diffuser–type designs. The first full proof of applicability to volute–type pumps was in 1976 for single–stage double– suction high–energy–input booster pumps. All other pumps previously modified were diffuser types. This case opened a new road for research on volute–type pumps. Many single–stage double–suction and single–suction multi–stage volute–type pumps now have been modified. These fully support the evidence that the technology applies to volute–type, as well as diffuser–type pumps.

Radial changes of axial thrust as the rotor position changed was reported to create major unexplained problems. With sufficient Overlap, the thrust variation becomes insignificant when the impellers are shifted in the axial direction. The Overlap assures that the gap will not be unsealed fluid mechanically and remains in full control for the flow path. Whenever an impeller has a sideplate with an incorrect thickness, as in a very large number of older designs, the Overlap and Gap A cannot be effectively utilized. Even if Gap A is reduced to a very small value, several problems exist: (1) there is insufficient Overlap to make it effective; (2) if b3/b2 is correct (1.15 or larger), the Overlap is insufficient; and (3) if the ratio is incorrect (say 1.0), the flow catches on the diffuser sideplate whenever

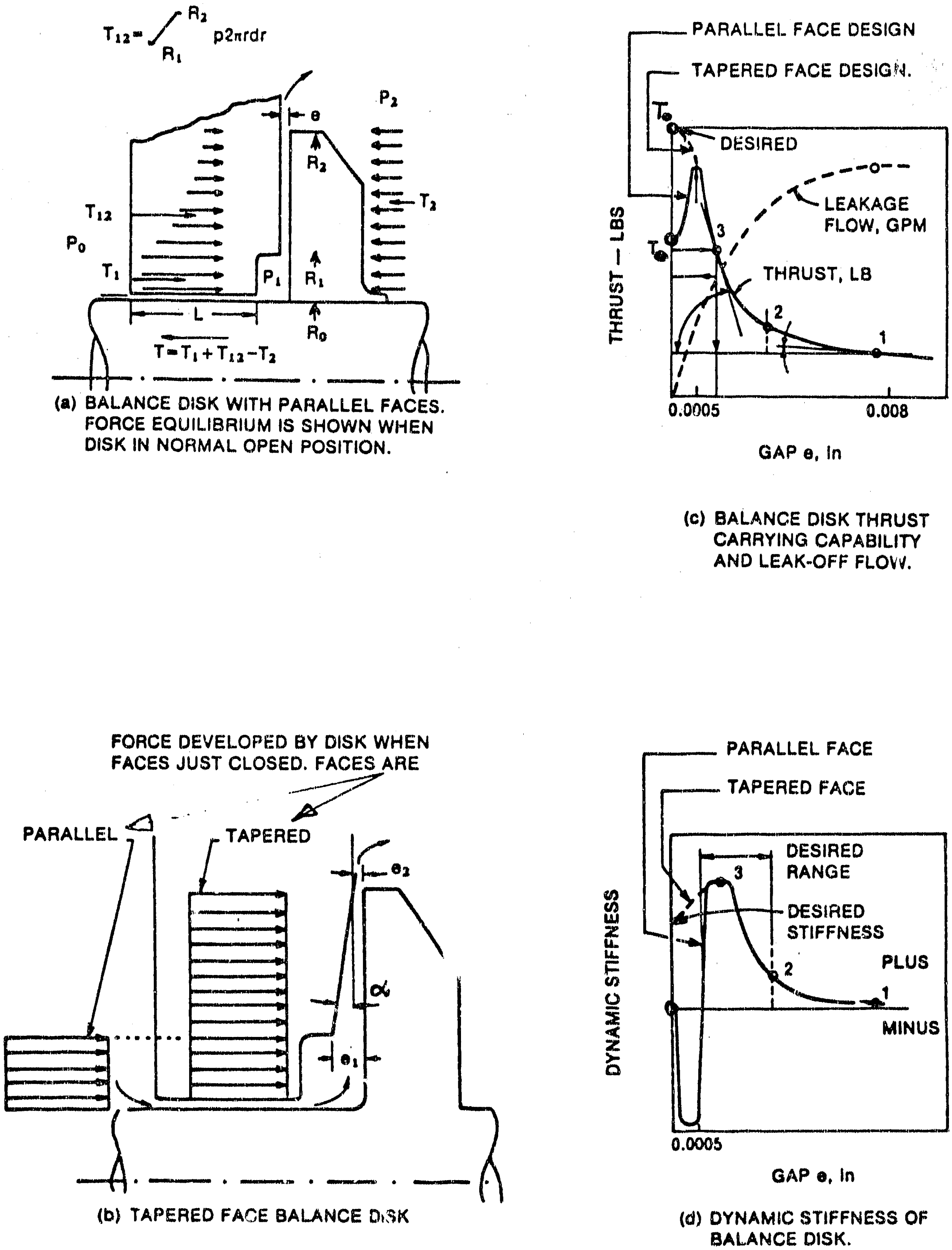

Figure 19. Details of balance disks.

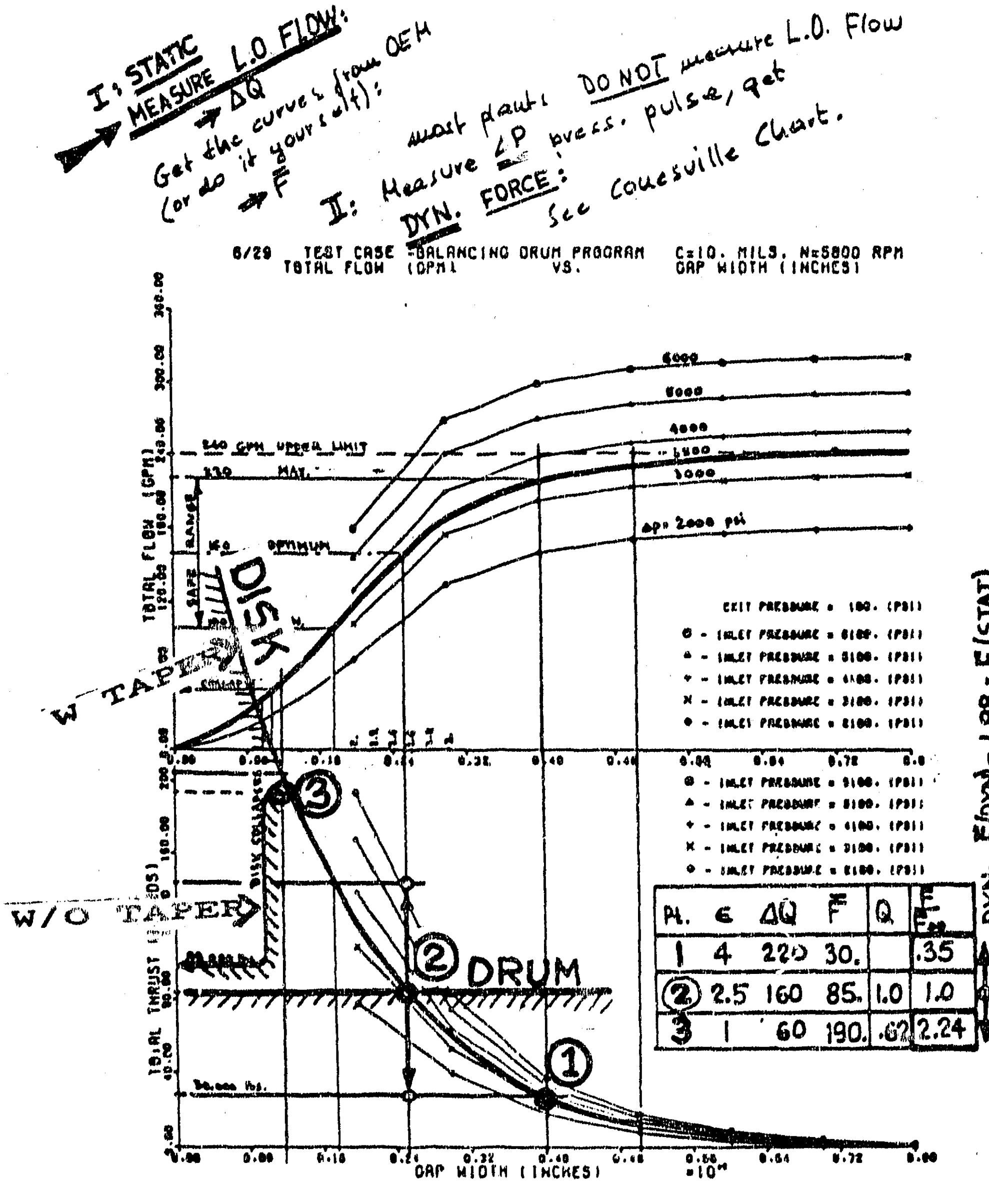

Figure 20. Balancing disks (a) without taper between faces (if thrust increases above a certain load, disk collapses. Major rotor failure is unavoidable); (b) with taper load carrying capability, keeps increasing, prevents failure of rotor. Balancing drum is a constant thrust design. If thrust changes (+ or −) thrust bearing will take the extra load. Thrust bearing cannot take the increased load between point 2 and 3, or the decreased load between points 1 and 2. Actual field data: disk without taper failed five times. Drum design would have failed more. Introduction of taper prevented further failures (Reference 9).

there is axial misalignment, causing thrust reversal and large vibration amplitudes.

Some designs specify a radius or a chamfer on the impeller sideplate at the O.D., or specify a very thin sideplate. This reduces the effectiveness of the gap. A wall thickness of not less than 3/8 inch is recommended, preferably 1/2 inch on large feed pumps. The available axial clearance behind the impeller sideplates also must be examined. Insufficient clearance greatly increases fluid friction and reduces pump efficiency. Axial float of the rotor may also become a problem.

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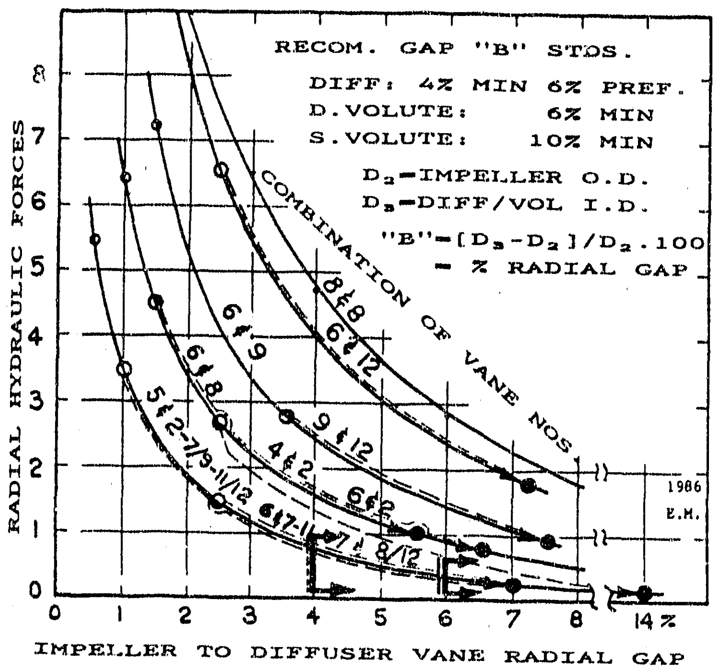


INFLUENCE OF IMPELLER TO DIFFUSER/VOLUTE RADIAL GAP "B" ON PRESSURE PULSATION AND MAGNITUDE OF RADIAL DYNAMIC FORCES. THE FREQUENCY OF THE FORCES IS VANE PASSING OR ITS MULTIPLES. IF THE COMBINATION OF THE NUMBER OF VANES OF THE IMPELLER AND DIFFUSER IS NOT AS SHOWN ON THE BOTTOM LINE ABOVE, THE RADIAL GAP "B" MAY HAVE TO BE CONSIDERABLY LARGER THAN RECOMMENDED ABOVE.

COMBINATIONS OTHER THAN SHOWN ON THE BOTTOM LINE ARE NOT RECOMMENDED FOR HIGH SPEED AND HIGH ENERGY INPUT FEEDPUMPS IN POWER PLANT APPLICATIONS FOR TROUBLE FREE OPERATION.

Figure 21. Variation of radial forces with Gap "B" as a function of vane combination.

Descriptions of the Radial Gap – The comprehension and physical visualization of the Gap B geometry effect is comparatively simple relative to that of Gap A. Because of the finite thickness of the vanes at the impeller exit and the formation of a boundary layer adjacent to each impeller vane, a pressure wake originates that is interrupted as the vane passes each diffuser/volute vane. The interaction between the wake and the diffuser/volute creates a dynamic force that can be particularly great at part–load flow conditions. The magnitude of the force is a function of the gap between the impeller exit and diffuser/volute inlet. The variation of this force is a function of percent radial gap, or Gap B. This force is also responsible for the noise generated by a centrifugal pump. The smaller the radial gap, the greater the shock caused by the wake or the dynamic force of the pressure pulsation.

The smaller the radial gap between the impeller vane exit and diffuser, volute vane/volute tongue (often called cutwater) inlet, the higher the shock (pressure pulsation magnitude). The frequency of the shocks depends on the number of impeller vanes times rpm or multiples of this number. The effect depends on the vane number combination. It can have a detrimental effect on various structural parts of the pump, as well as being responsible for the pump–generated noise level. This gap is the "Noise Generator" of a centrifugal pump. The larger the radial Gap B, the longer the free unguided flow–path of the liquid becomes. This increases the possibility for the flow finding a more favorable inlet path to the diffuser, especially under conditions of high incidence angles that occur at partial capacities.

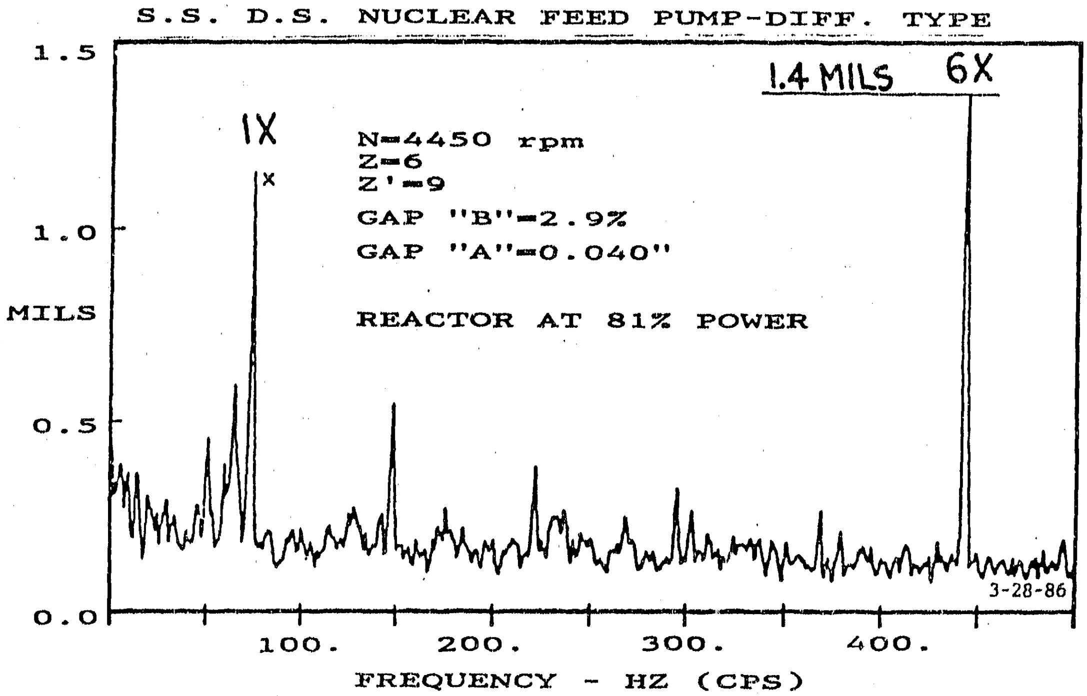


Figure 22. Damaging vibration amplitude at vane passing frequency of a large nuclear feed pump due to the unfavorable impeller diffuser vane combination (6 and 9).

For a high–energy, high–speed BFP, the gap must not be less than a certain minimum value to minimize the dynamic forces. A smaller gap can induce forces that fracture components such as diffuser vane tips, impeller sideplates, diffuser retaining bolts, etc., and cause failure of many other minor and major components internal and external to the pump.

An impeller is more likely to fail if its natural frequency coincides with the vane–passing frequency or its multiples. This natural frequency can be determined only be measurements, which unfortunately can be made only after the impeller has been designed and manufactured. The amount of shrink–fit applied between the impeller hub and the shaft also influences the frequency.

While the results from the application of Gap A are very powerful, it is mainly effective for high–energy–input centrifugal pumps. The effect of Gap B, however, is universally valid and effective for any pump type, any specific speed (N_s) or suction specific speed (S_s), and any application. Of course, the failure rate caused by an incorrect radial gap between the vanes is highest and most noticeable for high–energy feed pumps.

When a tight Gap B is increased to the "preferred" size, the following are some of the most notable improvements of this geometry corrective modifications:

- Reduction of vibration and pressure pulsation amplitude at vane–passing frequency, reducing failure rate, and reducing maintenance. The fatiguing effect of this phenomenon is a leading cause of various failure mode, including shaft breakages.
- Reduction of vibration amplitudes at other than vane–passing frequency, especially at synchronous (rpm) frequency. When rubbing excites a subsynchronous component for a flexible rotor, the vibration can be eliminated by applying the proper radial gap.
- Substantial noise reduction is a direct result of an enlarged radial gap.
- Elimination of diffuser vane–inlet cavitation and of breakage of diffuser inlet tips. This is one of the most commons symptoms of a tight Gap B for high–energy–input applications.
- Reduction of impeller structure failures at the impeller periphery. A reinforced impeller structure had several more

failures before Gap B was increased to over 10 percent.

- When properly combined with Gap A, the correct radial gap eliminates large variations in both static and dynamic axial thrust, which reduces maintenance and significantly increases pump reliability.

- When properly combined with the Gap A modification, Gap B eliminates or greatly reduces hydraulic instability at part–load operation.

Standards for Gap B and for Impeller–Diffuser Vane Number Combinations. Gap B fully controls the magnitudes of vibration and pressure pulsation at the vane–passing frequency, its direct harmonics, and its combinations with the diffuser/volute vanes. Although this is a very important dimension in a high energy pump, design standards have not been established for Gap B. In previous EPRI publications, a three percent minimum gap was recommended for reasonable operation (References 2 and 3). Later publications called for at least four percent, but preferably six percent or more. This does not contradict the previous guidelines (References 1, 2, 3,). As investigations have continued, field experience showed that for high–energy–input feed pumps, a greater gap is needed than was previously believed. Figure 25 of Reference 1 states that six percent is preferred and warns against even–even combinations of impeller and diffuser vanes. This warning was also published in Reference 2 and 3.

In addition to the established design parameters such as head per stage, flow, speed, tip–speed (or diameter), etc., a pump designer should seriously consider one more parameter; namely, the combination of the number of impeller and diffuser/volute vanes. It is shown in Figure 21 that designers have managed to apply virtually all possible combinations of vane for high energy pumps, thus presenting additional field complications.

As the number of unexplained cases of high vibration or pressure pulsation amplitudes has increased along with repeated field failures, the vane number combination also has come into the picture. Warnings against even–even combinations of impeller and diffuser vanes were given in References 2 and 3. An unconventional nine times rpm component is shown in Figure 22, and others are shown in Figure 21, such as vane combinations of 2–4, 6–9, 6–12, 9–12, and others.

The vane combinations effects present some difficulties in creating standards for Gap B dimensions. The EPRI guidelines were prepared assuming normal designs. For a 5–2 or 5–8 vane combination, only one impeller vane creates a pressure pulse at an instant of time. With a 6–12 combination, six impeller vanes pass a diffuser vane simultaneously. Such combinations have been solved reasonably successfully by increasing Gap B to unusually large dimensions. In some cases, a gap of over 20 percent gap was necessary. In such cases, Gap A, Gap B, and the Overlap combination had to be optimized to maintain pump efficiency.

Interaction between Radial Gaps A and B, and the "Overlap". The design range for effective application of the radial Gap A and its Overlap is limited by the specific speed and suction specific speed of utility pumps with high energy input. The effectiveness of the radial Gap B applies to any centrifugal pump design, and to all N_s and S_s values used in utility pumps. The combination "fix" for boiler feed pumps and other high energy feed pumps has become a powerful design modification to eliminate failures and to control various system problems that were previously not connected to pump component design. This has also shown to be the least expensive modification in most instances. Rotors must be periodically repaired and overhauled especially if a fundamentally hydraulic design deficiency causes failures or requires high maintenance. With a small additional effort, a corrective modification can be performed during routine maintenance to eliminate the root cause of high maintenance. A routine overhaul without modifying the faulty hydraulic components leaves the opportunity for another failure and for more maintenance than required.

Because Gaps A and B both participate in related fluid mechanical actions in regions between the impeller exit and the diffuser/volute inlet, the two geometries cannot be considered independently. The interaction between the effects of these gaps is very strong and distinct, and the results apply equally to diffuser and volute pumps.

Single–Stage Double–Suction Reactor Feed Pump. During start–up of an 1100 MW BWR nuclear unit, feedwater system instability and high levels of radial and axial vibration occurred in a reactor feed pump at below 60% of pump BEP. The plant required stable operating capabilities down to 46% BEP (at reduced pump speed). The pumps are S.S., D.S., diffuser–type with very high energy input. An intensive investigation was started by testing one pump with its controversial suction piping configuration at the OEM's elaborate testing facilities with a well instrumented test program. Test No. 1 was performed with the original full diameter impeller, and is not shown in Figure 23. Test No. 2 was the "bench mark" test, for which the impeller vanes were modified during start-up, as shown in Figure 24. Test 2 was performed without modification of the impeller-exit to diffuser-inlet dimensions; these were changed for Test 3.

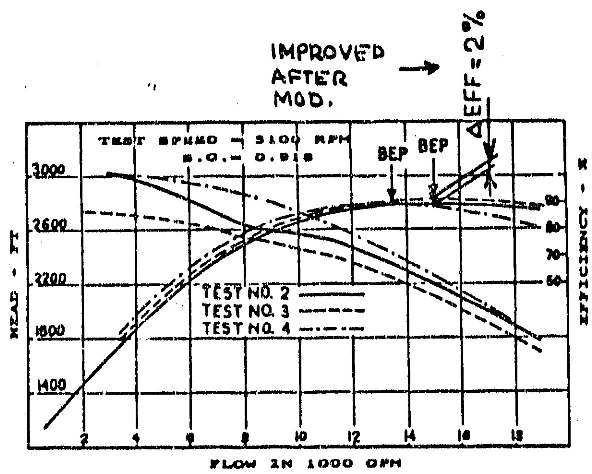
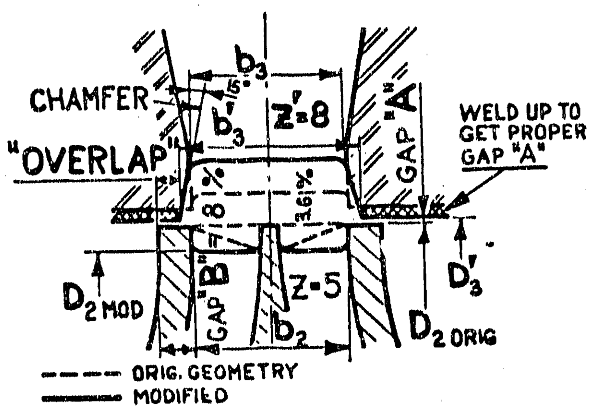


Figure 23. Nuclear reactor feed pump test data before and after impeller-diffuser modifications (efficiency improved by 2%).



PECO: LIMERICK 1

Figure 24. Nuclear reactor feed pump stage modifications.

The Gap A overlap was only partially correct during Test 3, because the tested impeller (a) had thinner-than-desired and uneven wall thickness, (b) had a large amount of balance weight removed from the sideplates up to the O.D., and (c) had to be the same as for Test 2 to properly compare results. A new impeller and diffuser were designed and cast with special emphasis on low-flow stability. Sideplates were made wider at the O.D., and balance weight removal did not interfere with uniformity of the "Overlap".

Full performance, vibration, and pressure pulsation tests were made between 130% and 25% of BEP flow at full speed (5100 rpm) and at a reduced speed (4500 rpm) down to 15% flow. The head-capacity curve rose steadily toward low flows, and was stable for both Test 3 and 4, as shown in Figure 23. Reference 13 describes more details. The stability criterion was met at 35% flow with Test 3, and 25% flow with Test 4 at full speed. The change in shaft axial vibration was similar to the results shown in Figures 25-a and 25-b. There was a substantial improvement in rotor damping. Pump configurations are shown in Figures 26 and 27. Table 3 provides the parameters that were varied to find the optimum performance and the actual data from the testing of a boiler feed pump. Figures 28 through 38 provide additional examples.

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3. E. Makay, "Eliminating Pump-Stability Problems." *Power, Vol. 114*, No. 7, pp. 62-63, July 1970.

4. E. Makay and O. Szamody, "Survey of Feed Pump Outages," EPRI Report No. FP-754, April 1978, Survey of ~1500 Feed Pumps.

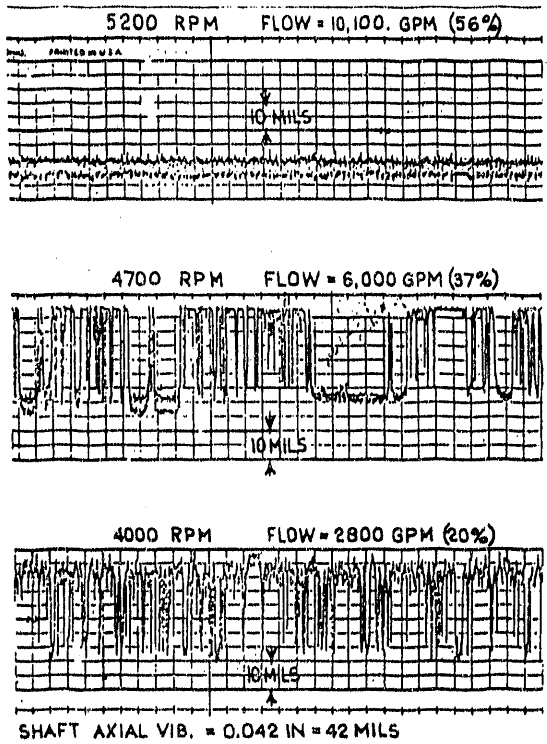

Figure 25a. Low frequency axial shuttling of high speed S.S., D.S. feed pump rotor. Vibration levels normal at "BEP" with the thrust bearing setting of 12 to 14 mils, shaft movement clearly visible below 50% flow. Maximum axial vibration level is 42 mils with reduced speed.

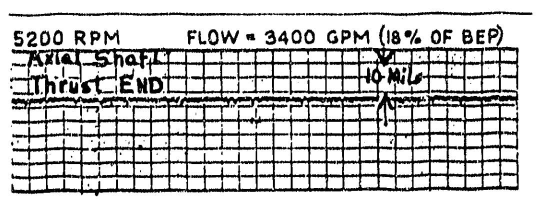

Figure 25b. Axial vibration of pump greatly reduced from 42 mils to 5 mils maximum (at full speed) after the impeller exit to diffuser inlet geometry redesigned with Gap "A" below 50 mils and Gap "B" over 20% radial gap.

Figure 25. Axial vibration of rotor before and after Gap "A" modification (see Figures 23 and 24) (Reference 2).

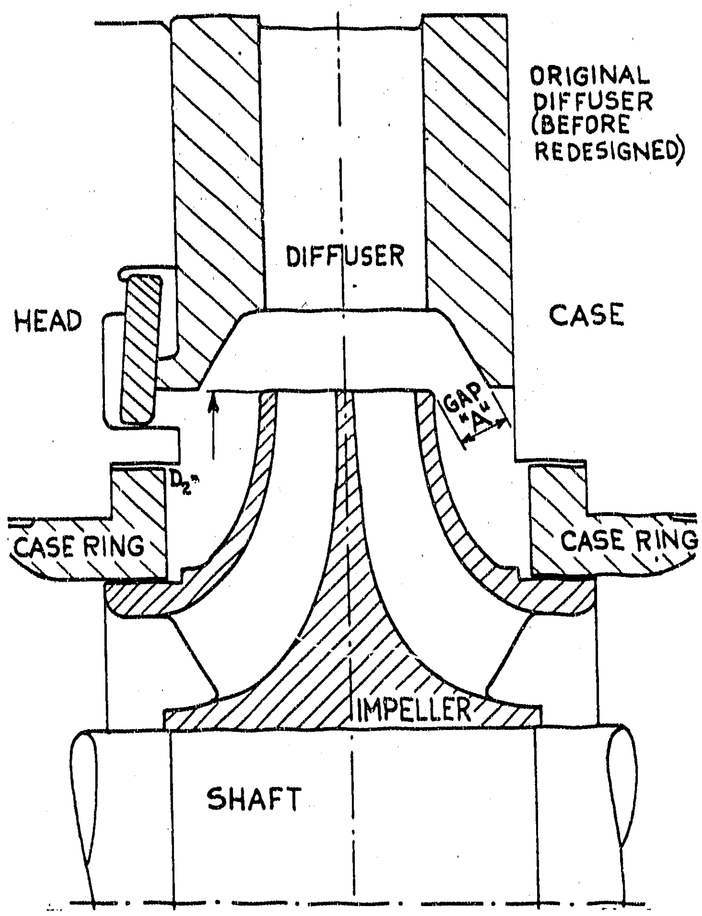

Figure 26. High speed reactor feed pump before diffuser was redesigned. Axial vibration of rotor was 42 mils at 26% of BEP at reduced speed of 4,000 rpm. (Reference 2)

5. E. Makay and O. Szamody, "Recommended Design Guidelines for Feedwater Pumps in Large Power Generating Units," EPRI Report No. CS-1512, Sept. 1980.

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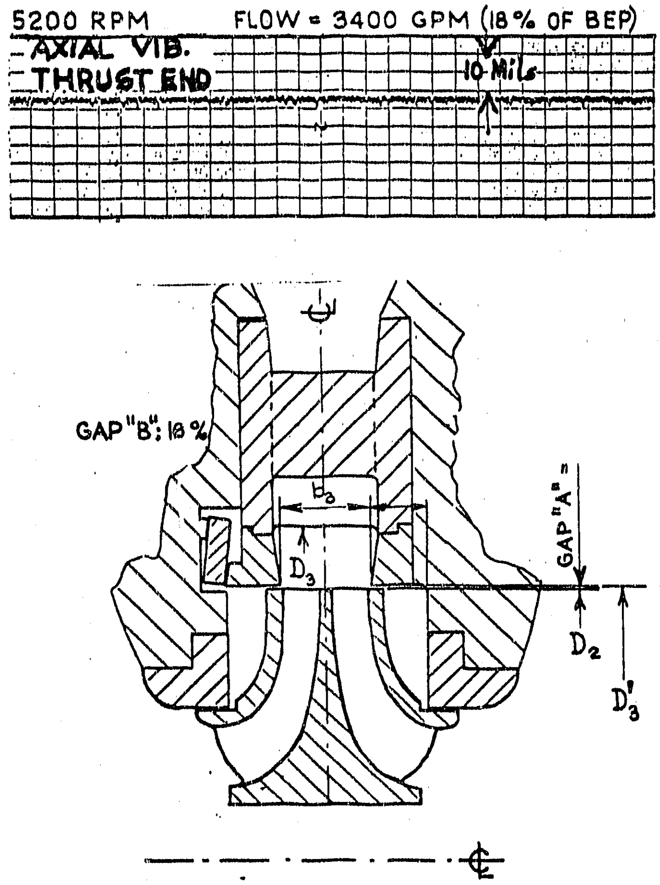
12. I. Massey, "Subsynchronous Vibration Problems in High Speed Multistage Centrifugal Pumps," 14th Turbomachinery Symposium, Texas A&M University, 1985.



AXIAL VIBRATION OF A HIGH SPEED NUCLEAR FEED PUMP GREATLY REDUCES (FROM 42MILS TO 2 MILS MAX.) AT 18% OF BEP MINIMUM FLOW AFTER THE DIFFUSER INLET GEOMETRY WAS REDESIGNED WITH GAP "A" = 35 MILS.

Figure 27. Gap "A" modification geometry and test results after modifications at 18% of BEP flow (Reference 2).

13. P. Cooper, "Roto–Dynamic Pumps with a Backflow Recirculator," U.S. Patent No. 4,375,937, assigned to Ingersoll–Rand, Issued March 8, 1983.

14. E. Mackay: In Service Testing of the Palo Verde Nuclear Aux. F. W. Pumps in the Normal, Minimum and Extreme Run–Out (LOCA) Flow Modes. ERCO Rpt. No. 10–756, March 28, 1985.

15. M. L. Adams and E. Makay: Aging and Service Wear of Aux. F.W. Pumps for PWR Nuclear Power Plants. NUREG/CR–4597, Vol. 1 (ORNL–6193/V1), July 1986.

16. E. Makay: Performance Behavior of the Aux. F. W. Pumps under Extreme Conditions with a Main Steam Line Break at the Rancho Seco Nuclear Power Station. ERCO Rpt. No. 10–693, February 16, 1984.

17. E. Makay: Connecticut Yankee Charging Pump Critical Speed Review. ERCO Rpt. No. 70–08–004. May 1973.

18. US NRC Bulletin No. 88–04, Potential Safety–Related Pump Loss. May 5, 1988.

Table 3. Parameters tested and actual data from a boiler feed pump (March 3, 1988)

Parameters Varied to Find Optimum BFP Performance
(ranges tested in mm)

Gap A	Gap B (%)	Gap E & F	L/A	b_3/b_2
0.035 – 0.9	0.5	0.07 – 1.8	NE	0.9
0.045 – 1.15	4	0.26 – 6.6	2	0.9
0.065 – 1.65	6	0.35 – 9.0	4	0.9
0.085 – 2.16	12	0.50 – 12.7	6	0.9
0.100 – 2.6	20	0.80 – 20	8	1.5
0.165 – 4.0			16	

BFP Tested with Dimensions, Parameters in Ranges
(min–max)

D_2	b_2	HP	RPM	N_s
14.2 – 360	0.8 – 20	1,000	3,480	1100 – 21
15.5 – 394	1.25 – 32	1,000	3,480	1100 – 21
16.5 – 420	1.4 – 36	1,000	3,480	1100 – 21
18.5 – 470	1.625 – 41	1,000	3,480	1100 – 21
21 – 533	1.75 – 45	40,000	9,000	1800 – 35

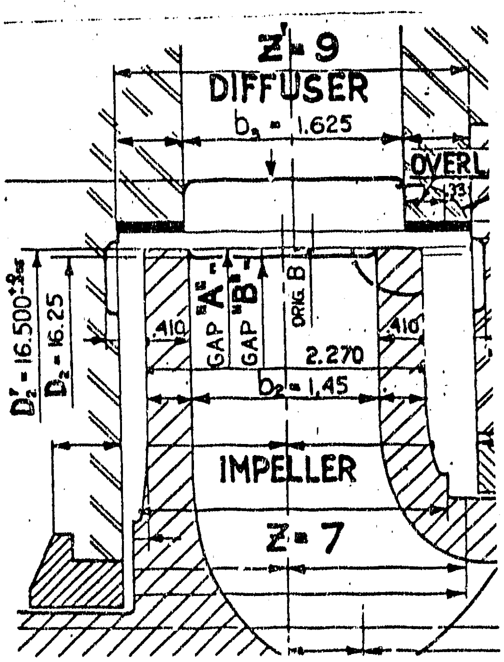


Figure 28. Misaligned rotor toward outboard. Gap "A" ineffective on suction side. Thrust reversal at low flows. Thrust bearing will fail on the unloaded side.

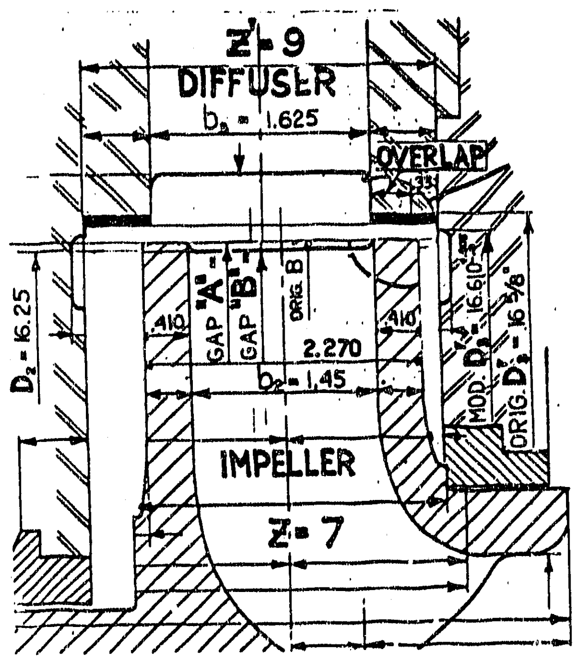


Figure 29. Misaligned rotor toward inboard. Gap "A" ineffective on discharge side. Drastic increase of axial thrust. Thrust bearing will fail on the active side.

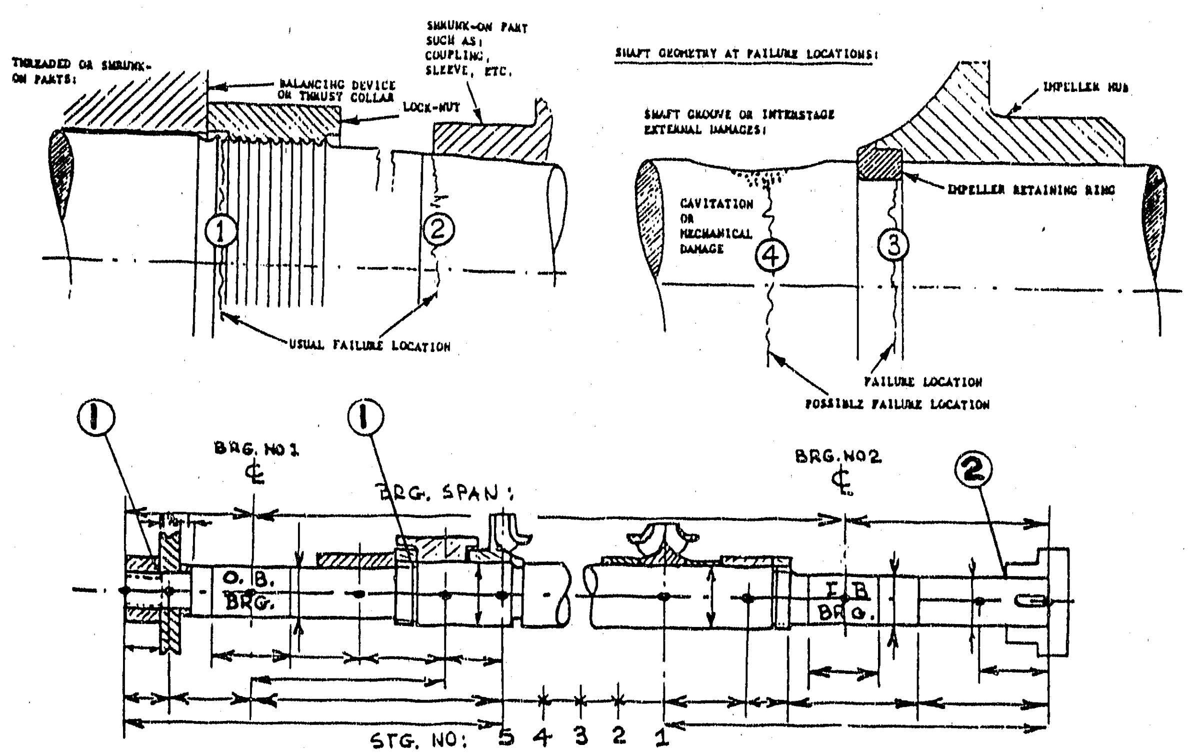
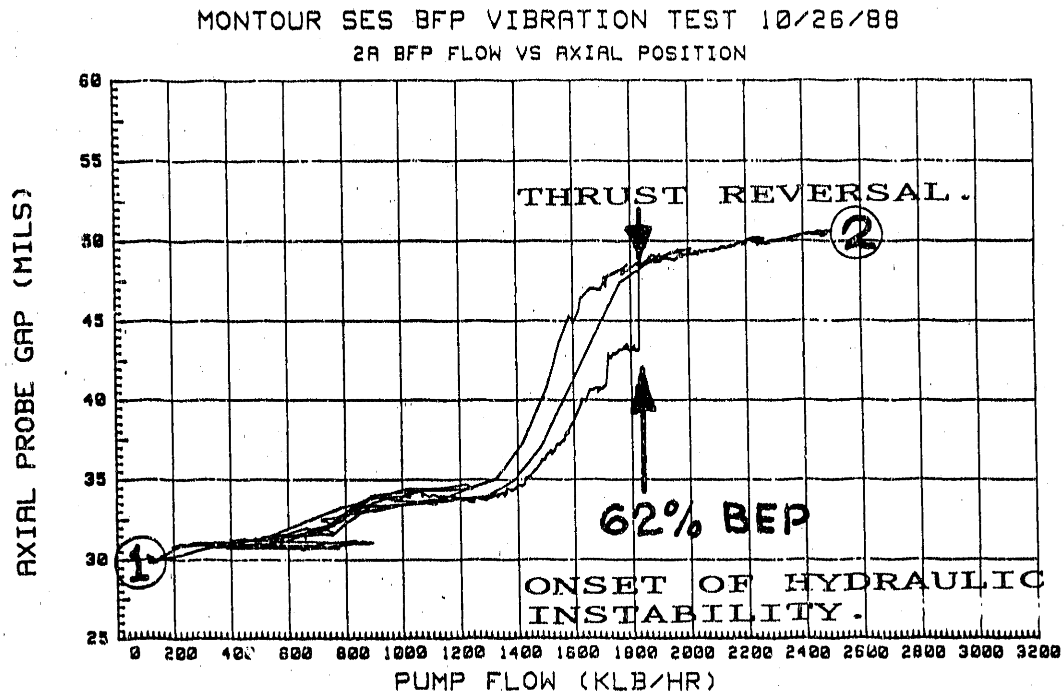
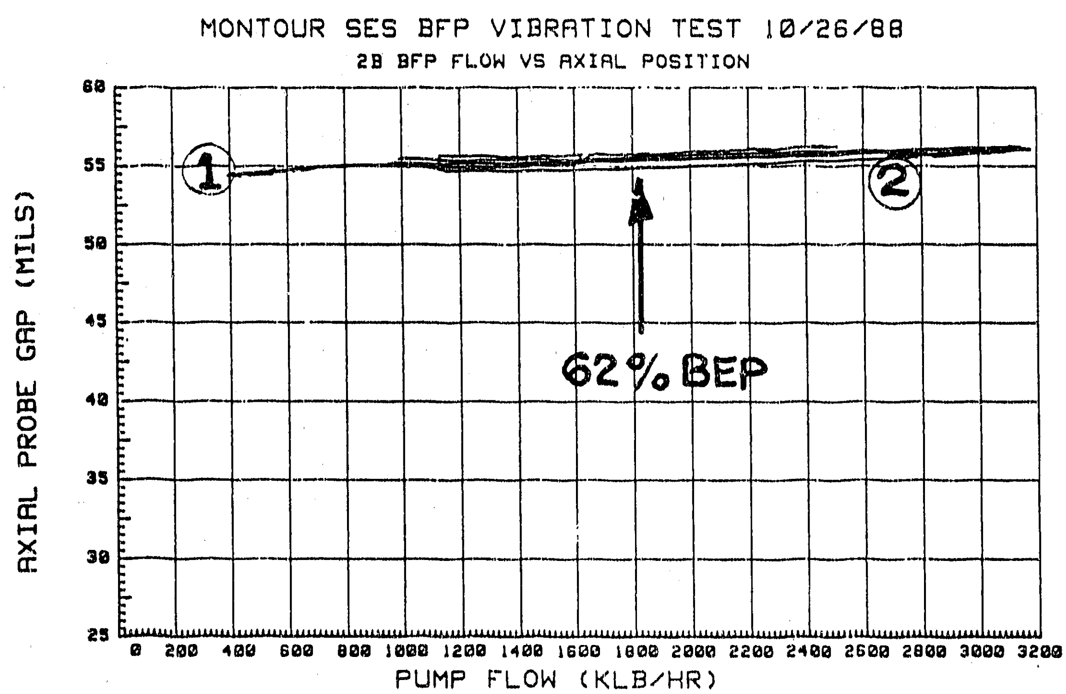


Figure 30. Typical multistage centrifugal pump shaft failure locations. Most frequent locations are in order of failure frequency.

Figure 31. Typical appearance of a vibration fatigue shaft break of a boiler feed pump (location 1).



14 WNC-156 TYPE BOILER FEED PUMP AXIAL POSITION BEFORE GAP "A" AND "B" MODIFICATION. SHAFT MOVES TOWARD OUTBOARD.



AXIAL THRUST STABILIZED AFTER GAP "A" AND "B" MODIFICATION INTRODUCED.

Figure 32. Boiler feed pump axial probe gap at 62% BEP before and after Gap "A" and "B" modification.

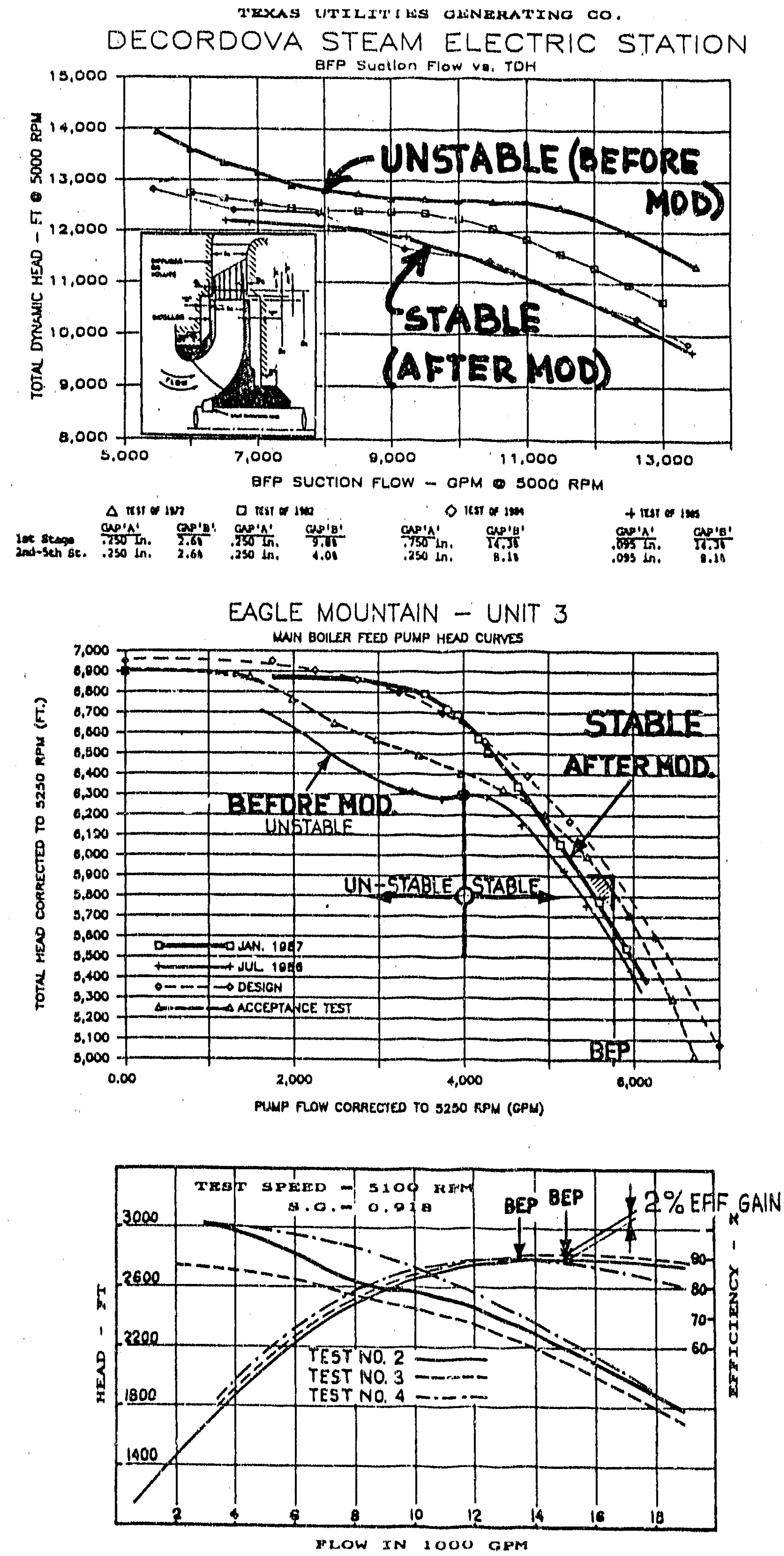


Figure 33. Nuclear reactor feed pump test data before and after impeller–diffuser modifications. Test No. 1: original factory acceptance test (not shown). Test No. 2: pump retested. Test No. 3: Gap "A" and "B" modifications. Test No. 5: New impeller design for low flow stability and all modifications of test No. 3.

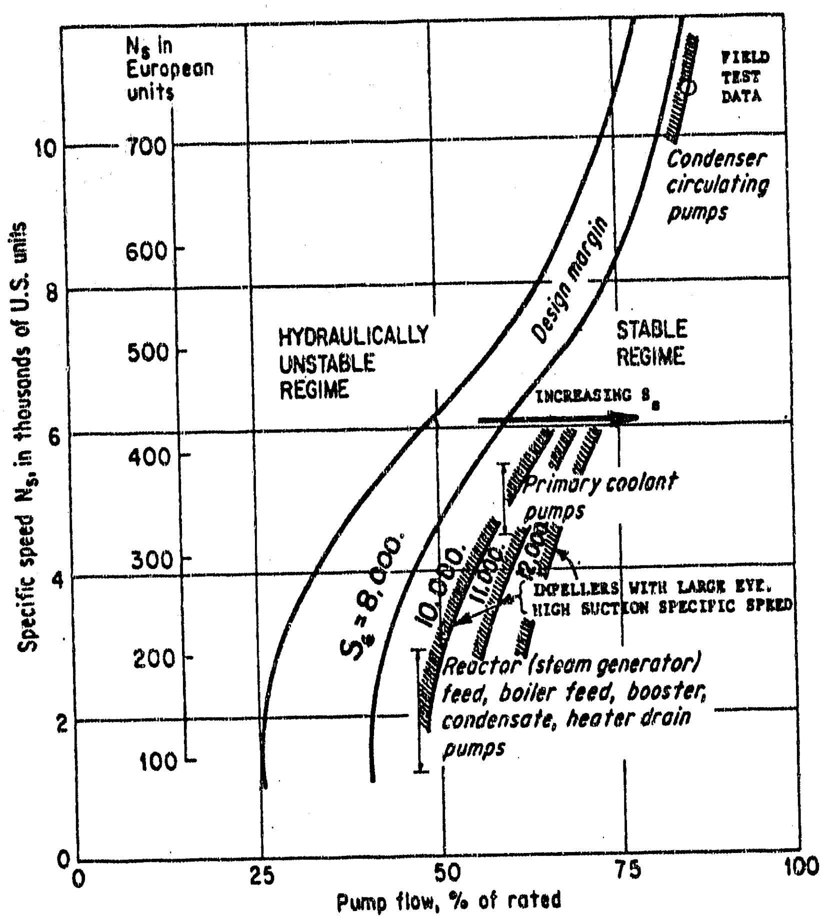


Figure 34. Anticipated useful operating ranges of centrifugal pumps used in large nuclear and fossil power generating units as a function of specific speed (impeller shape), and suction specific speed (NPSH).

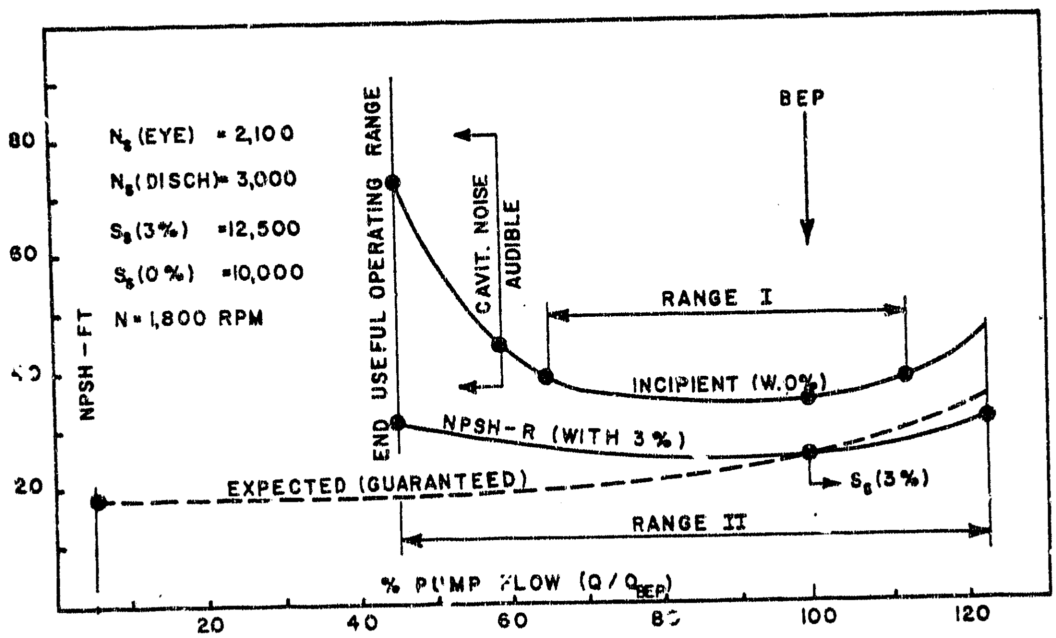


Figure 35. Guaranteed–vs–tested NPSH–REG. for a single–stage, double–suction, service water pump in a 1,100 MW PWR nuclear power station.

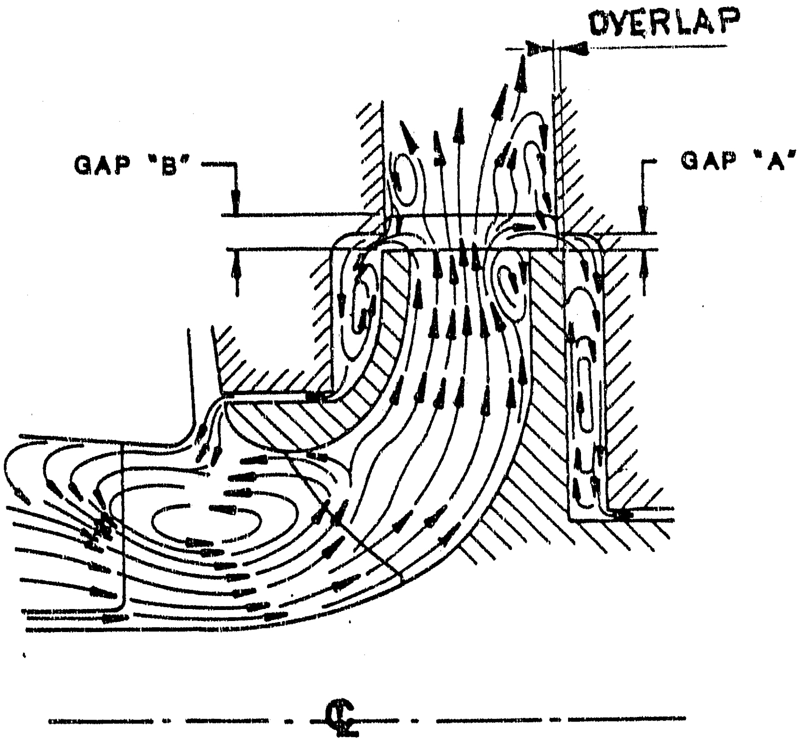


Figure 36. Recirculating flow pattern in the meridional plane at part load (at or below the onset of hydraulic instability).

THIS DRWG. IS TO SCALE (x 0.8 reduction) 1-10-80
MADE AS AN IMPRINT FROM THE ACTUAL DIFF. E.MAKAY

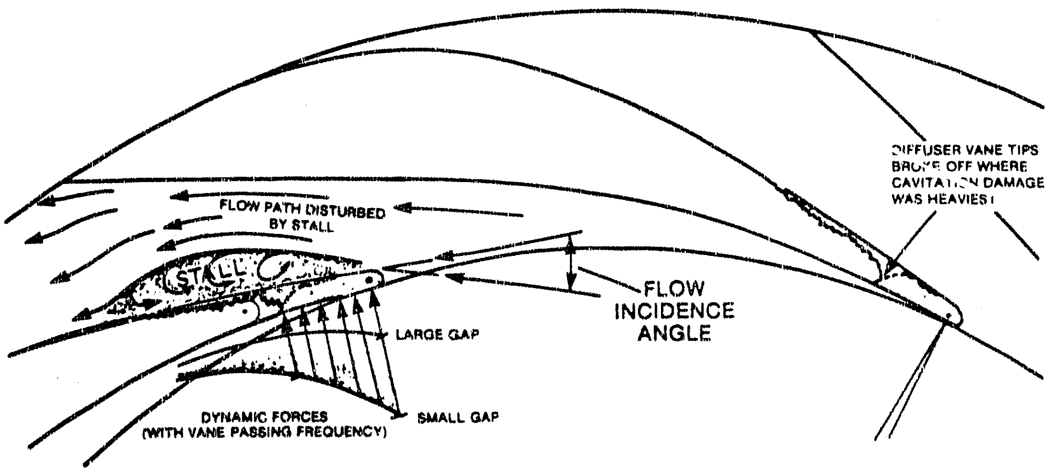


Figure 37. Origin of failure mechanism: flow incidence angle induced stall. Four identical failures at PECO's Eddystone PW Station between 1976 and 1978. No failure since diffuser vanes cut back to the point of breakage. Two similar failures at NIPCO's Schahfer No. 15 Station. Vanes broke at the same location. Eighteen identical failures at other stations. Same modification remedied all cases. (For more details, see EPRI Report FP-754, pp. 4–5 and Figure 10.)

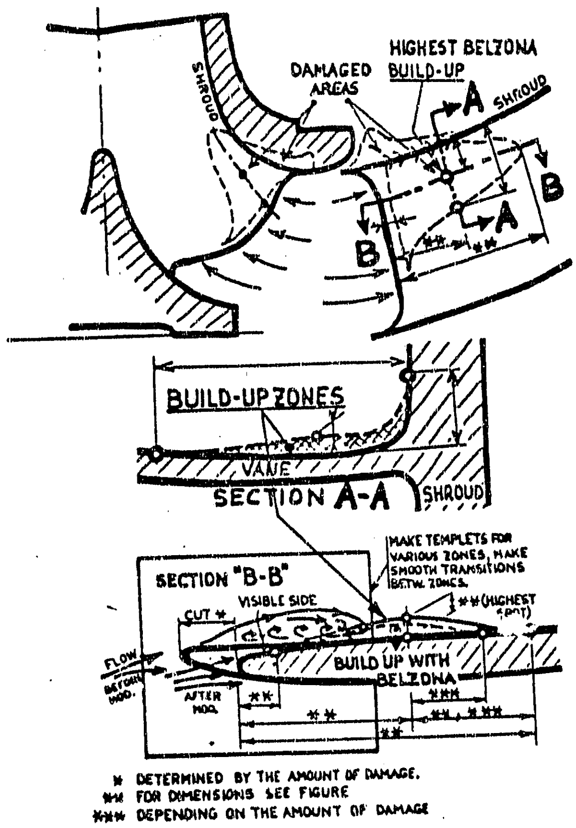


Figure 38. Application of the "anti–stall hump" to prevent cavitation damage at the impeller inlet (from Reference 1). To date, it has been applied to 58 condenser (cooling tower) pumps by the author with success (March 28, 1989).

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

No Questions/Comments

PANEL DISCUSSION: IST OF PUMPS; EXPERIENCE AND LESSONS LEARNED

T. HOYLE MODERATOR

PANEL MEMBERS: J. HEALY, J. MAXWELL, W. GREENSTREET, AND E. MAKAY

Question 1 (Mini–Flow on Pumps)

It has been strongly suggested that testing at low flow will cause pump failures. The examples of some pump failures caused by low flow operation were for service water type pumps, which run a lot.

My understanding of the phenomenon is that it is a long–term degradation mechanism. Contesting a pump for 30 minutes every three months at low flow; will that result in failure or damage to the pump? This only adds up to 80 hours of run time in 40 years.

Response 1

Panelist There are many replies along that vein to 88–04; that is, the pump can run a certain period to mini flow. One has to look at that in light of current knowledge.

It's very difficult to say how long a pump can operate in mini flow because of the forces and things that are there and that are not there.

How can you consider an isolation operation at mini flow when your pump is degraded but not to the point of failure at any flow that you have.

A great deal of failure is caused by fatigue, and fatigue is something that happens slowly. For example, if you consider something in fatigue you have a crack initiation phase and a crack propagation phase. Once you have a crack you can see that there's a problem. You can measure the crack points or whatever.

But up to that point, which is a major portion of life, you have no way to know, really, how much damage has occurred. So, you may be right that your pump will operate that period of time, but you may be wrong if you haven't got the history right, and you could have failure within that small amount of time at mini flow. So it's a difficult question to answer.

Question 2 (Pump Testing)

It was stated that pump degradations can be corrected by heat exchanger performance. It was also stated that heat exchangers can't be tested. They can be tested, and we do test them. Can heat exchanger can improve 'egradation of a pump?

Response 2

Panelist Testing a heat exchanger for flow capacity is one thing. My experience has been that without a true measurement of what the heat transfer coefficient on the UA of a heat exchanger is, there may be some assumptions based upon a safety analysis that, to perform a certain duty of rejecting heat, you need a certain amount of flow.

You really don't know whether or not even without a certain amount of fouling factor, a certain percentage of tubes plugged—whether you really need 10,000 or 8,000 or 6,000 gallons a minute.

I worked with a number of specialists at Stone & Webster in Boston, and one of my associates is a heat exchanger specialist. He's been in the industry for 30 years and very glibly and very candidly relates to me stories that, typically, manufacturers of heat exchangers really don't know how many tubes they can lose or what the true UA of a heat exchanger is.

Indeed, to ensure margin there's a tendency to overload the surface area to ensure that, if it was a contractual problem later on, they wouldn't be caught short. In a way, this is what pump manufacturers do. We have a design flow and a design head.

A manufacturer rarely designs an impeller or trims an impeller to get that point right on the money. Whether it be the hydraulic institute standards for pumps or HTRI for heat exchangers, there is always the tolerance above the design guarantee point before that pump or heat exchanger can be accepted.

It's been my experience that we know a lot more about the magnitude of that tolerance on pumps, and we happen to know very little at all about the magnitude of tolerance on heat exchangers.

Based upon what I know to be the sizing criteria, however, I suspect that if we are able to do more field testing or shop testing, indeed we might find that the heat exchanger to reject a certain heat load– which is our objective, obviously—to pass so many million Btus an hour, that we might not need the 10,000 gallons. We might be able to get away, let's say, with 8,000 gallons a minute by looking at it that way.

If the pump is only sized for 10,000 gallons a minute or slightly larger than that, it's like taking money out of one pocket and putting it in the other. Instead of having real pump margin, what you might find yourself living with is system margin, which gets back to the thrust of OM-20 and other aspects of what we've been talking about the last couple of days or so.

The margin can exist in several places. It can either exist in the control valve type of system, pump or heat exchanger. The focus of what we've been talking about, is looking at all the margin that exists on a pump and finding in some cases that it's wanting.

I think if we find systems that are wanting, and we're looking for greater margins, we should look in the areas that we haven't looked at in the past.

So, it's a proposal based upon knowing how heat exchangers are designed, and it's a challenge for those who think they have a problem to go back to find out whether or not it's 10, 15, or 20%. But I believe you'll find a lot more than you'll ever find in pumps.

Question 3 (Design of Mini-Flow Pumps)

It has been stated that our pumps are failing because they weren't designed to be run on standby and testing in mini flow.

I believe purchase specifications stated these conditions when the pumps were bought. Why weren't they designed to these conditions? Isn't it possible to design a pump to normally run at low flow and yet have high flow capabilities?

Response 3

Panelist I agree. I'm partly in the realm, perhaps as other people, of writing these design specifications.

But if we take ourselves back 15 and 20 years, we find that there might have been lines in our data sheets or our specifications asking for minimum flow. We didn't then know, and certainly the manufacturers were no smarter than us, as to what that meant.

Typically, in a multi-pump system when you go to an Ingersol Rand or Byron Jackson—and I'm not singling out any manufacturers, even though I can see some of them in the audience—what they were looking at is typically what they would expect to see in a cogeneration plant or a municipal obligation where if there was a two-pump system and you had a problem with the A pump, you'd be expected to start the B pump while you were powering down the A pump, a transition which might take all of five or ten minutes.

No one knew that we were going to be testing nuclear plants for weeks, months, and sometimes years. I can cite Stone & Webster's experience at Shoreham. At least a decade's worth of testing has been applied to some of these pumps without any service whatsoever.

I think when you look at the amount of time spent at minimum flow, no one could have imagined what that could mean. As pumps in the nuclear plants got bigger and bigger, the high-energy density levels spoken about—whereby we're exposing these impellers and shafts, sometimes very limber shafts, very high radial deflections—were unanticipated, because in the past in the '50s and '60s we never accumulated any running time on top of this.

And the further enhancement of this argument I'd like to propose is that no one ever expected to qualify a pump on minimum flow: (a) we weren't expected to run at min flow, and (b) we weren't expected to qualify. You buy a pump for 10,000 gallons a minute at 2,000 feet a head, such as a reactor. That's the point that you demonstrate before the pump is shipped.

I think we've been spending a lot of time looking at the problems at the low end and trying to qualify things that no one ever imagined.

Can we redesign these pumps? The answer is yes in many instances. It's more difficult to redesign a 14-stage split case auxiliary feedwater pump than a three. But within certain parameters, if we can communicate the need to demonstrate, or the need to spend 90% of the pump's life at 20% flow, there's a chance that we can go back and ensure that not only mechanically the pump doesn't destroy itself or degrade itself, but also we can have greater confidence that the pump is performing as well as it should there, as well as at design flows which are higher than the minimum flow.

I hope that addresses the issues. I think we've learned a lot in the last 15 to 20 years.

Question 4 (Vibration Data on Vertical Pumps)

It was stated yesterday that vibration data on vertical pumps is near useless. Do you agree?

Could the value of the vibration data be increased by taking shaft proximity probe data in addition or instead of casing vibration?

Response 4

Panelist For pumps where the impeller is 20 or 30 feet below the upper bearing, you can wreck the pump without being able to tell anything in the vibration. The only way to detect problems there is to put probes in the water. That has been done for a certain amount of testing, but it is really difficult to do.

You can pick up some failures by taking data at the top of the motor or as close down to the stuffing box as you can get, but it is really hard to tell what's going on down on a pump in that situation.

Panelist It's been my experience that vertical wet pit pumps that we've been talking about, or deep draft vertical—there may be a rare exception—but it's been my experience very few of them spend any amount of time in minimum flow.

If you are looking at a safety-related service water pump that is normally in service, the minimum flow happens to be a turndown of about 2 to 1. The pump is designed for 16,000 gallons a minute. It might come down to 8,000. So, that is a rare problem for a vertical pump, if it is in either normal operation or even in standby where it spends all of its time at or near design flow.

Another interesting thing is that we also look at a number of ranges we've been talking about. The OM–6 standard talks about pumps operating at less than 600 rpm. Again, unless there's a rare example, I don't know of an end-stamped or safety-related piece of equipment that runs at 600 rpm. Ninety-nine point nine percent of them run between 1,800 and 5,000 rpm. So, in a way, that kind of limits the field of concern that we're talking about when it comes to vibration also. We're looking at a narrow class of typically horizontal pumps, either single or multiple stage, an occasional one– or two–stage vertical that might only see service at test line capacity.

Panelist By the way, the slow-speed pumps are generally the reciprocating pumps, and there are a number of those.

Question 5 (Accelerometer Versus Velocity Transducers)

Could you comment on accelerometer versus velocity transducers for casing measurements—advantages and disadvantages?

Response 5

Panelist Generally, I prefer accelerometers. Velocity transducers have the advantage that you only integrate once if you are going to measure mils, and for low frequencies that's generally helpful. But generally, accelerometers go to a lower frequency range. They have no moving parts and tend to be more reliable for that reason, and in a lot of cases they're smaller. To try to get a small velocity transducer, you wind up only being able to use it in the horizontal direction or vertical direction because of the spring loading. So generally I prefer accelerometers.

Question 6 (OM–6 Instrument Calibration)

In your opinion, (a) is the requirement in OM–6 to have the instrumentation calibrated to one–third of the lowest running speed with the specified accuracy really necessary, and (b) would a range of from 10 to 100 Hz be more realistic in view of the usual calibration facility capability?

Response 6

Panelist I'm not sure that I can answer a question about what the philosophy is behind OM–6. When you specify the accuracy of a vibration transducer, a vibration monitoring system, you can't just say I want 5% accuracy. You have to specify half a dozen different numbers, including frequency response. What do you mean by frequency response?

Generally in the electronic industry, frequency response is measured by 3 db points. You can have 3 db on voltage or you can have 3 db on power, which turns out to be on voltage if you have a 3 db power, a 60 db voltage; that's a 2 to 1 or 50% decrease. So I'm not sure if they want 5% across the entire frequency range, or they want 5% at the calibration frequency and 3 db across the entire frequency range. It's a problem.

It is possible to measure very low frequencies accurately, but it's not easy. You have to use seismic transducers. They get to be big transducers or force voltage type transducers. It's difficult. The National Bureau of Standards only calibrates down to 5 Hz, so—if you are going to calibrate lower than that—you have to do something exotic in your calibration lab. So, it gets to be a real problem and I'm not sure whether the Committee would like to address whether OM–6 has thought of that or what they're planning on.

Panelist I want to comment on the "down to 10 Hz" comment.

As you can see on this chart, that item 5, which goes up to 10 Hz, turned out to be the biggest headache in the large nuclear pumps and boiler feed pumps. When most people take measurements, they go down to 60 cpm, 10 Hz, and they don't measure below that. That sometimes turns out to be the most important barometer in troubleshooting that pump. I want to kind of caution those people who take measurements down to only 10 Hz, because frequencies below 10 Hz may also indicate problems with that pump, and if you only go down to 10 Hz you don't see the problem.

Panelist Let me just give a little bit of background. The reason for the one-third of the lowest running speed is that we really wanted to pick up the one-half times running speed, and to ensure that we had that we went to one-third.

Generally, we had never discussed the problems in the working group of what happens on some of these reciprocal pumps where you get very low speeds. So, that's an item that we have on our agenda to relook at. We're reformulating the task group on vibration, and we're hoping to look at some of these areas, as well as the one that we discussed yesterday that was also on our agenda, should we have a minimum that you have to worry about. If you have a 0.01 or can you just say anything below that you are not going to worry about.

We're going to be looking at more of those in detail in the working group in the next months.

Question 7 (Instrument Accuracy)

As OM-6 requires the total loop accuracy of digital vibration instrumentation to be 5% over the calibrated range, in your opinion is this an absolute accuracy requirement? Can it be met?

Response 7

Panelist Let me answer the question of can it be met. Yes, it can be met. It is not easy. It is very hard.

Is it necessary? I don't think it's necessary. Especially, if you are using relative standards and looking for a 2.5 change or a six times change, and you are using an instrument that has a good repeatability or a single design of instrument where all the transducers have the same low-frequency roll-off characteristics and the meters have the same low frequency roll-off characteristics. You can still get very good data at low frequencies.

The example I gave earlier in my paper of that charging pump—that's a 3.3 Hz pump—and we were able to get good data there, and good data at 10 Hz. Even though we may not be getting 5% accuracy, you can still tell what's going on with the pump if you are careful and pay attention.

So, I'm not sure that it's necessary to maintain the 5%, but you have to be careful that you are not using a transducer that rolls off so fast you are not going to pick up any of those low frequencies.

Question 8 (Instrumentation of Low Flow Pumps)

The question is concerning low flow rate pumps, in particular diesel generator fuel oil transfer pumps. These pumps are usually overdesigned to begin with, 300 to 400% pumps gpm.

With a flow rate at 20 to 22 gpm, slight instrument inaccuracies can cause the test data to fall outside the acceptable and alert range.

What is the best way to handle this? A relief request? If so, what range is acceptable?

Response 8

Panelist Again, I think we're looking at an unusual application. Diesel generator fuel oil transfer pumps happen to be typically rotary positive displacement pumps, and I think you've got yourself a Catch-22.

If they are centrifugal, then I would wonder why the reference flow is as high as it is if the pump is 400% its rating. I've run across both kinds of pumps. If they're centrifugal and the pump is rated, if the design flow of the pump is 22 gallons a minute and you only need 5 or 6, why are we demonstrating capability at 22 gallons?

I'd like to have the questioner clarify this because I've run across the other style of pump, which is a PD. Positive displacement, of course, is a function of speed. If the motor runs at one speed, then you are forced to run at 22 gallons per minute even though you only need 5 or 6. So any small perturbation of back pressure or lift, or whatever it happens to be with the fluid that day, can cause you to get a blip in data.

Attendee It's a positive displacement pump.

Panelist That's the problem, then. It would appear that a relief request would be the way to go. I can't imagine the argument being anything other than accepting it if the pump is three to four times the size. Why you would have to be demonstrating either 20, 21, or 22 gallons a minute so consistently that you couldn't avoid a deviation outside that window, especially if the pump is three to four times the size it need be?

Attendee We can avoid the deviation, but I guess I should have saved that question for later this afternoon when we have the NRC and say what is an acceptable range.

When you are talking 22 gpm, you don't have much to play with.

Panelist I think this question is really good not only for positive displacement but for any low head pump. It really has to do with where you start having a fairly low differential pressure across a pump. You are talking about very small amounts. If you have a DP of 25 psi, you take a few percent of that and that's not much at all. You start talking in the range of 0.5, 0.3 as being how much over you can go for an acceptance criteria.

That's always going to be a concern, and I think you have a couple of options. One is to install better test instrumentation, and two is the relief request.

Attendee (NRC) This is a difficult issue to deal with. We have had a number of relief requests that come in along these lines. One thing that makes it difficult is that we almost never have articulated very well what is actually going on. Why is it that the ranges cannot be met? That puts us in an impossible situation, because we can't process a relief request if we don't know physically what the reason is that the test is behaving the way it is. Irrespective of that, it meets the definition. The other question is if you know what's going on, we need to know why the test is adequate for demonstrating the condition of the pump.

What I'd like to avoid is a situation where we say that the solution is through a relief request without us having the opportunity to know what's going on. We need to know why the test that is being done is accomplishing the intended objectives.

I personally don't think just saying that there's a wide margin between what the system needs and what the pump can deliver is the answer, because that doesn't tell me what the condition of the pump is.

Question 9 (Low Energy Pumps)

Most of your examples refer to high/energy multi-stage pumps, such as reactor coolant or a steam generator feed pumps. However, safety-related nuclear pumps are normally moderate performance pumps.

Do you agree, and would you discuss this point?

Response 9

Panelist Yes, you are right. But remember, at the beginning I said the reason I bring in the high-energy input pump is because we conclude clearly on those pumps what the problems are. So, when you look at the low-energy inputs or lower-energy input pump, it's very difficult to pin down what the problem is if you develop a problem. The low-flow and low-frequency problem does come up with low-energy input pumps. Also, as a matter of fact, the low-energy input pumps have one additional vibration component there, same as the zero-to-10-Hz but generated at the impeller eye. That we don't have at least in the large-energy input boiler feed pumps, but it's easier to judge if you look at, 1,500 large energy input high-speed boiler feed pumps and conclude what breaks the shaft or what those vibration components come from. Then you go to low-energy input pumps, and instead of being in the dark and trying to outguess what the problem is, your conclusion is based on the large energy input pumps. But all those components are present in those pumps.

Now most of the zero-to-10-Hz components are not noticed on those low-energy inputs. They are not as damaging. Perhaps you can run two, three, four years with that kind of component, while a boiler feed pump would have failed in 10 minutes.

But the biggest problem is that people don't measure it. Your velocity pickup doesn't go down that far. You have the IRD (manufacturer of vibration monitoring equipment) equipment that goes only down to 10 Hz. So you skip that area. It's there. You just don't notice it.

Obviously, you are right. It is not as obvious and is not as damaging, but it is there.

Panelist When we look at boiler feeds or large reactor feeds or steam generated feeds (in a nuclear plant, a reactor feed pump or a steam generated feed is one, or two stages, with energy levels of about 5,000 horsepower per stage) we can't just jump over to a split case or a small, auxiliary feedwater pump or charging pump by just looking at energy density, because there's another factor that creeps into this. That is specific speed.

When you look at an auxiliary feedwater pump which might be a 500–horsepower pump, it might have anywhere between eight to ten stages. So, horsepower per stage is low, but you now introduce the other phenomenon, which is the rotodynamics problem. Supporting a 14–, 10–, or an 8–stage pump with a large bearing span introduces dynamic problems and mechanical problems that the high–energy pump doesn't really have but the three stages or two or even five stages on a fossil plant are more related to what we'd refer to as a stiffer shaft pump, maybe not a stiff shaft pump design.

But some of the problems that are incurred at low–flow work are more readily on a multi–stage pump with a greater bearing span with a smaller shaft diameter per pound of horsepower or per pound of pump, let's say, where it's a lot more sensitive even if the energy level is lower per stage, especially if the bulk of the time the auxiliary feedwater pump is back on minimum flow and run infrequently.

A boiler feed pump that comes up once or twice a year running at or near BEP, or perhaps at night if the cycle comes down, creates one kind of a problem. But I don't believe if you were to put that on a scale it would weigh equal to what happens with a pump that is started monthly, quarterly, or the bearing span might be twice as great as the pump that has one–tenth the horsepower. So, in my experience, it's more than just energy level per stage.

Question 10 (Full Flow Versus Mini–Flow Testing)

Should utilities be designing full flow test lines and getting away from mini flow test lines on Section XI pumps?

Response 10

Panelist From all the evidence that I have found, I would agree that you need to test near design flow. It's not always possible, but certainly you should test above minimum continuous stable flow. I think that's a clear point.

Panelist I'm probably not qualified to speak to this because it is a regulatory position as to whether or not pumps have to be tested at all. But without sounding too comical I'd like us all to remember that pumps, if they're designed correctly, probably won't wear if you don't run them. So, it's not like the '58 Buick that you have in the driveway where if you keep it up on blocks it will last for 20 or 30 years. But if we design a quality product and we build a quality product and we test it in the shop and test it in the field and we never run it, the only concern that I have about pumps is whether or not they will respond to the demand to come up to speed, and that's not a pump problem. It's a problem with a motor or a turbine or an engine, something that pushes or pulls the pump.

If, however, we are convinced that we have to do testing, then to ensure that we have adequate results that can mean something and at the same time don't destroy the very machine that we're trying to protect, then we have to do full–flow testing. But to jump from the need to test, the issue is something that I don't feel that I can quote to.

Question 11 (Low Flow Testing)

Since low–flow pump testing is of little value and can cause damage, do you recommend suspending the testing until full–flow testing can be accomplished?

Response 11

Panelist You should monitor so that you are in control of the situation and it doesn't control you. I would like to see you get away from mini–flow, but I don't think that's going to hold true in every case.

Panelist But concerning mini flow testing, I have to admit, many times when I test pumps out there I get very annoyed by the NRC's request to test the pumps in such extreme conditions.

I have one here at Palo Verde. In this particular case we had to test the minimum flow and we had to show that in case of a LOCA condition the pump would survive. I said let's test the pump for LOCA when we have a LOCA, but don't purposely ruin the pump.

On the other hand, I was very happy that we had to perform this test, because within a year these pumps had two miserable failures, both at Palo Verde and at the Texas South Project, Houston Lighting and Power. If you did not request such extreme conditions, we would not have yet debugged the design. It was clearly a design problem in the pump.

So I'm torn between the two. At once I'm annoyed that I have to run such tests. On the other hand, it debugs the pump.

Now my thought would be—again, I don't represent anyone here; I'm just speaking for myself—once we debug the unit, I would prefer not to run the pumps in those extreme conditions. As a matter of fact, if it were up to me I would say bump test the pump, don't run it for 30 minutes, because what are you trying to do?

The important thing is when we need the pump, we want to make sure that the pump will start up and will pump. But again, I'm no authority. I'm only expressing my opinion. In many cases these extreme conditions did do some good to us by debugging the pump.

Attendee (NRC) I want to clarify first that it's not the NRC's position that plants need to test on minimum flow. That's not our requirement. I think that's a system result. If you are looking at technical specifications that talk about the flow that needs to be exhibited when you are on minimum flow, that is there because plants test in that condition and a value had to be specified, and that's why it's in the technical specifications.

It's not an NRC requirement that plants test in minimum flow. Just the opposite. We would prefer plants do meaningful testing, and if that means full flow, by all means do that. Whatever we can do to entice the Code to go that way, to entice plants to put in full flow loops, do that. In fact, when we review advanced reactors, we're discouraged to see that there isn't any advancement in piping systems. There isn't any addition to full-flow loops. We're in the same kind of configuration that we've always been in, which is the minimum flow. That's the wrong way to go. By all means, design your plants so that you can test them at the best point available.

Panelist If I can be allowed a P.S. on this, I have been privileged to witness the testing of pumps that have been installed for the last 15 to 20 years, and I think the root thought behind the question is whether or not we should suspend this testing. Perhaps I touched on this. I'd like to elaborate a little bit more on my original opening remarks earlier this morning. So unless we treat all centrifugal pumps the same way, then we're going to be surprised that some of them fail. I think we really almost have to treat pumps like people. I can cite some examples: my experience at Ft. Calhoun, where some pumps have been run for 14 years on minimum flow.

Now the only thing that we're uncertain about is whether or not the pumps were able to go to design flow and make design head. Recent testing has demonstrated that the pumps can do that, but the bottom line is that the pumps installed in that plant run as periodically as it has been for the last 17 years, has sustained absolutely no failures. There have been no wearing failures, no shaft failures, no diffuser failures, no bearing failures. If that's the way the pump is designed and running, then it's that pump and that plant that can apparently sustain and tolerate that service.

Panelist I think on the other side of the fence, without mentioning units, there are pumps that continually have maintenance problems, auxiliary feedwater pumps that come down annually or have bearing cracking. There is a clear indication of the pump telling you that that style (whether it be that manufacturer size, or construction) just can't tolerate that service.

I think in those cases we need to upgrade metallurgy. We need to upgrade designs, change the internal fits within the pumps, and make them less tolerant. But I don't think we should suspend this testing until and unless we put full flow testing in, because there are some pumps out there that can take it. Then the only issue becomes hydraulic proof.

Question 12 (Temperature Affects on Failures)

The data and failures discussed come from boiler feed pump and RHR pumps. Both types of pumps spend many hours pumping fluids that are hot.

Does temperature of the fluid being pumped play a role in the damage or failures? Please explain how the fluid temperature has been addressed in your research.

Response 12

Panelist The fluid temperature does come into play. We know in particular it comes into play in the NPSHR, so it can have an influence. I am not sure that the temperature played an important role in the feedwater pump failures that I gave as examples.

Panelist Temperature comes into the picture. During warmup deformation it can be a problem, and we face seizure problems. The boiler feed pumps, if they are turbine driven, are put on turning gear. When you get ready to start up, you are on turning gear 5 to 10 rpm, and you increase the temperature from ambient to, say, 420 degrees or 350 degrees.

You go through a period when the shaft and the barrel are both in such a mode that the pump will seize. Seizure is a very serious problem. The impeller eye and the impeller bearing must be of a certain material, and the basic rule is that they must be at least 10 Rockwell C difference in hardness. Certain materials are what we call seizure proof or nongalling, but if you use common materials, seizure is indeed a very serious problem for all the feed pumps, especially if the boiler feed pump has many stages, like 6, 8, 10, and then seizure is the main result of temperature- caused problems. The majority of them are seizures on the turning gear.

Question 13 (Instrumentation of Mini-Flow Lines)

Please talk more about the necessity (or lack of it) or desirability of installing minimum flow line flow instrumentation. It was indicated yesterday that NRC thinks it should be installed for IST purposes.

Response 13

Panelist If minimum flow is a severely reduced percentage of the design flow whether it be an auxiliary feed water pump or a similar pump where the rise to shutoff results in a pump curve that is essentially flat from zero flow to perhaps as high as 50% of BEP, it's been my experience that, even if you put a flow transmitter of any kind in that line, the indication is going to be cyclical. It will vary all over the place as the system tries to find a home. So, the reading, even though you are going to get a reading of minimum flow, if the pump is designed for 300 gpm but the curve is flat to as high as 150 and the minimum flow is 50 gallons a minute, the reading will be jumping all over the place, perhaps between 25 to 75 gallons a minute. It's like asking five people for a reading and you get six answers. It depends upon your viewpoint, how you are looking at the gauge or if it's a digital output. I don't think measuring an unstable point should give you a warm and fuzzy feeling.

I think the issue is either we design the pumps to run at minimum flow and, more importantly, if we're actually out to demonstrate operability on one hand, which is response, and then performance, we shouldn't be testing at minimum flow at all.

So, by putting an instrument in the minimum flow path to perhaps enhance our data, if I can be allowed to say this, is a nickel fix for a dollar problem, so long as you are way back on the flat end of the pump curve. I think as has been pointed out, the only example or, if you will, exception to this rule are the high specific speed pumps. These again are the large circulating water pumps where the curve shape lends itself to the constantly rising characteristic to shutoff. By and large, the bulk of the centrifugals, whether they be all radial or mixed flow that we find in nuclear service, just don't lend themselves, if the minimum flow is a flow or another when the pump is operating at the low end of the curve.

So the bottom line is I don't really think it enhances anything. In fact, it may actually contribute to the uncertainty as to what the pump is doing.

Panelist There is a very important point in measuring flow in recirculation lines when you have a characteristic curve such as he's described because you don't know whether you have flow at all, and it's very important that you do have flow. The line can be clogged.

Question 14 (Consistent Instrumentation of Pumps)

Since, as stated, there can be alleged variation in the readings obtained by different vibration instruments due to different ways of determining the peak-to-peak values, should the code address controls to ensure that instruments with equivalent response are always used to measure vibration on safety-related pumps?

This is important because changing instruments could allow a degraded pump to go undetected.

Response 14

Panelist I would like to emphasize that when we develop Code rules they are really minimum requirements. It's very difficult to get a consensus standard through the system. There are so many people, and they haven't all heard the discussions. So, we generally have minimum requirements. It is up to the owner. Many times the licensee has to know what equipment he's using, and when pump testing or valve testing or any other testing, we have to know what we're doing and we have to be consistent.

I think we must look at pump testing or any other testing this way and keep in mind what we're doing. We're placing more emphasis on the reference values. The reason we're doing that is we want people to be consistent in their test methods. Some of this you are going to know when you get your test results. If your test results start jumping all over, you are going to find out that you haven't done your test right, and that should show up. So if you have an instrument like this and you change it, I think you will see the difference, and you know that you need to take some action.

Question 15 (Identifying Pump Degradation)

How could we identify pump degradation by using pump head curves for the pumps that normally run on the flat portion of the pump curve?

The second part of the question is how close could we get to the pump shutoff head and pump runout to perform pump operability?

Response 15

Panelist I don't know of a pump application where the pump is sized and required to run on the flat portion of the curve, or the design flow happens to be I think the issues that I'm aware of relate back to testing at minimum flow or testing at flows where the curve is flat. I'd be interested in knowing where a manufacturer was asked to design in the flat region only.

One would expect that the typical application of a pump would place the pump's design flow which is at best efficiency—one would hope that the application requirement for the system would be somewhere near that. That's not always the case. It typically is a problem on auxiliary feedwater pumps where the pump's BEP flow might be actually another 1.. or 20% to the right of where the application flow happens to be.

Again, it's a function of the specific speed and the application of the pump. But I'd be curious to find the application where the pump which would have perhaps a best efficiency of some 600 gallons a minute, the pump specificity being 600, where the service or the system flow was so low that it would be in the flat region, say perhaps 300 to 250 gallons a minute. I'd certainly like to know a little bit more about the application because this type of characteristic is not that which you typically find in a nuclear pump in a nuclear plant. There are characteristic curves for pumps that are typically found in balance of plant, and this usually isn't a problem, at least not in my experience, and I don't know if anyone else can speak to it.

Panelist If you look at the pump curves in figure 33, you can see that there was a bad droop in the curve before the pump was modified. The modification completely removed the droop. The modification involved installing a ring in the pump volute to change its shape and the size of the gap.

If the volute shape is improper, there's a gross misalignment, but if the geometry is not correct the flow shoots back from the diffuser behind the impeller side plate, and that energy loss is the cause of the hump on the pump curve. It's all fluid mechanical, and it's the hydraulic instability caused losses.

Panelist There was a second part of this question, which asks how close should we get to the pump shutoff head and pump runout to perform pump operability, testing, and is this we're talking really beyond the best efficiency point.

Panelist Again, it will be a function of the pump and the size of the pump and the curve shape and the particular pump design. It's been my experience that we should avoid minimum flow where the curve is flat. From a pump designer's standpoint, we really should be looking at points where the curve is dropping initially to the left of BEP so that you have a discreet intersection between a system curve, whether it be a manual, a test line, or the actual system itself, where instead of getting a low side of an infinite number of flows for a given head, you've got one head for one flow.

Typically, our rule is that we'd like to find an intersection where the head is about 2 to 3% less than shutoff, if I'm looking to the left of best efficiency. To the right, certainly again I would stress that we would want to recommend to the vendor that we may be doing this. I would expect to be able to say, generally, that pumps are generally stable to about 115% of their best efficiency flows.

If you need to go beyond 115 to 120 to 125, then I would recommend that we consult our pump designer to ensure that the pump is equally stable far to the right of best efficiency as we would expect it to be stable to the left of best efficiency. But as a general rule, my experience is that you can go to at least 15% past BEP and expect to get a reliable, stable, repeatable point of performance.

Question 16 (Restrictions on Running Pumps in Mini-Flow)

Do the pump manufacturers adequately define the restrictions to running various pumps on minimum flow? More precisely, how should they provide this information and how should they state it?

Response 16

Panelist I'll reply by referring to replies from 88-04. Usually, the plants go back to the vendor and get his recommendation as far as minimum flow is concerned. Many of the replies have come back saying you can operate at a given number of hours at that flow rate.

I really wish I knew what was behind their decision, because you can't make that decision in isolation. You have to take into account the remainder of the operating history of the pump in order to do that, and you also have to have a very good knowledge of the forces and so forth that are imposed on the pump when you operate at minimum flows.

I am not aware that one is able to calculate forces that are imposed on the impeller or on the shaft at minimum flow conditions. For design flow, of course they do make those calculations, and that gives them a way to predict what life one should expect from a pump. But at minimum flow it's not clear to me how they do that or even whether they can.

Panelist It's been my experience that manufacturers do know how to calculate a number of factors. We've talked about them up to this point. We've talked about recirculation at the eye of the impeller, recirculations that just showed vane tips. There's also been a lot of work done to calculate the radial thrust on impellers so that if we look at differentiating between the point that I've been exposed to over the last 15 years—if we can differentiate between saying minimum flow, and clearly make it understood that we're looking minimum intermittent flow or minimum continuc ' flow.

I think the word "intermittent" or "con... ..us" was missing from the specifications in the past. The manufacturers treated the minimal flow issue as if it was a minimal flow on a boiler feed pump on a 300 megawatt fossil plant or on pumps that they had experience with in earlier generations. I think if you pose the question two ways, you'll get two completely different answers, intermittent being perhaps minutes. Once in a while a minimum continuous would be a reading that can be sustained continuously.

Attendee Shouldn't they be more clear what intermittent means precisely?

Panelist My opinion is we shouldn't ask for intermittent flow requirements. I think you'll get into subjective evaluations. You'll go to three or four engineers. You'll get five and six different answers because it will be a function of the age of the machine, the speed of the machine, the horsepower, etc.

It's been our experience if we insist on only one term, minimum continuous, and make it clear that this is an operating point where the pump could spend 8,000 hr a year, you'll get a completely different answer from the vendor, and he knows more about what that flow happens to be.

It's very subjective to talk intermittent unless you can say it's five minutes once a day, five minutes once a month, or five minutes once a year. But since we don't know what that is and it can vary from site to site or from job to job, it would probably be safer not to even enter the realm of intermittent if you are going to protect a vital piece of equipment in the plant because it only enters into the subjective evaluation and the personal evaluation that may not be substantiated by calculations or actual test data.

Question 17 (Flow Instability Effects)

Does OM–6 provide the latitude to allow for flow instability effects created by testing pumps on minimum flow pass; for example, induced vibrations at greater than 0.325 in. per second due to operation close to shutoff head.

Response 17

Panelist No, at this time the standard does not allow that. You would be in the alert range at that level. So I guess the answer is just no.

Question 18 (Flow Instability Effects)

A lot of excellent work has occurred to identify flow instability concerns. Are pump vendors involved? If so, is there any indication that they are cauti... ; use of their products in unstable regions. The Code cautions that owners need to check with vendors? However, if they are not involved, the industry may suffer.

Response 18

Panelist When I see a pump failure problem, or system problem, usually the vendor is involved. It wasn't so 20 years ago. People weren't used to an outsider being a consultant other than the pump vendor. Normally, consultants were consultant only to the vendor. Everything was kept in great secrecy. That is changed. Normally, we prefer to work with the utilities, like the work I showed you and the Limerick nuclear feed pump. We did it as a complete uniform team. As a matter of fact, when I worked on that pump I had the feeling that I was employed by them, because they treated me as if I were one of them.

The flow instability problem was a long, long road for me to persuade them that it is there; it is a nasty problem. Lots of testing has been done and there is agreement on the hydraulic instability. It applies to Delaveau and the others. As a matter of fact, we have European pump vendors involved in our research. So the answer is yes, it's a fairly accelerated research.

So there are three parties involved there: the pump vendor, the utility, and then the consultant. And the exchange of information is very free in this area because we all realize that it's a very damaging, very nasty, not well–understood fluid mechanical problem. You cannot calculate it. You have to test it, and then later on you have to depend on statistical data if you need it. But I have not seen anyone yet who could calculate it. There are a couple of computer programs available out on the market, and to give you one example, one computer program calculated onset of hydraulic instability at 45%, and when we tested the pump the result was 85%.

If that is accurate enough for you and if you call that a calculation, then I accept it. But I have to admit I don't know how to calculate it, and I have not yet met anyone who guesses 20%. I guess 80%. And if we get 50% then it's half and half.

Question 19 (Replacement Pump Specifications)

When specifying replacement or new pumps for safety class applications, what code and/or specs give guidance to ensure impeller valute geometry and impeller-to-diffuser spacing of the gap? Are there any references?

Response 19

Panelist More and more companies are specifying these numbers, like distance between the impeller and the diffuser, the gap A. But it's a beginning, and I have not seen too many of them unless I was personally involved before they finished up the specs. We have so-called specification guidelines published by the Electric Power Research Institute, but we were very careful not to call it a specification guideline because who are we to put down exactly the specifications. It may apply to a class of pump and people will misunderstand it and they will religiously apply it to some other pump or it doesn't apply. So there is no set rule. You can look at some of my publications, and I try to caution everybody. Don't use it without thinking this pump may be different than that. You can't use the same rule for a diffuser pump as for a valute pump, or if the specification reads 1,200 you cannot use a specification that was written for a 2,000.

Normally, the specifications are written for that particular application, and it will change from application to application.

Panelist I agree on avoiding specifying design features of a pump.

Typically, working for an architect engineer, we design systems, and we write performance specifications. Every now and then we might get into the design of a particular pump by specifying a bearing type or a coupling type. But when it gets down to the hydraulic design, the more you put into the spec as to what you think you want—even with all of the papers over the last five or ten years—the more you will find that you will be taking responsibility for that pump or finding that the manufacturer's own unique experience runs counter to that. So I think we should focus on performance specifications—that specify a design flow, design head, and other specific performance data. The other design detail, I think should be left in the hands of the turbo machinery designers.

