NUREG/CR-6089
ORNL-6765

Detection of Pump Degradation

Prepared by
R. H. Greene, D. A. Casada

Oak Ridge National Laboratory

Prepared for
U.S. Nuclear Regulatory Commission

AVAILABILITY NOTICE

Availability of Reference Materials Cited in NRC Publications

Most documents cited in NRC publications will be available from one of the following sources:

1. The NRC Public Document Room, 2120 L Street, NW., Lower Level, Washington, DC 20555-0001

2. The Superintendent of Documents, U.S. Government Printing Office, P. O. Box 37082, Washington, DC 20402-9328

3. The National Technical Information Service, Springfield, VA 22161-0002

Although the listing that follows represents the majority of documents cited in NRC publications, it is not intended to be exhaustive.

Referenced documents available for inspection and copying for a fee from the NRC Public Document Room include NRC correspondence and internal NRC memoranda; NRC bulletins, circulars, information notices, inspection and investigation notices; licensee event reports; vendor reports and correspondence; Commission papers; and applicant and licensee documents and correspondence.

The following documents in the NUREG series are available for purchase from the Government Printing Office: formal NRC staff and contractor reports, NRC-sponsored conference proceedings, international agreement reports, grantee reports, and NRC booklets and brochures. Also available are regulatory guides, NRC regulations in the *Code of Federal Regulations*, and *Nuclear Regulatory Commission Issuances*.

Documents available from the National Technical Information Service include NUREG-series reports and technical reports prepared by other Federal agencies and reports prepared by the Atomic Energy Commission, forerunner agency to the Nuclear Regulatory Commission.

Documents available from public and special technical libraries include all open literature items, such as books, journal articles, and transactions. *Federal Register* notices, Federal and State legislation, and congressional reports can usually be obtained from these libraries.

Documents such as theses, dissertations, foreign reports and translations, and non-NRC conference proceedings are available for purchase from the organization sponsoring the publication cited.

Single copies of NRC draft reports are available free, to the extent of supply, upon written request to the Office of Administration, Distribution and Mail Services Section, U.S. Nuclear Regulatory Commission, Washington, DC 20555-0001.

Copies of industry codes and standards used in a substantive manner in the NRC regulatory process are maintained at the NRC Library, Two White Flint North, 11545 Rockville Pike, Rockville, MD 20852-2738, for use by the public. Codes and standards are usually copyrighted and may be purchased from the originating organization or, if they are American National Standards, from the American National Standards Institute, 1430 Broadway, New York, NY 10018-3308.

NUREG/CR-6089
ORNL-6765

Detection of Pump Degradation

Manuscript Completed: June 1995
Date Published: August 1995

Prepared by
R. H. Greene, D. A. Casada

Contributing Authors
C. W. Ayers*, C. C. Southmayd*

Oak Ridge National Laboratory
Managed by Lockheed Martin Energy Systems, Inc.

Oak Ridge National Laboratory
Oak Ridge, TN 37831-8038

Prepared for
Division of Engineering Technology
Office of Nuclear Regulatory Research
U.S. Nuclear Regulatory Commission
Washington, DC 20555-0001
NRC Job Code B0828

*Oak Ridge National Laboratory

Abstract

This Phase II Nuclear Plant Aging Research study examines the methods of detecting pump degradation that are currently employed in domestic and overseas nuclear facilities. This report evaluates the criteria mandated by required pump testing at U.S. nuclear power plants and compares them to those features characteristic of state-of-the-art diagnostic programs and practices currently implemented by other major industries. In addition, since the working condition of the pump driver is crucial to pump operability, a brief review of new applications of motor diagnostics is provided that highlights recent developments in this expanding technology.

Vibration spectral analysis is the most powerful diagnostic tool for the pump analyst. The routine collection and analysis of spectral data is superior to all other technologies in its ability to accurately detect numerous types and causes of pump degradation, such as misalignment, unbalance, looseness, and various bearing anomalies. Existing American Society of Mechanical Engineers (ASME) Code testing criteria do not require the evaluation of pump vibration spectra but instead overall vibration amplitude. The mechanical information discernible from vibration amplitude analysis is limited, and several cases of pump failure in the nuclear power industry (domestic and overseas) were not detected in their early stages by vibration amplitude monitoring. Since spectral analysis can provide a wealth of pertinent information concerning the mechanical condition of rotating machinery, its incorporation into ASME testing criteria could merit a relaxation in the monthly-to-quarterly testing schedules that seek to verify and assure pump operability.

Pump drivers are not included in the current battery of required testing. Numerous operational problems thought to be caused by pump degradation were found to actually be the result of motor degradation. Recent advances in nonintrusive monitoring techniques have made motor diagnostics a viable technology for assessing motor operability. In particular, motor current or power analysis techniques can detect rotor bar degradation and ascertain ranges of hydraulically unstable operation for a particular pump and motor set. Damaging low-flow pump phenomena, such as cavitation and recirculation, may be avoidable if the pump is not operated in these unstable configurations. The concept of using motor current or power fluctuations as an indicator of pump hydraulic load stability is presented.

Contents

List of Figures

List of Tables

Summary

This Phase II Nuclear Plant Aging Research study examines the methods of detecting pump degradation that are currently employed in domestic and overseas nuclear facilities. This report evaluates the criteria mandated by required pump testing at U.S. nuclear power plants and compares them to those features characteristic of state-of-the-art diagnostic programs and practices currently implemented by other major industries. In addition, since the working condition of the pump driver is crucial to pump operability, a brief review of new applications of motor diagnostics is provided to highlight recent developments in this expanding technology.

Vibration spectral analysis is a powerful diagnostic tool for the pump analyst. The routine collection and analysis of spectral data is generally regarded as a superior means of detecting numerous types and causes of pump degradation, such as misalignment, unbalance, looseness, and various bearing anomalies. Early detection and diagnosis often provides the capability to correct the source of degradation before significant degradation has occurred, as well as the ability to plan ahead for preventive maintenance, thereby avoiding unplanned corrective maintenance.

Existing American Society of Mechanical Engineers (ASME) Code testing criteria do not require the evaluation of pump vibration spectra but instead, overall vibration amplitude. The mechanical information discernible from vibration amplitude analysis is limited, and several cases of pump failure in the nuclear power industry (domestic and overseas) were not detected in their early stages by vibration amplitude monitoring. Since spectral analysis can provide a wealth of pertinent information concerning the mechanical condition of rotating machinery, its incorporation into ASME testing criteria could merit a relaxation in certain existing code requirements that seek to verify and assure pump operability.

New advances in thermographic measurement equipment have enabled machinery analysts to detect anomalies not readily identified through other conventional diagnostic techniques. Thermography is nonintrusive and can be used to scan for hot spots that are indicative of impending degradation. For example, misaligned couplings that create excessive friction are easily seen through the lens of the thermographic camera. Other anomalies such as overheated bearings, misaligned and rubbing shafts, and components exhibiting inadequate lubrication have all been detected through thermographic evaluation.