PLANNED SESSION PAPERS

PART 3: GENERIC CONSIDERATIONS FOR VALVE IST PROGRAMS

INTRODUCTION TO OM–10
TECHNICAL DIFFERENCES BETWEEN IWV AND OM–10

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ABSTRACT

ASME/ANSI OMa – 1988 Part 10, developed over the last ten years has recently been adopted by the Boiler and Pressure Vessel Code, Section XI. This new standard will provide new inservice testing requirements for valves and replaces IWV. The paper will highlight the significant changes to valve testing requirements, and provide the philosophy behind the new requirements.

Background

Nuclear power plants are required by Federal Regulation [10 CFR 50.55a(g)] to test certain ASME Code Class 1, 2 and 3 pumps and valves in accordance with ASME Section XI of the Boiler and Pressure Vessel Code. The Board of Nuclear Codes and Standards created an organization, Operations and Maintenance (O&M), whose charter was to develop standards for testing of components for nuclear power plants. The working group on Pumps and Valves under directions from the O&M Main Committee was to take Section XI IWP and IWV and use them as the starting point for new standards for inservice testing of pumps and valves. These new standards were to replace IWP and IWV of Section XI.

The new O&M pump and valve standards (Part 6 and Part 10) have just been published as the 1988a addendum of ASME/ANSI OM–1987 Operation and Maintenance of Nuclear Power Plants. These new standards are referenced in the 1988 addendum of Section XI and replace Subsection IWP and IWV of that code.

Part 6 and 10 Goals and Objectives

The O&M working group on Pumps and Valves was directed to review Section XI IWP and IWV, and use the lessons learned to develop new O&M standards. One goal of the working group was to develop standards which could be more easily followed and yet contain all of the important aspects of the current Code.

Most nuclear plant technical specifications require the utility to obtain specific written relief from the Nuclear Regulatory Commission (NRC) for any deviations from the Code. This has been very difficult for both the nuclear plant operators and the NRC, who must review and approve each deviation. For example, if a utility wants to determine flow rate by the change in level of a tank divided by time instead of measuring flow directly, then the utility must request relief from Code requirements. This situation arises when there is no installed flow instruments—usually for small pumps. The working group has tried to incorporate alternative testing into the standard when plant conditions make some testing impractical.

At first glance, one would think OM–6 and OM–10 are quite similar to the IWP and IWV sections that they replace. However, on closer inspection, an inservice test (IST) engineer will note many significant differences. These differences will make IST programs based on O&M Part 6 and Part 10 easier to implement. The remainder of this article will look at these differences in detail.

Overview of Part 10 Changes

As with Part 6, the new valve testing standard, Part 10, has increased the scope of testing to all valves which perform a safety function, regardless of construction code. Additionally, any safety relief valve which protects a system performing a safety function must also be tested. This part will require additional non–ASME valves to be tested. The expansion of scope is justified in that each important valve required for accident mitigation should require testing.

Each nuclear utility must submit an inservice testing program to the Nuclear Regulatory Commission at the start of each ten–year interval. If there are parts of the Code (or standard) which the licensee has determined to be impractical, Federal law requires the Licensee to demonstrate the basis of this determination to the satisfaction of the NRC. One of the prime goals of the working group at the onset of OM–10 development was to lessen the number of relief requests required. This goal was particularly true for valve testing.

Two areas which should require fewer relief requests is in Section 4.2.1.2. Exercising Requirements for

Category A and B Valves, and 4.3.2.2, Exercising Requirements for Check Valves. These sections were written specifically to allow testing at various frequencies depending on plant conditions. For example, safety injection valves according to NRC guidance should not be tested every three months if testing these valves could put the reactor in a less safe condition. Sections 4.2.1.2 and 4.3.2.2 allow testing at refueling outages if testing cannot practicably be done during operation or at cold shutdown.

Stroke Testing

One of the major tests for valves in both Section XI and OM–10 is stroke testing. In the past, valve testing results were required to be compared to the last test, and if the stroke time increased more than 25 % (or 50 % depending upon stroke time), the test frequency was increased until the condition was corrected or the valve was determined to be acceptable as is. Contrary to this philosophy, OM–10 requires a valve's test results to be compared to a reference value. As with pump testing, a reference value is established when the valve is operating acceptably. Then each test is compared to this reference value. This way, a slowly degrading condition must be corrected instead of possibly being acceptable under Section XI rules.

The acceptance standards for valves have also been narrowed. For example, a fast acting motor operated valve must stroke within ±25% of its reference value instead of within ±50% of the last test.

It is expected that this new acceptance criteria will make stroke time testing of power operated valves more effective in assuring operability.

Containment Isolation Valves

One area of inservice testing is the testing of containment isolation valves. These important valves must isolate the reactor containment in case of an accident. Federal law (10 CFR 50) requires, via Section XI, that valves which must be leak–tight (i.e. containment isolation valves) be tested under the owner's inservice testing program. This same 10 CFR 50 in Appendix J also contains rules for leakage rate testing of containment isolation valves. In an attempt to simplify the IST Programs, Part 10 requires those valves whose only function is containment isolation to be tested in accordance with Appendix J only. However, if a containment isolation valve also functions as a pressure isolation valve, then the valve would require additional leakage testing for the pressure isolation function in accordance with Part 10.

Relief Valve Testing

Part 10 requires certain relief valves to be tested per ASME OM–1. This standard, which is Part 1 of OM–1987, has been specifically written for testing of nuclear power plant relief valves.

Earlier versions of Section XI require relief valve testing in accordance with PTC–25.3. This standard was written for testing at the manufacturer's facility and is very difficult to apply to an operating power plant.

Other. Table 1 of Inservice Test Requirements of Part 10 has been expanded from the corresponding Section XI tables. This table now more clearly identifies required valve inservice tests. It specifies each test and clearly defines when a test is required.

The records and reports section has also changed. The records requirements for Part 10 have been increased but are consistent with what is required for pump testing in Part 6.

Future Changes

At the February 1989 ASME Board on Nuclear Codes and Standards meeting, a motion was passed to recognize that the Operations and Maintenance Committee is the appropriate committee to establish inservice testing requirements. The O&M Committee is to proceed with making OM–1987 stand on its own with the objective of eventual deletion of inservice testing from Section XI when appropriate. The O&M Main Committee at its March 1989 meeting approved a resolution to organize a task group which would determine what changes would be necessary to implement this change. The standard for testing snubbers (OM–4) is affected in the same manner as are the pump (OM–6), valve (OM–10), and relief valve (OM–1) standards.

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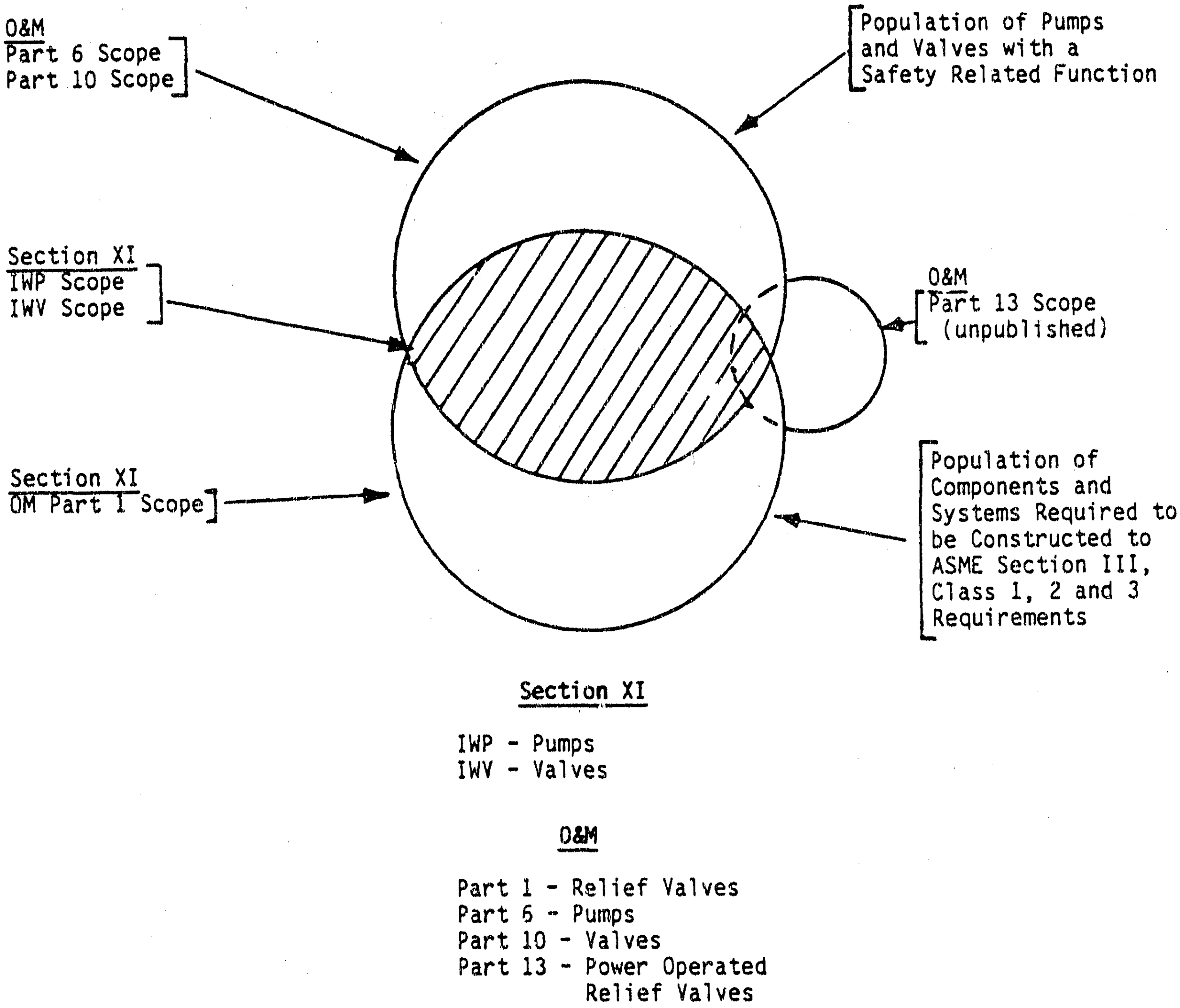

Figure 1. An overview of how the scope of the new O&M pump and valve standards relate to the scope of pump and valve subsections of 1986 edition through the 1987 addendum of Section XI.

Table 1. Comparison of IWV to OM–10

	IWV	OM–10
IWV–1100	Scope	All safety related valves included deletes Class 1, 2, and 3. IWV requires testing of all Class 1, 2, or 3 pressure relief devices. OM–10 only requires testing of those which protect a system which performs a safety related function.
IWA–9000	Definitions	Obturator Preservice test period Reactor coolant system pressure Isolation Reference values
IWV–2100	Categories of valves	OM–10 ties the function of the valve to the scope statement for purposes of classification.
IWV–3100	Preservice testing	Clarifies that preservice testing is required prior to the time the valve is "required" to operate.
IWV–3200	Valve replacement, repair and maintenance	Places more emphasis on reference values. Deviations shall be identified, analyzed and documented.
IWV–3300	Valve position indicator verification	Requires, where practicable, to observe flow or other suitable parameter to verify obturator movement.
	Inservice tests	New requirement – valve testing program shall commence when valves are required to operate per 1100.
	Reference values	New paragraph – places emphasis on reference values. "Shall only be established when valve is known to be operating acceptably.
	To Establish an additional set of reference values	New paragraph – tight controls on how reference values are established.
IWV–3411	Test frequency	Change "at least every three months" to "nominally every three months".
IWV–3412	Exercising procedure	Paragraph rearranged in logical order. Must start cold shutdown testing within 48 hours or complete all cold shutdown tests.
IWV–3413	Power operated valves	Adds requirement that any abnormality or erratic action be recorded and evaluated for corrective action.
IWV–3410	Valves in systems out	Changes required testing from of service "within 30 days" to "within three months."

Table 1. (continued)

IWV–3417 Corrective action

	2 s		<10 s		>10 s	
	IWV	OM–10	IWV[a]	OM–10[b]	IWV[a]	OM–10[b]
Electric motor operated valves	—	2 s	+50%	±25%	+25%	±15%
Air or hydraulic operated valves	—	2 s	+50%	+50%	+25%	+25%

a. From previous valve test.

b. From reference valve.

IWV		OM–10
	Corrective Action	New paragraph • If valve fails to stroke – immediately declare inoperable. • If stroke time does not meet limiting values of stroke time – immediately declare inoperable. • If stroke time does not meet 2 s/15%/25% criteria retest 96 h to analyze. • Documentation requirements added for analysis.
	Containment isolation valves	To be tested IAW 10 CFR 50 Appendix J.
	Leakage rate test for other than containment isolation valves	Seat leakage test specifically required for valves which perform a "function" other than containment isolation.
IWV–3423	Differential test pressure	Deletes option of testing valves in either direction if 15 psi function differential pressure. Deletes option to correlate mediums; i.e., air to water.
IWV–3424	Seat leakage	Allows a group of valves to be measurement tested at one time; e.g., a containment penetration. Provision made for determination of leak rate by pressure decay (not previously allowed).
IWV–3426	Analysis of leakage rates	Changed water leakage rates from 30D ml/hr to 0.5D gpm or 5 gpm.
IWV-3427	Corrective action	Deleted provision for change of test frequency based upon increase of leak rate.

Table 1. (continued)

IWV		OM–10
IWV–3522	Exercising procedure	Logically ordered from full stroke during normal plant operation to stroke only at refueling outages (same as for Cat A and B valves). For mechanical exercising of check valves, requires break away torque to be compared to reference value.As an alternate to quarterly testing allows disassembly every refueling outage.
IWV–3523	Corrective action	Deletes 24 hour grace period to correct valve anomaly.
IWV–3610	Explosively actuated	Requires a record be kept for each valves charge and to verify charge service life every two years.
IWV–6210	Summary listing	Deleted in OM–10.
OM–10	7000 records and reports	New section added Valve records Test plans Valves subject to test Category of each valve Tests to be performed Justification for deferral of stroke testing Record of tests Record of corrective action.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Please define an OM–10 accident, safety function, and safety equipment. If we can't nail down the scope, we can't tell where to start our test program.

Response 1

I absolutely agree. However, it is not OM–10 that decides. It is you, as the owner, that has to decide. You have to know exactly what equipment is required to perform in an accident condition or to mitigate an accident. You have to know every system designed to mitigate an accident, analyze that system, and determine the safety function of every pump and valve.

Once you know that, the program is fairly easy. The tough part is knowing the design basis of your plant. Start with the FSAR, which is the system concept, and then go down to specific components within the system or connecting systems.

Comment

As far as using the FSAR to define an accident since the word "accident" is used only in the Code—inspectors asked me if a tube rupture of a heat exchanger (service water component heat exchanger) was an accident, and are the valves to isolate that heat exchanger required.

In the past we have used the accidents, as discussed in Chapter 15 of the FSAR, as a basis for an IST program.

Discussion

Panel That is correct. Those are exactly the accidents discussed by the NRC.

Panel That is a definition we can work from to defend testing or not testing components in the plant. But it is a fairly narrow definition of an accident because some FSARs don't discuss flooding issues in Chapter 15. They're discussed elsewhere in the FSAR. So, if we are going to work with Chapter 15 accidents, I would hate for someone to come up and ask about other components that are not covered by Chapter 15 accidents or required for Chapter 15 accidents.

Question 2

Some older plants have design basis advance consideration and safety–related functions based on the capability to shut down and maintain the reactor in hot shutdown, not cold shutdown. When reviewing IST submittals and preparing IST submittals, what consideration, if any, is given to submittals from plants whose original design basis accident analysis was based on hot shutdown and maintaining the reactor in hot shutdown.

Are IST submittals reviewed based on ASME Section XI and O&M standards only, which require safety–related functions as a specific function in shutting down reactors to the cold shutdown condition?

Response 2

The NRC does recognize hot shutdown conditions for plants licensed to hot shutdown. For reviews the reason we base the questions and our considerations on cold shutdown is that cold shutdown is the only mode specifically addressed in Section XI. If you look in IWV, it specifically states the equipment required to shut the plant down to the cold shutdown condition.

However, if your plant is licensed to hot shutdown, the NRC will look at that to evaluate your program and grant relief for your relief request.

GENERIC CONSIDERATIONS FOR INSERVICE TESTING OF VALVES

ROBERT E. MARTEL III[a]

ABSTRACT

The issues dealing with valve reliability in the Nuclear Industry have increased substantially over the past few years. As a result, any papers, studies, and proposed industry changes have been published on this subject by Electric Power Research Institute (EPRI), Institute of Nuclear Power Operations (INPO), utilities and the Nuclear Regulatory Commission (NRC). While the cure to these problems are in the embryonic state, industry has made great strides in their commitment to maintenance, testing, and identifying the root cause of valve problems. This paper briefly describes where the industry is today and where we may go tomorrow. Generic considerations for Inservice Testing (IST) of valves will be addressed in accordance with recently approved SERs, Industry problems, and regulatory directions as follows:

- Full flow testing of check valves
- Back flow testing of check valves
- Containment isolation valves
- Pressure isolation valves
- Power operated relief valves
- Stroke testing
- Timing rapid–acting valves
- IST program scope.

BACKGROUND

Nuclear power plants are currently mandated in accordance with 10 CFR 50.55a(g) to perform periodic surveillance tests on safety–related valves per the ASME Boiler and Pressure vessel code, Section XI, Subsection IWV. These tests are intended to provide information for assessing operational readiness of components required to perform a specific function in shutting down the reactor to the cold shutdown condition or in mitigating the consequences of an accident.

Today the utility industry and the NRC realize that current testing maintenance requirements may not be comprehensive enough (References 1 to 9) to provide information on valve degradation. As you heard from Tom Hoyle and John Zudans previously, the valve testing methods and requirements have come a long way since the mid–sixties and the ANSI N45 Committee (Figure 1). Through the Committee on Operations and Maintenance (O&M), the Working Group (WG) on pumps and valves with individuals from utilities, the NRC, research groups, consulting companies, and manufacturers are improving test criteria for assessing operational readiness. Valve degradation and operational readiness go hand in hand.

A total of 35 of the 108 plants as of this writing have approved safety Evaluation Reports (SERs) on their Section XI programs. Due to the dynamic effects of the code and utilities gaining experience in testing many changes (relief requests) are being submitted to update programs. Despite the number of approved SERs, the NRC has evaluated all IST programs and performed IST reviews through a list of drafted questions. Unfortunately, utilities are skeptical to act on items through reviews and inspections that are not specified in SERs. However, with instances such as check valve failures (sticking open or discs coming loose) and MOV failures (torque switch, limit switch, spring pack failures) the industry and the NRC are faced with weaknesses in IST programs and operational readiness that cannot be tolerated.

VALVE TESTING

The results of IST programs are based on a number of different variables, i.e., experience level of their people, finances available, type of diagnostic equipment available, number of people available, and management commitment to excellence. This does not imply that the IST people at this symposium are not in search of excellence to the letter of their code edition and addenda. It is important to realize that plant

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Mid	1960's	Atomic Energy Commission began to develop ISI criteria
	1967	ANSI/ASME setup ANSI N45 Committee
Late	1960's	ISI in effect with no common exam methods or techniques
Oct.	1969	AEC announced licensing regulation to include ISI developed by N45 Committee
Jan.	1970	1st Edition of ASME Boilers and Pressure Vessel Code introduced for Class 1
Summer Addenda	1973	Required operability testing of ASME Class 1, 2 and 3 pumps and valves
	1974	Required inspection of ASME Class 3 systems and supports
Jan.	1975	AEC licensing and regulatory functions transferred to the NRC
Late	1975	N45 Committee disbanded and committee on Operations and Maintenance under ASME created
Feb.	1976	ASME O&M Committee established the subcommittee on Performance Testing
Late 70's & 1980's		ASME Working Group developing OM6 & 10
	1989	IWV & IWP removed from Section XI reference only to Consolidation Document

Figure 1.

management must be in tune to Section XI in order to be committed to such a program. A high priority with top notch people must be dedicated to Section XI in order to evaluate testing, determine root causes of problems and establish good procedures. Coupled with a good plant maintenance program, this action will help provide assurance of valve performance and reliability under accident conditions.

A utilities, diagnostic methods are of prime importance. In the early 1970s, most stations timed valves with a stop watch. Today, many stations utilize strip chart recordings, signature analysis, and actual dynamic testing in addition to timing valves with stop watches.

The diagnostic methods required by Section XI/O&M 10 are quite simple; either an exercise test and/or leak rate test is performed. Studies of Nuclear Plant Reliability Data Systems (NPRDS) found some problems not identified by code methods such as; valve aging (loss of spring pack life); insufficient testing (no actual dynamics); design deficiencies (sizing), and maintenance practices. Examples of MOV failure are demonstrated by significant events at San Onofre and Davis–Besse (Reference 9):

- San Onofre experienced a rapid primary system cooldown while switching to the shutdown cooling mode. The cooldown was caused by two partially open shutdown cooling system valves that were indicating closed.
- Davis–Besse experienced a loss of all main and auxiliary feedwater for a period of about 12 minutes. A major contributor to this event was the failure of both auxiliary feedwater containment isolation valves to reopen upon demand following their closure.

Today the industry is playing catch up in these vital areas to maintain safe environment both inside and outside the plant protected area. The generic considerations for inservice testing for valves is based on SER reviews and inspections. The NRC Staff has identified a number of generic areas (Reference 10) that affect plant safety and have appeared as programmatic weaknesses in past IST or other NRC inspections. In order to remedy these generic IST problems, clarify the status of current programs with respect to the Technical Specification requirement, and to alleviate the problem with respect to review of program revisions, the NRC with comments from the industry proposes the following policy and guidance:

GENERIC ISSUES

Full Flow Testing of Check Valves and Alternatives

The ASME Code requires check valves to be exercised to the positions in which they perform their safety functions. The Commission feels a check valve's full--stroke to the open position may be verified by passing the maximum required accident condition flow through the valve; anything less would be considered a part-stroke exercise. There are many situations where this requirement is physically impractical and overly restrictive: An example would be when the maximum required accident flow is at pump run out flow. This is physically impractical.

PERSONAL PERSPECTIVE

Some check valves can be shown full open at much lower flow rates than those of the design conditions. If the check valve could be shown to be open at a lower flow rate by comparing differential pressure across the valve to a baseline when the valve was known to be operating acceptably, this should qualify as a acceptable method. Additionally, there are new techniques which are currently being marketed which claim to show check valve position such as MOVATS, Inc. Checkmate.

There is a condition in many plants where flow is directed through parallel paths. Flow is balanced through each of the parallel lines by a throttled valve which is administratively controlled. These parallel paths have been carefully balanced to ensure equal flow is provided through each line. Most plants administratively control throttled valves by locking or removing the handwheels. Due to special precautions taken, it was the consensus of the ASME Working Group that these lines should not require individual flow rate of the branch line to be measured to take credit for a full stroke of the check valve. Generic Letter No. 89-04 (Reference 10) states: Knowledge of only the total flow through multiple parallel lines does not provide verification of flow rates through the individual valves and is not a valid full--stroke exercise.

Other alternatives for testing check valves have been allowed by the NRC (Reference 2) and the generic letter. One example is utilizing a sampling disassembly/inspection program. At each disassembly, it should be verified that the disassembled valve is capable of full stroking and its' internals are structurally sound (no loose or corroded parts). In the event that the disassembled valve's full-stroke capability is in question, all valves in this group shall be disassembled.

The ASME Working Group is reviewing check valve disassembly as a future technical change to the valve standard. Check valve disassembly is a high priority in the areas of loose, corroded, and worn parts. ASME XI and OM10 lacks direction in the area of visual inspection during disassembly.

Back Flow Testing of Check Valves

Section XI requires that Category C check valves (valves that are self-actuated in response to a system characteristic) performing a safety function in the closed position to prevent reverse flow be tested in a manner that proves the disks travel to the seat promptly on cessation or reversal of flow. In addition, for Category A/C check valves (valves that have a specified leak rate limit and are self-actuated in response to a system characteristic), seat leakage must be limited to a specific maximum amount in the closed position for fulfillment of their function. Verification that a Category C valve is in the closed position can be done by visual observation, by an electrical signal initiated by a position-indicating device, by observation of appropriate pressure indication in the system or by other positive means. Examples of check valves that perform a safety function in the closed position that are frequently omitted from IST programs are:

- Main feedwater header check valves
- Pump discharge check valves on parallel pumps
- Keep full check valves
- Check valves in steam supply lines to turbine-driven AFW pumps
- Main steam non-return valves

- CVCS volume control tank outlet check valves.

PERSONAL PERSPECTIVE

Leak testing as a method of back flow testing of check valves should be recommended. The commission and industry should specifically address back flow testing of two or more check valves in series. This is a significant problem from a testing standpoint.

Utilities choose to disassemble one of two check valves in series while checking for seat leakage rather than adding a test connection between the two valves. It is important to know which valve is preventing backflow and if there is valve degradation, however, we must look at long-term solutions and not always take the easiest way out. More important in this issue is, does the removal of one check valve so that you can test only the remaining one improve nuclear safety?

At a EPRI Power Plant Valve Symposium in October 1988 (Reference 8), a paper submitted by William Suslick of Duke Power on Acoustic Emission Monitoring of Check Valve Performance takes a step in the right direction toward not disassembling check valves. Mr. Suslick and his co-author, Brian A. McDermott, conclude based on test results to date, "acoustic emission monitoring of installed check valves may be performed to monitor for tapping or instability. Acoustic emission monitoring will greatly reduce the number of valves that need to be disassembled for inspection." This paper and others will be produced by EPRI in a study on Pump and Valve Surveillance Test Optimization sometime this year.

Pressure Isolation Valves

General. Pressure isolation valves (PIVs) are defined as two normally closed valves in series that isolate the reactor coolant system (RCS) from an attached low pressure system. PIVs are located at all RCS low pressure system interfaces. 10 CFR 50.2 contains the definition of the Reactor Coolant Pressure Boundary (RCPB). In most cases PIVs are within the reactor coolant pressure boundary. In a few cases the staff has allowed individual licensees to consider a valve in an interfacing high pressure.

Class 2 pipe as a PIV. The following summary is based upon the staff's review of responses to Generic Letter 87-06, Periodic Verification of Leak Tight Integrity of Pressure Isolation Valves. All plants licensed since 1979 have a full leak test requirements and limiting conditions for operation (LCOs). The plants licensed prior to 1979 fall into several categories. Some pre-1979 plants have a full list of PIVs along with leak test requirements and LCOs in the plant Technical Specifications. Some pre-1979 plants have only Event V PIVs in the plant Technical Specifications. Some pre-1979 plants have no Technical Specification requirements regarding PIVs and therefore are not leak testing any PIVs.

All PIVs listed in plant Technical Specifications should be listed in the IST program as Category A or A/C valves. The Technical Specification requirements should be referenced in the IST program.

Event V PIVs. Event V PIVs are defined as two check valves in series at a low pressure/RCS interface whose failure may result in a Loss of Coolant Accident (LOCA) that bypasses containment. Event V refers to the scenario described for this event in the WASH-1400 study.

On April 20, 1981, NRC issued "Orders" to 32 PWRs and 2 BWRs which required these licensees to conduct leak rate testing of their PIVs, based on plant-specific NRC supplied lists of PIVs, and required licensees to modify their technical specifications accordingly. These orders are known as the "Event V Orders" and the valves listed therein are the "Event V" PIVs.

Based upon the results of recent inspections it has been determined the following implementation problem still exists with respect to testing of PIVs: The staff has determined in some cases the procedures were inadequate to assure these valves are individually leak tested and evaluated against the leakage limits specified in the Technical Specifications. In other cases the procedures were adequate but were not being followed. Specifically, some check valves were tested in series as opposed to individually and some check valves were not tested when required (i.e., for one plant inspected, whenever primary pressure was within 100 psig of the system design pressure on the low pressure side of the check valve).

Licensees should review their testing procedures to ensure the Event V PIVs are not leak rate tested in series.

PERSONAL PERSPECTIVE

The main concern is that in many cases PIVs are not individually tested. Since the concern is "in series testing of valves," the industry should endorse the position of testing these valves individually.

Limiting Values of Full–Stroke Times for Power Operated Valves IWV–3413(a) of the ASME Code requires that the licensee specify the limiting value of full–stroke time of each power operated valve. The corrective actions of IWV–3417(b) should be followed when these limiting values are exceeded. The Code does not provide any requirements or guidelines for establishing these limits nor does it identify the relationship that should exist between these limits and any functional operating limits identified for the relevant valves in the plant Technical Specifications or Safety Analysis Report (SAR).

The primary reason for measuring the full–stroke times of power operated valves is to detect valve degradation. The function of the limiting value of full–stroke time is to establish a value for taking corrective action on a degraded valve before the valve reaches the point where there is a high probability of failure to perform its safety function if called upon. The NRC has, therefore, established the position described below regarding limiting values of full–stroke time for power operated valves.

The limiting value of full–stroke time should be based on the valve reference or average stroke time of a valve when it is known to be in good condition and operating properly. The limits should be a reasonable deviation from this reference stroke time based on the valve size, valve type, and actuator type. The deviation should not be so restrictive that it results in a valve being declared inoperable due to reasonable stroke time variations. However, the deviation used to establish the limit should be conservative enough that corrective action would be taken for a valve that may not perform its' intended function.

When the functional operating limit for a valve identified in the plant Technical Specifications or SAR is less than the value established using the above guidelines, the appropriate Technical Specification or SAR limit should be used as the limiting value of full–stroke time. The limiting value of full–stroke time for a valve should not exceed a Technical Specification or SAR limit specified for that valve.

When the functional operating limit for a valve identified in the plant Technical Specifications or SAR is greater than the value established using the above guidelines than the limiting value of full–stroke time should be based on the above criteria instead of the plant Technical Specifications or SAR

PERSONAL PERSPECTIVE

Even though the WG had no comments on this section, some utilities officially responded. It was felt the ASME Code states that values are to be determined by the plant owner [IWV3414(a)]. Valve stroke times are established individually and are based on the time that is required for the valve to perform its safety function unless otherwise specified by a Technical Specification requirement. Establishing a valve stroke time limit based on a average (reference) stroke time is not defined in the present edition of the ASME Code and would render the corrective action requirements specified in IWV–3417 useless.

It is understood that the above reference stroke time method of assessing valve performance is specified in OM–10 but this method is a substantially different practice than what is presently specified in the Code and would require licensees to revamp existing programs to comply.

Stroke Time Measurements for Rapid–Acting Valves

The Code requires the following for power operated valves with stroke times 10 seconds or less: (a) limiting values of fullstroke times shall be specified [IWV–3413(a)], (b) valve stroke times shall be measured to the [at least] nearest second [IWV–3413(b)] and (c) if the stroke time increases by 50% or more from the previous test [or from a reference value] then the test frequency shall be increased to once each month until corrective action is taken [IWV–3417(a)]. Paragraph IWV–3417(b) specifies corrective actions that must be taken.

Most plants have many power operated valves that normally stroke in two seconds or less and they encounter difficulty in applying the 50% increase of stroke time corrective action requirements for these valves. The purpose of this requirement is to detect and evaluate degradation of a valve. For valves with stroke times in this range, much of the difference in stroke times from test to test comes from inconsistencies in the operator or timing device used to gather the data. These differences are compounded by rounding the results as allowed by the Code. Thus, the results may not be representative of actual valve degradation.

The following discussion illustrates the problem that may exist when complying with the Code requirements for many of these rapid–acting valves:

> A valve with a measured stroke time of 1.49 s during one test (rounded to 1 s), and

a measured stroke time during the following test of 1.51 s (rounded to 2 s) would exceed the 50% criteria and would require an increased frequency of testing until corrective action is taken. This can result from a stroke time difference of 0.02 s, which is usually not indicative of significant valve degradation.

Power operated valves with normal stroke times of 2 s or less are referred to by the staff as "rapid–acting valves." Relief may be granted from the requirements of Section XI, Paragraph IWV–3417(a) for these valves provided the licensee assigns a maximum limiting value of full–stroke time of 2 s to these valves and, upon exceeding this limit, declares the valve inoperable and takes corrective action in accordance with IWV–3417(b).

Licensees are required to either comply with the Code stroke timing requirements or the staff's rapid–acting valve position stated above. Since this represents a deviation from the Code requirements, a relief request must be included in the IST program. This relief may be requested for any or all of the rapid–acting valves in the IST program.

PERSONAL PERSPECTIVE

Assigning a maximum limiting value for rapid–acting valves is more conservative than the Code and is consistent with OM–10 By doing so, a valve can be timed to the nearest 1/20 or 1/100 of a second which is common practice.

Starting Point for Time Period in Technical Specification Action Statements

ASME Section XI, IWP–3220, states: "All test data shall be analyzed within 96 hours after completion of a test." IWP–3230(c) states, in part, "If the deviations fail within the 'Required Action Range' of Table IWP–3100–2, the pump shall be declared inoperative....."

In many cases, pumps or valves covered by ASME, Section XI, Subsections IWP and IWV, are also in systems covered by Technical Specifications and, if declared inoperable, would result in the plant entering an ACTION statement. These ACTION statements generally have a time period after which, if the equipment is still inoperable, the plant is required to undergo some specific action such as commence plant shutdown.

The potential exists for a conflict between the aforementioned data analysis interval versus the Technical Specification ACTION statement time period. Section XI, IWP–6000 requires the reference values, limits, and acceptance criteria to be included in the test plans or records of tests. With this information available, the shift individual(s) responsible for conducting the test (i.e., shift supervisor, reactor operator) should be able to make a timely determination as to whether or not the data meets the requirements.

When the data is determined to be within the Required Action Range of Table IWP–3100–2, the pump is inoperable and the Technical Specification ACTION statement time starts. The provisions in IWP–3230(d) to recalibrate the instruments involved and rerun the test to show the pump is still capable of fulfilling its function are an alternative to replacement or repair, not an additional action that can be taken before declaring the pump inoperable.

The above position, which has been stated in terms of pump testing, is equally valid for valve testing.

In summary, it is the staff's position that as soon as the data is recognized as being within the Required Action Range for pumps or exceeding the limiting value of full–stroke time for valves, the associated component must be declared inoperable and the Technical Specification ACTION time must be started.

PERSONAL PERSPECTIVE

It is believed that 96 hours was put into the Code in the 1970s to account for a long weekend when resources were not available for analysis. Today, some utilities put the acceptance criteria in their test procedures to avoid potential conflicts. In any case, it is imperative to know all safety limits and assure operational readiness when necessary.

Containment Isolation Valves

The NRC would like to see all Containment Isolation Valves (CIVs) that are 10 CFR 50, Appendix J, Type C, leak–testing included in the IST program as Category A or A/C valves. The Staff has determined that the leak test procedures and requirements for containment isolation valves specified in Appendix J are equivalent to the requirements of IWV-3421 thru 3425 (see the white paper for corresponding OM paragraphs). The licensee must also comply with the Analysis of Leakage Rates and Corrective Action requirements of Paragraph IWV-3426 and 3427(a).

IWV-3427(b) specifies additional requirements on increased test frequencies for valve sizes of six inches and larger and repairs or replacement over the requirements of IWV-3427(a). Based on input from many utilities and staff review of testing data at some plants, the usefulness of IWV-3427(b) does not justify the burden of complying with this requirement. Since this position represents a deviation from the Code requirements, it should be documented in the IST program.

PERSONAL PERSPECTIVE

Over the past five years, the commission has been consistent in allowing the utilities to use their Appendix J program to meet the requirements of IWV-3421 thru IWV-3425. At the same time, the commission has required utilities to specifically comply with IWV-3426 and IWV-3427a while allowing relief from 3427b. The reason for requiring compliance with IWV-3426 & 3427a was that Section XI is a component test code and, therefore, each component (valve) should be treated on an individual basis.

The Industry at the same time felt that containment isolation valve testing was subject to dual testing requirements of Appendix J and Section XI. The Industry, via the OM-10 Standard, has changed the philosophy by requiring only Appendix J testing for CIVs. The basis of this change is: (a) adequate requirements exist in Appendix J to ensure public safety, (b) the only function of these valves is containment isolation; therefore, relying on Appendix J program must be specifically tailored due to the plant's design. There are exceptions taken to Appendix J programs which must be granted by the commission.

Another reason to allow all CIV testing to be done in accordance with Appendix J is to avoid conflicting regulatory requirements. For example, the generic letter requires all valves with a safety-related function to be tested and allows the testing to be in accordance with Appendix J for IWV-3421 – 3425 requirements. However, there are a number of valves which are safety-related but are exempt from Appendix J requirements. The Containment Systems Branch within the NRC would have reviewed these exemptions and approved them. Under this generic letter, which is prepared under the Mechanical Engineering Branch, these valves would have to be tested.

IST Program Scope

The NRC Staff has taken the position that 10 CFR 50.55a(g) requires all valves performing a safety-related function to be included in the IST program and tested in accordance with the ASME Code, Section XI. This position is consistent with IEB 85-03 Supplement 1 as stated previously. In Supplement 1, the term "safety-related" refers to those systems and components that are relied upon to remain functional during and following design basis events to ensure (i) the integrity of the reactor coolant pressure boundary, (ii) the capability to shut down the reactor and maintain it in a safe shutdown condition, and (iii) the capability to prevent or mitigate the consequences of accidents that could result in potential offsite exposures comparable to the 10 CFR Part 100 guidelines. The definition follows closely with the new scope statement of OM-10 The Working Group felt the standard should not be limited to ASME Class 1, 2, and 3 components. In the newer ASME III plants, most of the valves intended by the Working Group to be covered by this standard would already be covered. However, there are usually some valves which are not ASME Class 1, 2, or 3 which do perform functions described in the scope. The Working Group stressed the point that function is more important than the code class. This point was well documented in the WG white paper.

PERSONAL PERSPECTIVE

Today, industry does not believe the regulation requires all safety related valves be included in IST programs. Instead, the utilities feel that the regulation requires only certain ASME valves to be tested as specified in Section XI, Subsection IWV of the ASME Boiler and Pressure Vessel Code. Based on the previous definition provided and discussions with the NRC during SER reviews (References 2,3), it is believed that the gap on safety-related definitions is narrowing.

The first addenda to ASME/ANSI OMa – 1987 published in February, 1989 does not specify Class 1, 2, and 3 valves in its scope statement. The new Operations and Maintenance Standard on Valves; OMa Part 10, does not address safety class. Part 10 does address active and passive valves in a IST program that are required to perform a specific function in shutting down a reactor to the cold shutdown condition, in maintaining the cold shutdown condition, or in mitigating the consequences of an accident.

- Examples of valves that may be required for shutdown and accident mitigation are often erroneously omitted from IST programs are:

- Valves in emergency diesel generator air start and cooling water systems

- Valves in fuel oil transfer systems for emergency diesel generators

- BWR scram system valves

- Control room chilled water system pumps & valves

- Accumulator motor operated isolation valves, or accumulator vent valves

- Pressurizer auxiliary spray system valves

- Valves in the emergency flow path

- Control valves that have a required fail-safe position.

Currently, many of the valves listed above are not being tested by some utilities. The implementation schedule should make provisions for this and allow an adequate amount of time to put testing procedures in place. It is very important to get plant operations input with a good understanding of the design basis when debating the safety-related issue. Unfortunately, the early ANSI B31.1 plants do not have well documented design basis and will require major backfit work for company engineering groups. A benefit to this will be a better understanding of plant design and upgraded documentation.

SUMMARY

To summarize, the current industry initiatives have identified areas where emphasis is needed to improve IST valve performance and reliability. long-term reliability is dependent on comprehensive valve programs which include the generic issues raised in this paper. The sole purpose of raising these issues in a Generic Letter is to clarify and update all IST programs with the latest safety concerns to alleviate problems with respect to review of program revisions.

GENERIC ISSUE SUMMARY

- Full Flow Testing of Check Valves and Alternatives A check valve's full-stroke to the open position may be verified by passing the maximum required accident condition flow through the valve. The alternative is disassembly or an approved (relief request) sampling disassembly inspection program. Review the design basis to include all safety-related valves into the IST program with respect to Generic Letter No. 89-04.

- Back Flow Testing of Check Valves
  - Review the check valves that perform a safety function in the closed position and are erroneously omitted from IST programs.

- Pressure Isolation Valves
  - Utilities should review their testing procedures to ensure PIVs are not leak rate tested in series.

- Limiting Valves of Full-Stroke Times for Power Operated Valves Utilities should be comparing average stroke times with Technical Specifications and SAR limits, however, corrective actions of IWV-3417(b) should be followed when limiting values are exceeded.

- Stroke Time Measurement for Rapid-Acting Valves
  - Relief may be granted from the requirements of Section XI, Paragraph IWV-3417(a) for rapid-acting valves provided the licensee assigns a maximum limiting value of full-stroke time of 2 seconds to these valves and, upon exceeding this limit, declares the valve inoperable and takes corrective action in accordance with IWV-3417(b).

- Starting Point for Time Period in Technical Specification Action Statements
  - The pressurizer PORV block valves should be included in IST programs as an NRC augmented requirement and tested to code requirements.

- Containment Isolation Valves
  - All containment isolation valves that are Appendix J, Type C, leak tested should be included in the IST program as Category A or A/C valves or relief must be granted by the Commission.

- IST Program Scope
  - Review the design basis to include all safety-related valves into the IST program with respect to Generic Letter No. 89-04.

REFERENCES

1. EPRI Application Guidelines for Check Valves in Nuclear Power Plants. Palo Alto, CA: Electric Power Research Institute, January 1988 – NP 5479.

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3. Safety Evaluation Report Pump and Valve Inservice Testing Program Millstone Nuclear Power Station, Unit III. Idaho National Engineering Laboratory, EG&G Idaho Falls, Idaho, November 1987 Fin No. A6812 for USNRC Washington, DC 20555.

4. A Review of Motor-Operated Valve Performance. Office for Analysis and Evaluation of Operational Data, USNRC, December 1986, AEOD/C603.

5. United States Nuclear Regulatory Commission, IE Bulletin No. 85-03: "Motor-Operated Valve Common Mode Failures During Plant Transients Improper Switch Settings," November 15, 1985.

6. United States Nuclear Regulatory Commission, IE Bulletin No. 85-03, Supplement 1: Motor-Operated Valve Common Mode Failures During Plant Transients Due to Improper Switch Settings, April 1988.

7. United States Nuclear Regulatory Commission, E. J. Brown and F. S. Ashe, "Inoperable Motor-Operated Valve Assemblies Due to Premature Degradation of Motors and/or Improper Limit Switch/ Torque Switch Adjustment," AEOD/E305, April 13, 1983.

8. EPRI Power Plant Valves Symposium, Charlotte, North Carolina: Electric Power Research Institute, October 1988.

9. United States Nuclear Regulatory Commission, NUREG-1184, "Loss of Main and Auxiliary Feedwater Event at the Davis-Besse Plant on June 9, 1985."

10. United States Nuclear Regulatory Commission, Generic Letter No. 89-04: "Guidance on Developing Acceptable Inservice Testing Programs," April 3, 1989.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

This report was based on the original draft of generic 89–04, not the recent 89–04.

One of the areas of my concern is the ISI program scope. A lot of the words, safety–related valves, and the list you included are no longer covered for 89–04. I think that is very clear to a lot of people. For example, diesel–generator systems are not part of 89–04.

The last part of 89–04, part 11, is very ambiguous. Part of the meetings we had with the Commission originally defined these systems which are not per 10 CFR regulations, that the utilities were asked to put in. So people who have diesel generator air start systems installed don't have to have them, and that should be clear to a lot of people.

I think these equipment lists should be updated to the new generic letter.

Response 1

You are correct. I do have a copy of the latest letter, and it does include BWR scram valves, control room chilled water, system pumps and valves, accumulated motor operation isolation valves, auxiliary pressurizer spray valves, boric acid valves in emergency boration, control valves that have a required failsafe position.

Unfortunately, the argument you will have to defend with the NRC is in respect to the diesel and air stop valves, valves in minimum flow lines.

Question 2

In the situation where a check valve functions as a barrier between a high and a low pressure system, how is a meaningful leakage criteria assigned to the valve; that is, how much back flow back leakage will compromise the valve's function?

Response 2

Again, if you don't know the design, if you don't know what your tech specs say, it is difficult for me to respond. When we did our SER reviews, we reviewed what was required for safety and what was needed to make a system function. Leakage has a lot to do with that.

We just went through it on our RHR system. I think we were allowed a magnanimous amount of leakage, as far as our RHR valves were concerned. But for every system it is difficult to put a number on leakage.

Comment

Due to a recent occurrence at Crystal River, where an auxiliary feedwater check valve leaked back and caused overtemperature conditions upstream of the valve which exceeded the piping design limits, the NRC has taken a closer look at that situation. They have come to the conclusion that there may be situations where valves have been categorized as B or C, but their leakage limit is of greater concern than just preventing a gross diversion of flow or a gross leakage.

If there is a specific leakage value of concern on a particular valve and that leakage is fairly small, the NRC has taken the position that the valves may be recategorized as either A or AC, as appropriate, and may be leak tested to ensure that they will perform their safety functions to prevent the backflow or leakage past the valve.

NRC EXPERIENCE WITH PUMP AND VALVE INSERVICE TESTING PROGRAMS

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ABSTRACT

This paper presents a discussion of the experience of NRC and their contractors in reviewing pump and valve inservice testing (IST) programs. Topics that will be discussed include the overall purpose of an IST program to assure continued component operability and to meet the NRC's regulatory requirements, general information that should be addressed in the IST program, and the level of technical detail needed to meet the licensee's and NRC's objectives. Other topics that will be addressed include the experience with the diverse quality of IST programs, scope of past NRC program reviews, problem areas encountered, and a discussion of frequently omitted systems and components.

NRC EXPERIENCE WITH PUMP AND VALVE INSERVICE TESTING PROGRAMS

INTRODUCTION

In 1976 the NRC adopted the requirements of the ASME Boiler and Pressure Vessel Code, Section XI, for the routine inservice testing of safety related pumps and valves. The Code of Federal Regulations has endorsed various editions and addenda of this Code through the years as these have changed and evolved. All commercial nuclear facilities are required to perform routine testing of these components as identified in the Code to verify their operational readiness and monitor for degradation.

The NRC, with technical support from the Idaho National Engineering Laboratory (INEL), has performed reviews and evaluations of pump and valve inservice testing (IST) programs for over 10 years. During this interval, detailed technical reviews have been performed on over one hundred nuclear plant IST programs. These reviews have generated detailed requests for additional information, which were used as agendas for working meetings between NRC, INEL, and licensee representatives. The result of these meetings has been IST program resubmittals upon which Safety Evaluation Reports have been written. This process has brought about substantial interface and discussions on the topic of inservice testing between the NRC, the INEL, and the nuclear industry in general. This has resulted in the identification of problem areas which are frequently encountered during inservice testing. It has also given those involved much insight concerning the identification of pertinent pumps and valves, the types of information that should be included in an IST program, and the level of detail needed to evaluate deviations from and alternatives to the ASME Section XI requirements.

While the requirements for inservice testing have been in effect since 1976, there are still many areas that need improvement. Many IST programs are submitted to the NRC for review in which components that perform a safety function are excluded and inadequate technical information is provided. Further, many justifications for cold shutdown testing and requests for relief are incomplete or inadequate.

In addition, several general problem areas have become evident during the review and evaluation of numerous IST program submittals. These have hampered the NRC's review and approval process for IST programs. They are the result of differences in interpretation of the Code requirements, inadequate information regarding inservice testing, or special considerations not addressed by the Code.

PURPOSE OF INSERVICE TESTING

The purpose of inservice testing is to assess the operational readiness of individual safety related pumps and valves and to determine whether degradation has occurred. The intent is to ascertain if a component's operational parameters provide reasonable assurance that it will perform its safety function when called upon. The Section XI testing is component specific and is not intended to verify the operability of complete safety systems. Conversely, system operability testing does not necessarily provide assurance of the future operational readiness of the individual components in those systems.

Section XI testing and acceptance criteria are designed to monitor component condition and require corrective action when there is indication of excessive degradation. Corrective actions are required when a component's measured test parameters are outside of the allowable ranges (i.e., into either the "alert" or the "required action" range). Components may be put on increased surveillance testing or declared inoperable and be repaired or replaced even if the system can still meet its operability requirements.

Inservice testing of pumps and valves must comply with 10 CFR 50.55a, which identifies the provisions for licensees to request relief from the requirements of the Code as well as the criteria utilized by the NRC to evaluate those requests.

CONTENTS OF INSERVICE TESTING PROGRAMS

General

Subsections IWP–1100 and IWV–1100 of Section XI of the ASME Boiler and Pressure Vessel Code define the scope of the IST program for pumps and valves. Both Section XI and 10 CFR 50.55a(g) specify Code Class 1, 2, and 3 pumps and valves, which perform a specific safety function, as those required to be included.

On the basis of 10 CFR 50.55a(g), the NRC Staff may require inservice testing of ASME Code Class 1, 2, and 3 pumps and valves, in addition to those included in the IST program, if these components are required to perform a specific function in the following:

1. Mitigating the consequences of an accident,

2. Shutdown of the reactor to the cold shutdown condition, or

3. Maintaining the reactor in a safe shutdown condition.

Section XI, Paragraphs IWP–1200 and IWV–1200, specifically exempt some components from testing such as valves used only for operating convenience (such as manual vent, drain, instrument, and test valves), valves used for system control (such as pressure regulating valves), and valves used only for maintenance. However, control valves that are required to perform a specific safety function as identified above, are not exempted from inservice testing. Examples of such valves could include the residual heat removal (RHR) heat exchanger bypass valves, control valves in the diesel cooling water, and control valves in chilled water systems.

Systems and Components Which Are Often Overlooked

IST programs frequently do not contain some of the pumps and valves that perform specific safety functions. The following list, while not all–inclusive, gives examples of both pressurized water reactor (PWR) and boiling water reactor (BWR) subsystems and components that are occasionally overlooked when preparing IST programs. This list uses the most commonly used terms for components, subsystems, and systems. However, terminology may vary from plant to plant. It should be recognized that the following systems or components are not necessarily safety related and Code Class 1, 2, or 3 for all plants.

PRESSURIZED WATER REACTORS

- Power–operated relief valves and associated block valves

- Reactor coolant system high point vent valves

- Any proposed flow path utilized for long–term core cooling and/or safety–grade shutdown

- Reactor coolant system pressure boundary isolation valves

- Steam generator atmospheric relief or dump valves
- Auxiliary feedwater turbine steam supply check valves
- Emergency boration flow paths
- Containment spray additive tank valves
- Containment isolation valves
- Containment combustible gas control
- Spent Fuel Pool Cooling System
- Emergency diesel generator fuel oil storage and transfer
- Emergency diesel generator external cooling water
- Emergency diesel generator air start
- Control room ventilation including cooling water
- Instrument/plant air supply to safety related valves
- Post–accident sampling containment isolation valves.

BOILING WATER REACTORS

- Automatic depressurization valve vacuum breakers
- Pressure boundary isolation valves
- MSIV leakage control
- Reactor Core Isolation Cooling System
- Containment isolation valves including excess flow check valves
- Containment combustible gas control
- Containment pressure suppression and vents
- Portions of the Control Rod Drive System
- Emergency diesel generator fuel oil storage and transfer
- Emergency diesel generator external cooling water
- Emergency diesel generator air start
- Control room ventilation including cooling water
- Instrument air or nitrogen supply to the main steam isolation valve accumulator
- Instrument air or nitrogen supply to the automatic depressurization valve accumulator
- Traversing incore probe squib and ball valves
- Spent Fuel Pool Cooling System
- Post–Accident Sampling System.

CONSIDERATIONS FOR A COMPLETE INSERVICE TESTING PROGRAM

The following information should be considered during the preparation of an IST program. It also provides a discussion on the recommended level of detail and considerations for presenting relief requests and cold shutdown test justifications.

The IST program should include a list of all safety related pumps and valves that undergo inservice testing as required by 10 CFR 50.55a. This list should clearly identify the components, the testing required by Section XI, and the testing that is actually being performed. Licensees have found that pump and valve listing tables are a convenient way to organize and present this information.

The pump and valve listing tables can be used to ensure that all identified components receive the specified testing at the stipulated testing intervals. The tables can also be used in the development of component test schedules and for schedule adjustments due to changes in plant operating conditions.

Pump Listing Table

The pump listing table should contain the following information for each pump that performs a safety function. The information comprises headings and a description of the text that should be included under each heading.

IST Program Revision or Revision Date: Include the program revision number or revision number and date on each page. Each revised program that is submitted should be distinguished by a specific revision number.

System: Identify the plant system in which the pumps are located or give a brief description of the pump.

Pump Identification: A unique identifier should be provided for each pump included in the IST program. This identifier should be used consistently in all plant IST documentation, such as system piping and instrumentation drawings (P&IDs), test procedures, and relief requests.

P&ID Number: This should be the unique identifier for the system P&ID on which each pump is shown.

Coordinates: This should be the coordinates on the listed P&ID where each identified pump is located.

Test Parameters: The table should address each of the seven in–service test quantities that are identified in Section XI, Table IWP–3100–1. The licensee should identify whether that parameter is being measured or observed for each pump in the IST program. Footnotes can be used to identify factors affecting testing. Relief requests should be provided where the testing cannot be performed as required.

Valve Listing Table

The valve listing tables should contain the following information for each valve that performs a safety function. The information comprises headings and a description of the text that should be included under each heading.

System: Identify the plant system in which the valves are located.

Valve Identification: A unique identifier should be provided for each valve included in the IST program. This identifier should be used consistently in all plant IST documentation, such as system P&IDs, test procedures, and relief requests.

P&ID Number: This should be the unique identifier for the system P&ID on which each valve is shown.

Coordinates: These should be the coordinates on the listed P&ID where each pertinent valve is located.

Valve Type: The valve type (i.e., gate, globe, check, relief) should be identified in this column. The valve type is an essential item in determining the valve category, which establishes the required testing (refer to Table IWV–3700–1).

Valve Size: Valve size should be specified in inches or decimal fraction of an inch of nominal valve size.

Actuator Type: Valve actuator type (i.e., motor, solenoid, air, hydraulic, self) should be listed. This is used in conjunction with valve type and function to determine the Code category and the required testing.

Code Category: Code category, as defined in IWV–2100, should be included in the table. The valve category determines the Section XI subsections that are applicable for a particular valve (i.e., IWV–3400 for Category A and B valves, IWV–3500 for Category C valves, and IWV–3600 for Category D valves).

Active/Passive: The valve listing table should indicate whether a valve is active or passive, as defined in IWA–9000. Valves that may be out of their safety position during routine power operation, would have to change position to perform their safety function and cannot be considered passive valves, even if their normal position is their safety function position. Testing requirements vary based on this classification. Table IWV–3700–1 indicates that Category A passive valves should be leak–rate tested, but no exercise test is required. Note 1 for this table states that no test is required for Category B, C, and D passive valves.

Safety Position: The safety function position(s) of a valve should be indicated on the valve listing table. For valves that perform a safety function in both open and closed positions, both positions should be indicated on the table. This is important for power operated valves which must have their stroke times measured as they are exercised to their safety function position(s) as well as for check valves because they must be exercised to the position or positions required to fulfill their function (refer to IWV–3522). The safety function position(s) of the check valves affect whether the valve should be full–stroke exercised open, exercised closed (verify reverse flow closure), or both full–stroke exercised open and closed.

Tests Performed: The table should show the tests that should be performed on each valve in the program. This list is the central focus of the valve testing program. Thus, the information should clearly show what tests are being performed for each valve [e.g., leak-rate testing according to IWV-3420, fail-safe actuator testing according to IWV-3415, valve remote position indicator verification according to IWV-3300, valve stroke time measurement according to IWV-3413(b)].

Test Frequency: Closely related to the type of test is the test frequency. The Code specifies the test frequency; however, performing the testing at this frequency is sometimes impractical. As a result, relief canbe requested or a cold shutdown justification can be provided. The table should show the actual test frequency for each required valve test.

Relief Requests: When it is impractical to comply with the Code requirements, 10 CFR 50.55a(a)(3) states that the NRC can grant relief and authorize alternative testing in lieu of that specified by the Code. To obtain NRC approval of a deviation from a Code requirement, where the condition is not specifically addressed in Attachment 1 of NRC Generic Letter 89-04, the licensee must submit a request for relief as part of their IST program. If a component is tested consistent with the positions taken in Attachment 1 of Generic Letter 89-04 as an alternative to the Code requirements, this situation should be noted in the IST program and approval is granted through the Generic Letter, provided the adequacy of the proposed alternate testing for detecting degradation is justified as discussed in the Generic Letter. These deviations from the Code should be identified for each applicable valve in the valve listing table using a unique identifier for each deviation.

Cold Shutdown Justifications

Section XI, Paragraphs IWV-3412(a) and 3522, permit exercising valves during cold shutdowns if exercising the valves quarterly during power operation is impractical. The Code requires the licensee to identify the valves to be tested at cold shutdowns. The NRC requires technical justifications demonstrating the impracticality of quarterly testing to be provided in the IST program. Cold shutdown justifications for each applicable valve should be identified in the valve listing table.

The justification should address the specific technical concerns that make quarterly part-stroke and/or full-stroke exercising of these valves impractical. Concerns such as damage to equipment, hazards to personnel, and possible plant trip should be thoroughly addressed. These concerns should clearly demonstrate the impracticality of performing the required testing quarterly for both full-stroke and, if applicable, part-stroke.

The licensee should also specify the method of testing that will be performed and the test frequency for the affected valves. For example, certain valves will be part-stroke exercised open quarterly using the pump's minimum flow path and full-stroke exercised with flow into the reactor coolant system during cold shutdowns and refueling outages. Further, the frequency of testing should not be presented in a manner which can be considered conditional. For example, an IST program might indicate that certain valves will be exercised during cold shutdowns when the reactor coolant pumps can be stopped. It is unclear in this example whether the pumps will be stopped for inservice testing of valves during each cold shutdown or just during those cold shutdowns when conditions permit stopping the reactor coolant pumps. If there are valid concerns that prevent full-stroke exercising valves during cold shutdowns, then relief must be requested from the Code requirements.

Requests for Relief From Code Requirements

For the licensee to obtain relief from the requirements specified in the ASME Code, Section XI, Subsections IWP and IWV, a request for relief must be submitted to the NRC for review and approval unless the alternate testing is consistent with the positions taken in Attachment 1 of Generic Letter 89-04.

The licensee has the responsibility of providing the technical basis which demonstrates that the proposed alternative provides an equivalent level of safety, performance of the Code required testing would result in a hardship without a compensating increase in safety, or that complying with the Code requirements is impractical. The discussion should thoroughly address specific technical concerns such as damage to equipment, hazards to personnel, or the possibility of a plant trip and should present the proposed alternative testing. It should also present the bases to support that the proposed alternative testing will accomplish the purpose of inservice testing discussed above.

When references are made to testing components as identified in the Final Safety Analysis Report, Technical Specifications, or other plant documents, the IST program should be self contained and include a description of the proposed testing frequency and methodology.

The following are examples of bases for not complying with Code requirements which if presented in a request for relief may result in the granting of relief (these are valid bases for relief, however, relief requests should provide more specific technical detail):

- Compliance with the Code requirements would be a hardship and would not provide better information for evaluating a parameter than the information currently available. For example, installing a gage with a range of three times the reference value or less to comply with IWP–4120 may not provide any better accuracy or readability than is provided by the presently installed instrument.

- Performance of quarterly or cold shutdown testing as required by the Code could result in a high probability of causing personnel harm due to high temperature environment, high radiation levels, high energy steam or fluid systems, etc. ALARA concerns may present a suitable justification; however, the licensee should provide information about the general area radiation field, any local hot spots, plant radiation limits and stay times, and the amount of exposure that would be received by personnel performing the testing.

- Quarterly or cold shutdown testing as required by the Code could result in a high probability of causing equipment damage. For example, isolating cooling water flow to a pump required to remain operating could result in overheating and subsequent damage to the pump.

- The licensee may propose to adopt a new technology for testing components. This new technology should provide a result either equivalent to or better than that required by the Code. For example, pump vibration measurements in units of velocity generally provide a better indication of pump degradation as opposed to measurements taken in displacement units as required by the Code.

Inconvenience is not a suitable justification for deviating from the Code requirements. Any testing places a burden on the licensee. However, unless the alternative testing provides adequate assurance of component operability and unless the burden of the Code testing outweighs the benefit of increased assurance of component operability, the Code testing should be performed as required.

The fact that testing necessitates removing a redundant safety system train from service is not a valid basis for relief from the Code. As long as the length of time required to perform the testing is less than the allowed out of service time of the Technical Specification action statement before commencing plant shutdown, the testing should be performed. Single failures are not postulated concurrent with taking a component or train out of service for testing. If the testing takes a train or system out of service and places the plant in a condition so that the design bases cannot be met, the testing may be temporarily postponed to cold shutdown. However, if this situation exists, the appropriateness of the design and testing should be evaluated and corrected as necessary.

Relief requests should propose alternate testing that provides reasonable acceptance criteria for the test data or a mechanism to determine component degradation. For example, licensees frequently request relief from measuring the hydraulic parameters for the emergency diesel fuel oil transfer system due to the lack of installed instrumentation. They propose to test system components by demonstrating that the system is capable of performing its intended design function. This alternate test does not necessarily evaluate the mechanical or hydraulic condition of the pumps or the condition of system valves.

Many systems have components whose design is well in excess of system requirements. The component could meet the system requirements and be degraded sufficiently that its future operability is questionable. For example, the above mentioned fuel oil transfer pump may normally produce a flow of 25 gpm while only 12 gpm is necessary to meet system safety demands. If the output of the pump dropped to 13 gpm, it meets system requirements; however, it is seriously degraded and may fail during subsequent operations. Therefore, the satisfactory performance of a system test with an acceptance criteria of 12 gpm would not provide a reasonable alternative to Code testing.

PAST EXPERIENCE WITH INSERVICE TESTING PROGRAMS

During the review of licensee IST programs and requests for relief from the Section XI requirements, the NRC and their contractors have identified several problem areas. Some relate to the quality of information provided to the NRC for reviewing and evaluating requests for relief while others relate to interpretations of Section XI requirements and their implementation. In an effort to address the significant problem areas that have been identified, the NRC has issued Generic Letter 89–04 and followed with Regional meetings with the nuclear utility industry. Other concerns that have been identified, which are not addressed in the generic letter, are listed below. Resolutions of these concerns should be developed by the ASME Code or the NRC Staff as appropriate.

There are many valves, which are identified to be tested only during cold shutdowns. However, for short duration cold shutdowns it may not be possible to test all those valves without delaying the plant's return to power. Should all valves identified to be tested at cold shutdown be tested during each cold shutdown?

Most boiling water reactor plants inert their containments with nitrogen to reduce the chance of a hydrogen explosion during an accident. De–inerting containment during each cold shutdown to allow personnel entry for pump or valve testing would be costly, time consuming, and could delay the plant's return to power. Should the containment of BWRs be de–inerted during each cold shutdown solely to facilitate inservice testing?

Section XI recognizes test frequencies of quarterly, annually, cold shutdown and biannually. The NRC recognizes other frequencies for inservice testing of components and plant technical specifications allow an interval margin of 25% for surveillances. What are the frequencies at which pumps and valves should be tested and should the technical specification allowable margins be applied to those test intervals?

Emergency diesel generating systems have been determined to perform a safety function. What components in the diesel generator subsystems should receive individual inservice testing to assess their condition and verify their operational readiness? Further, should there be differences in the way skid–mounted and nonskid–mounted components are treated?

Several plants have series check valves with no provisions to individually verify the closure of either series valve. What testing should be performed to individually verify the operational readiness of each of these series check valves?

Many plants have check valves, both inside and outside of the reactor containment, that can be verified in the closed position only by performing a leak test. What is the appropriate frequency for exercising these valves?

Some power operated valves perform a safety function in both the open and closed position. Since different failure mechanisms can affect these valves as they are stroked in each direction, must they be exercised and timed in both directions?

Many valves can be exercised only during cold shutdown periods. If one of these valves exceeds the "alert" stroke time limit during exercising at cold shutdown, monthly testing is required until corrective action is taken. This could require returning the plant to the cold shutdown mode monthly to test the valve. Should these valves be repaired or replaced prior to returning the plant to power?

How accurate should stroke time measurements be and is rounding of stroke times to the nearest second required?

Many pneumatically operated valves have attendant solenoid valves that must operate for the main process valve to function (stroke) correctly. Should testing be performed on these solenoid valves separate from testing of the associated process valve?

Section XI, Paragraph IWV–3300 states that "Valves with remote position indicators shall be observed at least once every two years to verify that valve operation is accurately indicated." When it is impractical to directly observe a valve to determine the accuracy of its position indication, what other positive methods can be employed?

What testing needs to be performed on relief valves whose only function is to protect piping systems from overpressurization due to thermal expansion of trapped fluid?

Many BWRs have experienced difficulties testing automatic depressurization system (ADS) valves. What is the appropriate test method and frequency for these valves?

Many plants have experienced difficulties testing pressurizer power operated relief valves (PORVs). What testing should be performed on these valves and at what frequency?

NUREG 0737 requires plants to have a post–accident sampling system for accident assessment. Post accident, this system performs a safety function to help mitigate the consequences of the accident. However, most of the valves in the system are not Code Class 1, 2, or 3. What inservice testing should be performed on valves in this system?

Operation of the reactor coolant pumps during cold shutdowns precludes the testing of certain valves in the seal water and component cooling water systems. Should the reactor coolant pumps be stopped during each cold shutdown solely to permit the testing of these valves?

Some plants have containment penetrations with multiple valves branching from a common header at the containment interface. Often it is not practical to perform a seat leakage test on individual valves. How should leakage limits be assigned to a group of these valves to assess the operational readiness of the individual valves?

Many safety related pumps do not have an observable or measurable lubricant level or pressure due to the pump bearing design. How should licensees address this situation in their IST programs?

Many pumps are in systems where the system design prevents reestablishing reference conditions during inservice testing. Can variable values of flow rate and differential pressure be used to evaluate the operational readiness of these pumps?

Many pumps, such as deep draft service water pumps, have no provision for the direct measurement of inlet pressure. What alternate methods should be used to determine the inlet pressure to these pumps and subsequently the differential pressure across these pumps?

Some pumps have no provision for direct measurement of bearing temperature as required by the Code. Measurements taken on the casing, which in many cases is the only accessible point, can be affected by a large variety of external factors unrelated to the condition of the bearing. What testing should be performed on these pumps instead of the annual measurement of their bearing temperature?

CONCLUSION

Through the IST program review process, the NRC and their contractors have obtained significant insight regarding the current industry practices as well as the problems associated with inservice testing of safety related pumps and valves at nuclear facilities. Certainly one of the major problems with reviewing IST programs has been insufficient information and an adequate level of detail in the cold shutdown test justifications and relief requests. Another, has been the perception that system tests meet the requirements of Section XI for component testing. Other problems include the lack of published guidance available to licensees preparing IST program submittals as well as a variety of concerns relative to interpretations and implementation of the Code requirements. The greater attention given to IST should result in increased and perhaps more focused dialogue that will lead to solutions to the problems attendant to implementation of inservice testing and result in increased assurance of operability of nuclear power plant pumps and valves.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Premise Generic Letter 89-10 extends the MOV safety-related valve scope to include position changeable valves which would affect the operation of reactor safety systems. Example: Normally open feedwater isolation valves could conceivably isolate BWR high pressure core injection and RCIC systems. Many of these valves were not considered safety related in accordance with the design basis of most plants.

Question Is a similar scope change being considered to the overall valve IST program requirements developed in response to Generic Letter 89-04?

Response 1

On a lot of these issues I cannot speak for the NRC, so I'm giving my opinion in all of these cases. I believe that the valves you identified in your question, if those valves are passive valves and perform no active safety function, will still be considered passive valves for in-service testing.

In that case, the only testing that might be required would be a remote position indication verification or something of that nature.

I think, as far as Generic Letter 89-10 is concerned, they are looking at a way to block out those valves so they will not be mispositioned to disenable safety systems. At least that is what I understand from your question.

Comment

I'll go along with the same interpretation. I think we ought to treat that issue as a 201 type of situation from the old construction days. If a "nonsafety-related component" has a potential to compromise the integrity of the safety-related component, prudence dictates that we are to take care in maintaining that component in such a way that the effect will be minimum. It does not have to be in the IST program, but there should be some way of controlling that component in a reliable fashion so it will be there when you need it.

Question 2

Should equipment required to maintain the environmental qualification of safety systems be considered "required" for accident indication and be included in the IST program? The specific example I'd like to bring up to see if OM- 10 considered it is at Vermont Yankee. The water cleanup system is not in the IST program, except for the valves that isolate it from the vessel in an accident.

However, once it is isolated, check valves are installed in the system in case of a system break to prevent outflow from the system, and that outflow would affect the equipment qualification of systems in the area.

So since it is not a direct accident mitigator, but may affect the qualification of other systems, should that really be in the IST scope?

Response 2

I would think yes.

Comment

I'd have to think about that specific example a little more, but we have required licensees to include that type of equipment in their IST programs.

For example, consider active valves in a cooling water system to an air cooler for a safety-related pump. If you take credit for that safety-related pump, you have to have cooling water to the air cooler, so you do not damage the pump prior to taking credit for its use.

Comment

I think this would fall into about the same situation. If a failure of those valves could compromise other safety-related equipment, they should be included in the IST program and tested to the Code requirements.

Question 3

How does cooling the fuel pool fall under the scope of IST? For most stations this system does not mitigate an accident, as defined in the FSAR. How does the NRC justify the system as being required under the scope of IST?

Response 3

I can't answer that other than discussing a couple of programs. I started at Maine Yankee, which was a B31.1 plant, and could not coordinate the relationship.

I was convinced when I left that if we wanted to get a licensed program, we had to include it. But I certainly concur with you. And you certainly can have an accident from a fuel pool.

Comment

Panel Rather than mitigating a Chapter 15 accident, you can mitigate an accident such as an off-site release of radiation. If you have a problem from inadequate cooling, you can have off-site release of the radiation if you boil the pool down and expose the fuel. That would be an accident in and of itself if you had the right type of failures.

The testing of that system is identified in the standard review plan, NUREG 0800, and the NRC has taken the position that the inservice testing program was the most appropriate vehicle for performing the testing required by that document.

Comment

Panel There may be some additional components in the requirements that require augmented inservice testing. Generally, I think we have discussed what should be in these programs and not. However, I think this question needs some additional communication.

NRC PERSPECTIVE AND EXPERIENCE ON VALVE TESTING

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INTRODUCTION

Testing of safety related valves is one of the major activities at commercial nuclear power plants. In addition to Technical Specification, valve testing is required in 10 CFR 50.55a and 10 CFR 50 Appendix J. NRC inspectors (both resident and specialists) spend a considerable amount of time in following the valve test activities as part of their routine business. In the past, depending on a licensee's organizational structure, a valve could be tested more than three times to verify conformance with Technical Specifications, 10 CFR 50.55a, and 10 CFR 50 Appendix J. The regulatory reviewers were isolated from each other. Licensee test personnel were also not communicating among themselves. As a result, NRC inspectors found that certain valves in the IST program were inadequately tested. The typical licensee response was to say that this valve is exempted from testing under Appendix J. Others would say that the technical specification does not require fast closure of a valve in question

In addition to the above, the inspectors had to deal with exemption requests that were not dispositioned by the NRC. In the seventies there was a gentlemen's agreement to allow the licensee to do the testing in accordance with the exception, without waiting for the NRC approval. Needless to say when the new NRC inspection procedure was issued in March 1989 for implementation, the Regional inspectors had extremely difficult time to cope with the "gray" areas of valve testing.

In August 1987, NRC Region I was reorganized and the special test program section was established to perform inspections in the IST area. This section was chartered to optimize resources and develop a meaningful inspection plan. The perspectives and insights used in the development of a detailed inspection plan is discussed below.

Perspective on Valve Testing

The inspection objective was to determine whether valve testing was conducted in accordance with regulatory requirements and licensee commitments. The regional staff used the following perspectives to develop a detailed inspection plan to accomplish the above objective.

1. *Defense in Depth*

The staff strongly believed that the valve testing program is an integral part of Defense in Depth. Valves at a nuclear plant assure that the three barriers namely clad, Reactor Coolant System and the containment are intact. Improper and inadequate testing of a crucial valve can severely challenge and disable anyone of the above boundaries. Examples are containment isolation valves, pressure isolation valves and various valves in the ECCS systems. The IST program for valves enables a licensee to detect degradation and prevent deterioration during operation. This assures operability with a high degree of reliability. Additionally, it will preclude accidents caused by component failures.

A well established IST program will verify the design features of a valve that are essential for the safe operation of facility. This program will be established by carefully selected and trained personnel under effective supervision. The insights gained on the components from testing will be used to develop quality maintenance program and emergency procedures. Inservice testing of valves would provide a safety net and minimize the incidents that will components the integrity of systems that are required to protect the operators and the public. Inservice testing of valves will also assure additional system such as the ECCS will function when needed.

2. *Design Basis and Safety Related Function*

IST program generally verifies the component's ability to perform the design basis required safety related functions. This requires the test personnel to be knowledgeable about the design basis and safety related function of the valve. Unfortunately, a good many test personnel and NRC inspectors are overly concerned about the operation of the valve and they often forget the safety related design function completely. Therefore the regional staff concluded that the IST inspector should develop a good understanding of the safety function of the valve being selected for inspection.

This will enable the inspector to discuss the safety significance of identified concerns, if any, with the licensees and the NRC management. Unfortunately in the past, neither the inspector nor the licensee personnel knew the design function of the component at the time of the inspection and at times enormous resources were expended by the licensee and the NRC to disposition a matter of little or no safety significance. Our inspectors study the FSAR and the system description to gain this knowledge. This knowledge is especially handy when one judges the full open function of a valve using its ability to pass the design flow. It also helps our inspectors to represent the safety significance of enforcement matters.

3. *Probabilistic Risk Analysis (PRA)*

NRC inspection is a sampling process. We need to select components with significant safety function. Plant specific or generic PRA is used to select components that contribute maximum to the core melt probability.

Failure modes of such components are further studied using failure analysis and the information available in LERs and NPRDs data bank. The information gathered from the PRA leased study will be used to judge the adequacy of technical requirements for testing the component. Selection of components in this manner enabled our inspectors to focus our inspection in potentially weak and vulnerable areas of the plant.

4. *SALP Attributes*

Our inspectors are now trained to review the IST program implementation and make assessments using the SALP attributes. In other words our inspectors assess management involvement, adequacy of technical resolution, enforcement history, responsiveness to NRC initiatives, staffing, training and qualification of personnel and other SALP attributes using carefully selected testing activity.

This usually forms the basis of the inspector's SALP input at the end of the inspection.

In summary, our inspection plan requires the inspector to assess valve testing activities from various angles using safety, reliability and regulatory attributes.

Inspection Methods

Region I used three different types of inspections to assess the implementation of IST programs. The first and the shortest inspection is to send an inspector for a day or two to review IST organization, programs and implementation incidental to another routine inspection.

Next type of inspection is to dispatch a specialist to review selected aspects and components in detail for a week when known problems exist in limited areas.

The third inspection is a major effort which utilizes resources from contractors and headquarters and it reviews several system and assures overall program adequacy.

During the past two years, all of the above inspection were identifying significant safety concerns in Region I.

Our instruction to the inspectors was to assess the valve testing adequacy using the IWV Sections endorsed by the licensee including applicable approved and requested exemption. However, if a genuine safety concern is identified in approved or requested exemption areas, actions were taken promptly to correct the safety concern.

Inspection Experiences

Region 1 inspection effort was instrumental in refocusing licensee attention in the IST area. Since the region did not perform detailed inspections in this area in the early 80's, the licensee attention was diminishing. Each of the inspection conducted to date identified violations or weaknesses. The concern of these violations ranged from uncontrolled IST programs to inoperable check valves that resulted in a civil penalty. The region and NRC headquarters gained first hand information about the problems in the IST area. We also believe that these inspections assisted us to develop meaningful and practical positions in generic letter 89-04.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

No Questions/Comments

PANEL DISCUSSION: THE PURPOSE OF VALVE IST PROGRAMS – AVAILABILITY AND RELIABILITY

J. ZUDANS (MODERATOR), T. HOYLE, R. MARTEL, C. RANSOM, P. EAPEN

Question 1 (Documentation of Deviations)

The new OM standard refers to the proper documentation of deviations. What does the NRC consider proper documentation?

Response 1

I can give you our perspective, as inspectors, when we inspect a plant. We send knowledgeable people, industry experienced people, and as far as I'm concerned, if you can make a logical and technically sound argument to our inspectors, that is definitely good in my books.

We ought to have good documentation. Base the facts on the design basis requirements and the operational requirements of a particular component. If you can do that, no one is going to find fault with your conclusions and assumptions.

Question 2 (Definition of Continued Testing)

OM–10 states that cold shutdown testing shall commence within 48 hours and continue until all testing is completed or the plant is ready to return to power. What does the term "continue" mean: (a) procedure performance testing back to back; (b) one test per shift; (c) one test per day?

Response 2

I don't think this requirement could be stated that in documents such as OM–6 and OM–10, but I think the inspectors will be looking for a reasonable and good faith effort to perform as much testing as reasonably possible.

If you are making a reasonable and good faith effort, I don't think anyone will have any problem with what you are doing. But if you are delaying and dragging your feet, the inspectors will probably be bothered by that.

Comment Has anybody so far incorporated OM–6 or OM–10 into their program? Has anybody done that so far?

(A show of hands.)

When you do that, if you take an exception to portions of the standard, be sure to state what you are doing and how what you are doing is equal to or better than the standard.

Our inspectors will be looking for a document like the tech spec interpretation. A lot of plants have tech spec interpretations in their control room. In an argument between an inspector and operating personnel, the tech spec interpretation provides information to the inspectors. Most of the inspectors, to the best of my knowledge, will go along with the tech spec interpretation. So we can probably make the same thing happen if we have difficulty with certain portions of the OM–6 and OM–10 standard. If you can do that in a prudent manner and if you can put it under the docket, that would be even better because headquarters would get an opportunity to review and comment on it.

Question 3 (Inclusion of Relief Valves in OM–10)

Isn't it the intent of the new OM–10 scope statement to include relief valves designed for component protection, as well as those designed for system protection; that is, in–line pump motor cooler relief valves?

Response 3

I think the answer has to be yes, it includes both of those. If it is required, as stated in the standard, to protect the system or portions thereof, it is required to be tested. So either of those would have to be tested.

Question 4 (Full Flow Testing of Check Valves)

Regarding Generic Letter 89–04 requirements for full flow testing of check valves, acknowledging that there is no Code requirement for accuracy of instruments used to measure check valve flow, is it necessary to know the accuracy of the instruments, whatever it may be, and compensate the flow for the maximum design accident condition referred to in the generic letter by this amount to ensure that the check valve is passing the required flow given the worst case accident condition?

Response 4

Panelist I'm not sure the NRC is going to like what I say, but my position would be no, because you are trying to show the valve is full open. You are not trying to prove that the flow meets any kind of a tech spec limit, which is the intent of these tests.

Panelist On the other hand, if your error in the accuracy of the flow measurement is significant enough to call attention to the assumption that you are meeting the accident flow, I think that would require some scrutiny, but I don't know what that inaccuracy is.

Panelist I agree. If your accuracy is close to the accuracy limits included in IWP for testing pumps, I don't think the NRC would have any problem with using those instruments to verify a flow-through valve. However, if you are talking about an instrument accurate to plus or minus 10 percent, then you are right on the borderline for the flow through the valve. I think there is a concern because there would be a serious question whether the valve is, in fact, opening sufficiently to perform its safety function.

Panelist I'd like to add one more thing. It looks like the industry would like the NRC to be prescriptive. As it is, I think the NRC has reasonable regulation on the books, especially in 10 CFR 50, Appendix B, prescribing what ought to be your test control.

You are to develop a procedure that is appropriate for the test and to make sure that adequate controls are in place. If we abide by that, the responsibility is strictly on the industry, and you could regulate yourselves. If the NRC tried to regulate a very prescriptive standard, we may cause the industry and the NRC a big problem. So my plea to you will be to stay within the confines of your own quality assurance requirement. Make up your own mind and determine what it is that you have to do with that particular instrument and have justifiable and defendable documentation on what you're doing.

If you did that, the regulators would be happy, and you, as the utility, would also be happy because you would know exactly the whys and wherefore of the test and the accuracy of that instrument. And on top of that, you would know why you are doing it and why it is relevant with respect to design

Panelist I have one more comment on that. When we are dealing with instruments, we have to recognize that they have some inherent characteristics, like these flow meters. We are not going to get a 2 percent flow meter all the time. We may be lucky to get 5 or 10 percent, depending on the piping configuration. There may not be enough straight pipe to get good flow. Even with a questionable amount, we are trying to show that the equipment is operating acceptably. It was never the intent of the code to try to prove safety performance, and I think we need to be careful about this. Otherwise we are going to create a situation where we could never satisfy everybody and would go well above what the Code and standards were written to.

Question 5 (Parallel Branch Path Full Flow)

Reading your personal perspective in full flow testing of check valves gives the impression that you, as well as the ASME working group, feel that verification of full flow of parallel branch paths is not required. Our technical specifications are strict regarding the flow through these parallel lines, ECCS, and the verification of flow if there is a change to the system; that is, throttle valve repositioning, pump impeller change-out, etc.

Do you or the ASME working group members have justification for not measuring branch line flow rates, and instead measuring full flow header flows? If not, where do you get your basis for your opinion on branch line flow measuring?

Response 5

Number one, I didn't try to mislead you. Basically it is something that we all have to do. Generic Letter 89-04 says knowledge of only the total flow through multiple parallel lines does not provide verification of flow rates through the individual valves and is not a valid full stroke test. This is something that we, as a utility, are wrestling with also. It all balanced, and that is good enough. We don't really want to charge, but we will be putting in flow measuring devices, as I'm sure most of you will. No one said that this was going to be easy. If I mislead you by saying that I don't want to do it or we are not going to do it or we have justification, that is not true.

We are not doing it now. We are using the old method; balance it and from loop to loop verify that there are safe margins. We will be pu' g in some sort of flow device.

Panelist I don't believe the ASME working group ever intended that you do not measure individual branch flow lines if you need to measure those branch flow lines.

I think this issue is one that affects a lot of people. Specifically, it may affect people who have parallel branch lines that can only be measured by the header flow because they have no flow instrumentation and ability to isolate different parallel paths of the loop.

I do have a problem verifying that my check valve is passing the right accident flow in some plants.

Panelist In some cases the accident analysis takes credit for injection through all the parallel branches, and if only the header flow is measured, you cannot verify that the system meets the safety analysis. It is not a case of redundancy or loss of redundancy. It is a case of not meeting the safety analysis. In most cases, it is extremely important to verify the flow through each parallel path.

Question 6 (Disassembly in Lieu of Testing)

Is a sample disassembly and inspection of check valve an acceptable test for back flow of check valves?

If so, can an approved relief request which already permits another method for testing such a leak be revised to allow a choice between the two methods?

Would this relief request be preapproved, based on the positions in Generic Letter 89–04?

I can unequivocally state that I have been told that if you write a new relief request it will not be approved by the generic letter. Only the old ones are approved.

Response 6

Panelist For any positions that fall within the discussion of attachment 1 of Generic Letter 89–04, relief can be approved.

As discussed, for a relief request that does not fall within the guidance of attachment 1 of the generic letter, if you modify the relief request, it will have to be submitted for review.

In this case, the NRC's position on sample disassembly is that it can be used to demonstrate the reverse closure because you are observing the valve disk on the seat. It cannot be used as leak testing to verify the integrity.

In other words, if it is a category A valve, the disassembly does not take the place of leak testing, but can be used to verify reverse closure.

Attendee (NRC) I'd like to amplify that. There is a hierarchy articulated in our "road show" meetings concerning sample disassembly or the process of disassembly. We don't really do this as a test. It is a maintenance procedure, and we received a number of questions about whether or not disassembly could be used in lieu of full flow testing when full flow testing was possible.

We said that disassembly is not a one–for–one substitute in situations where the test can be performed.

We articulated that both in terms of forward flow and reverse flow testing.

Question 7 (Relationship of IST to ISI)

Is it the opinion of the panel that the inservice testing program is totally independent of the inservice inspecting program, which is prepared using the Regulatory Guide 1.26 and NUREG 0800 guidelines?

Specifically, do ISI boundaries include all valves and pumps to be included in the IST program?

Response 7

Panelist I think our feeling on this is, yes, they are independent, and the criteria are not necessarily the same criteria as specified in the standard or the Code.

Panelist Unanimous agreement.

Comment

Attendee There is not a unanimous view of your stated purpose of IST; that is, determining whether degradation has occurred. This is an added benefit if you can get it. Verification of operational readiness is the basic purpose.

Panelist Some of the tests identified by Section XI were specifically designed for detecting degradation, like the stroke time limits on power operated valves. The leak rates of category A valves provide a trending device to project corrective action based on the increase in leak rate. That is looking at degradation.

Also, some of the hydraulic measurements for pumps (flow rate and differential pressure) are measured to detect hydraulic degradation of the pump, while vibration is measured to detect mechanical degradation in a pump bearing.

Degradation is an essential part of the IST testing. I don't think operational readiness is the only area of concern.

Question 8 (Cold Shutdown Testing of Pumps and Valves)

Why the difference in philosophy between pump and valve testing?

IWP 3400 and OM-6 stated that pump testing during shutdown periods is not required. IWV and OM-10 do not address continued testing of valves that are normally tested during operation during plant shutdown periods.

Response 8

Panelist I don't think that is correct. Let me read what this section from OM-10, Part 10 says: "Valves and systems out of service. Where a valve and a system declared inoperable are not required to be operable, the exercise in test schedule need not be followed. Within three months prior to placing the system in an operable status, the valve shall be exercised."

The same philosophy is used for both pump and valve testing.

Attendee But pump testing specifically says that you don't have to test a pump when the plant is shut down, whether it is required to be in operation or not, where the valve testing does not address that.

Panelist It says, in a system declared inoperable and not required to be operable, the exercise in test schedule need not be followed. Isn't that saying the same thing?

Attendee No. The shutdown cooling system is required to be operable when you shut down. but the LPSI pumps that are tested during operation would not be required to be tested when you shut down.

Panelist For the valves obviously, they would. Where does it say that pumps are not required to be tested?

It says that pumps in systems out of service need not be tested, but if the pump is in service, you need to test it, and that's still in service.

Basically, if it is in regular use, you would have to run the tests. I don't think you can get out of testing. That is certainly not the intent.

Panelist Maybe we need to clarify that issue, and I would invite you to send a letter to the chairman of the working group asking for a clarification in that area. I think that is a very positive comment that you interpret something a little bit differently than what we intended.

Panelist IWP 3400 states, after recommending a schedule of every three months during normal plant operation, that this test frequency be maintained during the shutdown periods if this can reasonably be accomplished, although it is not mandatory.

I think that the NRC hopes that you do maintain that frequency.

Question 9 (NRC and Licensee Manpower)

The NRC has indicated that they evaluate the limitations of their resources in making decisions as to where and in what areas inspections will be conducted.

Does that same consideration apply to evaluation of the licensee's resources to respond to NRC concern?

Response 9

No. Remember, if I understand the intent of the question, we do consider resources. That is one of the factors that goes into where and when we are going to inspect.

That is by no means the predominant factor. The primary responsibility is yours. We are not going to do a 100 percent inspection. You have to make that decision of what your resources will allow in responding to our initiatives, in responding to our generic letters, in responding to our information requests, and in negotiations with the project manager. I think that is a reasonable thing to do within limits and constraints.

For example, in responding the Generic Letter 89–04, resources will have to be expended. It is reasonable to state to your project manager that other safety-related work must be done that will cause a reasonable delay. It doesn't mean it is going to be accepted, but I do think it is something reasonable to approach the NRC with.

Question 10 (Check Valve/Relief Valve Classification)

The subject of classifying a check valve or relief valve as passive appears to be interpreted differently at different utilities?

Response 10

Panelist My interpretation of a passive valve is one that need not change position to perform a safety function. A passive valve is used only during the shutdown periods for maintenance or during startup, such as a bypass valve for the feedwater system. Once the plant is operational, no accident analysis accepts credit for it changing positions.

However, if a valve is normally in one position but periodically, during operation, changes to the nonsafety-related position, an accident would cause that valve to change positions to perform its function; therefore, it is not a passive valve. It is an active valve. Just because a valve is normally open and a safety position is open, doesn't make the valve passive. If the valve may be required to change positions to perform its safety function, it is an active valve.

Panelist My opinion is quite similar, in that I look at check valves in this kind of a program. Even though they don't have an active safety function, they are going to open normally. Therefore, they almost always have to be considered active. Even if the only time they are used is during the shutdown period, they are still active, unless they are in their proper position before you start up and they are not going to move for the duration.

Panelist One example in our plant is the breathing air line going through the containment that has a check valve. That is closed before the plant starts up. The containment isolation valves are check valves that are basically passive for the time that the plant is operating. They will not perform any safety function in the accident, except to stay closed.

Question 11 (Stroke Testing Multi-Solenoid Valves)

For valves with multiple safety-related solenoids, should the valve be stroked quarterly using all safety solenoids, or is engineered safeguards testing at each refueling outage sufficient? Note, safeguard testing times the valve with all solenoids.

Response 11

Panelist My feeling is that you would have to request relief to do anything other than test with all solenoids.

Panelist I agree. If you can demonstrate an impracticality or hardship for testing all solenoids quarterly, then there is a good chance that relief could be granted. But otherwise you would have to either perform the full testing quarterly or get approved relief.

Question 12 (NRC Acceptance of OM–6 Parameters)

In the past, EG&G/NRC review of IST programs addressing pump testing to OM–6 parameters has not been accepted.

One, what is the status of NRC view of OM–6 parameters?

Two, is it still based on individual justification and at lower parameters?

Response 12

Panelist The staff accepts OM–6 for the vibration measurement program. As far as the other limits (differential pressure and flow), the staff does not accept those limits. You have to request relief based on individual pump requirements stated in Section XI of IWP. You cannot just go to the OM–6.

Other points of difference between OM–6 and Section XI IWP will have to be addressed individually. The staff has indicated that they will accept the vibration measurement program from OM–6.

Question 13 (OM–10, Section 4.2.1.8, "Stroke Time")

Is the stroke time acceptance criteria of OM–10, Section 4.2.1.8, intended as alert or maximum limit criteria?

If it is maximum limit, why is the option given to analyze within 96 hours or declare inoperable immediately? Generic Letter 89–04 states the requirements to declare inoperable immediately.

Response 13

Panelist I can answer that if you give me a little bit of help. I want to make sure I am answering the right questions. No, this is not intended to be a maximum limit. These are the limits of plus or minus 15 percent, plus or minus 25 percent, etc. These are intended more as an alert limit, and that is why there is a 96 hour period to analyze it.

There are two types of stroke times. One is called the limiting stroke time, or the maximum beyond which you must declare a valve inoperable. If you go beyond that limiting stroke time, you must declare the component inoperable.

The acceptance criteria are stated in 4.2.1.4, as the limiting stroke time. The important thing to note is that limiting stroke time is usually a bit higher than 4.2.1.8 criteria.

Attendee Can you picture an analysis that says the valve is inoperative on stroke time, but the stroke time does not exceed the maximum time?

Panelist Yes. Your question asked about 4.2.1.8, which is not the maximum limit. 4.2.1.4 is the maximum limit. If you exceed that 4.2.1.4, you are right. The valve is declared inoperative immediately, as stated in 4.2.1.9. But if you exceed that 25 or 50 percent from a reference value, but less than the maximum value that is set, then you have the 96 hours.

Attendee At that point, you can do an analysis like the analysis already done, saying that it is okay up to the maximum limit.

Panelist That is correct.

Attendee I can't picture a case where we would say that it exceeds those ranges, but our maximum limit is higher than that, so it is inoperative. We are already saying that we looked at the time and this is our maximum limit. I am having trouble understanding what the analysis is going to accomplish.

There are many plants whose maximum limiting stroke time is much, much higher than the normal stroke time of the valve. I'm talking about a 15–second valve with a 60–second limiting stroke time. Under those conditions you would have difficulty justifying 60 seconds.

Panelist But that is more of a question, according to the generic letter, why the maximum limit is so high.

Panelist You could still have a limiting stroke time that is slightly higher than 50 percent or 25 percent above your reference value. For example, a 12–second valve could have a 20–second limiting stroke time, and at that point if 18 seconds is the stroke time, you might have to analyze, but you wouldn't have to declare it inoperable.

The philosophy behind this is we want to allow some time for testing techniques or other things that may influence results, and that is the idea. First of all, you have met your maximum limits. You have already told me that. You have run your test, you have met your maximum limit. Therefore, you say I know the system is still operable, but I do have a problem. The time allows you to rerun the test and to look at the testing results to see if that performance is acceptable. You might say yes, marginally so, or you might say no, it is unacceptable, and then you have to declare it inoperative. It allows you some time.

Attendee So it is just more or less intended as—

Panelist As an alert.

Attendee Making you look at it seriously.

Another Attendee When we did this change? As I remember it, the analysis was not that you were still below your maximum stroke time, so you were okay. The analysis had to answer why this valve changed by 25 percent.

So you don't have an analysis. Just because the stroke time is less than the maximum stroke time doesn't mean an analysis, though. You have to say why this changed. That is the analysis.

So it's a different analysis than what you're talking about. It's not the analysis of the system and the valve in the system. It's the analysis of why did the valve change.

Attendee The reason I asked is that the maximum limits should be based on a reasonable deviation. But what is a reasonable deviation?

You are setting a limit based on a reasonable deviation. If it exceeds that limit, it is inoperative. It should be one limit. We are also adding an alert range now.

Other Attendee It is not an alert range if you cannot explain why it changed. They are not maximum limits because you can use an analysis to say why it changed. Maybe your analysis says there was no flow when the baseline was done and there is flow now. That may be a good reason for change.

Panelist Essentially there are three limits. You have a system operability limit. If you bought a five–second valve that has 60 seconds to change position, you have a system operability limit of 60 seconds. If you have a valve that normally strokes in five seconds, you have a plus or minus limit to determine if it is okay.

Then you have a reasonable limit, which could not be 60 seconds for a five–second valve. It might be 10 seconds, or whatever. You have a reasonable limit based on the normal data scatter, and if you exceed that limit you conclude that the valve is seriously degraded and you should declare it inoperable. So you have three limits.

Question 14 (Quarterly Stroke Test)

In effect, the O&M document drops the monthly testing by putting it in an alert range, if I understand the sense correctly. I have 96 hours to either set a new limit or do an evaluation, if the stroke time is okay, and then test it next quarter?

Response 14

That appears to be true.

Comment

On the last comment about dropping the monthly testing, for the increased frequency testing, that was, in fact, the case. Let me read from the white paper. We talked about changing that.

It says, "If a valve does not meet the acceptance criteria of 5218, a change has occurred. If the change is acceptable, the analysis will show that. If not, then corrective action should be taken. An increased frequency of testing will not improve the situation and may cause additional damage to the valve." So we definitely did pull that out, and that is the reason.

Question 15 (Parallel Path Full Flow Testing)

For the case of parallel path full flow testing, can you comment on the use of analytical techniques to demonstrate that total flow can provide adequate indication of individual branch line performance; that is, degradation in any one branch line can be detected using total flow?

I can tell you that we attempted that at my utility, but it was not easy because we tried to sense the change in flow from possibly one check valve being partially closed and the kind of reaction from that pump.

I would say that it is not going to be fruitful in my mind, but I will pass it on.

Response 15

If you did provide empirical data that demonstrated you could predict or make that determination accurately, I'm sure the NRC would review that data.

But it would be very difficult, and to date no one has successfully provided the adequate data or information to demonstrate this.