The historical trending of lubricant analysis data can verify normal and abnormal machinery wear and degradation. Nuclear applications of lubrication analysis have been greatly assisted through the recent development of on-site lubrication analysis devices, which obviate the need for off-site transfer of samples.

The next major thrust of development in machinery diagnostic methodologies is focused on the development and refinement of expert systems. Expert diagnostic systems have been available for the past decade, but time has seen their abilities expand and their costs decline. Most expert systems marketed in the United States today do have a good diagnostic success rate; they provide an excellent starting point for diagnosing equipment problems, can provide guidance and insight regarding a variety of different machinery abnormalities, and can save valuable time by identifying incipient failures. The use of these artificial expert systems on the more routine and mundane tasks can also relinquish time for human experts to study the more time-intensive problems. With an increasing number of nuclear plants entering their third decade of operation, maintenance organizations that merge state-of-the-art expert systems with their normal diagnostic procedures may be able to increase their capability to detect and correct machinery problems as well as save on the maintenance and operational costs associated with unscheduled downtime.

Pump drivers are not included in the current battery of required testing. Failure of the pump motor to function obviously has the same effect, from a system perspective, as failure of the pump. Recent advances in nonintrusive monitoring techniques have made motor diagnostics a viable technology for assessing motor operability. Off-line and to a limited extent (at present) on-line motor parameter monitoring methods can detect stator faults. Motor current analysis techniques can be used to detect rotor degradation; examples of degraded rotor testing techniques and results are presented.

Low-flow operation and unstable flow conditions can result in substantial forces that can damage both the stationary and rotating parts of pumps. The failure and degradation modes associated with this type of pump operation include seal failure; occasional shaft, impeller, or diffuser breakage; accelerated bearing wear; premature aging of the wear rings; and cavitation damage. The historically required monthly-to-quarterly testing using minimum flow loops may have created unstable flow configurations that contributed to pump degradation at some nuclear plants.

Modifications to the ASME Code are helping to minimize operation at this condition.

A variety of analysis techniques have been developed by different organizations to evaluate pump performance parameters as well as pump head curves in order to ascertain an acceptable range of pump operation under various load conditions. The concept of using motor current or power fluctuations as an indicator of load stability is presented.

Acknowledgments

The authors wish to acknowledge the support of a number of persons and organizations who have been invaluable in their assistance with the research activities associated with this report.

Appreciation is expressed to Mr. Hirokai Yasui and the Tokyo Electric Power Company (TEPCO) for providing insightful information regarding TEPCO's pump monitoring programs. Similarly, special thanks are also extended to Mr. Borje Nilsson and the Statens Vattenfallsverk for furnishing this study with a comprehensive report on the diagnostic monitoring program at the Barsebaeck Nuclear Plant in Sweden.

The authors render special recognition to Mr. Ron Endicott of Public Service Electric and Gas, Messrs. Roger Carr and John Lewis of ITI MOVATS, and to Messrs. John Easter and Jerry Miller of the Tennessee Valley Authority for their assistance and contribution of pump and motor operational data that were used in the analysis of pump and motor degradation.

The assistance of Ms. Michelle Destefano and Brenda Smith in the preparation of this report is also gratefully acknowledged.

List of Acronyms

AE	acoustic emission
AF	analytical ferrography
AFW	auxiliary feedwater
API	American Petroleum Institute
ASME	American Society Mechanical Engineers
BEP	best efficiency point
COMOS	condition monitoring system
DIAPO	diagnostic pompes
DOE	Department of Energy
DRF	direct reading ferrography
EDF	Electricité de France
EPRI	Electric Power Research Institute
FFT	fast Fourier transform
FRF	frequency response function
HFIR	High Flux Isotope Reactor
IR	infrared
KWU	Kraftwerk Union
LPMS	loose parts monitoring system
MAINS	Maintenance Support Expert System for Rotating Machines
MCSA	Motor Current Signature Analysis
NPAR	Nuclear Plant Aging Research
NPSH	net positive suction head
NRC	Nuclear Regulatory Commission
OA	oil analysis
OM	Operations and Maintenance
PSAD	Poste de Surveillance et d'Aide au Diagnostic
PSE&G	Public Service Electric and Gas Company
PWR	pressurized water reactor
QA	quality assurance
RBOT	rotating bomb oxidation test
RCM	reliability centered maintenance
RF	radio frequency
rms	root mean square
STP	standard test procedure
TAN	total acid number
TBN	total base number
TEPCO	Tokyo Electric Power Company
WPA	wear particle analysis

1 Introduction

The pump is one of the earliest human inventions to convert natural energy to useful work. As with most machines, efficient operation and productivity have been the goals of both the pump designer and user. However, if a single item distinguishes pumps for nuclear service from conventional pumps, that item would be the attention to safety. Most of the rules and regulatory standards that have been prepared for nuclear pumps are dedicated to the achievement of public safety. The assurance of safe and reliable pump operation is a goal more readily achieved today from technological advances that have occurred in machinery diagnostics in the past 15 to 20 years. The degree to which these advances in detecting pump degradation have been implemented into the U.S. and overseas nuclear plants is the basis of study of this report.

To adequately evaluate the pump monitoring and diagnostic practices in U.S. nuclear utilities, it was necessary to understand the primary causes of pump degradation and examine the relevance of the current monitoring and testing methodologies deployed by nuclear facilities to ascertain pump operational health. Chapter 2 of this report documents the historically predominant causes of pump degradation. Chapter 3 provides a description of the state-of-the-art techniques currently being implemented by many different industries to detect and assess pump degradation. Descriptions of the program characteristics common to the more preeminent industrial and commercial pump monitoring practices are contained in Chap. 4, along with a discussion of the testing requirements mandated for safety-related pumps at U.S. nuclear plants. In addition, selected features of pump monitoring programs at overseas nuclear plants are provided as a comparison to the attributes of domestic programs.

Chapter 5 contains a selection of case histories for four differently sized pump and motor sets of various horsepower ratings to illustrate the dependence of pump operation on the performance of the driving motor. The mechanisms of low-flow pump degradation are discussed in Chap. 6. The effectiveness of monitoring hydraulic, mechanical, and electrical parameters to understand pump instabilities is discussed. The conclusions of the study are presented in Chap. 7.

2 Predominant Causes of Pump Degradation

When trying to determine the predominant causes of pump degradation, it is imperative that the operation of the entire pump system be evaluated. For the purpose of problem solving, a typical pumping system can be divided into eight components:[1]

1. foundation,
2. driver,
3. mechanical power transmission,
4. the driven pump,
5. suction piping and valves,
6. discharge piping and valves,
7. instrumentation for controlling pump flow, and
8. alignment anchoring devices.

Pump degradation can result from abnormalities associated with the performance in any one of these aforementioned areas. There are a host of advanced technologies currently used to determine the source of pump problems: vibration spectral analysis, modal analysis, acoustic emission (AE) analysis, and motor current or power analysis. Other types of evaluations include lubricant and wear particle analysis (WPA), the historical trending of process parameters such as temperature, pressure, and flow rates, and contact pyrometry and infrared (IR) imaging. In addition, the use of computerized problem analysis programs, commonly referred to as "expert systems," is also becoming another diagnostic tool for predictive maintenance personnel at various commercial production facilities. Currently, the Electric Power Research Institute (EPRI) is conducting numerous research projects on expert systems for individual components, such as feed pumps (RP3485-20) and multiple plant components (RP2923-03).

The motivation for implementing state-of-the-art predictive maintenance technologies at non-nuclear, commercial production facilities is primarily an economic one: *improving plant performance by increasing machine availability leads to greater production capacity*. This is one desired result of the practice of reliability centered maintenance (RCM). RCM is not a single condition monitoring nor a predictive analysis method, but an overall program designed to marry several monitoring technologies with a component's historical performance data that, ideally, yields an educated prediction of the component's future maintenance requirements necessary to achieve a desired performance. One goal of RCM is to avoid unnecessary maintenance and its associated costs. A properly completed RCM project should result in optimized condition monitoring methods in many systems in any type of industrial plant. Nuclear utilities, both domestic and foreign, have been experimenting and implementing RCM pilot programs.[2,3]

Since the successful application of the aforementioned diagnostic technologies has been proven in the commercial realm, it is anticipated that they will also be incorporated into the maintenance activities of most nuclear plants. The following sections discuss the most likely causes of pump problems and how the newer diagnostic technologies are being used to identify them.

2.1 Vibration Factors

For decades pumps and other types of rotating machinery have been monitored for excess vibration levels. Many vibration problems involve the deterioration and wear of the mechanical components of the pump, but hydraulic-related problems are just as common. As compiled by Charles Jackson, P.E., these are top vibration-related problems for pumps in all types of services:[4]

1. improper suction conditions (e.g., suction specific speed >12,000);
2. recirculation caused by pump not properly designed for application;
3. vane pass frequency vibrations caused by improper Gap B;*
4. pump not accurately aligned for actual heat rise of pump, driver, and pipe strain;
5. no attention to coupling fitting, balance, or keys,
6. improper selection of seals;
7. improper lubrication (e.g., too cold, water ingestion and condensation, contamination);
8. improper selection of baseplate and grouting procedures; and
9. improper maintenance procedures in shop or bearing selection/fitting wrong.

Today, pump manufacturers do provide acceptable "as-installed" vibration levels based on varying load conditions that are usually evaluated in the pump's performance acceptance test. However, acceptable overall vibration levels do not guarantee overall machinery health. The state-of-the-art in machinery condition monitoring programs incorporates vibration spectral analysis, which can reveal many types of anomalies not apparent from excessive overall vibration levels, such as bearing defects. Although overall vibration levels were being monitored, vibration spectral analysis *was not* being performed prior to the January 1989 Tokyo Electric Power Company's (TEPCO's) major casualty of a reactor circulating water pump, which was caused by a defective weld in the pump-end hydraulic bearing that failed and caused the seizure of the pump shaft. After this

*Gap B is the gap between the pump impeller and diffuser vanes.

failure, TEPCO permanently installed instrumentation on *all* its reactor circulating pumps that would allow vibration spectral analysis to be performed.[5]

Section 4.3 contains the American Society of Mechanical Engineers (ASME) Sect. XI requirements for pumps given in Subsect. IWP, In-Service Testing of Pumps in Nuclear Plants. This subsection discusses the in-service testing of Class 1, 2, and 3 pumps required to achieve the cold shutdown condition for a reactor or to mitigate the consequences of an accident. Although they are currently being revised, the existing standards were published in 1983 and describe pump testing that requires the examination of several process variables along with vibration amplitude measurements; that is, the pump's overall vibration level. Application of spectral analysis technology can yield significantly more information on machinery health than is obtained with the 1983 vibration amplitude criteria. Since there is so much pertinent machinery performance information available in spectral analysis, the importance of pump vibration spectral data cannot be understated. As further discussed in Chap. 6, even hydraulically unstable conditions, such as recirculation, rotating stalls, and cavitation, are detectable through vibration spectral analysis. As might also be expected, anomalous vibration signatures can also originate from the damaging dynamic forces inherent to the moving fluid in the piping system that frequently cause cavitation.

2.2 Hydraulic Factors

Centrifugal pump operators hope to avoid five major hydraulic phenomena: cavitation, recirculation, axial thrust, radial thrust, and pressure pulsations. Each one of these conditions manifests itself in excessive vibration and noise and can be avoided through proper design and operating conditions.

Cavitation is the formation and subsequent collapse of vapor bubbles in any flow that occurs in an ambient pressure equal to or less than the vapor pressure of the liquid being pumped. As the vapor bubbles collapse on metal surfaces, resultant pitting occurs on the surface finish. Deterioration of pump internals can be severe. Cavitation is normally evidenced by a steady crackling noise in and around the pump suction that is accompanied by a substantial increase in vibration and noise level with a reduction in total head and output capacity. It should be noted that cavitation in the presence of significant background noise may be difficult to distinguish unless acoustical signals are properly conditioned. Cavitation can be minimized by operating with an available net positive suction head (NPSH) greater than the NPSH required by the manufacturer's rating curve, but it can be difficult to completely eliminate this phenomenon.

Pump recirculation, the reversal of a portion of the flow back through the impeller, can be potentially more damaging than cavitation. Recirculation at the inlet of the impeller is known as suction recirculation and at the outlet is referred to as discharge recirculation.

> If located at the impeller eye, recirculation damages the inlet areas of the casing. At the impeller tips, recirculation alters the outside diameter of the impeller. If recirculation occurs around impeller shrouds, it damages thrust bearings. Recirculation also erodes impellers, diffusers, and volutes and causes failure of mechanical seals and bearings. The tendency toward recirculation strongly influences the minimum flow under which a pump can operate. Traditional pump curves assume uniform flow conditions and thus can be misleading and even invalid.[1]

Minimum flow conditions are also influenced by the temperature rise of the working fluid. Under steady-state conditions, friction and the work of compression increase the temperature of the liquid as it flows from suction to discharge. A further temperature rise may develop from fluid returned to the pump suction through wearing rings or a minimum-flow bypass line that protects the pump when operating at or near shutoff conditions.

The temperature rise of the fluid is calculated to be[6]

$$T = \frac{H}{778(C_p)\eta - C} ,$$

where

H = total head, ft
C_p = specific heat of the liquid (Btu/lb°F)
η = pump efficiency
C = correction factor (Fig. 2.1).

General service pumps handling cold liquids may be able to stand a temperature rise of 100°F (56°C). Most modern boiler feed pumps may safely withstand a temperature rise of 50°F (28°C). The NPSH required to avoid cavitation, prevent flashing, or to establish acceptable minimum flow conditions is highly influenced by the liquid returning to pump suction whose thermal energy has been increased.