Question 16 (Functions of Category A/C Valve)

Does an AC valve have two safety functions: one, to shut on cessation of flow, and two, to be leaktight?

Therefore, does this valve require testing in the closed position quarterly and quantitative leak testing at least every two years?

Or does this valve have one safety function, being leaktight; which would require only quantitative leak testing at least every two years?

Response 16

That depends on whether it is an active or passive valve. If it is an A passive, then it has one function, and that is to prevent the reverse flow—or to prevent flow through the valve. It only need be leak tested to verify its category A function.

However, if the valve is an active valve and has an active safety function other than just this category, then you must exercise it quarterly and leak test it every refueling outage, or as required by the 10 CFR Part 50 Appendix J program.

PLANNED SESSION PAPERS

PART 4: VALVE TESTING TECHNIQUES AND LIMITATIONS

IN SITU TESTING OF MOTOR–OPERATED VALVES IN NUCLEAR POWER PLANTS

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ABSTRACT

This paper presents a perspective of the status of in situ testing of motor operated valves in nuclear power plants. The objectives of in situ testing are discussed. A short history of in situ testing of motor–operated valves in nuclear plant applications is offered. Recent developments regarding in situ testing are discussed followed by a perspective on needed research and development.

IN SITU TESTING OF MOTOR–OPERATED VALVES IN NUCLEAR POWER PLANTS

Motor–operated valve (MOV) failures and the subsequent consequences have been concerns of the Nuclear Regulatory Commission and its staff for some time. Reports, Bulletins, Circulars, and Notices dating at least as far back as 1972 document failures and resulting NRC recommendations to the nuclear industry. The critical problem is that MOVs have been experiencing high failure rates when called upon to operate against pressure or flow. The major root–causes include:

1. Defective parts or sub–components
2. Misadjustment of switches or components
3. Poor coordination, on a life cycle basis, of design, installation, maintenance, and testing
4. Vague maintenance, test, and failure analysis procedures
5. Ignorance of the design basis parameters that control MOV size and adjustment.

Motor–operated valves are to be found in most fluid systems of every commercial nuclear power plant. The vast majority of MOV operators in the United States are made by Limitorque Corporation of Lynchberg, Virginia. Most of the remainder are made by Rotork Controls Incorporated of Rochester, New York. In most cases, identical MOVs are installed in the parallel trains of safety systems. Where two MOVs are installed in a train of a particular safety system to act as two valves in series they are usually identical. MOVs in parallel trains are usually served by identical power and control systems and have similar operating environments. These MOVs are serviced by the same personnel and procedures for each particular plant. Therefore, MOVs are especially subject to generic and

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common mode problems. Since the reliable positioning of valves in a nuclear power plant is critical to the safe and economical operation of the plant, generic MOV problems are a serious concern.

The MOV operators in nuclear plants are close adaptations of a commercial design originally developed in the late 1930s. The design has evolved, but the basic concept of a torque-limited motorized gear box remains. The equipment design has not changed radically over the years. Several problems arose in the application of the operator design to nuclear power plants, some of which were not recognized for some time thereafter. Problems that are inconvenient in the commercial field are intolerable in nuclear power plants. For example, spring pack relaxation has only recently been recognized as a significant problem in the industry. Improperly loaded spring packs can disable an operator or cause damage to a valve.

MOVs are subject to loads and stresses from the control systems and power systems that serve them, as well as from the fluid systems in which they serve. MOVs are subject to partial damage or degradation that will leave them operable for normal or no-load situations, but may cause failure at design basis demands.

MOVs are somewhat unique as a class of components, because they exist at a junction of several systems and other components that must all work in order for the MOVs to function properly and without damage. There are a number of effects that can operate either individually or synergistically and that can prevent an MOV from functioning, either partially or fully.

In situ testing is only one aspect of a comprehensive program to assure the operability of MOVs under all required conditions. The role of in situ testing of MOVs is burdened by several factors. First, the testing that can be done in a nuclear plant is limited by safety and plant operational considerations. Second, there is an implicit assumption, justified or not, that a particular component that is in place has been properly designed, fabricated, and installed.

In situ testing may indicate that a MOV is deteriorated or inoperable, but may not be able to provide all of the information needed to verify design basis operability. If the assumption of design basis operability is to be validated by in situ testing, then appropriate methods must be available. The methods may be somewhat different than those used for the detection of degradations or other anomalies. Thus, the in situ test methods should provide plant operators with techniques that will verify that each MOV has the ability to meet design basis requirements and allow diagnosis of anomalies as well. Such in situ testing may not be possible, depending on a number of considerations, some of which are discussed below.

In spite of these limitations, the NRC staff has focused attention on in situ testing primarily because virtually all of the components are in place and operational. Initially, the focus on in situ testing developed when it was recognized that the then existing MOV operability test was inadequate and would have to be modified.

The assumption was made, and still remains to be completely proven, that in situ testing can be used as the primary tool to maintain assurance of design basis operability. This assumption was bolstered by the emerging signature analysis techniques that made it possible to diagnose misadjustments, internal damage, or wear without extensive disassembly. The assumption was weakened, however, by results of recent full scale blowdown tests that were recently conducted under NRC sponsorship. The blowdown test results indicate that some of the extrapolation techniques that were used to predict MOV operability under design basis conditions may be unconservative. This may be due to the limitations of the existing diagnostic equipment as well as the variables associated with converting torque to thrust in the MOV operator. The design basis conditions modeled in the NRC blowdown tests were extremely severe. Signature analysis diagnostic techniques, as they evolve, may provide reliable extrapolations for less severe conditions. Certainly, signature analysis techniques are valuable to maintain normal service operability of MOVs.

A proper in situ test of an MOV should provide objective assurance of future operability under the required conditions. The ASME Code (Section XI) stroke-timing test, that is the test mandated by 10CFR50.55a(9), does not provide such assurance. That test indicates that a MOV may be capable of moving for a particular stroke, but provides little information about future operability. The Section XI stroke-timing test is almost always conducted under no-load (no flow or no differential pressure) conditions. Very little information about wear, misadjustment, excessive loadings, broken parts, deteriorated parts, etc., can be gathered from the stroke-timing test. In several instances MOVs have been left inoperable after a stroke-timing test and only found to be inoperable when called upon to function later. Such a situation is obviously unsatisfactory but, until the recent development of signature analysis techniques, there were few in situ testing options available. Disassembly is not a particularly attractive option because the degraded condition might not be detected and the MOV could be reassembled incorrectly.

The advent of a system to record and allow subsequent detailed analysis of pertinent MOV electrical and load parameters was truly innovative. The recognition that such data could provide insights into the misadjustments and deteriorated conditions that might occur in the working mechanisms of an MOV was a significant milestone.

Signature analysis diagnostic techniques provide unique insights into the on-line performance characteristics of an MOV and allow detection of wear, deterioration, or misadjustment without major disassembly of the equipment. The technology is developing rapidly. For example, on-line monitoring of MOV operating parameters is expected to become available in the near future.

The first signature analysis diagnostic system for MOVs became available just about the time of the Davis-Besse incident that prompted the development of *Bulletin 85-03* by the NRC. That signature analysis system was originally intended to serve as a diagnostic tool for misadjustments or degraded conditions. The system was adapted by the vendor to accommodate the need to verify design basis operability by means of an in situ test that would allow extrapolation from the conditions that were encountered at the time of a particular test to design basis conditions. The basic assumption was that a linear relationship exists between thrust, torque, and motor load. A direct, if not perfectly linear, relationship has been verified to exist between motor load and operator torque. However, the relationship of torque and thrust, as applied to motor-operated gate valves, is not well understood. The usual practice in the industry has been to provide sufficient torque to envelope the losses that occur in the operator when it converts thrust to torque. It was discovered that the losses may not have been conservatively estimated in the usual engineering methodology. The various diagnostic system vendors are now developing strategies and hardware to measure thrust directly. There appear to be other factors such as the number of strokes and time between strokes that affect the thrust developed by a MOV. Again, it still remains to be shown that required thrust can be reliably extrapolated from a test conducted at less than design basis conditions (either in situ or prototype) because of the variability of the thrust developed by the operator. Factors such as valve stem design, valve trim material, number of strokes, time between strokes, temperature, wiring size, voltage, stem lubrication, and spring pack adjustment, among others are all pertinent. Load duration and intensity may be a major consideration. The individual and combined effects of such conditions are not completely understood.

A number of safety-related MOVs are ball, plug, or butterfly types and, as such, do not require the development of thrust in the valve stem in order to operate. These types of MOVs do not depend on the adjustment of a torque switch in order to achieve proper closure position of the valve. The majority of MOVs covered by *Bulletin 85-03* are gate or globe valves. These MOVs usually develop thrust in the valve stem during operation. The extension of design basis operability verification to all safety-related MOVs is expected with the forthcoming publication of a NRC generic letter. This will place several additional MOV types under closer examination and it is expected that problems other than those that have been previously identified will be brought to our attention. These new problems may require additional or modified testing and evaluation schemes.

Alternatives and supplements to in situ testing include bench testing and prototype testing. Environmental qualification tests provide a data base of prototype test information. The environmental qualification tests do not seem to provide much information about testing of the complete MOV as an operational unit under design basis conditions or, for that matter, any conditions where the MOV is subject to pressure or flow loads. Other prototype tests include those performed by valve manufacturers under various ANSI standards, the EPRI/Marshall PORV block valve tests, a number of proprietary tests performed in the U.S. and other countries, and the tests conducted by the NRC as part of the resolution of Generic Issue 87, "Failure of HPCI Steam Line Without Isolation." All of these tests may provide some needed information, however the detailed information needed to predict design basis behavior of a particular MOV will probably not be conveniently available.

Bench tests refer to tests performed on an MOV that is removed from its installed location, or perhaps prior to installation. Such tests may be conducted by a valve vendor prior to shipment, or by a licensee prior to installation, or as part of a maintenance program. One recent innovation is the use of a device to simulate force and/or torque on a valve stem in order to adjust and load test the operator. Several such devices are now being used in the industry. Even if the loads do not duplicate design basis load conditions, the use of such tests provide some insights about the behavior of the operator under load. Use of such testing, in parallel with use of suitable diagnostic equipment, is far superior to the Section XI stroke–timing test.

The action statements in *Bulletin 85–03* were focused primarily on switch setting adjustment. The implication was that switch adjustment was all that was necessary in order to assure design basis operability. The NRC generic letter that extends the scope of *Bulletin 85–03* to all safety-related MOVs contains virtually the same emphasis in the first few action statements. However, it is now known (due in part to industry actions taken in response to *Bulletin 85–03*) that design basis operability is influenced by a number of other factors besides switch settings. Three of these items are of particular concern. First, MOVs may not be sized properly to meet design basis requirements. It became apparent to NRC that many licensees did not have complete design basis information on all safety–related MOVs. Second, the steps taken in an effort to assure design basis operability might compromise the operability of the MOV at conditions other than those considered for the design basis. For example, torque switch settings made to assure MOV closure against design basis flow could, under certain conditions, cause damage to the valve or operator at no–load conditions that may be encountered, for example, during a stroke test. Third, a number of misadjustments, degraded conditions, material problems and design deficiencies not related to switch settings have been identified.

All of these items are at least mentioned in the new NRC generic letter; however, the guidance provided for resolution is necessarily vague. The problem of determining what capability is needed in order to meet design basis force requirements should be fairly straightforward. It may be somewhat complicated and expensive to make the needed hardware changes in the plants. The competing demands of design basis operability and normal or no–load operational requirements may also not be easily resolved without hardware changes and could be difficult to resolve at all. The problems associated with misadjustments, material deficiencies and degraded conditions will require excellent coordination within the industry, as well as education of all concerned. Finally, it will probably be expensive for the industry to research design basis operability parameters, particularly at older plants, but there is no alternative to this effort.

The NRC staff will soon produce a Temporary Instruction to assist inspectors in making uniform judgements about licensee compliance with the recommendations of the generic letter on MOVs. In addition, the staff plans to sponsor workshops on the NRC generic letter. NRC staff members may attend industry sponsored meetings and workshops if invited. The efforts by NUMARC, EPRI/NMAC, INPO, licensees, and various vendors have produced a large amount of information in a very short time concerning the particular problems of motor–operated valves. All of these efforts represent an educational process for the industry that must continue if the problems described above are to be solved. A coordinated effort involving the use of shared information might prove to be cost–effective for the industry as a whole. The problems associated with the production and use of proprietary information could be difficult to overcome, however. The industry would also profit greatly by an increased awareness by utility managers of the technical issues associated with in situ testing and maintenance in general, as well as some awareness of MOV operation and associated problems in particular.

The results of the recent research on MOVs, as well as the increase in meaningful testing, can be expected to suggest generic hardware improvements in not only the test equipment but in the operators and valves as well. The NRC efforts under the Aging Research Program and toward the resolution of Generic Issue 87 have provided some useful data. Improvements in lubrication technology, valve trim materials, operator mechanical operation, and motor design might be reasonably expected from future industry efforts. Further, it has already been observed that the industry efforts on maintenance programs have resulted in increased coordination of MOV in situ testing and maintenance. The competing demands of design basis operability and normal or no–load operability remain to be resolved. Testing methods that will demonstrate design basis operability of installed equipment need to be developed and improved.

This paper was written from the point of view of one who is somewhat familiar with the particular problems and history of motor–operated valves. For those who may be interested in a description of the equipment and some of the various problems associated with MOVs, a short bibliography is provided.

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ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

When will the temporary instruction be produced to assist inspectors in making uniform judgments about licensee compliance with Generic Letter 89-10, and will it be available to licensees? That is the first part.

The second question is when will NRC-sponsored workshops begin?

Response 1

The second part of the first question is, "will the instructions be available to licensees?" Absolutely. All of our NRC documentation is available to the public.

We discussed the timing of the temporary instruction when we produced the generic letter. We decided that it should be produced in about a year from the issue of the generic letter. As you may or may not know, the timing listed in the generic letter is that licensees are asked to reply within six months of receiving the generic letter as to when they will or will not comply with what it requests. And their program is to be in place and operational within five years. Their outline is to be in place within a year. We think that if we have the instructions out within a year, that is reasonable. We will be glad to hear from industry if they think it isn't.

Comment

We have not completed formal plans for conducting the workshops. When we are at that point, we will put the information out in the Federal Register, much as we did for the "road shows" on Generic Letter 89-04.

Our current plans, though, are for the end of September. Keep checking with your project managers, who will issue the information, as well as the Federal Register.

Question 2

What kind of adjustments do you mean when ensuring that the high loads required to overcome maximum design basis conditions will not damage valves under static no-load conditions?

Response 2

The valve manufacturer has an obligation to furnish a valve that will withstand a certain force. As I understand the way these things work, the valve manufacturer chooses or has the actuator manufacturer choose an operator that will close the motor operator under the worst case conditions of flow and pressure. It is adjusted so that the operator, when closing under no-load, will not exceed the forces that the valve can withstand.

There may be valves in the plant that could be damaged, if they were closed under no-load conditions using the full force of the motor operator. If you adjust the force that the motor operator delivers so that it will not damage the valve, it is possible that it will not close under design basis conditions of flow. That has to be determined for each particular motor operator.

CHECK VALVE DESIGNS AND DISASSEMBLY FOR THE PURPOSE OF IST

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ABSTRACT

The objective of this paper is to discuss problems encountered by the Nuclear Utilities when faced with check valve disassembly. INPO Significant Operating Experience Report (SOER 86–3) was developed in response to the increased failure of check valves important to plant operation. Most recently the NRC has issued Generic Letter 89–04, "Guidance on Developing Acceptable Inservice Testing Programs." This generic letter re–emphasizes the importance of compliance with 10 CFR 50.55a(g) and highlights areas of inadequacies in the current scope and methods of testing. The NRC in the text of this generic letter clearly establishes disassembly and inspection as a preferred alternative when full flow testing of check valves is not practical. Additional clarification is presented on the inspection criteria during disassembly and sampling techniques. With greater emphasis being placed on utilities to implement SOER 86–3 recommendations, Generic Letter 89–04 and ASME Boiler and Pressure Vessel (B&PV) Code Section XI IWV testing/inspection requirements, several areas of difficulty have been identified. To address these recommendations/requirements each utility is attempting to develop its own check valve program.

This program approach identifies the necessity of verifying proper check valve application and establishing an adequate preventive maintenance program. An integral part of this program would include check valve testing. One of the most common methods for testing check valve performance has been disassembly and inspection.

This paper will outline the methods and problems associated with check valve disassembly and inspection.

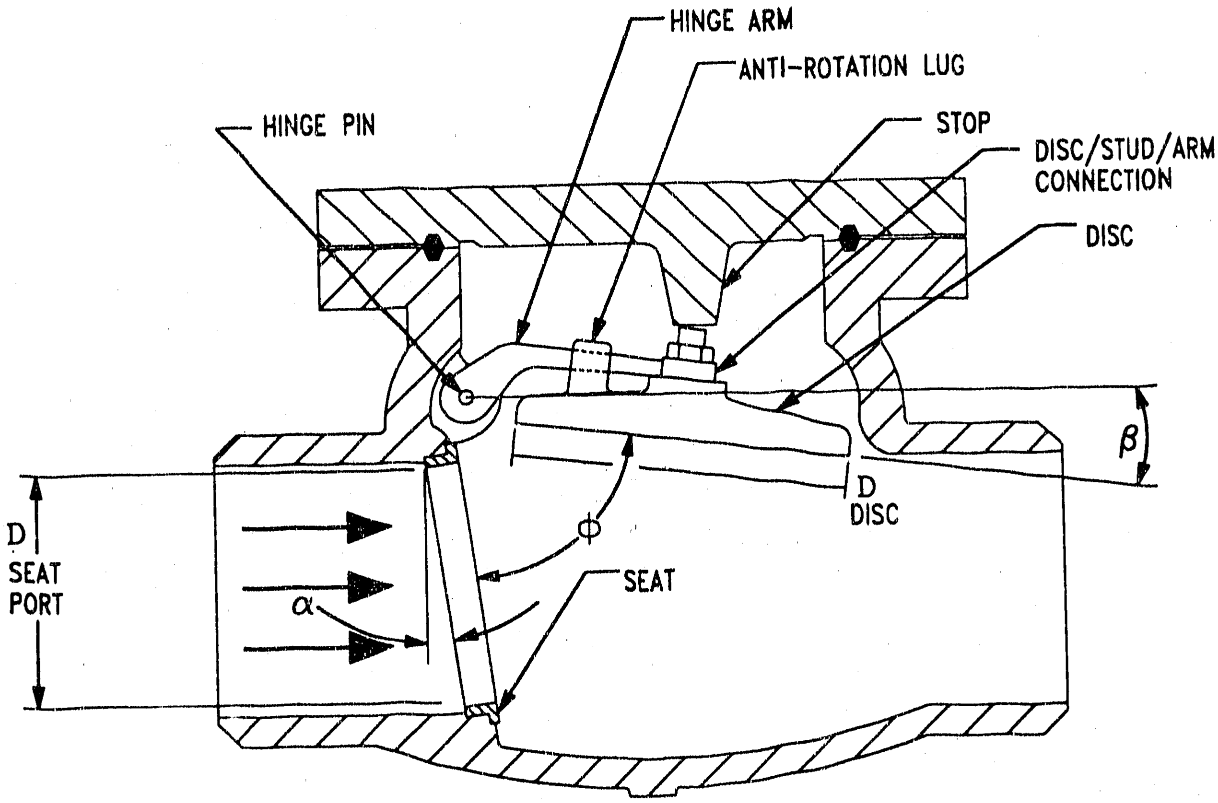
INTRODUCTION

Until the issuance of, and events contributing to SOER B6–3, little attention outside of the IST program was given to check valve testing, inspection, maintenance programs, and design applications relative to other valves used in Nuclear Power Plants. The check valve failures in 1985 which resulted in the INPO review and publication of SOER 86–3 caused an increased awareness of the importance of check valves in the industry. Prior to these events, power plants had been required to perform testing and maintenance activities on check valves covered by the Appendix J and ASME B&PV Code Section XI Programs. However, it was not until this new, increased awareness that plant check valves were given the same priority as other vital components such as motor operated valves (MOVs). The industry response to the check valve issue was to initiate an effort to develop a check valve program on an individual utility basis.

A comprehensive check valve program involves long term testing, inspection, and maintenance to ensure operability and facilitate trending degradation of the valves. As a minimum, utilities have been including safety related check valves and those check valves that are important to plant operations in the program.

The catalyst for including a check valve for disassembly has been an application review, identification of valves with a high incidence of failure, or the inability of a system configuration to allow full flow (i.e., full–stroke) testing. Verification of full–stroke for check valves is a requirement of ASME B&PV Section XI IHV. Disassembly and inspection has become a vital element of the check valve program to trend degradation and verify operability when other methods are not possible or practical.

Disassembly and inspection provides the most positive means of determining check valve condition. Potential problems of valve flutter, loose retaining fasteners, improper disc orientation, corrosion, etc. can be observed (See Figure 1). Unlike non–intrusive methods of inspection, disassembly provides a visual benchmark of incipient degradation and wear.



α = SEAT PLANE TILT WITH RESPECT TO VERTICAL

β = DISC ANGLE WITH RESPECT TO HORIZONTAL

$\theta = (\alpha + \beta)$ = TOTAL IMPINGEMENT ANGLE

ϕ = DISC OPENING ANGLE

D = DIAMETER

Figure 1. Typical swing check valve.

DESCRIPTION OF PROBLEM

Since it has been documented that check valves have caused forced shutdowns in nuclear power plants, it is important to monitor check valve performance/degradation to preclude potential failures. ASME B&PV Code Section XI IWV requires that utilities endorsing these requirements perform quarterly full–stroke verification of all check valves included in the IST program. Full–stroke testing can be verified by passing the maximum required accident flow through the valve, but occasionally this is not practical due to plant operating conditions or potential damage to major plant equipment. In these instances, utilities can request code relief to perform full–stroke verification at alternate time intervals, while observing, if practical, partial stroke testing quarterly. Where this can not be accomplished, a relief request can be submitted to allow for valve disassembly and inspection during reactor refueling outages. Check valves identified for disassembly can be grouped together and a random sampling performed. To group valves for disassembly requires that they be of the same manufacturer, model number, size, material of construction, and service conditions including valve orientation. Groupings are generally limited to four valves or less. Once a group is identified, one valve can be disassembled and credit taken for the condition of the remaining valves. If the valve being disassembled exhibits degradation then the remaining valves within the group will require disassembly during that outage. The valves selected for disassembly must be alternated during subsequent outages. Generic Letter 89–04 provides direction for the disassembly and inspection interval, grouping methods and extension of intervals.

Check valve disassembly and inspection programs need to include requirements for recording critical design dimensions and any other indications of wear or corrosion. An important element of a successful disassembly and inspection program is documentation (See Figure 2). Prior to valve disassembly it is essential that a review be performed to identify manufacturer's mark numbers, vendor's drawings, existing maintenance history, previous IST test reports, internal dimensions/tolerances, etc. while accumulating valve background information necessary to baseline the disassembly criteria, one area of weakness has been the identification of internal valve tolerances, dimensions and wear margins. In working with various utilities and valve manufacturers we have found that manufacturing tolerances in some cases can be obtained from the vendor. Since it does involve some amount of research, the vendors usually charge a fee to obtain this information. In other cases we found the vendor's design drawings alone contained sufficient information to baseline the disassembly criteria.

In discussion with several valve manufacturers it was conveyed that they were reluctant to provide detailed design criteria to preclude the replicating of parts. Although some valve manufacturers stated they could provide recommendations for wear and replacement tolerances, it is generally up to the utility to provide an engineering assessment for specifying an acceptance criteria.

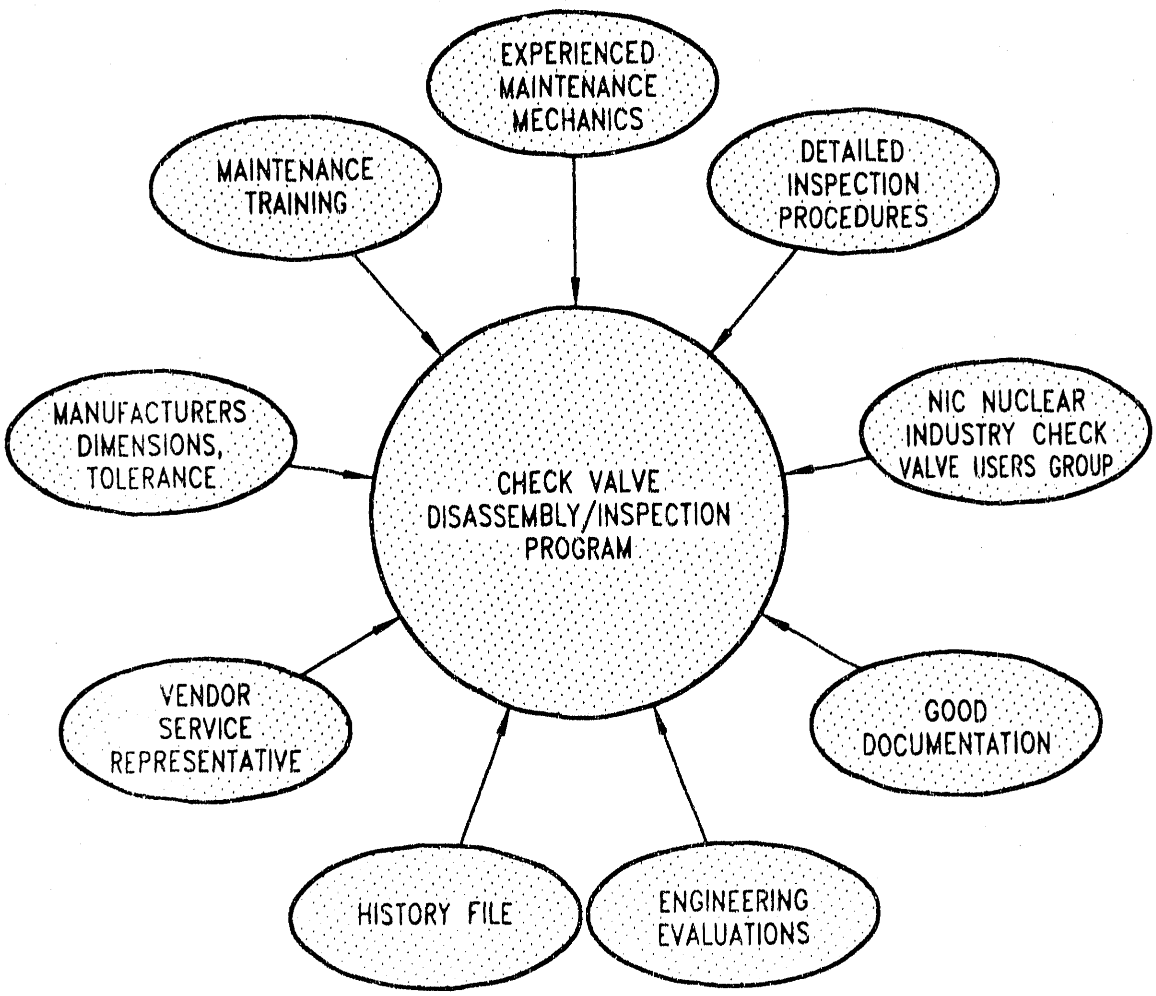


Figure 2. Elements of a check valve disassembly/inspection program.

One major valve manufacturer identified that all internal valve tolerances and dimensions for their valves are proprietary. It was their position that utilities can solicit their services to provide an inspection team to evaluate a valve during disassembly. This manufacturer also stated that they would not make a determination on the acceptability of a dimension without physically inspecting the part. This position was based on the liability of providing a determination without their inspection. A problem commented on by several utility maintenance groups addressed the lack of product improvement notices. Some utility maintenance supervisors felt that only after they had approached the valve manufacturers to discuss disassembly criteria that they were informed of design changes. This criticism was directed to the valve manufacturers and some agreed that design updates were not typically issued to utility groups. Other manufacturers identified that product improvement/notices were provided through their service representatives.

One method discussed with the utilities for baselining valve wear has been to compare the dimensions obtained from the disassembly of a valve to the design dimensions provided by some manufacturers. An alternative used at one utility has been to disassembly a new valve of the same manufacturer, size and design maintained in spare parts and compare these values to those of a valve in service. Utilities with older nuclear facilities need to be cautious in this approach. Valves installed at older plants may not be in production and the replacement design may have been significantly altered.

These methods do not provide a basis for the determination of acceptable wear and the utilities engineering department is generally still required to resolve dimensional concerns.

USER'S GROUP

Presently, a user's group, "The Nuclear Industry Check Valve Group," is being organized to present a forum for tabling information to the industry on internal check valve dimensions, flow limits and alternative non-intrusive test methods. This group may provide utilities an avenue to obtain information already identified by other utilities, valve manufacturers, A/Es, etc.

ALARA

Prior to valve disassembly it is important that a thorough review be performed for ALARA concerns. For any valve that is in a high radiation area, other methods of monitoring performance/degradation should be evaluated. If disassembly and inspection has been identified as the only viable method for monitoring the valve, integrating this with other methods of testing/monitoring should be considered. Grouping of similar valves for disassembly and inspection would also reduce the number of entries for a particular valve.

DESIGN APPLICATION REVIEW

As a result of the design application review being performed for SOER 86–3, valves indicating poor application are being disassembled and inspected for abnormal wear. Several utilities have chosen to include these valves within the IST Program. Surveillance requirements for these valves are being determined in accordance with the severity of the misapplication or based on their maintenance history.

DESIGN VS. MAINTENANCE

In general, the disassembly of check valves is viewed negatively by utilities because of high radiation concerns, spare parts required to be maintained, maintenance training and potential plant down time. Discussions with several utility maintenance departments revealed a higher incidence of valve leakage and maintenance requirements after a valve has been disassembled.

The majority of valves being disassembled tend to be larger valves, therefore the greatest majority are swing check valves. No specific difficulties with disassembly for one design of check valve over another has been concluded. However, check valves operating in severe conditions, such as service water, having poor water quality and stagnant flow conditions have exhibited a much higher incidence of degradation during disassembly and inspection.

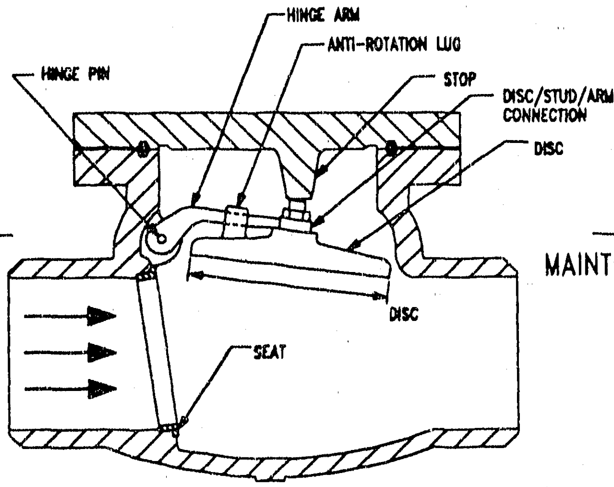
VISUAL INSPECTION

While disassembling a check valve it is critical to perform a complete visual inspection. It is this detail of information that will help identify potential check valve failures. Any abnormalities present at the time of the disassembly can be observed and resolved. The maintenance groups solicited are generally applying "good maintenance practice" to this inspection process. This means looking for "wear marks" on the valve bodies, "wear steps" on hinge pins, verifying the disk moves freely and opens and closes properly, etc.

In several plants, maintenance work orders utilize a checklist for inspection criteria. (See Figure 3). The checklist includes a list of all valve parts the mechanic and QC inspector will inspect. Comments on the condition of the valve noting any indications of wear, erosion. corrosion or damage are recorded. One utility has developed a generic maintenance guideline to highlight the areas of concern for visual inspection. This guideline discusses possible problems and the potential root causes. The mechanics are trained on this guideline and it is referenced on the Disassembly/Inspection checklist. Engineering support is requested if anything is found that the mechanic or QC inspector feel requires further evaluation. Engineering would evaluate the deficiencies and determine the cause. If the deficiency indicates abnormal wear, Engineering would perform an application review of the valve. The application review would identify improper valve location or type. If the deficiency is caused by normal wear, Engineering would determine if the wear is

NOTE: N/A SECTIONS OF THIS CHECKLIST NOT APPLICABLE. ATTACH CHECKLIST TO THE MWR.

VALVE MARK NUMBER ____________



MAINTENANCE REQUEST NUMBER ____________

VENDOR DATA (THIS TYPICALLY WOULD BE RECORDED PRIOR TO DISASSEMBLY TO IDENTIFY SPARE PARTS, DRAWINGS, ETC.)

MANUFACTURER ____________

MODEL NUMBER ____________

SIZE ____________

PURCHASE ORDER NUMBER ____________

NOTE: PRIOR TO DISASSEMBLY, REVIEW MAINTENANCE PROCEDURE 1-MP-*** FOR DETAILS OF VISUAL INSPECTION CRITERIA. VISUALLY INSPECT THE VALVE'S INTERNAL COMPONENTS. RECORD ANY DEFICIENCIES OBSERVED TO THE COMMENT SECTION AND NOTIFY THE MAINTENANCE SUPERVISOR AND SYSTEM ENGINEER.

1. HINGE PIN/BUSHING
2. HINGE ARM
3. ANTI-ROTATION LUGS
4. BACKSTOP
5. DISK/STUD/ARM CONNECTION/FASTENERS
6. DISK
7. VALVE BODY

Figure 3. Example of a checklist to record visual inspection of valve internals.

RESULTS OF ITEMS 1 THRU 7 ________________________________

INITIAL \ DATE

8. PERFORM A BLUE CHECK OF THE VALVE DISK/SEAT TO MEASURE SEATING SURFACE CONTACT.

RESULTS OF BLUE CHECK ________________________________

INITIAL \ DATE

GENERAL COMMENTS ________________________________

PERFORMED BY ________________
INITIAL \ DATE

* Q. C. INSPECTOR ________________
INITIAL \ DATE

* REQUIRED IF VALVE IS SAFETY RELATED.

Figure 3. (continued).

within the service limits of the valve. In cases where service limits are not available, it would be Engineering's responsibility to specify allowable wear for that type of valve or possible design modifications. The drawbacks in this practice are existing manpower restraints imposed on Engineering during an outage and potential plant delays waiting for an Engineering response.

Maintenance planning can reduce the above mentioned drawbacks by reviewing valves scheduled for disassembly before an outage. This allows any valve with a poor maintenance history to be identified to Engineering for an application review and identification of service limits.

Often the process of visual inspection has been limited to removing the bonnet, verifying free movement of the disk and a cursory observation for missing or loose parts. As discussed, observations such as gouges on the internal valve casting can provide warning of design misapplication prior to a valve failure.

As previously mentioned, some check valve manufacturers are providing technical services to perform valve inspections. Inspection services from the vendor are beneficial to the utility when critical dimensions and tolerances cannot be obtained. The disadvantage is that the utility becomes dependent on the vendor and without vendor support is unable to establish a program that measures valve wear to determine service limits.

SUMMARY

In summary, the ideal check valve program would include a history of every valve in the program. The history would include the manufacturer's critical design dimensions and tolerances, or, as a minimum, service limits. Each valve disassembly would be included in the history file and a checklist would be completed for each inspection. The checklist (See Figure 3) would include a list of all valve parts and a series of questions associated with each part. The questions would not only require the recording of critical dimensions, but would also look at the general condition of the part.

For example, when looking at a hinge pin from a swing check valve, in addition to measuring the pin diameter, indications of step changes, corrosion, etc. should also be noted. In some cases, photographs of the valve internals would be helpful. The documented results of a valve inspection should be reviewed and a determination made regarding repair/replacement and the inspection interval for the valve.

Since the vendor data required to make this program complete is not always available, a typical check valve program would rely on good maintenance practice policies and sound engineering judgment. Without having design dimensions and wear tolerances to baseline a valve's degradation, the program must include inspection procedures that require detailed documentation of a valve's condition. Also, good maintenance training and experienced maintenance mechanics are a key part of a successful disassembly/inspection program.

As the engineering manager of one valve manufacturer stated, "a lot can be learned about a (check) valve just from looking at the internals." Although this isn't the ideal situation this method can be very helpful provided it is accompanied by experienced maintenance mechanics, good maintenance training, quality inspection procedures, and detailed documentation.

When a check valve has been disassembled the operability of the valve should be verified (i.e., disc properly seats and moves freely). Each internal part of the valve should be evaluated and commented on. Measurements of valve wear should be recorded where practical (i.e., seat surface, excessive play, bushing wear. etc). Any abnormalities noted during the inspection should be documented and evaluated by engineering.

A check valve disassembly/inspection program that does not include manufacturers wear tolerances and design dimensions can be as effective as the program that includes these benefits. Regardless of whether the design and wear data is available, the same level of documentation should apply to each valve inspection. Initially, not having the manufacturer's parameters will make it more difficult to benchmark a valves wear. Using good maintenance practices and procedures the current condition of a valve can be evaluated to allow for baselining degradation. Data recorded during a valve disassembly should be evaluated against previous inspections to establish a trending program.

The utility should continue to work with the manufacturer when questions arise concerning valve wear or abnormalities found during the inspection process. Even if the vendor is reluctant to provide valve design and/or wear data, it is in his best interest to provide support for his product.

As non-intrusive testings (e.g. Acoustical, Ultrasonic, Magnetic Particle, Radiography, etc.) develop, utilities may be capable of supplementing their disassembly programs with this technology. At this time, the general industry feeling is a limited confidence in any of the existing non–intrusive test methods.

The results of the research for this paper strongly suggest that at present the most viable method of determining check valve condition is disassembly. Recognizing the maintenance concerns expressed by the utilities, to meet the requirements of ASME B&PV Code Section XI, disassembly looks to be the best method.

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1. *Check Valve Failures or Degradation*, Atlanta, Georgia: Institute of Nuclear Power Operations, Significant Operating Experience Report 86–3, October 15, 1986.

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3. *Check Valve Application Manual and User's Guide*, Raleigh. NC: Rockwell International, Bulletin V–303, June 1988.

4. *Guidance on Developing Acceptable Inservice Testing Programs*, Generic Letter B9–04 United States Nuclear Regulatory Commission, April 3. 1989.

5. *Nuclear Industry Check Valve Group Meeting Agenda for April 19–20, 1989*, May 1, 1989.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

What is an acceptable level of inspection to ascertain operational readiness of a check valve? Is visual enough or does a dimensional inspection improve the ability to make such a determination?

Response 1

Our research indicated that dimensional analysis would be an improvement to the program. It would supplement strictly visual inspection. However, the Code requests simply verifying that the valve full strokes through its operation.

Question 2

I am interested in what we need to do in IST. Are there other implications of check valve problems, such as severe wear?

According to the Code, do we need to go beyond a visual inspection?

Response 2

The answer to your question is in the generic letter.

DIAGNOSTICS OF STROKE TIMING TEST FOR POWER OPERATED VALVES

GARRY GALBREATH
JIM WITHERSPOON
DUKE POWER COMPANY

ABSTRACT

The paper provides a historical perspective of the application of valve stroke timing since the early 1970s and how the issues have evolved. An assessment of how effective valve stroke timing is in finding and predicting problems is discussed. Research findings will be presented regarding the complete test and maintenance history of a small, random sample of different types of valves, highlighting where the industry may be heading in the future.

INTRODUCTION

I would like to talk to you this morning about valve stroke timing. When we first got involved in this area about 15 yr ago, we took a pretty simplistic approach to timing valves. This approach has changed over the years as our experience has grown. I would like to cover three aspects of valve timing today, and I hope that this perspective will be of benefit to you.

First, I want to cover some history. Where did valve timing come from, and how has it changed over the years. Second, I want to take a look at what valve timing is doing for us. Its surprising how little data exists in a retrievable form which would let us know how effective valve timing is. We have done some selective research on our own test history, and I will cover what we have found. Finally, I want to touch on where we are going with valve stroke timing. The short answer is that no one knows for sure, but there are some interesting possibilities.

As I'm sure you have all experienced, in the surveillance area, we tend to get watched from time to time. Quality Assurance, INPO, and the NRC all audit us, and in addition, we sometimes find things on our own that we don't feel comfortable with.

In the interest of brevity, I want to share just a few of the issues that have been brought up over the years regarding valve timing, which I hope will demonstrate that this is not as simple as it looks at the outset:

- Comparative valve times for similar valves in similar applications. We initially looked at each valve with blinders on, determining whether it met the code. Now we look at the corresponding valves on the other units, as well as similar design valves, and see if the times are consistent. If not, we may have a problem.
- Air–operated valves which are activated from multiple solenoids. Should you use the same solenoid each time? Are the times different when using different solenoids?
- Timing from limit–switch to limit–switch vs. timing from actuator to limit–switch. We originally made the argument that limit–to–limit was a better test, since we could use automatic triggering for the timing. We felt that by using a consistent trigger point, we would have a better chance of detecting changes. The NRC agreed with us, at least on Catawba, because they approved our SER with this relief written into it. However, this is a continuing sensitive issue.
- Retest issues—We have had to examine our approach to post–maintenance retest evaluations. For example, if work is done that would affect the bleed–off rate for an air actuated valve, the stroke time is probably affected, and the valve should be retested. I think we are doing a better job now of identifying these types of situations.

I'm sure there are lots more, but these are just some examples of the types of issues that have arisen. I'm sure that any of you who administer these programs have seen the same types of issues.

CURRENT DATA

Now lets move on to something potentially more interesting, namely, what's valve stroke timing doing for us, anyway? This is an obvious question, yet we don't seem to have a ready answer. Most people who have been involved with valve timing for a few years have opinions, but rarely is the data available to back up the opinions.

In preparing for this presentation, we decided to do a little research, admittedly limited, to see what we could find out. What we did was to pick at random several valves of different types, and research the history of these valves. We gathered both the maintenance and test history for the life of the plant (our plant was McGuire, so the records go back about 8 yr). Once we had all the information, we tried to see if we could learn anything from it. Some things we were interested in included:

- Has valve stroke timing identified problems before the component actually broke?
- Could we have used the information we had to predict a failure?
- Were there failures which valve timing did not catch?
- How many test failures actually turned out to be valve failures? That is, in how many cases had the valve really been unable to perform its safety function?

We selected what we believe is a representative cross–section of valves. We looked at a couple of MOVs, several air–operated valves, and a couple of electric solenoid valves. Although the sample was generally random, we did pick a couple of valves which we "perceived" to be a problem.

I want to discuss our findings according to valve type, since it seems that this is the most important factor in what we found. I first want to say that our preconceptions with regard to valve timing were in most cases fulfilled. We didn't find a lot of surprises, but we did find some interesting results.

The least interesting were the electric solenoid operated valves (See Attachments 1 and 2). The stroke times were virtually unmeasurable, ranging from 0.0 to 0.2 s. Both valves had maintenance performed due to seat leakage. Valve stroke timing had not identified any problems, although after one test one of the reed switches stuck, requiring a work request to repair it. If these findings could be extrapolated across a wide sample of this type of valves, I would conclude that stroke timing is doing us no good for these valves. A periodic stroke test, probably at a reduced frequency, would provide as much information.

We looked at two electric motor operated valves, one a gate valve and the other a butterfly valve (See Attachments 3 and 4). The conventional wisdom is that valve stroke times do not vary for EMOs, except when limit switches are reset, and we found this to be true in these cases. There had been several work requests performed on these valves. these were for seat and external leakage and for preventive maintenance. Valve stroke timing had identified no problems on either of these valves.

We found that following valve maintenance in which the limit switches were reset, the measured stroke time may or may not change. For example, on one valve the stroke time had stayed within a band of 6.0 to 7.2 s for over 5 yr. Following limit switch setup, the time changed to 4.6 s. In another case, however, the time stayed within a band of 31.2 to 32.0 s, even though limit switches had been reset.

One interesting item we noted was that although the stroke times for this valve were very consistent, we did get a jump from the 31 to 32 s range to around 40 s for two tests, and then the time returned to normal. We can only speculate that the valve may have been timed using different indications for these tests. Our records indicate that no work was performed on the valve during this interval.

One last item I want to cover on MOVs concerns the question of limit–to–limit versus initiation–to–limit timing. Prior to 1987, McGuire had been using limit–to–limit timing. At that time, they switched to initiation–to–limit. We found that a step change was observed when this change was made. On one valve, times went from ~32 s to ~42 s, and the other went from 4.6 to 7 s. Following the change, the times have been very consistent at their new values.

Summing up for MOVs, based on our sample, we are not detecting any problems via valve stroke timing. It would appear that a simple valve stroke, without measurement of time, on some frequency would be as useful as what we are doing now.

Our final sample was three air operated valves (see Attachments 5, 6, and 7). We found in general that these valves had been much less reliable and much less consistent with regard to stroke time, although we still detected only a couple of "real" problems with these valves. I am defining a "real" problem to be one in which the valve may have not been able to fulfill its safety function.

In general, most of the problems identified with these valves involved leaks or control problems which were identified during normal day–to–day operation of the plant.

We did identify several cases in which these valves did not meet their required stroke time. Upon investigation, no apparent problems were found. It appeared that the act of stroking the valve for the test was enough to free it up such that it successfully cycled afterward. Our policy is to generate a work request to investigate if the valve fails to meet its required stroke time on the first attempt.

As I mentioned earlier, the range of times varied widely for the air–operated valves. The best one ranged from 5.2 to 10 s, the worst from 5.8 to 57.2 s. In the latter case, a total of three work requests have been written due to failure to meet stroke times. It is interesting to note that this valve, a Main Steam PORV, was tested 23 times in 1987 and 27 times in 1988.

The same transition in timing methods, from limit–to–limit to actuation–to–limit was made for AOVs in 1987. The resultant times do not show any significant variation due to timing method, which is somewhat contrary to conventional wisdom.

For AOVs, there seems to be more indication of timing related problems than with the other types, and it would appear based on our limited sample that continued timing as we are now doing is appropriate for these valves.

I want to give you an idea of what it took to come up with this data, since that was interesting in itself. We basically had one engineer working for two weeks to develop this history on seven valves. I took a combination of reviewing documents in our Master File, data searches on computer, and retrieval of microfilm information. Although the sample we have here is limited, I wanted to give you a feel for what it might take to do a more extensive study.

I was glad to have an opportunity to do this research, because I had been wondering for a while now what it might tell us. To summarize, it does not appear that stroke timing EMOs and electric solenoid valves is doing much for us. A simple stroke of the valve, which is done for a lot of valves during the course of normal plant operations, would accomplish as much. For AOVs, stroke timing is more likely to find problems, but the act of stroking the valve itself appears to resolve the problem in some cases (This obviously doesn't address the issue of how long the valve has been sitting there in a less than optimum condition).

FUTURE

I want to close by looking a little bit at what the future might hold for valve stroke timing.

As I'm sure has been discussed elsewhere during this conference, O&M Part 10 is now approved and will presumably be referenced at some point in 10 CFR 50, opening the way to its use in defining valve testing programs. I don't want to get into the requirements of Part 10 here, but this will definitely affect our programs in the not too distant future.

Looking farther ahead, there seems to be interest in the industry in the type of research that I have discussed today. There has been an EPRI study which has laid some groundwork for assessing the effectiveness of the current program, and the OM Part 10 Working Group is looking at various aspects of potential changes to the Standard requirements.

Another potential factor is the more extensive diagnostic testing that is now being done on valves. Equipment is available which allows a comprehensive assessment of valve health. It may be that this testing will be able in the future to justify a reduced or eliminated valve timing requirement.

One parameter of potential interest with regard to EMOs is motor current. It may be that recording a motor current trace as the valve cycles may be a better way of determining on an ongoing basis the health of the valve.

However, given the usual momentum in this industry, I wouldn't hold my breath waiting for any imminent changes in valve stroke timing requirements.

I hope that this presentation has been somewhat enlightening, or has at least caused you to think of some aspects of valve timing that you hadn't considered before. As I said at the outset, its a subject that can appear to be cut and dried on the surface, but its anything but that once you get into the details.

HISTORY

Valve stroke timing began in the early 1970's. ASME Section XI, subsection IWV mandated that ASME Class 1, 2, and 3 valves which have a safety function would be timed on a quarterly basis. (In cases where quarterly testing is not possible, deferral to cold shutdown or refueling is permissible). I remember writing the first procedures for doing valve stroke timing at McGuire around 1977, and our concerns at the time were basically with identifying test conditions and equipment lineups such that we could physically stroke the valves. We had response times that our Design Engineering Department had given us, and we felt that we were in good shape with regard to meeting the code.

Our first surprise came in 1981, when we received a package of a hundred or so questions from EG&G, Idaho regarding our program. Our meetings with EG&G resulted in a total rewrite of our program, primarily due to necessary changes in relief requests and differing philosophies as to what should be included in the program. These meetings were helpful to us, but it would have been much more helpful if the information that we learned could have been obtained years earlier.

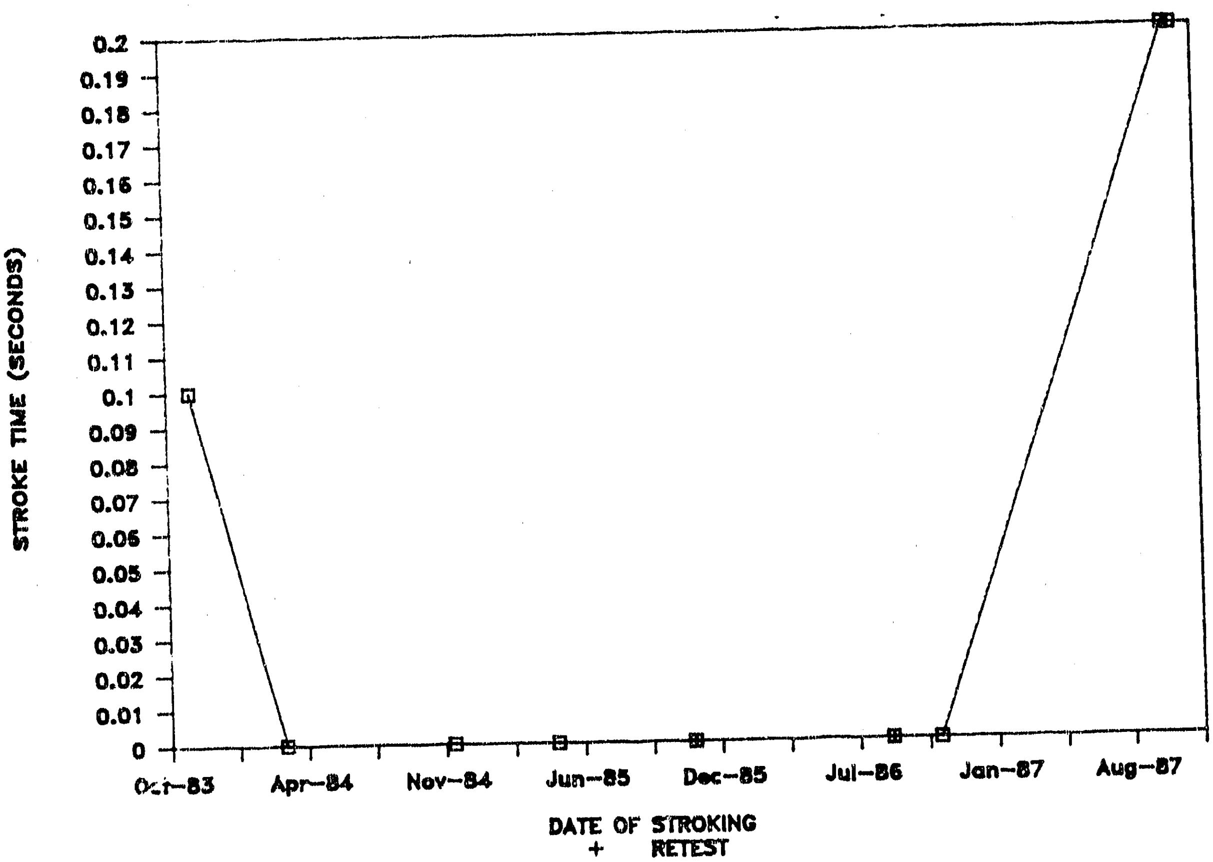
Following the EG&G meetings and subsequent program revision, we felt that our program was in pretty good shape. That's when the subtleties of implementation began to surface.

ATTACHMENT 1

Solenoid operated globe valve (1" target rock) Reactor vessel vent line isolation (lNC272AC)

Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)	Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)
11/14/83	0.1	60	No	08/22/86	0.0	60	Yes
03/26/84	0.0	60	No	10/31/86	0.0	60	No
11/26/84	0.0	60	No	10/08/87[a]	0.2	60	No
04/23/85	0.0	60	No	10/19/87	0.2	60	No
11/08/85	0.0	60	No				

a. Stroke times were measured limit to limit prior to this date. Stroke times were measured initiation to limit on and subsequent to this date.



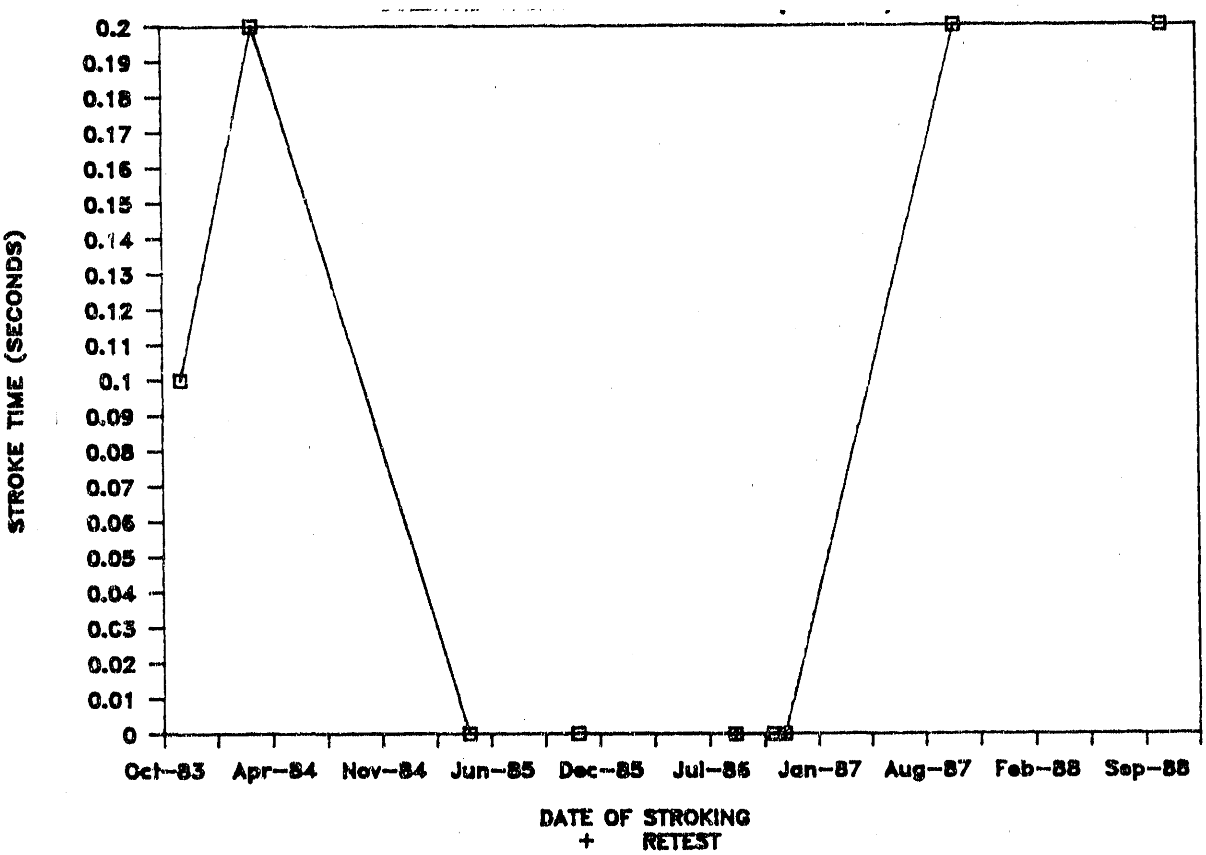
Attachment 1. Solenoid Operated Globe Valve (NC272AC).

ATTACHMENT 2

Solenoid operated globe valve (1" target rock) reactor vessel vent line isolation (INC273AC)

Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)	Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)
11/14/83	0.1	60	No	10/31/86	0.0	60	No
03/26/84	0.2	60	No	11/21/86	0.0	60	Yes
04/23/85	0.0	60	No	10/08/87[a]	0.2	60	No
11/08/85	0.0	60	No	10/19/88	0.2	60	No
08/22/86	0.0	60	Yes				

a. Stroke times were measured limit to limit prior to this date. Stroke times were measured initiation to limit on and subsequent to this date.



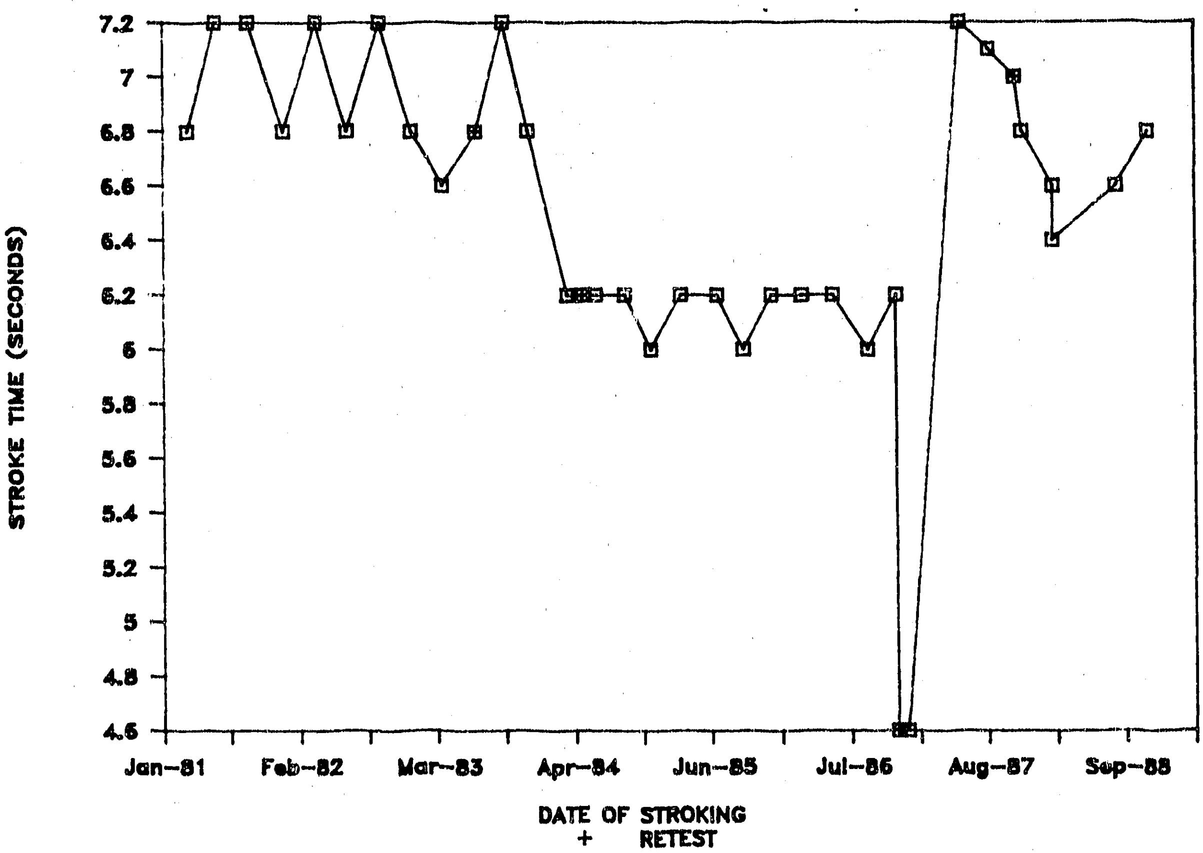
Attachment 2. Solenoid Operated Globe Valve (NC273AC).

ATTACHMENT 3

Electric motor operated gate valve (8" Walworth) NS pump discharge containment isolation (INS15B)

Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)	Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)
03/26/81	6.8	10	No	06/16/85	6.2	10	No
06/17/81	7.2	10	No	08/30/85	6.0	10	No
09/23/81	7.2	10	No	11/18/85	6.2	10	No
12/30/81	6.8	10	No	02/12/86	6.2	10	No
04/07/82	7.2	10	No	05/12/86	6.2	10	No
07/07/82	6.8	10	No	08/23/86	6.0	10	No
10/08/82	7.2	10	No	11/13/86	6.2	10	No
01/10/83	6.8	10	No	11/18/86	4.6	10	Yes
04/10/83	6.6	10	No	12/15/86	4.6	10	No
07/13/83	6.8	10	Yes	05/18/87[a]	7.2	10	No
10/04/83	7.2	10	No	08/13/87	7.1	10	No
12/12/83	6.8	10	No	10/29/87	7.0	10	Yes
04/03/84	6.2	10	No	11/17/87	6.8	10	No
05/16/84	6.2	10	Yes	02/15/88	6.6	10	No
06/27/84	6.2	10	No	02/13/88	6.4	10	No
09/19/84	6.2	10	No	08/18/88	6.6	10	No
12/05/84	6.0	10	No	11/14/88	6.8	10	No
03/01/85	6.2	10	No				

a. Stroke times were measured limit to limit prior to this date. Stroke times were measured initiation to limit on and subsequent to this date.



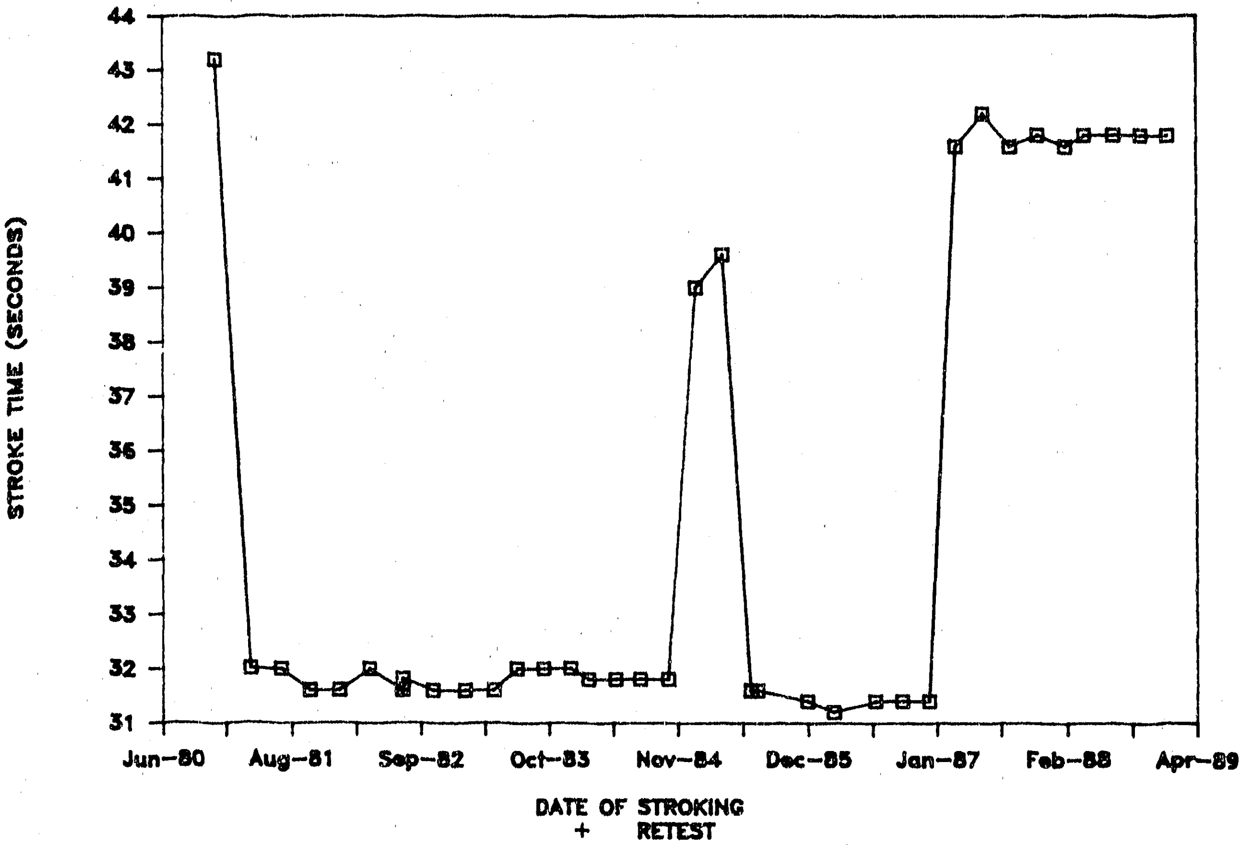
Attachment 3. Motor Operated Gate Valve (1NS15B).

ATTACHMENT 4

Electric motor operated butterfly valve (20" Pratt) KC Train to nonessential supply header (IKC53B)

Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)	Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)
12/11/80	43.2	60	No	01/08/85	39.0	60	No
03/27/81	32.0	60	No	04/01/85	39.6	60	No
06/30/81	32.0	60	No	06/26/85	31.6	60	No
09/28/81	31.6	60	No	07/17/85	31.6	60	No
12/28/81	31.6	60	No	12/16/85	31.4	60	No
03/31/82	32.0	60	No	03/07/86	31.2	60	No
07/12/82	31.6	60	Yes	07/14/86	31.4	60	No
07/13/82	31.8	60	No	10/06/86	31.4	60	No
10/13/82	31.6	60	No	12/29/86	31.4	60	No
01/20/83	31.6	60	No	03/23/87[a]	41.6	60	No
04/21/83	31.6	60	No	06/15/87	42.2	60	No
06/30/83	32.0	60	No	09/09/87	41.6	60	No
09/22/83	32.0	60	No	11/30/87	41.8	60	No
12/15/83	32.0	60	No	02/23/88	41.6	60	No
02/09/84	31.8	60	No	04/21/88	41.8	60	No
05/03/84	31.8	60	No	07/19/88	41.8	60	No
07/20/84	31.8	60	No	10/10/88	41.8	60	No
10/16/84	31.8	60	No	01/03/89	41.8	60	No

a. Stroke times were measured limit to limit prior to this date. Stroke times were measured initiation to limit on and subsequent to this date.



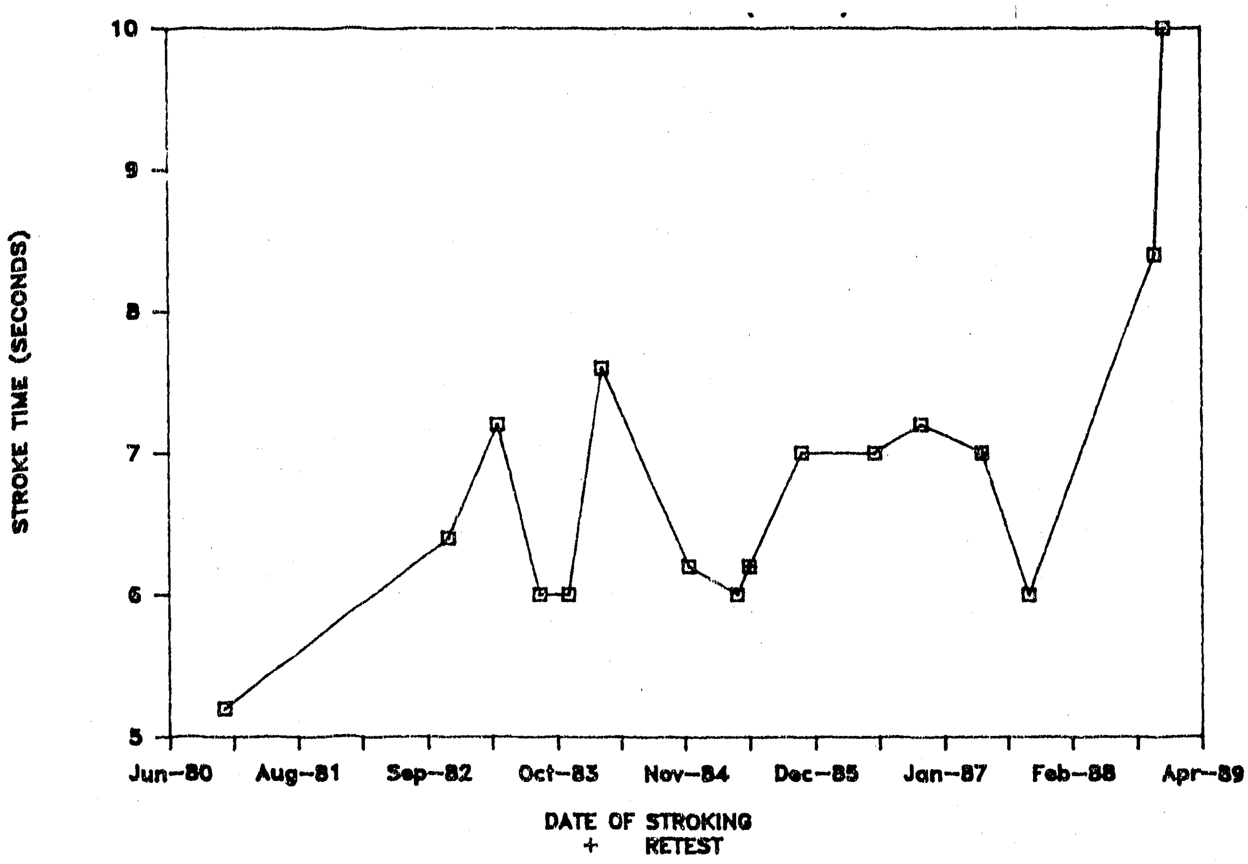
Attachment 4. Motor Operated Butterfly Valve (1KC53B).

ATTACHMENT 5

Air operated gate valve (6" Pacific) Strainer backflush isolation (IRN25B)

Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)	Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)
12/16/80	5.2	60	No	05/31/85	6.2	60	Yes
11/09/82	6.4	60	No	11/08/85	7.0	60	No
04/12/83	7.2	60	No	06/23/86	7.0	60	No
08/18/83	6.0	60	No	11/12/86	7.2	60	No
11/14/83	6.0	60	No	05/21/87[a]	7.0	60	Yes
02/29/84	7.6	60	No	10/15/87	6.0	60	No
11/26/84	6.2	60	No	11/08/88	8.4	60	No
04/23/85	6.0	60	No	12/08/88	10.0	60	No

a. Stroke times were measured limit to limit prior to this date. Stroke times were measured initiation to limit on and subsequent to this date.

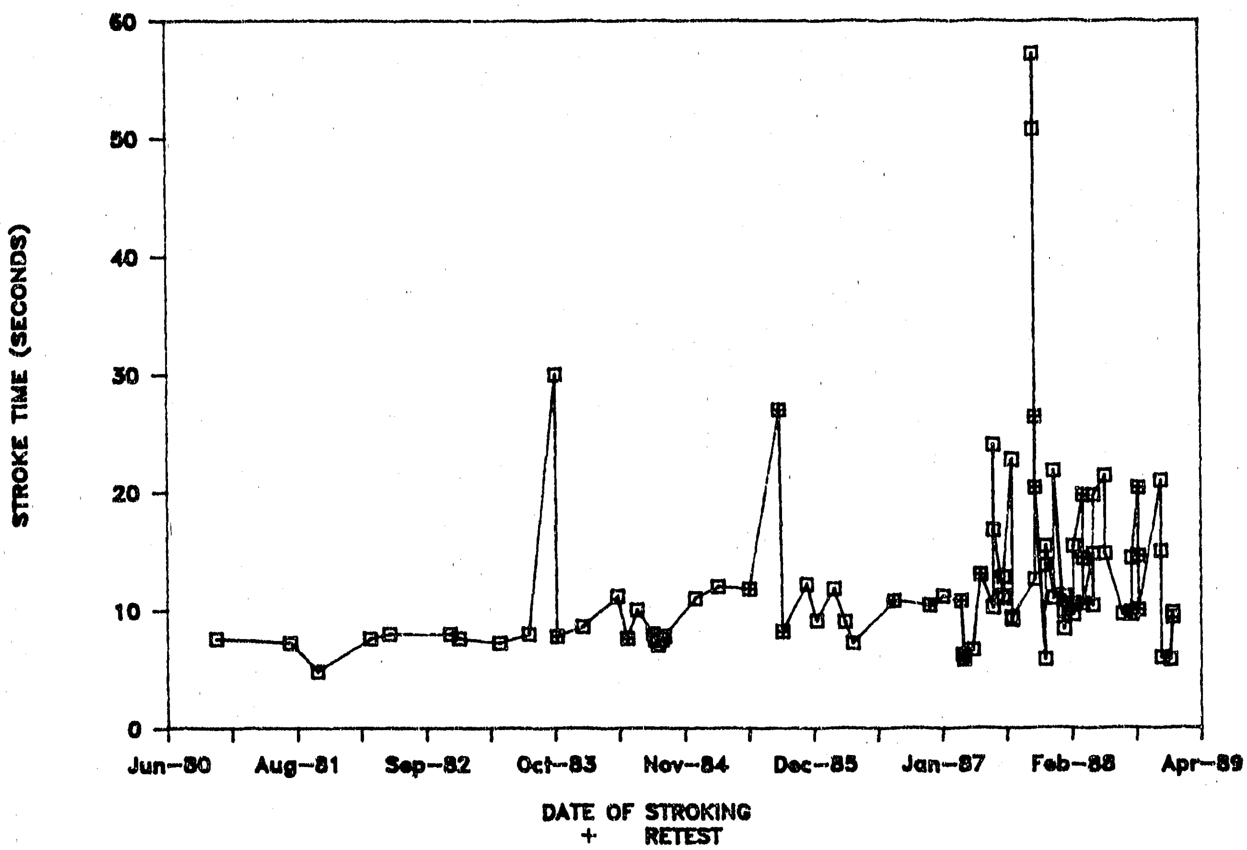


Attachment 5. Air Operated Gate Valve (1RN25B).

ATTACHMENT 6

Air operated velocity control valve (6" Crosby) main steam PORV (ISVIAB)

Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)	Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)
11/25/80	7.6	30	No	07/31/87	11.0	30	No
07/11/81	7.2	30	No	07/31/87	12.8	30	No
10/04/81	4.8	30	No	08/28/87	22.8	30	No
03/16/82	7.6	30	No	08/28/87	9.2	30	No
05/15/82	8.0	30	Yes	08/28/87	9.4	30	No
11/19/82	8.0	30	No	11/06/87	12.6	30	No
12/16/82	7.6	30	Yes	11/06/87	50.8	30	No
04/21/83	7.2	30	No	11/06/87	57.2	30	No
07/21/83	8.0	20	No	11/07/87	26.4	30	Yes
10/13/83	30.0	20	No	11/07/87	20.4	30	Yes
10/17/83	7.8	20	Yes	12/08/87	5.8	60	No
01/05/84	8.6	20	No	12/10/87	13.8	60	No
04/21/84	11.2	20	No	12/10/87	15.4	60	No
05/21/84	7.6	20	Yes	01/04/88	11.0	60	No
06/20/84	10.0	20	No	01/04/88	21.8	60	No
08/09/84	8.0	20	Yes	02/04/88	8.4	60	No
08/15/84	7.4	20	Yes	02/04/88	9.8	60	No
08/23/84	7.0	20	No	02/04/88	11.2	60	No
09/03/84	7.4	20	Yes	03/03/88	10.4	60	No
09/12/84	7.8	20	No	03/03/88	9.6	60	No
12/17/84	11.0	20	No	03/03/88	15.4	60	No
02/27/85	12.0	20	No	04/01/88	19.8	60	Yes
06/03/85	11.8	20	Yes	04/01/88	14.4	60	Yes
09/03/85	27.0	20	Yes	04/01/88	10.6	60	Yes
09/11/85	8.2	20	Yes	05/03/88	14.8	60	No
11/25/85	12.2	20	No	05/03/88	10.4	60	No
12/27/85	9.1	20	No	05/05/88	19.8	60	No
02/18/86	11.8	20	No	06/09/88	21.4	60	No
03/24/86	9.0	20	No	06/09/88	14.8	60	No
04/17/86	7.2	20	No	08/02/88	9.6	60	No
08/26/86	10.8	20	Yes	08/29/88	9.8	60	No
12/15/86	10.4	20	Yes	08/29/88	9.6	60	No
01/26/87	11.2	20	No	08/29/88	14.4	60	No
03/23/87	10.8	20	Yes	09/20/88	20.4	60	Yes
03/30/87	5.8	20	Yes	09/20/88	14.6	60	Yes
03/26/87	6.2	20	Yes	09/20/88	10.0	60	Yes
04/28/87	6.6	20	No	11/29/88	21.0	60	No
05/22/87	13.0	20	Yes	11/29/88	15.0	60	No
07/01/87	10.2	20	No	11/29/88	6.0	60	No
07/02/87	24.0	27	No	12/26/88	5.8	60	No
07/02/87	16.8	27	No	01/03/89	9.8	60	No
07/31/87	11.2	30	No	01/03/89	9.4	60	No



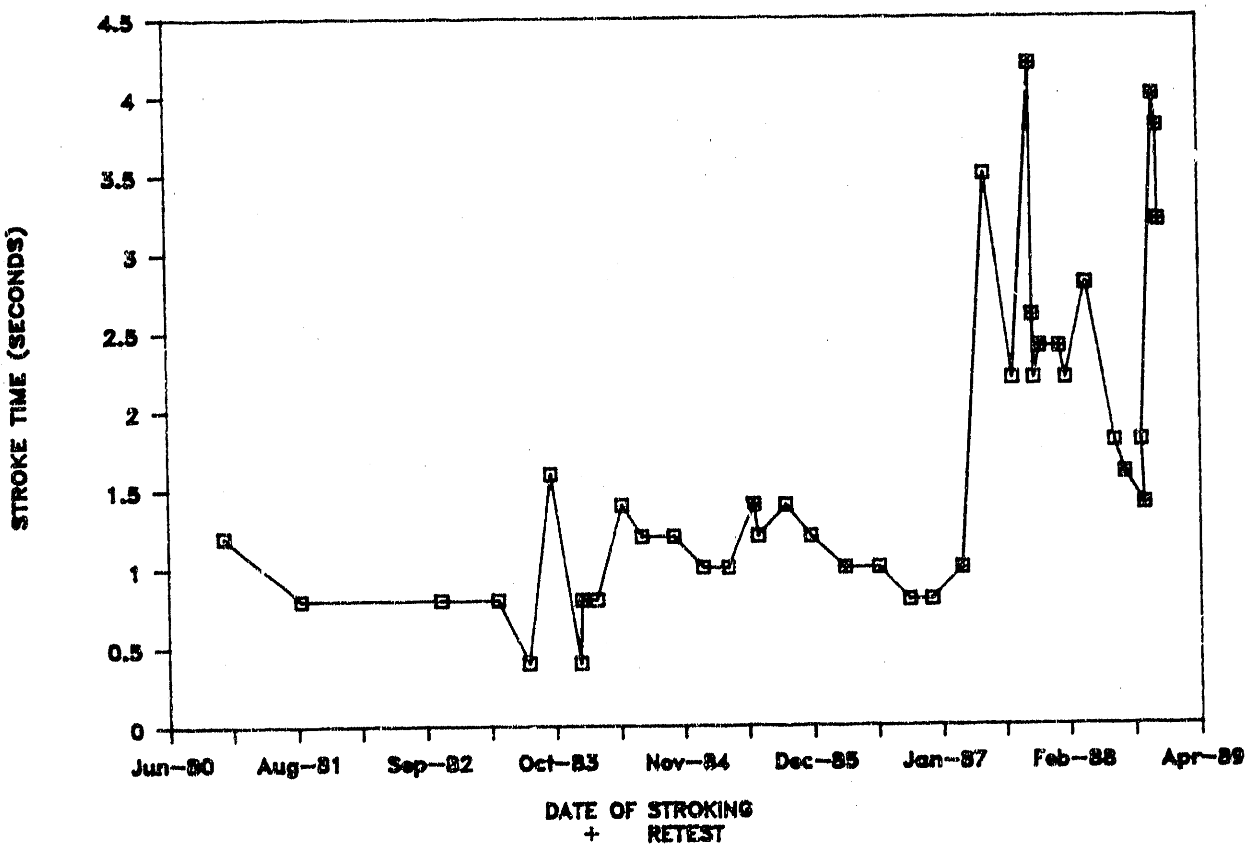
Attachment 6. Air Operated Relief Valve (1SV1AB).

ATTACHMENT 7

Air operated butterfly valve (20" Fisher) KC HX outlet temperature control (IRN59A)

Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)	Date of Cycle (MMDDYY)	Measured Stroke Time (s)	Required Stroke Time (s)	Performed as a Retest (Yes/No)
12/17/80	1.2	60	No	12/22/86	0.8	60	No
08/12/81	0.8	60	No	03/25/87	1.0	60	Yes
10/20/82	0.8	60	No	06/11/87	3.5[a]	60	No
04/15/83	0.8	60	No	09/03/87	2.2	60	No
07/19/83	0.4	60	No	11/01/87	4.2	60	Yes
09/27/83	1.6	60	No	11/03/87	2.6	60	Yes
12/28/83	0.4	60	No	11/07/87	2.2	60	No
01/01/84	0.8	60	Yes	11/25/87	2.4	60	No
02/16/84	0.8	60	No	11/25/87	2.4	60	Yes
05/03/84	1.4	60	No	01/29/88	2.4	60	Yes
07/05/84	1.2	60	No	02/17/88	2.2	60	No
10/16/84	1.2	60	No	04/19/88	2.8	60	No
01/10/85	1.0	60	No	04/21/88	2.8	60	No
04/01/85	1.0	60	No	07/14/88	1.8	60	No
06/18/85	1.4	60	Yes	08/15/88	1.6	60	Yes
07/02/85	1.2	60	No	10/14/88	1.4	60	Yes
09/25/85	1.4	60	No	10/05/88	1.8	60	No
12/11/85	1.2	60	No	11/17/88	4.0	60	Yes
03/27/86	1.0	60	Yes	11/29/88	3.8	60	Yes
07/10/86	1.0	60	No	12/02/88	3.2	60	Yes
10/15/86	0.8	60	No				

a. Stroke times were measured limit to limit prior to this date. Stroke times were measured initiation to limit on and subsequent to this date.



Attachment 7. Air Operated Butterfly Valve (1RN89A).

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

No Questions/Comments

HOW CHECK VALVES FAIL

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ABSTRACT

Because of the five check valve failures at San Onofre Unit 1 on November 21, 1985, the NRC issued NUREG-1190. This NUREG discussed the reasons for the failures of the five check valves and how the check valve failures induced water hammer to the feedwater system. In parallel effort INPO issued Significant Operating Experience Report SOER 86-3 on October 15, 1986. This SOER discussed the San Onofre Unit 1 check valve failures as well as previously issued SER's, IE Information Notices, SER's and information on industry experience with check valves.

As a result of the above effort, the Nuclear Industry Owner Group formed a task force to address the issue of check valve failures, and developed and issued an Application Guidelines for check valves as EPRI NP-5479 Project This publication provides guidelines for proper application and correct design of specific check valves.

The purpose of this paper is to enhance the above information and explain many of the subtle and salient features of check valve failures and their effects on the piping systems.

This paper will discuss and give examples of

1. Check valve vibration modes
2. Check valve failure modes
3. Check valve effects on the piping system
4. Case histories to illustrate the problems.

INTRODUCTION

Check valve failures continue to occur in nuclear power plants, causing concern for plant availability and integrity. To identify various failure mechanisms for root cause analysis, one has to understand the many different failures of check valves. There are many failure mechanisms possible for check valves, but the most significant ones are the following:

1. The minimum velocity required to lift the disc to the full-open and stable position.
2. The minimum L/D's from upstream disturbances.
3. Vortex shedding from the valve seat ring and the trailing edge of the valve disc.
4. High velocity through the valve seat.
5. Control valve and orifice cavitation.
6. Water hammer in piping systems.
7. Pump induced pulsations.

Items 1 and 2 will not be discussed in this paper since an excellent and complete discussion of them is given in References 1 to 9.

The most common problem associated with check valve failures is due to system flow oscillations or system piping vibrations; these oscillations and vibrations induce check valve component wear and thus the check valve components may fail. The seven above items are either singularly or collectively responsible for the following type of physically damaged check valves:

1. Disc separated from hinge arm
2. Disc stud broken
3. Disc nut loose
4. Disc partially open
5. Disc caught on inside of seat ring
6. Antirotation lug lodged under hinge arm
7. Cracked dis, seat ring, and bushings

8. Worn hinge pin and bushing

9. Missing bushing

10. Elongated hinge pin holes

11. Bent hinge pin, disc, and hinge arm

Before we discuss the last five failure mechanisms, we will discuss check valve vibration modes.

CHECK VALVE VIBRATION MODES

Figure 1 illustrates a typical swing check valve that is installed in many nuclear power plants; it also shows the many possible vibration modes the valve may have. The disc is attached to the hinge arm which can swing freely or vibrate backwards and forward through an angle ϕ about the hinge pin, as in the case of a compound pendulum. This disc oscillation or natural frequency in quiescent water is between 0 and 10 Hz. This has been verified in Reference 11. Table 1 shows a comparison of the theoretically calculated natural (in air) frequencies of major internal swing check valve components for different size valves. The "added mass" effect of the water on the valve components cannot be neglected and it appreciably lowers the natural frequency (or about 15%) from the in air value.

The process of periodic vortex formation and shedding or fluctuating pressures from the piping system produces periodic pressures on the valve disc and other internal valve components. If the frequency of the vortex shedding or fluctuating pressures from the piping system coincide with the disc and other internal valve components *Natural Frequency*, then a resonance will occur and the valve disc and other valve components will experience vibratory displacements. These vibrating displacements will induce abrasive wear, surface fatigue wear, and fretting wear and the valve components may fail.

One may see from Table 1, that flow–induced vibration, pressure pulsation, and fluid turbulence in the zone surrounding the check valve disc and other valve components may have sufficient energy to set up many different modes of vibrations in the check valve at their natural resonant frequencies.

VORTEX SHEDDING

Vortex Shedding from the valve seat ring and the trailing edge of the valve disc can cause disc oscillations in the flow stream (see Figure 2). When a fluid flows over a valve disc, the wake downstream of the plate is no longer regular or streamlined, but exhibits turbulence induced by separation from the valve disc. Vortices along a surface of separation result from instability of the flow, which occurs whenever there is a discontinuity or a very great velocity gradient fluid flow. This separated flow exhibits distinct counterclockwise vortices in a regular, periodic manner. This vortex shedding from alternate sides of the plate and along the surface of discontinuity imposes a varying force on the plate perpendicular to the normal flow of the fluid; or the vortices attach and detach the trailing edge of the valve disc, which protrudes slightly into

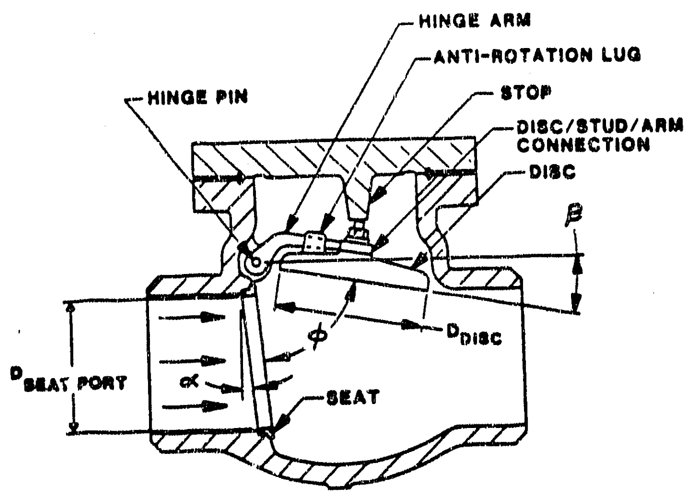


Figure 1. Swing check valve nomenclature.

Table 1. Natural (in air) frequencies of major internal swing check valve components

No.	Valve Size and Class	Spring	Disc	Hinge Arm	Hinge Pin
1	3"–150	292	4054	930	3710
2	3"–900	292	8936	930	3710
3	4"–150	259	4468	640	7024
4	6"–150	198	1733	315	2177
5	6"–900	261	3466	239	2183
6	8"–150	161	1745	312	4198
7	10"–150	83	1316	307	3370
8	12"–150	53	614	231	3852
9	16"–900	67	1574	87	982
10	36"–150	48	391	40	873

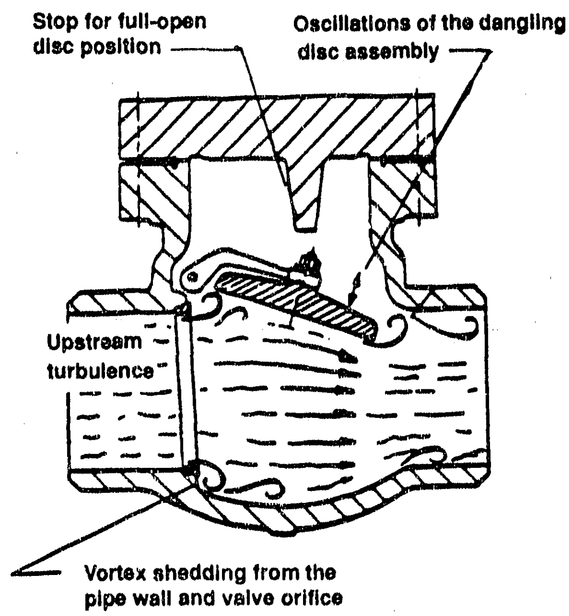


Figure 2. Vortex shedding responsible for the flow induced motion of a disc assembly dangling in the flow stream.

the flow stream. As the vortices are swept downstream, smaller vortices form around the main ones until the entire vortex pattern degenerates into random turbulent motion. This cycle is periodic. When the frequency of the vortex shedding approaches the natural frequency of the valve disc, severe disc flutter and oscillations occur and the disc may oscillate in the flow stream between 1 and 10 Hz or the disc may repeatedly hit the check valve stop (see Figure 1). But in both cases the wear will be on the hinge pin and the bushing; but for back tapping—repeatedly hitting the stop—the wear will be on the disc stud and stop.

Figure 2 shows vortices being shed over the seat and the downstream edge of the disc. It is believed that this vortex shedding is responsible for the check valve failures at San O'Nofre (see Reference 5).

Figures 3 and 4 are taken from Reference 12; note the similarity between Figure 2 and 3. The purpose of Reference 12 study was to analyze flow through abrupt two–dimensional expansions and to analyze the effect of forced oscillations of the gate on vortex formation. The shedding of vortices for abrupt expansions and

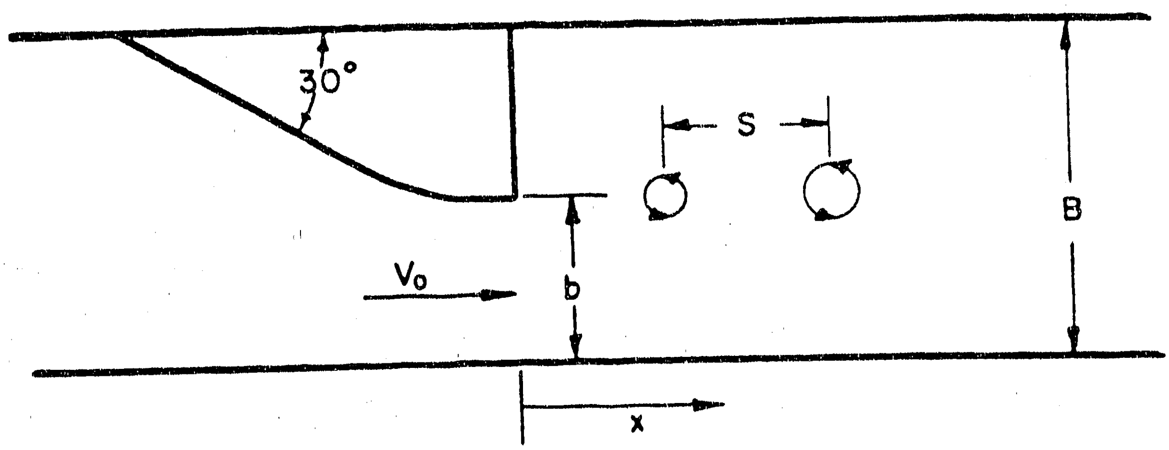


Figure 3. Notation for vortices on a surface of separation at an abrupt expansion.

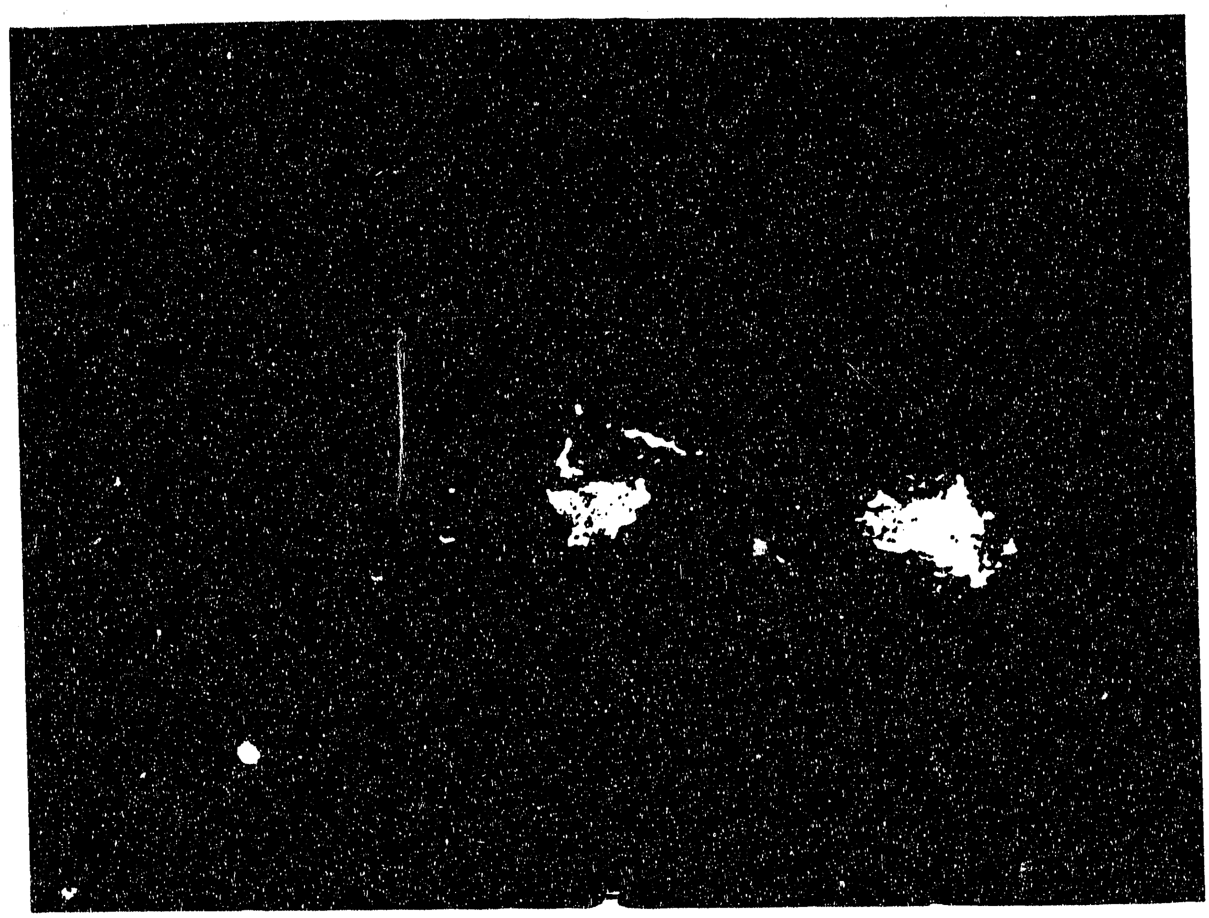


Figure 4. Vapor cavities in vortices on a surface of separation.

oscillating gates was observed; Figure 4 shows a typical vortex shedding observation: a gate oscillating in a plane normal to flow through a conduit causes generations of vortices. The vortices are stable and form periodically at a frequency equal to that of the gate oscillation. These regular vortices are not formed unless the strouhal number of the gate motion is greater than about one–half. For lower values of the strouhal number, vortices occur irregularly and at an average frequency higher than that for the gate. If we can make the analogy between an oscillating check valve disc and an oscillating gate, then the Reference 12 experiments would imply that check valve disc oscillations are caused by vortex shedding.