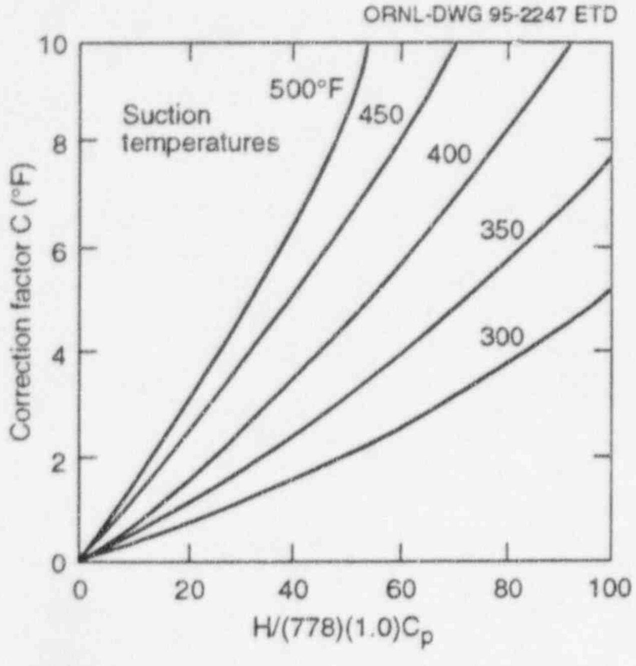


Figure 2.1 Correction factor for water at elevated temperatures. Redrawn with permission from A. R. Bush, "Calculate Temperature Rise through Boiler-Feed Pumps," *Power* 107(4), 69 (1963)

2.3 Maintenance Factors

Although manifested as a vibration problem, proper alignment is a maintenance issue. Outside of serious unbalance of pump components, there is no single contributor to poor mechanical performance more significant than poor alignment.[7] Incorrect alignment between a pump and its driver can cause

1. extreme heat in couplings,
2. extreme wear in gear couplings and fatigue in dry element couplings,
3. cracked shafts and totally failed shafts,
4. preloading on the bearings (evident by an elliptical and flattened orbit resembling a partially deflated beach ball), and
5. bearing failures plus thrust transmission through the coupling.

Significant changes in the cold, nonrunning alignment of a pump and driver can take place if the temperature rise in each machine is different and if the piping imposes forces on the pump. Therefore, alignment under actual operating conditions must be predicted or, if unknown, confirmed by instrumentation. In either case, particularly for safety-related pumps whose anticipated operation would occur only during emergency conditions, an allowance must be made in the initial cold alignment to compensate for changes in alignment from cold idle to hot running. There are numerous techniques and instrumentation available on today's market to accomplish a "hot running" alignment under "cold idle" conditions.

2.4 Motor Factors

When evaluating pump performance, it is imperative that the condition of the pump motor be taken into consideration, along with the configuration of the coupling between pump and motor. Even if the coupling between the two machines has an acceptable alignment, both angular and parallel, degradation in the pump can be translated to the motor, and vice-versa, through excessive thrust components to bearings. Ball bearing motors have reasonable thrust capacity and are more capable of accepting thrust force than those having sleeve bearings, which have a negligible thrust capacity.

Historically, the major emphasis of motor condition monitoring has been based on the evaluation of motor insulation integrity. Frequently, preventive maintenance programs use megohmeters to routinely measure the insulation's resistance to ground. Variable temperature and humidity conditions make it difficult to obtain repeatable, trendable readings, especially for those motors that are not totally enclosed and sealed from the effects of the surrounding environment. Although monitoring the resistance to ground in motors is important, other motor monitoring techniques have been developed that yield a more accurate assessment of overall motor condition.

One of the simplest forms of motor condition monitoring that is used today is visual inspection, although it is not routinely performed on large motors. This technique uses bore scopes to physically examine the inside of the motor. Discolored insulation due to thermal stress, worn surface finishes on motor windings, and contamination from dirt and moisture in the motor housing are indicative of motor degradation. The most significant problem associated with this method is attributed to limited motor clearances available for the inspection.

A study sponsored by EPRI and completed in 1985 reviewed the distribution of failures in a population of over 6000 utility motors. The failure distribution for these motors was as follows:

Bearing related—41%
Stator related—37%

Predominant

Rotor related—10%
Other (balance, alignment, etc.)—12%

Another study by EPRI evaluated the applicability of nine predictive technologies to typical motor problems and assessed the correlation between these technologies.[8] Table 2.1 shows the correlation relationship for the nine methods evaluated. Blocks containing asterisks indicate that a correlation exists between the technologies named.

Table 2.1 Correlation table for motor degradation predictive technologies

	Breakaway and coastdown	Motor circuit evaluation	High potential testing	Surge comparison tests	Motor current analysis	Temperature/thermography	Lubricant wear particle analysis	Vibration analysis
Vibration analysis								
Lubricant wear particle analysis								*
Temperature/thermography							*	*
Motor current analysis						*	*	*
Surge comparison tests					*			*
High potential testing				*				
Motor circuit evaluation			*	*	*	*	*	*
Breakaway and coastdown		*			*	*	*	*
Insulation resistance monitoring	*	*	*	*	*	*	*	*

Source: Adapted with permission from J. R. Nicholas, Jr., "Predictive Condition Monitoring of Electric Motors," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.

Frequently, vibration analysis techniques are used as universal measures of component degradation. In motors, however, this practice can overlook significant component degradation.

> One of the most commonly experienced motor-related vibration frequencies is 120 Hz. This component of the vibration spectrum can originate from a variety of sources, including power supply phase imbalance, motor winding phase imbalance, looseness in the stator iron, and other phenomena that are electromagnetic in nature. The 120-Hz component is frequently mistaken for a harmonic of running speed when low-resolution spectra are acquired. For two-pole motors, the twice running speed frequency is often greater than 119 Hz and will fall into the same bin as 120 Hz in low-resolution spectra. As a result, analysts may mistake an electrically induced vibration for a mechanical condition (e.g., misalignment).
>
> If the vibration persists it can cause mechanical damage to components of the rotating machine such as bearings and couplings. Where lubricant sampling is possible, the mechanical damage can be confirmed by the presence of wear particles using methods such as spectrometric or ferrographic analysis. For example, when rolling element bearings deteriorate mechanically, earliest indications normally come from the presence of wear particles. Ultrasonic, shock pulse and/or spike energy monitoring, which uses high frequencies (25–44 kHz) will give somewhat later indications. Vibration levels due to increased levels in bearing forcing frequencies (10–20 kHz) [*sic* 10–2000 Hz] will generally be the last to appear. By the time vibrations in the lower frequencies appear, ultrasonic and shock pulse readings have risen substantially and wear particles will be appearing in great quantity and larger size. Heat energy from deteriorated bearings, evidenced either by temperature sensors or infrared monitors, confirms the presence of mechanical damage. By the time these indications appear, the root cause becomes obscured as the focus of attention shifts to correction of collateral (bearing) damage rather than its true cause, which may be the magnetically induced vibration, which is electrical in nature.[8]

Motor current spectral monitoring has been used to assist in the identification of rotor faults. Conventional technologies rely on the measurement of the slip-related sidebands of the carrier frequency (60 Hz in the United States) and their harmonics. Recent studies by Kueck, Criscoe, and Burstein[9] have also verified the detection of rotor bar degradation through demodulated motor current analysis. Hydraulically related noise may obscure important motor-related parameters, however, and additional signal conditioning may be required to provide a clearer picture of motor degradation. Although rotor-related failures represent only 10% of the failure distribution in the aforementioned EPRI study, the continued development of this technology will serve to verify its applicability to the other major causes of motor degradation. Chapter 5 provides additional discussion on motor monitoring and analysis.