Figure 5 illustrates actual test data (plot of velocity as a function of valve opening) for two different swing check valves. Note the scatter of test data at valve openings of approximately 94 to 100% for the 16 in. check valve. Note also, that when the valve disc moves from 94 to 100% opening the flow will change from 3.5 to 9 fps which corresponds to an instantaneous change in velocity of 5.5 fps. This velocity change will produce a high pressure rise in the piping system; known as a water hammer wave. If the valve disc is open 100%, but because of system flow perturbations the disc moves to the 94% open position, then the flow will change from 9 to 5.5 fps; this will also generate a water hammer wave. If this process is repeated many times, the check valve will be tapping the back stop and wear will occur at the hinge pin and disc stop. This valve is highly unstable, probably due to some vortex shedding.

HIGH VELOCITY THROUGH THE VALVE SEAT

High velocity through the check valve may induce disc tapping by allowing the disc to unseat from the backstop and then repeatedly impact the stop. This is due to vortex shedding cavitation. Figure 6 shows actual test data for a 12 in. swing check valve; the figure shows that incipient cavitation started at about 30 fps. At this flow rate the valve is only 80° open. Note that it takes approximately 40 fps to actually or fully open the valve. For this valve, the upstream test pressure was 90 psig. The disc oscillation is due to downstream pipe wall–pressure fluctuations which are induced by cavitation collapse. Pressure fluctuations due to cavitation may reach a maximum value of six time that of non–cavitating flow.

As an interesting example, one utility experienced seat damage and hinge pin wear due to low flows through their check valves in the RHR system. After extensive engineering analysis and studies, they installed new check valves with reduced port area to increase the flow velocity to 25 fps; and thus, keep the disc fully open. The new check valves also had integral hinge arm and disc. But after eight months of operation, they found premature hinge pin wear. This disc oscillation was due to vortex cavitation at high flow velocities; it also extended their outage.

CONTROL VALVE AND ORIFICE CAVITATION

Control valve and orifice cavitation and its effects on piping systems have induced cost of many millions of dollars to the utilities. For example, cavitation may limit the flow through the valve and thus the valve is undersized; cavitation may cause material damage to valve parts, trim, or valve body, or erodes downstream piping and thus the valve or piping leaks; and cavitation may cause noise and vibration, which may cause major damage or destruction to equipment such as valve positioners, actuators, pipe supports and sometimes to other downstream control and check valves.

Cavitation is a two–stage phenomenon, but the mechanism by which it can be initiated in a control valve or an orifice is best described by considering the sequence of events as the pressure drop across a valve increases while the inlet pressure remains constant. Consider the flow of cold water through a valve. The maximum water velocity is at the minimum cross section area known as the vena contracta, just downstream from the valve orifice. As the pressure drop across the valve increases, so does the flow rate and also the water velocity at the vena contracta. By Bernoulli's equation, this causes a decrease in pressure at the vena contracta. As pressure drop increases, so does the flow until the static pressure at the vena contracta is below the vapor pressure of the water. At this point, water vaporization starts and vapor bubbles are produced.

In the second stage of cavitation, the water bubbles move into a region where the existing pressure exceeds the water vapor pressure in a larger volume that is the downstream valve body or pipe. This allows the water stream to decelerate and regain a portion or the lost pressure. Since the vapor cannot exist at this higher pressure, the bubbles collapse into the liquids state, and thus produce cavitation.

Further increase in pressure differential across the valve increases the number of water bubbles that collapse in the downstream side of the valve and produce more cavitation. Eventually, a point is reached at which any further increase of pressure differential does not increase flow, and the valve chokes; this point represents severe cavitation.

Also, it is essential to review supercavitation and its effect on valves and piping. When supercavitation occurs, the valve and pipe for several diameters downstream are completely filled with water vapor. When this happens,,the valve and adjacent pipe may not be

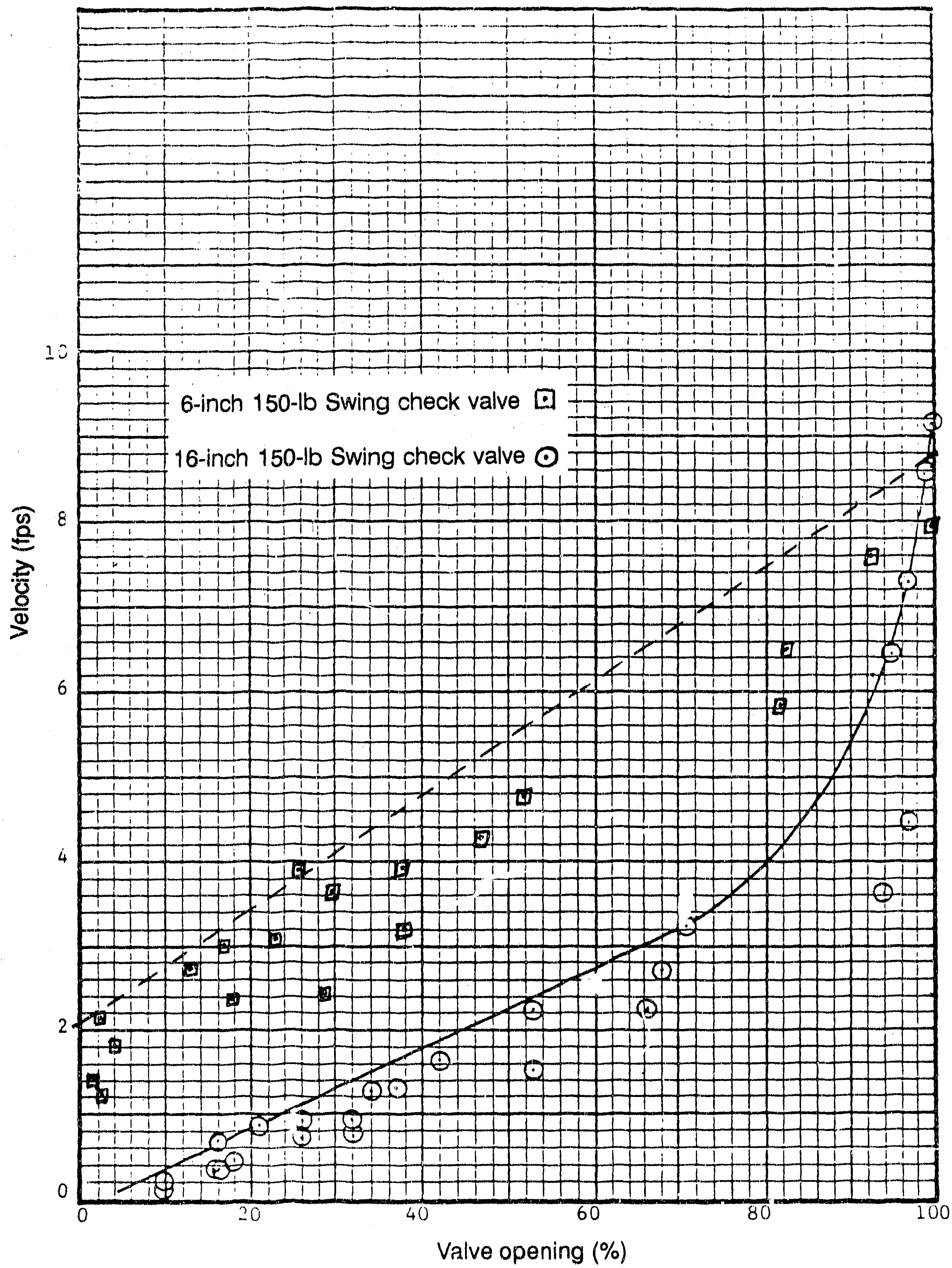


Figure 5. Velocity required to open valve.

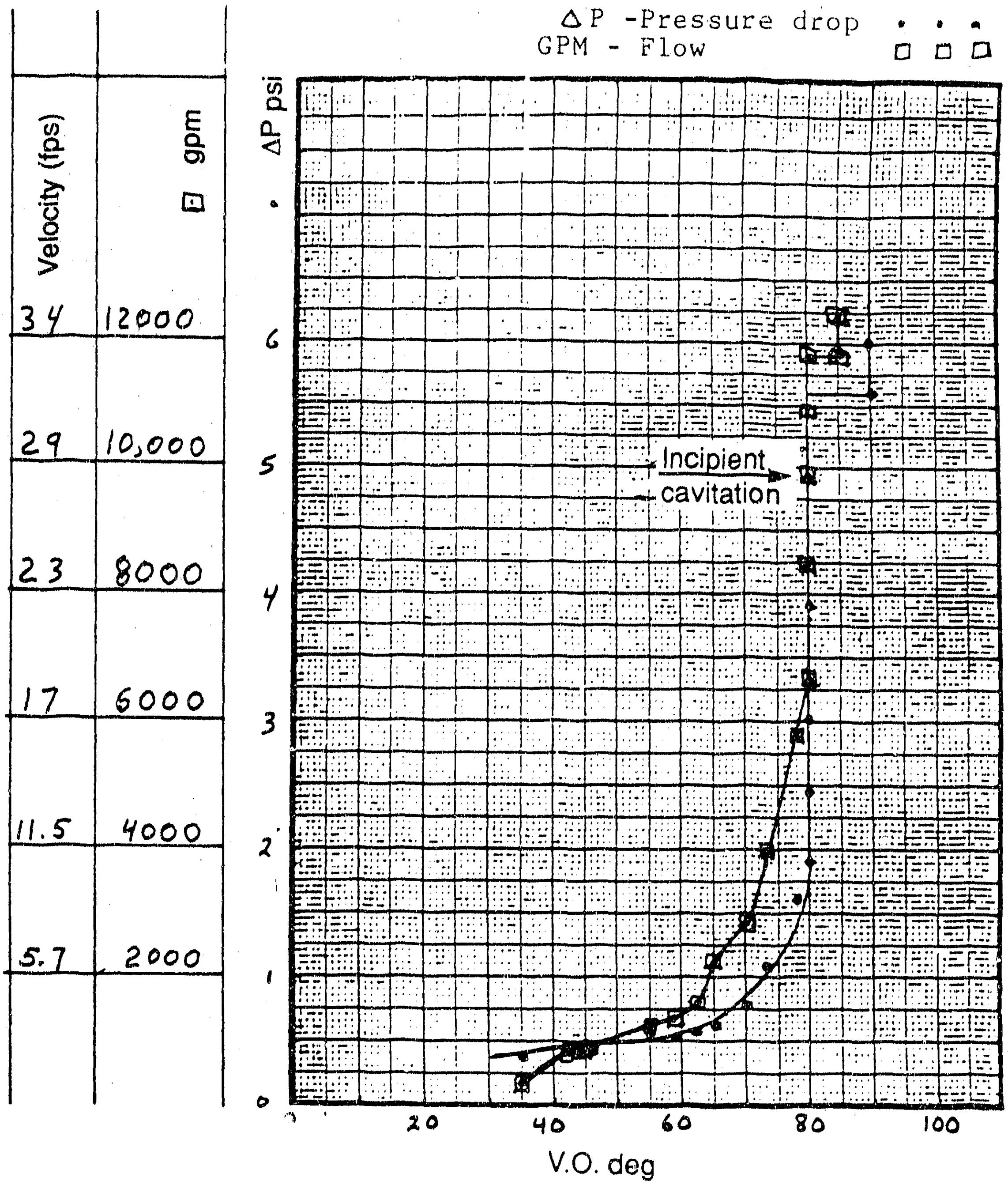


Figure 6. Velocity required to open a 12 in. check valve.

subject to any cavitation damage. However, at the location downstream where the vapor pocket collapses, the system is subjected to sever cavitation.

To show the destructive damage from supercavitation, a 20-in. butterfly valve was tested. During these tests, the valve and piping were fastened to the 3-ft thick concrete floor with 1/2-in. diameter cables at 10-ft intervals. The cavitation caused the pipe and valve to have a displacement amplitude on the order of 1/2-in.; also, it generated a force near the valve of approximately 1,500,000 lbs.

Cavitation collapse pressures are large enough to start fatigue cracks or cause deformations that may later lead to failure. The formation and collapse of large vapor pockets may initiate flow oscillations and periodic pressure pulsations in the piping system. These pressure pulsations may be forcing functions for piping vibrations and check valve disc oscillations (See Table 1 for the possible modes of vibrations). They may also set up acoustic resonances in the piping systems and check valve parts.

For example, see INPO(SER1-84) for "Vibration Failures of Safety-Related Valves." This SER discusses the following valve failures:

- Yoke had pulled away from bonnet
- Bent valve stems
- Valve disc separated from stem
- Loss of valve packing
- Cracked welds on saddle
- Motor casting failed
- Motor bolts sheared
- Stem key was sheared.

All of the above problems are due to control valve cavitation (Reference 17).

But more recently, NRC issued "Information Notice No. 89-01: Valve Body Erosion." This also is a cavitation problem.

As one can see from the above, cavitation induced problems are not only dangerous, but also costly.

In conclusion, control valve and orifice cavitation induced pipe vibrations may be responsible for some of the reported check valve failures. Therefore, it is recommended, that all check valves downstream from cavitating valves and orifices should be inspected. One should not use Criteria 1 and 2 of the Introduction of this paper for selecting check valves for inspection.

WATER HAMMER IN PIPING SYSTEMS

Water hammer is caused by the sudden change in velocity of water. The kinetic energy of the moving water is transformed into pressure energy. At the point of sudden velocity destruction, a pressure wave (or a series of pressure waves) is generated and travels upstream at a very high velocity that depends upon the velocity of sound in water and the elasticity of the pipe. The magnitude of the pressure rise depends on the rate of velocity reduction, the magnitude of the change in velocity, and the velocity of wave propagation.

The pressure wave (or series of pressure waves) travel upstream to the origin of pressure and are reflected from there back toward the point of flow restriction. The pressure waves continue oscillating until the energy is dissipated by friction, pipe deflection, or in extreme cases, equipment destruction. Most explanations of water hammer involve the rapid closing of a valve, which indeed causes water hammer. However, water hammer is caused by the rapid change in velocity no matter what mechanism causes the rapid change. For example, an instantaneous change in velocity of only 9 fps for a 12-in. schedule 80 steel pipe produces a pressure rise of approximately 522 psi. A pressure wave near this magnitude traveling at 4300 fps, hitting check valves and other pipe obstructions can be damaging.

Through the review of plant water hammer experiences (see References 13, 14, 15, 16) seven severe mechanisms have been identified, these are:

1. Condensation collapse in steam pipe after initially injecting steam into subcooled water (water cannon).
2. Steam and subcooled water interactions in horizontal and near horizontal pipes.
3. Subcooled water flow into a vertical initially steamfilled pipe.
4. Hot water entering lower pressure line with subsequent flashed steam bubble collapse.
5. Steam propelled water slug flow.
6. Rapid valve operation.
7. Water column separation and rejoining

Wilkinson made a survey of water hammer incidents (Reference 15) he found 35 cases of failure, of which about 30 were gate valves. He also made a survey of American Nuclear Power Stations water hammer incidents (See Table 2). Also, Reference 14 has documented 281 water hammer events.

What can be concluded from the above water hammer experiences is that, some of the water hammer events are responsible for check valve disc failures; for example, bent hinge pin, disc, and hinge arm.

Table 2. Water hammer incidents in nuclear stations

Site	Type	Fault location	Assumed cause	Damage etc.
Indian Point	PWR	450 mm feed pipe, 19 mm wall thickness	Water slug propelled by steam followed drop in water level after a trip	Fracture of pipe. water loss. flooding, damage to containment
San Onofre	PWR	Feed?	Lack of venting? followed a trip due to LP blade failure	Hangers damaged. safety injection valve actuation failure
Ginna	PWR	Feed	Valve plug broke and shut off flow	Pipe supports damaged
Dresden 2	BWR	Steam line to ECCS turbine	Water slug on re–instatement after maintenance and trip	Broken pipe supports damaged snubbers and pipe
Quad Cities 1	BWR	CW system	Sudden closure of 2540 mm butterfly valve	Burst rubber pipe joint. Flooding caused damage to 17 pumps
Beznau	PWR	Hammer in steam line	Debris in feed valve causing boiler flooding	—
Dresden 3	BWR	Feed	Sudden partial feed valve closure, cause unknown	Pipe supports damaged, control lines broken
Nine Mile Point	BWR	—	Lack of venting and condensate drainage	—
Palisades	PWR	—	—	Seismic pipe restraint failed
	BWR	ECCS	Lines empty on pump start	62 mm movement of header. Minor damage
	BWR	Residual heat removal system	Inadequate venting	Pipe supports damaged valve actuator failed
Fitzpatrick	BWR	Service water system	Lines not full on pump startup	Pipe damage supports broken
Cook 1	PWR	Feed	—	Hydraulic snubbers damaged
Browns Ferry	BWR	Turbine steam line	—	Components damaged
Tihange	PWR	Feed	Water slug following drop in water level	None

The seven causes listed in previous listing of water hammer brings to mind two interesting check valve failures: stable disc limit cycle oscillations and steam propelled water slug flow.

In the first case, the problem was encountered when swing check valves were used in applications where rapid pump shutdown occurred (see Reference 11). On pump shut down, the check valve disc would bounce several times on its seat at a well defined frequency. The severity of the repeated disc slammings caused shearing of the bolt securing the disc to the hinge arm. It was also, discovered that as the disc damping was increased, stable disc oscillations developed. That is, the disc would never seat. This was due to vortex shedding from the seat ring.

In the second case, many turbine exhaust check valve failures have been reported. The failures are due to steam propelled waters lug flow. Typical failures are bent hinge pin, disc, and hinge arm, as well as broken disc stud.

PUMP INDUCED PULSATIONS

Pump pulsations are considered to be the major cause of pipeline vibrations. If the piping system is subjected to a forced frequency from the pump, severe oscillations of pressure and flow may develop in the piping system. This may result in serious damage or failures of the piping system. Many incidents and accidents caused by pump induced resonance have been reported (Reference 18). Pumps may induce many different problems to the piping system but the three most important ones are:

1. Radical GAP A
2. Radical GAP B
3. High suction specific speed(SS).

Figure 7 shows a typical boiler feed pump stage and gives the nomenclature of the impeller and diffuser geometry. If gap A is too large, low frequency vibrations and oscillations are developed in the piping system; these frequencies are between 1 and 10 Hz. This is the natural frequency of the check valve disc and thus, would induce check valve disc oscillation.

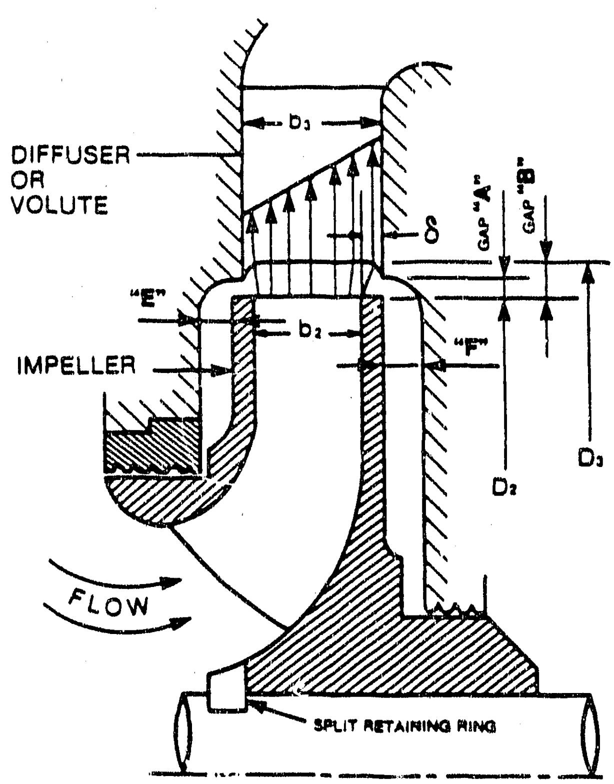


Figure 7. Nomenclature of impeller and diffuser geometry.

If gap B is too small, a strong shock is generated each time that an impeller vane passes a diffuser vane inlet or, in a volute type pump, a volute tongue. The distinct frequency of this pressure shock is equal to the pump rpm times the number of impeller vanes. For a 6–vane impeller operating at 5400 rpm, the vane passing frequency is 540 Hz. This frequency may very easily interact with one of the check valve modes (see Table 1) and thus, would resonate the check valve to destruction.

As an example, Figure 8 shows a GAP A and GAP B hydraulically induced instability. Note the peak to peak pressure pulsations of up to 150 psi. These pressure pulsations may be destructive to any check valve downstream of this pump. GAP A and B modification corrected the problem.

Impellers with high suction specific speed (higher than 10,000) induce hydraulic instabilities in the piping system; flow oscillation between 1–10 Hz. This pressure oscillation would also induce check valve disc oscillation. Figure 9 shows the operating range of these pumps. This flow instability will increase as the flow decreases below design flow. For this same problem, NRC issued Bulletin 88–04. The above points have been the subject of an excellent discussion by Dr. Makay (Reference 18–20).

As a second example, because check valves, control valves, and actuator components downstream of the boiler pump extraction piping were failing, which required extensive maintenance, an analysis was performed. Measurements showed that fluid pressure pulsations were the probable driving mechanism for the mechanical vibrations observed on the extraction lines of the pumps. The vibrations were generated by the interaction of high pressure pulsations within the fluid and mechanical resonances in the piping. The pressure pulsations originated at the interstage discharge of the boiler feedwater pumps and were induced by the blade passage of the pump's third stage impeller. Analysis showed that the pulsations could be reduced by installing hydraulic acoustic filters in the extraction line of each pump; and Figure 10 shows the results.

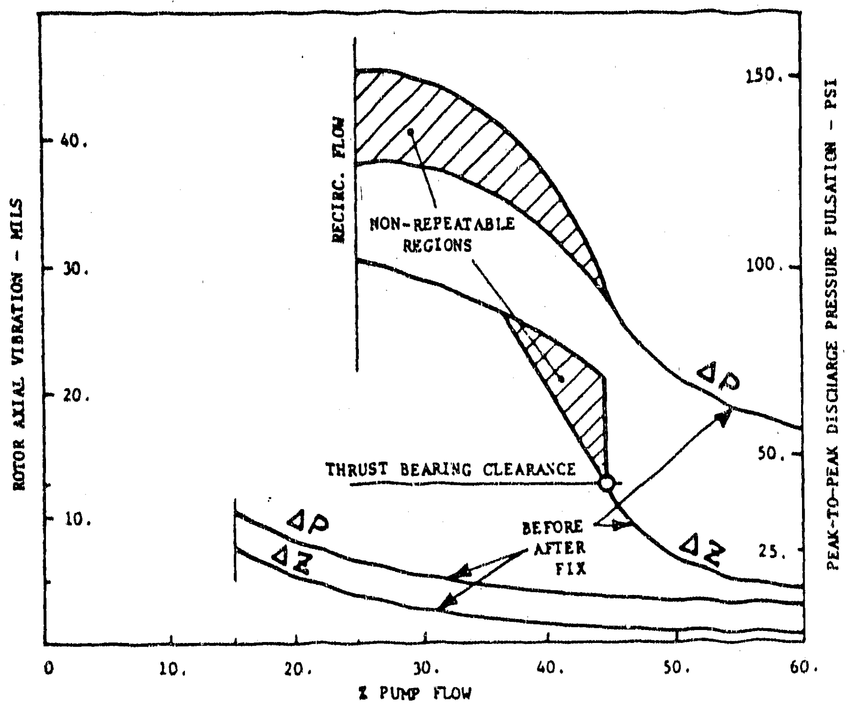


Figure 8. Peak–to–Peak pressure pulsation (ΔP) measured in the discharge nozzle, and rotor axial vibration (ΔZ) of a high speed reactor feed pump with two different impeller discharge configurations. Pump efficiency at 100 percent flow was the same in both cases.

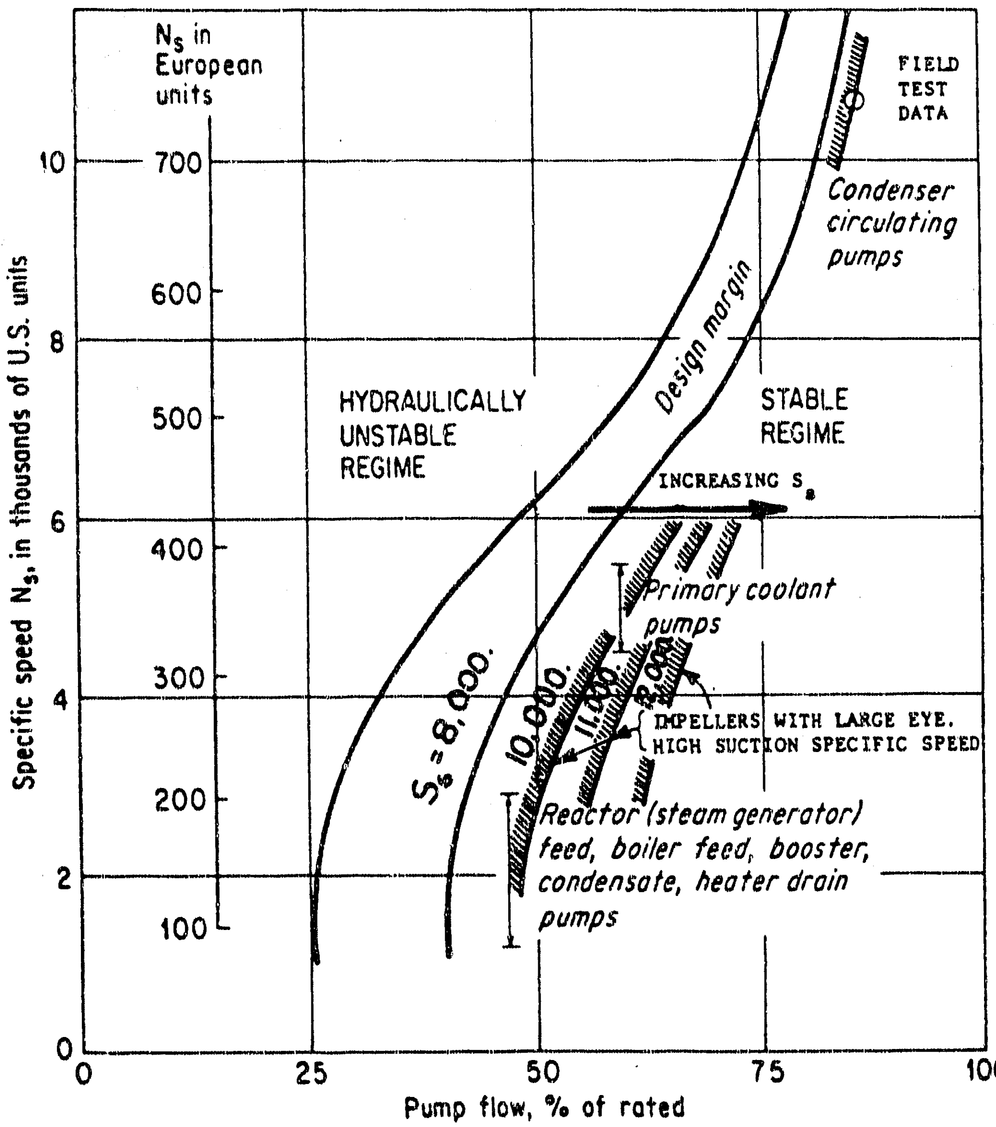
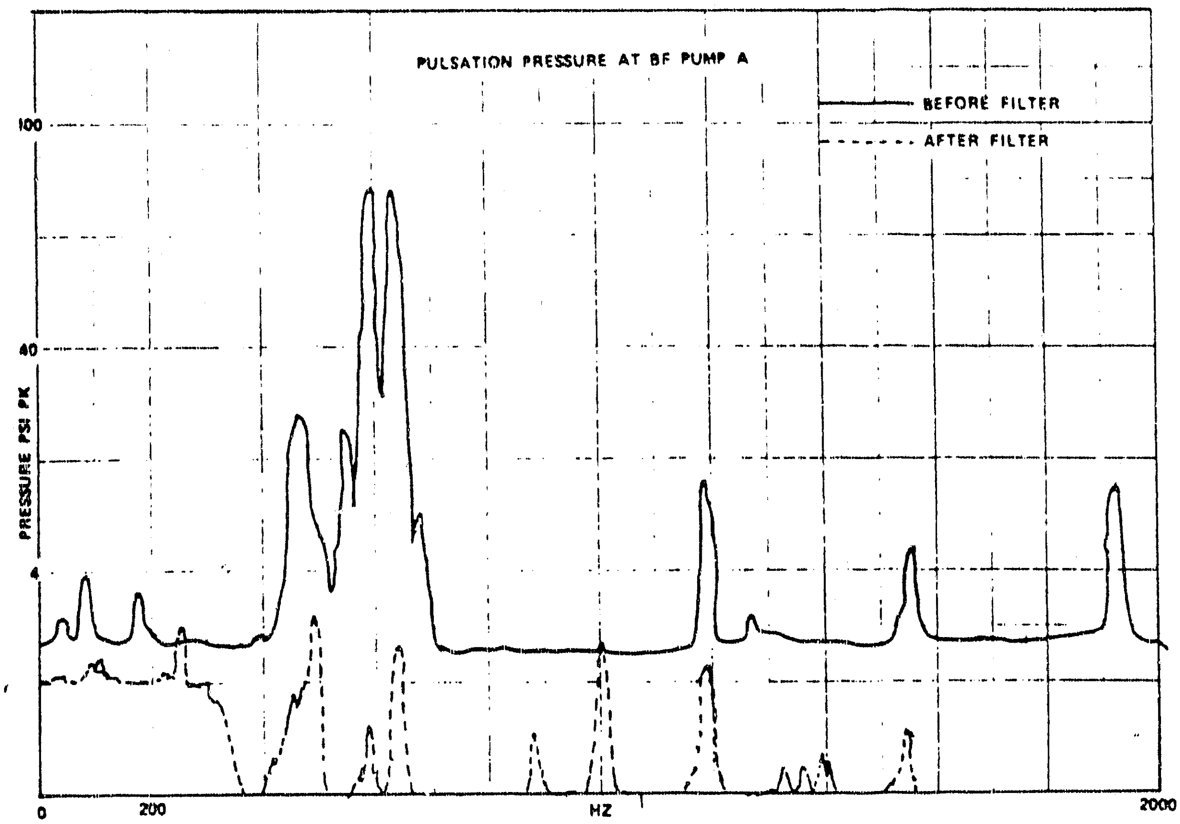


Figure 9. Anticipated useful operating ranges for pumps used in large units. Both nuclear and fossil power generating units are included. The inner line is for a properly–designed impeller stage. The design margin is a function of the designer's ability and experience. Impellers with eye larger than normal, or with high values for suction specific speed (ss), will have a more limited operating range at off–design flows that may require very large recirculation flow rates.



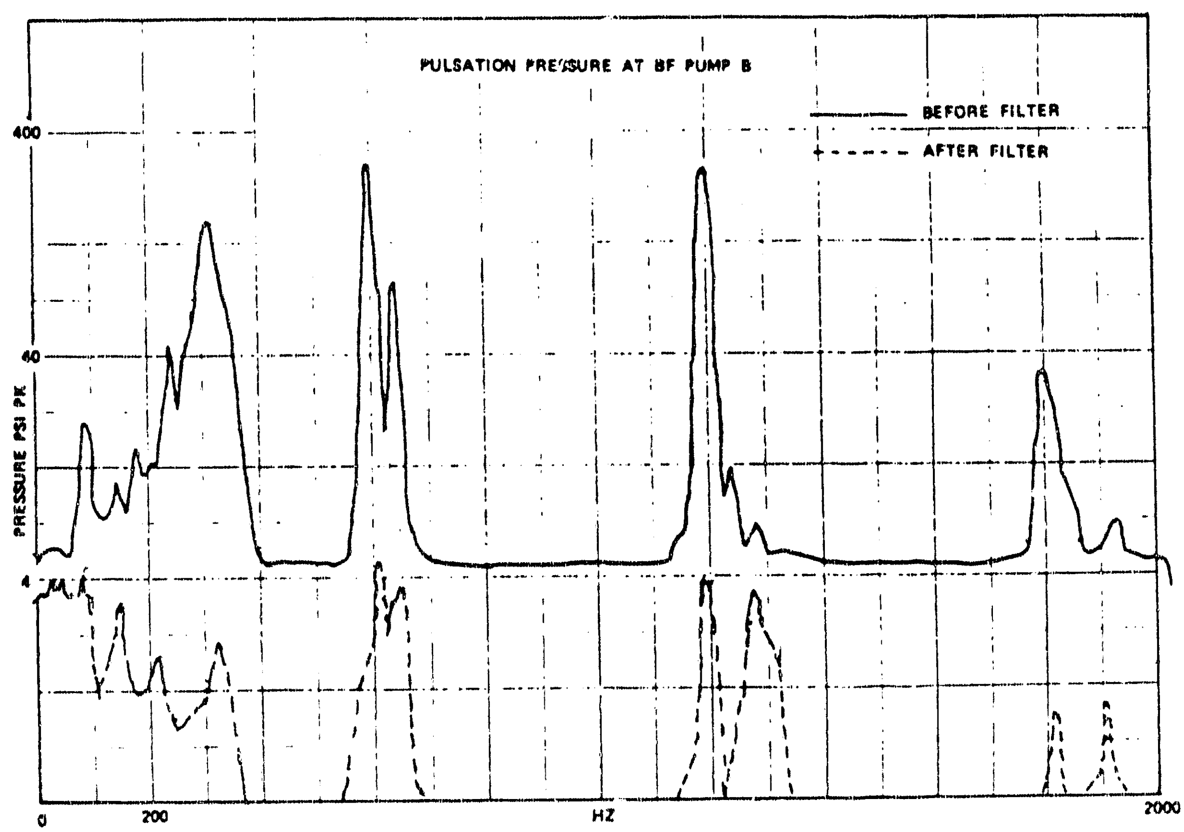


Figure 10. Before and after pulse pressure measurements of the fluid in the extraction piping for BF pumps A and B respectively.

CASE HISTORY TO ILLUSTRATE THE PROBLEMS

The pump discharge through discharge valve configuration is shown in Figure 11 and a typical installed check valve is shown in Figure 12. The diesel generator trip events of October 23 and November 19 occurred due to high generator cooling jacket temperature, indicating that there was insufficient cooling flow to the diesel generator heat exchanger from the diesel generator cooling water pumps. The cooling water pump discharge check valves were disassembled and found damaged. The valve discs had become disconnected and lodged in the valve discharge.

Because of the above problem, six swing check valves in the discharge side of the DGCWP system were inspected; three from Dresden and three from Quad Cities. All six check valves had failed. The valves were 8–in. swing check valves mounted horizontally. The failures involved broken hinge arms, excessive wear, and corrosion. Figures 13 to 16 show typical damage found. Interestingly there was no hinge pin wear (see Figure 16).

The check valve installation can be seen in (Figure 11 to 12) and consists of an 8–in. check valve with inlet of a 4 x 8 reducer and a 4 in. elbow. The minimum velocity through the Dresden check valves was 7 fps while through the Quad Cities it was 12 fps. Failure of these check valves was probably due to three factors:

1. Possible low flow velocity through the valve which induced valve disc slamming against the backstop.

2. Pump start up water hammer induced by pumping water into a voided pipe line; this would induce disc slamming against the backstop. Figure 14 shows the washer partially bent.

3. Possible pump instability problems.

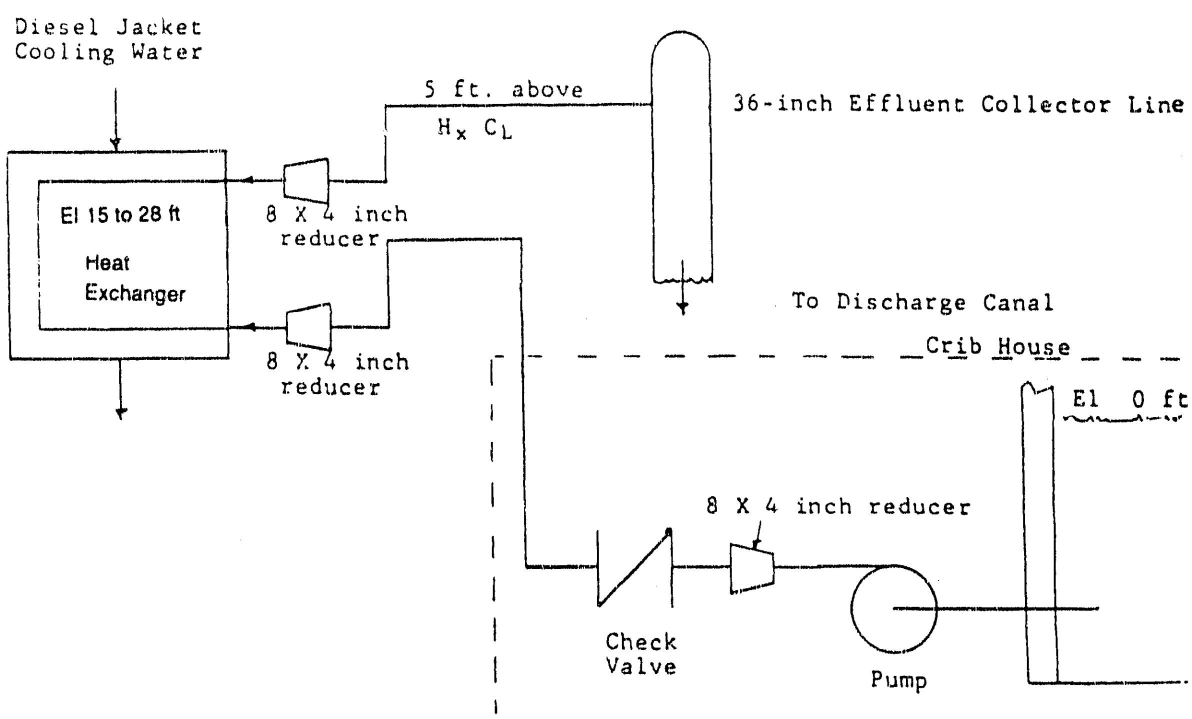


Figure 11. Diesel generator cooling water system.

Figure 12. Typical diesel generator check valve installation.

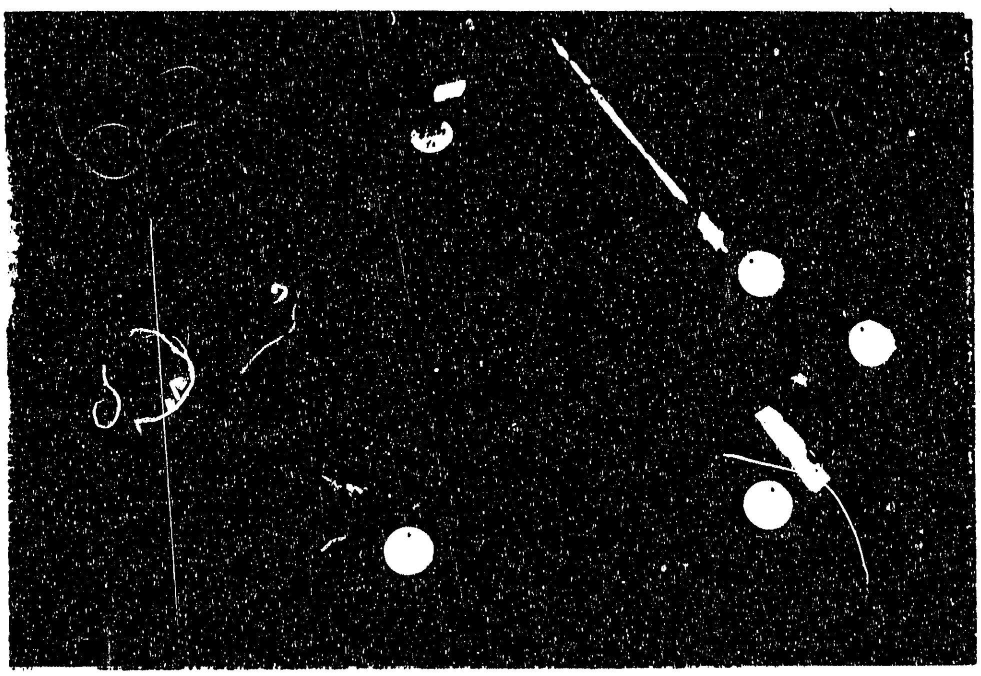


Figure 13. Damaged check valve components from Dresden and Quad cities.

Figure 14. Close–up of disc hinge connection on Dresden No. 3 disc. Note washer used to retain the disc on the hinge is partially bent.

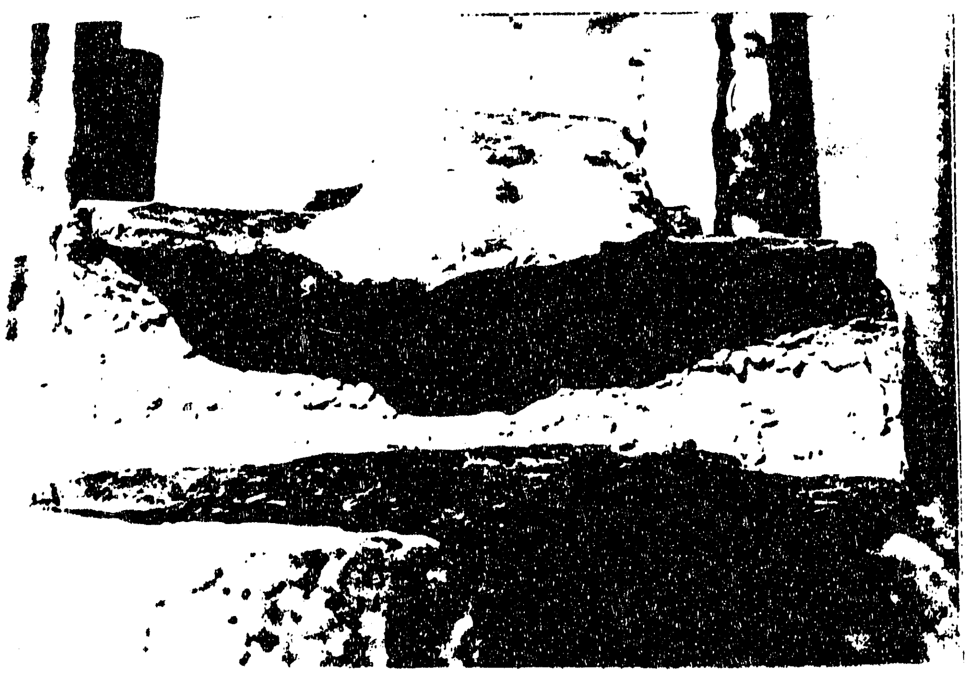


Figure 15. Fractured and missing area at pin hole from Dresden No. 2 hinge.

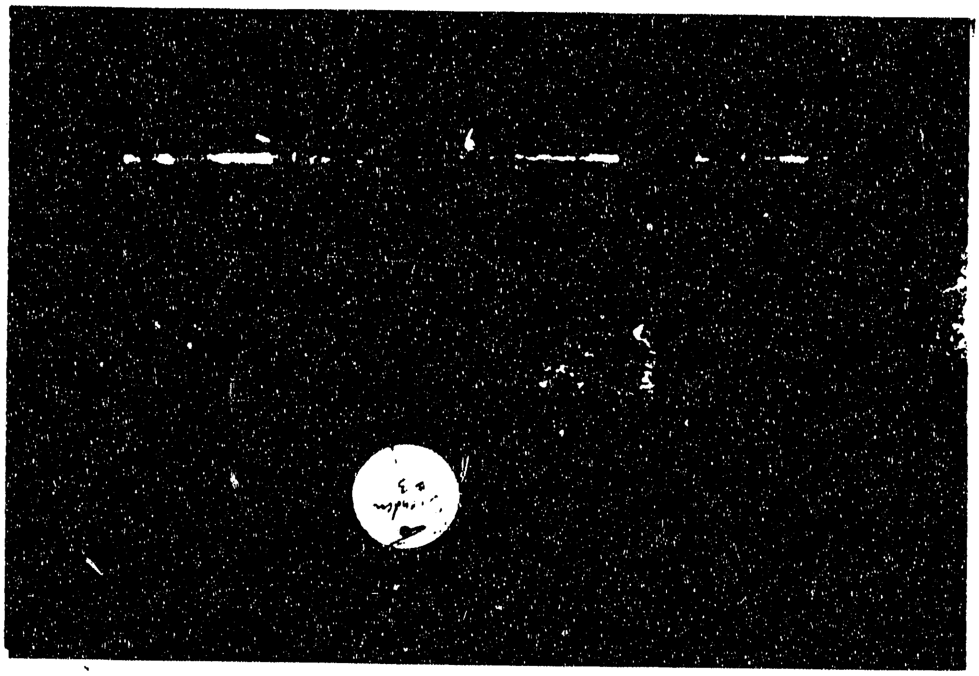


Figure 16. Dresden No. 3 hinge and pin.

The above last item will be discussed further. The original DGCWP specification called for a pump flow of 1304 gpm at a pump head of 210 ft. However, the pump operates at 1850 gpm, but this induces a different sort of a pump instability. Namely cavitation and thus, high pump vibrations.

Station maintenance records show that significant corrective maintenance has been required on the Unit DG Cooling Water Pumps and Motors. Corrective maintenance over the past 7 yr has included:

1. Six rebuilds of the rotating element including replacement of one shaft.
2. Replacement of two complete pump rotating elements.
3. Two motor overhauls.
4. Replacement of two motors.

CONCLUSION AND RECOMMENDATIONS

From the above, check valves may fail in many subtle and salient ways. Even if the check valve had sufficient velocity to lift the disc in an open position, and even if the minimum L/D was more than 10 ft, the check valve may still fail due to one or a combination of the five failure mechanisms listed in the Introduction of this paper.

To have an excellent check valve program, it is recommended to either inspect or monitor the following check valve installations:

1. Check valves that have higher than 25 fps flow through them.
2. Check valves that are located either upstream or downstream of cavitating valves and orifices.
3. Check valves known or possibly have experienced water hammer.
4. Check valves that are installed either upstream or downstream of pumps that induce pulsations.

The above requirements are an addition to the INP SOER concerns and requirements. To accomplish an effective check valve program one must first start with an effective maintenance history review of failed check valves. For example, the three feedwater check valves that failed at San O'Nofre, namely FWS–345, FWS–346, and FWS–398 also failed before; their maintenance history can be summarized as follows:

- 1975 Refueling—Inspected three main feedwater regulator check valves and installed new internals in all three check valves.

- 1977 Refueling—Inspected and installed new internals in all three check valves.
- 1978 Refueling—Inspected all three check valves, found no problems, cleaned parts, and reassembled valves.
- 1980 Refueling—Inspected all three check valves, replaced cotter pins, washers and nuts on flappers, and reassembled.

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ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Comment 1

We are having problems with feedwater check valves, back leakage problems. It seems to be a generic problem.

In my paper I referenced Weaver's work in Canada. He calls it limit cycling. If the check valve is shut down, it will go into the seat and oscillate.

Sometimes it will just go limit cycling, that is open in the seat and come back in four or five degrees.

I spoke with Weaver. He did an analysis for an actual installation. And if you use dampers, he says how the valve manufacturers perceive the dampers incorrectly.

Limit cycling on a high energy bumping system may explain some of the back leakage problems and also any cracks on your disc. The type of damage is discussed in my paper. But that is an interesting point because we are having problems with the feedwater system check valves, back leakage, high energy—something worth looking at.

PANEL DISCUSSION: IST OF VALVES; EXPERIENCE AND LESSONS LEARNED

T. SULLIVAN (MODERATOR), O. ROTHBERG, S. MCLEAN, J. HIGGINS, J. OZOL, J. WITHERSPOON

Question 1 (Relationship Between GL 89–10 and GL 89–04)

Please discuss the relationship between the MOV generic letter and Generic Letter 89–04. Are they closely tied together, or are they independent?

Response 1

It is an administrative nightmare. They were obviously conceived independently. As I understand 89–04, it deals with the ASME Code testing. 89–10 deals primarily with a specific problem that we have with motor–operated valves. Right now I am hesitating because I see a lot of problems. Potentially, it may not be a good idea to stroke test motor–operated valves periodically. The ASME Code says you should. We wrote the generic letter assuming that the stroke timing tests would continue. The Germans don't particularly care for the stroke timing test. But the Germans use a lot of "self–lubricating materials."

You have this administrative problem. 89–04 is associated with the ASME Code, and the ASME Code is mandated by 10 CFR 50.55, and there is a problem of compliance. Whereas, 89–10 is driven by a fact of life, a problem we have with motor–operated valves. So I don't know that there is a conflict. I know that I would like to see the actions done that 89–10 talks about.

Question 2 (Alternatives to Stroke Testing)

Conceptually, what would be an alternative to the existing standard, which might improve stroke time testing for MOVs and AOVs?

Response 2

As I stated, I am not sure that stroke time testing is appropriate for electric motor–operated valves or solenoid–operated valves. An alternative might be to derive a stroke time out of a signature analysis for valves. But of course, a problem with that is coming up with an accepted criteria for signature analysis. I am not sure that there really is an alternative at this point, and I don't know if there is an alternative for AOVs.

Question 3 (Check Valve Inspection)

Can we say, at this time, whether there is any case of check valves which have been inspected, which did not need to be inspected?

Response 3

We have not seen that in our research for the paper, nor is there enough industry experience on a particular class or application. However, the NIC group indicated that they may accumulate that kind of information to disseminate it to the utilities.

Comment

I would just like to add, that if it works, don't fix it. But to apply that, we do have to gather initial data.

Question 4 (Check Valve Inspection)

Could we clarify whether there isn't enough information because the data doesn't exist or because they are finding problems with every application that they look into?

Response 4

Our research did not come to a conclusion on that. We didn't specifically look for it. So now, I will talk about our personal knowledge on the subject. Disassemblies have sometimes indicated problems, sometimes they have not. Whether there is enough information on a particular class or design or application to take a generic relief request, I do not know that knowledge is available today, although it may be.

Comment

I want to mention something about disassembly of valves. The valve manufacturers indicate that if there is a problem with a particular valve, maintenance records will show a repeat problem.

If we consult maintenance records, we may be able to take valves off our program. For example, the valve that failed at San Onofre had no lifting force. The disc was all the way horizontal. But many people use the same type of valve in power service. That valve had had no problems whatsoever. Looking at the maintenance records closely, if you eliminate all of your recurring problems, then you can cut back the frequency of inspection. And incidentally,in San Onofre case, they took 48 valves apart after the five failures and found no problems with the 48 valves. Other stations have disassembled many valves and found no problems with them.

Question 5 (Vortex Shedding on Check Valves)

If a check valve is properly and fully back seated, does vortex shedding present a problem with disc oscillation? Is your answer based on actual field results and observations?

Response 5

I'll give you an interesting story that really happened in a power station. The original problem was a worn-down main pin. There was not enough minimal velocity, so the disc was oscillating. The personnel put in a new valve and reduced the port. Now there was high velocity through the valve. The valve had a 25-feet-per-second maximum velocity. The pin was worn down, and that definitely was from a vortex. If the disc is stuck further and further in the stream, a different sort of vortexing occurs. I have seen this from other dissertation work. Perhaps acoustics could distinguish this vortex. I believe that we could use acoustics to pick up the sound.

Question 6 (Check Valve Disassembly)

One of the speakers implied that the only way to properly determine that check valves function properly is disassembly. Are non-invasive methods acceptable and useful? If no problems are found and system parameters do not change, can future inspections be eliminated? If not, what are your reasons?

Response 6

If I implied that disassembly was the only method of verifying proper function of the check valve, I did not mean to imply that. The Code says that full flow testing is the method to test check valves. Disassembly and inspection are performed if full flow testing cannot be performed.

The other part of the question is, "Are other non-invasive methods acceptable and useful?" Some utilities now utilize non-intrusive methods. I don't know of any particular non-intrusive method that is widely accepted, although the NIC group is evaluating that. Their charter includes the evaluation of the current non-intrusive methods.

The next part of the question is, "If no problems are found and system parameters do not change, can future inspections be eliminated?" You have to live to the Code. You can use that as a basis for a relief request. Generic Letter 89-04 has specific instructions as to the inspection interval and frequency.

Comment

Panelist Disassembly can be utilized only when full flow, i.e., full stroke, testing can not be performed.

Panelist We need to distinguish between testing and maintenance. I think disassembly is basically a maintenance procedure, and I think this presentation concerned taking valves apart for different purposes. But in either case disassembly is basically a maintenance procedure that we are following, albeit a disassembly for the generic letter purposes where the valve is stroked manually. It is really not a test.

We need to get the most out of testing we can, whether it is partial or full flow. I also think that, whether or not a full flow test can be done, valves need to be looked at internally on some occasions. The best way to find out what the condition of the valve is to take it apart.

Different things are going on here. We need to be clear about testing for operability purposes versus maintenance, to tell what is going on inside the valve.

Question 7 (Check Valve Disassembly)

Would inspection procedures for a check valve be different for a valve that is normally open during operation versus one that is normally closed?

Response 7

No, the disassembly procedure simply identifies a visual inspection of the valve. I don't see that you would differentiate a procedure between whether the valve was in a normally open or normally closed position.

Question 8 (Check Valve Disassembly)

It appears that check valve disassembly and inspection provide some information that cannot be determined by non- intrusive testing, such as the existence of valve component corrosion, erosion, wear, chatter, misalignment, and general mechanical integrity. Should this testing be performed on all check valves periodically, or are the disadvantages and costs too great?

Response 8

Some utilities have performed an application review similar to the EPRI application guideline for the check valves in their program. As a result of that review, those valves that indicate a potential problem are added to the disassembly and inspection list. That eliminates the need to go in and look at every check valve. The review identifies only the potential problem check valves.

Comment 8

Panelist I'd agree that you do not want to overtask the system by including valves randomly into the program, because disassembly is certainly costly, as mentioned in the question. Again, if there is no method to test via full flow, you would disassemble. Or on a case by case basis, if the utility identifies valves that have had repeat failures and chooses to investigate them through disassembly, that may be appropriate. But you certainly would not want to add any great number of valves into the program.

Panelist Let's say that I take a valve, I do not know its condition, and I flow test it. And if I do not know the position indication, I get my full flow through it, although the valve may be open only 60 percent. What if the valve has been damaged previously. You would not pick that up. Now, if it is a linear check valve where the C_v is proportional, a full flow test would go all the way up and the valve should reseat. You could say that you may infer something from it.

But you will not detect erosion/corrosion problems and minor wear of the valve by full flow test. The results of further inspections and the literature indicate that the best way to test a check valve is seat leakage test. For example, if the disc shimmies, it will wear off one side. Then if it catches, it will drop a little bit. And the valve does not pass the seat leakage test, the test will pick up something. Look at each individual valve design. Now, incidentally, Commonwealth Edison contracted an architect engineer to do a European survey on check valves. We did the survey in the German way. The Germans have a dedicated team of people that go out and take valves apart periodically to look at them and replace all the parts.

If there is a chronic problem with a check valve, they will install a larger check valve than line size. Perhaps one of the reasons is water hammer, because the waves are slowed down. They have a dedicated group of people who know what to do and how to do it. I know it is quite costly, but it seems to work for them. In my opinion, the seat leakage test is number one, and next would be flow testing. With these, you will pick up perhaps 50 percent of the problems.

Comment

Panelist It sounds like there are two diverse issues here. One would be check valve reliability. You may be evaluating it in accordance with the SOER 86–03 or an EPRI application guideline review. And, then the requirements for testing from ASME Section XI. Section XI wouldn't endorse necessarily all the requirements of an application review.

There are two things to look for in disassembly, check valve reliability and testing requirements for Section XI.

Question 9 (Flow Test at Check Valves)

It was just mentioned that you may be able to pick up 50 percent of the problems doing flow testing. I would like more information about the parameters measured and the problems that can be found by flow testing.

Response 9

Yes, that is a tough one. The reason is that it has to do with the gain. For example, in a high gain valve, the steep curve, if the pin wears down just a little bit, it will drop and suddenly increase or decrease the flow. This is in normal operation. The valve is working, but it may be affecting your system hydraulically. The plant people pick up a vibration problem, perhaps a damaged snubber or something else. If this valve is flow checked, what will it tell you? It is going to tell you that the disc swings up and down, goes open, closed, and passes flow through it. However, a flow check does not pick up that the pin is worn down a little bit.

Now, the next question is, how far can the pin wear down? We really don't know. You would have to do a stress and strain analysis on the pin. In fossil stations, I have seen valves damaged badly and chewed to pieces, and they back leak once in a while but, they run the system.

Now, what is interesting about a fossil station is that their systems have much more energy than a nuclear station. It seems to work. If the valve is leaking, or if a problem is found they just replace the whole valve.

But I have not seen any correlation. The physical question is, what can you get by flow testing? What does it tell you? If you do not get your full flow through, you have a problem. The valve is stuck somehow. The valve could be stuck in the body. My opinion is to use seat leak tests. Those are excellent. It is hard to imagine a seat leak test not picking up a problem. But for corrosion or erosion, the only way you would find that is to take the valve apart and look at it.

Question 10 (Flow Test of Check Valves)

We talk in the regulatory arena about partial flow testing and full flow testing. I wonder if you can comment on what you see to be the value of either one or both of them?

Do you get anything that is really worthwhile from going to the extra trouble to do a full flow test as opposed to a partial flow test?

Response 10

We know that if you do a full flow test, the valve will do what it is supposed to do, which means passing the flow needed for emergency conditions. That is guaranteed. And we should do those tests. Now we know our system is functioning properly, and that is what we are after. If we can put the full flow through a partially damaged valve, we have satisfied the safety conditions.

Many times these valves will work even though they are damaged. Such as San Onofre's case, or the Dresden problem, where the disks were sheared off and stuck in the line. We don't know how long those valves were stuck in the line.

We ran the system and it worked. It is an interesting example. The only time you worry is if you lose a disk that may block the line.

I know one fossil station that lost a disk. It jimmied in an elbow and restricted flow to such a degree that it was cavitating the elbow. The fitters went by and said, something is wrong, all this noise coming from the elbow. The next outage, they cut the line out and found a check valve disk, there. And the system was running. So it is not intuitive. But if you do a full flow test, you can say the system works, but it does not necessarily follow. Then you can say perhaps you have no major damage to your valve. So we should do some tests. How much damage can a check valve take before it is picked up? This is a problem of experimenting.

The only people I know in the United States doing work like this are Professors Tullis and Rameyer at Utah State, Professor Martin at Georgia Tech, and a professor at MIT.

What we should do is fund research to use a damaged valve, run full flow through, and see what happens. It would probably run a few million dollars, which in a country of our size should be no problem. It would be a worthy experiment. It would settle the issue. That is the whole point.

Question 11 (Flow Test of Check Valves)

Can you tell me the probability of a valve that is partially open going to the full open position? Is there any estimate of that?

Response 11

Panelist I would say from my experience, if a valve passes a partial flow test, it will pass the full flow test. The best way to know is through a flow gain curve, but those are hard to come by. Then you can get the C_V and put a pressure gauge across it. You can actually calculate it accurately.

But we do not have flow testing. Most of the check valve manufacturers do not flow test their valves.

But when you counter-phrase the question asking can I guarantee the valve is not damaged, I am saying there are some situations where it may be damaged and pass a flow test. But I am saying also that the damage might be minor, but could ruin the system.

Panelist So you are saying that you get some assurance, but it is still not a substitute for making sure by an actual test that it can pass the design flow?

Panelist Right.

Comment

Attendee Before we decide to disassemble a check valve, often we acoustically monitor it to see if we can determine if the pin is rattling or there is any tapping, anything of that nature. And whatever we detect will go into making the decision of whether or not we ought to disassemble the check valve.

Question 12 (Radiographing Valves)

Is there any evidence that radiographing a valve can be useful in assessing its condition or position?

Response 12

At Pacific Gas & Electric we have done some radiographic examinations of check valves and have fairly successfully determined the internal conditions of those valves. That is, we have been able to observe the configurations of the nuts on some of the valves and the position of the discs on those. And I think it is a reasonable effort if you do it properly.

Question 13 (Radiographing Valves)

Have you been able to do that kind of examination with flow through the valve?

Response 13

These radiographic examinations were conducted with flow in the valve with the disc full back seated. There could be distortion in the flow stream and wiggling of the disc itself that will blur the geometry of the disc and nut assembly. But if it is fully back seated against the valve, you can get some good indications of the state of the internal disc assembly. The only problem with that, of course, is the radiation level needed for that and the restrictions of plant personnel around the area. But other than that, from a technical standpoint, it is perfectly feasible.

Comment

I am the chairman of the nuclear industry check valve group. One of the items we're going to look at under phase one of our test program is real time radiography. We have asked industry what they use to look at check valves under real time radiography. Only two companies have said they have done any of this testing. One response was from Pacific Gas & Electric. Their results have been favorable. The cost of this particular type of testing would probably be more than any utility would spend. The particular method that we evaluated shows some good promise, though it is beyond the scope of one utility using it, unless it is one application, because of cost.

Question 14 (Radiographing Valves)

What power supply are you working with?

Response 14

They told us that it would probably take 4 kV. One particular company told us it may take up to 4 kV for them to be able to read a 16–inch tilting disc check valve. I think we were talking to them about a 160–pound valve.

A committee called the Non–Intrusive Examination Committee went out to all non–intrusive vendors, including Real Time Radiography, and obtained this information. If you come to our meeting in August, there will be some talk about that, what we received from the vendors, and other information available.

Comment

We haven't radiographed check valves, per se, but we have radiographed some small valves to determine valve position with excellent results. We had no problem at all seeing the disc position.

So, I think the same thing would work.

Question 15 (Oversize Check Valves)

The impression was given during the talk that a larger valve is sometimes installed in a line. This is probably the wrong way to go. I think that a check valve should be installed with a flow velocity of 7 to 20 feet per second.

Response 15

It is a good question. I agree with you 100 percent. But that is what the Germans do. Our AE told us that. We asked why the Germans are doing this, because they are good in hydraulics. There are two reasons.

The number one reason is the classical water hammer problem with the check valve. If the valve is shut down, the valve comes down. If a reverse flow occurs, that is, a high velocity reverse flow through the check valve, then it closes and tremendous water hammer waves occur that might do damage to your system in terms of supports.

However, if a larger body is used, in the case of the Germans, the velocity coming into the body is much lower. Now it is protected against water hammer, that is the water hammer induced by the check valve when a pump is shut down.

Now, I do agree with your philosophy. Seven f[illegible] to 20 feet per second is a very good velocity, and the only reason that the Germans might use [illegible] larger size is, perhaps, to alleviate water hamm[illegible] [illegible] line. And I was mentioned that a system's [illegible] [illegible]mer induces failure to your check valves. That is the opposite problem.

Comment

Could I add something to that? Usually the way these things were designed in the old days—and the old days are not that old—nobody sized check valves.

The piping system was sized and then the check valves were added the same size as the pipe. The criterion that has been around with several manufacturers for years uses an equation, with different equations for swing checks and tilting discs. But generally, the minimum velocity that has been used to keep a check valve open is 7 feet per second. In most cooling water systems with heat exchangers, you try to keep the velocity low, so you normally size the piping for 4 to 7. You are already having a problem.

For the feedwater system, you are looking at 10 to 20 on the other side of the spectrum. And for safety-related components, like certain RHR systems, flow rates range from 15 percent of design point all the way up to the design point and beyond.

No one check valve will furnish that range of flows. Normally, piping systems in plants, included too large valves in the cooling water system.

In a main system, like a feed or a condensate system, it was not as bad. In a safety-related system that normally was dormant except for tests, the valves were grossly oversized, too. So that is why it is strange that they would say they would put a bigger valve in a smaller line.

Comment

That was a very good point. Incidentally, when you start up your main feed systems, you start up with your small valves bypassing—you put in a bypass control valve to your feed system. You start and run the plant the big check valves will start oscillating right off the seat. That is right.

I believe the EPRI guidelines address part of it, but that is something that is worthy of note. I agree with him 100 percent that perhaps we should build bypasses. I still don't know why the Germans are putting in larger valves. When they have a problem today, they will put in a larger check valve. But he is right.

Question 16 (Detection of Check Valve Failures)

It appears that position verification by means of full flow or seat leakage is preferred by the NRC.

Have you had experience where the full flow and seat leakage tests pass their test acceptance criteria and later found the check valve borderline or falling apart? We have. If so, how do you detect this?

Response 16

I've known a couple of cases from other utilities' experiences where they did pass the seat leak test and the valve was, in fact, failed. I believe one of those was a broken bolt. You're 100 percent correct and I really can't answer that question. But I would say the best we have is seat leakage testing and I have great faith in it.

What does the valve do, how does it fail, what will the test pick up? All the failure mechanisms in the EPRI guidelines and elsewhere may damage it, but the disc will drop somehow or cock, that is, tilt. For example, the limit cycling one, you'll crack the disc. If you have a cracked disc, a partial crack across the seating surface, you are not going to pass the leak rate test. Or if it is a fine crack, perhaps you will. But that is rare. I am talking about the probability of passing the seat leakage test. And with a damaged valve it is highly unlikely, although it does happen.

Comment

Panelist You are indicating that you have found problems and you are asking how to detect it. I guess I would like to ask you, for the cases that you are mentioning, how did you detect it?

Attendee In this particular instance, there were a couple of indicators. First, due to questionable design the check valve chattering could be heard in line.

Also, leakage was detected, during inservice conditions and this was in a critical check valve, a PWR pressure isolation valve. When I specified acceptance criteria, that is a one gpm seat leakage criteria. And the full flow was design accident full flow requirements.

Panelist You said you heard the valve chattering, right? Back-tapping or hitting the body, the disc?

Attendee I think it was back–tapping.

Panelist Any time you hear that, you may have a problem. So you found the problem.

Question 17 (Check Valve Disassembly)

If check valve disassembly inspection is considered maintenance, not testing, how is the requirement for retesting after maintenance satisfied?

Response 17

Panelist Basically, if you are taking the valve apart because you can not do full flow testing, for post–maintenance testing, verify that the valve is capable of passing that flow, i.e., a partial flow, after the maintenance activity has been completed.

Attendee What about check valves that pass steam? How big a problem do we have with steam? Some big problems have been encountered in the industry. Can we talk about inspection results and non–intrusive methods?

Panelist I know of a couple of failure incidents. Incidentally, what I covered was strictly water hydraulics. However, in steam I know the big main steam check valves lost a disc a couple of times, it fell off, back–tapping, same problem, minimum flow velocity or high turbulence.

This does happen, yes. The same EPRI guidelines apply to steam. And you can also get steam hammer from the system. One utility actually lost a main steam disc. And in one case, they had a real refined design, and the valve manufacturer took a patent out on the design, which was a spring pack behind the disc to absorb the shock as the valve opens up. A few years later the disc fell off on a main steam check valve.

People are really trying. They worked very hard at it. They ran computer codes and did everything else. But people just do not know. It is really complicated. So we just have to learn how to live with that.

Panelist So are you saying, that from a design applications point of view this is an extremely tough problem to come to grips with?

Panelist That is the problem. In one case I know, in a duocheck, there are pin and bushings with maybe 20 thousandths clearance. It will shut off once in a while. If you analyze the valves, were they not designed to take shock waves like this. But people are trying to do the best.

And it is that subtle. We are talking about 20 thousandths clearance, a duocheck for example.

Panelist What about steam cutting?

Panelist When the disc is off its seat, very high velocities come through. In one case, if soft seats are not dovetailed, they will be cut out, sucked out from high velocities as the disc comes open. With very high (or jet) velocity, the forces change. The higher velocity, the lower force you have.

And I have seen where the soft seat is pulled out partially, because it is not dovetailed. It is rectangular. That is how they sealed it. It is a manufacturing process. Some of these problems are manufacturing process. But the bottom line is that people are really trying to design.

Panelist Yes. I have seen problems with steam valves, and it is a situation of the valve not being sized properly.

The same equation used for water could generally be used for steam. It is a function of specific volume. The problem is twofold: one, if the valve just happens to drop into the right flow velocity regime at normal flow, the problem comes when you are starting up and shutting down, when you are in a transient situation, because you start burping the valve. But the other part of it is a valve on the exhaust from a turbine, a safety–related valve, and a situation where the valve was sized for LOCA flow and tested at 15 percent flow monthly. Originally, a swing check valve was not suited for that particular service, and the valve destroyed itself. And it was not a function that the valve was a bad design. The valve was a very good design. It was just being used for something it wasn't intended to be used for.

Panelist That is a very good point.

Rameyer produced an excellent curve on the stability of check valves in terms of the flow ranges. He and other manufacturers has said that a check valve can be oscillated midstream.

Actually, it is supposed to float from the pin, such as the Dresden valves. Our people at Dresden took out the check valves last outage and found problems with them. However, there was no pin damage, and we know there is oscillation. We can hear it tapping. That is an example.

Many times the designers, if they are creative, know that the pinhole will float. Now, if the pinhole is 50 percent open floating slowly, it is not going to affect the valve. If it is right off the seat, you will start slamming. Then the back-tap is solid, and as velocity increases, another instability curve occurs. I should have copied that curve from William Rameyer's paper. So if you get his curve, that is the best we have. We should look at the ranges he stated. If we are too low we may have a problem, if we are in a middle range we are okay, if we are very high, it may be unstable for testing purposes. When you do your flow tests, listen to the valve, put acoustics on it, see if you hear it tapping. That is a very good point.

OPEN SESSION PAPERS

DIGITAL COMPUTER ANALYSIS OF INSERVICE TESTING DATA IN ACCORDANCE WITH ASME CODE

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ABSTRACT

Inservice Testing (IST) requirements for nuclear–powered generating plants demand accuracy and attention to detail in order to meet regulatory standards and to avoid unnecessary plant shutdowns. The use of computers and microprocessor–based data collectors can be extremely helpful in eliminating human error. The primary functions of the hardware are to effectively collect, transfer, analyze, trend and archive the thousands of bytes of data generated in performing these tests. This paper will discuss electronically managing IST data and illustrate how human error can be reduced by systematic programming of the entire process using technology which is currently available.

SUMMARY

Sources of repeated human error in the Inservice Testing process can be reduced significantly through the use of electronic data processing systems and programs which are now available on the market. As these systems become more sophisticated and plant installations are modified to accommodate them, many other advantages, such as reduced stay times in hazardous areas can be realized. Today only a small segment of the utility industry is benefiting from these new developments.

ACKNOWLEDGMENTS

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NOMENCLATURE

EDP — Electronic Data Processing—Management of information via automated means, usually by microprocessors and/or computers.

IST — Inservice Testing—In this case for pumps and valves in accordance with ASME Code Section XI Pump and Valve Testing Requirements (all revisions) including those recently developed under OM/6 and OM/10 committee sponsorship.

SOTA — State Of The Art—A generic term which refers to what is possible with today's technology in management of (in this case) Inservice Testing.

INTRODUCTION

The purpose of this paper is to define "State Of The Art" methodology in obtaining, analyzing, trending, storing, and retrieving data involved in the IST requirements.

In 1987, a comprehensive hardware and software system was custom designed by a Florida based company, Precision Mechanical Analysis, Inc., to assist a nearby utility in meeting its IST requirements. This "State Of The Art" system, designed for compliance with ASME Section XI Code requirements for nuclear plant pump and valve testing, has been implemented by the utility.

CURRENT IST METHODOLOGY

Presently, the Inservice Testing procedures at most nuclear facilities require manual entry (on paper) by the technician onto a log or data sheet. After the data are on paper, they will be reviewed by the appropriate personnel and then either filed or manually entered into a computer system.

The IST process is normally carried out as follows:

- An IST procedure is prepared, approved, and issued by the utility.
- The data are collected by technicians who interpret the procedures, insure plant conditions, and take the readings (i.e., which piece of equipment, what type of reading, which location on the equipment, etc.).

- The technicians then interpret the readings. (i.e., which scale on the instrument to use, what type of measuring unit to use, etc.).

- The readings are transferred to paper, written out manually.

- The IST engineers then review the completed procedures and associated data.

- If the plant has computer data storage and analysis capabilities, the data are once again transferred manually, this time into a computer data format.

Opportunities abound for human error in a manual system such as previously described. The critical information will be interpreted twice, then manually transferred, then transferred again in many cases.

Many times those steps in the process which require interpretation, translation and review are addressed through training programs and the utilization of a second person to confirm that steps are performed correctly (i.e., dual verification). Training and dual confirmation do not address the fundamental problems perpetuated by performing this process manually.

THE STATE OF THE ART METHODOLOGY

A properly implemented computerized IST system will offer the following benefits:

- The digital collection unit will actually guide the technician to the proper equipment, to the proper location on the equipment and will instruct him/her as to position of the reading, if pertinent.

- Sign–offs for completion of specific procedure steps can be verified and stored for future recall. Training and qualifications of the technician for each procedure can be verified.

- Though the technician taking the readings will know immediately if the reading is out of tolerance, field interpretation should be kept at a minimum.

- The system should have the ability to digitally collect the different types of data monitored under all revisions of ASME Code Section XI including the latest OM/6 and OM/10 requirements (i.e., overall vibration readings, flow and pressure).

- In addition to storing data necessary to complete procedures, the SOTA system should also allow the user to store more complete machine data (i.e., vibration spectrums, notes, etc.). In fact, the system should be programed to collect additional information automatically. For instance, diagnostic data could be taken only on readings that are out of tolerance, completely independent of the technician. In other words, if a particular vibration reading is found to be "out of alert" the digital collection unit would automatically take a vibration spectrum.

- Data should be transmitted via automatic, electronic means and not written out or transferred manually at any point in the system, once entered. In this system human error would be minimized.

- The SOTA system should have the ability to review tens of thousands of bytes of data and call only exception data to the attention of the IST engineer.

- In addition to producing completed procedure/data sheet printouts and exception reports, the system should include full data manipulation and comparison capabilities and have the ability to produce a variety of reports.

- The data storage methods should be compatible with the main stream of technology and should easily interface with other software packages. This would allow the use of future diagnostic packages that might be available later.

OPERATION STEPS OF THE IST PROCESS UTILIZING EDP

The "State Of The Art" approach should follow a systematic methodology described as follows:

- A computer generated version of the procedure in ASCII (American Standard Code for Information Interchange) format is put into the computer containing the IST software.

- The IST engineer then utilizes the software to specify the characteristics of the data points to be collected according to the procedure. These characteristics are location of the point, point I.D., description, unit of measure, tolerances, etc.

- These procedures with their defined points are then electronically down-loaded to a digital collection unit.

- The digital collection unit then directs the technician to the equipment to be measured, on a machine-by-machine and point-by-point basis.

- The technician operating the data collector places the probe as directed by the unit which then obtains the data. Any equipment which does not output electronic data must have its individual gauges and meters read by the technician and entered via the keyboard on the collector.

- Having taken the measurement, there is no interpretation of the readings by the technician because the digital collection unit knows what type reading is being taken, what unit of measure is in use and that data are electronically entered. However, if the reading is in the "alert" or "action" range, the unit notifies the technician by displaying data and tolerances in both digital and graphic format.

- The collected data are then electronically up-loaded to the IST computer.

- The computer reduces the data, provides an exception report (listing out of tolerance and missed points), prints out completed procedures (if desired), and stores the data histories for future reference or comparisons.

- For purposes of review or reporting, the SOTA system will print out graphs and charts illustrating and comparing a history of the readings with static descriptive data, "alert" levels, "action" levels, baseline readings, the statistical mean, standard deviation and other criteria.

The major advantages of this systematic approach are that human factors are taken into account and the potential for error reduced.

THREE MAJOR COMPONENTS OF THE SOTA IST SYSTEM

This methodology is broken down into three major categories:

- Automated Procedure Management—Setting up point models. Each point represents one reading. Several points make up one procedure. With automated procedure management the procedures are easily created, modified and replicated in the computer.

- Digital Data Collection—Vibration, pressure, flow rates, and any other type of data that can be measured are collected and recorded. Ideally, the reading is taken and recorded electronically, with no keyboard operation. Meters, gauges, and vibration instruments with electrical signal transmission capabilities, requiring only the insertion or other connection of a probe, are the most error-free installations.

- Computerized Tracking, Trending, and Reporting—The computer reduces the data, builds histories, compares the histories and reports any readings outside preset tolerances.

A closer look at each of these three parts is contained in the following paragraphs, along with sample computer screens from a SOTA program which is presently in use in a nuclear power plant.

Automated Procedure Management

A SOTA system supplies the operator with a complete menu driven, automated sub program to set up points for data collection and trending in the computer. We will call this sub program "The Model Manager" (an example is shown in Figure 1).

Point models are defined for each point. A point model is essentially a formula for each point that is collected and interpreted. The model tells the system what type of measurement is being considered, (i.e., vibration, pressure, etc.), what unit of measure is used, and what the tolerances are ("alert" and "action" levels) for the point. Some types of readings, like a pressure measurement, will have a high and low "alert" and "action"; other readings will have just one "alert" and one "action" tolerance.

The Model Manager will link any rules (e.g., Technical Specifications) to the points that are monitored.

Digital Data Collection

The data collection unit and associated probes are selected by an authorized technician. He/she checks the laboratory calibration stickers attached to each component to assure completion within the last calibration cycle. Steps are executed to down–load the procedure(s) scheduled to be conducted from the IST computer to the data collector.

Modern data collection units also permit a simple, informal calibration check of certain functions, such as the vibration data probe and associated circuitry. The vibration probe is attached to a "shaker" which has a current laboratory calibration. A reading is then taken on the "shaker" in the same manner as during data collection on a rotating machine in the plant. In this way the "shaker" signal is automatically transferred to the unit with no interpretation by the technician. After the signal is transferred, the unit then either approves or disapproves its own calibration. This type of check helps assure that the unit is in calibration for each and every procedure. In future programs the temperature data collection probe and circuit could also be checked using an appropriate calibration device.

When taking the vibration readings for the procedure, the technician is instructed by the data collector as to which location and in which position to place the accelerometer. When the accelerometer is in place, he simply presses a button and the vibration signal is automatically transferred to the unit. This is done with no interpretation by the technician. He can read the level recorded and is notified if it exceeds the "alert" or "action" tolerances. The same is true of any other type of reading (pressure, flow, etc.) taken with a direct connecting probe to sensors transmitting electronic signals.

After each data collection step of the procedure is completed, an internal comparison is performed to immediately tell the technician if that particular portion of the procedure is satisfactory. If the reading is in any way inconsistent with what the data collection unit is expecting, the technician is notified and then has the option to either continue or take corrective action.

This system will tell the technician instantly the status of the machine while he is still at the machine. It completely eliminates the necessity for any manual comparison of the data.

Once all of the data are entered into the unit, the technician is notified that the procedure has been completed and the technician(s) performing the procedure will be required to enter a personal code to identify who performed the procedure.

When the technician returns to the control room, the recording device will be connected to the interface of the IST computer. The data are then transferred into the computer part of the system for tracking, trending, reporting, storage and rapid recall/retrieval.

Computerized Tracking, Trending, and Reporting

After the data are entered into the computer system the appropriate personnel are notified by the system that the procedure has been satisfied and if there are "alert" or "action" concerns.

"Alert" or "action" tolerances can be changed by the IST Supervisor if the reference values are changed by rebaselining, or the method in which the values are calculated is changed through the model manager. Security levels should be included to prevent unauthorized access.

Once the data have been transferred into the computer system they will be entered into a trending program that will allow data plotting for a selected time interval.

The system can virtually eliminate all paperwork presently involved with IST procedures. However, the data are always available and can be printed at any time. The entire completed procedure or the trend (history) for any machine can be produced with just a few keystrokes.

The main data base for the IST system would be maintained in the control room to enable the Shift Supervisor to have immediate access to the procedure histories for any safety related component.

Automatic digital communication packages should be available which allow the data to be accessed from any normal work station within the plant.

The procedure schedules should also be computerized to insure that the procedures are performed on schedule.

Options Made Possible by SOTA Methodology

Future programs would allow the use of bar codes; this would further guarantee proper readings and eliminate the need for dual verification. In this implementation, the technician would have to scan a bar code at the proper location before being allowed to take the reading. This gives positive identification of the point where the reading is taken.

Sign–offs can also be performed via bar codes, thus providing positive identification of the plant personnel performing the procedures.

If desired, manual input can be locked out entirely.

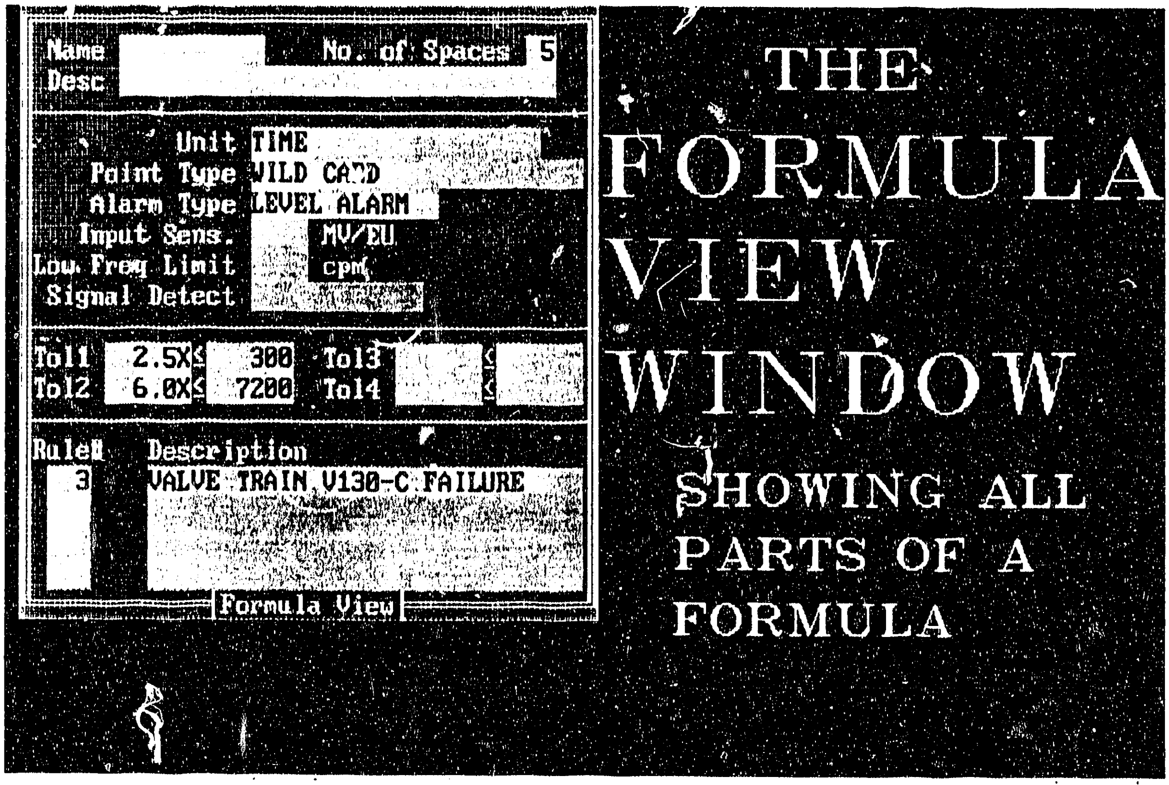

Figure 1. This is an example of the model manager which is used to create, view, edit and replicate the formula describing a specific point.

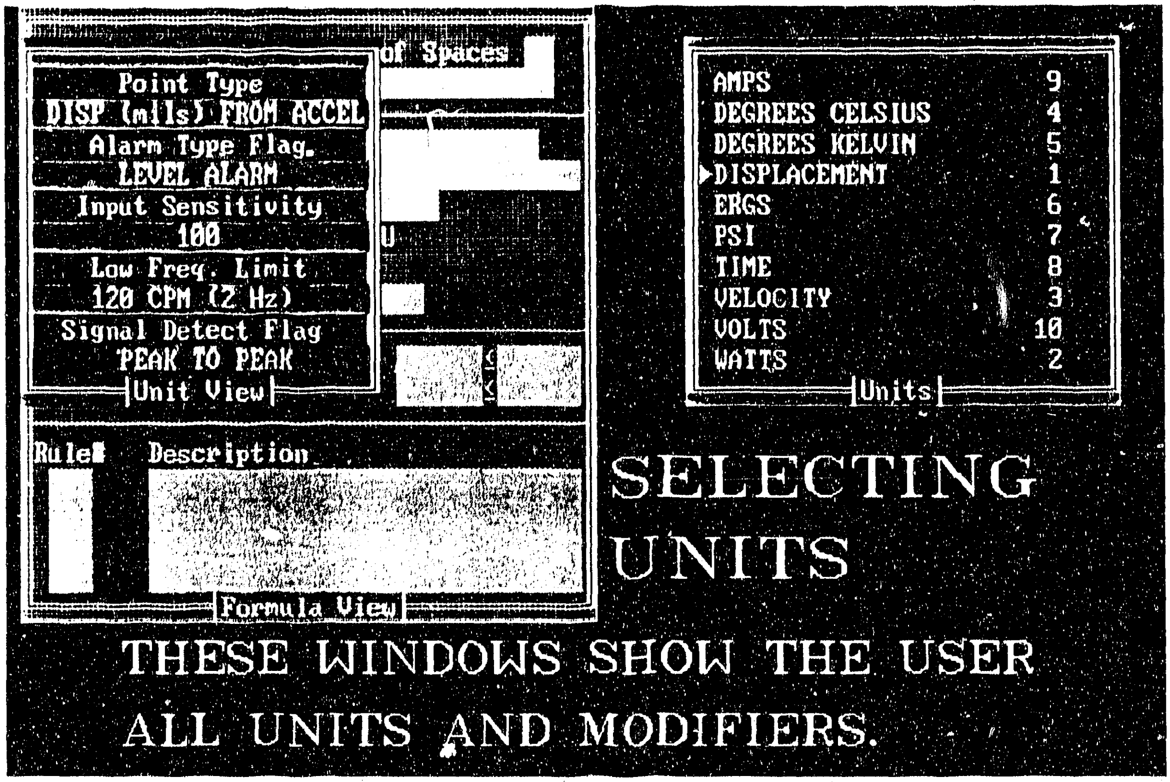

Figure 2. Illustration showing how the model manager can select units for a point model by using pop–windows.

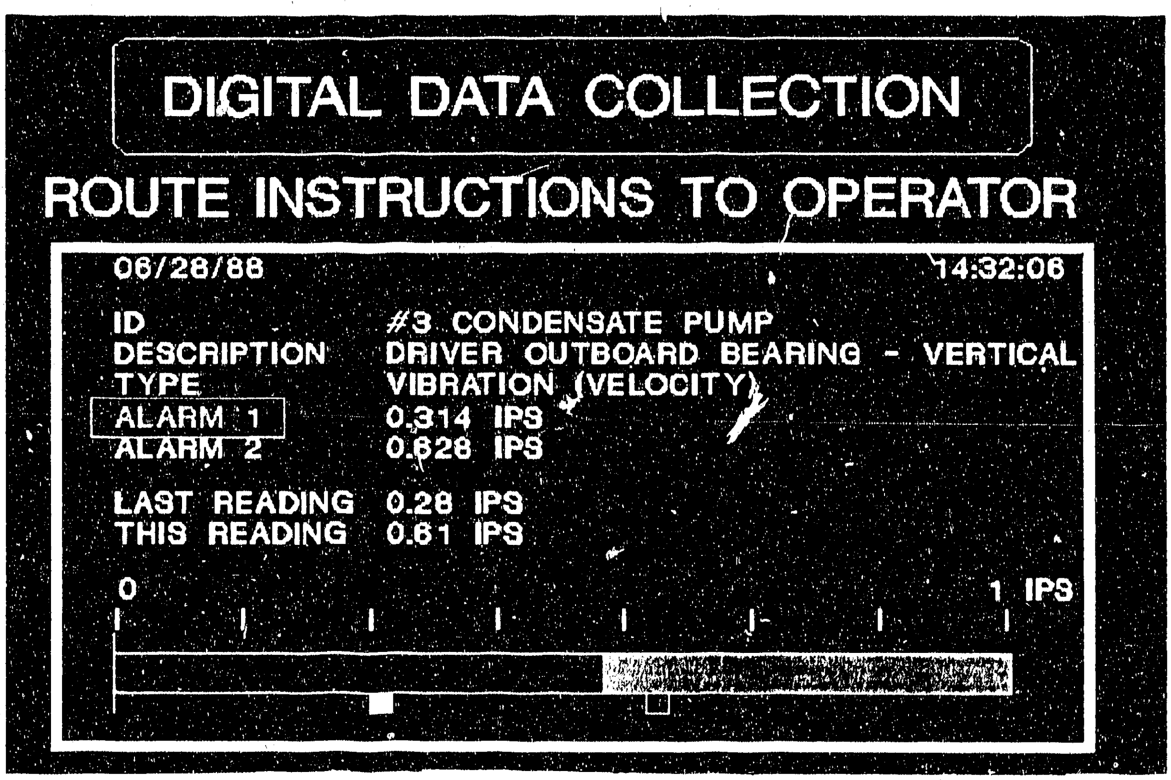

Figure 3. Example of the readout on data collector unit.

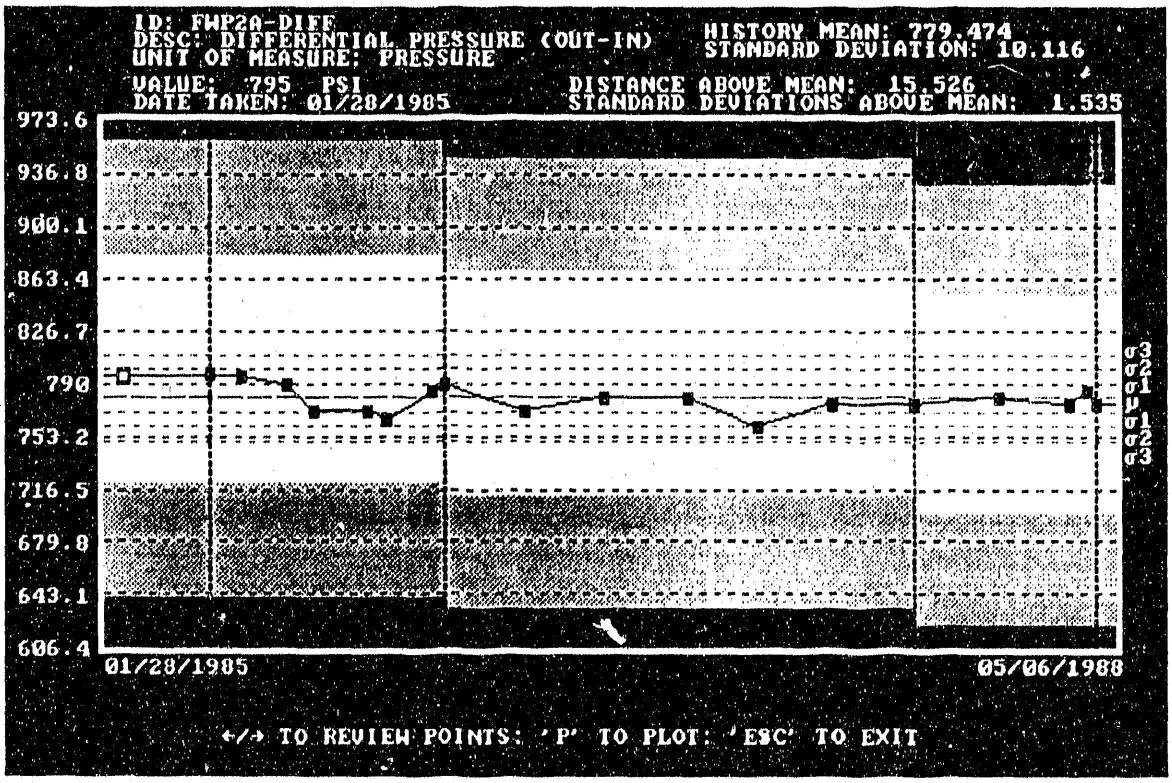

Figure 4. Example of a computer generated graph, showing data from 01/28/1985 to 05/06/1988 and plotted against means ("μ"), standard deviations ("σ") and tolerances (represented by shaded regions). New baselines are shown at each vertical dotted line.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

When you implemented systems, such as your example from Crystal River, how long did it take to work out the bugs, load the data base, and make this a usable approach?

Response 1

It takes anywhere from weeks to months, depending on who is assigned to do it. In this case, the person assigned to enter the data was not a technical person, and we had to teach her how to do a lot of the entry. But in the end, she was very, very knowledgeable of the information and able to retrieve, handle, and manage it very effectively. So it depends upon who is assigned to actually get the information and put it in, how much training that person needed and what they are trying to do with inservice testing.

THE IMPACT OF ASME O&M STANDARD ON OPERATING NUCLEAR POWER PLANTS

C. WESLEY ROWLEY, P.E.[a]

ABSTRACT

This paper discusses the origin and mission of the O&M Committee, the Parts of the OM Standard that it has issued to date, the Parts it plans to issue within the next few years, and the status of regulatory enforcement of its Parts. These Parts include pump and valve performance testing, as well as eighteen other topics.

Origin of the OM Standard—In the late 1960s the ASME decided to develop a Power Test Code (PTC) for the testing of nuclear pumps and valves. Before these PTCs were issued a decision was made in the early 1970s to transfer the requirements for pump and valve testing into Section XI of the Boiler & Pressure Vessel Code. As a result parts IWP and IWV were first published in 1974. In 1975 the ASME made the decision to create a new main committee to develop standards for operating nuclear power plants. This committee was named the "Operations and Maintenance Committee" and its mission was:

> To identify, develop, maintain, and review codes and standards that are considered necessary for the safe and efficient operation and maintenance of nuclear power plants, particularly as they relate to structural and functional adequacy.

The intent for this committee was to take over the testing of active components from BPV Code Section XI.

Issued Parts to the OM Standard—During the first decade (1975–85) of its existence, the O&M Committee issued standards that monitored or tested specific components and systems.

- The first O&M Part issued (in 1981) was for the testing of pressure relief valves (OM–I). This Part was intended to test those relief valves described by ASME Section III Class 1,2,3 and which have a specific safety function (in shutting down a reactor or in mitigating the consequences of an accident).
- The second O&M Part issued (in 1981) was for the inservice monitoring program to detect significant loss of axial preload at the core barrel's upper support flange in PWRs (OM–5).
- The third O&M Part issued (in 1982) was for the examination and testing of snubbers (OM–4). Notwithstanding the requirements for preservice examination, inservice examination, and repairs & replacements as described in the BPV Code Section XI IWF–5000, this Part defines the requirements for periodic examination and testing of snubbers through the life of the plant.
- The fourth O&M Part issued (in 1982) was for the system performance and functional testing of closed cooling water systems (OM–2) to check the degraduation of the capability to transfer heat to a heat sink.
- The fifth O&M Part issued (in 1982) provided the general requirements for the assessment of piping system vibration for BPV Code Section III piping and applicable ANSI classified systems (OM–3).

Note that most of these Parts now have either revision 1 or 2 issued thereby refining the requirements or guidance to the nuclear industry.

So far during the second decade of its existence (1985–present), the O&M Committee has issues Part for testing specific components.

- OM–6 was issued in 1988 to provide requirements for the testing of pumps, which are required to perform a specific safety function (in shutting down a reactor to cold shutdown condition, maintaining the cold shutdown condition, or in mitigating the consequences of an . accident). This standard was specifically developed to replace IWP in BPV Code Section XI.
- OM–7 was issued in 1988 to provide requirements for the development of procedures and

a. The author, C.W. Rowley, is an independent consultant to the nuclear industry from Cashiers, NC: he is also a member of the O&M Executive Committee and Chairman of its Task Group on Standards Utilization.

the assessment of thermal expansion response to Class 1,2,3 piping systems and applicable ANSI classified systems.