3 State-of-the-Art Pump Diagnostic Methods

It was only several decades ago when the most sophisticated type of machinery behavior assessment in an industrial environment involved the use of a technician whose job was to conduct an observational walk-through in a facility, place a screwdriver on the housing of an operating machine, and with his sensitive "calibrated" eardrum, listen for the resonating and distinct, yet anomalous, sounds. He might also use his hands to feel the intensity of the vibration that might be present; explore the housing for excessive, localized heat; look for signs of dripping oil; or smell the nearby air for burning or unique odors. The quality of a lubricant sample was examined by a visual scan, often rubbed between the fingers to evaluate viscosity, and sometimes even tasted by these old-timers for unwanted contaminants. Whether these maintenance practices were effective is certainly debatable, inasmuch as a person's inherent physical senses are difficult to quantify and trend in order to establish a cause and effect scenario. Although there are no substitutes for the eyes and ears of a good mechanic, industry is, thankfully, no longer solely dependent on the variance of human inferences for machinery diagnostics. The state-of-the-art of machinery behavior analysis has advanced by leaps and bounds in the past 20 years and grown from an art form into a proven and profitable science.

The following sections detail the more prevalent scientific methods employed today to evaluate pumps as well as other rotating machinery and discuss ways they are being implemented to monitor pumps in both industrial and nuclear power applications.

3.1 Vibration, Acoustic, and Modal Analyses

More than any other technique, vibration analysis is the most prevalent and successful analysis tool used to diagnose pump and rotating equipment problems that occur in nuclear plants and other industrial facilities. Three types of dynamic measurements can be made to quantify the operating vibration produced by the periodic motion of a rotating machinery (such as a pump shaft): displacement, velocity, and acceleration. Although it is not the purpose of this report to be a tutorial on the mathematical fundamentals of vibration, it is important to understand the physics of these different dynamic measurements in order to correctly apply them to vibration-based predictive maintenance programs. The following mathematical descriptions of mechanical motion contain numerous excerpts from J. S. Mitchell's book, *Machinery Analysis and Monitoring*:[10]

The combination of displacement, velocity, and acceleration exhibited by operating rotating machinery produce mechanical forces that are reasonably approximated by equations of sinusoidal motion. Displacement, X, is defined by the equation as:

$$X = A(\sin \omega t)$$

where

- X = instantaneous displacement amplitude from a zero or neutral axis
- A = length of the rotating vector (peak value)
- ω = its rotating speed, radians/second
- t = time, seconds

Expressed mathematically, velocity, the first derivative of displacement is:

$$\text{Velocity} = \dot{X} = A\omega(\cos \omega t) \quad .$$

Velocity is the change of linear displacement over a specific time interval. In the phase relationship between velocity and displacement in Fig. 3.1, velocity leads displacement by 90°.

Acceleration is the rate of change of velocity over a specific time interval and is defined as the derivative of velocity:

$$\text{Acceleration} = \ddot{X} = -Aw^2(\sin \omega t) \quad .$$

Looking at a simplified case of damped forced vibration, a model that closely approximates the sinusoidal motion of rotating machinery, displacement, velocity, and acceleration—when multiplied by stiffness, damping, and mass, respectively—combine to describe a total force, F(t), as a function of time:

$$F(t) = kX + c\dot{X} + m\ddot{X}$$

where

- k = stiffness or resistance to a change in length (lb/in.)
- c = damping or resistance to motion through a fluid (lb-s/in.)
- m = mass (amount of material present in the vibrating system (lb-s^2/in.)

Next it is important to examine the significance of ω, the rotational speed. Recognizing that $\omega = 2\pi f$ (where f equals

ORNL-DWG 95-2248 ETD

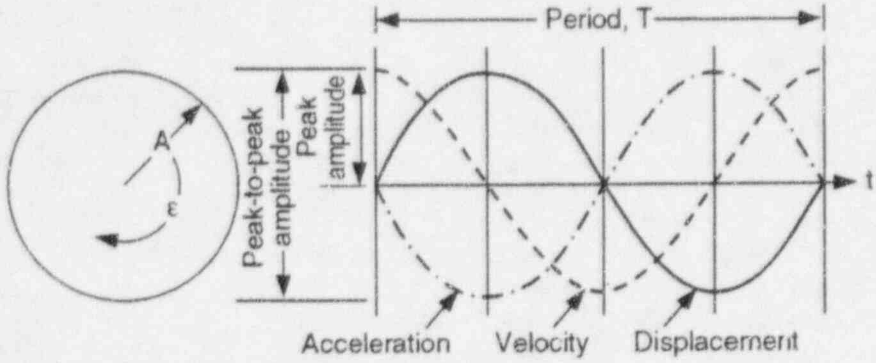


Figure 3.1 Simple periodic motion. Redrawn with permission from J. S. Mitchell, *Machinery Analysis and Monitoring*, Penn Well Publishing Company, Tulsa, Oklahoma, 1981

frequency in Hz), and substituting this value in the three terms for displacement, velocity, and acceleration, then it is apparent that the amplitude of the displacement term ($A \sin\omega t$) is independent of frequency. On the other hand, the velocity amplitude ($A\omega \cos\omega t$) will increase in direct proportion to frequency even though peak displacement A remains constant. Likewise, the acceleration amplitude ($-A\omega^2 \sin\omega t$) will increase with the square of the frequency at a constant amplitude of displacement. This variation in the values of velocity and acceleration with frequency is extremely important, for it forms the basis for vibration severity criteria, provides guidelines for selecting the variable that will be most representative for a particular measurement purpose, and explains how failures can occur without warning if the wrong dynamic variable is monitored. *The contribution to the total dynamic force shifts from displacement to acceleration as frequency increases.* At low frequencies displacement multiplied by stiffness is the dominant factor; at high frequencies, mass multiplied by acceleration dominates. Velocity appears to be a valid indicator of condition across the entire range of frequencies; it has the advantage of being linearly weighted with frequency and therefore much more representative of force than the more commonly used displacement measurements.[10]

This relationship between displacement, velocity, and acceleration provides the best indication of which parameter should be measured to assess condition. For example, if one is examining the low frequencies around or below the running speed of most machinery, velocity measurements are likely to produce the best quality signal. On the other hand, numerous pump bearing frequencies are manifested at 5 to 10 kHz and may be best measured in terms of acceleration. Figure 3.2 illustrates this principle by plotting displacement and acceleration vs frequency at a constant velocity amplitude of 0.3 in./s over a frequency range of 1 to 10,000 Hz.

The dominance of acceleration at high frequencies due to frequency squared as well as its rapid attenuation at low frequencies is readily apparent. The advantages of a displacement measurement at low frequencies as well as its disadvantage at high frequencies are also obvious. Because velocity has the advantage of being weighted by frequency, it is much more representative of the total force. The ASME Code has been revised to allow either velocity measurements or displacement measurements (refer to Sect. 4.3). Since velocity is considered to be a more revealing measurement, many maintenance staffs in nuclear power stations have implemented equipment monitoring programs (separate from the required testing programs) that evaluate not only the overall vibration in terms of velocity but use portable data collectors to scan and record vibration spectral data for diagnostic purposes.

It should be pointed out that in most cases, pump vibration signals, particularly in the acceleration domain, are complex signals that are distinctly nonsinusoidal in nature. As a result, the relationships that hold true for sinusoidal waveforms are invalid. For example, the peak of a sinusoidally varying signal is exactly the rms amplitude times $\sqrt{2}$, but the true peak amplitude in complex vibration signals is often many times the peak that would be calculated from the rms amplitude. Mitchell emphasizes this by pointing out that "RMS times 1.414, commonly labeled peak in the United States, is not equivalent to true peak, is not sensitive to the same condition characteristics as true peak, and should not be labeled peak."[10] The historical common labeling practice in the United States has resulted in considerable confusion. More discussion on this subject will be provided in Chaps. 4 and 6 of this report.

Vibration measurements require the physical attachment of a sensor to the housing of operating equipment. Depending

ORNL-DWG 95-2249 ETD

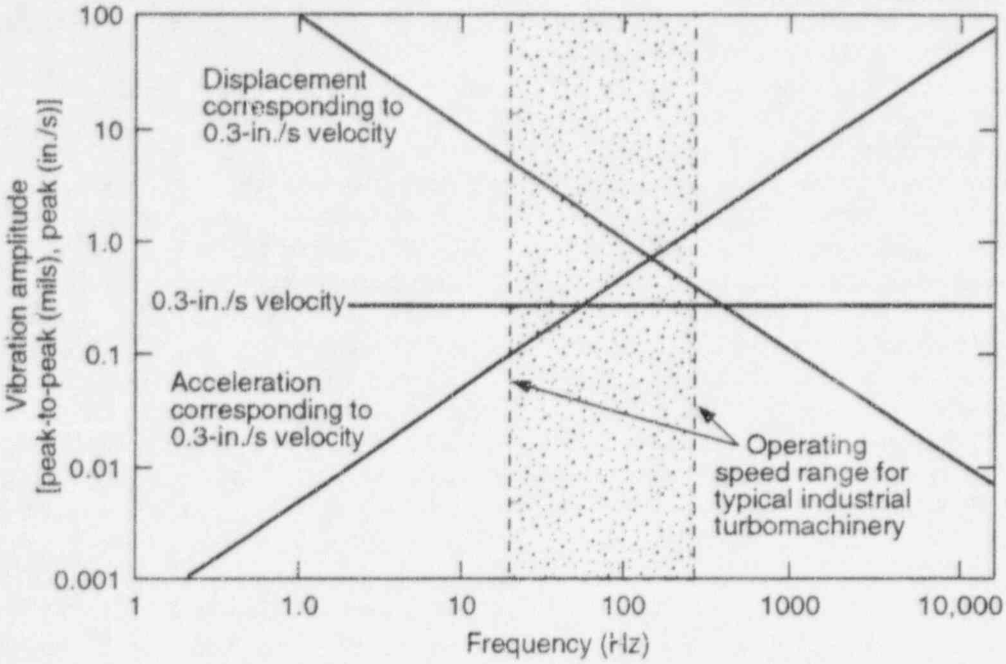


Figure 3.2 Displacement and acceleration corresponding to 0.3-in./s velocity. Redrawn with permission from J. S. Mitchell, *Machinery Analysis and Monitoring*, Penn Well Publishing Company, Tulsa, Oklahoma, 1981

on the facility, vibration data are acquired from permanently mounted or hand-held transducers. Even though the benefits of analyzing pump vibration data for predictive maintenance are tremendous, there are radiological exposure concerns to personnel who are responsible for the manual collection of this data.

Although primarily used to detect defect growth in metals, AE monitoring (the collection and analysis of very high frequency data for the purpose of detecting degradation) can also be used to discover flaws in rolling element bearings and gearbox abnormalities (i.e., high-frequency phenomena). Recent advancements in AE transducers and signal conditioning have enabled this technology to be sensitive to stress waves generated by a host of other source mechanisms, such as impacts, friction, turbulence, and cavitation.[11] Table 3.1 provides a list of the various faults that have been identified under laboratory test conditions with AE monitoring and analysis.

As with vibration monitoring, acoustical data analysis is most productive when both overall mean values and spectral signatures are trended and stored for analysis and comparison. Some types of acoustic monitoring do not require physical attachment of transducers to the machines undergoing surveillance. For example, a directional microphone can be secured on the end of an extension rod that allows acoustic data to be recorded on machinery at a distance.

Table 3.1 Sources of AE from operating machinery

Fault type	Source process
Mechanical looseness	Impacts
Inadequate lubrication	Friction and impacts
Overloading/misalignment	Friction and impacts
Fatigue damage	Cracking and friction
Rotor/stator rubbing	Friction
Wear of surfaces	Friction and impacts

Source: Adapted with permission from T. J. Holroyd and N. Randall, "Use of Acoustic Emission for Machine Condition Monitoring," *British Journal of Non-Destructive Testing*, 35 (2), 75–78 (February 1993).

The disadvantage of acoustical monitoring is that the measurements are susceptible to contamination from other sources of noise. Even the best transducers will acquire the background noise of the localized surroundings, and unless analysts are wary of this in their measurements, the data can be falsely interpreted. The guidelines for various AE signal processing techniques are still undergoing development and refinement, but since AE analysis involves the detection of high-frequency stress waves (usually somewhere within the range of 50 kHz to 1 MHz), it is important to filter out the low-frequency components during the signal processing because they can be of very high amplitudes and, therefore, mask pertinent spectral information.

Because of inevitable noise contamination, the best acoustic measurements do require the most stringent controls of the surrounding environment. To reduce the presence of unwanted and unrelated noises, continuity in the measurement conditions is a prerequisite of correct data trending. Extraneous noises from other plant activities should be held to a minimum; for this reason, it may be more practical to acquire measurements during a work shift that minimizes the presence of other plant personnel. To establish correct measurement trends, operating loads and speeds of the monitored machine and, if possible, those of the surrounding machines need to be maintained at levels near those obtained during previous measurements. Any exceptional conditions should be noted. Some analysts occasionally obtain a "background noise" spectral measurement without the machine operating so that when acoustic measurements are acquired with the machine running, the background spectra can be subtracted from the spectra of the operating machine in order to isolate its true contribution to the acoustic signature.

Many of the operational problems that cannot be identified through conventional vibration or acoustic analysis methods often require more complex diagnostics such as modal analysis. *Operational modal analysis* involves the simultaneous data acquisition of horizontal, vertical, and axial movement of an operating machine so that a simulated image of the measurement points' vectors can be constructed to represent actual operational movement. Obviously, the more sensors placed on the equipment, the better the analysis technique is able to characterize the motion of the machine. In comparison, *experimental modal analysis* uses an impact force to excite the natural resonances of a structure and/or component that is not operating and acquires data at multiple measurement points to record and analyze its response to the impact. By knowing the natural frequencies of a component or system, it is possible to avoid operational frequencies that may undesirably excite and damage its components.