- OM–10 was issued in 1988 to provide requirements for the testing of valves, which are required to perform a specific safety function (in shutting down a reactor to the cold shutdown condition, maintaining the cold shutdown condition, or in mitigating the consequences of an accident). This Part was specifically developed to replace IWV in BPV Code Section XI.

- OM–16 was issued in 1989 to provide guidance relating to the testing and maintenance of diesel drives which are required to perform a specific safety function (in shutting down a reactor or in mitigating the consequences of an accident).

Parts Under Development to the OM Standard—Currently there are another eleven O&M Parts under development; a brief description of each follows:

- OM–8 Performance testing of MOV assemblies used in nuclear power plants

- OM–9 Periodic inspection and performance testing of radioactive material handling cranes in nuclear power plants

- OM–II General procedures for the assessment of heat exchanger vibration in nuclear power plants

- OM–12 Loose parts monitoring and diagnostics for light water reactors

- OM–13 Performance testing and periodic monitoring of power operated relief valve assemblies in nuclear power plants

- OM–14 Guidelines for an in situ vibration monitoring program for rotating equipment important to nuclear power plants

- OM–15 Performance and functional testing criteria for ECCS in PWR nuclear power plants

- OM–17 Periodic inspection and performance testing of instrument air systems in nuclear power plants

- OM–18 Performance testing of electro–hydraulically operated valve assemblies in nuclear power plants (excludes PORVs covered by OM–13)

- OM–20 Performance and functional testing criteria for ECCS in BWR nuclear power plants

Historical Thrust of the OM Standard—To discern the emphasis of the O&M Committee in its development of Parts, it is interesting to categorize the various OM Parts as follows:

Specific Equipment Oriented		Mechanical System Oriented	Programmatic Oriented
OM–1	OM–10	OM–2	OM–3
OM–4	OM–11	OM–15	OM–7
OM–5	OM–13	OM–17	OM–12
OM–6	OM–16	OM–20	OM–14
OM–8	OM–18		
OM–9	OM–19		

So it is obvious that some 60% of the O&M Committee effort to date has been directed toward specific pieces of equipment or components. Of the twelve specific equipment oriented Parts, note that at least five of them (OM–6, –8, –16, –18, –19) have an electrical interface.

Potentially New Parts to the OM Standard—A number of nuclear plant problems can best be addressed via an additional Part to the OM Standard. Some topics under current consideration are:

- Service water systems

- Refueling systems

- Checkvalves

- Piping operability

- Heat exchangers

- Auxiliary feedwater systems.

In the INPO Summary of Maintenance Activities dated July 21, 1988, the current "top equipment problems list" is as follows:

- Feedwater control
- Motor operated valves OM–8
- Diesel–generators OM–16
- Standby turbine driven pumps
- Main turbines
- Heat exchanger tubes
- Relief valves OM–1
- Piping
- Instrument air OM–17
- Checkvalves
- Reactor coolant pumps
- a–c converters.

Of these twelve industry problem issues, the O&M Committee already has four of them at least partially covered by issued (OM–I and OM–16) and by developing (OM–8 and OM–17) Parts.

Regulatory Utilization of the OM Standard—The ASME BPV Code Section XI specifically references Parts 1, 4, 6, and 10 for requirements. Since Section XI is endorsed by 10 CFR 50.55a, these four parts are required to be used by nuclear power plants, depending on the following factors:

- Exemptions approved by NRC
- IST interval for pumps and valves currently in effect for a particular plant
- Specific plant tech spec snubber requirements versus adoption of OM–4
- Specific plant FSAR commitments.

Recently the BNC&S (Board of Nuclear Codes and Standards) has approved a motion to transfer the IST program from Section XI to the O&M Committee. This will require the NRC direct endorsement of the OM Standard vice Section XI for the IST (inservice testing) Program. An O&M/Section XI Transition Joint Task Force is already working on the implementation of this BNC&S motion, hopefully to be completed by the end of 1990.

Difference Between Codes, Standards, and Guidelines—The ACME, as well as the other technical societies, have not defined the differences between these three terms very well. The O&M Committee is in the process of developing definitions and meanings to these three terms, perhaps as follows:

- Code—a document that is developed to be implemented by a statutory or regulatory requirement; the requirements of the document must be implemented in its entirety.
- Standard—a document that is developed to be implemented on a voluntarily basis by the plant owner/operator; the requirements of the document must be implemented in its entirety, if owner/operator is to be credited for compliance.
- Guideline—a document that is developed to share good technical practice with the industry; the recommendations of the document may be utilized in total or partiality, as desired by the plant owner/operator. Compliance with a guideline is not usually a source of credit to the owner/operator.

Implementation of the OM Standard—To date the OM Standard has been or is planned to be implemented mainly as a guideline by many nuclear power plants.

Apparently few nuclear power plants have implemented any Part of the OM Standard as a "standard". This topic is being looked at closely by the O&M Committee via their Task Force on Standards Utilization. More information will become available over the next year or so.

A few plants, due to the "roll–over" of their IST interval (ten years) have been required to implement OM–I and OM–4 via Section XI, which is mandated by the NRC and by some states. As more "roll–overs" occur over the next few years, OM–6 and OM–IO will also become mandatory (OM–6 and OM–IO were first endorsed by the Section XI winter 1988 Addenda).

Will the fact that OM–I, OM–8, OM–16, and OM–17 address issues listed in the INPO "top equipment problems list" have a catalyst effect on the implementation of these documents as a "standard" vice a "guideline"? Some observations follow:

- Many people in the nuclear industry are not yet aware of the OM Standard

- The OM Standard does not yet have the visibility of the BPV Code Section 111 and Section XI.

This will probably change over the next few years.

O&M Committee Management Team—The chairman of the O&M Committee has always been a nuclear utility officer:

- 1st chairman—Wendell Johnson, VP at Yankee Atomic Electric

- 2nd chairman—Ed Williamson, Sr VP at Southern Company Svcs

- 3rd chairman—Jerrold Dewease, Sr VP at Louisiana Power & Light

This has helped ensure that the products of the O&M Committee are documents useful to the 54 nuclear utilities with nuclear power plant operating responsibility.

In addition the vice-chairman of the O&M Committee has always been a nuclear utility executive.

- 1st vice-chrm—John Kufel, Millstone Plant Manager

- 2nd vice-chrm—Jim Stacey, Manager at Yankee Atomic Electric

- 3rd vice-chrm—Dick McGaughy, VP at Iowa Electric Light & Pwr

The ASME consensus process requires that no one interest group have more than one third of the total main committee membership. The O&M Committee has conformed to that rule, but keeps its owner/operator utility membership right at the one third limit. In addition most O&M Committee members are utility and operations oriented by virtue of their experience and training.

Since NUMARC has been formed, the O&M Committee liaises with the NUMARC staff to ensure complementary efforts. In addition O&M Main Committee members from INPO, EPRI, ANI, and NRC have helped to provide a comprehensive nuclear industry perspective.

SUMMARY

In summary the O&M Committee is very determined to carry out its charter (mission) with effective nuclear industry leadership, so that the OM Standard will become a premier industry technical reference document for codes, standards, and guidelines.

The O&M Committee and its sub-tier membership represents the greatest resource of nuclear operational expertise of any consensus code and standard organization in the industry. As such it is dedicated to producing useful, practical consensus standards and guidelines that serve the needs of the user (owner/operator) as well as the regulatory agencies.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

We have already been granted interim approval to use a 1983 edition of Section XI for IWV and IWP. Since the OM-6 and 10 standards have not yet been adopted into the 10 CFR 50, are we going to need to resubmit another program and wait for another SER? We have not received the one that we are in right now, so I do not understand how we can use it.

Basically all the people on Table 1 for the generic letter have interim approval, but no SER.

Response 1

Basically, since the OM-6 and 10 was just adopted in the Code 1988 addenda, and 10 CFR 50 has not yet adopted that, there is no direct legal mechanism to satisfy. On the other hand, if you were to negotiate with the NRC via your licensing organization, you might be able to take credit for that.

Question 2

Yesterday, the transition committee between Section XI and O&M was briefly mentioned, and you mentioned again this morning the effort to incorporate O&M standards directly into 10 CFR 50. Could you elaborate on the transition team and what their direction is?

Response 2

The Board of Nuclear Codes and Standards passed a resolution in February that they would like to transfer the inservice testing program from Section XI over to O&M at some appropriate date when we are ready for it. The first thing we did was to create a transition team made up of three members of O&M, three members of Section XI, two members from the NRC representing the third party inspection aspect. These nine members are chaired by Ed Williamson, who stood up yesterday and was introduced. Their approach has been to determine all the problems in making this transition. They have completed their report, and that report will be submitted to the O&M Committee and Section XI and the Board of Nuclear Codes and Standards in the next several months, and then it is a process of implementing that approach.

Basically it means deleting IWV, IWP, and Section 5,000 out of Section XI with a stroke of the pen. But it also means putting into O&M the kind of code support words that make it easy to use these standards in a code-type sense. Then the NRC has to process their word changes to 10 CFR 50.55a. Our goal right now is to have both Section XI and an O&M addenda issued, which would transfer the IST program to O&M some time in 1990. We are shooting for a new edition of the O&M code that will come out in the summer of 1990.

That is our schedule goal. And then however long it takes the NRC to process their 10 CFR change after that may be anywhere from six to eighteen months.

Question 3

You talked about feedwater. Are you developing a standard for feedwater stability? You mentioned it on the graphs. Could you please talk about it?

Response 3

The feedwater instability was listed on the INPO top dozen problems. That is not currently one of the problem areas that O&M has addressed. But I will take your comment to the task force on standards planning and see if they think it is appropriate.

ADVANCED VALVE MOTOR OPERATOR DIAGNOSTIC SYSTEM M.C.S.A./VMODS

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ABSTRACT

This paper explains Wyle's new system for diagnostics, signature analysis and direct measurement of actual load on Motor Operated Valves (MOV) in the closed direction. The following topics are discussed:

- System description
- System capabilities
- Features.

The specific features of Wyle's system are:

- Measures directly the actual closing force
- Easy to install
- "Le.ıp frog" working feature minimizes test time
- Capability of up to 14–channel simultaneous input and acquisition
- Non–intrusive monitoring at a remote location (MCC)
- High sensitivity and selectivity to a variety of mechanical disorders affecting MOV operational readiness
- Provision of useful and trendable signature features.

INTRODUCTION

The Wyle Advanced Valve Motor Operator Diagnostic System (AVMODS) approach and equipment has undergone over a year of development to provide a relatively quick and non–intrusive capability to monitor, analyze and document the condition of motor operators on safety--related nuclear valves driven by an electric motor. The system measure valve stem thrust directly. Data is also provided on all limit switch and torque switch operations, as well as, motor voltage and current. Also, through detailed research programs, it has been determined that any mechanical perturbation occurring in the operation of a MOV (Motor Operated Valve) can be detected in the motor current supplied to the operator electric motor. This can be accomplished remotely from the Motor Control Center (MCC) with a clamp–on ammeter.

Wyle's AVMODS comprises two complementary segments:

1. VMODS (Valve Motor Operator Diagnostic System) is utilized to establish base line parameters and was developed entirely by Wyle Laboratories, Scientific Services and Systems Group.
2. M.C.S.A. (Motor Current Signature Analysis) has been developed by Oak Ridge National Laboratories (ORNL) under the auspices of the NRC. Wyle is the ORNL licensee for this technology.

VMODS Description

VMODS consists of several entities, mainly: (See VMODS Schematic,).

- Sensors: clamp on ammeters, load washers and position transducers.
- Interconnect box
- Cable sets
- Data acquisition module
- Computer/software/display.

Interconnect Box

Limit and Torque switch cables are routed through this junction box which is located near the valve and allows for 250 feet of umbilical cable to be run back to the Data Acquisition Module. The Motor current, motor voltage, stem travel, spring pack displacement and load washer cables also feed through this box.

Data Acquisition Module

The unit contains a Kistler charge amplifier and Simpson voltmeter and acts as a signal conditioner for all inputs to the computer. All ranges are selected through this module and input to it comes from the interconnect box via umbilical cable. End–to–end checks of all signals are performed prior to test.

Load Washers

Kistler series 9000 load washers are used to measure actual loading of the stem during valve closure. Location, mounting torque and range of the load washers have been precisely determined from extensive testing on different actuators (SMB–000 and up). Also tested were repeatability and precision. The results are contained in Wyle test reports. They show good repeatability and an accuracy of the stem load better than $\pm$10%. The load washers are supplied with calibrated sensitivity by the manufacturer and were tested by Wyle against a load cell traceable to the National Bureau of Standards during the research programs reported above.

Clamp–on Ammeters

There are 10 clamp–on ammeters supplied with each system, which make up two complete setups for use with two interconnect boxes to allow "leap frog" working. Four of the amprobes detect current through the torque switch, bypass and limit switches and the fifth, and more accurate, measures motor current (ac or dc).

Linear Transducers

A Celesco position transducer (string potentiometer) and waters displacement transducer measure stem travel and spring pack displacement, respectively.

Interconnect Cable Sets

There are two switch monitor cable sets for use with the two Interconnect Boxes, i.e., for instrumenting the valves to the interconnect boxes.

Umbilical Cable

Two hundred and fifty feet of umbilical cable is available for connection of the Data Acquisition Module to the Interconnect Boxes allowing for remote operation of the Data Acquisition Module/Computer.

Software

The software package of VMODS features built–in screen, text and array editors, multiple graphic windows, array handling function and plotting functions. This program allows 14–channel simultaneous input and acquisition. All required information is gathered in one complete cycle of a valve; close–open–close (or vice versa). The display can show one or any combination of eight of the fourteen channels on a computer screen.

System Capabilities

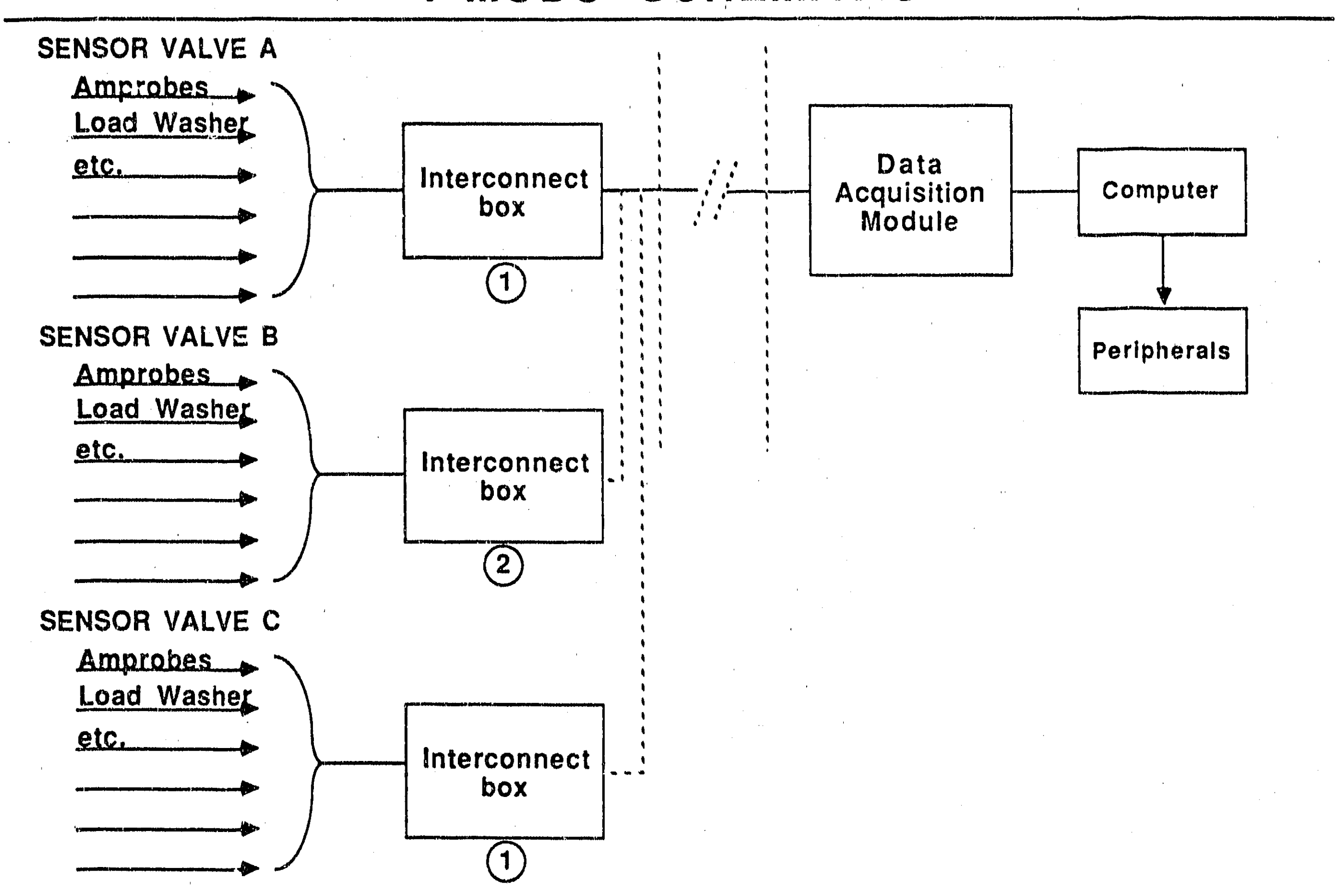
VMODS verifies torque switch, torque bypass and limit switches settings and stem load on valve motor operators by gathering valve operational data, either during static actuation or under full differential pressure conditions. It also provides information useful for predicting valve condition and for performing failure evaluation. The system measures eight operational parameters during valve opening and closing.

Parameters Measured

As the valve is operated from full close to open or vice versa, the following parameters are measured:

Measurement	Output
1. Valve stem travel	Stem travel (inches)
2. Valve stem load	Valve stem thrust (force pounds)
3. Motor current	Amps
4. Motor power	Motor power (kW)
5. Spring pack displacement	Spring pack displacement (inches)
6. Torque switch activations	Current change
7. Limit switch activations	Voltage change across contacts
8. Torque by–pass switch activations	Current change

V•MODS SCHEMATIC



P-10-88-48 (03)

Data Reduction

All measurements are analyzed and graphic output can be displayed, plotted or printed. Cursor control can enlarge any section of the graphic display and present elapsed time between any points on the screen. Therefore, power, stem travel, current switch activation and/or other parameters amplitudes can be evaluated on the same time basis using actual engineering units.

Data Output

After on-screen data manipulation and evaluation, any selected information can be output to the printer/plotter. A sample of VMODS hardcopy output is shown on the following page. This plot is indicative of the flexibility and comprehensiveness of the VMODS data system output. Figure 1:

Shows a full cycle of SMB-2 Limitorque actuator (close-open-close).

Channel 1: Shows stem travel in both directions (inches). Stem position at any point during the stroke can be found from this signature.

Channel 2: The load trace shows the load being transmitted to the valve by use of load washer(s). There is running load during the stroke, then the valve seats, the load increases and the torque switch shuts down the motor.

Channel 3: Shows current signature from close to open to close direction. The trace shows time and amplitude of: Close to Open—

- Inrush current
- Hammerblow time
- Unseating time
- Running current

Open to Close—

- Inrush current
- Running current
- Seating current

- Stroke time

Channel 4: Shows spring pack signature in both directions. Spring pack displacement (inches), hammerblow and unseating time can be obtained from this trace.

Channel 7: Shows open limit switch actuation. When the valve reaches the set point, this switch will trip and stop the valve (open stroke).

Channel 10: Shows close torque switch actuation. When the preset thrust has been reached, the close torque switch actuates and stops the valve in the closed direction.

Channel 11: Shows the open bypass switch actuation. This switch should cover the first hammerblow and unseating peak. This trace verified the valve unseating.

M.C.S.A.

Once the baseline data has been established, continued operability of the MOVs can be verified, by monitoring Motor Current Signatures from the Motor Control Center. This can be accomplished using Motor Current Signature Analysis (MCSA). MCSA possesses several innovative features which include on–site computerized data acquisition, remote monitoring capability and the use of such instrumentation as signal conditioner and clamp–on ammeters for quick setup. The system identifies, characterizes, and trends over time the instantaneous load variation of mechanical equipment in order to diagnose changes in the condition of the equipment, which if allowed to continue, may lead to failure. The motor current noise signature is detected, amplified and further processed as needed to examine its *time domain* and *frequency domain* (spectral) characteristics.

Parameters Measured

The motor current signature is obtained as the valve operates from fully closed to fully open or from fully open to fully closed. MOV motor current signatures provide diagnostic information on many levels:

- Mean value
- Gross variations
- Transients
- Frequency spectrum.

From the time domain signature, the following parameters are determined:

- Inrush current
- Running current
- Seating current
- Hammerblow time and amplitude
- Valve unseating, time and amplitude
- Limit switch trip
- Valve seating
- Torque switch trip
- Stroke time
- Torque bypass switch.

And from frequency domain, the following parameters are measured:

- Motor slip frequency
- Motor speed (rpm)
- Worm gear tooth meshing (WGTM) frequency
- Valve stem velocity
- Operator gear ratios
- Impact of packing tightness.

Samples of MCSA hard copy output are shown on the following pages. These plots are indicative of the flexibility and comprehensiveness of the MCSA data system output.

CONCLUSION

Wyle's Advance Valve Motor Operated Diagnostic System (AVMODS) is a unique system which can help the nuclear power industry to analyze and improve the performance monitoring of the Motor Operated Valve (MOV).

This diagnostic system can perform faster, more efficiently and more accurately than other methods previously available.

The advantages of AVMODS are:

- ALARA
- Reduce man hours
- Non–intrusive
- Real time analysis.

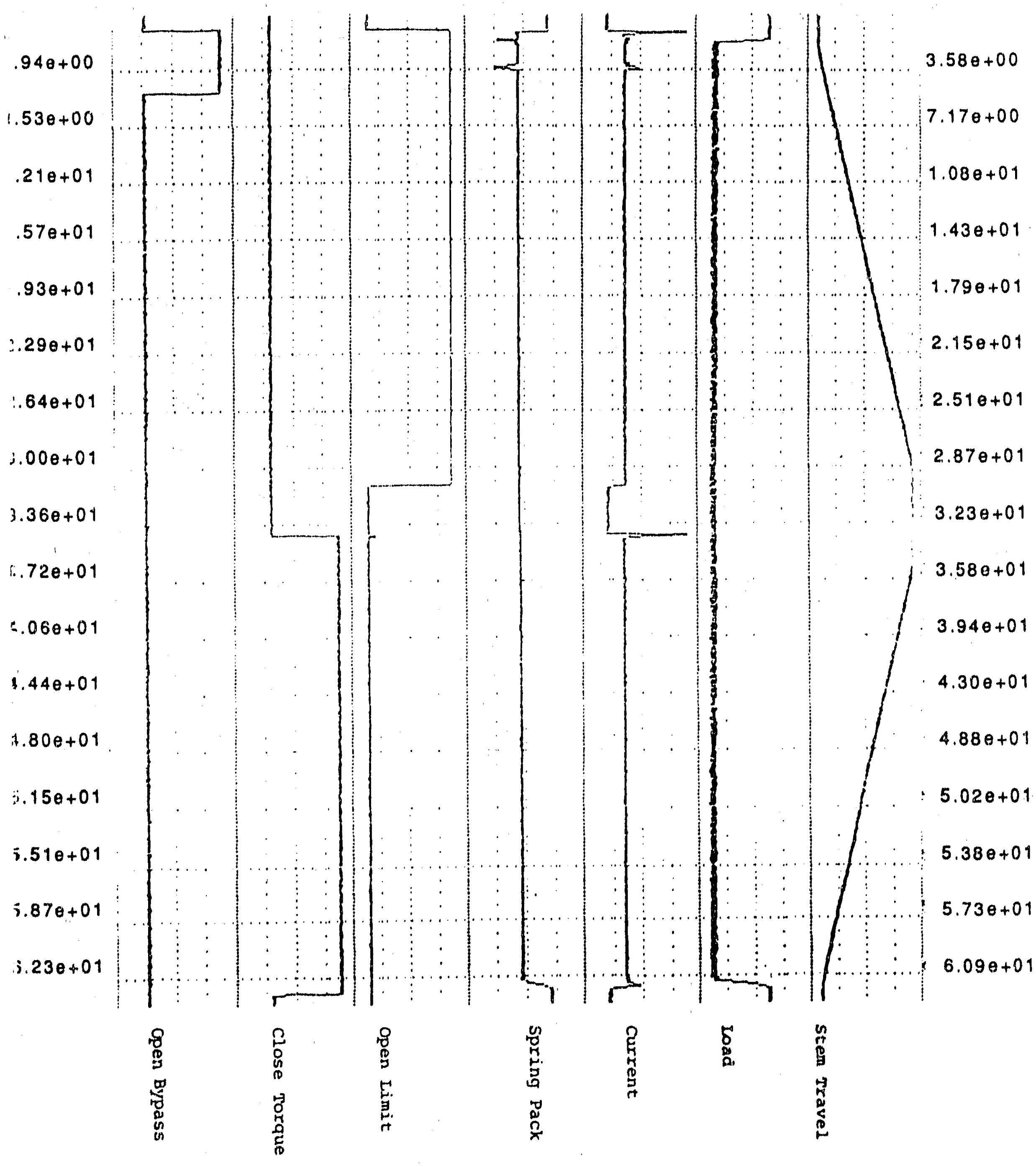


Figure 1. VMODS output—full stroke.

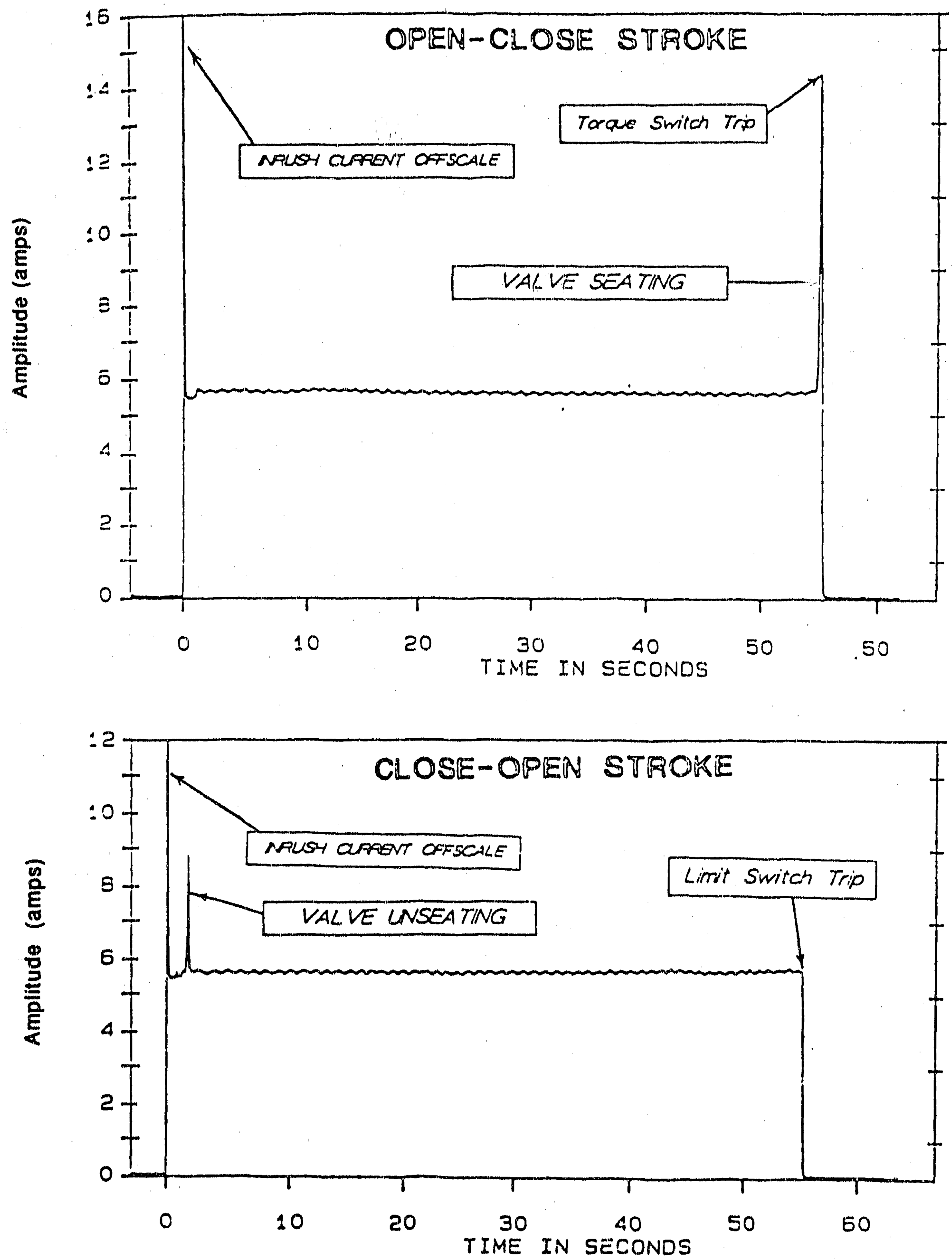


Figure 2. Motor current signature–full stroke.

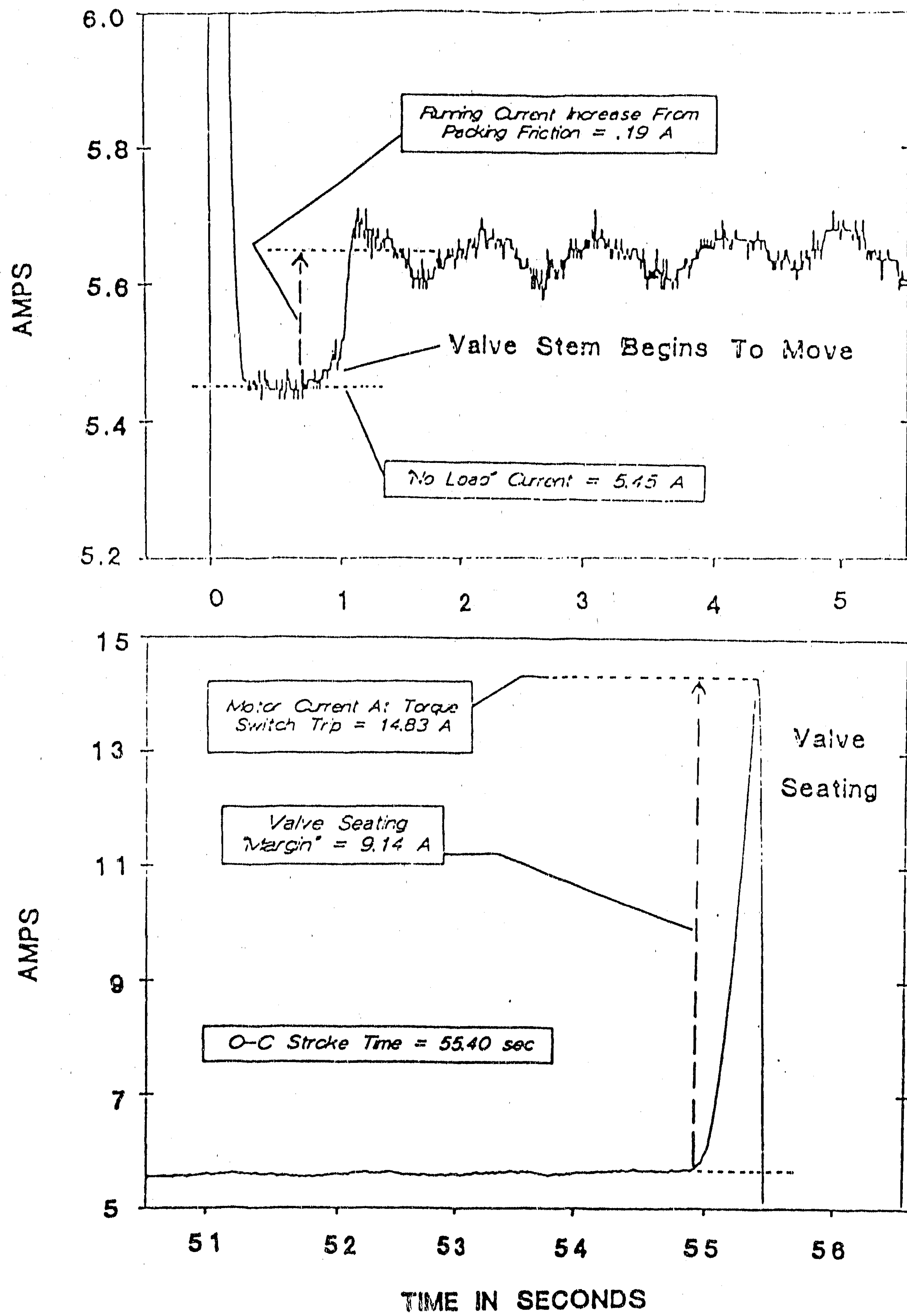


Figure 3. Motor current signature–transients (O–C stroke).

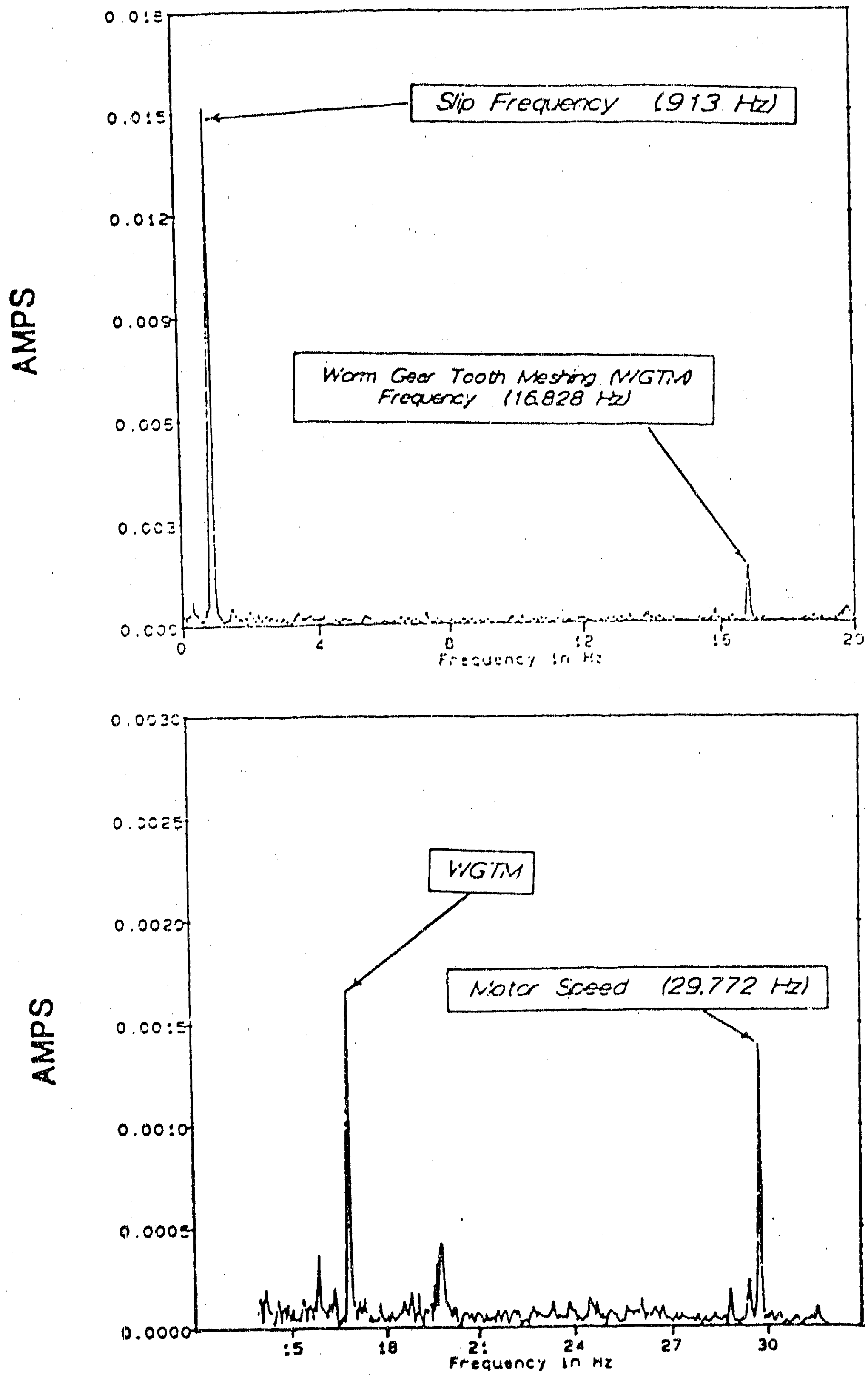


Figure 4. Motor current spectrum analysis (O–C stroke).

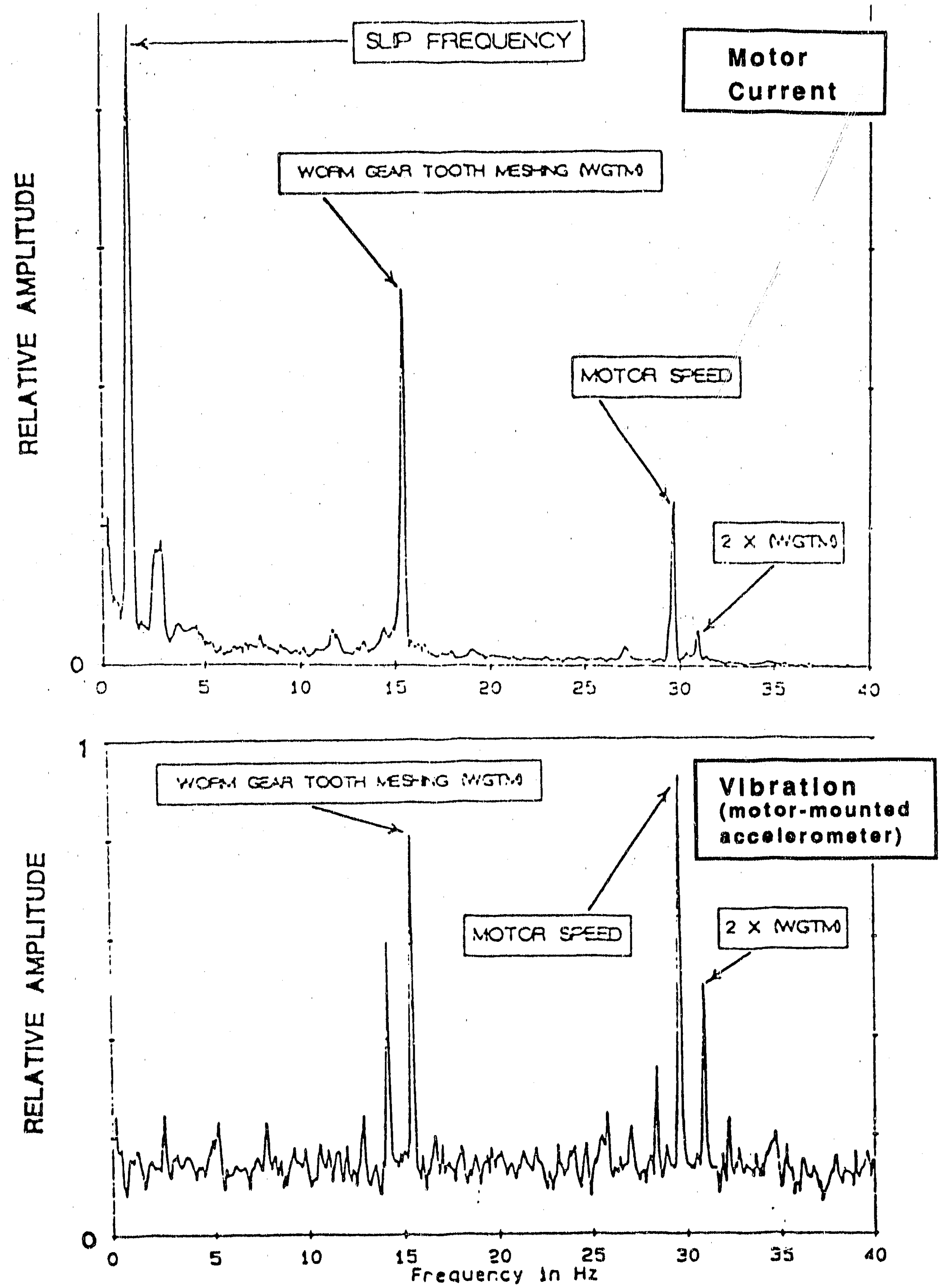


Figure 5. Comparison of motor current and vibration.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

What are the pros and cons of the application on this latest MOV industry problem go-around that we are in the midst of?

Response 1

You are referring to the Generic Letter 89–10. To me 89–10 is the old 85–03. 89–10 also includes some kind of testing that is supposed to be performed. We want the valve to operate in design basis conditions which, 89–10 has highlighted even more than 85–03. I think 89–10 goes deeper into the number of valves that need to be tested and also involves doing it in situ, which requires more testing under delta P. It is very hard today to extrapolate a signature from static conditions that you can take to dynamic conditions, and I think that has been proven by INEL tests at Wyle.

The friction factor used commonly was 0.3, and in some cases we know that it can go to 0.5.

But it seems to me that the valve industry, even if we have a good diagnostic system, does not know where we put the limit, how far we should go before we should do an overhaul and maintenance on the valve. Also, someone from KWU mentioned that the more you test a valve, the more chance you have to destroy that valve before you have to use it. That is true also.

So we have seen the result of the EG&G tests, we have seen the weardown on the seats. But whether that happened when we were in static conditions or when we were under dynamic conditions, that is very tough to know.

Question 2

How can you pick up OM–6 and OM–10 now, even though the regulation hasn't endorsed it?

Response 2

The regulation tells you how to update your plant to a later code version or to a different standard than the one that is officially in the regulation. And the answer is, it is an exemption from the regulation, and you can apply for staff approval. The technical thinking and the logic, and a lot of the basic thrust of OM–6 and 10, is in Generic Letter 89–04 on inservice testing.

The best way to get the OM–6 and 10 flavor into your inservice testing programs to ease the burden on you and get it done quickly is to implement the generic letter. Then you have to wait for the regulation to endorse OM–6 and 10. Now that process is nearing completion.

OM–6 and 10 and the regulation to endorse them should be out for public comment within a couple months. The formal endorsement of OM–6 and 10 will probably be out within eight to ten months. That really overshadows the Generic Letter 89–04 schedule. For most of you, the best approach is to look at 89–04, look at the discussion that is in Attachment 1, and follow those areas.

If there is a specific part of OM–6 and 10 beyond Attachment 1 of the generic letter that you want to employ now, the only way to do that is to include a relief request in your response to the generic letter, which is due in October. The request relief would use the OM–6 or OM–10 philosophy, and that must be approved prior to implementation by the generic letter's process.

Question 3

Our utility is on Table 1 of the generic letter, we have interim approval for a program already submitted that conforms with the 1983 edition of IWP and IVP. How then can we adopt legally the OM–6 and 10 criteria?

Response 3

After you get your SER, after you go through your changes, which means procedural changes and updating your program to reflect the SER and those things that are outstanding, if you then want to change to OM–6 and 10, you have to go through the process. You can update your plant to the generic letter at any time you want to, which means that after you get the SER it should already reflect much of the generic letter anyway, because we were thinking in the process when we wrote the generic letter.

So your SER should reflect much of the generic letter thinking, and, hence, OM–6 and 10. But if there are some aspects that you want to put into your program that go beyond your SER from OM–6 and 10, but aren't really in Attachment 1 of the generic letter, you have to apply for relief to employ specific parts. If you are going to update your plant to OM–6 and 10 in their entirety after you get your SER, that would be an exemption, and that is a difficult road. I would not recommend that. The easiest road to take is to do the generic letter and then wait for the regulation to approve OM–6 and 10.

CERTIFICATION TESTING OF SAFETY RELIEF VALVES

FOR

THE NUCLEAR POWER INDUSTRY

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ABSTRACT

This paper presents a summary of current test methodology used to perform recertification testing of Code Safety Relief Valves (SRVs). This paper discusses current issues in SRV testing including the following:

- Alternate media testing including a discussion of EPRI Report NP–4235.
- In situ testing of SRVs using lift devices.
- Effects of handling and transportation on set point.

SRV testing over the years at Wyle in close cooperation with the nuclear industry, NRC, and valve manufacturers provides the experience necessary to discuss lessons learned. These lessons may be helpful to those setting up inservice Inspection (ISI) Programs to effectively monitor SRV performance and meet the requirements of OM–1.

INTRODUCTION

Safety Relief Valves are the primary over–pressure protective device for nuclear power reactors. These devices are designed for high set point accuracy in order to perform their emergency safety function. They must also remain trouble free during routine plant operation, neither leaking excessively nor actuating prematurely which could lead to costly plant downtime.

Wyle has performed thousands of tests over the past fifteen years. These tests include "As Received" set pressure tests, leak tests, full flow blowdown tests, and other special tests, conducted for nuclear utilities with the valve manufacturers.

Current issues related to set pressure testing of SRVs include:

- Techniques and limitations of current test methodology including steam testing, testing with assist devices, and alternate media testing.
- Transportation issues.

There are two types of SRVs: Spring Operated Valves (Figure 1) and Pilot Operated Valves (Figure 2). Requirements for set pressure testing of these SRVs are set forth in References 1 through 4. This paper deals exclusively with PWR Class 1 Pressurizer Code Safeties and with BWR Class 1 Main Steam Safety Relief Valves.

CURRENT TEST METHODOLOGY

ANSI/ASME OM Part 1 and the industry recognize three basic methods of testing SRVs for set pressure, operability, and leakage:

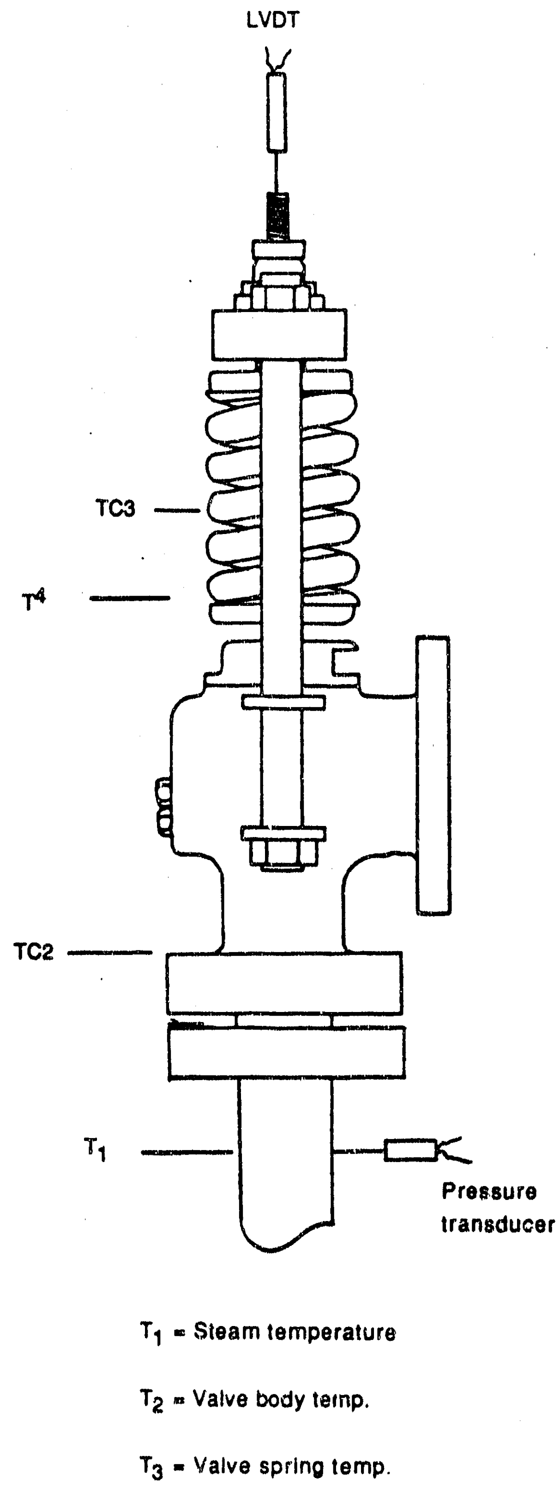

Figure 1. Test instrumentation — spring operated SRV.

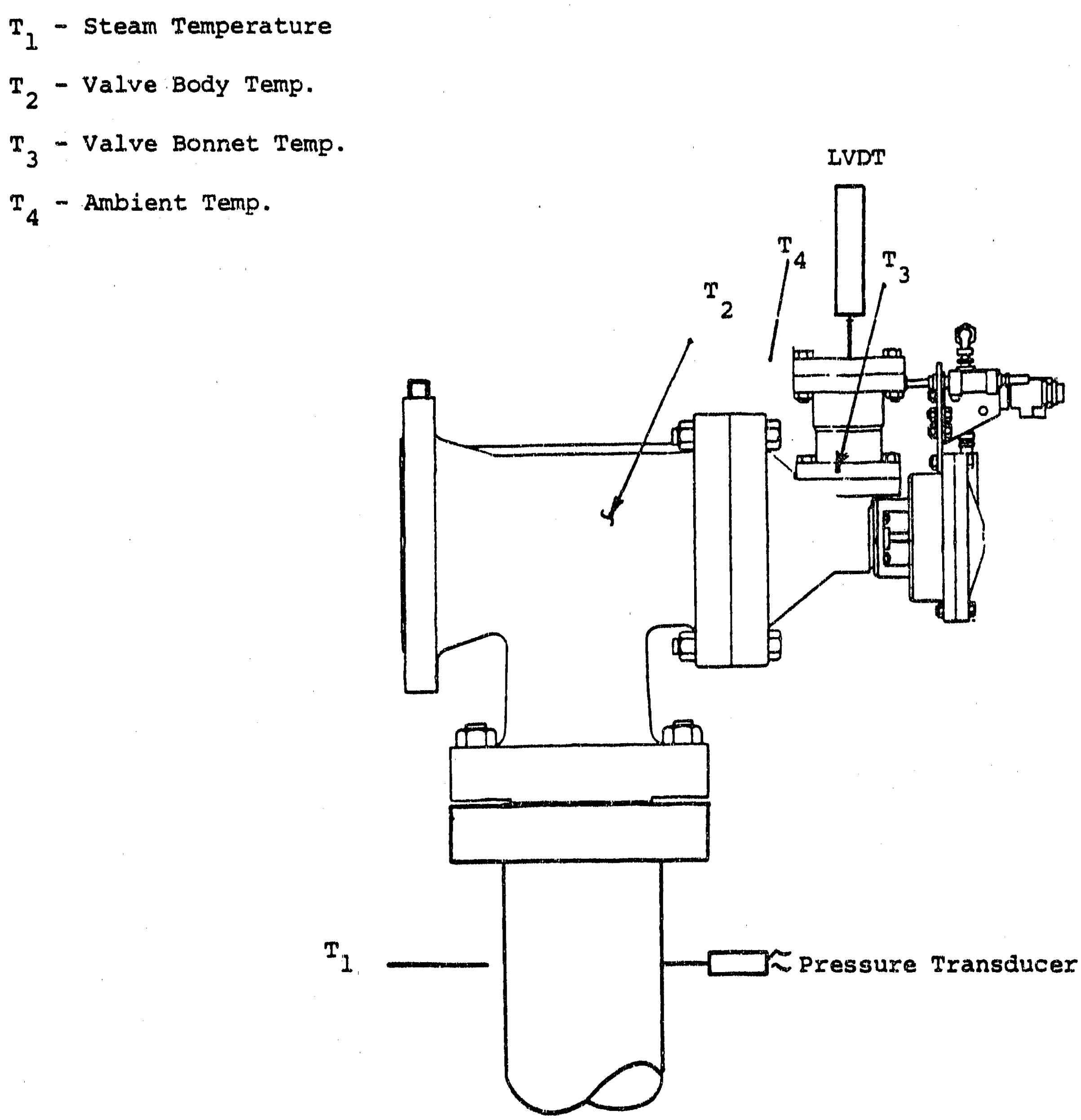

Figure 2. Test instrumentation — pilot operated SRV.

- Testing SRVs with service media at temperature and pressure conditions experienced inservice.
- Testing SRVs with service media at operational temperature and pressure using an assist device.
- Testing SRVs using an alternate pressure media.

ANSI/ASME OM Part 1 prefers Method (1) but allows Methods (2) and (3) for reasons of economy and practicality. Method (2) may be used if the overall combined *accuracy* meets the requirements of the OM. Method (3) may be used as long as *correlation* between the test medium and service medium including parameters of temperature and pressure have been established.

Accuracy

Most power plants require a set point tolerance of ±1% of nameplate set pressure. The valve is considered to be in set when actuated three consecutive times within this limit regardless of the method chosen.

The ±1% tolerance for set pressure should not be confused with the ±3% Acceptance Criteria of OM–1, Paragraph 1.3.3.1(d), for the "As Found" test which triggers testing additional valves if valve set pressure exceeds this limit.

OM Part 1 states that the test method including instrumentation, test equipment, calibration, etc., must be sufficiently accurate to perform its measurement task. Accuracy can be defined as the percent difference between an actual valve, X_A, and a measured valve, X_M, i.e.

$$\% \text{ Accuracy } = \frac{X_A - X_M}{X_M} \times 100 \quad .$$

OM Part 1 specifies an overall combined accuracy of +2/−1% of nameplate set pressure resulting in an actual set pressure within +1%/−2% of the measured valve, i.e.,

$$0.99\ X_A < X_M < 1.02\ X_A$$

or

$$0.98\ X_M < X_A < 1.01\ X_M \quad .$$

Figure 3 depicts a Pressurizer Safety set pressure in the region of interest. Three set point measurements are plotted within the ±1% tolerance band required by most plants. This ±1% corresponds to ±25 psi for a valve nameplate set pressure of 2500 psig. The dashed lines represent the limits of the overall combined accuracy. The actual set pressure value may lie anywhere within these limits.

Obviously, when selecting the test method, the *overall combined* accuracy must be considered in order to have assurance that the actual set pressure values lie within the desired limits. That is, if the tolerance of ±25 psi of nameplate is specified, then a method of testing including instrumentation that will deliver an accuracy on the order of ±2.5 psi would be desirable.

Method 1. Although all three methods have been or are in use at Wyle, depending on the requirement of the customer, the most common is Method 1.

A typical valve receives an initial receipt inspection. Then the valve is tested "As Found" for set pressure, operability, and leakage. The valve is installed in the test cell in its normal operating position on a steam accumulator. The valve is instrumented, insulated if required, and surrounded by an environmental box. Steam at 90% of nameplate set pressure is applied to the valve inlet to establish thermal stabilization. Following thermal stabilization, the valve is checked for leakage. Next, the steam pressure is increased at a controlled rate until the valve actuates. Additional actuations are performed with five-minute intervals between actuations. Set pressure adjustment may be performed, if required. Finally, a post-test leak check is conducted.

If rework is required, the valve is refurbished by the manufacturer or other repair agency following cooldown. At this time, ring settings and dimensional clearances are checked. Finally, the valve is tested again for recertification, if required, leak tested, then returned to the customer.

Leak Checks. Three methods of leak checks are conducted:

1. Cold mirror leak check
2. Quantitative leak check
3. Nitrogen bubble leak check.

If no audible or visual leakage is immediately evident, a cold (i.e. room temperature) mirror is inserted into the valve outlet and passed around the valve seating surfaces. The mirror is then examined for condensate formation. If quantitative leakage is required, a condensing coil is connected to the valve outlet, and the condensate collected in a graduated cylinder. This method is not effective unless the bonnet area is sealed from the body bowl. If nitrogen bubbles are required to be counted, the method of API Standard 527 is used. Up to 100 bubbles per minute may be counted this way. This test is performed with GN_2 with the valve at operating temperature or room temperature.

Instrumentation. Prior to test, the valve is instrumented to monitor the required test parameters. To prevent seat damage, spring-type valves are restricted at the spindle to a lift of approximately 50/1000 inch. Pilot operated SRVs are flow restricted by inserting a plug into the main seat outlet.

A Linear Variable Differential Transformer (LVDT) is installed on the spindle to measure the valve lift. In case of a pilot operated valve, the pilot disc is instrumented. The signal output is processed through conditioning electronics to the recording device. LVDTs are calibrated using a micrometer prior to test. The LVDT verifies lift and provides a timing mark for set point and reseat on the X–Y Plotter.

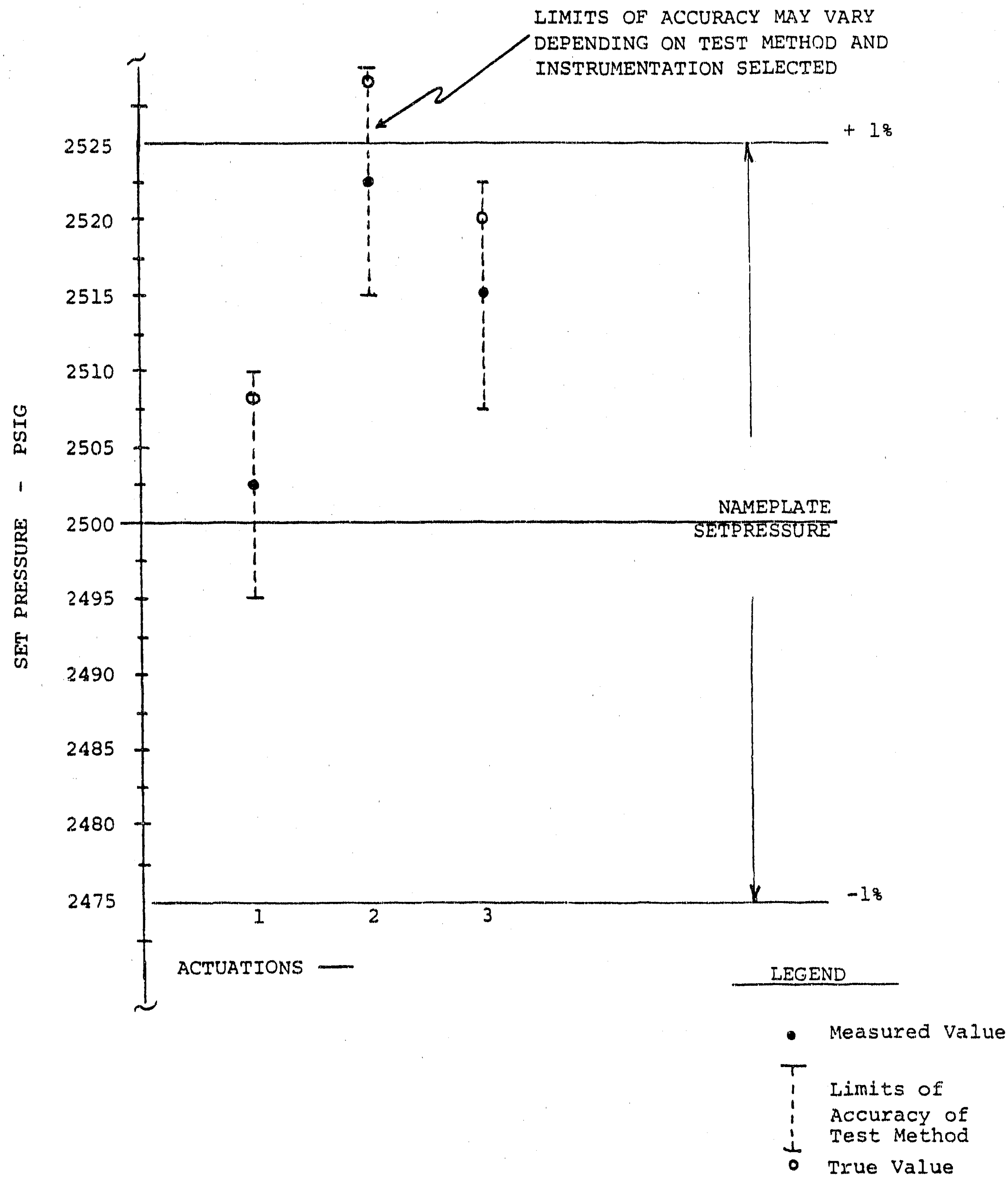

Figure 3. Set pressure measurement vs. accuracy.

K–Type thermocouples are installed on the valve where required. The number varies depending on the valve and test procedure. Typically, the following parameters are monitored:

- Steam inlet temperature as close to the valve inlet as possible
- Valve inlet neck
- Valve body
- Valve spring (upper, middle and lower)
- Valve spring ambient temperature.

The thermocouple output signal is monitored and recorded by a data logger. Printouts are at one–minute intervals and are used to monitor the valve during heatup and stabilization.

Accelerometers are used for testing pilot operated SRVs. The accelerometer output is on an oscillograph and is used to monitor valve stroke and response times on the order of 10 milliseconds.

Valve inlet steam pressure (and thus valve set pressure) is monitored using a pressure transducer. Following signal conditioning, the output signal is processed to the X–Y Plotter. A Heise test gage is also used in parallel and for comparison purpose. Both instruments are calibrated end–to–end using a deadweight tester of high accuracy prior to and following testing. Set pressure can be determined to an accuracy of $\pm 0.1\%$ on the X–Y Plot by means of point calibration methods (See Figure 4). Since the accuracy of the test results depends entirely on the method used to measure and record set pressure and, in this method, the Pressure (P_{Steam}) is measured with a pressure transducer ($P_{Transducer}$), rather than a pressure gage, the accuracy ($\pm \Delta P$) of Method 1 can be expressed as:

$$(P \pm \Delta P)_{Set\ pressure} = (P \pm \Delta P)_{Pressure\ transducer\ output} \quad (1)$$

TESTING USING ASSIST DEVICES

Method 2

Method 2 uses an assist device to test SRVs. This device can be used with the valve in place at operation conditions of temperature and pressure. The assist device is a pneumatic or hydraulic cylinder which applies a load to the valve spindle to assist the steam pressure to overcome the spring force. The steam pressure is measured by a test gage or pressure transducer. The load is measured by a gage or load cell. The set pressure must then be *calculated* using a formula which sums the forces acting on the valve spindle. These forces include the spring load, the load imposed by steam pressure acting on the seat area, and the load applied by the assist device:

$$F_{Spring} + F_{Steam} + F_{Assist} = 0 \quad (2)$$

Set pressure is calculated as follows:

$$\text{Set pressure} = \text{Steam pressure} + F_{Assist}/A_{Effective\ seat} \quad (3)$$

The effective seat area, A, is a factor which is usually provided by the valve manufacturer and varies depending on the valve orifice size, disc flexibility, and other factors. This is why we say that the set pressure is a calculated value.

Accuracy

The theoretical accuracy $(\pm\Delta P)_{Set}$ of Method 2 can be formulated as:

$$(P \pm \Delta P)_{Set\ pressure} = (P \pm \Delta P)_{Steam\ pressure} + (F \pm \Delta F)_{Assist}/(A \pm \Delta A)_{Seat\ area} \quad (4)$$

The solution of this equation for $\pm \Delta P_{Set}$ can be found in Appendix I.

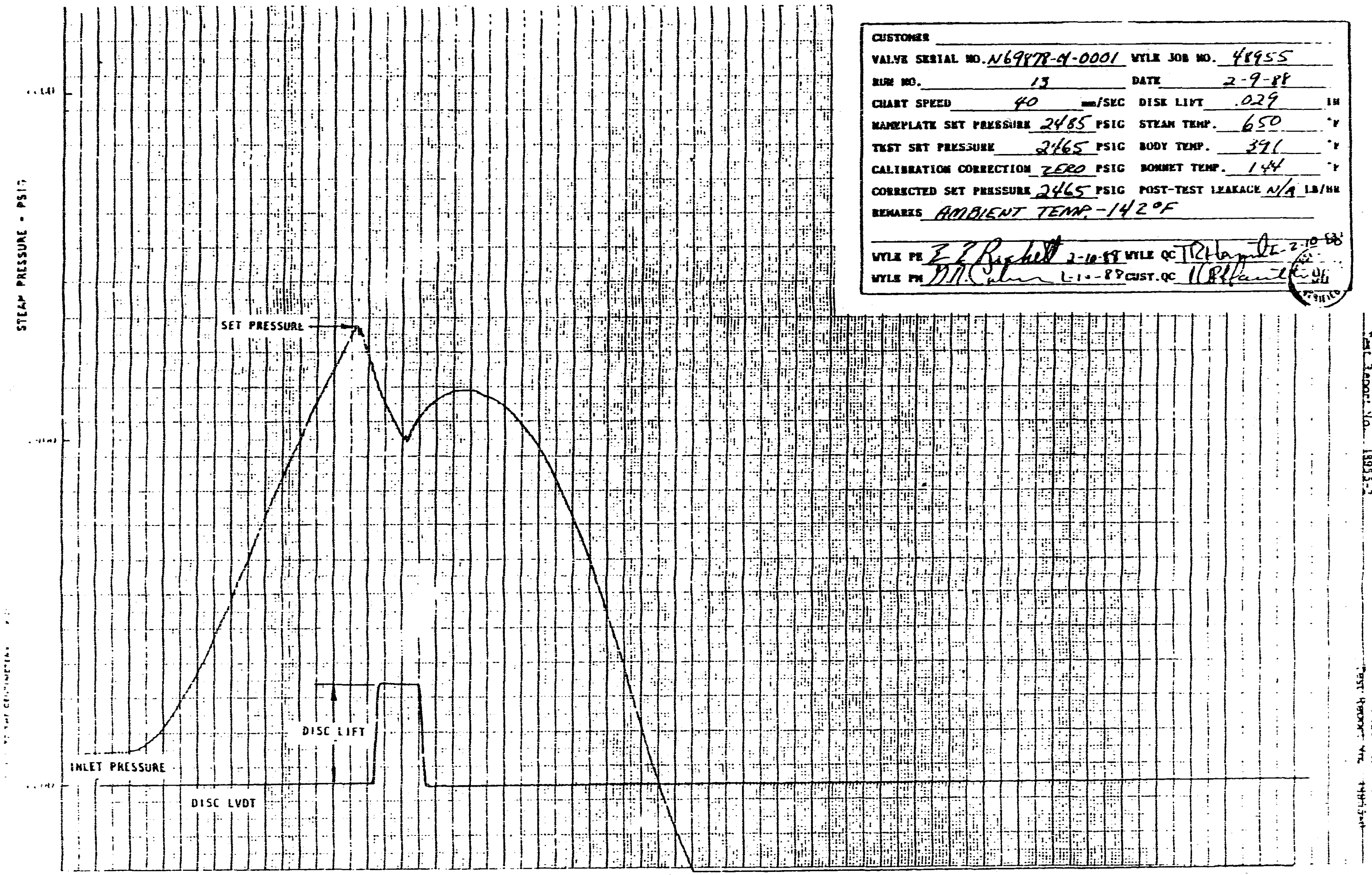

Figure 4. Example of X–Y plot.

$$\pm\ \Delta P_{Set} = \pm\ \Delta P_{Steam} \pm\ \Delta F/(A \pm \Delta A)$$

$$\pm\ \Delta A/(A \pm \Delta A)\ (F/A) \qquad (5)$$

which demonstrates that the accuracy of Method 2 is highly dependent on the overall combined accuracies of $\pm \Delta P_{Steam}$, $\pm \Delta F_{Assist}$, and $\pm \Delta A_{Effective\ seat\ area}$.

ALTERNATE MEDIA TESTING

Method 3

Method 3 of testing valves provides for the use of an alternate media such as gas or water in place of steam. Temperature must also be considered as the valve may be tested at room temperature or heated in an environmental enclosure. EPRI is proposing a method for alternate media testing of SRVs based on the results of a test program reported in EPRI Test Report NP–4235. The objective of this report is to establish a methodology which would achieve correlation required by OM–1987, Part 1.

EPRI tested ten Crosby HB–86–BP Pressurizer Safety Valves designed for loop seal service. The valves were tested for set pressure at various ambient temperatures three ways: (1) gaseous nitrogen, (2) saturated steam, and (3) using a lift device with steam at 90% set pressure. A matrix of all the set pressures is presented in Appendix D of the EPRI Report. Much can be learned from studying these results.

Steam With Lift Device

The variance between the results of testing on steam and the results of testing using a set device are notable. The data show a +1% to +3.4% difference between the two methods. We have already discussed the accuracy of lift devices which is highly dependent on the known accuracy of the effect seat area.

Nitrogen

The results of the alternate media nitrogen tests reported by EPRI are also interesting. The set pressures for nitrogen vs. steam at ambient temperatures of 100°F vary from –3.4% to +3.8%. What is surprising is the wide variation between nitrogen and steam for the identical type of valve. Why plus in some cases, minus in others? In spite of the results, the EPRI report claims to establish "correlation" for each specific valve. The problem is that this report assumes that a correlation exists in the first place; that is, if one has at hand, set pressures for nitrogen and steam then they necessarily correlate. At Wyle, valves are tested on both steam and nitrogen. We have seen valves which leak on nitrogen but do not leak on steam, and valves which leak on steam but are bubble tight on nitrogen for the same style valve. Some valves have nitrogen and steam set pressures which are close, others not close. The variance seen in the EPRI results leads one to ask if correlation exists in the first place. It goes without saying that anything which upsets this correlation, such as spring replacement, leakage or alignment valve, would require *a re–establishment* of the correlation by steam testing in order to comply with OM–1987, Part 1.

TRANSPORTATION

Regarding transportation, does removal expose the SRV to risk during handling and shipment? What effect does transportation actually have on valve set pressure performance? Let us look at two case histories.

A Dresser Pressurizer Safety Valve, Model 31739A, NPSP 2495 psig, has been last tested in 1985. The "As Left" set pressures were 2486, 2475, and 2479 psig. The valve was returned Plant A where it remained in storage as a backup spare and was not placed in service. It was returned to Wyle and retested on April 28, 1989. The "As Found" set pressures were 2498, 2489, and 2496 psig. Six years of storage and transport to and from Plant A, a distance of 2,560 miles, resulted in little difference.

In February 1988, Plant B dropped a valve on–site. This valve, a Crosby Pressurizer Safety Valve, Style HB–BP, had been recently recertified at Wyle. The recertification set pressures were 2482, 2502, and 2495 psig. Plant B decided to retest this valve. Upon return to Wyle, the valve was retested. The "As Found" set pressures were 2479, 2511, and 2520 psig. Again, little difference even after dropping.

The data which exists does not support the claim that travel causes set pressure drift provided the manufacturers recommendation that the valves be shipped in a vertical position to preclude damage to the valve internals, such as bending the spindle, be followed.

SUMMARY

Most plants still send their SRVs off–site for test. While handling and transportation pose some risk, case histories show that the valves are rugged enough to withstand this type of handling without upsetting the set pressure or producing leakage.

At Wyle, valves are exposed to inservice conditions experienced in the plant. The temperature profile and ambient conditions are reproduced very accurately. Valve set pressure testing is achieved by accurate measurement of valve inlet steam pressure using a pressure transducer system, calibrated prior to and after test by a deadweight tester. Results are recorded on an X–Y Plotter. Pressure can be determined to be nearest +0.1% (i.e., 2.5 psi for Pressurizers, 1.5 psi for MSSVs).

When the valve is returned to the plant, a high level of confidence exists that when the valve goes back on–line the valve will not leak or discharge prematurely while still remaining below the 3% level demanded by the NRC.

REFERENCES

1. ASME Boiler Pressure Vessel Code, Section III.

2. ASME Boiler Pressure Vessel Code, Section XI, Subsection IWV, Inservice Testing of Valves in Nuclear Power Plants.

3. ASME/ANSI OM–1987, Operation and Maintenance of Nuclear Power Plants.

4. ANSI/ASME PTC 25.3 – 1976, Safety and Relief Valves.

5. ASME PTC 25.2 – 1966, Safety and Relief Valves with Atmospheric or Superimposed Back Pressure Before Discharge.

6. EPRI NP–4235, September 1985, Setpoint Testing of Safety Valves using Alternative Test Methods.

7. ANSI/ASME PTC 19.1 – 1985 Instruments and Apparatus, Part 1, Measurement Uncertainty.

8. ANSI/ASME PTC 18.2 – 1964, Instruments and Apparatus, Part 2, Pressure Measurement.

BIOGRAPHY

Gerald R. Carbonneau is Manager of the Steam Test Services Department at Wyle Laboratories, Huntsville, Alabama. He received his Bachelors degree in Aerospace Engineering from the University of Florida in 1970. Following four years in the military, Mr. Carbonneau has had 15 years of professional, technical, and supervisory experience. Since 1980, Mr. Carbonneau has been directly associated with testing safety relief valves for the nuclear power industry.

Mr. Carbonneau is a member of the ASME and is a Registered Professional Engineer.

APPENDIX I

CALCULATIONS

Calculate Accuracy ($\pm \Delta P)_{Set}$ of Method 2

Given:

(1) $$(P \pm \Delta P)_{Set} = (P \pm \Delta P)_{Steam} + \frac{(F \pm \Delta F)_{Assist}}{(A \pm \Delta A)_{Seat}}$$

and

(2) $$P_{Set} = P_{Steam} + \frac{F_{Assist}}{A_{Seat}}$$

Subtract Equation (2) from (1) gives:

(3A) $$\pm \Delta P_{Set} = \pm \Delta P_{Steam} + \frac{(F \pm \Delta F)_{Assist}}{(A \pm \Delta A)_{Seat}} - \frac{F_{Assist}}{A_{Seat}}$$

or

(3B) $$\pm \Delta P_{Set} = \pm \Delta P_{Steam} + \frac{F_{Assist}}{(A \pm \Delta A)_{Seat}} + \frac{(F \pm \Delta F)_{Assist}}{(A \pm \Delta A)_{Seat}} - \frac{F_{Assist}}{A_{Seat}}$$

Since:

$$\frac{F}{A \pm \Delta A)} - \frac{F}{A} \equiv \frac{FA}{(A \pm \Delta A)A} - \frac{F(A \pm \Delta A)}{A(A \pm \Delta A)} \equiv \frac{F(+ \Delta A)}{(A \pm \Delta A)A}$$

Then:

(3C) $$\pm \Delta P_{Set} = \pm \Delta P_{Steam} + \frac{\pm \Delta F}{(A \pm \Delta A)} + \frac{(\pm \Delta A)\, F}{(A \pm \Delta A)\, A} .$$

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Do you have any explanation why these valves demonstrate different set point settings when, in fact, they were not used at all in the plant and were just returned to them?

From a physical standpoint, with the repeatability of measurements that you got from the first and second series of tests, it doesn't make much engineering sense to me. Do you have any explanation for that?

Response 1

A lot of the set pressure changes we see are an increase in set pressure after a long time in storage, and there are theories that perhaps a binding takes place. That is, the valves lose some of their lubricity over a period of time. The bonding phenomenon is well known, at least in the Target Rock valves regarding the stellite, where there is set point drift. Usually it is from valves inservice, sometimes with differences as high as 200 psi. This is caused by a bonding of the two metals, which takes more pressure to pull them apart.

Question 2

If you exercise the valve through some growing pains, you may not wind up with the same thing. But when you have taken it from a test facility, stored it under ambient conditions, and returned it, that I would suspect would have a lesser effect.

Response 2

It is less, that is true, by an order of magnitude.

Question 3

You mentioned about the wide variations between nitrogen and steam, that you need a correlation between the two. How do you obtain this correlation? What is this correlation you are referring to?

Is that printed by the valve manufacturer or do you have to do your own tests, or what?

Response 3

The correlation is established by the OM. The idea is to obtain a steam set pressure in the plant. But if you are going to be testing on nitrogen, you have to demonstrate that the nitrogen set pressure that you tested it at is equivalent or will result in a set pressure within the tolerance established by the plant, plus or minus one percent of nameplate set pressure, with steam under it.

And it is incumbent upon the owner doing the test to come up with that correlation, and that is what people have been trying to achieve.

Question 4

With the lift limited to 50 thousandths, do you get any information at all which is representative of the in–plant blowdown of the valve? That is, what is the reseat point, and is that a realistic reseat point?

Response 4

No. In the case of spring–operated valves, no. We have a full flow system that we have to go to in order to establish blowdown. And in the case of pilot–operated valves like Target Rock's, there is a correlation which has been established by test, where we have correlated or the valve manufacturer, Target Rock, has correlated the limited flow testing with full flow testing blowdowns. There is a Table.

Question 5

Could you please comment on a related matter, on the association between volume under the valve and accuracy of the test in regard to any of the three methods?

Response 5

You want to have sufficient volume under the valve in order to lift the valve. What we have seen on very, very small volumes where we have tested on gas, if the valve does not lift completely, you will get a one–sided lifting of the disc.

You do not have sufficient volume to blow the valve full open. It also results in the LBDT trace is very weak and it doesn't give you a good crisp lift and show you where the actual opening point may be.

As far as how much volume is required, I could not tell you. We usually use about 30 cubic feet of steam.

ASSESSMENT OF THE EFFECTIVENESS OF ASME SECTION XI PUMP AND VALVE SURVEILLANCE TESTING

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ELECTRIC POWER RESEARCH INSTITUTE

ABSTRACT

The effectiveness of the pump and valve surveillance testing currently performed in accordance with ASME Section XI requirements was assessed. The assessment was based on a pilot comparison of actual inplant Section XI testing results to reported component degradation/failure histories obtained from NPRDS for two nuclear units over a period of four years.

An evaluation was performed on a component by component basis evaluating Section XI testing results, i.e., measured stroke times, vibration levels, etc., for all pumps and valves for which component degradations/failures were reported. The intent was to determine the extent to which specific testing methods actually detected or predicted specific component degradation types.

The statistical data base (two plants over four years) was insufficient to allow firm conclusions. However, the results do provide an indication of the effectiveness of specific Section XI tests in detecting/predicting specific degradation/failure types.

The seat leakage testing for valves was found to be effective in detecting existing degradation but could not be trended to predict future leak rates. Stroke time testing of motor and air operated valves was, in general, ineffective in either detection of existing degradation or prediction of incipient failure.

Vibration testing (displacement) of pumps was found to be somewhat effective in both detection and prediction of high vibration levels. Likewise, DP/Flow testing was found to be somewhat effective in detecting low flow conditions. However, no assessment could be made regarding the relationship between a low flow test result obtained under mini-flow conditions and actual pump degradation. The data were insufficient to assess the effectiveness of DP/Flow testing in predicting future flow anomalies.

ACKNOWLEDGMENTS

The author would like to acknowledge the contributions of the EPRI Pump and Valve Surveillance Testing Utility/Industry Task Group which, under the chairmanship of Garry Galbreath (Duke Power Company) provided valuable guidance throughout the effort. The author also acknowledges the contributions of Bruce Hinkley, John Moran and Robert Stanley of Cygna Energy Services who compiled and evaluated the information summarized herein.

NOMENCLATURE

Pi	Pump Inlet Pressure
DP	Differential Pressure across pump
Q	Pump Flowrate
V	Pump Vibration Displacement
Tb	Pump Bearing Temperature
Pr	Pump Reference Inlet Pressure
Qr	Pump Reference Flowrate
DP_r	Pump Reference Differential Pressure

INTRODUCTION

Currently, utilities operating nuclear power plants maintain pumps and valves in accordance with established maintenance programs. Such programs may involve preventative maintenance activities involving periodic maintenance, as well as, predictive maintenance actions involving use of sophisticated diagnostic equipment to allow early detection of component degradation.

In addition to these established maintenance programs, the Nuclear Regulatory Commission (NRC) requires periodic surveillance tests to be performed on safety–related pumps and valves. The NRC requires that such testing be performed in accordance with ASME Section XI testing requirements for pumps and valves. Although such testing does provide some indication of the availability of the component at the time of test, it is primarily intended to detect degradation in component performance prior to an actual failure on demand.

As part of an ongoing program to improve nuclear plant predictive maintenance methods, the Electric Power Research Institute, in conjunction with an industry advisory group, contracted with Cygna Energy Services to conduct a pilot study to quantitatively assess the effectiveness of ASME Section XI testing of pumps and valves in both detection of existing degradation and prediction of incipient failure.

The results of this evaluation provide a basis for focusing future EPRI research on improved predictive maintenance testing methods.

SCOPE

This project assessed Section XI testing requirements for motor and air operated valves and check valves. Hydraulically operated valves, safety relief valves and valves which are capable of only one operation (Category D valves) were excluded from the study.

Following a review of the population of pump/driver types in IST programs, the following pump/driver combinations were included in the study:

- Vertical motor driven turbine pumps
- Horizontal motor driven centrifugal pumps
- Horizontal turbine driven centrifugal pumps
- Motor driven positive displacement pumps.

APPROACH

In order to assess the effectiveness of Section XI pump and valve surveillance testing currently performed, two nuclear units (one BWR and one PWR) were selected for evaluation. Over 400 pump and valve degradation/failure reports obtained from the Nuclear Plant Reliability Data System (NPRDS) over the time period 1985 through 1988 were analyzed to ascertain the nature of the degradation and the method of detection. For each component degradation/failure reported, the previous surveillance test results for that component were analyzed to determine whether trends in parameters measured during Section XI testing would have indicated an incipient degradation/failure.

The intent was to assess the effectiveness of each specific Section XI test in detection/prediction of specific component degradation/failure types.

Analysis of NPRDS narratives and cross correlation with Section XI testing results allowed an assessment of the effectiveness of specific testing methods in detection of specific existing degradation/failure types.

For each component included in each plant's Section XI testing program and for which degradations /failures were reported in NPRDS, time history plots of measured Section XI test parameters were developed covering the entire period of evaluation. Analysis of trends depicted in these plots allowed an assessment of the effectiveness of trending specific Section XI test parameters in predicting incipient degradation/failure.

ASSESSMENT OF SECTION XI VALVE TESTING EFFECTIVENESS

Summary of Section XI Valve Testing Requirements

Each specific test required under ASME Section XI was evaluated for effectiveness in detecting and or predicting degradation/failure. Section XI testing requirements for valves are summarized in Table 1.

For the purpose of this study, the effectiveness of the stroke time, seat leakage and full stroke exercise tests in detecting component degradation/failure was evaluated.

Table 1. Summary of Section XI valve testing requirements

Valve Type	Stroke Time	Seat Leakage[a]	Valve Position Indication Verification	Full Stroke Exercise
Power operated	3 month interval	2 year interval	2 year interval	3 month interval
Check	—	2 year interval	—	3 month[b] interval

a. Only valves for which seat leakage is limited to a specific maximum amount in the closed position to fulfill their function are required to be leak tested.

b. For valves which cannot be full stroke exercised to the position required to fulfill their function during plant operation they must be so exercised at each outage if outages are more than months apart.

The following summarizes the acceptance criteria for each Section XI test required for valves:

Stroke Time. For power operated valves with nominal stroke times greater than 10 s, an increase in stroke time of greater than 25% from the previous test requires an increase in testing frequency from 3 to 1 month intervals until corrective action is taken.

For valves with nominal stroke times of 10 s or less a stroke time increase of 50% requires the same increase in testing frequency.

Seat Leakage (all category A valves). Maximum allowable leakage rates are to be specified by the owner based on functional requirements. Also, maximum allowable leak rates are specified in Section XI as 30XD (ml/hr) for water and 7.5XD (SCF/day) for air where D is the nominal valve size in inches.

For valves NPS 6 or larger, trending of the leakage rate is required. If a change in leakage rate relative to the last test reduces the margin to the maximum allowable rate by 50% or more, the test frequency must be doubled. If, based on projection of 3 or more test points, the leakage rate is expected at the next scheduled test to exceed the maximum permissible value by more than 10%, the valve must be replaced or repaired.

Full Stroke Exercise. All safety related power operated and check valves must be full (or if impractical part) stroke exercised to confirm operability. It should be noted that in many cases demonstration of full/part stroke capability for check valves is not practical and the NRC has allowed disassembly and inspection as an alternative to full/part stroke exercising.

NPRDS Reported Valve Degradations/Failures

Table 2 presents a summary of NPRDS reported valve degradations/failures for the two units selected over the period examined. A total of 349 valve degradations/failures were reported. Of these, 37% involved motor operated valves, 28% air operated valves and 9% check valves. The remaining degradations/failures primarily involved relief or hydraulically operated valves. It should be noted that many "degradations" reported in NPRDS are not actual functional failures of the component i.e. they would not affect its ability to perform its intended function and that Section XI testing is not necessarily intended to detect all degradation types. Examples of such "degradations" include packing leaks and body to bonnet leaks.

Apparent Valve Degradation/ Failure Detection Efficiency

Motor Operated Valves. Table 3 presents the apparent degradation/failure detection efficiency for each Section XI test for all reported motor operated valve degradation/failure types. The detection efficiency is defined as the percentage of total reported degradations/failures of a given type which were actually detected by a specific Section XI test. From Table 3 it can be seen that the seat leakage test was relatively effective in detecting seat leaks, bent stems and incorrect limit switch settings. Although detection of a bent stem and/or incorrect limit switch settings are not objectives of the seat leakage test, the fact that the valve

Table 2. NPRDS reported valve degradations/failures

Valve Type	Failure Type	Number	Total NPRDS Reports
Air operated gate	Excessive seat leakage	2	11
	Excessive stroke time	3	—
	Bonnet leak	2	—
	Air leak	2	—
	Defective solenoids	1	—
	Other	1	—
Air operated globe	Excessive seat leakage	5	27
	Excessive stroke time	4	—
	Packing leak	7	—
	Bonnet leak	1	—
	Tubing leak	1	—
	Limit switch failure	1	—
	Defective solenoid	4	—
	Electrical failure	3	—
	Other	1	—
Air operated butterfly	Excessive seat leakage	7	28
	Excessive stroke time	4	—
	Packing leak	6	—
	Tubing leak	3	—
	Gear failure	6	—
	Limit switch failure	2	—
Air operated relief	Excessive air leakage	12	28
	Setpoint drift	16	—
Air operated angle	Excessive air leakage	2	5
	Excessive seat leakage	1	—
	Solenoid failure	1	—
	Other	1	—
Motor operated gate	Excessive seat leakage	10	79
	Excessive stroke time	5	—
	Packing leak	21	—
	Bonnet leak	4	—
	Torque switch failure	2	—
	Limit switch failure	5	—
	Failed motor operator	11	—
	Spring back failure	2	—
	Electrical contact failure	4	—
	Gear box failure	3	—
	Leakoff plug failure	4	—
	Other	8	—
Motor operated globe	Excessive seat leakage	3	31
	Excessive stroke time	2	—
	Packing leak	4	—
	Torque switch failure	7	—
	Failed motor operator	2	—

Table 2. (continued)

Valve Type	Failure Type	Number	Total NPRDS Reports
	Electrical failure	3	—
	Spring pack failure	4	—
	Anti–rotation device failure	2	—
	Other	4	—
Motor operated butterfly	Excessive seat leakage	5	12
	Torque switch failure	2	—
	Limit switch failure	1	—
	Electrical contacts failure	1	—
	Actuator base block failure	—	—
Motor operated angle	Packing leak	1	2
	Other	1	—
Motor operated stop check	Excessive seat leakage	3	4
	Torque switch failure	1	—
Relief valves	Excessive seat leakage	7	19
	Bonnet leaks	1	—
	Setpoint drift	4	—
	Other	7	—
Hydraulic operated gate	Packing leak	1	—
	Limit switch failure	1	29
	Defective solenoid	2	—
	Hydraulic system failure	25	—
Swing check	Excessive seat leakage	11	28
	Bonnet leak	7	—
	Hinge pin leak	3	—
	Mechanical binding	3	—
	Worn hinge pin	2	—
	Other	2	—
Electro hydraulic gate	Packing leak	4	37
	Defective solenoid	2	—
	Hydraulic system failure	26	—
	Other	5	—
Electro/hydraulic globe	Defective solenoid	3	3
Excess flow check	Bonnet leak	1	1
Explosive gate	Other	1	1
Solenoid gate	Other	1	1
Safety	Other	1	1
Split disc check	Excessive seat leakage	1	1
Stop check globe	Other	1	1

Table 3. Apparent degradation/failure detection efficiency motor operated valves

Failure Type	Number of Failures	Percent Detected: Stroke Time Test	Seat Leak Test	Supplemental Diagnostics	Plant Operation
Deteriorated packing	1	—	—	—	100
Seat leak	19	—	68	—	32
External leak	8	—	12.5	—	87.5
Tight packing	1	100	—	—	—
Tight stem nut	1	50	—	—	50
Bent stem	2	—	100	—	—
Packing leak	25	4	—	—	96
Incorrect T.S. setting	2	—	50	—	50
Worn gears	3	—	—	—	100
Hydraulic lock of springpack	1	—	—	—	100
Motor failure	13	8	—	—	92
Declutch mechanism	2	—	—	—	100
Defective torque or limit switch	17	—	6	—	94
Mechanical failure springpack	5	—	—	60	40
Contacts failure	5	20	—	—	80
Electrical component failure	11	—	—	—	100
Anti-rotation device	2	—	—	—	100
Other	6	17	17	—	66

is operated and monitored during the performance of the test provides an opportunity to detect malfunctions other than seat leakage.

The stroke time test was effective in detecting tight packing and a tight stem nut and somewhat effective in detecting contact failures. Stroke time testing was ineffective in detecting all other degradations/failures reported. By far, the majority of valve degradations/failures were detected during normal plant operation and not during a Section XI test. Overall, stroke time testing per say, resulted in minimal incremental improvement in the rate of detection of degradation/failure of motor operated valves.

Air Operated Valves. Table 4 presents the apparent degradation/failure detection efficiency for each Section XI test for all reported air operated valve degradation/failure types. Review of Table 4 indicates that seat leakage test was found to be effective in detecting a bent stem, seat leakage, and somewhat effective in detecting broken gears, packing leaks, external valve leakage and limit switches out of adjustment.

With the exception of seat leakage degradations, the majority of degradations/failures were again detected during normal operation or by visual inspection of the component.

Stroke time testing was extremely ineffective in detecting any degradations/failures reported. Only one of the 83 air operated valve degradations/failures reported was detected by a stroke time test. As with motor operated valves, stroke time testing resulted in only a minimal improvement in the rate of detection of air operated valve degradations/failures.

Check Valves. Table 5 presents the apparent degradation/failure detection efficiency for each Section XI test for all reported check valve degradation/failure types.

The exercise test was successful in detecting one of the three reported instances of mechanical binding. The seat leakage test was quite effective in detecting excessive seat leakage. As would be expected, neither the exercise or seat leakage test were found to be effective in detecting worn hinge pins, hinge pin leakage or bonnet leaks. This indicates that the exercise test can be somewhat effective in detecting substantial internal degradation but that the seat leakage test may be more effective in detecting more modest internal degradation.

Apparent Valve Degradation/ Failure Prediction Effectiveness

For all valves for which degradations/failures were reported, time history plots of measured stroke times and seat leakage rates were developed using tabulated IST program results from the plants evaluated.

Motor Operated Valves. Figures 1 and 2 present typical stroke time and seat leakage rate plots for motor operated valves. Note that there was no apparent correlation between the failures indicated and prior trends in stroke times. This was true for all motor operated valve stroke time plots developed.

Seat leakage typically went from ~zero to in excess of allowable limits or remained at ~zero between the 2 yr test intervals. Gradual trendable leak rates were almost never obtained.

Neither stroke times nor seat leakage rates provided data which could be trended to predict any of the failures reported.

Table 4. Apparent degradation/failure detection efficiency air operated valves

Failure Type	Number of Failures	Stroke Time Test (%)	Seat Leak Test (%)	Plant Operation (%)
Air leak	9	11	11	78
Solenoid leak	16	—	—	100
Solenoid cable failure	2	—	—	100
Failure to open/close	12	—	8	92
Broken gears	5	—	40	60
Packing leak	12	—	42	58
Bent Stem	1	—	100	—
Seat leak	15	—	67	33
External leak	3	—	33	67
Limit switch adjustment	3	—	33	67
Other	4	—	—	100

Table 5. Apparent degradation/failure detection efficiency check valves

Failure Type	Number of Failures	Percent Detected: Exercise Test	Percent Detected: Seat Leak Test	Percent Detected: Plant Operation
Seat leakage	11	9	73	18
Bonnet leak	7	—	—	100
Hinge pin leak	3	—	—	100
Mech. binding	3	33	—	67
Worn hinge pin	2	—	—	100[a]
Other	2	—	—	100

a. Found during disassembly/inspection.

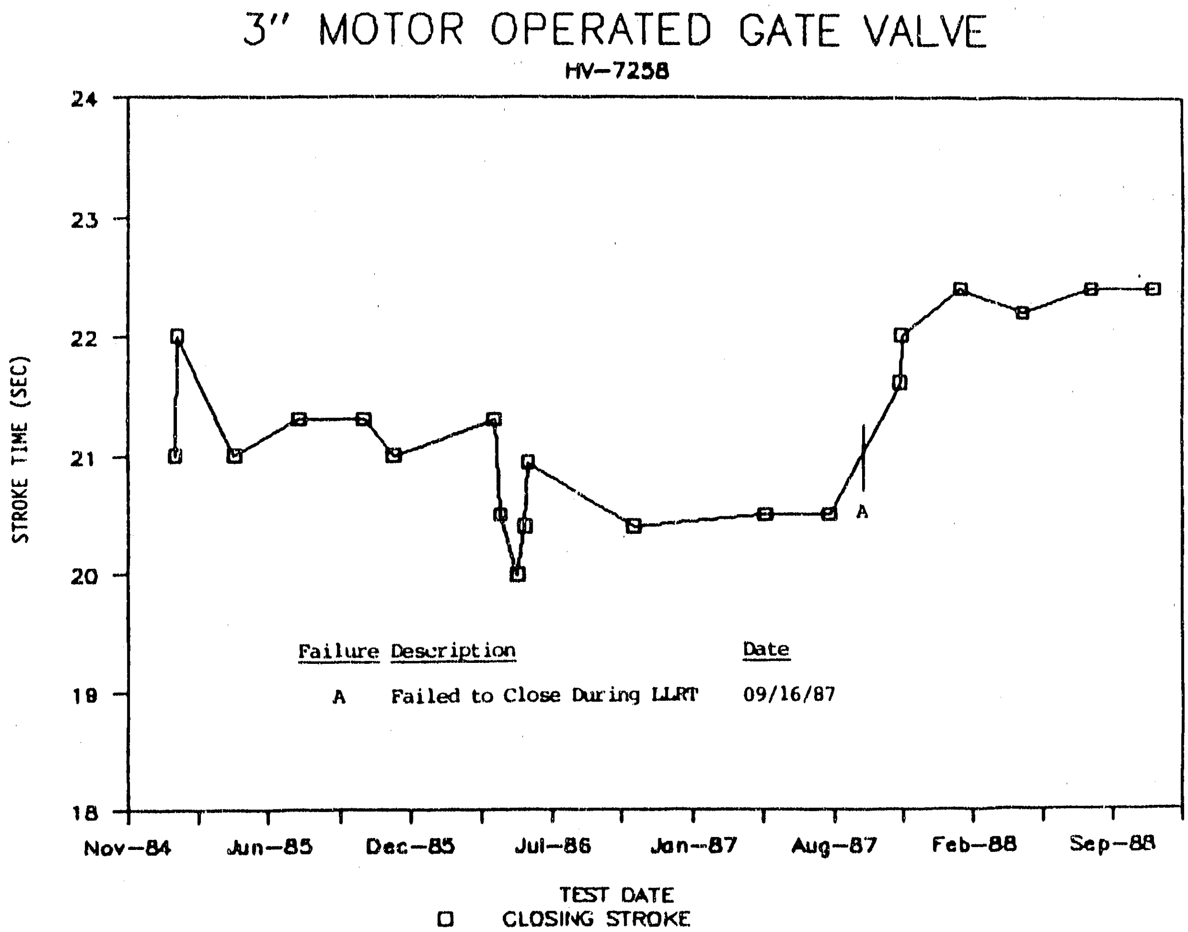

Figure 1. Typical motor operated valve stroke time history.