Modal analysis is widely employed for computer-based simulations of operating automobiles and aircraft, but is also used to study the dynamic behavior and interactions of a system comprised of multiple components, such as a pump, motor, and connected piping. For example, if vibration studies on a system reveal an abnormal motion in a particular direction that is reoccurring, even after actions have been taken to reduce or stabilize it, then modal analysis may yield an insight not obtained with other forms of machinery analysis. Simulations showing the combined vertical, horizontal, and axial movements to illustrate the dynamic pitch, yaw, and roll of the system can reveal detrimental operating conditions, particularly those that could excite resonant frequencies inherent to the system. Modal analysis may also help to identify resonant frequencies that exist above the operating speed that may not be obvious from a coastdown of the machine.

Historically, modal analysis is a technique that is not quickly or easily utilized to study equipment motion problems. Data from numerous sensor locations are required for accurate results. The analyst must be careful to specify transducers with a frequency response that is within an anticipated range. After sensors are properly mounted, the system must be "excited" by using an impulse hammer or vibration shaker to provide a quantified input force. The system response is recorded in a data format that synchronizes the reactions of the sensors to this initial, known input force. A sufficient input force is easy to achieve in small systems, but in large systems, adequate excitation may be difficult to accomplish. A correlation between the input (force) and the output (motion) is developed from modal analysis test results to determine the system's natural frequencies. This correlation may also be used to provide a computer-generated visualization of system motion.

In recent years, noncontact measurements of a structure's modal properties have been made possible through the use of a scanning laser beam vibrometer driven by geometry file information from a standard, personal-computer-based modal analysis program. One particular scanning laser system automatically positions the measurement beam for frequency response function (FRF) measurements at over 500 locations within a 22° scanning angle at distances of up to 100 ft from the structure. Once the test is initiated, data collection at each test point, FRF generation, curve fitting, and display of the resultant mode shapes are automated and require no operator intervention.[12]

3.2 Lubrication Analysis

Lubrication analysis has been utilized as a form of predictive maintenance in the nuclear power industry since the first plants were built. This analysis methodology has made numerous technological advancements in the past two decades. It can be as simple as evaluating the oil in a sight glass to determine if it has changed color or level, or as advanced as using a scanning electron microscope to study the particles in an oil sample. Nuclear plants use different types of lubricant analyses based on component requirements, and the maintenance goals of the particular organization. The common dilemma is that most nuclear plants must send their oil samples, some of which are radioactive, off-site to be analyzed. This is a very expensive and record-intensive procedure due to regulatory requirements. However, recent innovations to the predictive maintenance industry have enabled industries to not only perform their own oil analysis on-site but to collect, store, trend, and correlate the results with vibration analysis data on personal computers.[13,14]

Analyses performed to determine the quality and suitability of the oil for continued usage are commonly known as oil analysis (OA) techniques. Analyses performed to determine the mechanical condition of the lubricated components are commonly known as WPA techniques. While hundreds of different analyses can be performed on oil samples, a limited number are routinely used in predictive maintenance programs. Many of these techniques are also valuable in the evaluation of greases.

Spectrographic analysis is a technique also referred to as trace metals, spectrochemical, or wear metals analysis. This analysis provides information on the condition of the lubricated components by monitoring concentrations of wear metals such as iron, lead, tin, copper, etc. Coolant and dirt contamination are detected by monitoring the concentration of elements such as silicon, sodium, and boron. Lubricant consistency is monitored by trending concentrations of lubricant additives such as phosphorous, calcium, zinc, etc.

Particle count is a technique that provides a distribution of particles present in the oil sample. Particle count complements spectrometals analysis by monitoring particles greater than 10 μm, which are beyond the spectrometer's detection capabilities. This analysis technique is critical to systems that are extremely sensitive to particulate contamination. On the other hand, certain systems, such as gearboxes, generate particles in such quantities that particle count information is virtually useless.

There are two types of ferrographic analysis techniques: direct reading ferrography (DRF) and analytical ferrography (AF). DRF provides a ratio of large-to-small magnetically influenced particles and can provide insight into the wear rate of the lubricated component. AF is usually employed when other analyses indicate the presence of a potential problem: ferrous particles are examined by a trained analyst who can determine possible particulate source and wear severity. AF should be used when adverse wear trends are noted and before a decision is made to conduct a visual inspection of the component.

IR analysis monitors the chemical composition of the oil in certain key wavelengths. Contaminants such as glycol, fuel, and water can be detected as well as lubricant degradation products, such as oxidation and nitration.

There are also more commonly recognized lubricant analysis techniques. Viscosity, the most important property of an oil, provides a measure of its load bearing and lubricating properties. Solids content indicates the gross particulate contamination and can reveal potential mechanical and lubricant-related problems. Total acid number (TAN) is a measure of the amount of acidic agents present and indicates lubricant oxidation or contamination. Total base number (TBN) monitors the acid neutralization reserves of the lubricant.

In addition to these analyses previously discussed,

there are analytical techniques which are critical to the evaluation of the lubricant condition of reactor main coolant pumps. These pumps present a special problem for a lubricant analysis program, since they operate for extended periods without an opportunity for sample analysis and are generally radiologically contaminated. These additional analyses indicate the effective level of the lubricant additives. With this information determination of the remaining life of the lubricant can be made. A decision can then be made to change the lubricant or extend its use through another cycle. Analyses critical to the evaluation of main coolant pumps include: rotating bomb oxidation test (RBOT), which accelerates oxidation of the lubricant to determine the useful life of the lubricant's anti-oxidation additives, and an antirust test, which determines the ability of the lubricant to inhibit rust formation on lubricated components.

Selection of the types of analyses to be performed on each unit must take several factors into consideration: potential savings of the program in terms of radiation exposure, maintenance expense, lubricant changes, and downtime must be evaluated with the types of analyses and periodicity of sampling.

The following series of analyses is recommended for main coolant pump motor bearing samples:

- spectro-metals
- solids content
- acid neutralization
- particle count
- RBOT
- viscosity at 40°
- water content
- IR analysis
- analytical ferrography
- antirust

Samples should be drawn from main coolant pump motor bearings at the beginning of each outage. Analysis of the motor sample should be provided at the outset of the outage so bearing maintenance and lubricant changes can be planned and implemented, if necessary. Obtaining a representative sample from main coolant pump motor bearing is often a problem, since it may be several days from pump shutdown before personnel are able to draw a sample. During this time, the particles and water, if present, have settled to the bottom of the lubrication system. Drawing the sample from the bottom of the reservoir will provide the worst case sample, which is preferable and should be taken into consideration when evaluating the analysis results.