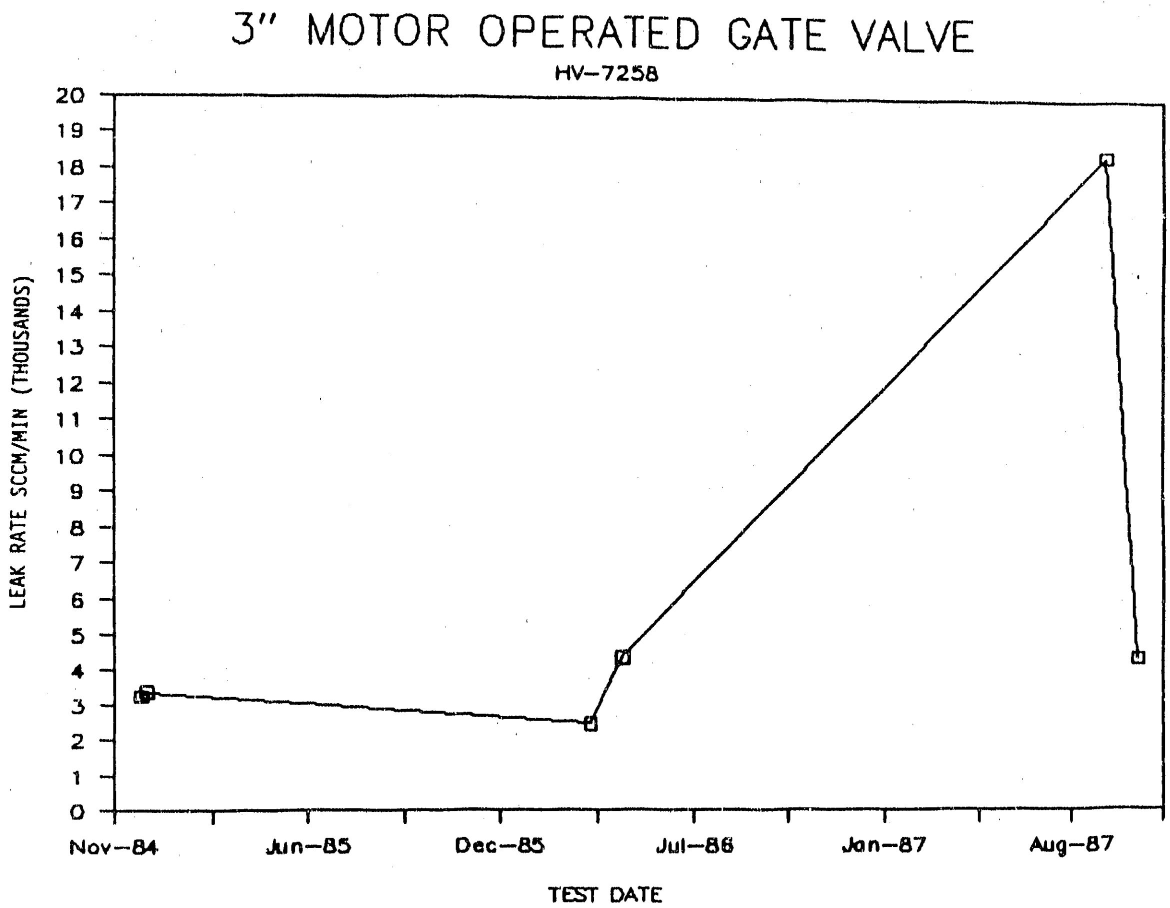

Figure 2. Typical motor operated valve seat leakage history.

Air Operated Valves. Figures 3 and 4 present typical stroke time and seat leakage rate plots for air operated valves. Again, as shown in Figure 3, there was no apparent correlation between the failures indicated and prior trends in measured stroke times. This was true for all air operated valve stroke time plots developed. In general, stroke times for air operated valves did display a significantly higher level of variation than did those for motor operated valves. This indicates that although trends in stroke times did not allow prediction of incipient failure, the stroke time may be sensitive to changes in air operated valve condition and/or air supply pressure variations.

As shown in Figure 4, seat leakage rates behaved essentially the same as for motor operated valves and were not trendable to predict failure or excessive leakage.

Check Valves. Seat leakage rate is the only parameter which is measured during Section XI testing of check valves, although flowrate is sometimes measured as an alternative to verification of full/partial stroke. Figure 5 presents a typical check valve seat leakage rate plot. Again, no correlation between reported leakage rate and reported failures were evident and the data were too erratic to allow trending.

ASSESSMENT OF PUMP TESTING EFFECTIVENESS

Summary of Section XI Testing Requirements for Pumps

Section XI testing requirement for pumps are summarized in Table 6. Note that delta pressure (DP) and flow (Q) are not evaluated independently. During testing, the operator generally sets the flow to equal the reference value and compares the DP reading to the allowables shown. The testing can, however, be done by setting the DP and measuring the resulting flow.

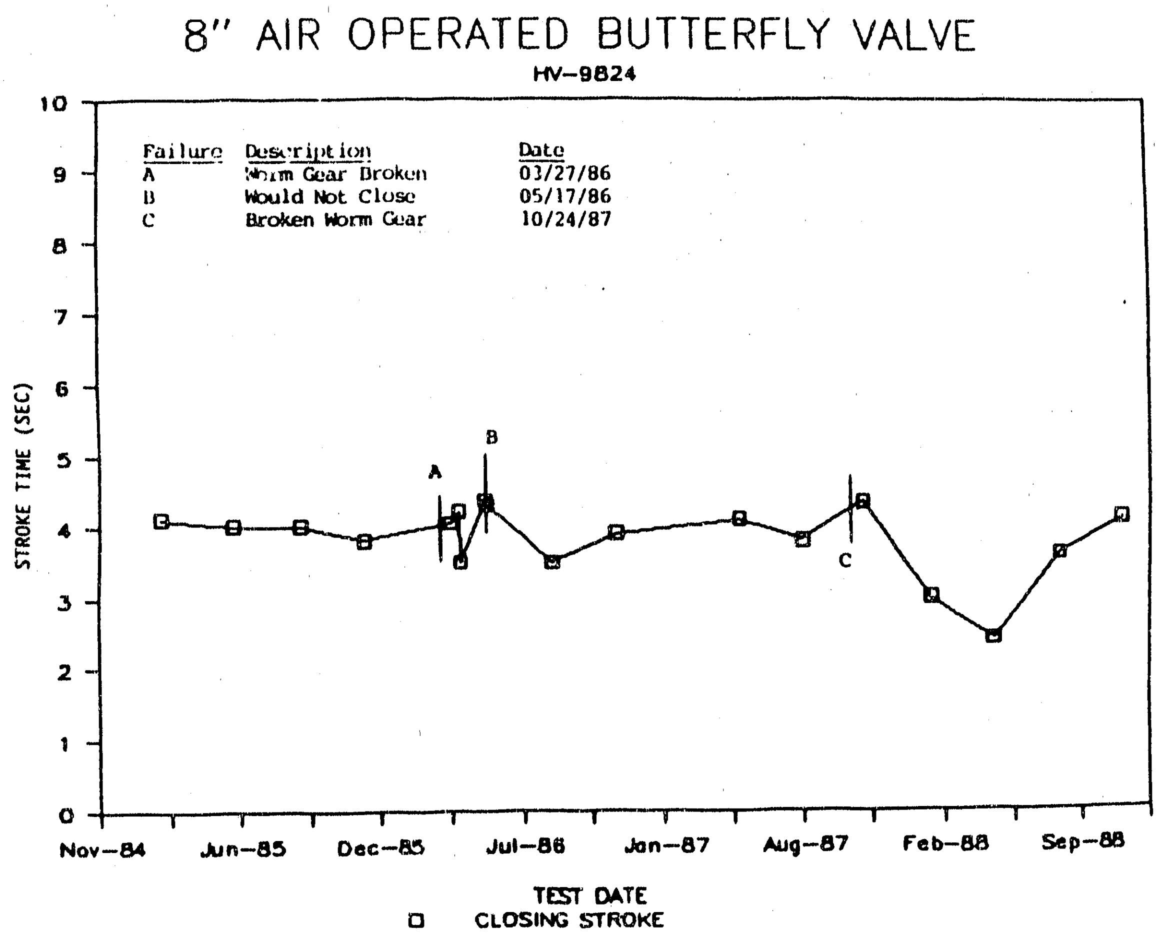

Figure 3. Typical air operated valve stroke time history.

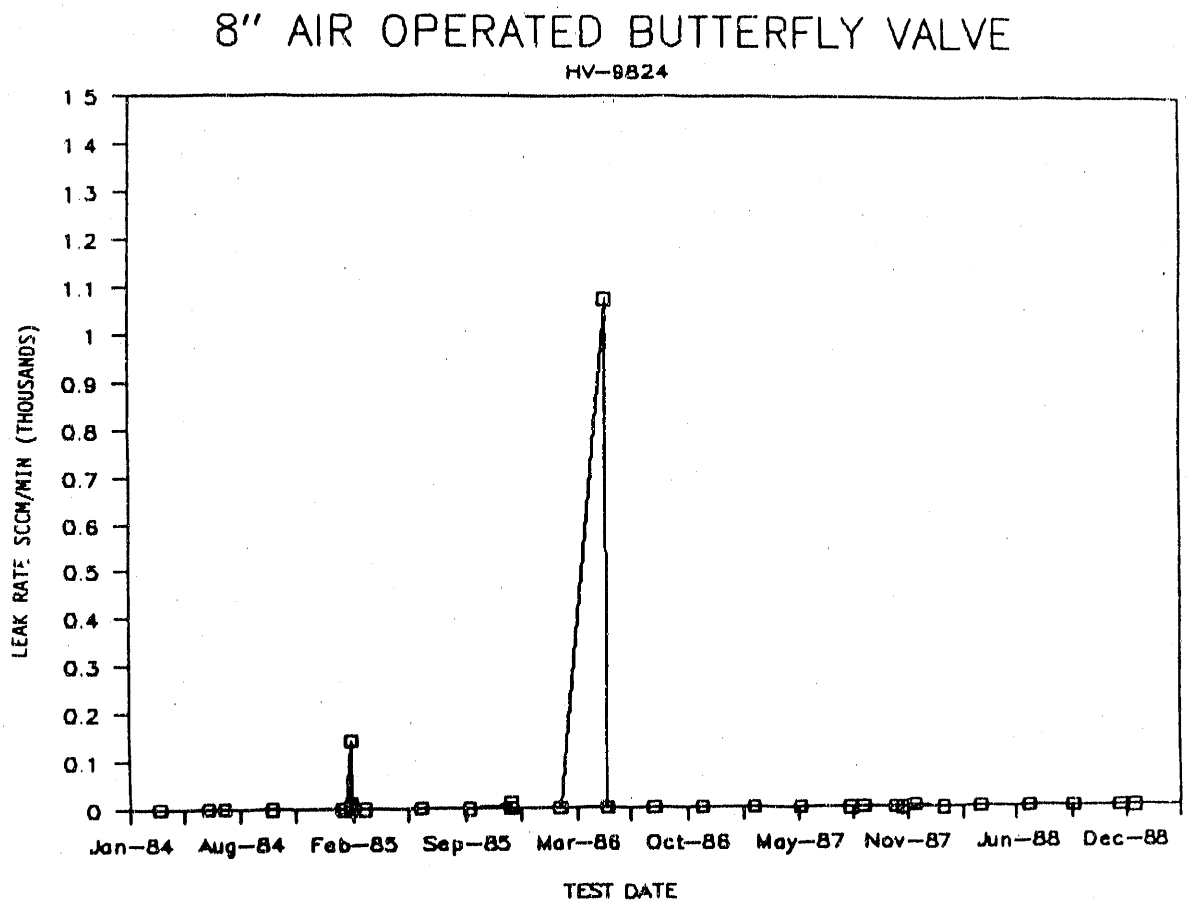

Figure 4. Typical air operated valve seat leakage history.

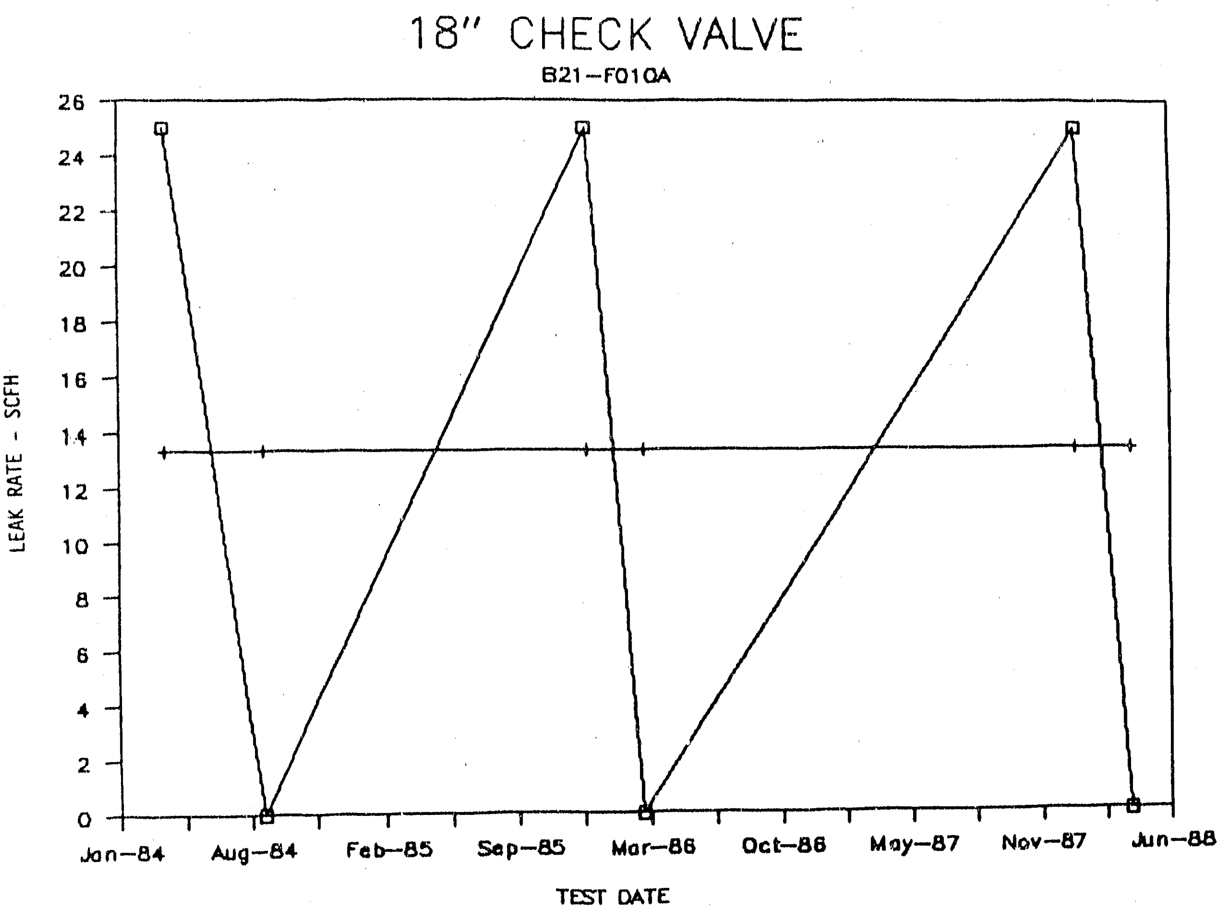

Figure 5. Typical check valve seat leakage time history.

Table 6. Summary of Section XI testing requirements for pumps allowable ranges of test quantities

Test Quantity	Acceptable Range	Alert Range Low Values	Alert Range High Values	Required Action Range Low Values	Required Action Range High Values
P_i	—[a]	—[a]	—[a]	—[a]	—[a]
DP	$0.93 - 1.02DP_r$	$0.90 - 0.93DP_r$	$1.02 - 1.03DP_{wr}$	$<0.90DP_r$	$>1.03DP_r$
Q	$0.94 - 1.02Q_r$	$0.90 - 0.94Q_r$	$1.02 - 1.03Q_r$	$<0.90Q_r$	$>1.03Q_r$
V when $0 \le V_r \le 0.5$ mils	0 – 1 mil	None	1 – 1.5 mils	None	>1.5 mils
V when 0.5 mils $< V_r < 2.0$ mils	$0 - 2V_r$ mils	None	$2V_r - 3V_r$ mils	None	$>3V_r$ mils
V when 2.0 mils $< V_r \le 5.0$ mils	$0 - (2 + V_r)$ mils	None	$(2 + V_r) - (4 + V_r)$ mils	None	$>(4 + V_r)$ mils
V when $V_r > 5.0$ mils	$0 - 1.4V_r$ mils	None	$1.4V_r - 1.8V_r$ mils	None	$>1.8V_r$ mils
T_b	—[b]	—[b]	—[b]	—[b]	—[b]

a. P_i shall be within the limits specified by the Owner in the record of test (IWP–6000).

b. T_b shall be within the limits specified by the Owner in the record of tests (IWP–6000).

All values shown in the table must be recorded every 3 mo except bearing temperature (Tb) which must be recorded once a year.

Deviations from reference values reaching the "Alert" range require a doubling of the testing frequency until the condition is corrected. Deviations reaching the "Required Action" range require the pump to be declared inoperable and not returned to service until the condition is corrected.

NPRDS Pump Degradations/ Failures

Table 7 presents the NPRDS reported degradation/ failure data for the two pilot plants. Note that only a small percentage of the pump degradations/failures reported represent conditions which Section XI testing is intended to detect i.e. internal component degradation resulting in excessive vibration or Flow/DP reduction.

Apparent Pump Degradation/ Failure Detection Efficiency

Horizontal Centrifugal Pumps. Table 8 presents the apparent degradation/failure detection efficiency for each Section XI test for horizontal centrifugal pumps.

Vibration testing was somewhat effective in detecting excessivevibration. All other degradations/failures were found during normal plant operation by visual monitoring.

Table 7. NPRDS reported pump degradation/failures

Pump Type	Failure Type	Number	Total NPRDS Reports
Horizontal centrifugal pumps	Mechanical seal leak	9	—
	High vibration	5	—
	Low pump flow	1	—
	Flange leaks	3	—
	Broken seal water line	5	—
	Scored shaft	3	—
	Oil leak	7	—
	Low cooling water pressure	1	—
	Pump casing cracked	1	35
Vertical pumps (turbine, centrifugal, in–line)	Low pump flow	1	—
	Flange leak	1	—
	Flush water insufficient	2	—
	Mechanical seal leak	3	—
	High vibration	1	—
	Low flow to bearings	1	9
Positive displacement pumps	Packing leaks	24	—
	Oil leaks	4	—
	Low pump flow	3	—
	Cracked casing	3	—
	Worn plunger	1	—
	Cooling system problem	1	36

Table 8. Apparent degradation/failure detection efficiency horizontal centrifugal pumps

		Percent Detected			
Failure Type	Number of Failures	Vibration	DP/ Flow	Bearing Temperature	Plant Operation
Low pump flow	1	—	—	—	100
Vibration	5	60	—	—	40
Mechanical seal failure	9	—	—	—	100
Oil leak	7	—	—	—	100
Seal piping leak	5	—	—	—	100
Cooling water insufficient	2	—	—	—	100
Score of shaft	2	—	—	—	100
Flange leaks	2	—	—	—	100
Cracked casing	1	—	—	—	100
Other	3	—	—	—	100

Vertical Pumps. Table 9 presents the apparent degradation/failure detection efficiency for vertical pumps. The DP/Flow test was effective in detecting the single reported low flow failure. All other failures were detected during normal plant operation. However, the statistical data base for the vertical pump failures evaluated is really insufficient to allow firm conclusions.

Positive Displacement Pumps. None of the 36 reported degradations/failures of positive displacement pumps were detected during a Section XI test. However, of the 36 failures, only three (three low flow failures) were of a type for which Section XI testing is intended to detect.

Apparent Pump Degradation/ Failure Prediction Efficiency

Vibration (displacement). Figure 6 presents a typical pump vibration (displacement in mils) measurement history. Two high vibration failures are indicated. The first (a) occurred during normal plant operation and was not preceded by any indication of incipient problem. The second high vibration failure (b) appears to have been predicted by the significant increase in vibration measured during the December 86 Section XI test. This indicates that trending of displacement measurements over time can be but is not always effective in predicting incipient failure or at least high levels of vibration.

Flow/DP. Figure 7 presents a typical pump DP measurement history. No correlation is seen between the high vibration failures and the virtually constant DP readings. The data reviewed did not include DP/Flow history information for any pumps which actually had low flow failures. As a result, no conclusion can be reached from this study as to the effectiveness of the DP/Flow testing in predicting such failures.

SUMMARY

A pilot study involving a review of pump and valve degradations/failures which occurred in two nuclear units over a four year period was conducted. The effectiveness with which Section XI testing either detected or predicted these degradations/failures was assessed. The statistical data base (~400 degradations/failures) most of which were not of a type for which Section XI testing is intended to detect, was not sufficient to draw firm conclusions. However, the results of the study provide some insight into the effectiveness of Section XI Testing. Based on this review, the following observations are made:

1. Seat leakage testing of valves was very effective in detecting existing valve degradation.

2. Seat leakage testing at the currently required intervals was not effective in predicting future leakage rates.

3. Stroke time testing of motor and air operated valves was not effective in either detecting or predicting valve degradation/failure. Obviously, if a valve is substantially degraded such that it will not operate, a stroke time/exercise test will detect the condition. However, the stroke time testing performed in accordance with Section XI requirements detected only an incremental increase in the number of valve degradations/failures detected

Table 9. Apparent degradation/failure detection efficiency vertical pumps[a]

Failure Type	Number of Failures	Percent Detected: Vibration	DP/ Flow	Bearing Temperature	Plant Operation
Low flow	1	—	100	—	—
Vibration	1	—	—	—	100
Flange leak	1	—	—	—	100
Cooling water insufficient	3	—	—	—	100
Mechanical seal	2	—	—	—	100
Other	1	—	—	—	100

a. Includes vertical turbine, centrifugal and in-line designs.

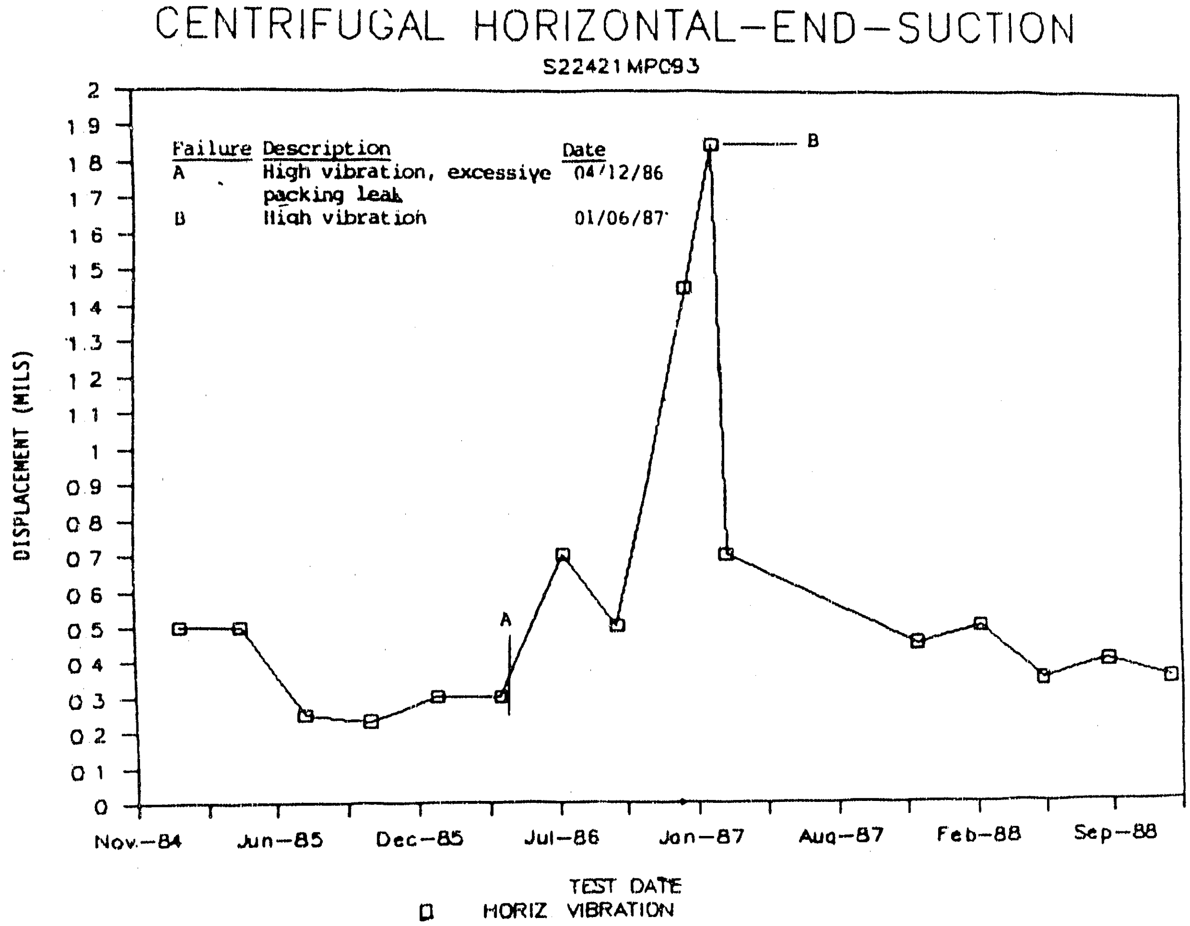

Figure 6. Typical pump vibration (displacement) time history.

during normal plant operation (even total failures to operate) and trending stroke time data provided no indication of incipient failure.

4. Vibration (displacement) measurements were somewhat effective both in detecting existing and predicting incipient high pump vibration levels.

5. DP/Flow measurements were effective in detecting existing low flow conditions, but the data was insufficient to determine whether such measurements can be trended to predict substantial flow reduction.

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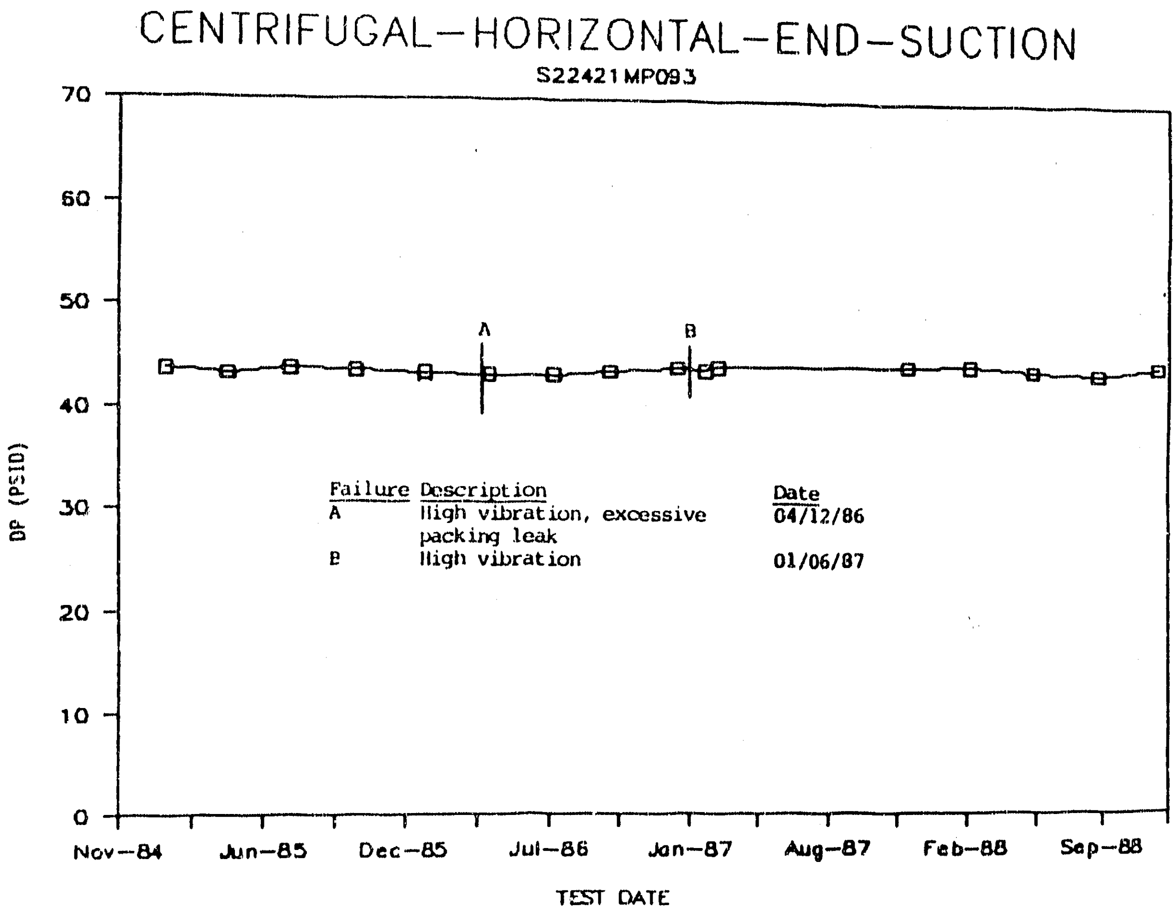

Figure 7. Typical pump DP time history.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Were either of the plants involved in the study using pump or valve diagnostic equipment, such as we have demonstrated here, that is, the Vision program on pumps or MOVATS?

Response 1

Panel Yes, they were.

Attendee Was that data reflected in the report?

Panel One of the categories here is plant operation, and we also had a category for supplemental diagnostics. Of the failures that were occurring, I can not say how many of them were picked up during MOVATS testing or some type of testing like that.

It may have been that in some cases there were actual readouts during a supplemental diagnostic type test. In a lot of cases, an NPRDS report does not say, we did a supplemental diagnostic test of this type and we have a problem. It just says, we have a problem.

Comment

I am also a member of the O&M Main Committee. Actually, I found this presentation very depressing from a standards development viewpoint. What it basically points out is that for a vast majority of the requirements that we have in the O&M standards, we cannot, from that data, get any semblance of assurance that what is happening in the nuclear plants is acceptable.

I think that, based on the data that we have here, EPRI and the industry should, in fact, expand the sample of the program to get a much larger data base, either to confirm that these results are typical of the industry or, if not, then to come up with some recommendations to O&M and ASME on how we can improve the testing to get a better predictive methodology incorporated into the standards.

Comment

I think that there is a philosophy that, if you go out and check the valves and do some tests on them, you can see things happening and you can therefore determine, if it is going to break before it breaks.

I think we can learn a little bit from the experience of the Europeans. The French and the Germans are not taking that approach. They do not go out and stroke time test motor-operated valves. They have good maintenance programs. They do testing on a much less frequent basis to confirm that a valve seems to work today, but not from the standpoint of, does it work a little different than it did three months ago.

So I think we should think about the philosophy of surveillance testing. This is a big NRC philosophical approach to evaluating what is going on in, not just pumps and valves, but all the components, and whether that really makes sense all the time.

Question 2

How effective are the tests required by Section XI in providing a high degree of assurance that the valve will operate the next time it is called upon to operate? Has that been included in the program? And are we detecting degradation, because, sure, you are going to have degradation.

Response 2

One way I would propose that you measure that is, how many failures did we have and how many of them did we catch by doing these tests? That was the first part of the study. The answer is, at least in terms of stroke time testing, it is a very, very small percentage.

Comment

Maybe that is the wrong parameter being measured. Perhaps we should look at another one if the goal of Section XI is to be accomplished.

Now, if you are looking at degradation of valves for maintenance purposes, that is a different subject, in my opinion.

Question 4

You had a suggestion that on a pilot study, you not use stroke time as your parameter in looking for degradation and operability, and you eliminate it.

Do you have a suggestion as to what you would substitute during that pilot study for monitoring degradation?

Response 4

It would be a good idea on that study to substitute several things and see how they all work out. So I would not propose one. I think part of that would to looking at other ways of doing it, hopefully as easy as stroke time testing.

MOTOR OPERATED VALVE STROKE TIMING; IS THERE VALUE?

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ABSTRACT

Both ASME Section XI, Subsection IWV and ASME/ANSI OMa–1988, Part 10 require stroke timing of certain power operated valves. This requirement is intended to detect valve degradation and subsequent maintenance, repair or replacement needs. However, the adequacy of stroke timing, especially for motor operated valves, has met much skepticism in the industry. This paper will demonstrate that stroke timing for ac motor operated valves is inadequate and provide a non–intrusive testing alternative. It will also discuss the value of stroke timing for dc motor operated valves.

ACKNOWLEDGMENTS

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INTRODUCTION

The stroke timing of certain safety–related power operated valves is presently required by ASME Section XI Subsection IWV and the future ASME/ANSI OMa–1988, Part 10 (OM–10).

The theory behind valve stroke timing is based on the assumption that degradation mechanisms that cause increases in the resistance of the valve to operate can be detected by changes in the valve's stroke time. Unfortunately, for some power operated valves such as solenoid valves, normal stroke time is so short it is almost impossible to measure changes. For others, such as ac motor operated valves, stroke time varies little with changes in the resistance of the valve. A recent EPRI/Utility/Industry task group study has determined that current valve stroke timing procedures can provide little or no indication of valve degradation that could cause failure.

The three most common power operated valve actuator types found in safety related nuclear applications are motor operated, air operated and solenoid operated. This paper will address only motor operated valves.

Factors Effecting Stroke Times

Many factors can effect stroke times. In fact, any condition which changes the resistance to valve operation will effect stroke time. Stem binding, lack of proper or dirty lubrication, corrosion, and overly tight packing are some of the degradation mechanisms detectable by increased stroke times. In addition, system conditions such as flow rate, differential pressure and temperature can also effect valve stroke times.

Motor Operated Valves

In the development, review and updating of inservice testing programs, clients have often questioned the requirement to measure stroke times on motor operated valves because the valves are driven by synchronous motors, and would fail before the stroke time increased by 25%.

The apparent inadequacy of the test raised a number of questions:

- How does degradation really effect stroke time of motor operated valves?
- If stroke time is ineffective, is there some other parameter that can be used to monitor degradation?
- Can testing of this other parameter be non–intrusive?
- What about dc motor operated valves?

How it Works

Regardless of the motor type (ac or dc) the working of a motor operated valve is the same. Figure 1 provides a simplified motor operator showing a lost

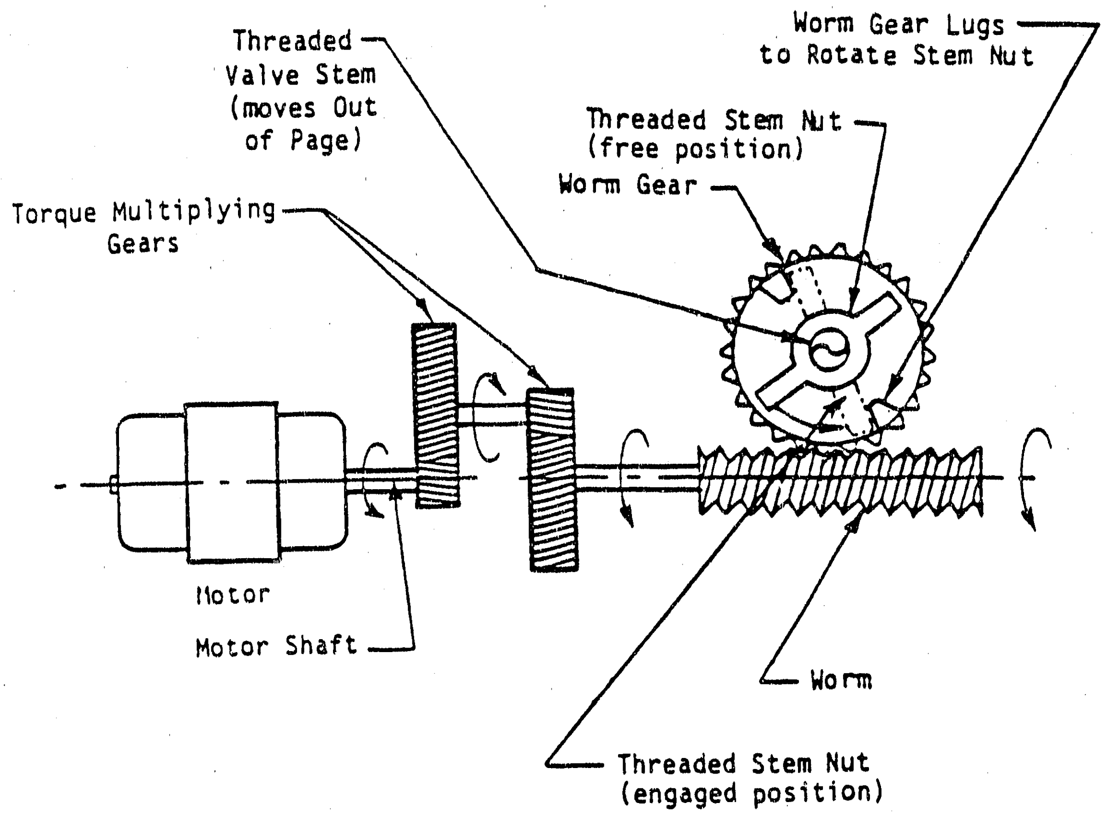

Figure 1. Motor operator showing lost motion device.

motion device. As shown in Figure 1, the motor turns a set of torque multiplying gears which drive the worm. The worm drives the worm gear which drives the lost motion device until it impacts the stem nut and the "hammer blow" is applied. Operation continues until the limit switch opens during an opening event and the torque switch opens during a closing event.

Motors

Safety related motor operated valves are typically actuated either by a 3–phase, 230/240 or 460/480 V ac squirrel cage induction motor or a 125 V/250 V dc compound–wound motor. The ac motor speed is normally 1800 rpm, however, a 3600 rpm motor may be used in some applications. The dc motor speed is normally rated at 1900 rpm.

A brief discussion of specific features of typical motors utilized is provided to form a basis for better understanding of the motor operated valve operation.

- *High starting torque* – The torque rating of the motor is based on the available starting torque which is usually between 65 and 90% of the motor stall torque. The motor is sized so that the rated torque is slightly higher than the torque required to provide the stem thrust to seat or unseat the valve. This torque is required for a very short time lasting only fractions of a second to several seconds.

- *Short duty–cycle* – The motors do not reach thermal equilibrium during operation. The rated torque can only be delivered for less than a minute, 15 to 30 s is typical, before the windings overheat. The duty–cycle is 15 min for three–phase ac motors and five min for dc motors. This duty–cycle is based on the temperature rise for a motor running at 20% of the rated torque. The justification for using this lower torque value to establish the duty–cycle is that the stem thrust required for most of the valve stroke is 20 to 40% of the seating force in most applications. Since the gearing is more efficient at running speeds than at starting, the torque at the motor is usually only about 10 to 25% of the rated torque. Even in cases where the running torque is 40% of the rated torque due to a tight valve packing or poor lubrication, there is usually adequate margin against motor failure if the stroke time is a few minutes or less.

- *Totally enclosed non-ventilated (TENV) motor* – TENV motor frames are used because a ventilating fan, which would increase motor size (inertia and weight), is not effective for the cooling motor which operates for short intervals. TENV motor frames are enclosed to prevent the exchange of air between the inside and outside of the case, therefore, making the motor susceptible to over heating if operated above design running currents.

- These motors are provided with thermal overload devices, however, in accordance with Regulatory Guide 1.105 these overloads are normally bypassed on safety-related valves.

Motor Operating Valve Stroke Timing

To demonstrate a stroke time effectiveness, a theoretical example will be provided for both ac and dc motor operated valves. In addition, a laboratory test is provided for the ac motor operated valves.

Figure 2 "Speed vs. torque and current for typical 1800 rpm, 60 Hz, 3-phase, 230 V ac motor" and Figure 3 "Speed vs. torque and current for typical 250 V dc compound-wound motor" are provided as a basis for the theoretical examples. Although, these figures represent motors with a maximum torque of 10 ft-lbs and a running torque of 2 ft-lbs, the relationship between the characteristics is similar regardless of motor size.

For both the ac and dc theoretical examples, the valve stroke time is assumed to be 30 s at normal running torque with a limiting valve stroke time of 45 s. Based on this stroke time, a 25% increase for IWV and an increase or decrease of 15% for OM-10 would occur before any action would be required to be taken. Because of the different characteristics of ac and dc motors, they will be discussed separately.

AC Motor Operated Valves

Because the ac motors are synchronous, stroke time will not vary significantly during changes in the valve's operating resistance. This is demonstrated in the following example and laboratory test.

Theoretical Example. Between normal testing of a valve, the packing is tightened to correct a leak. The tightening of the packing results in an increase in valve running torque from 2 to 4 ft-lbs. Based on Figure 2, this increase in torque would result in a decrease in motor speed of approximately 20 rpm or 1%. Since change of motor speed would result in an equivalent percentage change in stroke time, stroke time would increase by 1% or 0.3 s. Because stroke times are recorded to 10% of the limiting valve stroke time per IWV and to the nearest second per OM-10, this change would not be detected. In fact, if the running torque increased to the maximum torque value of 10 ft-lbs, the stroke time would only increase 17%, considerably below the 25% increase allowed by IWV and just over the 15% limit imposed by OM-10. Furthermore, due to the short duty cycle and total enclosure design of these motors operating at maximum torque would probably cause the motor to either trip on thermal overload (if installed) or burn out.

As in Figure 2, the same increase in valve torque from 2 to 4 ft-lbs would result in an increase in motor current of approximately 1.6 amps or 38%. Therefore, motor current is much more sensitive to changes in the valve's operating resistance than stroke time.

Laboratory Test. The laboratory test was performed on a 12" Walworth Gate Valve equipped with a Limitorque SMB-2 motor operator. The test valve was connected to a MOVATS series 2150 analyzer with a Simpson Amp Clamp Model 154-2 to measure running current. Since the valve was mounted in a test stand and not to any process piping, increases in valve thrust were accomplished by adjusting the packing. Due to limitations of resistance available by packing adjustment the nameplate running current of 12 amps was not able to be achieved during the test. However, the data obtained supports the theoretical conclusion.

Rated Torque = 10 Ft-Lb Current (At Rated Torque) = 11 Amps
Running Torque = 2 Ft-Lb (20% of Rated Torque) Current (At Running Torque) = 4.2 Amps
Locked Rotor Torque = 11 Ft-Lb Locked Rotor Current = 23.8 Amps
Duty Cycle = 15 Minutes (Based on Current Drawn at Running Torque)

Motor Data Courtesy of Limitorque

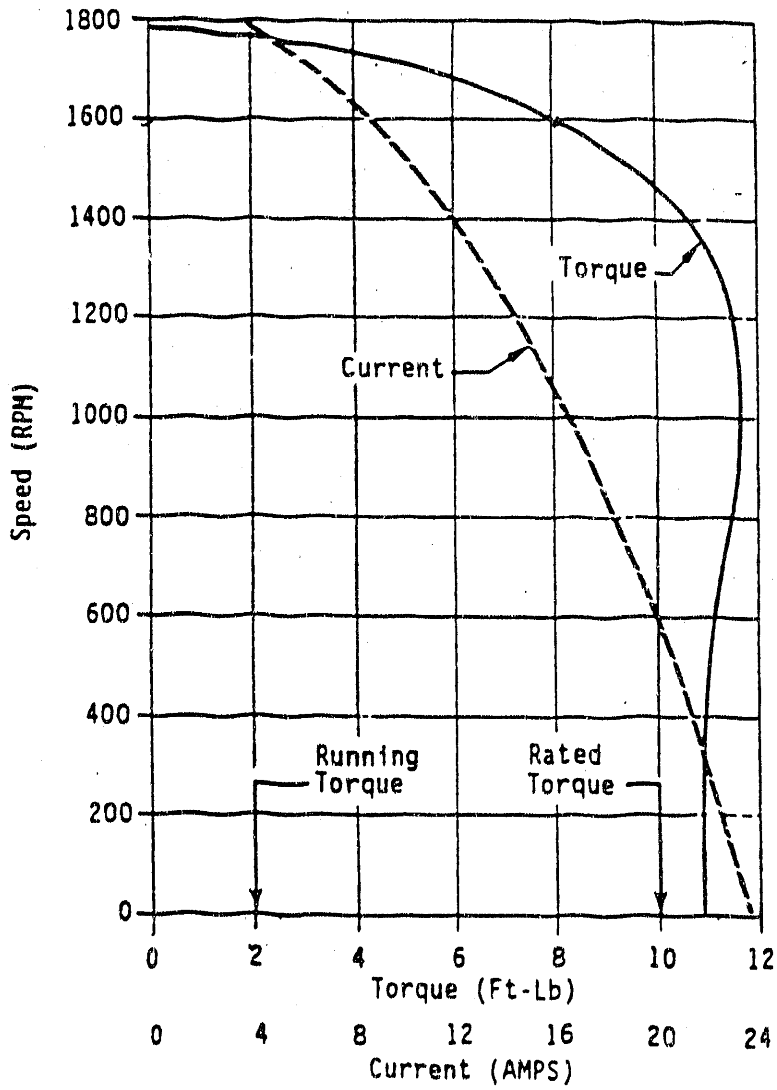

Figure 2. Speed vs torque and current for typical 1800 rpm, 60 Hz, 3-phase, 230 V ac motor.

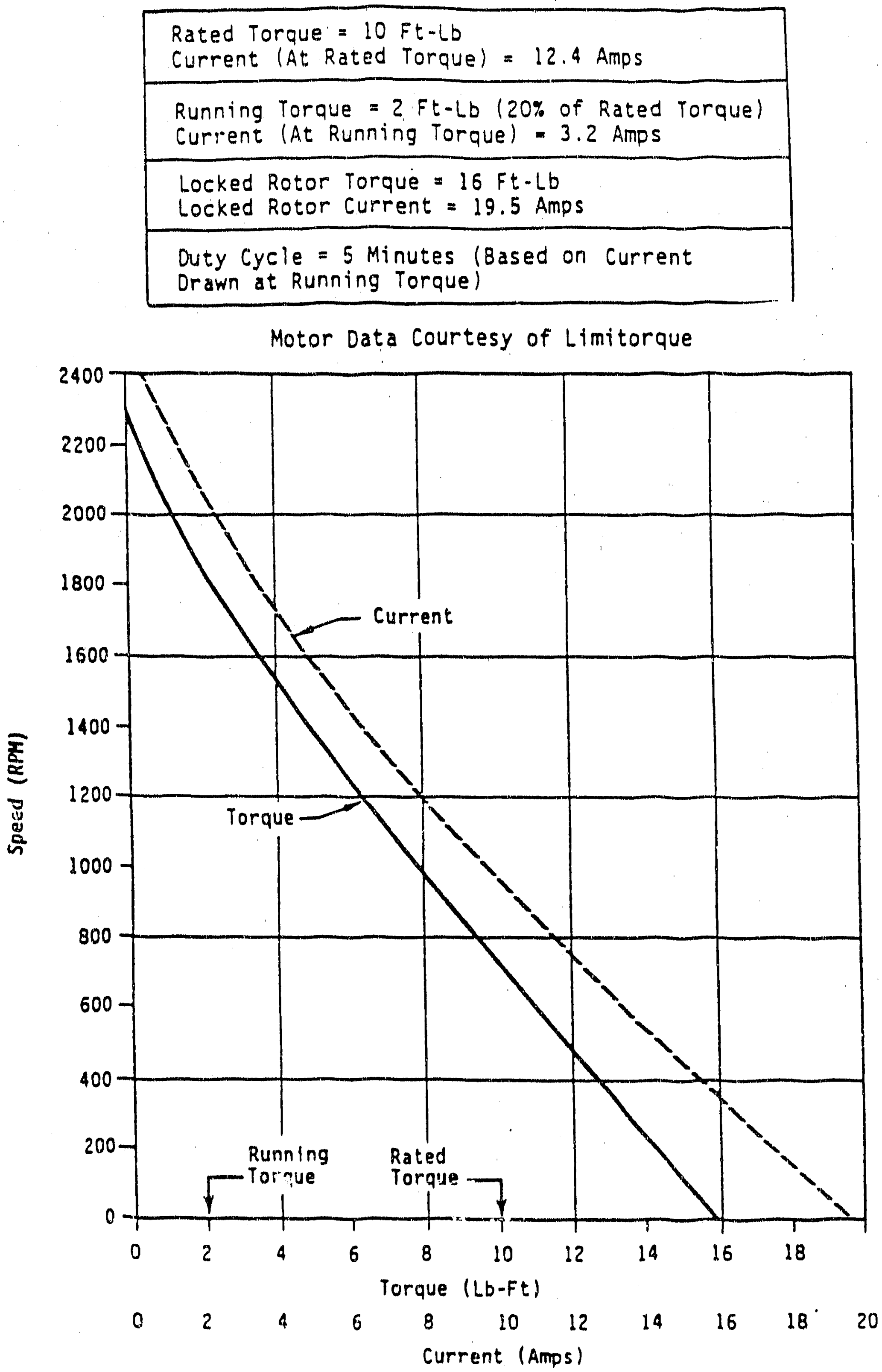

Figure 3. Speed vs torque and current for typical 250 V dc compound–wound motor.

Condition	Open: Stroke Time (s)	Open: Current (amps)	Closed: Stroke Time (s)	Closed: Current (amps)
	Test Results			
Packing installed	11.4	9	11.8	9.3
Bolts hand tight (≈ stem thrust 2400 lbs)				
Packing	11.5	9.7	11.8	10
Maximum tight (≈ stem thrust 5000 lbs				
Resulting change	0.9%	8%	0%	7.5%

AC Motor Current Signature Analysis

Based on the theoretical example and laboratory test, it appears that measuring motor current during valve operation would provide an improved indication of valve condition. By the use of a clamp on ampmeter and an analyzing recorder, motor current can be measured and recorded during the entire stroke event. The process is commonly called a motor current signature. Figure 4 shows a typical signature.

Motor current signatures are non–intrusive because they can be obtained from the motor control center of the valve and the required equipment to obtain the data would be available as part of normal plant test equipment. Extensive research and development activities are presently being pursued to improve methods for obtaining and analyzing motor current signatures.

DC Motor Operated Valves

Unlike ac motors, dc motor speed varies significantly as motor torque increases. Therefore, stroke timing of these valves may provide indication of the valve condition. The following example illustrates the effectiveness of stroke time measurement.

Theoretical Example. Between normal testing of a valve, the packing is tightened to correct a leak. The tightening of the packing results in an increase in the valve running torque from 2 to 4 ft–lbs. Based on Figure 3 this would result in a decrease of motor speed of approximately 230 rpm or 15%. This change would result in an equal percentage increase in valve stroke time and require corrective action to be performed according to OM–10, however, no action would be required by IWV.

In accordance with Figure 3, the same change in motor torque would result in an increase of motor current of approximately 2.2 amps or 59%. As shown in the example, stroke time changes can indicate degradation mechanisms resulting in a change in the valve's operating resistance. However, changes in motor current are much more sensitive to changes in motor torque. The measurement of motor current should, therefore, be considered as an alternative or supplemental test method.

CONCLUSION

The preceding evaluation resulted in the following conclusions:

- How does degradation really effect stroke time of motor operated valves?
  - For ac motor operated valves degradation has almost no effect on stroke time. However, dc motor operated valves degradation can be detected by stroke timing.

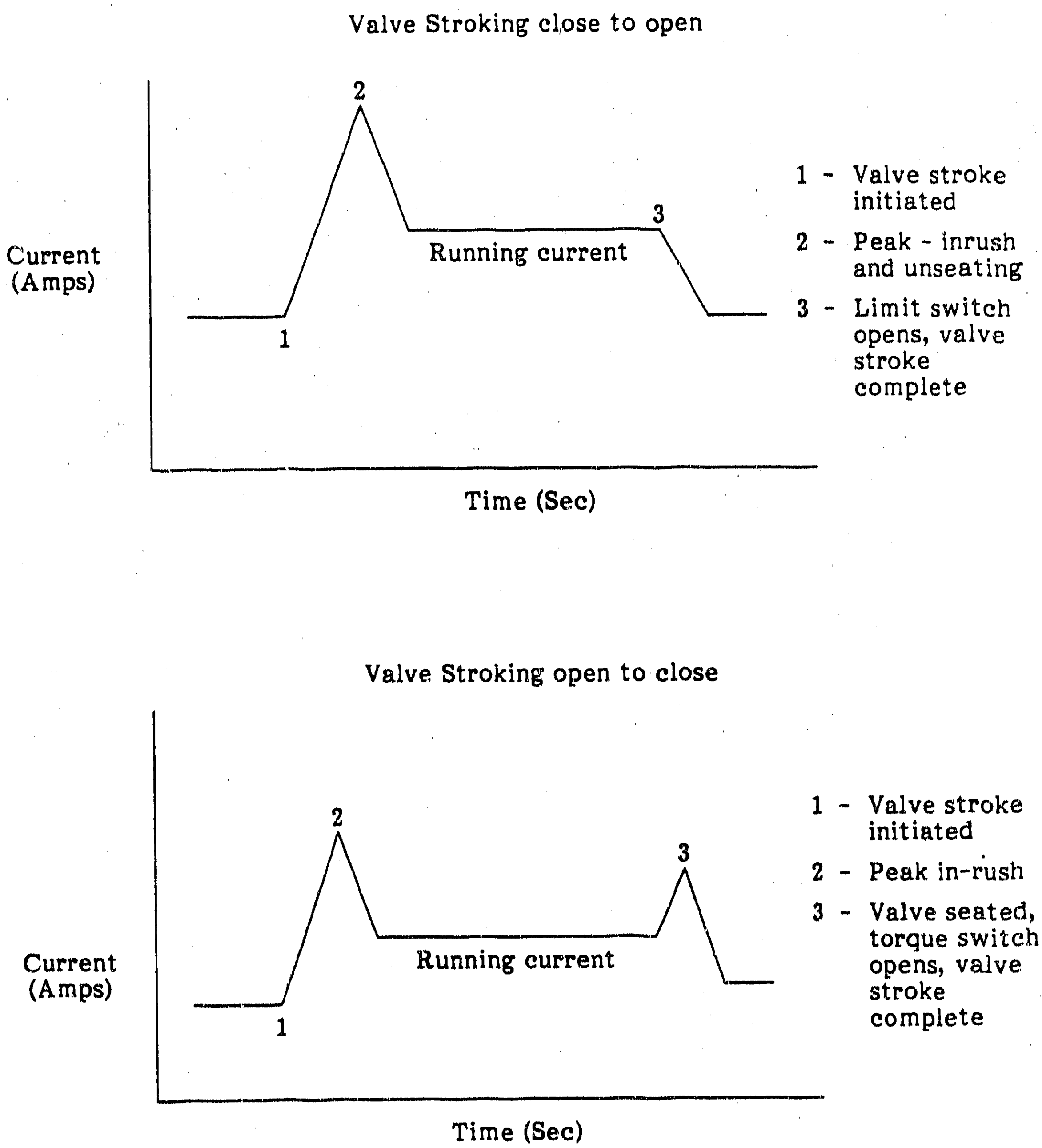

Figure 4. Simplified motor operated valve current signature.

- If stroke time is ineffective, is there some other parameter that can be used to monitor degradation?

  – Motor current signature analysis may be an early indicator of valve degradation for both ac and dc valves.

- Can testing of this other parameter be non–intrusive?

  – The performance of motor current signature analysis appears to be non–intrusive because it can be accomplished from the motor control center using normal plant test equipment.

- What about dc motor operated valves?

  – Because the speed of a dc motor changes as the motor torque changes, stroke timing does provide some indication of valve degradation. However, the measuring of motor current would provide an improved method of detection.

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ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Comment 1

I have a comment, rather than a question, on your dc measurements, on dc motor–driven valves. You are probably aware that voltage, as well as current, play a role in the speed of the dc motor. That is, a dc motor is not inherently a constant speed device, as is an ac motor within limits.

If you are going to use stroke time testing on dc motors, you will have to measure voltage as well as current draw, because the motor will speed up and slow down. If you do not account for that, you will get wild variations that have nothing to do with the conditions of the valve.

Presenter Yes, I understand. I agree with you on that. In fact, the testing we are doing is measuring stroke time, all we are doing is measuring stroke time, so it may be a problem. If our stroke time increases, like you said, it could be a fluctuation in our voltage.

Hopefully, in a nuclear plant, with it being on battery buses, the voltage is fairly constant throughout the stroke event.

Comment 2

Attendee A more sensitive parameter for ac–driven actuators is the actuator power, which is voltage times current times the phase angle between the two vectors. And it is possible to buy a chip which will do this conversion for you quite cheaply, and you get an even more sensitive measure of the true power going into the actuator, rather than simply the current.

Presenter What does it require to do that? Does it require installing something? Can it be something that is installed that can monitor this, or how is it done?

Attendee 1 I think you can still do it back at the junction box and not on the actuator.

You have got a three–phase wiring system. You have got a clip–on ammeter, which is looking at the current. You have another pair of wires looking at the voltage. This little chip is doing the vector multiplication of voltage times current times the cosine of the phase angle between them.

Comment 3

Attendee 1 We have been working on diagnostics for motor–operated valves for about four years, and we are about ready to issue a report that will be an NRC NUREG report. I think it will probably be out in the next couple of months. It describes all of the work we have done, both in identifying degradation and monitoring methods. And we did spend a lot of time on motor current signature analysis. It is as useful as you described. With regard to the issue of motor power or load rather than current, I certainly agree that voltage affects the result. If there is a voltage drop, then motor current will vary.

I want to note that in this NUREG report it will be noted that there is another way of identifying motor load other than the motor power indication, and that is the motor slip frequency. Using a spectrum analyzer, one can very easily measure the slip frequency of an ac motor. That slip frequency is as sensitive as VI cosine theta in identifying the load. In fact, it is the same. It is basically measuring the same thing.

Attendee 2 Except that it costs you $100 per chip and it costs you $1,000 for a signometer.

Attendee1 Well, I agree, as a capital investment. But let me note that the motor current slip frequency is measured truly non–intrusively. All it requires is a clip–on ammeter, whereas the chip must be installed on each valve and taken off or permanently installed, because you have to measure the voltage.

You have to put a bypass around, and you have to make a physical attachment, whereas the motor current probe is a clamp–on device or a current transformer, which is truly non–intrusive. But those are cost–benefits that have to be worked out.

Question 1

I do agree with the author's paper. Maybe the NRC and maybe the Code people could discuss this. We have already looked at pump bearing temperatures and decided they are not really all that good for what we want to accomplish. Maybe we should re–evaluate why, even in the OM's, we are requiring valve stroke time when we really do not need them. Maybe we should look at these newer techniques that have been developed, because it seems we are still down the road of sticking with old technically indefensible methods of measuring these parameters when the technology is leaving us behind.

The same goes for relief valve one percent parameters. Why are we sticking to one percent on 2500–pound valves that depend on a big spring to keep it closed?

Response 1

I think people involved with the O&M and the ASME Committees have to be sensitive because there are a lot of new technologies coming out. A lot of them have been tested and have been very successful, and some of them have identified more problems, or maybe there are problems in the methodology.

When you change a code or a standard, you are affecting a lot of plants, and you have to be very careful. They have to be very cautious on how they change that, because it is affecting the livelihood of many utilities, and the impact could be severe. We do not want to give them something that is not going to work again.

Comment 1

I am on the O&M Committee. I'm not on the OM–6 and 10 working group, but I would like to comment that they are very much interested in this new technology.

I know for a fact that the Oak Ridge signature analysis is under consideration. So I do not want you to think that the ASME is non–responsive to this information.

Question 2

The change you seem to be advocating on the stroke time to current sounds very, very good. Some of what we employ with MOVATS and other things could work, but has it been addressed how we go back to the FSAR and the tech spec time reaction requirements on those systems?

Response 2

No, it has not. But it is a good point. If you do some kind of a signature analysis, you do measure the time. It is measured electronically. From actual contact to contact, the time will be measured by the recorder. So you would not really be throwing away the time measurement. So therefore, you would still be able to take your requirements for tech spec surveillance or whatever to make sure a that valve does operate in its prescribed time.

Comment

I agree that stroke time verifies operability. We are reading into this degradation, as there are two separate things we are actually trying to accomplish simultaneously.

Now, if we at River Bend started trying to put current amp meters on all the valves at the motor control centers, we would have to have twice as many personnel hopscotching all over the plant to try to hook these things up, and it would add a considerable amount of time to our testing.

But if we implemented it as part of our maintenance, that could be done very easily. So if you start codifying these things, I would just like to caution you that there is operability, and there is also maintenance.

Comment

I think the point should be made that ac motor–operated valves cannot, in any way, meet the operability test of 25 percent.

There is no way that they could fail that test, theoretically and practically. They can not slow down 25 percent. They will either stall or they will operate. So your statement is correct, but you should be tempered by the reality of ac motors.

Comment

I notice that the curve you put on the overhead had a one percent change in speed going from synchronous to full load. In my experience, that is a much flatter curve than I normally see. I normally see a three to five percent slip from synchronous. So I think you probably chose the conservative side on the slip there.

But I think with a photoelectric timing device that is accurate to 0.1 percent changes, you could start to detect stroke time changes around one percent fairly accurately. And I think it is possible to use the stroke timing technique with better methods and enable degradation to be identified through that system.

USE OF A VALVE OPERATION TEST AND EVALUATION SYSTEM TO ENHANCE VALVE RELIABILITY

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ABSTRACT

Power plant owners have emphasized the need for assuring safe, reliable operation of valves. While most valves must simply open or close, the mechanisms involved can be quite complex. Motor operated valves (MOVs) must be properly adjusted to assure operability. Individual operator components determine the performance of the entire MOV. Failure in MOVs could cripple or shut down a unit. Thus, a complete valve program consisting of design reviews, operational testing, and preventive and predictive maintenance activities will enhance an owner's confidence level that his valves will operate as expected.

Liberty's Valve Operation Test and Evaluation System (VOTES®) accurately measures stem thrust without intruding on valve operation. Since mounting a strain gage to a valve stem is a desirable but impractical way of obtaining precise stem thrust, Liberty developed a method to obtain identical data by placing a strain gage sensor on the valve yoke. VOTES provides information which effectively eliminates costly, unscheduled downtime.

This paper presents the results of infield VOTES testing. The system's proven ability to identify and characterize actuator and valve performance is demonstrated. Specific topics of discussion include the ability of VOTES to ease a utility's IE Bulletin 85–03 concerns and conclusively diagnose MOV components. Data from static and differential pressure testing are presented. Technical, operational, and financial advantages resulting from VOTES technology are explored in detail.

BACKGROUND

Safe, reliable valves are important for successful operation of power plants. Initially, regulatory pressures prompted utilities to expand their valve programs. Today, however, maintenance staffs highly value all pertinent information on MOV performance.

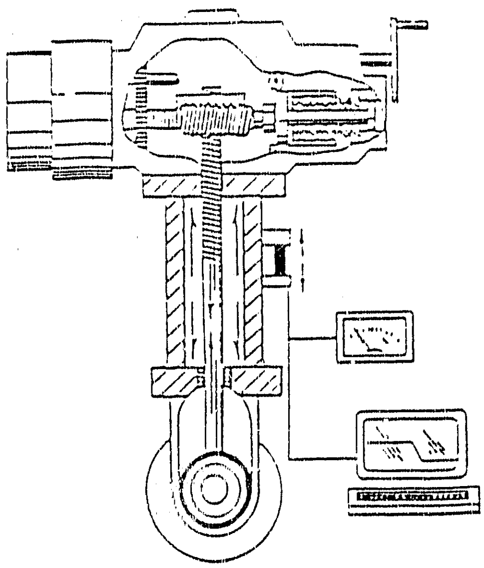
MOVs must be carefully adjusted to assure operability. Although manual operation is available as a back–up technique if the MOV does not open or close properly, effectiveness may be marginal in emergency conditions. Failures in MOVs may cripple or shut down a unit, causing costly, unscheduled downtime.

An appreciable amount of MOVs in nuclear power plants remain inactive for many months, yet must be operable in an instant under design basis conditions. Due to demands in certain fluid systems, some valves get stroked much more frequently. These latter valves, which control vital fluid flows on a daily basis, have gear wear patterns, spring pack deterioration, and stem taper problems that can occur at a rate greater than that for inactive valves. When valves in this category cannot operate, they have an immediate financial impact on the plant. Although system redundancy should allow safe fluid transfer, the chance exists that the sister valve will fail also, especially since it was designed, set–up, and maintained in a similar manner.

Moreover, a certain amount of valve testing is required due to the IE Bulletin 85–03. In order to complete these requirements quickly and accurately, VOTES captures crucial electrical or mechanical signatures from the MOV components. By providing conclusive diagnostic information through a simple and accurate MOV testing method, this system eliminates unscheduled downtime and maintenance expenditures.

THE VOTES® CONCEPT

Liberty VOTES to accurately measure stem thrust, without intruding on valve operation. A study by the Electric Power Research Institute (EPRI) determined that"... the use of strain gages attached to the stem would be the most desirable method for accurately monitoring the dynamic events..." within a MOV. Since mounting a stem gage is impractical, Liberty discovered that virtually identical data could be obtained through placing a strain gage device, Force Sensor, on the valve yoke. VOTES obtains the true stem force acting on the valve stem by measuring the equal and opposite reaction forces in the valve yoke (see Figure 1). This same force method for the stem–yoke combination applies independent of the actuator type: motor, air, or hydraulic.



GATE VALVE - SEATED

Figure 1.

The VOTES Force Sensor is calibrated by temporarily placing a National Bureau of Standards (NBS) traceable, diametral strain measuring device on the stem as the valve is closed. During closing, the stem "barrels" diametrally as it is compressed. This expansion is converted into stem thrust. Calibration is completed by relating stem thrust measurement to the electrical output from the Force Sensor (see Figure 2).

Importantly, Force Sensor calibration occurs over the same portion of the stem and on the same side of the stem and stem nut threads as utilized under normal closure. Tests have been performed under both static and differential pressure conditions to compare stem gage and Force Sensor (see Figure 3). The results indicate consistent capability of the Force Sensor to measure stem forces encountered during the entire opening or closing strokes.

CALIBRATION

DIAMETRAL STRAIN

↓

LONGITUDINAL STRAIN
(Poisson's Ratio)

↓

STRESS
(Young's Modulus)

↓

STEM THRUST
(Area)

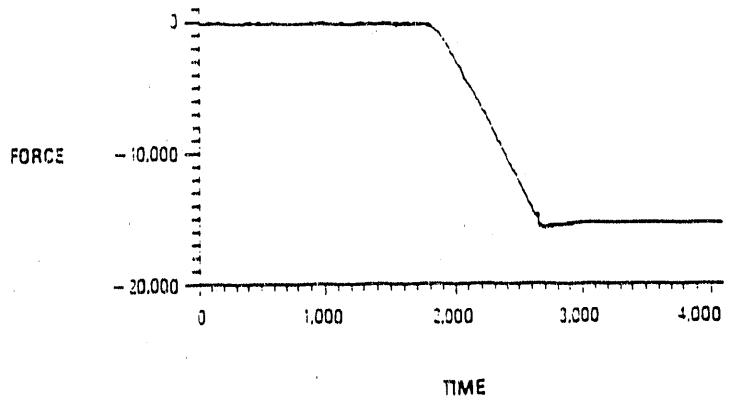
Figure 2.

The VOTES concept includes three primary components: signal conditioning box, breakout box, and portable computer as well as seven sensors: Force Sensor, four switch current probes, motor current probe, and accelerometer. The system gathers diagnostic data by sampling up to eight channels simultaneously, each at 1000 times per second. This rate permits a frequency response of 500 Hz to be maintained over the entire length of the stroke.

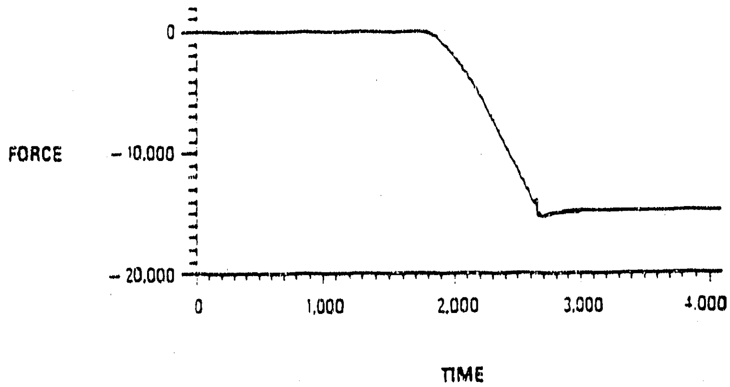
The breakout box consolidates the seven sensor cables into the main signal cable returning to the signal conditioning box. This feature permits the signal conditioning box and portable computer to be located in either a low dose or no dose area up to 500 ft away. Internally, sound–powered headphones allow the tester at the valve to communicate with the tester at the portable computer (see Figure 4).

IN–PLANT DIAGNOSTIC TESTING

VOTES® has been used by a number of utilities for multiple outage testing. The system has proven very effective in capturing the electrical and mechanical data required by maintenance engineers to characterize MOV performance.



Stem gauge.



VOTES Force Sensor.

Figure 3.

VOTES 100 SYSTEM

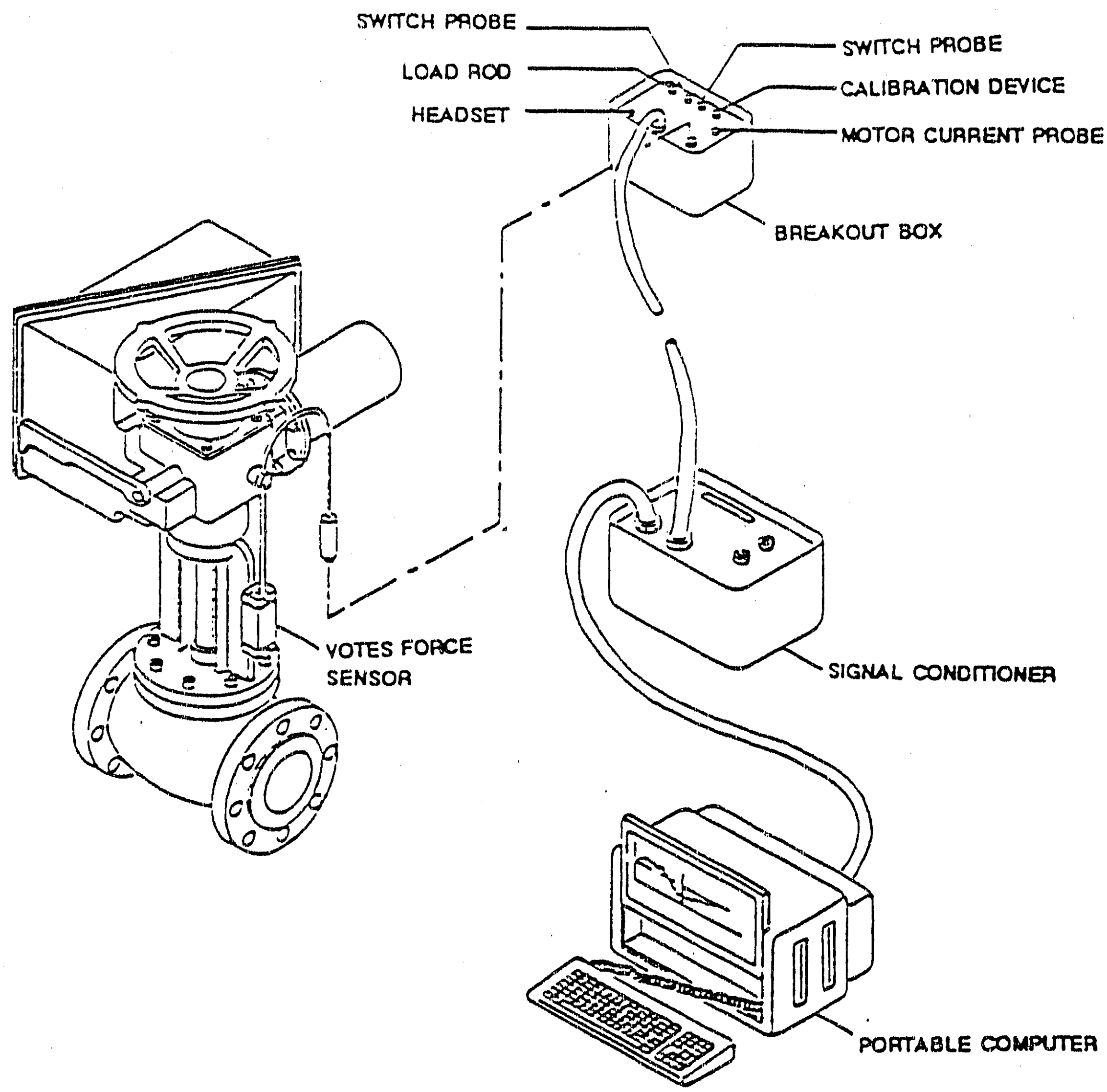


Figure 4.

Defining packing forces can be extremely important, especially for MOV's with small actuators. In these applications, improperly adjusted packing can erode the thrust available to overcome differential pressure to a level which inhibits valve closure or opening. With VOTES, these forces can be noted with striking clarity (see Figure 5).

In this VOTES Force Sensor signature of a 12 in. gate valve having a SMB 00 actuator, the packing force is given by cursor position as–1093 lb. The negative sign indicates a compressive force on the stem. This direct reading simplifies field procedures.

Packing forces also can be gained through use of a lightweight, portable meter (see Figure 6). Once a complete Force Sensor baseline test has been performed, a tester can merely attach the meter to the Force Sensor, then stroke the valve partially closed, then partially open from mid–position. A quick calculation allows the testing crew to know the increase or decrease of the packing adjustment in pounds.

As the industry has gathered more knowledge about the performance of aging MOV's under differential pressure, many questions have been raised. VOTES

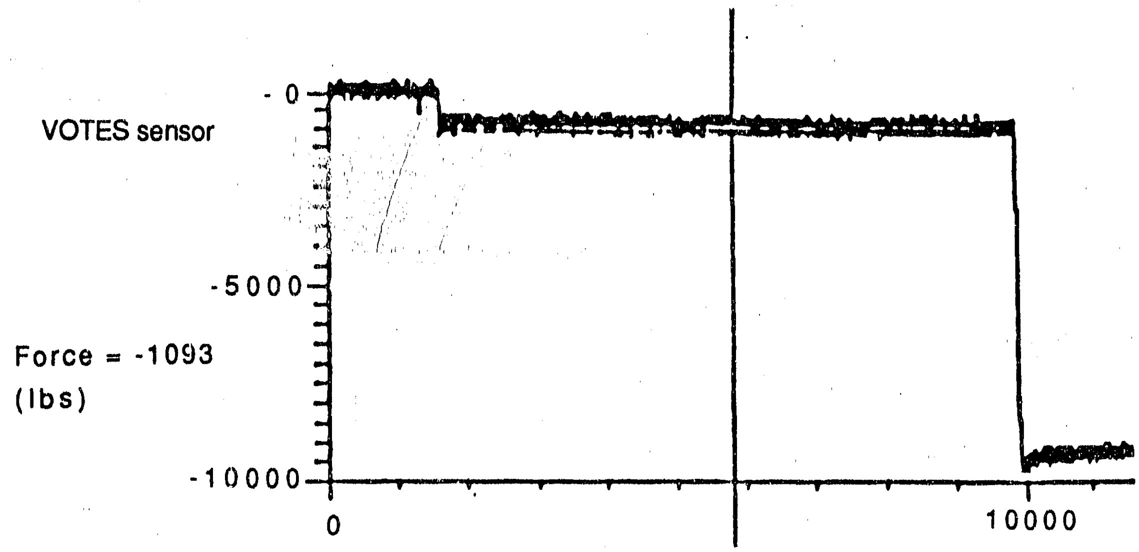


Figure 5.

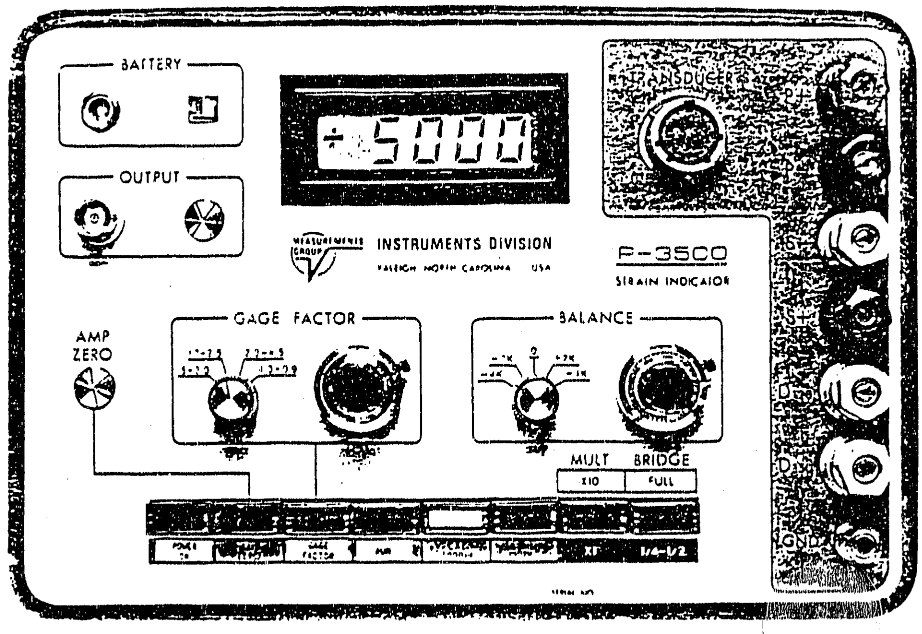


Figure 6.

gives utility owners an excellent tool for measuring how individual MOV designs and maintenance practices influence operability.

One critical quantity necessary for properly calculating the thrust to overcome differential pressure is the disc–to–seat ring friction factor, μ. Sometimes this term is referred to as the valve factor or disc factor. Many industry engineers believe that a number of external influences determine the actual friction factor such as: stellite–to–stellite disc–to–seat ring wear, levels of differential pressure applied to the MOV, the number of repeated valve closures or proximity of the MOV to flow obstructions (pumps, elbows, etc.).

With VOTES®, a tester can quickly quantify a valve's μ by stroking it under differential pressure. Under a valve closure, for example, a tester might expect to see a trace similar to one in Figure 7.

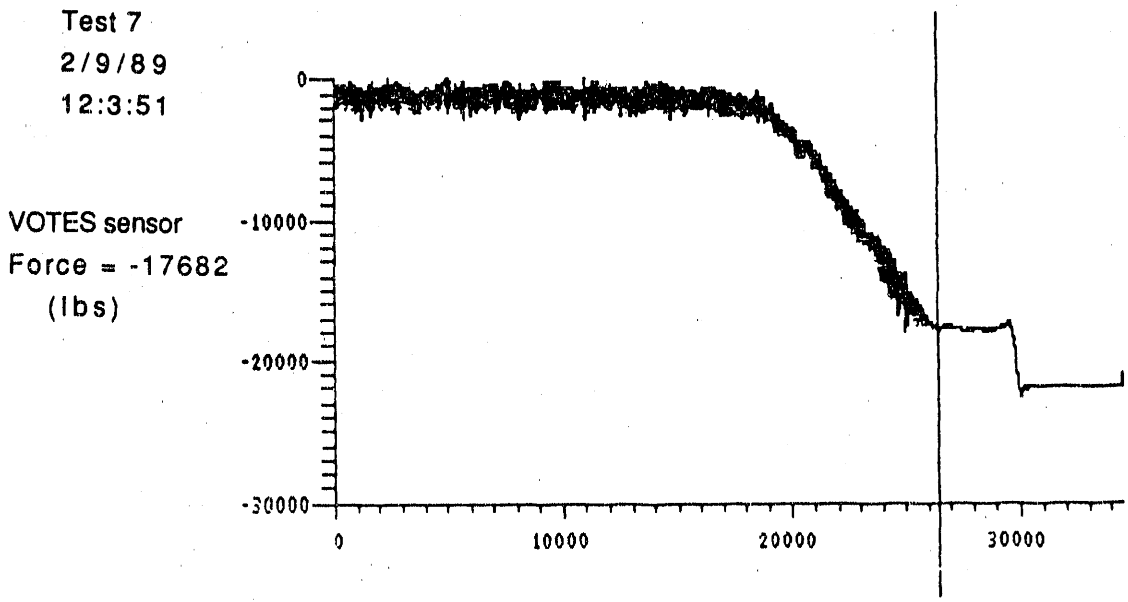


Figure 7.

The vertical oscillations in the trace are not signs of electrical noise, but demonstrate a useful indication of flow through the valve. As the tester zooms in to take a close look at stem thrust valves around the cursor placement, he is easily able to identify when flow stops to a millisecond (see Figure 8).

Utilizing the accepted formula for thrust required to close a valve, i.e.

$$T_c = \mu A \ \Delta P + A_s \ P + F_p \qquad (1)$$

where,

T_c	=	stem thrust at flow stoppage, read from Force Sensor trace
μ	=	friction `actor pipe
A	=	area of the seat ring opening, known
ΔP	=	differential pressure applied to the valve, measured
A_s	=	area of the stem, known
P	=	line pressure (generally upstream pressure), measured
F_p	=	packing force, read directly from trace

the tester can solve for a valve's μ, the only unknown in the equation.

$$\mu = \frac{T_c - F_p - A_s P}{A \Delta P} \qquad (2)$$

Values ranging from 0.15 to greater than 1.0 might be expected.

Liberty recommends multiplying the first term from Equation (1), $\mu A \Delta P$, by the ration of the design basis pressure to the test pressure to get the $\mu A \Delta P$ required for design basis differential pressure. Therefore, after subtracting the test pressure $\mu A \Delta P$, this increase in thrust must be added to the stem thrust at flow stoppage, T_c, to obtain the minimum required actuator thrust to close against full design basis ΔP. For instances, if

Test pressure	=	1200 psi
Design basis	=	1600 psi
A	=	3.14 in^2
ΔP	=	1200 psi
μ, calculated	=	0.4
T_c	=	3600 pounds

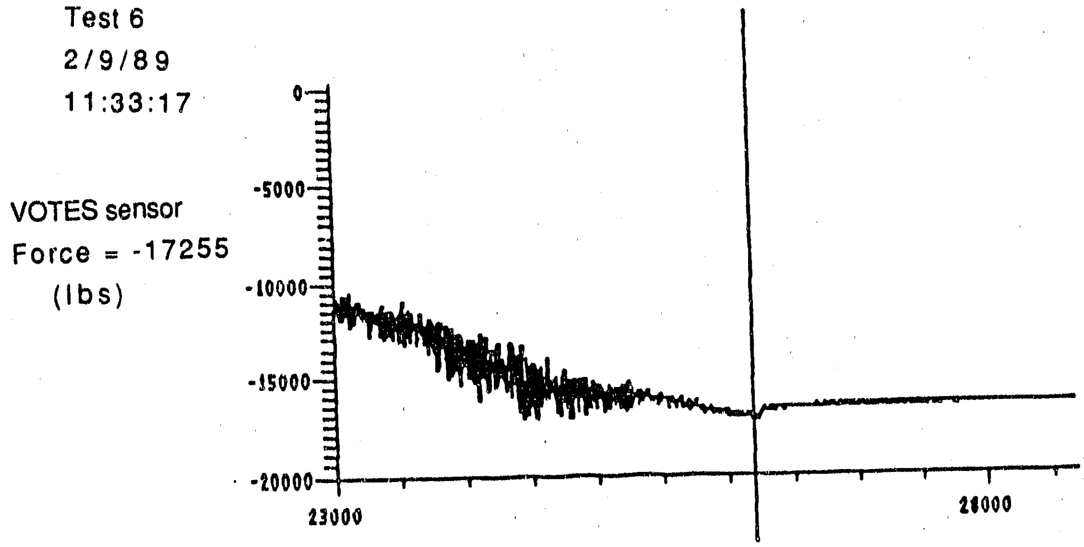


Figure 8.

then,

$$\mu A \ \Delta P = (0.4)(3.14)(1200)$$

$$= 1507 \text{ pounds}$$

therefore,

T required, design basis

$$= \left[\frac{\Delta\text{Pressure, design basis}}{\Delta\text{Pressure, test basis}} \right]$$

$$(\mu A \ \Delta P) - \mu \ A \ \Delta P + T_c$$

$$= [(1507) \ (1.33 - 1)] + 3600$$

$$= 4097 \text{ pounds}$$

The valve will close successfully under design basis conditions if the thrust at closed torque switch trip is greater than 4097 lb. If not, the torque switch setting would have to be increased.

Although much more testing is necessary, one major utility reports that the test pressure might have to be at least 60 to 70% of the design basis pressure to get stabilization of the friction factor. The industry's approach will certainly become refined upon additional testing.

VOTES diagnostics cover all aspects of the MOV. From closely defining packing forces to describing stem–related problems (taper, bending, etc.) to uncovering gearing maladies, VOTES® gives owners conclusive information about the health of their MOVs.

Babcock & Wilcox's Nuclear Power Division also has applied an approach which combines their torque switch and spring pack testing devices with VOTES' stem thrust information to find collapsed spring packs.

A typical example of VOTES' clear diagnostic power is shown (see Figure 9). The running loads in both the opening and closing traces reveal "smiles", indicating a bend in the stem. By moving the cursor across this area of the trace, a tester can identify the exact increase in stem thrust from the bend and decide whether to perform any maintenance.

VOTES® BENEFITS

VOTES® technology offers technical, operational, and financial advantages which reach far beyond first generation valve testing approaches.

The technical advances include greater accuracy and repeatability, better diagnostics, and a building block for future development. Previously, packing force or running load had to be estimated from a spring

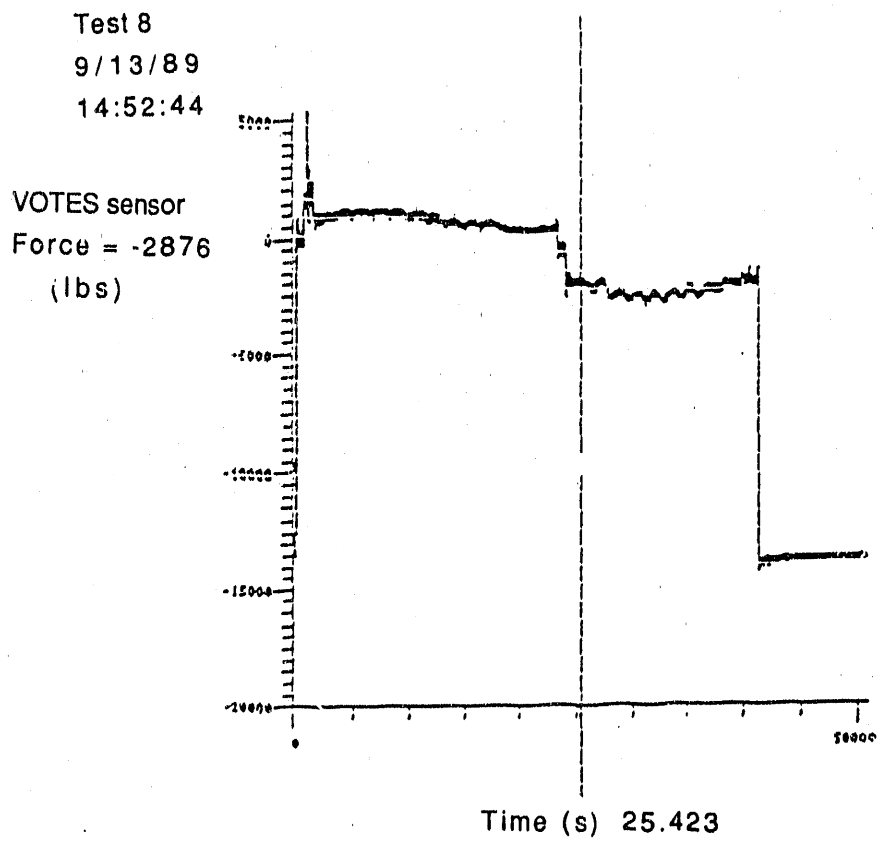


Figure 9.

pack motion trace. In many cases, spring pack preload prevented packing forces from being tested. MOV testers now have an easy, quantifiable way to define a term needed to calculate a valve's minimum required operating thrust. VOTES accuracy can be applied broadly to all operator/valve combinations.

VOTES was designed to provide a system capable of comprehensive trending and analysis. The permanent, non–intrusive Force Sensor allows a natural upgrade to an on–line monitoring system, particularly for such critical valves as those inside containment or those which present difficult testing conditions. Also, plant owners could ease outage testing requirements by capturing all pertinent signatures each time the valve is stroked.

Output from the VOTES Force Sensor is the cornerstone to organizing an effective predictive maintenance program for all plant valves. Through combining motor load and Force Sensor signatures, critical performance conclusions can be made for all combinations of actuators and yokes. As long as stem forces are transmitted to the yoke, correct data acquisition will occur. Operationally, VOTES offers tremendous speed in terms of data acquisition, analysis, and reporting. Field use indicates large reductions for manpower and man–hours compared to previous test methods. This feature will keep valve testing off of the critical path during future outages.

The technical and operational benefits have positive financial impacts on plant performance. Shorter test times mean significant radiation exposure savings. Labor costs drop for IE 85–03 testing, retesting, and post–maintenance verification. Remaining valve testing from the critical path allows for shorter outage schedules. MOV parts inventory levels can be minimized due to the improved component trending analyses.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

We have heard a number of times today about the disc factor normally being assumed to be 0.3. We heard 0.5 earlier. Now we are hearing up to 0.75. This is for wedge gate valves.

Has anybody really looked at other technology, such as parallel slide valves, where your VOTES system would still be able to pick up bent stems and the like? But as far as these high disc factors, they are not present with those type of valve systems.

Response 1

I can defer that to some people. I know that those particular valves are used in some applications, but I am probably not knowledgeable enough to tell you exactly why they are not used more broadly. There may be some increased complexity.

Comment

It would decrease the requirements of the motor operator tremendously, of course, if you did not have such a high factor.

Question 2

That is the second time I have seen today that your system meets the provisions of Generic Letter 89–10. The generic letter, in the very front, says that the Commission is concerned about current testing capabilities. I do not know that anybody knows how to address that generic letter. You have a good tool that will be helpful in addressing the letter, but it is a tool.

I have seen a diagnostic system used in the past. This whole thing depends on a comprehensive program and a technology or a system for using your tool.

Response 2

I agree totally. The comment has to be, that a diagnostic system is only a part of a programmatic approach by a utility to addressing the generic letter concerns, which of course involves design engineering, maintenance, and proper training of the personnel who are going to do the diagnostic testing. And certainly any diagnostic system, including Liberty's, cannot in and of itself satisfy the generic letter. It has to be a concerted effort by the utilities in order to complete that program.

Question 3

You did not say too much about the hardware connected to the valve harness. Is it a portable PC, or is it some sort of a wheel–around type of arrangement?

Response 3

Actually, the VOTES 100 system would be hooked up to a small, portable, breakout box, which weighs about three and a half pounds, as well as a signal conditioner portable computer. You make one connection to the connector that is coming out of the drain plug and that will return to the breakout box effectively. That is connected to the signal conditioner and the portable computer. You do not even have to take off the switch cover. You do not interfere at all with the operation or the operability of the valve during the test.

So you would hook up the system, arm the system effectively so it is ready to take data, and as soon as the valve is stroked, you would capture all that information again from the force sensor, from the switch current probes, and from the motor current probe.

Question 4

Could you please elaborate on the essential environmental data on the instrumentation? And how reliable are the strain gauges or the attached equipment on the valve, as far as maintenance activities being done on the valve, paint, grease, and that sort of thing.

Response 4

All the materials used inside the sensor have been tested for all those conditions that you just mentioned in terms of radiation, and high temperature.

The sensor is rated basically for 250 degrees long–term and up to 400–degree transients. Duke Power has a few hundred of these units installed at the various stations right now. And we have not seen any significant problems with maintenance interfering with the sensors themselves or any other extraneous material, such as shavings, oils, etc.

A COMPUTERIZED DATA BASE MANAGEMENT PROGRAM FOR INSERVICE TESTING OF PUMPS AND VALVES

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INTRODUCTION

This paper presents a description of an ideal computerized data base management program designed to control multiple plant and unit Inservice Testing (IST) activities of ASME Section XI. The program is intended to relieve personnel of the necessity to manually maintain inservice testing data in support of technical requirements of Subsections IWP and IWV of the ASME Boiler and Pressure Vessel Code, Section XI, "Rules for Inservice Testing of Nuclear Power Plant Components," Technical Specifications and other references.

The philosophy of the program is to utilize basic computer data base management techniques to coordinate inservice testing data, create a methodology for quick and easy input and output routines, and provide computerized test data storage and organization, component tracking and retrieval, report generation, trending and summarized test results.

From experience at a single unit BWR plant, it was found that the manual system of data management of inservice testing data may be cumbersome due to the large volume of information that has to be collected and recorded, thus wasteful of valuable time.

During daily input of inservice testing data, the appropriate test data chart has to be located in the main filing system, the new test data hand plotted, new limits calculated and evaluated, and possibly a new chart drawn due to (end of page reached, new limits required, change of surveillance, etc.), and the old chart sent to the main file for storage purposes.

The legitimate intent of any inservice testing information storage program is to use the data stored to identify and correct potential problems rather than react to component, and possibly plant failures. Given the nature of the current system, a computerized inservice testing data base management program would help to enhance the manual system which may expend vast resources on data management.

PROGRAM STRUCTURE

Figure 1 identifies a general structure of the software, and provides a list of primary user menus necessary for program function operation.

These menus provide user interface to the data fields for data entry, data maintenance and browse operations. They also provide the user with options for test result input and logging, component trending, data reporting, data queries, plotting and printing of charts and graphs, word processing, general utility features and program setup and security options.

DATA BASE MANAGEMENT PROGRAM

In brief, the program performs the following functions:

1. Maintains a component "file" for each component which must be tested according to the requirements of the ASME Code or other reference. The file contains a basic record with specific information and data pertaining to the component. The following is a list of possible trend summaries:

   Valve Trend Summaries

   - Full stroke exercise test to the open position

Plan Test Report Query pLot Utility Setup

ENTER/EDIT Pump/Valve/Other Component Info
EDIT Pump/Valve/Other Test Info
ENTER/EDIT Pump/Valve/Other Maintenance Info

INPUT Surveillance Results
LOG Surveillance Results
INPUT & LOG Individual Tests
DEFINE Surveillances

Pump/Valve/Other Tables Report
Pump/Valve/Other Trends Report
Pump/Valve/Other Status Report

Pump/Valve/Other Information Query
Pump/Valve/Other Exam Query
Pump/Valve/Other Test Query
Pump/Valve Other Maintenance Query

PLOT Pump/Valve/Other Trends
BATCH PLOT Pump/Valve/Other Trends
PLOT Pump Reference Curves
BATCH PLOT By Surveillance

BACKUP Mod. Data To Tape/Disk
BACKUP All Data To Tape/Disk
RUN ED (Text Editor)

SET Access Code
CUSTOMIZE
DEFINE Exam Codes
EDIT Plant Codes

Figure 1. Software organization logic.

- Full stroke exercise test to the closed position

- Containment isolation valve leak test

- Partial stroke exercise test

- Check valve exercise test to the open position

- Check valve exercise test to the closed position

Pump Trend Summaries

- Flow rate

- Differential pressure

- Discharge pressure

- Horizontal vibration amplitude

- Vertical vibration amplitude

- Horizontal vibration velocity

- Vertical vibration velocity

- Motor speed

- Motor amperes

- Inlet pressure (running, static)

- Bearing temperature (upper,lower)

- Pressure/Flow

2. Maintains test files containing a record for each required test performed on a component. The file contains the type of test being performed, and the frequency of testing.

3. Maintains files containing a record of each maintenance operation performed on a component.

4. Maintains a file for each surveillance in the program. Surveillances are used to simplify the entry of inservice test data. A surveillance can consist of a mixture of any number of tests.

5. Supports entry of inservice and reference tests for pumps, inservice tests for valves and other safety related test. Tests can be entered individually or through a batch process where multiple tests are entered for all components.

6. Generates pre-formatted and query-type standard reports based on the general information in the main component files. These reports include a list of:

- Component system inspection requirements of ASME Section XI,

- Test data trends showing performance of all components over a specified period of time.

- The current status of all unacceptable components in the Inservice Testing Program.

7. Generates graphical inservice test trend plots and reference curve plots from the test results data entered in the component files.

THE MENU SYSTEM

Each of the menus group functional activities in the program. A description of the types of functions each menu holds is described below.

The Plan Menu

The Plan Menu functions maintain component information in the program. This section provides the specific data pertaining to the component that is necessary for submittal to the NRC. The Plan Menu documents the basic information on all components included in the Inservice Testing Program, aids in response to NRC questions, and provides information for diagnosing plant problems quickly.

The Test Menu

One of the main purposes of the program is to keep track of inservice and reference tests for pumps and inservice tests for valves. The program uses a "batch" method of in-putting tests, that is, inservice tests are entered and collected on an input file screen, and then "logged", as a group. This section verifies the occurrence of an unacceptable or inoperable component, where, those components whose test results have exceeded the alert or action required ranges.

The Report Menu

The Report Menu has functions for displaying and generating multi-formatted reports based on the component information stored in the main component files. The report functions each have options for selecting various formats and for selecting reports containing only those records which match a set of selection criteria.

The Query Menu

The Query Menu has functions for printing or displaying query type or impromptu reports based on selected information stored in the main component files which match certain criteria. The query option interactively queries the file and directs the results to the printer, computer screen, or a user specified file. The program groups queries according to the file from which the information is drawn.

The Plot Menu

The program is capable of producing several types of plots: a trend plot of a sequence of component inservice tests, and a pump Pressure/Flow reference curve. The plots can be displayed on the screen, or generated on an external plotter and/or printer.

The Utility Menu

The Utility Menu contains functions for maintaining the various files in the program. Backing up the files can save all of the data included in the files or the most recent data entered. Regular backing up of files reduces data loss problems.

The Setup Menu

The Setup Menu contains functions to set up program and screen entry codes, customize information that tailors the program to the particular computer environment, define test types and/or change any of the labels on reports in the program, create multiple unit codes to support more than one unit at a time.

COMPONENT TESTING

The program provides data fields for tracking and maintaining specific Class 1, 2, and 3 pump and motor information, valve and actuator information and other safety related components. For each pump component, the Section XI allowable ranges of test quantities are maintained in the program. For each valve component, the Section XI requirements for allowable leakage rates, stroke time and motor operator information are maintained in the program. Component performance data is maintained for Acceptable, Alert and Action ranges for required examinations.

INSERVICE TEST ENTRY

The automatic batch entry and individual entry of inservice tests for pumps, valves and other safety related components is an important part of the program. The automatic batch entry process provides full screen direct input where tests can be entered quickly and accurately in a tabular form as shown in Figure 2.

Batch test data entry is used for two reasons:

1. It allows you to go back and double check (and correct, if necessary) each test entry for accuracy before it is logged.

2. It cuts down on the time it takes to enter tests (extra key stokes). You don't have to wait for a test to be logged before entering the next one. Once started, the logging process is automatic, which means that you can go about your business while the computer does the work.

During the entry and logging process, all necessary test information records are entered and logged, and the affected exam records are updated with new test and status information. During this process, the program compares the test value to the related Section XI acceptance criteria that has been previously set up in an exam records file.

The results are compared and the program generates a status report indicating whether the current test value is acceptable, in the alert range, or in the action range. This report can be reviewed by the IST Engineer to verify occurrence of an unacceptable or inoperable component.

GRAPHICAL OUTPUTS

The routines for generating a trend plot of a sequence of inservice tests for a particular component and a pump Pressure/Flow reference curve are available in the program. The type of graphics available to the user include a screen, plotter or printer display.

The inservice test trend plots allow trends to be established for a series of tests showing component performance over a specified period of time, and include a prediction of when component performance will become unacceptable.

Surveillance # 100-100			SURVEILLANCE INPUT	
Component: VALVE-1				
EXAM	MEASUREMENT 1	MEASUREMENT 2	DATE	R
LEAK-1	Leakage... 0.000		06/22/89	N
STROKE-1	Stroke Time. 0.000		06/22/89	N
CHECK-1	Pass/Fail P		06/22/89	

Figure 2. Batch test data entry example.

Figure 3 represents an example of a valve stroke test trend of the last 9 tests performed on the valve component.

During performance testing of pumps, flow rate and deferential pressure are two important parameters which are measured to support trending of pump performance. From the reference measurements of flow rate and differential pressure, a best fit flow curve plot can be generated to provide a graphical representation between the two.

Figure 4 represents an example of a pump Pressure/ Flow curve for a pump component.

SUMMARY

Because the daily management of inservice testing data is a major part of the broad effort to assure operational readiness of safety related components, the development of an easy to use computerized data base management program may provide information and techniques to assure component operability. A program with these useful operations may help to quickly detect problems affecting component performance and assess whether adequate margins are being maintained.

The basis for the full featured computerized data base management program is to accommodate the following basic activities:

1. User interface to major program functions
2. Storage of all safety related component information with controlled accuracy and security
3. Full screen data entry, maintenance and browse operations
4. Individual and batch entry of component inservice testing data
5. Component test data tracking and manipulation
6. Operational status verification of components
7. Pre-formatted data reporting for IST tracking
8. Query reporting for miscellaneous reporting
9. Graphical representations for performance trending.

REFERENCES

1. ASME Boiler and Pressure Vessel Code, Section XI, Subsections IWP & IWV.
2. NRC Generic letter 89-04, Guidance on Developing Acceptable Inservice Testing Programs.

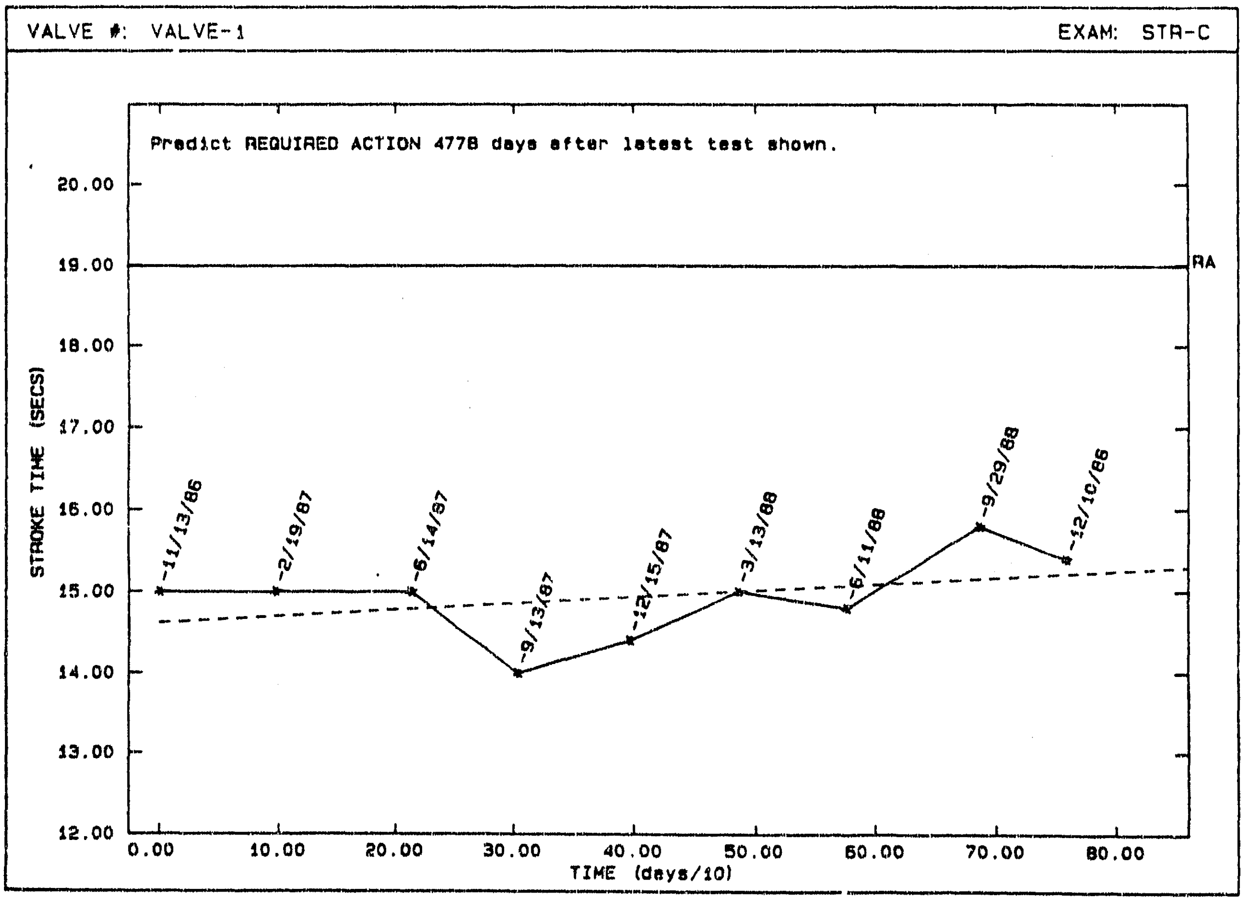

Figure 3. Example of valve stroke trend, last nine tests.

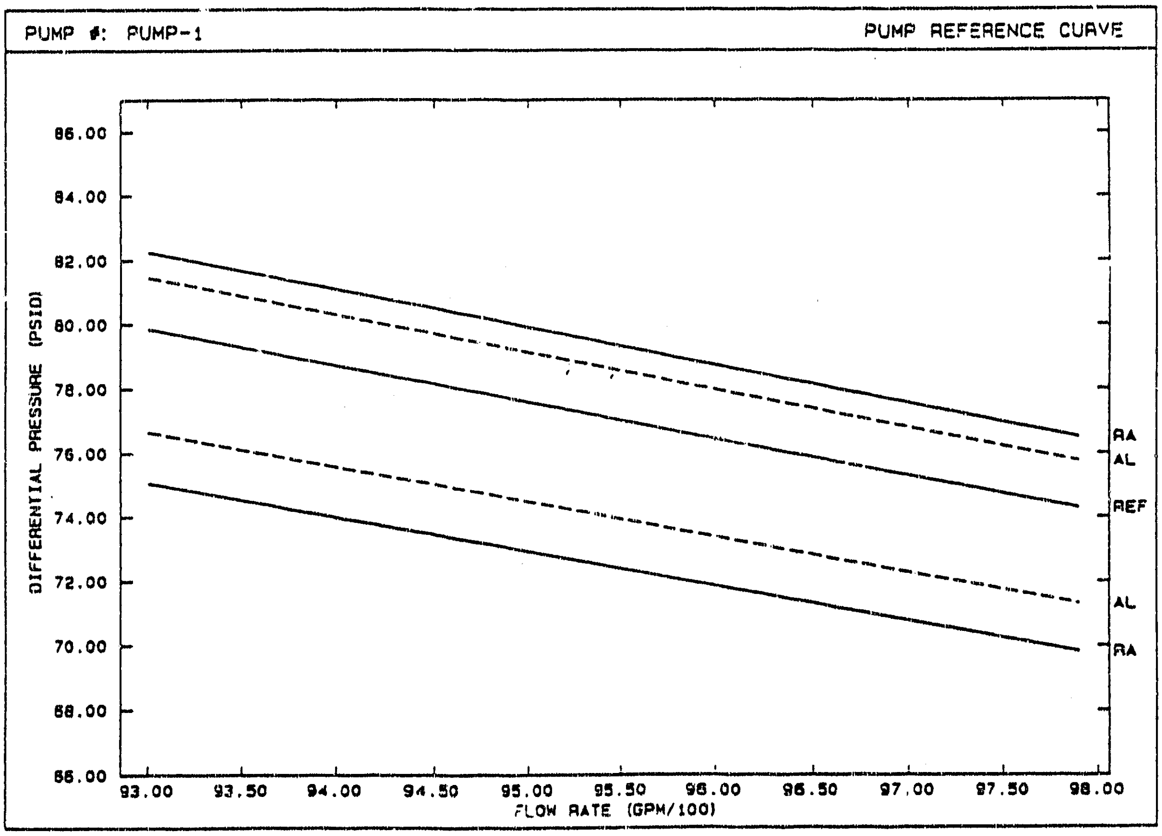

Figure 4. Example of pump pressure/flow curve.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

How do you set your reference values, your alert ranges, your action ranges, and so forth, on your trending output graphs? Is that menu–driven or is it automatically incorporated in the program? How do you do that?

Response 1

In the examination section of the component record, you are able go in and manually put your reference values in for alert and action ranges.

Question 2

How is your data entered? Do you record it in the field and then come back and enter it?

Response 2

It is taken from the field and brought back to the computer and entered directly.

AUTOMATED IST PROGRAM

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ABSTRACT

There are two methods used to manage a Section XI program: Manual and Automated. The manual method usually consists of hand written records of test results and scheduling requirements. This method while initially lower in cost, results in problems later on in the life of a plant as data continues to accumulate. Automation allows instant access to forty years of test results. Due to the lower cost and higher performance of todays' personal computers, an automated method via a computer program provides an excellent method for managing the vast amount of data that accumulates over the forty year life of a plant. Through the use of a computer, special functions involving this data are available, which through a manual method would not be practical. This paper will describe some of the advantages in using a computer program to manage the Section XI IST program.

The ISTBASE consists of program code and numerous databases. The source code is written and complied in CLIPPER (tm) language. Graphing routines are performed by dGE (tm) graphics library. Graphs are displayed in EGA form. Since it was estimated that the total complied code would exceed 640K of ram, overlays through the use of modular programming were used to facilitate the DOS restrictions of 640K ram. The use of overlays still require the user to gain access to ISTBASE through the PASSWORD module. The database files are designed to be compatible with dBASE III+ (tm) data structure. This allows transfer of data between ISTBASE and other database managers/applications. A math co-processor is utilized to speed up calculations on graphs and other mathematical calculations. Program code and data files require a hard disk drive with at least 28 Meg capacity. While ISTBASE will execute on a 8088 based computer, an 80286 computer with a 12 MHz operating speed should be considered the minimum system configuration. ISTBASE supports dot-matrix printers, laser printers, and plotters.

Entry into the program begins with the user selecting the path telling the computer where the data files are stored. Next, the user must enter a valid password to proceed any further.

PASSWORD MODULE

ISTBASE has two levels of password protection. An initial password is required to gain access to the program. With this password, the user can browse, query, and run canned reports from the databases. Any changes, deletions, or additions require a second password along with an account number. This password/account number combination is unique. Up to four users, other than the master level operator, can have access to the program. A master level password/account number controls the passwords/account numbers of the other four users. After entry into ISTBASE and correctly entering the browse password, the user can enter the access control module. Entry into this module is only accessible to the master level operator through entry of the master level password and account number. This module will display the other four users, passwords, and account numbers along with the master level operators information. The master level operator can modify the other four users passwords and account numbers as well as his own. If for some reason, the master level password/account number is forgotten, a 'back door' password and account number is available to reset the master level operator password/account number to some known value. The 'back door' password/account number becomes invalid after use by scrambling the password/account number by a known function.

General Valve Data

General valve data consists of the following fields:

VALDATA.dbf

	Name	Type	Length	Comments
1.	VALVEID	CHAR	15	Valve id number
2.	CLASS	CHAR	1	Class of valve
3.	DWGNO	CHAR	15	Drawing number
4.	DWGCOOR	CHAR	5	Drawing coordinate of valve
5.	CAT	CHAR	3	Category of valve
6.	SIZE	NUM	6.2	Internal size of valve
7.	TYPE	CHAR	3	Type of valve
8.	SYSTEM	CHAR	5	System where valve is located
9.	DISS	LOGICAL	1	Valve requires disassembly
10.	LLRT	LOGICAL	1	Valve requires 11rt
11.	PENET	CHAR	8	Penetration for 11rt valves
12.	COLD	LOGICAL	1	Cold shutdown test required
13.	REFUEL	LOGICAL	1	Refueling test required
14.	MODES1	CHAR	1	Valve test in plant mode 1
15.	MODES2	CHAR	1	Valve test in plant mode 2
16.	MODES3	CHAR	1	Valve test in plant mode 3
17.	MODES4	CHAR	1	Valve test in plant mode 4
18.	MODES5	CHAR	1	Valve test in plant mode 5
19.	MODES6	CHAR	1	Valve test in plant mode 6
20.	COMMENTS	CHAR	37	General comments
21.	ACT1	CHAR	2	Valve actuator type 1
22.	ACT2	CHAR	2	Valve actuator type 2
23.	STDIR1	CHAR	1	Valve stroke direction 1
24.	STDIR2	CHAR	1	Valve stroke direction 2
25.	STTIM1	CHAR	1	Valve timing direction 1
26.	STTIM2	CHAR	1	Valve timing direction 2
27.	CKDIR1	CHAR	2	Check valve stroke direction 1
28.	CKDIR2	CHAR	2	Check valve stroke direction 2
29.	NORMPOS	CHAR	2	Normal position of valve
30.	ACCMODE	CHAR	1	Active/passive identifier
31.	TEST1	CHAR	5	Code required test 1
32.	TEST2	CHAR	5	Code required test 2
33.	TEST3	CHAR	5	Code required test 3
34.	TEST4	CHAR	5	Code required test 4
35.	TEST5	CHAR	5	Code required test 5
36.	RR1	CHAR	6	Relief request 1
37.	RR2	CHAR	6	Relief request 2
38.	RR3	CHAR	6	Relief request 3
39.	RR4	CHAR	6	Relief request 4
40.	RR5	CHAR	6	Relief request 5
41.	ALTEST1	CHAR	5	Alternate test 1
42.	ALTEST2	CHAR	5	Alternate test 2
43.	ALTEST3	CHAR	5	Alternate test 3
44.	ALTEST4	CHAR	5	Alternate test 4
45.	ALTEST5	CHAR	5	Alternate test 5
46.	SURTEST1	CHAR	8	Surveillance test 1
47.	SURTEST2	CHAR	8	Surveillance test 2
48.	SURTEST3	CHAR	8	Surveillance test 3

Name		Type	Length	Comments
49.	SURTEST4	CHAR	8	Surveillance test 4
50.	SURTEST5	CHAR	8	Surveillance test 5
51.	SURTEST6	CHAR	8	Surveillance test 6
52.	SURTEST7	CHAR	8	Surveillance test 7
53.	SURTEST8	CHAR	8	Surveillance test 8
54.	SURTEST9	CHAR	8	Surveillance test 9
55.	SURTEST10	CHAR	8	Surveillance test 10
56.	OTVAL	NUM	7.2	Baseline open stroke time
57.	OTDATE	DATE	8	Baseline open test date
58.	OSURT	CHAR	8	Baseline open surv test
59.	OALERT	NUM	7.2	Baseline open alert value
60.	OMAX	NUM	7.2	Baseline open max value
61.	CTVAL	NUM	7.2	Baseline close stroke time
62.	CTDATE	DATE	8	Baseline close test date
63.	CSURT	CHAR	8	Baseline close surv test
64.	CALERT	NUM	7.2	Baseline close alert value
65.	CMAX	NUM	7.2	Baseline close max value
66.	LTVAL	NUM	9.4	Baseline 11rt leakage
67.	LTDATE	DATE	8	Baseline 11rt test date
68.	LSURT	CHAR	8	Baseline 11rt surv test
69.	LALERT	NUM	9.4	Baseline 11rt alert value
70.	LMAX	NUM	9.4	Baseline 11rt max value

When entering data fields such as class, category, actuator types, etc. pop-up windows display valid responses. Restricting input in this fashion ensures only valid data is loaded into the databases. Baseline data is entered for open/close stroke times as well as LLRT leakage. Space is provided for up to five required code tests, five relief requests, five alternate tests, and ten implementing surveillance tests. An example of this section would be as follows:

XI Test	Relief Request	ALT Test	Surveillance Tests
TSO–1	RR–1	TS0–3	ST–1043

TS: Stroke time valve that has a tech spec limit on max stroke time.

0: Time the valve from close to open position

1: perform test every 92 days

RR–1: Relief request listed in program plan

TSO–3: Perform this test every 18 months (refueling)

ST–1043: Surveillance that test the valve at refueling

Valve Test Data

Valve test database consists of the following fields:

VTEST.dbf

Name		Type	Length	Comments
1.	VALVEID	CHAR	15	Valve id number
2.	TESTTYPE	CHAR	2	Test type
3.	STROKEDIR	CHAR	1	Direction valve was stroked
4.	TESTDATE	DATE	8	Date when test was performed
5.	SURVTEST	CHAR	8	Surveillance test number
6.	PLANTMODE	CHAR	1	Plant code when tested
7.	TESTVAL	NUM	9.4	Test value

Name	Type	Length	Comments
8. TALERT	NUM	9.4	Current alert value
9. TMAX	NUM	9.4	Current maximum value

Valve test data is input by first typing in a valid valve id. Test type along with stroke direction is then entered. This will determine which baseline data is displayed. The data displayed consists of the alert, maximum, calculated increase test frequency, and last test value. Test date, surveillance test procedure number, and plant mode are then entered. When the test value is entered, a check is made to see if the value meets or exceeds the alert, max or increase test frequency values. If this occurs, a warning is displayed at the bottom of the screen.

CALCULATION OF INCREASE TEST FREQUENCY VALUES

There are two formulas for calculating the increase test frequency (ITF) value: the first is a 25% or greater increase in stroke time for power operated valves with baseline stroke times greater than 10 seconds. The second is a 50% or greater increase in stroke time for power operated valves with baseline stroke times equal to or less than 10 seconds. In order for the computer program to calculate the ITF value several items must be known. The first is whether the valve is being stroked open or closed since different baseline values can exist for each stroke direction. The second item required is the last stroke time (open or closed) for the valve. The third and final item required is the current baseline stroke time value. With these three values, the ITF value can be calculated. Since the calculation of this value is automatic, the user is relieved of the task of retrieving the last test value and baseline data for a valve, and manual calculation of the ITF value for comparison with the current test value in question. The computer will always check current test data that is being input against the ITF, alert, and max values.

AUTOMATIC ENTRY OF TEST RESULTS

Entry of test data requires the following information:

Valve id

Test value (seconds, secs)

Test date

Test type (full stroke exercise, stroke time test, etc.)

Plant mode

Stroke direction

Existing alert value

Existing maximum value

Surveillance test where valve was tested.

A completed surveillance test usually contains ten to thirty valves that have been tested. Inputting items 1–9 for each valve allows for data input errors as well as a chance of skipping a valve all together. One way to prevent these errors is to create a datafile that contains the following information:

1. Valve id
2. Test type
3. Stroke direction.

Valve ids are input into the datafile in the same order that they appear in the surveillance test procedure. ISTBASE will create this datafile. The program allows full editing capabilities including deletions of records. A window displays data already input into the datafile. The user inputs the valve id, test type, and stroke direction as they appear in the surveillance test procedure.

When the data from a completed test procedure is ready to be input into the computer, the user enters the surveillance test number. After verifying the existence of that particular datafile, plant mode and test date are input. The program then calls up each valve, test type and stroke direction as it appears in the test data file. Baseline information and valve type for each valve along with the calculated increase test frequency value is displayed. The user then enters the test value, or ENTER for test types such as check valve forward flow, etc. Inputting an 'S' will skip a valve, 'X' will exit the screen. A check is made of the entered value to see if any limits are met or exceeded. If limits are exceeded, a warning is displayed at the bottom of the screen that forces the user to pause to identify the problem. The data is then stored in the test data file. Autosatic entry of test data along with computer checking of ITF, alert and maximum values prevents some of the errors associated with data entry.

The surveillance valve list structure is listed as follows:

Surveillance Valve List.dbf **Valve Sequence List**

	Name	Type	Length	Comments
1.	SEQNUM	NUM	6	Sequence number
2.	VALVEID	CHAR	15	Valve id number
3.	TESTTYPE	CHAR	2	Required test type
4.	STROKEDIR	CHAR	1	Required stroke direction

VALVE GRAPHING

Graphing of valve test data requires a computer system with EGA graphics capability. EGA graphics allows up to 16 colors to be displayed at one time along with 648 points horizontal by 358 points vertical. Graphing valve test data begins by the user entering a single valve id, range of valves, or surveillance test procedure. Selecting a single valve will draw the graph of one valve, while selecting a range will graph multiple valves, one at a time. Selecting a surveillance test procedure will graph the valves that are tested in a specified surveillance test procedure. The program locates and displays the first and last test date for a particular valve id. The present alert and maximum test values are displayed for open stroke times, close stroke times, as well as LLRT limits. The user can then select the range of dates that the graph will display.

The graph of the test data is drawn showing the alert and maximum values that were current when the test data was recorded. A marker is displayed wherever maintenance was performed on the valve. Markers for full stroke exercise, partial exercise, fail open/close, disassembly of valve, and remote position indication are displayed at the bottom of the screen. This allows graphs of non–test value valves to be displayed such as full stroke exercising of check valves. Plant mode for each test record is displayed at the top of the screen. The program then computes a 'best–fit' straight line approximation of the plotted test data. Data used for the computation is based on all test data points AFTER the last time maintenance was performed on the valve. The computed line is extended out 98 days after the last test date. This allows checking for abnormal trends that may occur with the next test interval. The user can also specify how many days to extend the graph. By pressing PRINT SCREEN, the display can be routed to a printer or plotter.

QUERY

The QUERY module allows the user to generate reports from the VALDATA and VTEST databases. No field names or query language is required to be known. All of the functions are menu driven. Upon entering this module, the user selects from a flash bar menu with fields from VALDATA database that the report is to be based on. After selecting a field the user is prompted for a comparison sign. Logical connections (AND, OR, or NO MORE SELECTIONS) are the next choice. The user can then select fields from the VTEST database. After completion of this selection, the user is prompted to select data fields that will be printed out for the report. Upon selecting a field, the field name along with a running total of the output line length is displayed on the left of the screen. The running line length is important since a max of 80 characters can be displayed on the screen. Next the output mode is selected. The user can select printer, screen, query total, or exit back to the initial query screen. When the line length is greater than 80 characters, an output mode selection of Screen, will generate an error message. If printer output mode is selected, the program checks to ensure that the printer is on–line. If the printer is off–line, the program will give a warning message. Next the print size and paper width requirements are displayed. Output to the printer then begins. If the output mode selected is query totals, a running total is kept showing the matches found from the query statement. Screen output mode will direct the query responses to the computer screen, pausing after each full screen. After completion of the query, the user is returned to the initial query screen.

An example of the power of the query function is as follows:

A request is made for a report showing stroke times of all motor–operated gate valves ranging from 4 inches to 8 inches, tested during plant mode 5.

Items to Search:

From VALDATA:

TYPE = GATE * type of valve

ACT = MO * type of operator

SIZE > = 4 * valve size > = 4 inches

SIZE < = 8 * valve size < = 8 inches

From VTEST:

PLANTMODE = 5 * valves tested at cold shutdown

TESTVAL > 0 * retrieve only stroke times

From the VTEST database, select VALVEID. TESTOATA, and TESTVAL for the the fields to appear on the printout.

All of the above items appear in flashbar menus thus relieving the user from learning any complicated query language or remembering data field names.

Pump Baseline Data

Pump baseline database consists of the following fields:

PDATA.dbf

	Name	Type	Length	Comments
1.	PUMPID	CHAR	15	Pump id
2.	TESTTYPE	CHAR	1	Full flow/recirc test type
3.	SURVTEST	CHAR	8	Surveillance test number
4.	TESTDATE	DATE	8	Test date
5.	MODE	CHAR	1	Plant mode when tested
6.	ST1	CHAR	8	Surveillance test 1
7.	ST2	CHAR	8	Surveillance test 2
8.	ST3	CHAR	8	Surveillance test 3
9.	ST4	CHAR	8	Surveillance test 4
				Static Suction Pressure
10.	SS	NUM	9.2	Baseline
11.	SS1	NUM	9.2	Upper required action
12.	SS2	NUM	9.2	Upper alert
13.	SS3	NUM	9.2	Lower alert
14.	SS4	NUM	9.2	Lower required action
15.	SS5	NUM	9.2	Technical specification limit
				Dynamic Suction Pressure
16.	DS	NUM	9.2	Baseline
17.	DS1	NUM	9.2	Upper required action
18.	DS2	NUM	9.2	Upper alert
19.	DS3	NUM	9.2	Lower alert
20.	DS4	NUM	9.2	Lower required action
21.	DS5	NUM	9.2	Technical specification limit
				Discharge Pressure
22.	DIS	NUM	9.2	Baseline
23.	DIS1	NUM	9.2	Upper required action
24.	DIS2	NUM	9.2	Upper alert
25.	DIS3	NUM	9.2	Lower alert
26.	DIS4	NUM	9.2	Lower required action

Name		Type	Length	Comments
27.	DIS5	NUM	9.2	Technical specification limit
				Differential Pressure
28.	DP	NUM	9.2	Baseline
29.	DP1	NUM	9.2	Upper required action
30.	DP2	NUN	9.2	Upper alert
31.	DP3	NUM	9.2	Lower alert
32.	DP4	NUM	9.2	Lower required action
33.	DP5	NUM	9.2	Technical specification limit
				Total Pump Flow
34.	TF	NUM	9.2	Baseline
35.	TF1	NUM	9.2	Upper required action
36.	TF2	NUM	9.2	Upper alert
37.	TF3	NUM	9.2	Lower alert
38.	TF4	NUM	9.2	Lower required action
39.	TF5	NUM	9.2	Technical specification limit
				Vibration Outboard Horizontal
40.	VOH	NUM	9.4	Baseline
41.	VOH1	NUM	9.4	Upper required action
42.	VOH2	NUM	9.4	Upper alert
43.	VOH5	NUM	9.4	Technical specification limit
				Vibration Outboard Vertical
44.	VOV	NUM	9.4	Baseline
45.	VOV1	NUM	9.4	Upper required action
46.	VOV2	NUM	9.4	Upper alert
47.	VOV5	NUM	9.4	Technical specification limit
				Vibration Outboard Axial
48.	VOA	NUM	9.4	Baseline
49.	VOA1	NUM	9.4	Upper required action
50.	VOA2	NUM	9.4	Upper alert
51.	VOA5	NUM	9.4	Technical specification limit
				Vibration Inboard Horizontal
52.	VIH	NUM	9.4	Baseline
53.	VIH1	NUM	9.4	Upper required action
54.	VIH2	NUM	9.4	Upper alert
55.	VIH5	NUM	9.4	Technical specification limit
				Vibration Inboard Vertical
56.	VIV	NUM	9.4	Baseline
57.	VIV1	NUM	9.4	Upper required action
58.	VIV2	NUM	9.4	Upper alert
59.	VIV5	NUM	9.4	Technical specification limit
				Vibration Inboard Axial
60.	VIA	NUM	9.4	Baseline
61.	VIA1	NUM	9.4	Upper required action
62.	VIA2	NUM	9.4	Upper alert
63.	VIA5	NUM	9.4	Technical specification limit

Name		Type	Length	Comments
				Bearing Temperature
64.	BT	NUM	9.2	Baseline
65.	BT1	NUM	9.2	Upper required action
66.	BT2	NUM	9.2	Upper alert
67.	BT3	NUM	9.2	Lower alert
68.	BT4	NUM	9.2	Lower required action
69.	BT5	NUM	9.2	Technical specification limit
				Motor Current
70.	MC	NUM	9.2	Baseline
71.	MC1	NUM	9.2	Upper required action
72.	MC2	NUM	9.2	Upper alert
73.	MC3	NUM	9.2	Lower alert
74.	MC4	NUM	9.2	Lower required action
75.	MC5	NUM	9.2	Technical specification limit
				Pump Speed
76.	PS	NUM	9.2	Baseline
77.	PS1	NUM	9.2	Upper required action
78.	PS2	NUM	9.2	Upper alert
79.	PS3	NUM	9.2	Lower alert
80.	PS4	NUM	9.2	Lower required action
81.	PS5	NUM	9.2	Technical specification limit

Since some of the above parameters are not required for every pump, the user can select which Parameters are desired to be tracked. For example, pump speed is not required to be tracked for pumps driven by synchronous or induction motors. By not selecting this parameter, the program will skip data entry for this field. Where code limits exist, the user has the option of letting the computer determine the limits based on the baseline data. This prevents errors and saves time by letting the computer perform the calculations. When maintenance has been performed on a pump, the user can edit the applicable parameters thus establishing new baseline acceptance criteria.

The database structure is as follows:

PTEST.dbf

Name		Type	Length	Comments
1.	PUMPID	CHAR	15	Pump id
2.	TESTTYPE	CHAR	1	Full flow/recirc test type
3.	SURVTEST	CHAR	8	Surveillance test number
4.	TESTDATE	DATE	8	Test date
5.	MODE	CHAR	1	Plant mode when tested
				Static Suction Pressure
6.	SS	NUM	9.2	Test value
7.	SS1	NUM	9.2	Current upper required action
8.	SS2	NUM	9.2	Current upper alert

PUMP TEST DATA

Pump test database is similar to the baseline database. The exception being the parameter baseline field is now the test value field. The other fields are utilized to record the acceptable, alert, and required action limits that were in place when a test was performed. Recording the current acceptable, alert, and required action limits provides a history of acceptance criteria throughout the life of the pump.

Name	Type	Length	Comments
9. SS3	NUM	9.2	Current lower alert
10. SS4	NUM	9.2	Current lower required action
11. SS5	NUM	9.2	Current tech spec limit
			Dynamic Suction Pressure
12. DS	NUM	9.2	Test value
13. DS1	NUM	9.2	Current upper required action
14. DS2	NUM	9.2	Current upper alert
15. DS3	NUM	9.2	Current lower alert
16. DS4	NUM	9.2	Current lower required action
17. DS5	NUM	9.2	Current tech spec limit
			Discharge Pressure
18. DIS	NUM	9.2	Test value
19. DIS1	NUM	9.2	Current upper required action
20. DIS2	NUM	9.2	Current upper alert
21. DIS3	NUM	9.2	Current lower alert
22. DIS4	NUM	9.2	Current lower required action
23. DIS5	NUM	9.2	Current tech spec limit
			Differential Pressure
24. DP	NUM	9.2	Test value
25. DP1	NUM	9.2	Current upper required action
26. DP2	NUM	9.2	Current upper alert
27. DP3	NUM	9.2	Current lower alert
28. DP4	NUM	9.2	Current lower required action
29. DP5	NUM	9.2	Current tech spec limit
			Total Pump Flow
30. TF	NUM	9.2	Test value
31. TF1	NUM	9.2	Current upper required action
32. TF2	NUM	9.2	Current upper alert
33. TF3	NUM	9.2	Current lower alert
34. TF4	NUM	9.2	Current lower required action
35. TF5	NUM	9.2	Current tech spec limit
			Vibration Outboard Horizontal
36. VOH	NUM	9.4	Test value
37. VOH1	NUM	9.4	Current upper required action
38. VOH2	NUM	9.4	Current upper alert
39. VOH5	NUM	9.4	Current tech spec limit
			Vibration Outboard Vertical
40. VOV	NUM	9.4	Test value
41. VOV1	NUM	9.4	Current upper required action
42. VOV2	NUM	9.4	Current upper alert
43. VOV5	NUM	9.4	Current tech spec limit
			Vibration Outboard Axial
44. VOA	NUM	9.4	Test value
45. VOA1	NUM	9.4	Current upper required action
46. VOA2	NUM	9.4	Current upper alert
47. VOA5	NUM	9.4	Current tech spec limit

Name		Type	Length	Comments
				Vibration Inboard Horizontal
48.	VIH	NUM	9.4	Test value
49.	VIH1	NUM	9.4	Current upper required action
50.	VIH2	NUM	9.4	Current upper alert
51.	VIH5	NUM	9.4	Current tech spec limit
				Vibration Inboard Vertical
52.	V	NUM	9.4	Test value
53.	VIV1	NUM	9.4	Current upper required action
54.	VIV2	NUM	9.4	Current upper alert
55.	VIV5	NUM	9.4	Current tech spec limit
				Vibration Inboard Axial
56.	VIA	NUM	9.4	Test value
57.	VIA1	NUM	9.4	Current upper required action
58.	VIA2	NUM	9.4	Current upper alert
59.	VIA5	NUM	9.4	Current tech spec limit
				Bearing Temperature
60.	BT	NUM	9.2	Test value
61.	BT1	NUM	9.2	Current upper required action
62.	BT2	NUM	9.2	Current upper alert
63.	BT3	NUM	9.2	Current lower alert
64.	BT4	NUM	9.2	Current lower required action
65.	BT5	NUM	9.2	Current tech spec limit
				Motor Current
66.	MC	NUM	9.2	Test value
67.	MC1	NUM	9.2	Current upper required action
68.	MC2	NUM	9.2	Current upper alert
69.	MC3	NUM	9.2	Current lower alert
70.	MC4	NUM	9.2	Current lower required action
71.	MC5	NUM	9.2	Current tech spec limit
				Pump Speed
72.	PS	NUM	9.2	Test value
73.	PS1	NUM	9.2	Current upper required action
74.	PS2	NUM	9.2	Current upper alert
75.	PS3	NUM	9.2	Current lower alert
76.	PS4	NUM	9.2	Current lower required action
77.	PS5	NUM	9.2	Current tech spec limit

When test data is entered, baseline data along with the current limits are displayed for each parameter. Any test value that meets or exceeds any limit is flagged by the computer.

PUMP PERFORMANCE GRAPHING

The initial screen for pump graphing asks the user for the pump id. The program will then display the types of tests that are available for the particular pump (recirc flow, full flow, or both). The first and last test dates for the selected test types are displayed for reference. The user then inputs the required starting and ending test dates for the graph. A pop-up window then allows the user to select the type of graph(s) requested.

ISTBASE graphing of pump test data allows for single or double graphs of the following parameters:

1. Suction pressure (static and dynamic)

2. Discharge pressure

3. Differential pressure

4. Flow

5. Vibration (up to six points)

6. Bearing temperature

7. Pulp speed

8. Motor current

9. Plant mode

Also available for graphing is pump test data plotted on the actual pump performance curve.

Test limits (acceptable, alert, and required action) are also displayed on the graph.

As with the valve graphs, a best fit curve is computed and displayed. The time axis (x–axis) is also extended 90 days to allow the user to determine if any limits will be exceeded within the next required test date. After the graph is displayed, the PRINT SCREEN key will direct the graph to the printer or plotter.

Pump performance test data can be plotted on the computer screen against the actual pump performance curve supplied by the pump vendor. Pumps are supplied with a certified drawing of the pump's performance of total developed head vs flow. The Section XI limits, or user supplied limits, on differential pressure and flow can be graphically represented on the screen to show valid or acceptable areas of operation. Pump test results, consisting of differential pressure and flow, are plotted as a point on the graph showing the user where the data falls on the pump curve. Plotting of these points allows visual trending of pump performance.

ISTBASE has a datafile for each pump required to be in the IST program. This file contains enough data points, consisting of total developed head (TDH) and flow, to display a smooth curve when plotted out on the computer screen. Data from the pump baseline section provides the Section XI limits of Acceptable, Alert, and Required Action values. These values, when plotted, display rectangular boxes. The acceptable area is painted green, alert area is displayed yellow, and all other areas are displayed red. Since some pumps required a full flow test as well as a recirculation test, the program will automatically display the boxes at the low flow/high TDH section of the curve as well as the high flow/low TDH section. These databases can be modified by the user to allow for changes when major pump maintenance has occurred.

Using the time window selected in the initial graphing section, each data point is displayed against the pump curve. Flow, differential pressure, and the test date are displayed at the top of the screen. Pressing any key will select the next test data record. Selecting Print Screen will direct a copy of the screen to the printer/plotter.

MEMO AND MAINT DATAFILES

The last four databases are MEMO and MAINT. The memo database allow short messages to be recorded about a valve or pump. An example would be: VALVE IS LOCATED NEXT TO SNUBBER 1AF-6 ELEVATION 236.

MEMO database structure is listed is as follows:

VMEMO.dbf **General Memos for Valves**

Name	Type	Length	Comments
1. VALVEID	CHAR	15	Valve id number
2. MDATE	DATE	8	Memo date
3. VMEMO	CHAR	66	General memo field

PMEMO.dbf **General Memos for Pumps**

Name	Type	Length	Comments
1. PUMPID	CHAR	15	Pump id number
2. MDATE	DATE	8	Memo date
3. PMEMO	CHAR	66	General memo field

The MAINT databases record when maintenance has been performed on a valve or pump. A description field is provided to describe the maintenance activity. An example would be: TIGHTENED VALVE PACKING DUE TO BORATED WATER LEAK. When graphing test data, maintenance dates will be marked on the screen.

MAINT database structure is listed as follows:

VMAINT.dbf **Maintenance for Valves**

	Name	Type	Length	Comments
1.	VALVEID	CHAR	15	Valve id number
2.	MDATE	DATE	8	Maintenance date
3.	VMAINT	CHAR	66	Maintenance description

PMAINT.dbf **Maintenance for Pumps**

	Name	Type	Length	Comments
1.	PUMPID	CHAR	15	Pump id number
2.	MDATE	DATE	8	Maintenance date
3.	PMAINT	CHAR	66	Maintenance description

ARCHIVE DATA

Over the life of a plant, test datafiles can grow very large. A user might have a 10 year history of test results, but may only be interested in the last 3 years of data. In order to restrict the size of the test databases, an ARCHIVE routine allows the user to extract test data either by date, or by individual component. The selected test data is then removed from the active data bases and stored in separate databases. This 'archived' data can be returned to the active databases if required.

CONCLUSION

Use of a personal computer greatly simplifies the required data handling of an IST program. Forty years of test data can be manipulated into useful reports in minutes. Computer checking of input test data against established limits informs the user when test values are out of range. Graphing allows visual trending of test data along with 'best fit' line extended past last test date. The above described generic version of ISTBASE has been modified for some utilities to include scheduling capabilities, instrument calibration databases, variable databases used for data input validation as well as conforming with ANSI/IEEE Standard 730–1984 "IEEE Standard for Software Quality Assurance Plans." Due to the lower cost and higher performance of todays personal computers, an automated method of managing an IST program provides savings in time as well as greatly reducing personnel errors in interpretation of test data.

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ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

In one of your early example charts, you had shown on stroke timing a graph that kind of wiggled around. Is that actual data or is this an example of something you did for demonstration purposes?

Response 1

That is actual data.

Question 2

One of our earlier speakers gave a premise that stroke timing was not really effective because it could change. So your results seem to be a little different from his.

Response 2

Well, that particular valve, when I was at Harris, was a problem.

But with the majority of the valves you see, it is going to be essentially a straight line with small deviations. That one was chosen because it is flashy. Like I mentioned on blowdown valves, those valves did range from three to five seconds up to 30 when the plant was hot, until they fixed the problem with it.

FULL SCALE VALVE QUALIFICATION AND FLOW INTERRUPTION TESTING PROVIDES INSIGHTS ON INDUSTRY MOTOR OPERATOR SIZING SHORTCOMINGS AND DIAGNOSTIC TESTING LIMITATIONS

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ABSTRACT

Valve test programs sponsored by the United States Nuclear Regulatory Commission[a] (USNRC) and conducted by researchers from the Idaho National Engineering Laboratory (INEL) have yielded important insights concerning motor-operated valve operator sizing and in situ testing. The most significant of these test programs was a full-scale, high-energy qualification and flow-interruption gate valve test program conducted in the spring of 1988. For this program, we modified two representative 6-in, 900-lb valve assemblies to include a high-temperature load cell in each of the valve stems to directly measure stem thrust. Instrumentation in the flow loop and on the motor operators measured the important responses and parameters. At various times during the test program, each of the currently popular diagnostic test systems was installed to monitor the performance of the valve motor operators. The valves were subjected to the hydraulic and leakage testing requirements specified in ANSI B16.41 (QV-4) and to flow interruption (valve closure) testing at boiling water reactor (BWR) temperatures and pressures at line break conditions. Results show that for the tested valves, the variables used in current industry motor operator sizing equations under-predict actual valve thrust requirements at high temperature loadings; for one valve design, the equation may need an additional term to account for nonlinear performance. Results also show that the thrusts needed to close the valves were sensitive to the fluid temperature, and that the results of testing at lower pressures, temperatures, and flows cannot be extrapolated to design basis conditions.

INTRODUCTION

The INEL, under the sponsorship of the USNRC, is performing research to help resolve specific generic issues and to provide information to develop and improve industry qualification, operating, and maintenance standards for mechanical equipment. This research effort includes a program that tested the operability (opening and closing) of two full-scale gate valves subjected to the high temperatures, high pressures, and high flows of typical BWR line-break conditions. The purpose of the test program was to support NRC efforts regarding Generic Issue 87 (GI-87), "Failure of the HPCI Steam Line Without Isolation." This paper summarizes the impact of that test program on GI-87 and on another highly visible safety issue, Generic Issue II.E.6.1 (GI-II.E.6.1), "In Situ Testing of Valves," which represents the overall USNRC concern for valve operability in nuclear power plants. One effort in the resolution of GI-II.E.6.1 is addressed by *IE Bulletin 85-03*, "Motor-operated Valve Common Mode Failures During Plant Transients Due to Improper Switch Settings," and will be further addressed by a proposed generic letter on safety-related motor-operated valve testing and surveillance.

GI-87 applies to those process lines that communicate with the BWR primary system, pass through the containment, and contain normally open isolation valves. Two steam supply lines and one hot water supply line meet these criteria: the high-pressure coolant injection (HPCI) and the reactor core isolation cooling (RCIC) lines, and the reactor water cleanup (RWCU) line, respectively. The concern with these systems is

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

whether the containment isolation valves will close in the event of a downstream pipe break.

IE Bulletin 85–03 requested the utilities [both pressurized water reactor (PWR) plants and BWR plants] to develop and implement a program that would ensure that switch settings on selected safety–related motor–operated valves are chosen, set, and maintained correctly to accommodate the maximum differential pressures expected on these valves during both normal and abnormal events within the design basis. It is also our understanding that the USNRC staff is considering a generic letter that could expand the application of *IE Bulletin 85–03* to all safety related valves, including those that might be mispositioned.

To meet these new operating criteria, industry developed new motor–operated valve diagnostic test equipment and methods for in situ valve testing. One of the new requirements was that the valve control switches be set correctly for the design basis loading for each valve. *IE Bulletin 85–03* succeeded in significantly improving the operability of the selected safety–related valves because it caused the utilities to reanalyze the design basis load for the applicable motor–operated valves and reset the control switches accordingly. In many cases, these analyses were more complete than the analyses in the original procurements, and utilities reset the control switches in accordance with the improved analyses.

However, very little design basis testing of valves had been conducted outside the plant, and the utilities generally cannot test the valves at design basis conditions in the plant. This situation left the utilities relying on valve motor operator switch settings that were based on analyses of the design basis loadings. Utilities verified the torque or thrust levels for each valve through seat or back seat type loadings. The problem with this type of valve testing was that the motor operator is at full speed and is unloaded when the seat contact is made, and motor operator momentum constitutes a large part of the final measured thrust or torque. Such testing cannot verify the adequacy of the original operator sizing or the correctness of the torque switch setting. This problem is evidenced by the INEL gate valve test results, which show that because of the higher than anticipated friction factors at the disc guide to valve body guide surfaces and the variability of the valve stem–to–stem nut factor under load, the momentum and consequent high thrusts seen at low valve loadings will not be realized at higher loadings. Thus, valve operability verified under the lower loadings cannot ensure operability at design basis loadings.

BACKGROUND

Parallel with the *IE Bulletin 85–03* work, the USNRC contracted the INEL to provide technical insights in support of USNRC efforts regarding GI–87, which addresses the potential failure of selected BWR process line isolation valves to close in the event of a downstream pipe break. The initial INEL effort was to determine what valves are installed in these applications, their sizes and manufacturers, what testing had been previously performed, and which system would be the highest risk to plant safety. Surveys identified the flexible wedge carbon steel gate valve with a Limitorque motor operator as the predominate valve in the three systems (HPCI, RCIC, and RWCU) addressed by GI–87. The most common valve sizes are 4 in. for the RCIC system, 6 in. for the RWCU system, and 10 in for the HPCI system. A downstream break in the RWCU system would represent the highest risk to the plant, so it was decided that the initial flow isolation testing should provide information on valve operability questions associated with the RWCU system.

To avoid duplication, we reviewed applicable test programs that have been previously completed; that review showed that sufficient test information was not available for the support of the USNRC effort regarding GI–87. Among the test programs reviewed was the Electrical Power Research Institute (EPRI) power–operated relief valve/block valve testing at Duke Power in 1980. For applicability to GI–87, this program had three shortcomings: (a) the block valves were stainless steel as opposed to carbon steel; (b) the tests were go/no–go type tests where neither motor–operated valve thrust nor torque were measured; and (c) the EPRI test medium was steam, which would be more applicable to HPCI and RCIC systems than to RWCU systems. Also, Kraftwerkunion (KWU) of West Germany had tested a 3–in. stainless steel parallel disc gate valve at blowdown flows for the Central Electric Generating Board (CEGB), United Kingdom. Mechanical interference on the downstream disc prevented closure. KWU has performed full–flow interruption testing on a large number of valve types; however, our initial contacts indicated that the information at that time was proprietary. Since that time, Bechtel and KWU have formed an alliance, and the information may become more available.

GI–87 Hot Water Valve Test Program

We selected two 6–in., 900–lb class, flexible–wedge gate valves for testing. The design of the valves represents a large portion of the isolation valves installed in

BWR RWCU systems. The valves were manufactured and modified specifically for this program: in each valve, the stem was cut, the yoke lengthened, and a highly accurate, high–temperature load cell installed so that direct measurements of stem force could be obtained (see Figure 1). One of the valves, designated Valve A, was further modified by installing a very large motor operator. This modification ensured that sufficient thrust would be available to close the valve in the event of higher–than–anticipated closing loads. The other valve, Valve B, had a normal–sized motor operator with the torque switch set to deliver the thrust calculated for the highest test load (see test matrix, Table 1). Valve A had a normal guide design and large disc–to–body guide clearances, while Valve B had a hardfaced guide design and tighter clearances.

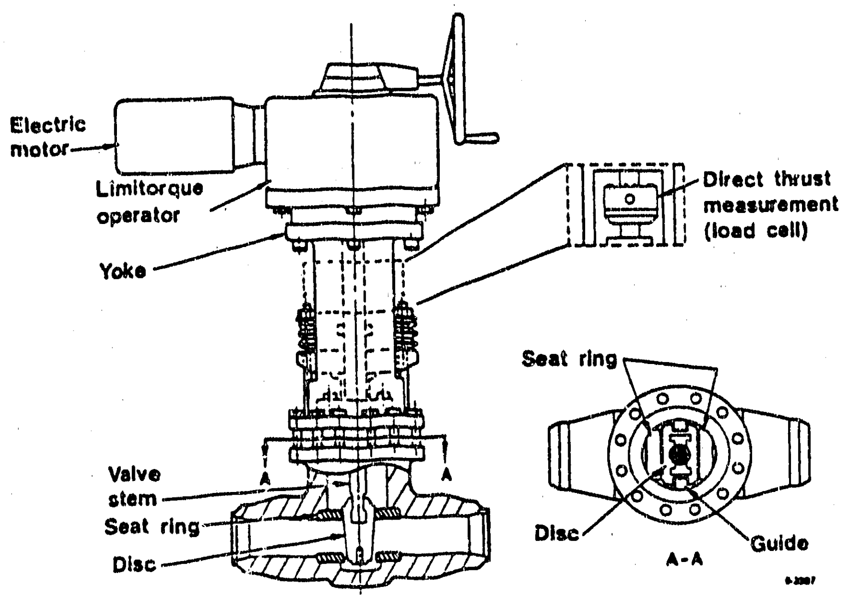


Figure 1. The test program used two motor–operated gate valves similar to the typical valve shown here; load cells were installed in the valve stem, as shown.

Table 1. GI–87 valve test matrix

			Initial Conditions		
Valve	Test Series	Description	Pressure (psig)	Temperature (°F)	Diagnostic Equipment
A	1	Qualification	—	—	MCSA
A	3	Blowdown	1000	480	MCSA
A	2	Blowdown	1000	530	None
A	4	Blowdown	1000	400	V–MODS
A	6	Blowdown	1400	530	V–MODS
A	5	Blowdown	1400	580	MOVATS
A	7	Blowdown	1400	450	MOVATS
A	9	Blowdown	600	430	None
A	8	Blowdown	600	480	None
A	10	Blowdown	600	350	MAC, VOTES
A	11	Blowdown	1000	530	MCSA
B	1	Qualification	—	—	MCSA
B	2	Blowdown	1000	530	MCSA, MOVATS
B	3	Blowdown	1400	580	V–MODS

Table 1. (continued)

Valve	Test Series	Description	Initial Conditions Pressure (psig)	Initial Conditions Temperature (°F)	Diagnostic Equipment
B	4	Blowdown	600	480	MAC
B	5	Blowdown	1000	530	V–MODS

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

The flow loop and instrumentation are shown in Figure 2. The test system featured a large water tank, heated and pressurized so that various system water conditions could be established and regulated throughout the valve cycle. The test program was designed to determine valve operability and to measure the forces needed to close the valves (flow interruption) and open them under various system conditions, as listed in Table 1. The 6–in. test valves were mounted in the test loop, as shown in Figure 2 with appropriate instrumentation for obtaining measurements of temperature, pressure, flow, motor current and voltage, and valve stem position. A fast acting, hydraulically operated valve was positioned so that when it was actuated, pipe break flow through the test valve would be initiated and the system's fluid abruptly dumped to the atmosphere. The flow loop included a small additional recirculation loop (not shown in Figure 2) that was used to heat the system to the initial temperature conditions before tests and to conduct low–flow valve testing.

In addition to the load cell installed in the valve stem and the other instrumentation listed previously, instrumentation was provided by manufacturers of the more widely used valve diagnostic systems. These manufacturers were invited to participate in the test program to validate the feasibility of extrapolating in situ plant diagnostic test results to design basis loadings. The diagnostic systems and their test participation are shown in Table 1.

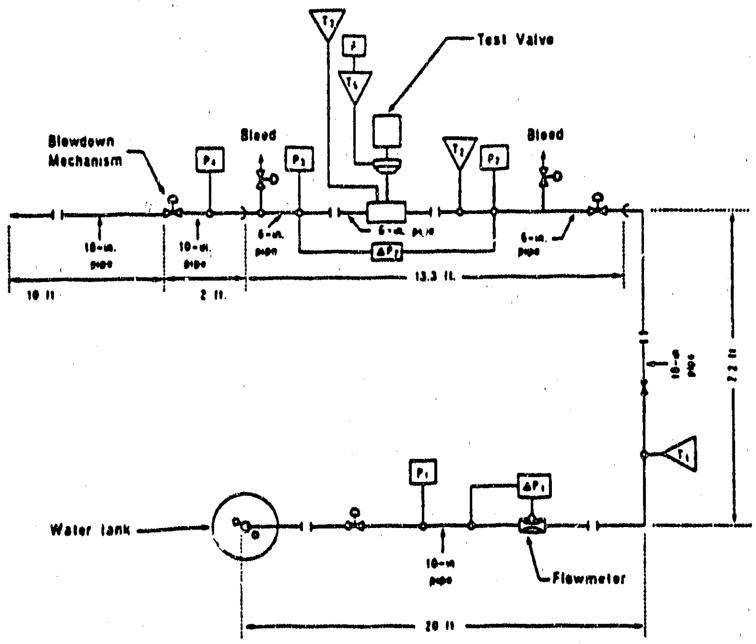
Before the flow interruption portion of the test program, both valves were subjected to the applicable hydraulic and leakage qualification tests outlined in ANSI B16.41 (QV–4), the nuclear valve qualification standard. The purposes of these tests were to establish baseline data and to ensure valve qualification before the valves were tested under high loads.

As a part of each flow interruption test, the test valve was opened against differential pressure with low flow and without flow and, when the test reservoir inventory permitted, was partially reopened after closure and reclosed with high flow. The parametric flow interruption testing for Valve A included varying both the starting pressure and the degree of fluid subcooling for a given pressure. For Valve B, only the pressure was varied at a constant 10°F subcooling.

TEST RESULTS

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

Valve Stem Force. Predicting the thrusts needed to open or close a gate valve is a critical step in sizing the



Simplified GI-87 test loop showing configuration, lengths, and pipe sizes

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

motor operator. This is typically done with the following equation:

$$F_t = \mu_d A_d \Delta P \pm A_s P + F_p \qquad (1)$$

where

F_t = total stem force

Dynamic Component:

μ_d = disc factor

A_d = disc area

ΔP = differential pressure

Static Component:

A_s = stem cross-sectional area

P = stem pressure

F_p = packing drag load (a constant).

Normally, the motor operator or valve manufacturer would use this equation to determine the maximum thrust needed to close the valve. The maximum predicted thrust at valve seating is the value the manufacturer would use for sizing the motor operator. In our analysis, we make incremental calculations of the predicted disc force throughout the closing and opening cycles, using the measured values for the parameters in the equation (differential pressure, disc area, etc.) and a constant disc factor, either the common 0.3 or the more conservative 0.5. We performed our calculations in this manner to see how well the equation modeled the actual stem forces during the entire cycle. A comparison of the predicted forces to the measured forces can show where the deviations start to take place and allows us to look at the fluid conditions and other parameters at that point to determine if there are other influences beside differential pressure affecting stem forces during valve operation.

The test results show that the portion of the equation designated as the static component is repeatable and linear with pressure. Industry has typically assumed 0.3 for the disc factor in the dynamic component of the equation. However, test results show that when a 0.3 disc factor is used in Equation (1), the resulting total stem force is inadequate for predicting valve thrust for the majority of design basis pressures and temperatures. Measurements of actual thrusts indicate that direction of operation (opening vs. closing), internal valve design (amount of disc to guide clearance), temperature, and fluid quality affect valve loadings in a manner not accounted for by the sizing equation.

Although both valves were 6-in 900-lb class flexible wedge gate valves, they responded to the flow interruption test quite differently. Figures 3 and 4 show the forces measured during valve closure (with normal BWR primary temperature and pressure) compared to forces calculated using both a normal disc factor of 0.3 and a higher factor of 0.5. The plots show that the 0.5 disc factor calculation bounds the flow isolation forces measured in Valve B but not those measured in Valve A. The pretest calculations using the 0.3 disc factor under predicts the actual closing force for both valves.

Valve B's measured forces follow the shape of the calculation quite well, and we describe this performance as linear. We describe Valve A's performance as nonlinear. We believe that the nonlinear performance of Valve A is the result of the greater disc-to-guide clearance in this valve design. Disassembly of the valve and inspection of the disc revealed a disc guide surface wear pattern indicating that the disc had tilted downstream as it closed, with a very small bearing area of the disc guide surface riding on the body guides. These small bearing areas show yielding and plastic deformation, which may account for the nonlinear performance. This hypothesis is further confirmed by the fact that at flow isolation, when the disc entered full contact with the seat ring, the forces dropped and the final force to full seating decreased to a level near that calculated with the 0.5 disc factor.

Figures 5 and 6 show forces measured during valve opening. The relationship between the forces calculated with the 0.3 disc factor and the measured forces for valve opening with full pressure, high temperature, and line break flows is similar to that for valve closing. However, the 0.5 calculation marginally envelopes the response of Valve A and slightly underpredicts the response of Valve B. Note that for both valves, the peak thrusts (excluding the spike caused by the hammer blow at unseating) are reached after flow is established, not while the disc is being moved off the seat.

The effects of temperature on valve response is shown in the next four plots. Figures 7 and 8 show calculated stem forces and forces measured during valve opening against differential pressure for Valves A and B with the test fluid at <100°F. The forces measured during opening are enveloped by the 0.3 disc factor calculation.

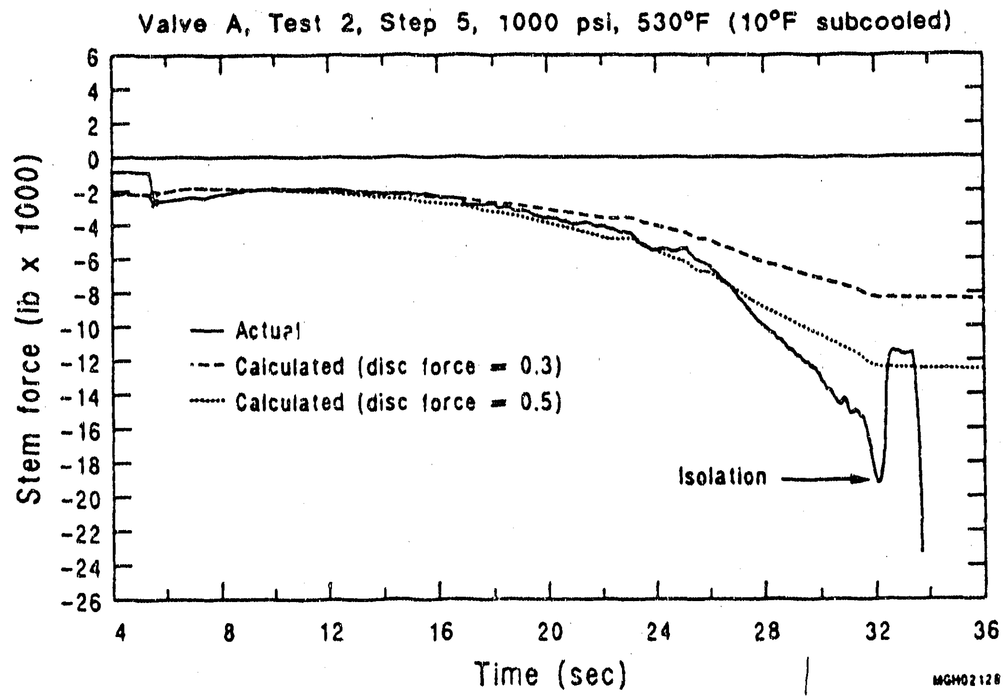


Figure 3. In this Valve A test, the loads measured during closing were greater than the loads calculated using 0.3 and 0.5 disc factors.

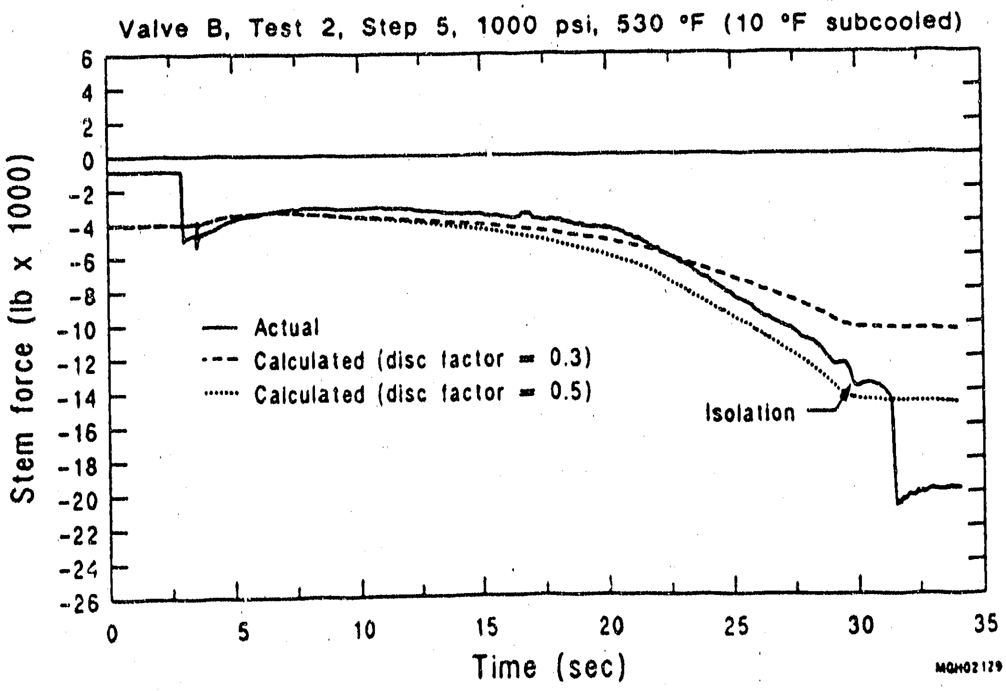


Figure 4. The response of Valve B is more linear than that of Valve A; the calculation using a 0.5 disc factor marginally envelopes the measured load, but the 0.3 calculation is not conservative.

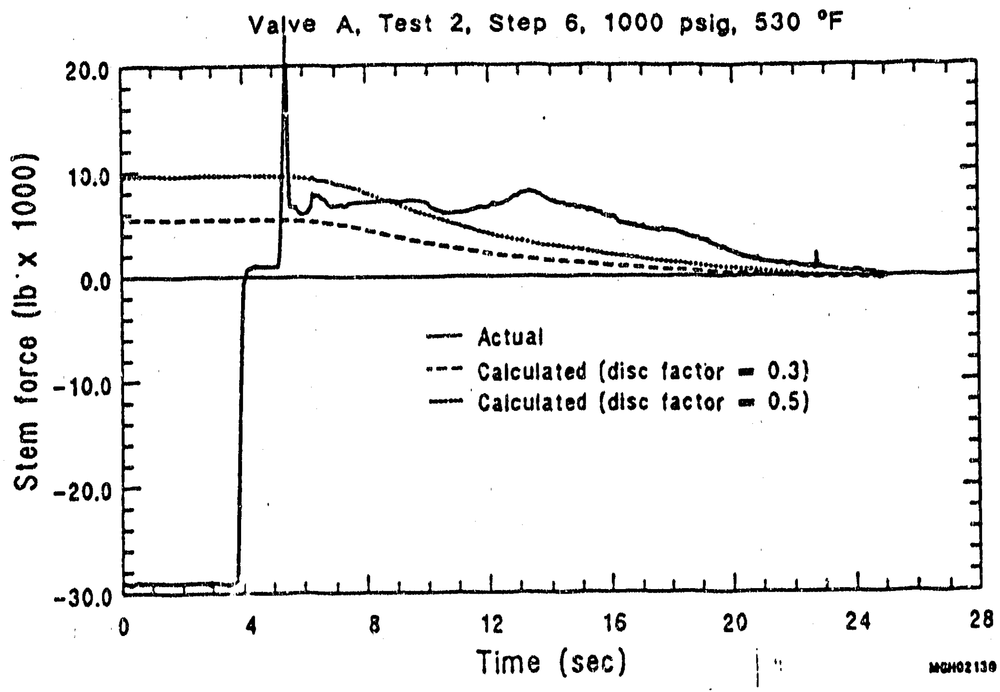


Figure 5. In this Valve A test, peak thrust encountered during opening was measured not while the disc was being lifted off the seat, but well after flow was established.

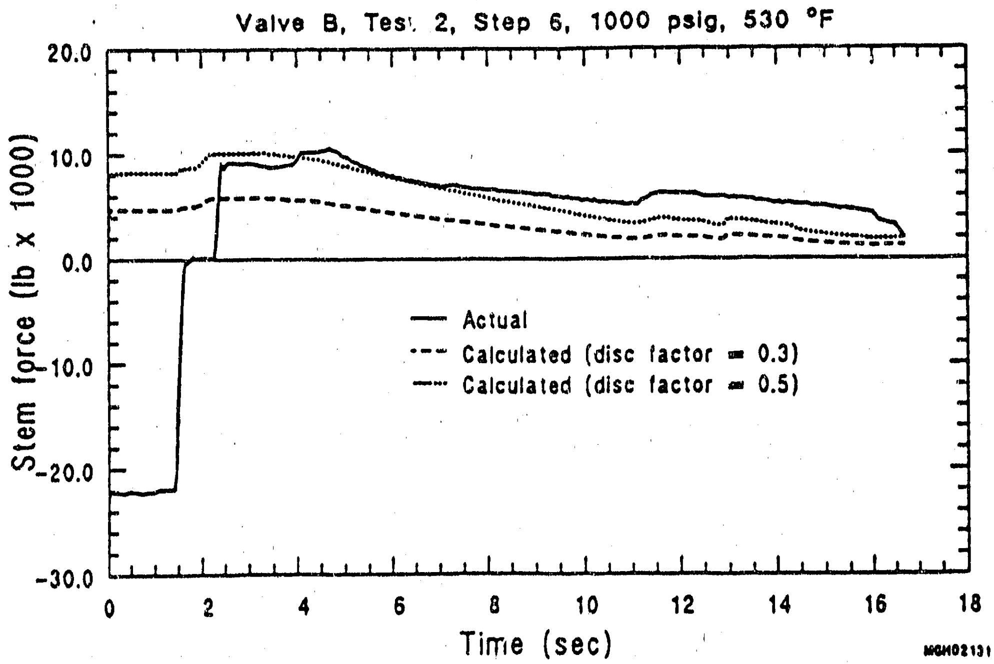


Figure 6. The response of Valve B is similar to that of Valve A (Figure 5); the absence of a spike at the hammer blow is because the valve was not fully seated at the end of the previous closing cycle.

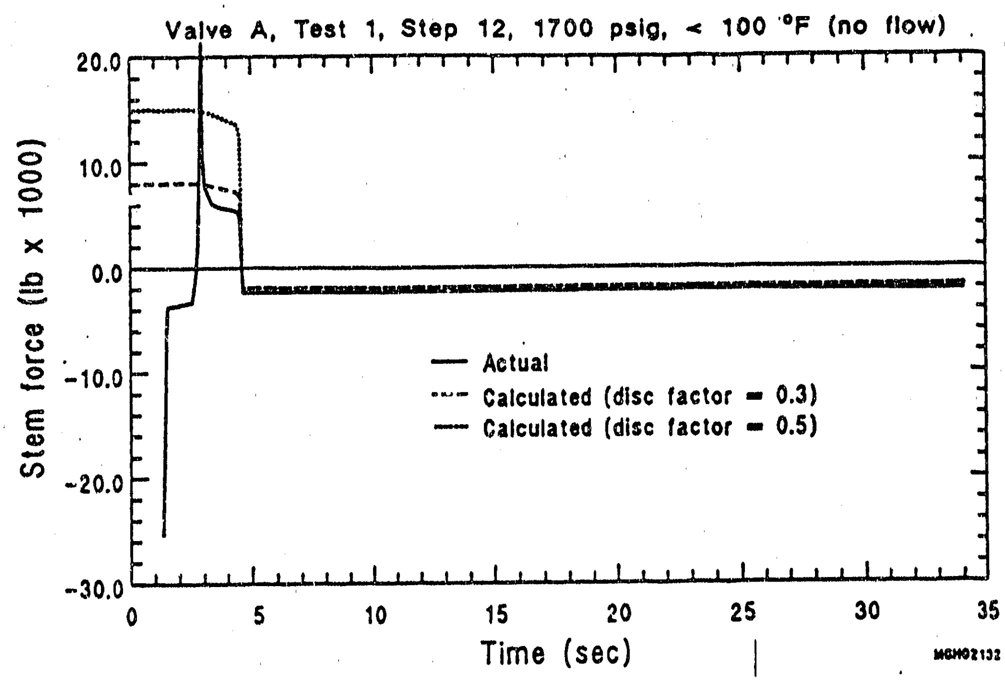


Figure 7. With Valve A opening against differential pressure only (no flow, cold fluid), the measured thrust is enveloped by the 0.3 disc factor calculation.

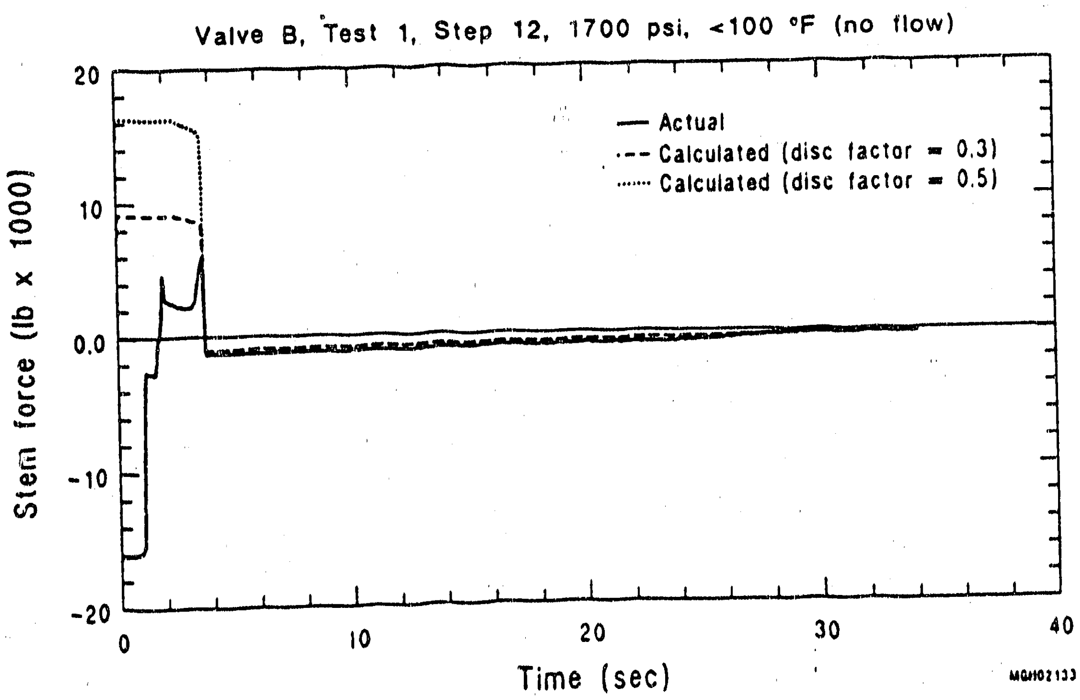


Figure 8. With Valve B opening against differential pressure only (no flow, cold fluid), the measured thrust is enveloped by the 0.3 disc factor calculation.

Figures 9 and 10 show the results of valve opening tests against a slightly lower pressure differential and with a fluid temperature of 580°F. The forces needed to open the valves are significantly higher than those measured during the cold test, and the 0.5 disc factor calculation only marginally envelopes the responses.

Stem Factor and Operator Momentum. The stem factor and the nature of the load that deflects the torque spring at the time that the torque switch trips are two valve operating characteristics that make it exceptionally difficult to set the valve control switches correctly during typical in–plant valve testing at loads lower than design basis loads. The two phenomena are different operating characteristics, but they are inseparable as they relate to in–plant testing.

The stem factor is used to calculate the conversion of motor operator torque to stem thrust. This relationship is normally expressed as

$$T = \mu_s F_t \qquad (2)$$

where

T = operator torque

μ_s = stem factor

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

where the stem factor is a function of stem diameter, thread pitch, and lead and the coefficient of friction between the actuator stem nut and the valve stem. As in Equation (1), the only variable that cannot be accurately measured or estimated in the stem factor equation is the parameter normally considered to be the coefficient of friction. Test results indicate that other phenomena may need to be included in this parameter. Most in the industry use a 0.15 or a 0.2 coefficient of friction for this parameter. Our testing and the work of others show that without hardware damage, the 0.15 value is marginally conservative and the 0.2 value may be excessively conservative. The stem factor in Equation (2) is treated as a constant, and many in industry believe it is a constant in operation. Again, our test results show this is not true; the stem factor increases with load. Also, as a motor–operated valve ages and as maintenance is performed, the correspondence between the torque switch position and the delivered output thrust becomes less reliable. Modern diagnostic test equipment for valve motor operators has allowed the utilities to recalibrate a motor operator torque switch in situ. During such recalibrations, the variability of the stem factor and the deceptively high thrusts of valve–seat–induced torqueouts can result in improperly set motor operator control switches.

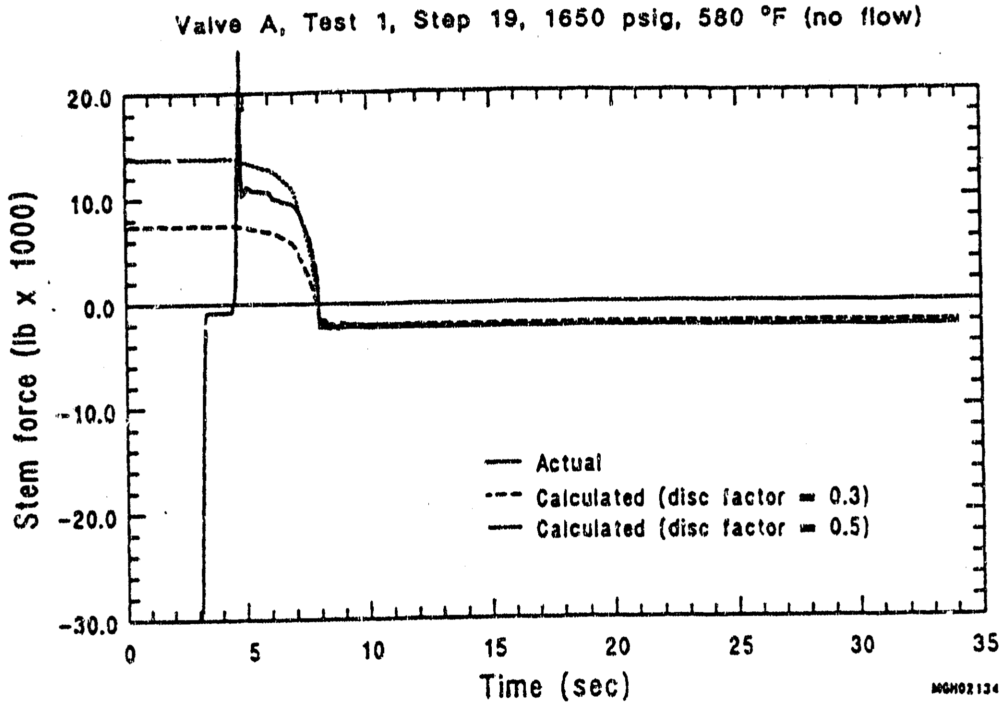


Figure 9. With an increase in fluid temperature, the measured thrust is not enveloped by the 0.3 disc friction factor (compare Figure 7).

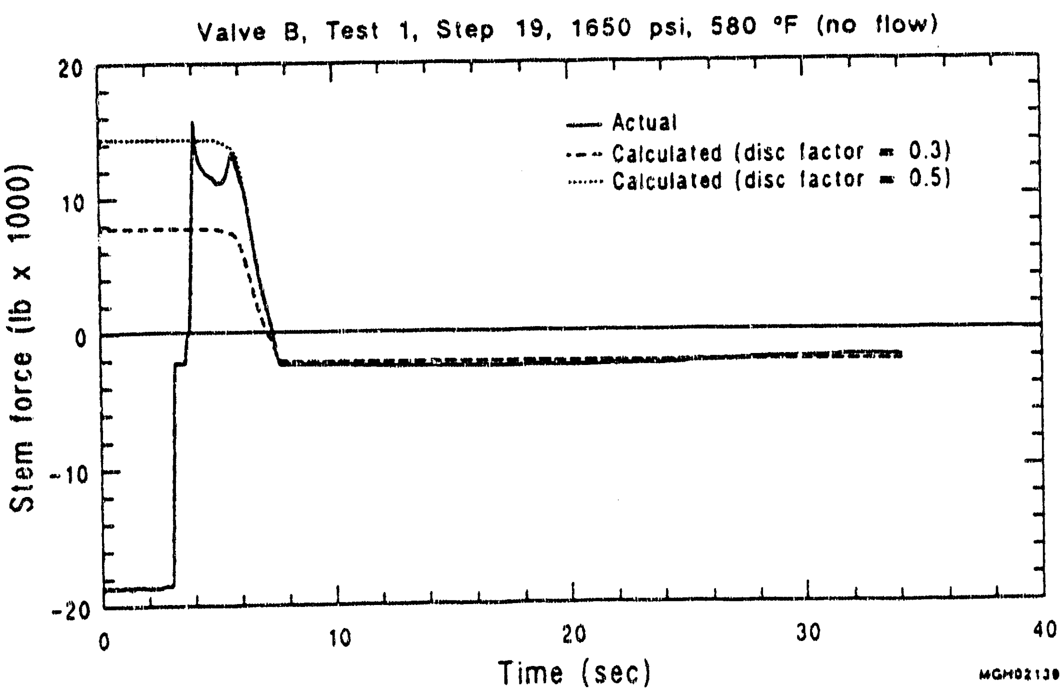


Figure 10. With an increase in fluid temperature, the measured thrust is not enveloped by the 0.3 disc friction factor (compare Figure 8).

The following test results show an example of this phenomenon. Valve B was selected for this analysis because of its linear performance during closing. The analysis shows that the final thrust in the valve stem varies depending on how the motor operator was loaded before and at torque switch trip. Initially we believed this variability to be a function of motor operator momentum; however, the measured motor operator parameters did not bear this out. The measured parameters did show that when the valve was lightly loaded, the stem factor was low until the moment the disc wedged in the seat. When the valve was highly loaded, the stem factor was higher, resulting in a poorer conversion of the torque to thrust. The measurements also showed that when the valve was lightly loaded and the disc wedging in the seat was the load that tripped the torque switch, the final thrust in the valve stem was higher than when the valve was highly loaded. With high loads, the torque spring was deflected by the disc load almost to the point of torque switch trip before the disc first contacted the seat. The initial contact with the seat, combined with the disc loads, was enough to trip the torque switch. From this point to the time the motor controller drops out and the motor operator momentum is spent, the worm acts like the input to a planetary gear, where the remaining revolutions of the motor are split between the worm turning the stem nut and the worm climbing the worm gear and compressing the torque spring past the torque switch trip point (see Figure 11). With light loads, however, the disc is already wedged very tightly at torque switch trip, and the remaining revolutions of the motor are not split; they all go into over compression of the torque spring, and thus the resulting final stem forces are higher.

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

Figure 12 shows the forces measured as the valve closed against pressure only. This test is typical of what a utility might be able to do. Note the final thrust (22,000 lb) with the valve lightly loaded and with torque switch trip induced by the disc wedging in the valve seat. When the disc contacts the seat and the torque switch is tripped, power continues to be supplied to the motor until the motor controller drops out (typically a time lag of 15 to 60 msec). At this time, the valve disc is wedged deeply in the seat. After the

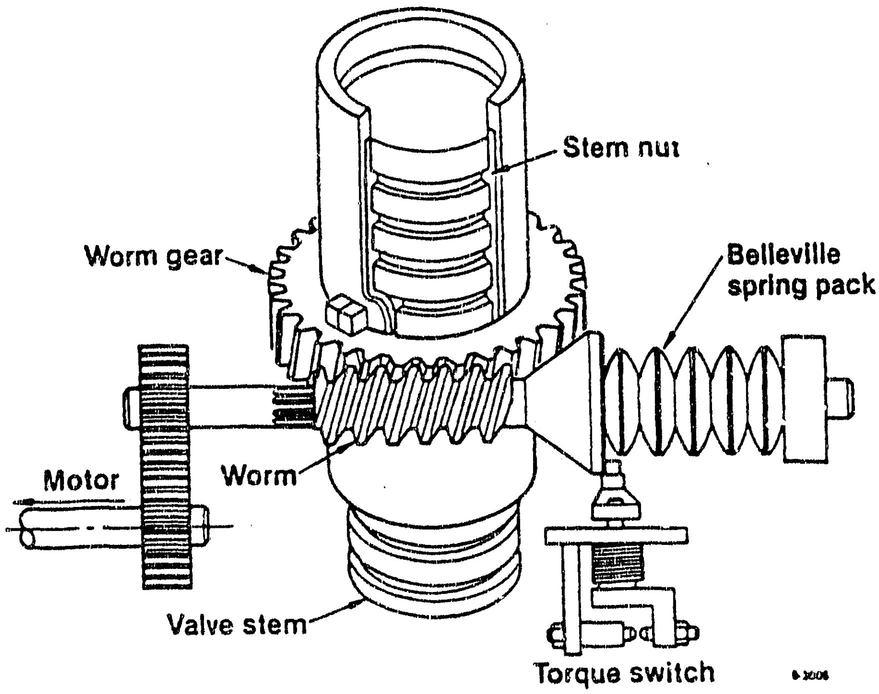


Figure 11. At the torque switch trip, the worm may either turn the worm gear and drive the disc deeper into the seat, or (if the disc will move not further) climb the worm gear, overcompressing the torque spring and producing additional thrust in the valve stem.

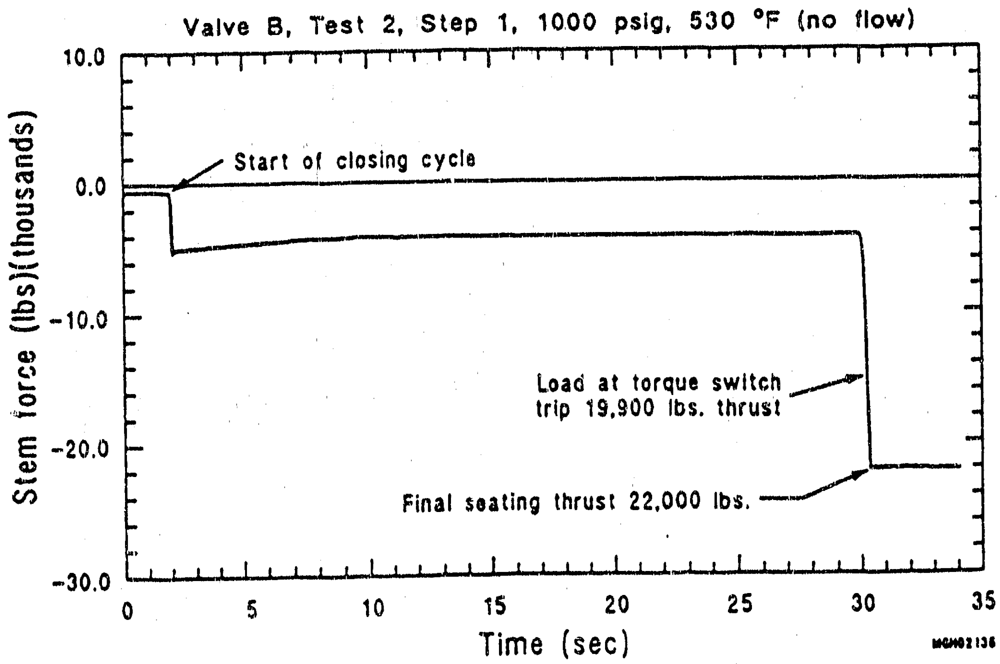


Figure 12. Though the torque switch was set to trip at 18,000 lb thrust in the absence of operator momentum, this lightly loaded valve achieved a significantly higher final seating thrust.

motor electrical power is broken, there is a period of deceleration of the motor operator components. This deceleration is proportional to the speed and mass of the motor operator, primarily the motor. With fast acting valves, there can also be significant momentum in the valve internals.

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

In Figure 13, we see the forces measured as the same valve closed against three different pressures at high flows. Note that with the same torque switch setting, the force when the torque switch tripped in the 600 psig test with high flow is less, at 18,100 lb, than the force when the torque switch tripped in the no-flow static pressure test, at 19,900 lb (Figure 12).

The valve closing at 1,000 psig shows a significantly higher load before flow isolation. Just before this test, the valve stem was lubricated, and a slightly higher thrust (18,600 lb) was obtained when the torque switch tripped. However, the valve stem position and the subsequent reopening of the valve indicated that the valve was lightly seated and the measured force was a reflection more of closing load than of seating load. During the closing at 1,400 psig inlet pressure, the design basis for operator sizing and torque switch setting, the valve marginally isolated flow but did not seat; the operator tripped on disc friction. The thrust when the torque switch tripped was lower, at 16,500 lb., a 17% reduction in the thrust at torque switch trip and a 25% reduction in final thrust, as compared to the lightly loaded case shown in Figure 12. The diagnostic equipment monitoring the operator performance showed that in contrast to the varying thrust, the operator output torque varied less than 3% for all valve loadings.

The variability of the stem factor under changing valve stem loads is shown in Figure 14. This stem factor history is derived from measurements of stem force and measurements of torque spring deflection mathematically converted to operator torque. This figure shows that the stem factor increased with load, resulting in less valve stem thrust at a given motor operator

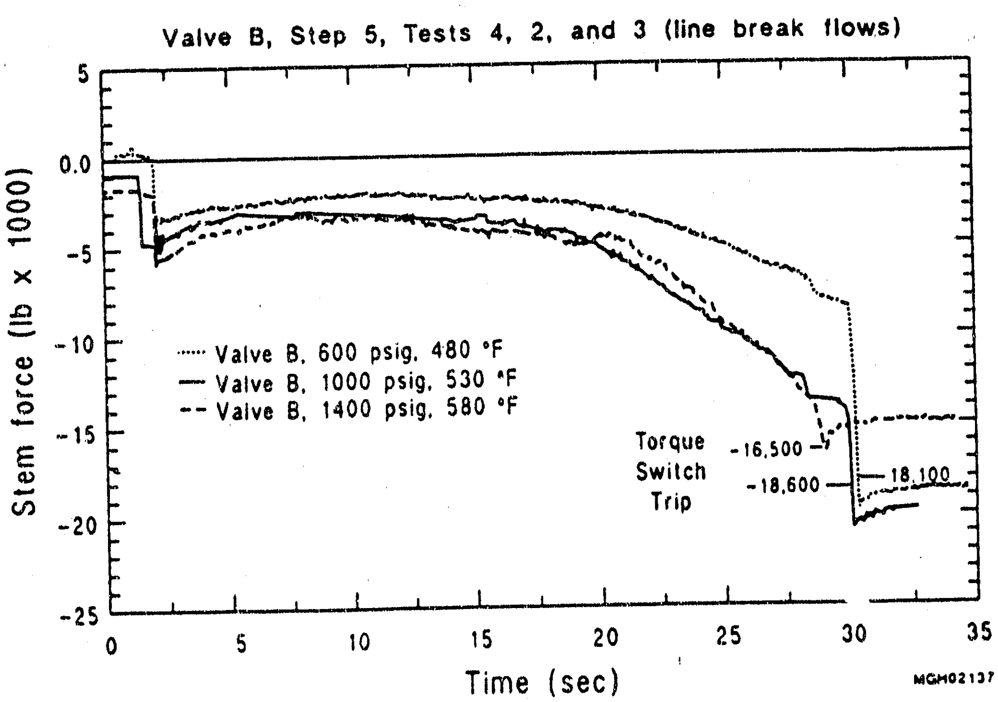


Figure 13. As stem factor and operator momentum are affected by increased loadings, the final thrust is less even though the torque switch setting is the same.

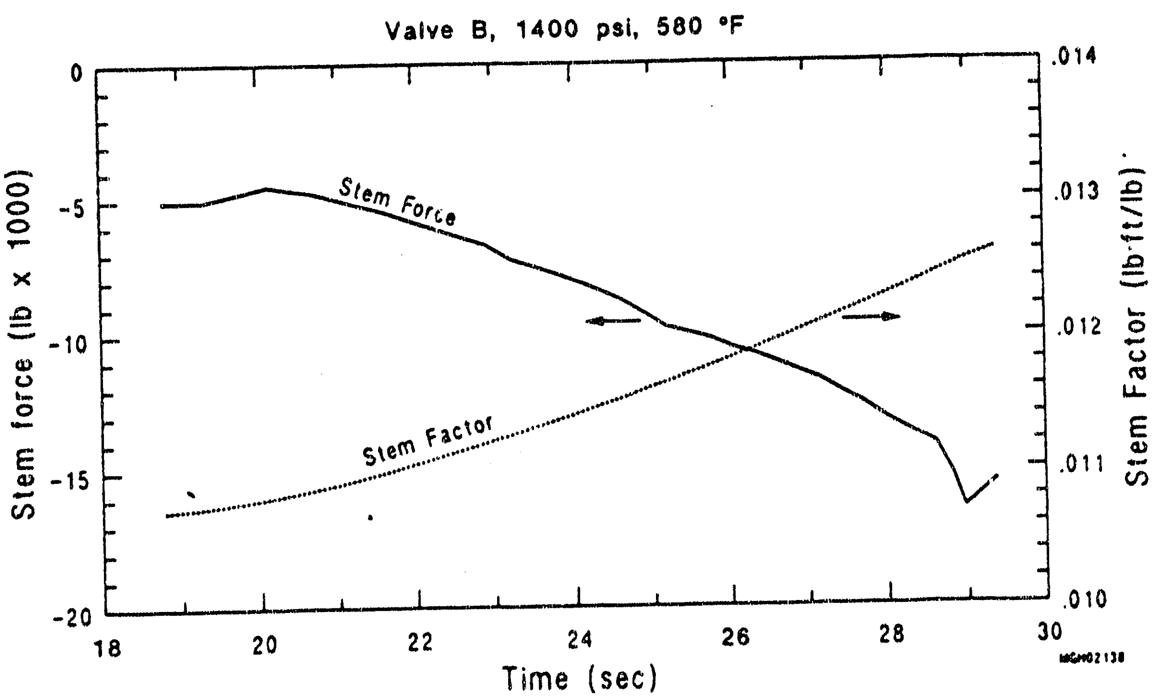


Figure 14. Calculations show that the stem factor increases with load, reducing the available thrust at a given torque.

torque. In this and other stem factor versus load comparisons, it appears there may be a proportional relationship between the increase in load and the increase in stem factor.

CONCLUSIONS

The disc factor of 0.3 typically used in industry is not conservative for either of the valves tested. A disc factor of 0.5 marginally predicts the forces for Valve B for both opening and closing. The response of Valve A is enveloped by the 0.5 disc factor during opening, but not during closing. Today's tools for analyzing valve response to loadings are not sophisticated enough to detect small design differences that make large response differences. Temperature also affects the thrust requirements of these gate valves. These facts justify continued qualification testing of prototypical valves at design basis loadings and point out the need for industry to take another look at the variables in the sizing equation. It may be necessary to add new terms to the equation or to increase the the disc factor to a very conservative number to account for the missing terms.

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

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Comment 1

You have found the friction factor to depend on the degree of subcooling of the gate valve flow interruption tests. We have found exactly the same as you in tests on a parallel slide gate valve when testing to ANSI B16.41 under flow interruption conditions for PWR primary circuit, where the degree of force depends on the design of the valve.

Indeed, the disc can become sucked into the seat, so that during the last critical five percent or so of closure of the valve, the disc has to ride over the top of the seat, and that gives you an apparently high friction coefficient.

In addition, for flow interruption testing qualification for PWR, another feature that may well affect the apparent friction coefficient is the chemistry of the fluid. The reason I say this is that we know that boric acid goes through a maximum friction coefficient at a temperature of about 300 to 350°C.

Now, that additional friction factor, combined with the friction due to working at high temperature steam or water conditions, could well cause the actuator to stall.

Question 1

Are you planning any separate effects testing of force stem factor using threaded stems and varying the thrust on them and matching the input torque to them at all?

Response 1

We have a test stand set up at our laboratory where we are looking into testing motor operators by controlling the torque that has to be overcome, thereby taking at least one factor out of the equation. Then, we have plans for some force type testing, where we take a motor operator and try to drive just an empty stem into a cylinder set up to duplicate some of these flow levels.

At the present time our testing is concentrated on doing similar tests with additional valves and encompassing steam systems, to verify this and to look at other valve designs and see what else is happening here.

This, of course, includes much better instrumentation than we used before, so that we can study the phenomenon of the stem factor.

Question 2

Considering the damage that was done to these valves, isolating flow under these severe high DP conditions, does this raise any questions in your mind of the advisability of the requirement in the new generic letter that plants DP test all their valves and then, after they have successfully completed this isolation, still consider the valves operable?

Response 2

From our inspection of our valve B, the valve that did not chew itself up, I would have no problem with using the valve again. It did not damage itself to the point where I would consider it inoperable.

The hard question comes in whether or not the valve you have is a valve B or a valve A, linear or non–linear. That is the point to be done in some sort of a prototypical test.

A PHOTOELECTRICALLY ACTUATED, MOTOR–OPERATED VALVE STROKE TIMER AND STROKE TIME DATA ANALYSIS AND EVALUATION TECHNIQUES

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ABSTRACT

Questions have been raised about the value of Motor–Operated Valve (MOV) stroke timing in nuclear power plants, and, reevaluations of requirements in this area have recently been done by the ASME O&M Committee Working Group on Pumps and Valves. Quoting an August 12, 1988, submittal to the Committee by W.L. Greenstreet from Oak Ridge National Laboratories, "stroke time data, if made more precise and carefully analyzed, can yield valuable operational readiness and diagnostic information."

This paper presents a battery–powered, photoelectrically actuated timer developed by the Chicago Nuclear Corporation for precision MOV stroke timing. The timer receives input from valve position indicating lights on the main control panel and can easily be used with existing plant procedures. The paper also describes analysis and evaluation techniques to be applied to the data in monitoring valve operability. The techniques described apply the data to specific valve operator characteristics, thereby increasing the quality of MOV operability assessments. Both ac and dc motor operators are considered.

Effective Stroke Time Testing

Valve stroke timing is already required by the ASME Boiler and Pressure Vessel Code, Section XI, and Title 10 of the Code of Federal Regulations. Failures to stroke are being identified through this requirement but partial or progressive valve degradations are not. While the timing is done, why not get accurate data that is suitable for effective operability evaluations and to detect degradations prior to expensive, total failure? This can be done by using an electronic timer with photosensitive inputs to monitor the OPEN/CLOSED lights near the valve control switch. The time measured is the time the motor carries its load between the switches that control the lights.

With the knowledge of a good baseline stroke time and where it relates to the motor performance curve, future stroke time changes can be very useful as a measure of continued satisfactory operation or of degradation. A stroke time increase can be the key for efficiently directing further test and maintenance resources; a small or no change can be an indicator of continued satisfactory operation.

From an evaluation of typical Alternating–Current(ac) motor characteristic curves, it appears desirable to detect small changes in motor running speed and stroke time of approximately 1.0%. Hence, a timer with approximately 0.1% accuracy should be used. For a 10 s stroke time, then, the repeatability of the technique including the timer should be approximately +/–0.01 s. This may not be practical, due to the variations in the light signal as an incandescent bulb heats up, but, repeatability of +/–0.05 s appears achievable and will still allow detecting changes of 1.0% for most valves.

While the timer times "light to light," the Code requirement has. been interpreted to mean "switch to light." For MOVs, this timing difference may be added through the use of a stop watch or could be considered uninteresting in many cases.

The Timer

The timer is a hand held, battery–powered device with a liquid crystal readout that has a range of from 0.00 to 200.00 s. Sensors connected by leads from the timer are placed over the indicating lights, the device reset, and the valve stroked. The coincident condition of the indicating lights in mid stroke, either both on or both off, is detected by the timer and the final stroke time is indicated. A hold feature keeps the time from changing while data is being recorded.

The repeatability of the stroke timing process is not timer limited but is limited by the repeatability of the sensed light level change from one stroke to the next. It appears that +/0.05 s repeatability is readily achievable and possibly down to +/–0.01 s with improved electronic triggering.

To get an idea of stroke timing repeatability, a twenty stroke experiment was performed, ten open and ten closed, each stroke followed closely by the previous one. A 3 in. gate valve was used, actuated by an SMB–000, Limitorque operator that used a 1/3 hp, 220 V ac motor. No attempt was made to separate actual stroke time changes from repeatability affects. The stroke timings ranged from 16.44 to 16.48 s, expressed as 16.46, +/– 0.02 s. Of the first 13 strokes, twelve timings did not deviate from 16.46 s. If the +/–0.02 s range of the twenty strokes are attributed entirely to repeatability, then the percent repeatability error would be:

100 X (+/–0.02/16.48) = +/–0.12 percent. (1)

MOV Stroke Time Relationship to Motor

The theory of the stroke timing process presented herein was developed by the Chicago Nuclear Corporation. It is based on using timing changes to detect changes in motor rpm and relate this to the "speed vs. torque" characteristic curve for the motor. An evaluation is then done to compare the motor torque change to acceptable limits for the motor and operator. Because of a fixed gearing relationship between valve stem travel and motor rotation, it can be said that the mathematical integral of motor rpm with respect to time for a given stroke distance is a constant. For a constant motor running torque and supply voltage, motor rpm is constant, and the integral becomes,

(rpm)(Stroke Time) = K, (2)

where the constant, K, represents the number of motor revolutions required to stroke the valve. If the valve load requirement changes or if the efficiency of the operator changes in converting motor torque to valve stem load, the motor load and rpm will change according to the motor characteristic curve.

Where S1 and S2 are the motor rpms and T1 and T2 are the respective stroke times, then according to Equation 1,

(S1)(T1) = (S2)(T2), and (T2)/(T1) = (S1)/(S2). (3)

From this relationship, it is seen that a stroke time change is inversely related to an rpm change. From this relationship, the motor characteristic curve can be used to determine the change in motor load and to quantitatively assess the changes occurring in terms of available margin in the motor and operator. This is explained further in the sections that follow.

Periodic Testing

Valve load increases, motor degradation, or a reduction in operator efficiency can all result in a slowing of the motor. Without additional information, one can not determine the cause.

Hence, the best approach in using periodic stroke time test information may be in looking for a limit in motor load change without trying to determine the cause until an arbitrary alarm point is reached. This is a matter of choice.

Periodic Test Analysis Example for an ac Motor–Operator. Assume the following characteristics taken from an actual operator motor curve:

Torque (ft–lbs)	rpm	% Change	Comment
0	1800	0	Synchronous speed
3	1750	3	Motor rating
6	1700	6	—
9	1640	9	—
12	1570	13	—
15	1420	21	Rated torque

The percent change indicated is the percent difference from synchronous speed. Normally, the motor running speed should be within the *motor rating* which means the rpm should be between 1800 and 1750 rpm. In this region and well beyond, per the table, a 3% reduction in rpm represents a torque change equivalent to the torque at 100% motor rating.

In this example for instance, if we wanted to detect a 1.5 ft–lb change in motor running load requirement (50% of the torque at 100% motor rating), we would look for a 1.5% stroke time increase. This is a very small change and the ability to relate this to the specific load changes may be masked by other affects such as changes in available motor voltage and other affects.

However, the motor characteristic curve chosen for this example was flatter than typical for many of the motors used in order to better understand the limits of the system. A steeper curve would give larger stroke time changes to evaluate for the same percent change in torque. This is obvious in the dc motor operator analysis, below. Also, a trend plot of the stroke time changes and experience will help in determining the significance of stroke time changes identified.

Periodic Test Analysis Example for a dc Motor–Operator. Assume the following from an actual operator motor curve:

Torque (ft–lbs)	rpm	% Change	Comment
0	2600	0	No load speed
12	1800	31	Motor rating
24	1350	48	—
36	1040	60	—
48	750	71	—
60	275	89	Rated torque

Comparing this data with the previous ac example shows a ten times difference in percent speed change going from no load to rated motor conditions. In the primary range of interest for motor running load the speed changes 31 percent for a load change from 0 to 100% of motor rating. Hence, stroke timing accuracy and repeatability becomes much less of a limitation in this example. For instance, if we want to detect a 6 ft–lb change in motor running load requirement (50% of the torque at 100% motor rating, as for the ac example), we would look for a 15.5 % stroke time increase. This is easily detected and typical of dc motors.

Other Possibilities

Operator Capability Test. If a valve has a significant stem rejection load because of a high system pressure existing during testing, it may be possible to detect the percent of motor load change due to the stem rejection load through measuring timing differences between opening and closing strokes. The efficiency of the operator in converting motor torque to stem thrust could be determined from an analysis of the data. The ultimate operator capability might then be determined by comparing this to the margin left in the motor to supply additional torque to the operator.

Post Stem Packing Tightening Test. If stem packing tightening is required, with the increased timing accuracy it is possible to measure and meaningfully assess the affect of tightening by measuring stroke times before and after.

Automatic Data Collection, Analysis, Trend Presentation, and Storage

Since the initial prototype timer was developed, Chicago Nuclear is proceeding with a laptop computer based timer that will use the timing principle developed, but will store the timing data to a disk, automatically compare the stroke time to reference values, initiate an immediate alarm if appropriate, and will be capable of downloading the data to a host computer for graphical presentation and trending.

CONCLUSION

A stroke timing tool has been developed that significantly improves the accuracy and repeatability of. valve stroke time test data. It can easily be incorporated into existing plant procedures because if already existing timing requirements. Analysis and evaluation techniques are available to take advantage of this increased accuracy to more effectively monitor motor–operated valve physical conditions; and, existing computer technology can be used to automate and efficiently handle the timing data.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

No Questions/Comments

ACTUATOR SETUP AND DIAGNOSTICS FOR PNEUMATIC–OPERATED VALVES

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INTRODUCTION

In the past few years the Nuclear Power Generating Industry has begun to realize the consequences of valve misapplication, improper actuator setup, undersized actuators and/or inadequate maintenance. The issuance of several NRC and INPO information notices concerning valve operation highlights these problems. Recently, the NRC issued Information Notice 88–94, which reports that certain pneumatic–operated valves may be equipped with undersized actuators which may not be capable of providing sufficient forces to properly operate, seat or unseat the valve at maximum design conditions.

Additionally, an industry–wide effort is in effect to reduce packing gland leakage. Many valve manufacturers and packing vendors are offering packing improvements such as with the use of die–formed graphite ribbon packing. It appears that often these packing modifications are incorporated without proper consideration of actuator sizing.

Corrective maintenance on valves to restore them to their original condition is an approach that many valve repair companies and utilities take. This approach addresses the symptoms of problems, but often does not consider root causes of problems. Original actuator sizing and setup criteria are based on estimates (intended to be conservative) of the forces acting on the valve and actuator. These estimates are seldom validated following installation, operation and wear to ensure the valve can perform its intended function.

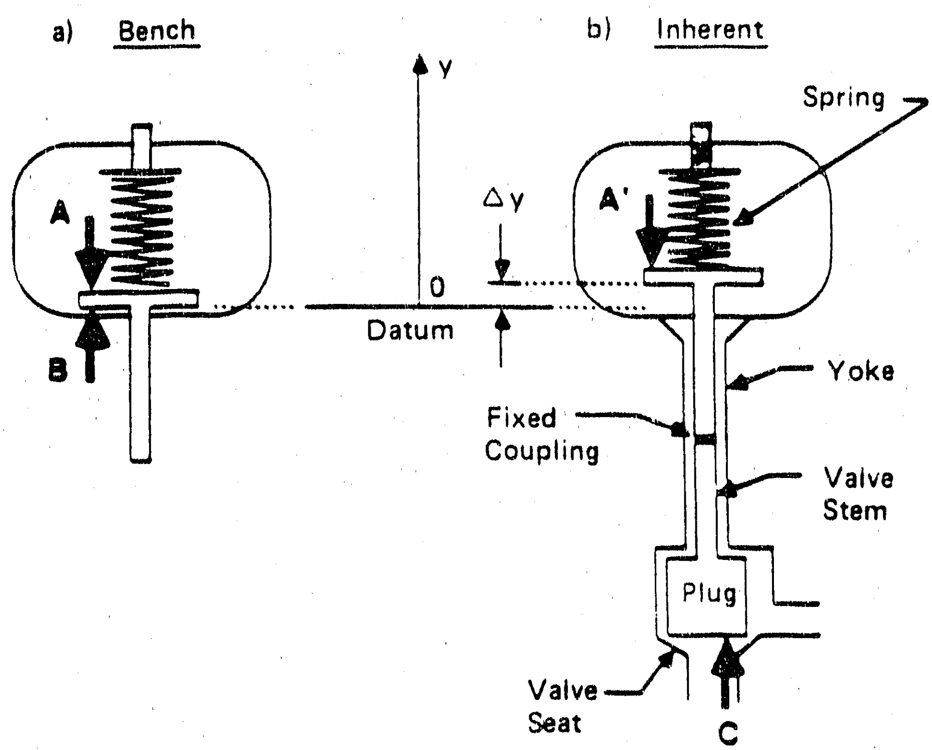
The Solution

A methodology has been developed to determine actuator setup criteria and perform diagnostics on pneumatic–operated valves. This type of diagnostic evaluation on pneumatic–operated valves will ensure that the valve can perform its intended function, pass in–service inspections, and improve valve reliability resulting in an increased plant availability. The methodology used to determine actuator setup criteria considers packing gland forces, seat loads, leakage classifications, system pressure forces, and valve/actuator capabilities including spring rate characteristics.

Basic valve, actuator and systems data are gathered and entered into a computerized valve software data base for future reference and calculations. This program can then be operated on a personal computer to calculate baseline actuator setup criteria for the bench, inherent and installed conditions (see Figure 1 for convention and terminology). A significant feature of this methodology is that feedback from actual measured conditions can be input into the program to validate or correct for conservative estimates used to determine the baseline actuator setup criteria. These criteria are presented in terms of actuator air pressure requirements versus valve position and a signature trace. Additionally, installed spring rates or actuator effective areas can be validated when this information cannot be verified through existing documentation.

To use the actuator pressure data correctly, a thorough understanding of three basic terms is required. The terms "bench", "inherent", and "installed" fully describe all conditions which the actuator will experience. Each of these three terms is discussed below, with reference to Figures 1 and 2.

Bench Actuator Pressure Settings. The "bench" condition refers to the actuator disconnected from the valve. The purpose of setting up the actuator in the bench condition (valve stem disconnected from actuator stem) is to preload the actuator spring, verify the spring constant (K), and verify available spring travel and margin available over the original design settings. The spring preload provides the necessary force to overcome packing friction (so that the valve may return to its deenergized state by spring force alone), to develop appropriate seat load for sealing (shut off) requirements, to overcome the effects of system pressure on the unbalanced area on the valve plug, and to overcome positioner preload. The application of the valve



FORCE TERMINOLOGY

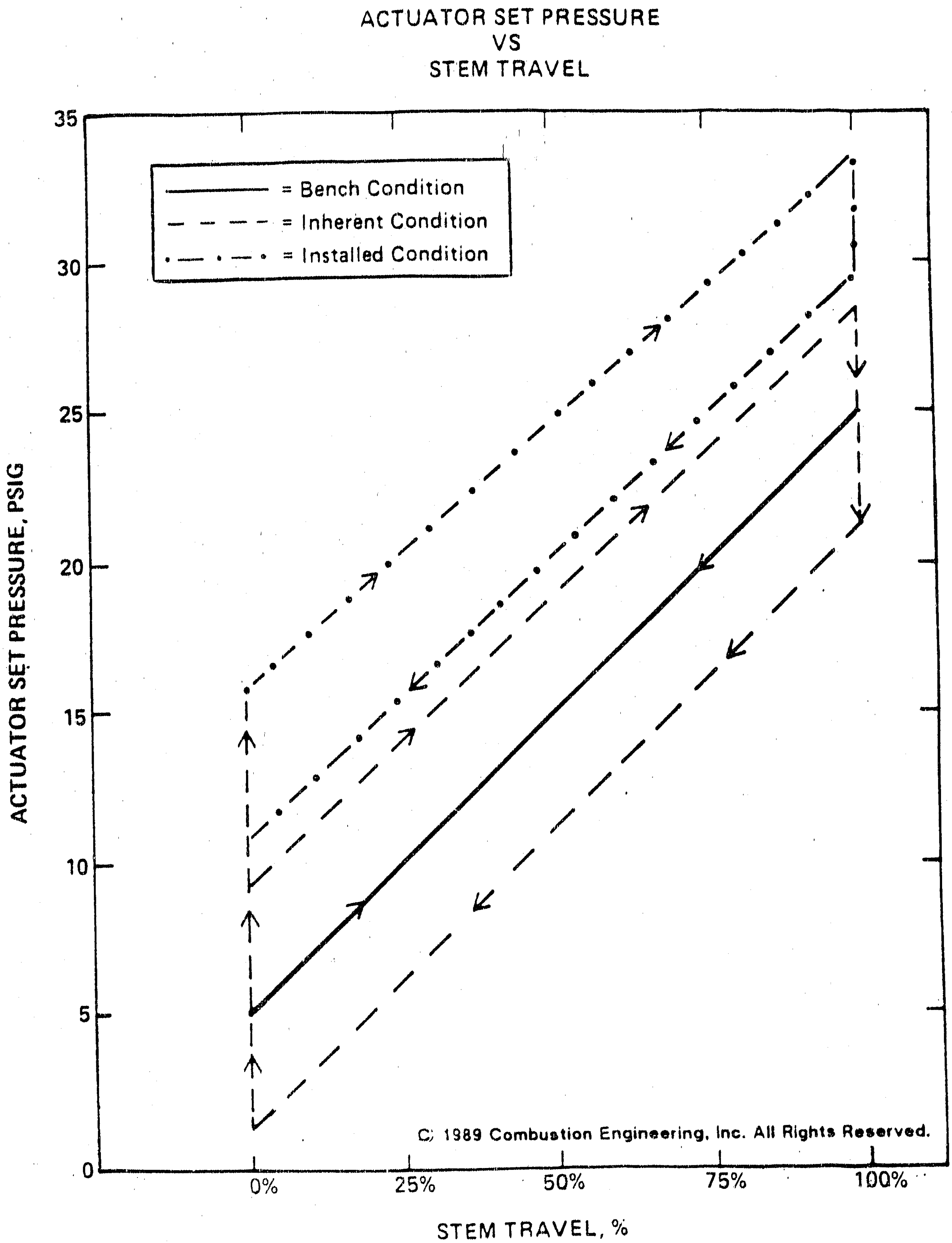
A = Force due to preload of spring acting down onto diaphram plate.

B = Reaction force from actuator casing acting up onto the diaphram plate.

C = Reaction force from valve seat acting up onto diaphram plate through valve stem and actuator stem.

Δy = Small deflection of spring required to transfer force through valve stem and onto valve seat. If Δy is large then preload must be corrected or stem lengths adjusted.

Figure 1.



ACTUATOR SET PRESSURE
VS
STEM TRAVEL
= Bench Condition
= Inherent Condition
= Installed Condition
35
30
25
20
15
10
5
0
ACTUATOR SET PRESSURE, PSIG
0%
25%
50%
75%
100%
STEM TRAVEL, %
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Figure 2.

(e.g., throttling versus isolation service) is an important factor to consider in determining seat loads.

Inherent Actuator Pressure Settings. The "inherent" condition refers to the actuator coupled to the valve with atmospheric pressure on both sides of the valve plug. All spring preload forces have been transferred to the valve stem from off the bottom of the actuator casing as in the bench condition (see Figure 1). As noted in the figure, if only a small deflection is required to couple the valve, no adjustments will have to be made. However, some valve designs include coupling of the valve as a contribution to preload. If this is the case the original preload will have to be corrected for this mismatch. This can easily be done by inputting the actuator mismatch into the data base and recalculating new bench, inherent, and installed settings.

The purpose of calculating inherent actuator pressure settings is to verify proper valve stroke and to compare actual packing friction values with those calculated previously from the data base input parameters. It should be realized that the only difference in these air pressure settings and those calculated in the bench condition should be the additional pressure required to overcome packing friction in either the open or closed direction plus any adjustment made for actuator mismatch. Additionally, it is noted that these factors will be present only when the valve plug is starting to travel (overcoming static friction) or is travelling (dynamic friction). These friction forces result in a hysteresis loop around the reference bench condition (See Figure 2).

The inherent set pressures provide an indication of the pressure demand required from the valve positioner and/or air system with atmospheric system conditions acting on the valve internals.

Installed Condition. The "installed" condition refers to the valve in-line, hot, with system pressure acting on it and the actuator coupled to the valve. The data base provides calculated actuator air settings based on input parameters and the unbalanced plug area.

The actuator set pressures obtained from field measurements for the installed condition allow for both empirical measurement of packing friction with the gland hot and the effect of system pressure forces. These field measurements can be input into the data base to correct for any difference between them and the values calculated. The installed actuator set pressures provide an indication of the pressure demand required from the valve positioner and/or air system at assumed operating differential pressures.

To perform the evaluation of a pneumatic-operated valve and determine the appropriate actuator settings, a common reference point should be established to serve as a foundation for the work. This reference point can be considered the basis for the evaluation and it is important to understand when interpreting the results. Reference points should be established for several important areas including spring available travel, seat load, seat angle, flow direction, packing, actuator preload, actuator mismatch, design bases (e.g., LOCA or MSLB) sizing considerations and the validity of the documentation from which critical data base input parameters were obtained. A thorough understanding of each of these considerations is required to determine appropriate actuator settings.

The pneumatic-operated valve diagnostic software used is a multi-purpose engineering and maintenance tool. It is assembled in modular format so that it may be applied as a stand-alone data base or expanded to perform actuator setpoint calculations and valve diagnostics for a variety of valves. For example, the software is able to calculate actuator set-up for the various combinations of actuator type, valve type, and flow directions, for unbalanced globe valves with or without bellows seals, as described in Figure 3.

One of the key features of the this software is its diagnostic capability. In addition to the air pressure requirements defined by the program for setting up the actuator, there exists the capability of inputting observed actuator performance data. This input data allows diagnostics to be performed on the valve/actuator. Actual spring rate, or actuator effective area, packing friction, and system pressure force are among the information derived from the input data. Air pressure requirements are also compared to design pressures of the various control components and available air supply.

Evaluation of the actuator is based primarily on spring rate, spring available travel, actuator effective area and system pressure conditions. The results of the baseline calculations, utilizing the necessary input parameters, are used to make the final judgement on the actuator. Design spring margin is the single best term used to define the actuator's ability to perform its function. Positive spring margins are desirable, however, excessive positive spring margins may be an indication of an oversized actuator and should be given additional consideration for potential valve damage and control stability, etc.

UNBALANCED GLOBE VALVE
AVAILABLE COMBINATIONS AND
VERIFICATION RUN DIRECTORY

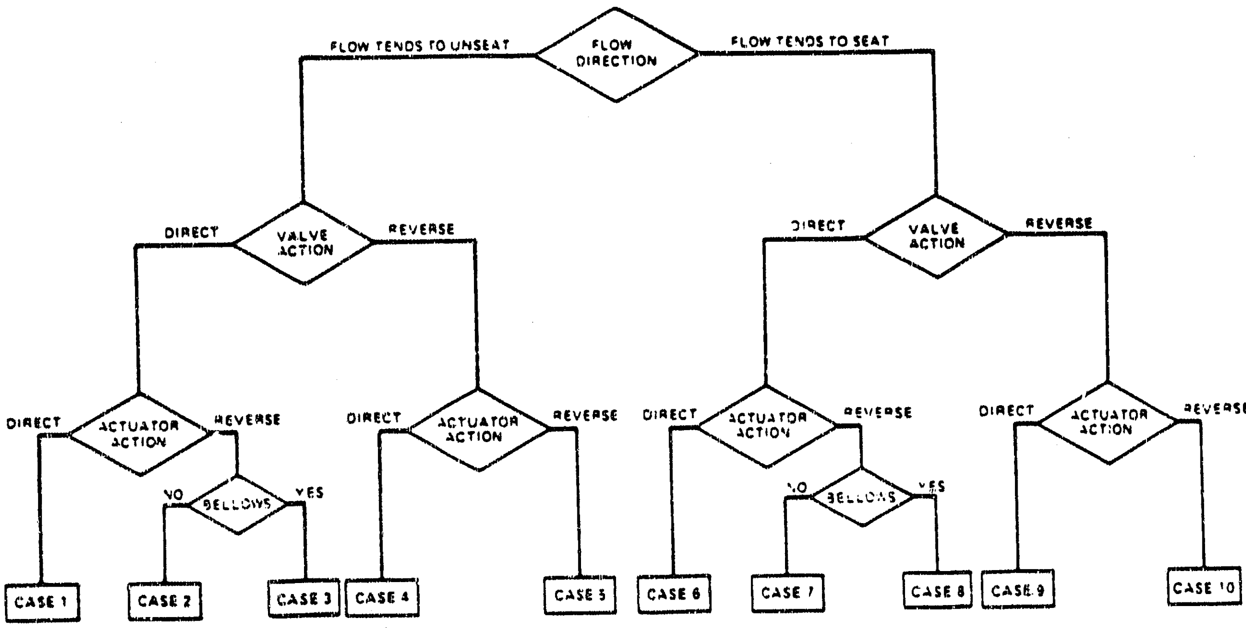

Figure 3.

Available instrument air pressure is also important because actuator set pressure requirements may exceed the maximum available supply. In general, high actuator set pressures provide large spring preloads but impose added demands on the air supply. In these cases where large spring preloads are required, it is therefore more desirable to increase actuator effective area to reduce the air pressure requirements.

CONCLUSION

Actuators for pneumatic–operated valves can be set–up using computer based technology. Diagnostic evaluations which consider packing gland friction, seat load and other forces can also be accomplished utilizing these techniques to determine if the actuator is sized adequately to enable the valve perform its safety function. However, a thorough understanding of the terminology and conventions is required prior to undertaking this type of effort.

Although calculations to determine the appropriate set–up criteria for pneumatic–operated valves can be performed quickly with the aid of the computer software mentioned, the measurement of actual valve performance parameters is currently manpower intensive. An analyzer to automatically develop a signature trace for pneumatic–operated valves is currently under development.

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CONTACT

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ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

No Questions/Comments

AN ENGINEERED INSPECTION PROGRAM TO MEET 1990 AND BEYOND

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ABSTRACT

The roots of ASME Section XI testing of components lie in the very first year of the commercial nuclear industry in the United States. With few exceptions, the content and overall goals of the testing program have not materially changed over the ensuing years. While the testing of equipment considered important to safety has been performed at plants consistent with the requirements of ASME Section XI, there are other methods not yet described in Section XI (or in the recently approved O&M standards) that can provide more effective measures of equipment reliability. The paper defines an argument for the use of advanced techniques of testing applied in an integrated Reliability Centered Maintenance framework in lieu of simply testing safety-related components via ASME Section XI. The paper presents the basis for an integrated testing program that would utilize industry failure data, tempered with plant-specific information, to develop a program of testing specific pumps and valves that would provide a maximum of component data with a minimum of component degradation and at a minimum expenditure in equipment, time, and manpower. The outline of the testing program is provided, along with a sample of the methodology used to develop the scope and content of testing.

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AN ENGINEERED INSPECTION PROGRAM TO MEET 1990 AND BEYOND

The present inservice testing program in use at utilities across the country is based on use of ASME Section XI subsections IWP and IWV, with editions ranging back to 1975. In the past decade, use of predictive and preventive maintenance methods, combined with Reliability Centered Maintenance techniques, has improved the level of knowledge of the condition of plant equipment at many utilities. This increased level of knowledge generally exceeds the information gained as a result of the testing performed per ASME Section XI on most plant components. Unfortunately, utility personnel are presently forced to implement both programs, at increased cost and with little increase in equipment reliability or availability.

To compound this problem, implementation of the Section XI testing has the concern of allowing complacency regarding the condition of equipment tested, assuming that the testing will detect all possible failure modes prior to their occurrence. With the goal of "assessing operational readiness," Section XI (and OM–6/10) is an intermediary role between the technical requirements of ensuring that equipment meets the *current* operational needs (i.e., maintained in an "operable' condition) and ensuring that the equipment *will* meet operational needs at some point in the *future* (i.e., evaluation for degradation that *could* negatively affect operational at some later time). A given piece of safety–related equipment at a nuclear plant can be subject to a wide variety of surveillance testing, preventive and predictive maintenance activities, as well as an optional utility–generated condition–monitoring program. Performance of all of these activities on a specific piece of equipment may indeed verify adequate operation at the current time, but very few activities have the capability of detecting specific degradation resulting in potential failure modes.

CONCERNS OVER THE PRESENT CONTENT OF SECTION XI AND OM–6/10

Presently, ASME Section XI requires, among other tests, an evaluation of the current and future condition of all safety–related power–operated valves via an evaluation of the changes in valve full–stroke time. In the case of valve condition, it is true that we can assess the current condition of a valve based on a measurement of valve stroke time, but it has not been clear that future failures are adequately indicated by changes in valve stroke time. The current NRC concerns regarding the operability of motor–actuated valves underline the level of doubt of stroke time measurements to be an adequate measure of future valve condition. In addition, a recently completed EPRI project has cast additional doubt on the ability for stroke time measurements, taken alone, to provide adequate predictive information regarding potential valve failure.

Based on the type of valve actuator, changes in stroke time may or may not indicate anything about future valve condition. Alternating current motor–actuated valves will not see significant changes in stroke time prior to complete valve failure, due to the relatively constant speed of the actuating device. While it is true that verification of identical ac motor–actuator stroke time will aid in ensuring present actuator operability, it does not provide useful information regarding degradation modes in progress. The typical stroke time data of an ac motor actuator shows a constant stroke time ending in a sudden failure, with no gentle or predictable rise to failure. Indeed the stroke time at failure for these valves is typically infinite, since the valve never completes its stroke. Therefore, measurement of stroke time in ac motor–actuated valves can assess current valve operability, but will give no useful data regarding degradation in progress The reliability of the test goals is then to assess the present status of the operability of the components under test, not to assess or to predict the condition of the valve during the next scheduled test. The present Section XI requirements specifying increased test frequency based on a change in observed valve stroke time will therefore not provide added assurance of acceptable component condition.

Direct–current motor–actuated valves will identify certain failure modes via stroke timing, but certainly the concerns regarding adequate torque switch settings, correct actuator/motor configuration, etc. will still remain.

Air–actuated valves will generally show evidence of some failure modes as a change in valve stroke time, but many cases have shown that slow constant degradation of actuator moving parts is not the most common cause of failure. Improper instrument air system maintenance, operation, or valve actuator maintenance has been much more likely to cause failure of an air–actuated valve than is the degradation that the equipment is undergoing. Therefore a testing program that relies on predicting failure by measuring degradation is not going to identify these system operation and maintenance–related types of failure modes.

Solenoid actuated valves have been a difficult problem for Section XI and the industry to deal with appropriately in the context of a regular surveillance type of test. Based on the control system position indication system, and general operational design of many of these actuators, performing stroke time tests may not verify the current condition of these valves in a reasonable fashion. Operational/functional leak tests to verify both position and leak rate through the valve are one method of ensuring that these valves currently perform their required operational function, but they give no assurances regarding future operability. Therefore, measuring stroke time, even to the extent of ensuring that the valves stroke within 2 s or less (the "rapid–acting" position taken by OM–10 and the NRC), will not provide us with assurance that the valves are ready to perform their intended function.

Check valves provide nuclear industry with an extreme challenge to monitor and evaluate actual valve performance and condition in a reasonable fashion over time. While valve disassembly is one method of performing this evaluation, it certainly is not the method of choice in the industry. By using the "accident design–basis" flow rate method promulgated by the NRC, in combination with a reverse flow leak test to ensure integrity of the moving parts of the valve, reasonable assurance of the current condition of the check valve can be made, but effectively nothing regarding degradation in progress can be discovered. In addition, the concept of passing "partial" flow through the check valve as a method of assessing current valve condition does not seem to provide even a limited assurance of operability unless care is taken when specifying the test flow rate to be used. Therefore measurement of check valve condition via monitoring of flow in both directions through the valve will provide assurance of current valve operability, but not of degradation in progress.

The results of this evaluation of concerns with the content of Section XI inservice testing requirements is that, while some limited knowledge of current component condition can be gained via testing with Section XI rules, most attempts to gauge future performance based on a simplistic condition assessment (e.g., valve stroke time) is not appropriate. Therefore the results of Section XI testing should only be used to assess present condition of safety–related equipment, and not as a method of uncovering component degradation in progress.

Technical Specification Interfaces

The "operability" of a nuclear power plant is most clearly stated in plant Technical Specifications, which outlines the requirements that must be met to continue operation with some level of operable components. The impact of Section XI on these requirements is to allow the licensee to assess the condition of specific plant equipment in order to make a determination regarding the operability of a given system at the current time. In addition to testing required by Section XI, the Technical Specifications will identify the level of acceptable performance of certain equipment when evaluated by test. Levels of equipment performance not meeting these Technical Specification acceptable levels result in an equipment condition which is "inoperable," regardless of the actual running condition of the equipment.

Section XI also requires that equipment be declared inoperable when unacceptable levels of performance are identified. Unfortunately, these levels of performance may not be identical, and therefore may come into conflict (i.e., a component which meets its Technical Specification requirements, but does not meet Section XI). Pumps tested via Section XI often fall into this situation. Section XI also makes an evaluation of the condition of power–operated valves based on a change in stroke time from test to test. In addition, OM–10 evaluates the condition of the valve via a comparison on stroke times from the initial valve baseline. In either case, we are put in the position of declaring a component (and possible a system) potentially inoperable based on the expectation that the component *may* fail (i.e., may not meet its Technical Specification requirements) at some *future* time. This determination is clearly not one to be made lightly, and should be based on more than one data point from one type of test. Even if the component "failure" does not cause plant shutdown, it does require initiating reports and documentation (safety system inoperability, Licensee Events Reports, Limiting Condition for Operation evaluations, etc.) that will have little useful purpose if the "failure" was spurious or random. By continuing this process, we find ourselves "crying wolf" when analyzing test data, with the expected eventual outcome of breeding complacency in response to data taken via Section XI.

In order to prevent this situation, components which meet their Technical Specification requirements but do *not* meet a requirement found in Section XI should be handled in a different fashion from those components which do not meet their Technical Specification minimum acceptable performance standards. Components which fail Section XI requirements but do not meet Technical Specification limits should first have the test results verified to ensure that human error or a random event was not the cause. Once the validity of the data has been ensured, additional, more sophisticated tests should be used to more fully define component condition. At the end of this evaluation process, the component's condition should be assessed in light of the data generated, and a determination as to the operability of the component be made. Keep in mind that during this entire process, the component can still meet its intended function, as identified in plant Technical Specifications.

In summary of this first section, use of Section XI testing alone, without recourse to other methods of examination of component condition, cannot fully define either the operability or the actual condition of the component under test. This means that the evaluation of the true condition of the components under the scope of Section XI will require more rigorous testing/examination/monitoring than is presently contained within Section XI (or OM–6/10) rules. We have discussed the application of Section XI rules in the context of equipment operability versus prediction of failure. The remainder of this paper will tie these concepts together, and establish the necessity of an integrated programmatic response to the question of degradation measurement that is tailored to the specific plant needs, condition, and practices.

An Alternative Inspection Program

Based on the programs presently in place at nuclear plants that could be used to verify the condition of equipment and to trend its condition over time, the most appropriate area for this effort is with the scope of the plant Preventive Maintenance (PM) Program. This program would encompass the condition monitoring aspects of Section XI (and OM–6/10) without addressing the operability of components that is presently the responsibility of the plant Technical Specification surveillance testing program. The end result of this process would be to evaluate the *condition* of plant safety–related equipment and to trend, to the maximum practical extent, those parameters that are indicative of the actual condition of the equipment. This program of upgrading the condition of equipment is based on use of a Reliability Centered Maintenance philosophy that provides the specific goals of availability and reliability for each piece of plant equipment, and then devises a component–specific program of condition monitoring, preventive maintenance activities, and diagnostic testing to realize those specific goals.

The Use of RCM for Section XI Equipment

Historically, a non–systematic approach has often been used to develop preventive maintenance (PM) programs -- plant personnel would sit down and "design" a program based on equipment manufacturers recommendations and any federal requirements that applied. This failure to reflect a systematic definition of PM program content has often resulted in generic program deficiencies, including not effectively and economically realizing the inherent reliability of plant equipment and systems.

Manufacturers develop PM recommendations based on certain assumptions relative to equipment duty cycles, ambient conditions and system parameters. No matter how detailed an equipment specification, the equipment will never be operated precisely as the designer envisioned. The manufacturer is also not aware of the criticality of the item to the operational capability of the plant, therefore his recommended PM program may not be totally appropriate. In some cases, as experienced at some nuclear power plants, and overemphasis of the manufacturers PM recommendations can actually result in a program that is potentially too extensive, not cost–effective, and impacts negatively on ALARA program goals.

This industry–wide lack of a systematic approach to maintenance program development, implementation and revision has resulted in a broad spectrum of PM program practices across utilities. Some organizations have created comprehensive programs that exceed any existing requirements and reflect an obvious corporate management commitment. At the other end of the spectrum exist organizations that are apparently interested only in satisfying minimum requirements. Nonetheless, the need to implement systematic approaches to maintenance program development and revision has been a major topic discussion in the power generation industry in recent years and presents a major objective for many utilities.

This systematic process of PM Program enhancement includes five major phases:

1. Objectives
2. Design
3. Development
4. Implementation
5. Evaluation and revision.

When used to revise an existing PM program, this process logically functions to make the best use of existing information and processes. In many cases only revisions to existing program elements may be needed, if at all. This systematic process will define what revisions are necessary to support a full integrated reliability–based program. Table 1 summarizes the key activities associated with each of the five phases.

The heart of this systematic PM program development/revision process is the Reliability Centered Maintenance (RCM) methodology. RCM offers a systematic step–by–step process for identifying the equipment and tasks for the preventive maintenance program in a logical manner, utilizing all of the information and knowledge that can be brought to bear on the problem. RCM is a proven technique that is the result of 20 yr of research by maintenance planners in the air transport industry. It has been adopted by the military services and is being introduced within the utility industry. RCM provides the methodology that can be applied to a PM program and results in improved equipment reliability, while minimizing overall program costs. The end result of these efforts, when applied to appropriate plant systems and components, can lead to overall improved plant, system, and component availability.

Preventive Maintenance (PM) has many definitions and variations in scope depending on the individual plant operating philosophy. For the purposes of this discussion, we use the INPO definition of Preventive Maintenance as "predictive, periodic, and planned, maintenance actions taken to maintain a piece of equipment within design operating conditions and extend its life." Using this definition, Preventive Maintenance includes equipment condition monitoring and diagnostic activities, periodically scheduled (time–directed) maintenance activities, and activities planned and performed prior to equipment failure, but does not include corrective maintenance (CM) tasks (i.e., repair and restoration activities for equipment that has failed or is malfunctioning).

When properly implemented this systematic approach provides a number of distinct advantages:

1. It provides a logical, consistent, and traceable methodology.

Table 1. Summary of PM program development/revision process activities

Phase No.	Phase Title and Activities
1	Program Objectives
	Evaluate existing program
	Determine PM program objectives
	Develop PM program philosophy
	Identify program performance indicators
	Identify desired program enhancements
2	Program Design
	Identify program functions
	Develop function specifications
	Identify organization needed
	Identify information transfers and reports
	Develop integrated program design
3	Program Development
	Develop/revise administrative procedures
	Develop PM program manual
	Develop enhanced program implementation plan
4	Program Implementation
	Prioritize systems for analysis
	Conduct RCM analysis
	Develop integrated PM task schedule
	Develop/revise PM task procedures
	Develop condition monitoring/performance
	Monitoring programs
	Establish equipment failure tracking program
	Revise training
	Evaluate PM spare parts requirements
5	Program Evaluation and Revision
	Track performance indicators
	Conduct general evaluations of program effectiveness
	Conduct equipment–specific PM task effectiveness evaluation
	Perform proactive program review

2. It provides the capability for using the PM Program to *drive plant availability* rather than merely react to it.

3. It systematically *prioritizes equipment* based on *calculated expected return* and provides *process feedback* to improve the prioritization.

4. It produces a *well–documented engineering basis* for applying PM resources that considers *the importance, functions*, and historical and plausible *failure modes and effects of equipment in its operating environment.*

5. It can include a *plant availability model* that can *predict those actions that result in the*

greatest increase in system and plant availability.

6. It is self-analyzible in that whenever maintenance-related goals are not met, the spec fic cause can be pinpointed for re-analysis and correction.

7. It provides a methodology for *continuing evaluation and revision of the PM Program* on a reactive and proactive basis to ensure that it continues to meets its objectives.

SUMMARY AND CONCLUSIONS

The result of this replacement of Section XI testing with a component-specific PM Program will ensure that each component achieves the desired level of reliability and availability (effectively 100% for safety-related equipment within the scope of Section XI) that was required by the RCM Program objectives for that equipment. In addition, the present resources utilized to track and "forecast" the failure of equipment as required by Section XI would be utilized in a more effective effort, one that provides realistic inspection and testing programs for individual components, and takes appropriate actions based on individually determined component conditions. Another benefit of this type of engineered inspection program is that it is constantly upgraded and improved over time, to ensure that the last advances in diagnostic testing and condition monitoring are appropriately incorporated. The latest technology and methodology improvements available to the industry are reviewed for potential future application to the plant. In addition, inputs from personnel associated with PM program are used to identify improvements related to program operation. This ongoing improvement in a testing and evaluation program is critical to success in maintaining equipment in its desired state. Rather than implementing a testing program and updating every decade, this living program would provide continuous improvement in methods and goals. The single most important factor in any equipment condition upgrade program would remain what it is (or should be) today: an uncompromising dedication to maintain the plant equipment in the best possible condition.

Regulatory approval of this type of program, as a replacement for Section XI testing activities, can presently only be generated via an individual plant Request for Relief, which undergoes extensive NRC review and evaluation before being granted. Based on the recent attention placed on plant maintenance by the NRC, an evaluation by the NRC of the potential benefits and problems in utilizing an engineered, committed RCM program as an alternative to Section XI component testing would seem appropriate.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

You obviously are involved in developing commercial or traditional Section XI pump and valve programs. You are obviously familiar with RCM programs.

How do you see the two related together? Do you see them as two separate entities right now, or will they be brought together, and how will other issues required by other programs be brought into this?

Response 1

This is an important point. How do the requirements generally get implemented at the plant? In many plants this person gets in charge of that one, this person is in charge of that one, this person is in charge of that one, and we have a sort of muddled response to equipment performance questions.

The RCM program, which in the present context is going to be totally separate from the Section XI program. And I think this is the unfortunate reality—the RCM program is going to be in place of that area. It is will be required to incorporate all of those requirements in a unified front, and the unified front will identify exactly what the plant wants to do in response to those equipment questions.

ACOUSTIC/MAGNETIC TEST TECHNIQUES FOR VERIFYING OPERATION OF CHECK VALVES

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ABSTRACT

Recent operating events involving check valve failure in the nuclear power industry have raised specific issues on how to verify operability of check valves by inspection. Philadelphia Electric has had a program to develop test techniques and procedures that can be used to address those specific inspection issues. This paper discusses acoustic and magnetic techniques used to measure check valve disk motion and internal impacts resulting from the disk motion. Full stroke capability, internal worn parts, excessive fluttering, leak tightness are all detected and confirmed by the acoustic and magnetic techniques. These testing techniques can be used to address certain INPO SOER 86–03 and ASME code Inservice Testing requirements.

BACKGROUND

Failures of a few check valves in applications directly related to safe shut down of a unit during nuclear power plant operation led to a review of all check valve maintenance actions and failures. INPO published the results in a Significant Operating Experience Report (SOER) number 86–03, "Check Valve Failures or Degradation," in October 1986. SOER 86–03 concluded that the major causes of check valve failures were primarily misapplication and inadequate preventive maintenance. As a result of INPO's SOER, the utility industry worked with the Electric Power Research Institute (EPRI) and formed a program to address the needs of the industry. In 1988, EPRI issued a report entitled "Application Guidelines for Check Valves in Nuclear Power Plants." This report provided guidelines recommending the use of non–intrusive inspection techniques to verify proper operation of check valves. There now exists in the nuclear industry a strong demand for an economical, viable means of inspecting check valves to verify proper operation.

CHECK VALVE DISCUSSION

A check valve operates by allowing flow in one direction while preventing flow in the other. The check valve has no external moving parts and, therefore, its position and integrity cannot be evaluated with normal visual inspection methods without valve disassembly. Because of recent industry problems with the operation of check valves, it has become essential to assure proper valve performance through valve inspection. The most common check valve is the "swing check". Two other types are in use but are not as common—the "tilting disk" and the "lift check". The following discussion focuses on the "swing check", but the techniques described apply to all check valves.

The "swing check" valve is comprised of (Figure 1) the valve body, valve bonnet, hinge pin, hinge arm, disk, disk washer, disk nut, disk stop, and the disk seat. When the valve is in good condition in a properly designed use, the disk closes tightly against the disk seat preventing back leakage. When the valve opens, tight clearances between internal parts allows proper disk motion and prevents excessive wear conditions. The valve disk assembly will contact the stop during the opening and should come to rest on the stop.

A valve with internal damage or in an inadequate design application may not close tightly against the disk seat and back leakage may occur. When the valve opens, wear on the hinge pin may cause the hinge pin to rattle, wear on the hinge arm may cause the arm to rattle, and wear on the disk nut may cause the disk to wobble. Insufficient flow causes the disk to fluctuate continuously and results in excessive wear on the hinge pin. If those fluctuations include impacting on the valve stop, excessive wear will occur on the disk nut and hinge arm assembly.

An inspection technique for check valves should be able to detect the following conditions:

- Disk Position
  - Stroke time
  - Fully open
  - Fully closed
  - Intermediate positions
  - Flutter magnitude and frequency
  - Contact with seats/stops

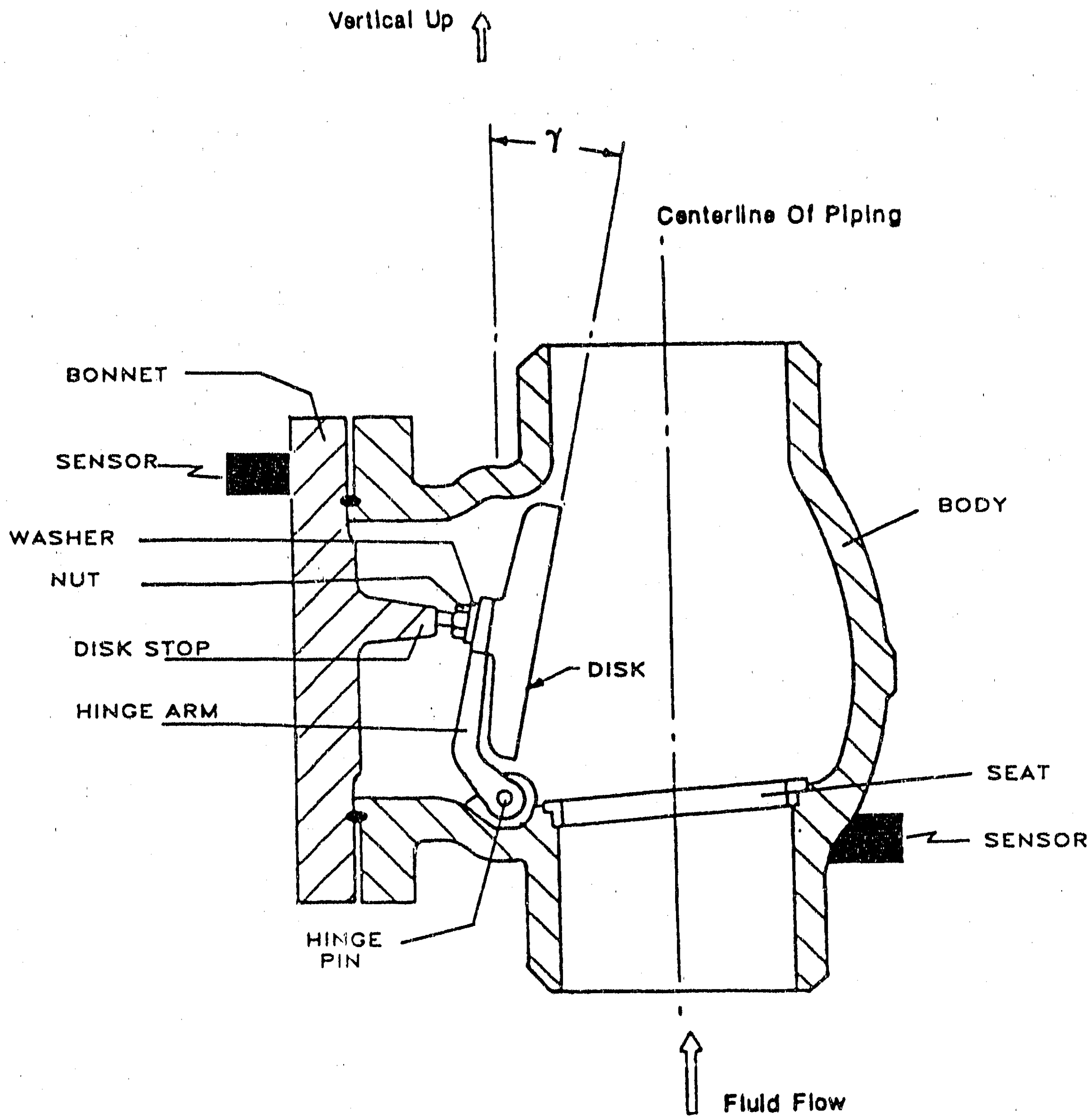

Figure 1. Swing check valve.

- Wear of Internal Parts
  - Hinge pin/bushing
  - Seat/disk facing surfaces
  - Disk to hinge arm connection
- Loose or Missing Internal Parts
- Seat Leakage
  - Presence of
  - Rate

ACOUSTIC INSPECTION TECHNIQUES

Acoustic techniques involve the detection of structural–borne noise emanating from the internal workings of the valve. The technique involves the use of a piezoelectric crystal sensor, such as an accelerometer mounted on the valve. All structural–borne energy waves are detected by the acoustic sensor. Back–leakage through the closed disk–disk seat area causes broadband frequency energy waves to be input to the valve structure as the fluid flows between the disk–disk seat surfaces. These energy waves are detected as a continuous energy level higher than the normal non-leaking condition. Therefore, back leakage is detected.

Worn internal parts create abnormal clearances allowing relative motion between those parts resulting in transient impact energy waves being input to the valve structure. These relatively low energy impacts are detected by the acoustic sensor as the disk assembly moves. Therefore, worn hinge pins or hinge arms, and loose, wobbling disks will be detected by the acoustic techniques.

A fluctuating disk that does not contact the disk stop or disk seat inputs no energy waves into the valve structure and, therefore, is not detectable by the use of acoustic techniques. Once the disk makes contact with the disk stop, energy waves are impacted into the valve structure and can be detected by the acoustic technique. The continued impacting of the disk to disk stop is discernible as discrete impact signatures with the acoustic technique. The valve disk closing against the disk seat is also detectable as the disk impacts the disk seat. Normal closure is seen as a single discrete event while valves with worn parts and excessive clearance provide signatures that indicate multiple impacts as the seat wobbles into a steady state condition.

There are a few disadvantages to utilizing acoustic techniques alone for analyzing check valves. First, although the indications can be detected by the acoustic sensor, the diagnosing as to which problem the acoustic signal is indicating is difficult. For instance, impacts caused by worn hinge pins or hinge arm and several impacts of the disk to disk stop are difficult to differentiate. Also, impacts created by worn parts as the disk fluctuates could be misinterpreted as disk to disk stop impacting. Another disadvantage is that no acoustic indication is available when the disk is fluttering without contacting the disk stop or disk seat. It is also possible for the entire disk to be missing with just the hinge arm remaining and get acoustic signatures that might be misinterpreted as disk to disk stop impacting as the hinge arm fluctuates and impacts the valve internals.

MAGNETIC INSPECTION FOR DISK POSITION INDICATION

The placement of a permanent magnet on the disk or hinge arm will provide a varying magnetic field as the position of the disk changes. The magnetic field strength (in Gauss) as measured from a stationary point outside the valve is proportional to the position of the disk.

A magnetic field strength sensor, placed exterior to the valve in proximity to the location of the disk stop, would detect the changing field strength as the disk was in motion. The magnetic field strength sensor would always indicate the position of the disk.

The technique involves locating a permanent magnet on the hinge arm/disk assembly with the North/South axis normal (perpendicular) to the disk. This assembly can be accomplished by epoxies, metal "putties", welding, or creating a magnetic washer for under the disk nut. A strong magnetic field in terms of magnetic moment is preferred. The magnetic field strength of the magnet measured at either pole should be greater than 2 kilogauss.

The magnetic field sensor is a Hall Generator consisting of a ceramic crystal such as indium arsenide, which has a transfer function of MV/Gauss. The actual millivolt reading obtained from the sensor/electronic system once calibrated would indicate disk position. When the sensor is mounted on the valve bonnet, the background magnetic field must be zeroed out in order to indicate disk movement. This accounts for changes in local magnetic field strength due to temperature changes, relative position of permanent magnet to sensor changes, etc.

Knowing disk position will allow for limited diagnostics of check valves. A fluctuating disk will be evident using the magnetic technique whether or not the disk is tapping against its stop. If the disk is tapping against the stop, it may or may not be evident from the magnetic technique depending upon the change in motion upon impact with the stop. Proper seating upon closing would not be evident using only magnetic techniques although the position of the disk would indicate closed. Back–leakage, worn hinge pins, hinge arms, and disk nuts would not be evident using just magnetics.

Acoustic/Magnetic Inspection Techniques

Acoustic inspection techniques are coupled with magnetic inspection techniques to form a system that clearly diagnoses deteriorated conditions within the check valve. The magnetic device indicates the position of the valve coincident with the detection of an acoustic signature indicating a valve condition. With the knowledge of the valve's position at the time of the detection of an acoustic event provides the necessary discernment required to analyze the diagnostic meaning of the acoustic events.

Figure 1 shows where the Hall generator/accelerometer sensor is located for inspection of the check valve. Figure 2 shows the Hall accelerometer/dual sensor. This sensor contains both the piezoelectric crystal accelerometer and the indium arsenide crystal Hall generator. The sensor is attached to the valve bonnet with a bonded mounting shoe. Figure 3 shows a schematic of the signal processing equipment (Liberty Technology, Inc.). The field unit provides signal conditioning and signal recording capability. The acoustic channel has a frequency response of 500 Hz to 10 kHz. The magnetic channel has a frequency response of 0 to 10 Hz. Test technicians will spend a minimum amount of time at the valve attaching the sensor and recording signals. Once the data has been recorded, it will be played back into the analysis unit. The analysis unit provides the capability to diagnose the condition of the valve.

Acoustic/Magnetic Inspection Results

Figure 4A shows the response of both sensors to a check valve closure. On the acoustic trace, the flow noise is seen to decrease followed by the impact signature of the disk making contact with the seat. The magnetic signature of the closing shows the magnetic field strength decreasing to the seat position.

Figure 4B shows the trace of a check valve. closing with several disk bounces and improper valve seating. The acoustic signature shows flow noise up to the point where the disk impacts the seat 3 times followed by substantial leak flow noise indicating either a degraded disk or an improperly seated disk. The magnetic trace shows the disk closing with a bounce signature present.

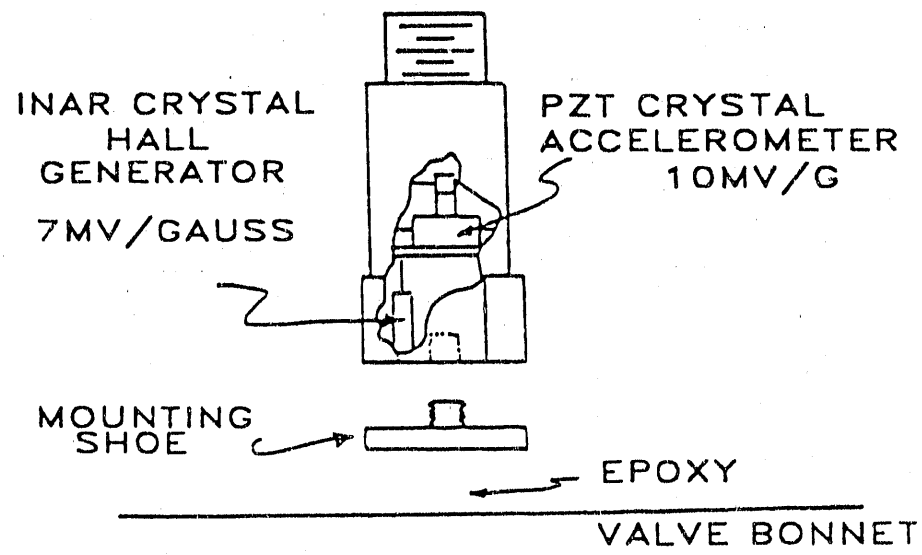

Figure 2. Hall accelerometer dual sensor.

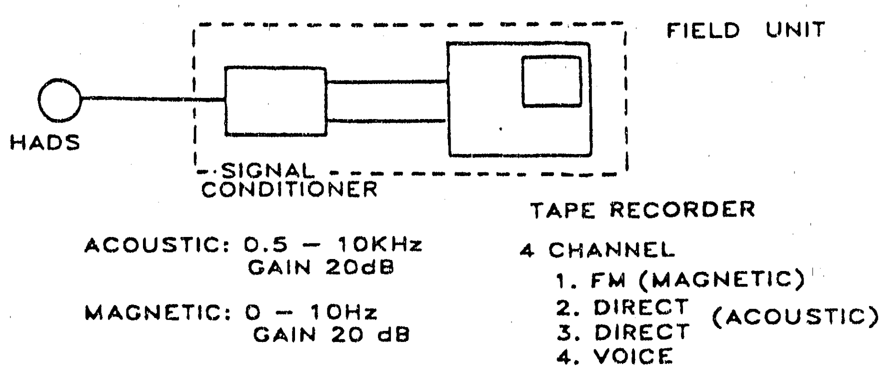

Figure 3A. Signal processing equipment–field unit.

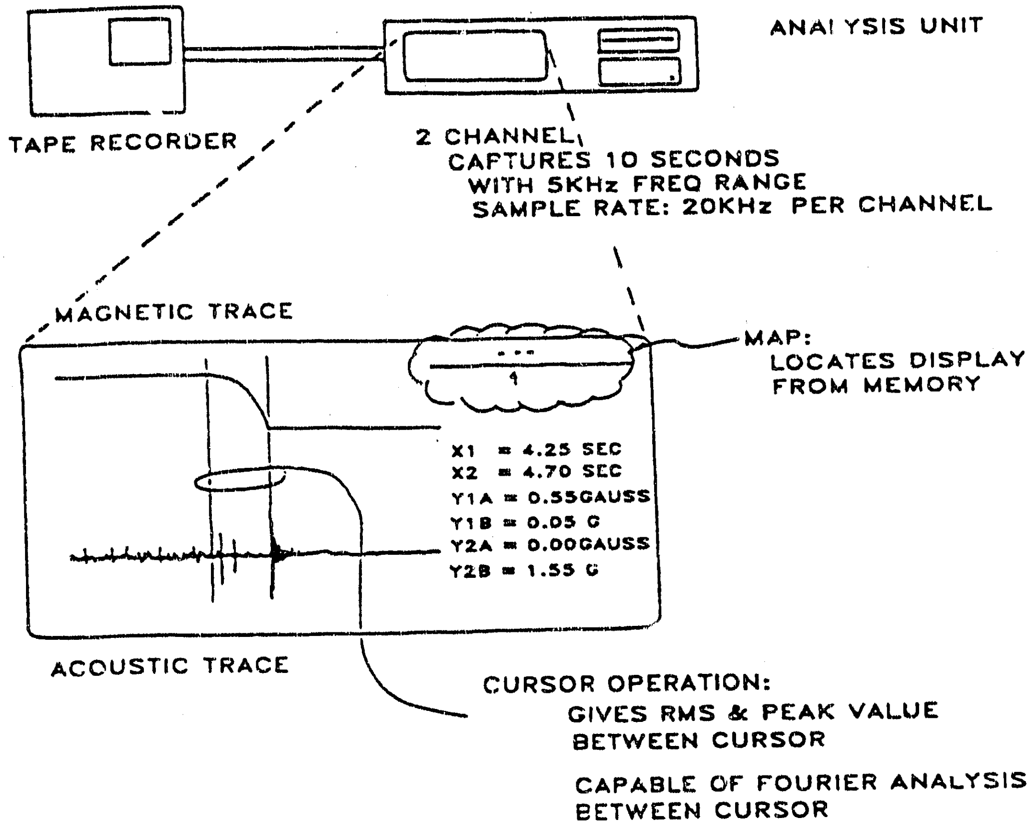

Figure 3B. Signal processing equipment–analysis unit.

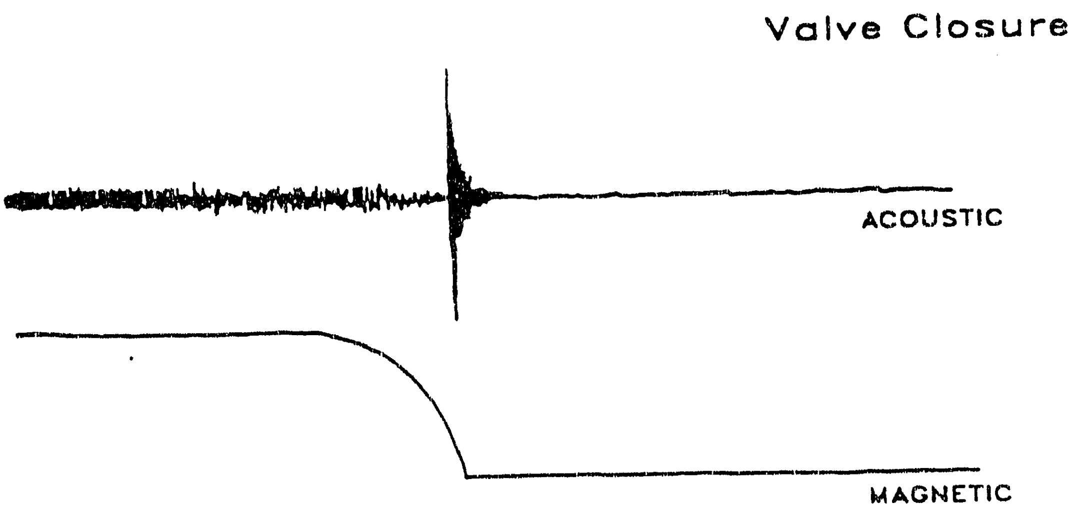

Figure 4A. Check valve closure.

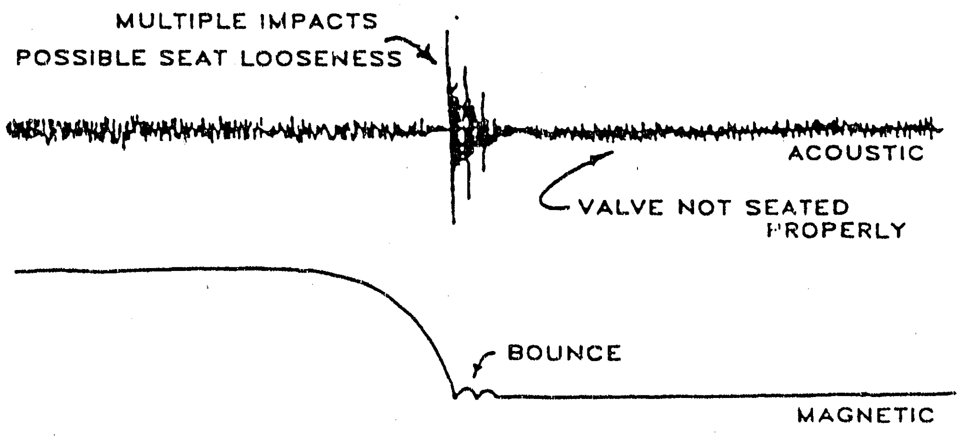

Figure 4B. Check valve closure.

Figure 5A shows the traces for a check valve opening. The acoustic signature begins to show flow noise followed by impact with the disk stop and consequent flow noise. The magnetic trace shows the valve opening with the disk coming to a stop at the impact with the disk stop.

Figure 5B shows a valve opening with possible hinge pin wear or excessive looseness. The acoustic trace shows several small impact signatures prior to the disk–to–disk stop impact. These precursor impacts are a result of the movement due to excessive clearances within the mechanical assembly of the check valve. This indicates possible hinge pin wear or excessive looseness.

Figure 6 shows a check valve opening in which the disk is unstable causing excessive fluttering with impacts between the disk and disk stop. The acoustic trace shows multiple impacts consistent with the magnetic trace indicating fluttering.

In the development of these techniques, reviewing acoustic traces separately did not provide complete diagnostics of the internal workings of the valve. Figure 7 shows an acoustic trace where multiple impacts occur. Diagnosing this acoustic signature alone would indicate several possible conclusions.

- Disk–to–Disk stop impacting
- Disk fluttering with hinge pin wear or looseness
- Disk–to–Disk seat impacting
- Any combination of the above,

Figure 8 is an acoustic trace with no apparent impacting. Diagnosing this trace without the corresponding magnetic trace would lead to two conclusions. Either the disk is fluttering with no impacting or the valve internals are not moving. In the case of the acoustic traces of Figure 5 and 6, the diagnostics can be more accurately focused by utilizing the additional magnetic traces.

The techniques can be used to address INPO's SOER 86–03 which requires the use of an inspection method for preventive maintenance of check valves. They can also be used for ASME Inservice Testing on valves requiring test verification that the check valve has full travel capability from closed to full design flow or full open positions.

SUMMARY

Recent operating events involving check valve failure in the nuclear power industry have raised specific issues on how to verify operability of check valves by inspection. Acoustic and magnetic techniques used to measure check valve disk motion and internal impacts resulting from the disk motion will be capable of meeting the needs of the industry. Full stroke capability, internal worn parts, excessive fluttering, leak tightness are all detected and confirmed by the acoustic and magnetic techniques. These testing techniques can be used to address INPO SOER 86–03 and ASME code Inservice Testing requirements.

Valve Opening

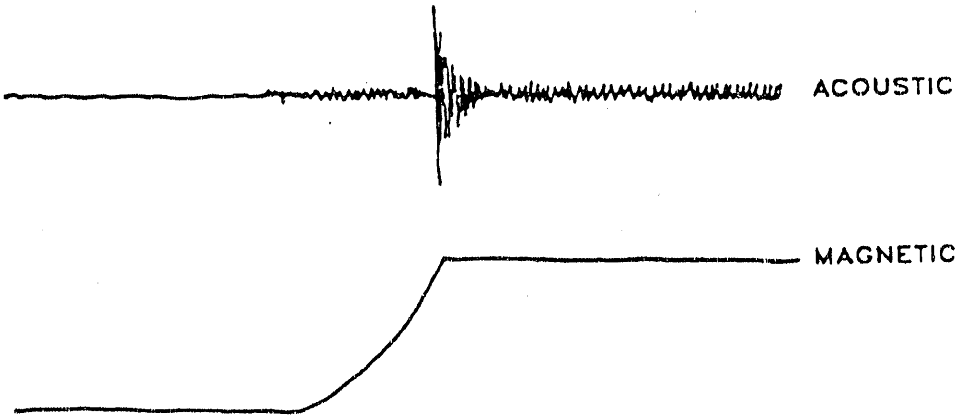

Figure 5A. Check valve opening.

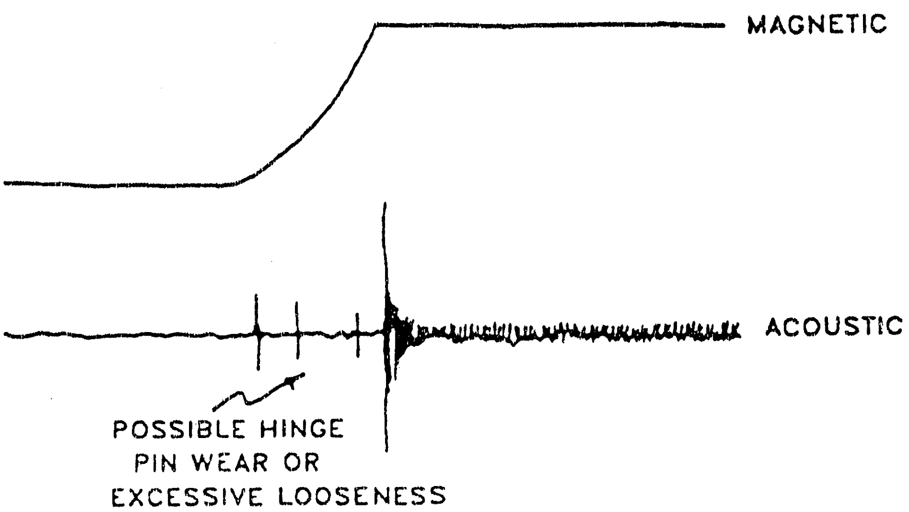

Figure 5B. Check valve opening.

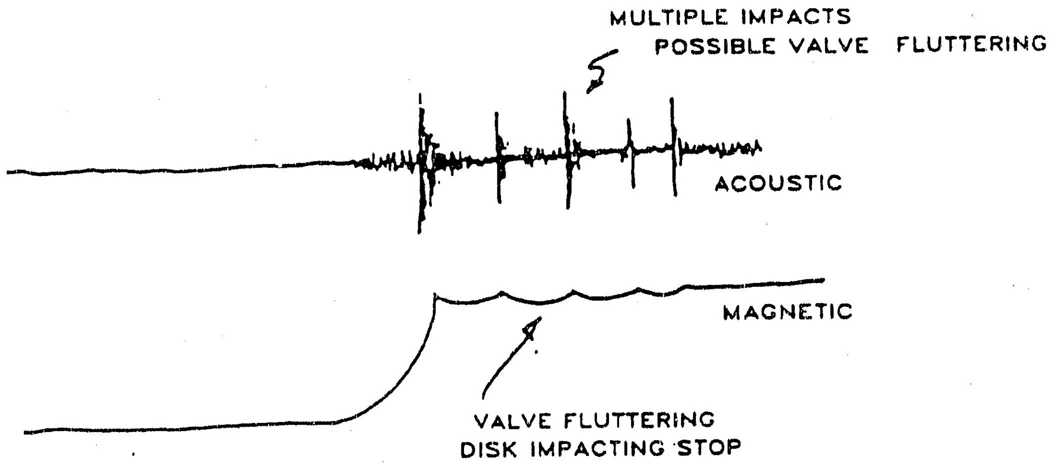

Figure 6. Check valve fluttering.

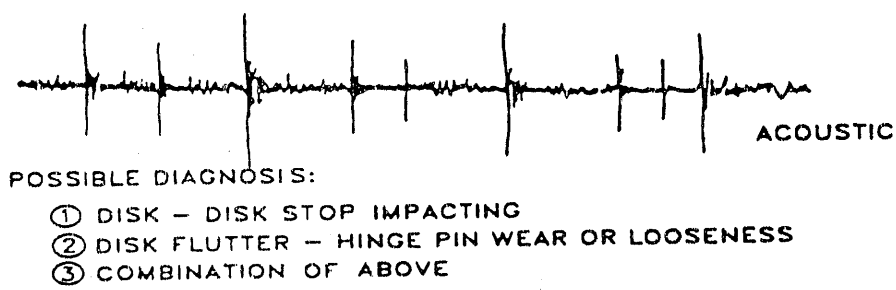

Figure 7. Acoustic trace of multiple impacting.

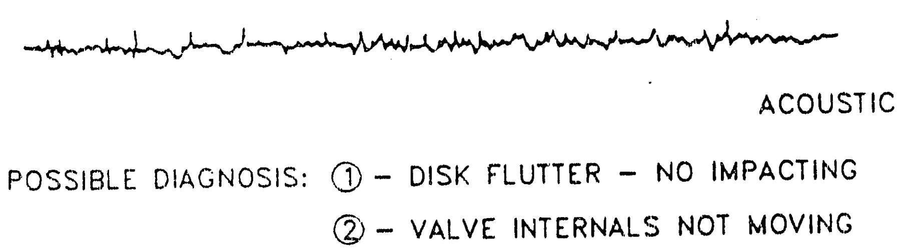

Figure 8. Acoustic trace with no apparent impacting.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Has any work been done on plug type check valves since the carrot test; and if so, how successful has that been?

Response 1

No, there has not been. Most of our work has been with the swing checks.

The acoustics—what I presented to you today—we have been doing in the field. The magnetic part is a development effort that has been in the lab and is just now about to get out into the field.

Comment

I would like to partly answer the question that was just asked. We have stuck a permanent magnet into a ledge gate valve and found that we could see the motion of the ledge coming up and down. So, I see absolutely no reason why a distant look would not show the same results. The work we did on the permanent magnet installation was done under a project for the NRC nuclear plant aging research program. In case the question would be asked, we have checked the magnetic flux indicator with a brass, a stainless steel, and a mild steel valve. The mild steel valve was a 10–inch swing check valve with a one–inch thick flange. We were able to easily detect a magnetic flux on the outside of a magnet placed inside, so that the limitations associated with the use of a magnetic steel could be overcome.

As a matter of fact, in some cases with the steel, we found a better signal because the ferrous material collected the magnetic field and focuses it for you. Whereas in a stainless or a brass valve, there is just no focusing.

Question 2

Have you had any problems with check valves in parallel and monitoring both of them, as far as the feedback?

Response 2

Presenter The answer is no. There is a problem, but it is not a severe problem. Let me explain. There are two sensors on the same valve. There is an impact. You can see the amplitude difference within the same valve, which is probably no more than 18 inches of material.

You can see the attenuation between the two signals. Now, if you were to go those 18 inches through another flange over two feet to the next valve, you will find that one signal really does not make it over to the other one, as far as impact. What you do get is increased background noise from perturbation in the system, another check valve causing the background noise, which is what will be the oscillating and causing the perturbation in the stream.

We found that it does not really affect the diagnostics. There is an effect, but it is not a severe effect.

Attendee Would the leakage then be harder to check for one particular valve?

Presenter If they are very close together, one is leaking severely and the other is not leaking at all, some of that noise might make it over to the second valve.

But the same ratio of attenuation occurs from one valve to the next, so it would have to be a very strong signal to affect a nearby valve. I can not say that they do not, so my answer to you is they have an effect but I think it is a very minimal effect.

Here is what you do if you have two valves, you put two sensors on, and you see this signal. Look at the one that is okay and the other one is leaking, and test both of them before this one started leaking to get a baseline. The next time you find that one signal is increased or place the sensor over here to find that this one has really increased, so this one gets fixed first. Then you repair that one, and a good logical decision would be just to fix that one and see what you have afterwards.

Presenter We do not attempt to power brake the magnetics so that you essentially have a position indication that, each time you go back, you would expect the same kind of value.

What we expect is to have is relative motion. If it is moving and hitting another stop, the abrupt changes in the magnetic signal are what we are after.

To power brake the thing is the next generation of technology development, so our intent now is not to calibrate these things to indicate position forever. It is to be a relative motion.

ADDITIONAL CONTRIBUTION TO THE PUMP TESTING SESSION ON WEDNESDAY, AUGUST 2
MCP–SHAFT CRACK DETECTION IN ISAR–2 PWR BY FREQUENCY–SELECTIVE VIBRATION MONITORING

R. SUNDER AND D. WACH
GESELLSCHAFT FUR REAKTORSICHERHEIT (GRS) MBH. 0–8046 GARCHING
R. HEINBUCH AND J. IRLBECK
BAYERNWERK AG, HEAD QUARTERS, D–8000 MUNCHEN
FEDERAL REPUBLIC OF GERMANY

SUMMARY

Mid of December 1988 the utility operating the ISAR–2 PWR decided to shutdown one of the four main reactor coolant pumps (MCPs) for a non–planned inspection. At that time, the overall shaft vibrations had reached only 0.08 mm, distinctly below the first alert level of a standard vibration monitoring system. The reason for this decision were abnormal trend courses of several shaft vibration frequencies, which had been monitored and provided by a novel computer–based surveillance system, the condition monitoring system (COMOS) developed by GRS. On January 8, 1989 the plant was started again to power operation after a MCP shaft replacement was finished. During the following more detailed inspections, a fairly progressed shaft crack (80%) was detected below a sleeve in the range of the lower shaft bearing. Without these intensive inspection measures, the crack could have grown eventually to a prompt shaft rupture during ongoing plant operation.

The full paper is being published in the technical journal

Power Plant Engineering No. 4, August 1989.

During the ASME/NP C–Symposium the highlights of this recent successful 80% crack detection are presented and explained by *V. Bauernfeind* of GRS Garching.

ATTENDEE – PANEL QUESTIONS AND ANSWERS

(EDITED; ATTENDEE QUESTIONS AND PANEL/AUTHOR RESPONSES NOT IDENTIFIED BY NAME)

Question 1

Why did you institute a shaft crack detection program? Had you been failing cracked shafts previously?

Response 1

Presenter Yes. It was two events in this case, one nuclear power plant in Switzerland and the other one in Germany.

Attendee Where was the idea conceived? How did you get this idea?

Presenter From previous work that was done in KSB. After this shaft crack, we had to do something for vibration monitoring. They had a system for the overall vibration monitoring of the primary circuit.

They asked us whether we could do this job with the system. It was quite easy to do. We had only to put the signals on the system and to make some changes in the software to take the signals and to make this analysis.

They did not have to install new sensors. They did not have to install a new system. They used the system they had for vibration monitoring and made this Mode 1 operation every 90 minutes, two signals for one of these pumps or another.

Note The remainder of the questions and responses in this session were inaudible, and therefore, have not been included.

CLOSING REMARKS

CLOSING REMARKS

LARRY CHOCKIE
CHOCKIE CONSULTANTS

From the ASME perspective and from the Board on Nuclear Codes and Standards, I did encourage our Board to take a positive vote to establish this symposium, co-sponsored by the NRC and the ASME.

By the measurements that we normally establish in how well a symposium such as this has been conducted, we look at the attendance, the facilities, the quality of papers, and the interest in the presentations.

Attendance has been excellent.

Facilities, we've had a little difficulty with the audio on a couple of occasions, but otherwise facilities were also excellent.

Quality of papers. You each have a copy of them in front of you. They are excellent also.

And of course, the interest in presentations. Now, what we had here is something you seldom see. Usually in a symposium of this type two-thirds of the audience is out in the halls exchanging business cards and talking about other subjects. Not this one. They were all in here listening to the papers. So the conclusion is, this symposium has been excellent. An outstanding success.

A couple individual points, again from the perspective of an officer of the ASME Board. First, we met on this subject and encouraged setting up this symposium to help those individuals who are not getting the message directly by participation on the Code committees themselves.

Second, we would like to encourage the attendance at our Code meetings, both O&M and Section XI of the Code, we encourage your participation. ASME does offer due process, you can get your questions answered through the formal ASME process, but that is rather time consuming. You can do a lot better by going to the meetings themselves, and I encourage you.

Thirdly, I would recommend and encourage continuing this symposium as a series, to bring the users, the Code developers, and the regulatory and enforcement personnel together to continue to iron out the problems.

Now, taking the advantage of this forum, I am going to give you a couple of personal perspectives from some 35 years in Codes and standards development confining my remarks to pump and valve testing.

The process of Codes and standards development is evolutionary. Things do change. Goals change. The ASME Boiler and Pressure Vessel Code is a safety code. It was designed to provide assurance of the integrity of that nuclear reactor pressure vessel.

I am observing a trend that we are performing tests that will enhance predictive maintenance activities. That is good. If pump and valve tests will provide that information, that is well and good.

I am also observing that the trend is to include pumps and valves in the testing programs required for plant operation. That also is good.

But I caution: Do not lose sight of the ASME Code's goal, to provide a high degree of assurance that the pumps and valves necessary to shut down the reactor, mitigate the consequence of an accident, are tested for that purpose—that they will operate when called upon.

And as we move to include the new population of pumps and valves, those for plant operation, and programs to assist predictive maintenance, I would advise extreme caution. Do not mix the operation and maintenance programs with the safety requirements of the Code.

It is very easy to put things in the Code, or the programs to meet the Code in your plant, but if they do not work, it is difficult to take them back out.

As a former officer of the ASME, I would like to thank Tad Marsh of the NRC and Kevin Ennis of the ASME for both their technical and administrative guidance, which made this symposium a success.

Finally, the next O&M meeting is in Kansas City, the week of September 11th. I personally invite you. Thank you.

CLOSING REMARKS

JOEL PAGE
U.S. NUCLEAR REGULATORY COMMISSION

I've broken this Session 1 into three parts, as follows:

The first part was put together by John Zudans and Larry Sage. They spoke from the White Paper, which is a document that was put together by the O&M Working Group on Pumps and Valves. The White Paper explains the differences between OM–6 and the existing ASME Section XI, Subsection IWP requirements.

The notable changes pointed out by John Zudans were an increased emphasis on vibration testing, with less emphasis on hydraulic criteria, and he noted that there has been a notable change in the scope. That is, the OM–6 scope requires testing pumps based on their functional requirements, as opposed to selecting pumps based on ASME Code classifications.

Larry pointed out in the details of the vibration criteria that we are asking people to monitor all the accessible bearings on the pump in all the relevant directions. We feel that is very important, as opposed to monitoring only one bearing in one direction.

He also pointed out that we are emphasizing the use of velocity versus displacement testing, and that the Task Group on Vibration Monitoring put together a two–phase approach; with the first phase based on a change from reference values, and the second based on absolute values.

For the second part of session, I have grouped together the presentations made by Kevin Guy and Bill Cleveland. They addressed both testing of non–nuclear and nuclear pumps. The amazing part was that I think they both said very similar things.

The two most interesting aspects they presented were the importance of predictive diagnostics and the importance of management commitment. Where you have the management commitment, you have an excellent pump IST program.

They discussed the use of filtered and unfiltered vibration; the importance of customized lubrication analysis; bearing temperature measurements and trending; and when all these are included, although it would not be realized overnight, you would see a cost effectiveness in three to five years.

The last part was of Session 1 was put together by Patricia Eng and myself. Patricia discussed the inspection aspects of IST. She discussed the IST inspection module and many of the findings she found during her regional experience. Pat also pointed out that there were a lot of inconsistent practices being used.

She tabulated many Region III violations, and she pointed out what she considered to be very good practices for inservice testing, including vibration spectrum testing and lube oil analysis. But I thought the thing that stood out over and above her points was the involvement of a system engineer in the IST.

When I presented my personal perspectives, I felt that our overall goals in the past ten years have been to optimize inservice testing. That is, to take meaningful data, to use predictive analysis, to maintain margins, basically not to experience operational or test failures, to not forget the importance of exercise of the component, and to help the working group by providing good data; something that we can use for future development of this code.

I had a few specific examples of what I'd like to see in the future. I think we should improve the quality of pump vibration testing instead of just increasing test frequencies. Also, I think we do need to add vibration flow instrumentation where needed, even into the recirculation loops.

CLOSING REMARKS

TOM HOYLE
WASHINGTON PUBLIC POWER SERVICE SUPPLY

As I look over this session, after I have a few minutes to reflect on it, I think the message that stood out to me was that we must be approaching testing from an engineering standpoint. The common thread of continuity that each author stressed was that testing must be done in a manner that looks at the overall system, not only the pump, but also the system we are testing.

Codes and Standards specify minimum requirements. What we are talking about in many of these cases is effective predictive maintenance and all of the industry to be more effective in the testing.

The first paper in the session, by James Healy, discussed pump reliability. He stressed that we must look at the pump and the system together.

He discussed minimum flow testing and that, really, it is undefined. We need to look at it carefully and be careful of the results from minimum flow testing.

He also stressed a point about designers who are maximizing efficiency and, therefore, minimizing margin. He feels that margin is out there and does exist, but it may be hidden, and we will have to find it. He also discussed the concept that a pump is not a precision machine, and we can expect some problems with repeatability.

Howard Maxwell's paper on Measurements of Vibrational Parameters for Pump Testing really went a step farther than the code in discussing spectral analysis, the types of measurement, and the types of equipment we must use.

We really need to know the equipment we are testing, the measuring equipment, and how they interact. He discussed the equations of motion so that we could see that relationship better.

Dr. William Greenstreet discussed Low Flow Operations and Testing of Pumps in Nuclear Power Plants. We must look very carefully at the running and testing of pumps. There may be very significant degradation occurring in these pumps. If at all possible, we should be testing these pumps at the best efficiency point.

We, as owners, must be able to determine that our equipment is operating and that we can test it effectively.

Dr. Elemer Makay discussed corrective measurements that can be taken for low flow instability. He notes that each pump must be considered as an individual piece of equipment, that each pump can be tuned up by changing critical clearances that can smooth out the pump performance curve.

Also, I think an important point that he stressed several times was that we need to apply lessons learned on large pumps to the smaller pumps used in the nuclear industry.

CLOSING REMARKS

JOHN ZUDANS
FLORIDA POWER & LIGHT

I viewed the session as a very good overview, both from the new valve standard point of view, and the NRC's views of IST program reviews and the inspection techniques that they're upgrading at this time.

Tom Hoyle presented the changes to Part 10, emphasized scope and reference value institution. We heard about the new changes with regard to the standard, and we generated significant discussion on stroke timing, check valves, and other issues discussed today.

Bob Martel presented to us the generic letter overview from the industry. I think it was a very healthy discussion. I particularly thank him because we had an additional session with the Nuclear Regulatory Commission in the evening, and I think we were able to talk about issues such as relief requests, the scope of the effort that the industry must now undertake, and other issues.

Clair B. Ransom provided us a very insightful presentation on the observations of IST program reviews, identified issues such as what they are finding—generally that scope programs are not always complete, there are some inadequate relief requests; and that we still have some inconsistent interpretations of how we develop our programs. And I think that was very insightful.

Dr. Eapen from Region I provided us a philosophy change in the inspection program in the region, from a compliance–based program to a performance–based one.

We also heard that the inspection plans are being developed based on safety–related determinations and PRAs, and I think that is very good.

The inspectors will be asked to justify their findings on safety significance, and I think that is healthy for all of us. And I hope that this type of approach becomes standardized in the industry in all the regions. In that way, we all will know better what to expect.

So we had a technically sound session, as well as an administrative session, where we were able to see how things are going to be handled in the future.

CLOSING REMARKS

TED SULLIVAN
U.S. NUCLEAR REGULATORY COMMISSION

We had four papers, as you all know from attending the session. The first one was given by Owen Rothberg, and Owen was talking about MOV problems. He talked about some realized experience that gave evidence that Code testing has not been adequate to reveal certain cases where MOVs couldn't perform as they were intended.

He indicated, through a discussion of Generic Letter 89–10 and what led up to it, that this is a design issue, a testing issue, and a maintenance issue.

Steve McLean went on to talk about disassembly and inspection of check valves, a very interesting area, one that led to a lot of questions in the panel discussion. He gave us a better understanding of what is involved in developing a successful program for evaluating valve conditions for this process of disassembly and inspection.

He indicated some of the benefits, some of the shortcomings and difficulties, particularly the difficulty of determining original design information that you really need when you complete valve disassembly maintenance.

Jim Witherspoon talked about stroke time testing, another somewhat controversial area in Section XI. He called into question the value of stroke timing as a method for assessing valve conditions or predicting imminent failures.

He substantiated this with data and with actual observations that failures are being found during other aspects of plant operation, rather than during the testing phase.

John Ozol brought a unique perspective to check valves by talking about system–induced conditions that can cause problems with check valves. He gave us a number of examples and, therefore, gave us some clues as to what to look for.

He indicated that this area requires strong capabilities in design and application assessment, as well as testing.

These papers all dealt with various aspects of the engineering process involved in keeping components operable. The overall observation that I would like to make from this session was that design applications and maintenance activities are all closely linked to testing, and that none of these activities can be a substitute for the others.

CLOSING REMARKS

JOHN BASILE
VICE–CHAIRMAN OF ASME O&M SUBCOMMITTEE ON PERFORMANCE TESTING
CONSOLIDATED EDISON COMPANY

In my participation in this symposium, I tried to take the approach of first, how we, in the operations and maintenance code development committees, improve development of the codes and standards we have been working on for these many years; and second, how we, the utilities who design and operate the stations, enhance safe plant operations, which is our ultimate objective.

The major purpose of a symposium of this type is the interchange of technical and programmatic information, in this case for the inservice testing of pumps and valves. And I feel very good about it.

I believe we have been very successful. What I have seen is the number of questions and the breadth of the issues that have been discussed, the active audience participation, and the behind–the–scenes discussions by people with a common concern, bringing us together, I believe, to make this a very beneficial experience for each of us.

As I sat through the presentations yesterday, I realized that one method of communications that many do not realize exists in the Code process is, as mentioned by Mr. Chockie, the inquiry process. It is utilized to get an official interpretation of the Code or standard rules.

In the ASME/ANSI Operations and Maintenance 1987 edition of the standard, a set of procedures in the preface that tells you how to get official interpretations. This inquiry process is for your use, and I would certainly encourage you to use it. It often will result in some future changes in the standards. Now, that is an official method.

There is another way, and that is to get involved in developing the standards. We always have need for technical expertise to participate in the standards development. That involves a lot of time for individuals, and you have to ensure that you have the management support for your participation. If you have any interest, I would be glad to hear about it.

In Commissioner Rogers' challenge the other night, we heard several things. He talked at great length about applying a systems approach to our use of standards in testing the nuclear power plants that we operate. And he also suggested that we look outside the industry for new methods. Well, I think for the industry this presents a number of challenges. We need to become more innovative, and to do that takes a concentrated effort to develop new procedures.

In looking at the papers presented in the open forum, had we known some of those things a couple of years ago, we would have been a lot better off. But the point is, there is a lot of activity out there. We ought to find it and make the best use of it.

We need to establish methods to utilize new technology and new methodologies in testing before they become official consensus–developed standards, so that they can be proven before we cast the requirements in concrete.

And finally, there are operations and maintenance standards being prepared right now that are oriented toward testing plant systems. It is incumbent upon us who develop these standards to be very careful to ensure that we have the pump and valve standards and the system standards that properly complement each other, so that they do, in fact, accomplish the challenge presented to us by the Commissioner.

All of us here have responsibilities to exercise great care in specifying requirements for inservice testing, so that we reinforce nuclear safety while not inadvertently challenging the safe operation of our plants.

I believe there are several challenges in meeting that responsibility. First, those who participate in the standards development have to find better methods. We have to improve the useability of the standards we prepare, and we need to speed up the process without losing the quality in that development.

Second, I believe the NRC has a real challenge on its hands, and that is to ensure that we keep our ultimate objective in mind of safe nuclear power plant operation, not just the compliance with rigid procedures, if those procedures do not serve safe plant operation.

Third, I think we who are in utilities and operate the stations, who have the ultimate responsibility for safe nuclear operation—and that includes the design, the operation, and the maintenance of the stations—have to make our inservice testing serve us as one of the primary methods we have to ensure nuclear safety in our operations.

As I realized observing a number of the presentations, we have some challenges to ensure we understand, as the experts, what is needed to have effective testing and to carry that message to our senior management, so that we will have the correct priorities and will have the necessary funds to carry out the testing needed to ensure our safe operations.

CLOSING REMARKS

TAD MARSH
SYMPOSIUM MODERATOR
U.S. NUCLEAR REGULATORY COMMISSION

I would like to begin my part of this presentation by thanking the members of the ASME staff for the excellent work they have done in pulling this together. There was a lot of behind–the–scenes work that had to take place, and it could not have happened without them.

I would also like to thank the organizing committee. They had a difficult job. They had to pick papers and authors in a very short time. And to a large extent, the papers you heard, the quality you heard, was their responsibility. I want to thank them.

Of course, the authors themselves—I think the papers that we heard, the quality of them, the manner in which they were presented, was terrific, and they should be specifically complimented.

I want to reflect for a moment on the purpose of this symposium. Remember, I said at the beginning that the purpose was to provide a forum for the exchange of technical, code, regulatory, and related information, and it was designed to allow the utilities, the NRC, and the Code developers to share perspectives, philosophies, and technical information. I ask myself and ask you, if it was successful in that regard, and I answer that I think, yes, by several measurements.

One measure we heard several times is the attendance. This symposium was widely attended, and until the last session, we had the room filled, and attentive listening, too. People were taking notes and were exchanging information. That was excellent.

We also saw, I think, the need for more dialogue. I think because we had so many people here, because they hung in there until the very end attentively, shows the need for continuation of this type of thing.

Another measurement in my mind is the questions themselves, the questions people posed to the panel members. And they were very insightful questions, very detailed questions, very hard–to–answer questions, which is exactly what you want.

That shows that there are problems. That shows the need for guidance, clarity of thought, new techniques. So that is another measure in my mind of an excellent quality symposium.

I also want to call to your attention, in case you have not noticed, to the spectrum of people that we have here. We had 48 utilities of the 55 in the country. We had all three ANI's. We had a representative of the national board. We had consultants, a wide variety of consultants. We had the government plant operations people. We had the regulatory authorities. We had INPO and NUMARC. We had representatives of five nations here.

This was a wide spectrum of people, and that was another indication to me, at least, of success in communication among us.

But we have work yet to do, and I think that theme has come through. I want to start with the challenge to the NRC. I think one of the high points for me was a statement made by Kevin Guy in the presentation on inservice or testing programs at a non–nuclear plant.

He began his presentation by saying that what you must have for effective testing is a management commitment. If you do not have a management commitment, you will not have continuity of people. You will not have resources. And you are bound to fail.

And I believe that is true, based on the things I have seen around the industry. I want to challenge the NRC to continue the management commitment that it has made.

There has not been a management commitment in the past. There is a management commitment now, and I see every indication that it continues.

I think we need to continue to be responsive to the various information needs and relief requests from licensees. We heard this theme come through in questions from you: When will you be able to process my relief request? How responsive will you be? Is it going to take you a month? Is it going to take you a year?

Those are concerns that come from the past, and they are real concerns. I understand that, and I think we need to continue to work on being responsive to the relief requests. I believe we put in place a method to help us be responsive. But we need to continue to be attentive towards that.

John Zudans brought up another challenge, that is, to ensure coordination among the various NRC parties, including the primary branch mechanical engineers, the project managers, inspectors, and other technical branches associated with inservice testing.

I do not think you know, and I am not sure whether you could know, but you had a wide spectrum of NRC offices here, too. You have had the Office of Research, Project Management, inspectors from all five regions, and, of course, the Mechanical Engineering Branch. You had representatives of the Advisory Committee for Reactor Safeguards. That is another measure, I think, of success and an indication of need for the symposium and of the desire for coordination within the NRC.

We need to continue to work with the ASME in developing and improving new ASME standards. That is a theme we heard both here in this room and also in the hallways outside.

Several people said to me: Why are we accepting this code in its current state? We know we can do better. We know there are better standards out there. Vibration monitoring can be better.

It's true, we need to work on those standards to make them better. What you need to do as users of the standards and what we need to do as endorsers of the standards is to make sure that there is a process in place to get us to where we need to be.

Industry consensus documents are very important documents. They come from you, they flow through us, and they result in standards that are achievable standards. That is why we need to continue to work in that process.

I think we also need to consider seriously more symposiums and workshops of this sort. I think the dialogue that has taken place here has been crucial. I think it has been extremely important.

Now, I want to challenge you. I think you have heard much of this, but I want to reiterate it. It was a theme that I began with. Inservice testing should not be considered as just one of those things that needs to be done in order to satisfy those guys back in Washington. Inservice testing needs to be thought of as something that is extremely important to plant safety and as a necessary ingredient in the maintenance program and other activities, like plant life extension or advanced reactors.

You need to look at problems in a systems context, and that is not an easy concept. When you think of a system, such as when the Commissioner was discussing systems and what this is all about, you could think that could mean only the pump and its hydraulic system.

It is more detailed in my mind. It involves the human being involved in testing that component. It involves, of course, the fluid circuit. It involves the electrical components. It involves the testing environment and the procedural environment.

A valve is composed of many small parts. What is going on in that valve to make it do its thing? And you need to do the right type of testing.

This is a challenge for, not just you, but for us. If we believe that certain types of testing are harmful to components, are deleterious to plant safety or put the plant at risk, we should fix that. That should be our first order of business. How do we do that? We take that on ourselves. You should take that on with the ASME and with other representatives to ensure that the standards themselves are doing what needs to be done.

Now, the overall summary point, and I find it difficult to summarize in a short way everything that has taken place. You have heard the main thoughts so far, I want to thank you for your attendance and for your interest and for your desire to put inservice testing on the best possible footing.

We all share a common goal of safe nuclear power plants, even though our perspectives are certainly different. And we are going to disagree, on many occasions whether a certain component can be tested in a certain way or whether something needs to be in the scope.

We may not agree. We need to understand each other's perspectives. We need to have this dialogue. I think we are all on the right track, even though we have work yet to do.

Thank you very much.

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The 1990 Symposium on Inservice Testing of Pumps and Valves, jointly sponsored by the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the Nuclear Regulatory Commission, provided a forum for the discussion of current programs and methods for inservice testing at nuclear power plants. The symposium also provided an opportunity to discuss the need to improve inservice testing in order to ensure the reliable performance of pumps and valves. The participation of industry representatives, regulators and consultants resulted in the discussion of a broad spectrum of ideas and perspectives regarding the improvement of inservice testing of pumps and valves at nuclear power plants.

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