The following series of analyses is recommended for equipment (and pumps) that are considered critical to plant operation and safety:

- spectro-metals
- solids content
- IR analysis
- acid neutralization (if extended lubricant usage is anticipated)
- viscosity at 40°
- water content
- particle count

This equipment should be sampled monthly or quarterly, depending upon duty cycle and importance. Quarterly analysis is the minimum recommendation to maintain consistent trend information.[15]

3.3 Thermography Applications

In December 1989, a catastrophic failure of the lubricated coupling of one of the reactor feed pumps at the Peach Bottom Atomic Power Station occurred without any prior indication of distress (vibration levels were low). This prompted the start of thermographic monitoring of the couplings to provide some indication of abnormal wear while operating. In the following year, it was necessary to replace 7 more couplings due to excessive wear, which was predicted by thermographic monitoring.[16]

IR thermography is the collection and analysis of thermal images to ascertain various types of component degradation. IR inspections are an ideal complement to a comprehensive monitoring program because the technology is noncontacting and analysis results are quickly and easily obtained. Thermography has always been recognized as an effective monitoring technique for electric and steam systems, but it should not be overlooked in evaluating the operating conditions of pumps and motors.

One advantage of IR monitoring of bearings is that it can quickly detect early signs of overheating by observing the temperature rises on the rotating shaft. This information is particularly beneficial during start-ups since it enables the monitoring of heat propagation through the shaft. This technology has also proven invaluable in assessing the true operational conditions of equipment that have no temperature sensors or where existing instrumentation is considered to be faulty.

Postmaintenance testing procedures that utilize thermographic analysis have been particularly helpful in diagnosing problems such as packing being too tight, inadequate cooling flows to packing glands, and seal clearances.

> Normally temperatures will rise fairly quickly when a newly packed pump is started and the packing 'breaks in.' Temperatures will then slowly drop and should usually be slightly hotter than the cooling water (within 5° or 10°F). The amount of time it takes for the packing to break in and the temperatures to return to normal will be, in part, a function of pump speed and shaft and packing ring contact area, and typical times can be determined for each type of pump. If the temperatures make the initial quick rise, and then continue to rise steadily, this usually indicates an abnormal situation such as inadequate cooling flow, or excessively tight packing. Temperatures can be monitored as adjustments are made to the packing.[16]

Thermographic data can be acquired with a portable IR scanner and subsequently interfaced with a computer where the IR data are downloaded. Data analysis software is then used to catalog the data and develop historical trends. Plots of component temperature vs time for the various types of equipment have been successfully utilized in the prevention of operation problems by numerous industrial and utility predictive maintenance programs.

3.4 Pump Performance Curve Comparison and Analysis

Historically, individual pump performance was primarily determined by comparing operating conditions to the pump's performance curves provided by the manufacturer. This reference curve is only an approximation of the true "as-installed" relationship between differential pressure and flow rate. If utilities use a manufacturer's curve in lieu of developing their own, the manufacturer's curve should be confirmed when the pump is known to be in good operating condition and verified over numerous operational points; the greater the number of points, the less the error of approximation is likely to be.

Particularly for load-following fossil power plants, pumps are frequently operated at conditions that are outside the design specifications for the pump. In particular, one fossil utility stated that their

> operating plants were experiencing serious reliability and performance problems with boiler feed pumps and they had very little reliable information available upon which to base decisions. As their pump testing program developed, it became apparent that most of the problems could be classified in one of three categories: the pumps were often operating in an unfavorable range due to low unit load conditions, the pumps were experiencing problems that were inherent to their particular design, or the rebuilds provided by the repair shops were inferior in quality.[17]

The utility decided to develop its own pump performance curves to help resolve some of its pump performance problems.

Unlike many tests required for safety-related pumps,

> there is no prescribed method for developing a pump reference curve. The methods vary and may yield substantially different results. The errors associated with different curve testing techniques should be understood and controlled within reasonable bounds. Manufacturer's pump curves, in general, are not sufficiently accurate to use as reference pump curves for in-service testing. Testing using reference curves generated with polynomial least squares fits over limited ranges of pump operation, cubic spline interpolation, or cubic spline least squares fits can provide a measure of pump hydraulic performance that is at least as accurate as the ASME Code method. Regardless of the test method, error can be reduced by using more accurate instruments, by correcting for systematic errors, by increasing the number of data points, and by taking repetitive measurements at each data point.[18]

The required parameters for developing a typical set of pump performance curves include flow, pressure (suction, discharge, and atmospheric), temperature (both suction and discharge), pump speed, and input power. At least five data points of pump flow rate and head are recommended for developing a pump curve for a centrifugal pump installed in a variable resistance system.[19] The test data are used to calculate the total developed head, water horsepower, brake horsepower, volumetric flow rate, and pump efficiency. The results are then corrected to design speed, utilizing affinity laws, and plotted against both the design curves and the previous test curves to evaluate the condition of the pump.

The more accurate the instrumentation, the less error is introduced into the test. The biggest source of instrumental error is typically found in the flow meter. In order for the flow meter to perform as accurately as possible, it is critical that the meter be properly located (with sufficient upstream and downstream distances from flow disturbances), regardless of the specific type of meter used. Strict calibration schedules also augment consistent test accuracy. Trending the historical performance of test data is also an invaluable predictive maintenance tool for evaluating rebuilding quality and/or the effectiveness of any maintenance modifications that may have been implemented to the pump.

Pump hydraulic testing requires comparison of an experimental test point to a point on the reference pump curve. Obviously, the precision of a plot can be a limiting factor, particularly when high-precision instruments are used. The difference between the reference point and the subsequent test point represents the pump's hydraulic degradation, usually expressed as a percentage of the reference value. Degradation usually causes the pump curve to droop, with a significant decrease in performance at high flow rates and little or no change at shut-off head. However, most pump degradation mechanisms do not result in a consistent reduction in hydraulic performance across the entire range of operation.[20] Since the ASME Code does not specify the range of pump operation where testing must be performed, pump testing has frequently and, perhaps, more conveniently been performed at low-flow conditions. As further discussed in Sect. 6.1 of this report, testing at low-flow rates provides little useful data with regard to pump operation near the best efficiency point (BEP), can accelerate degradation through cavitation mechanisms and uneven radial thrust forces that are accentuated by hydraulic instabilities, and may lead to a false sense of operational readiness.

3.5 Expert Systems and Automated Machinery Analysis

In the early 1980s, research projects were initiated by both the Department of Energy (DOE)[21] and EPRI[22] that established the foundation of today's predictive maintenance expert systems. Both of these projects were established and tested in nuclear plants. The primary goal of the DOE project was to reduce occupational radiation exposures by continuously monitoring the vibration spectral characteristics of a variety of major operating equipment and implementing predictive maintenance decisions on an "as-needed" basis rather than routinely applying procedural maintenance that may be unnecessary. By justifiably alleviating the need for maintenance, personnel doses are reduced. The EPRI project continuously monitored a select group of pumps and automatically diagnosed malfunctions that were recognized by unique patterns or changing trends in the vibration spectra; the goal was to imitate an experienced machinery analyst; hence the name, "expert system."

The expert systems marketed today perform the basics of these earlier systems and much more. Advances in computer hardware and data acquisition technology have fostered many of the innovations that expanded the diagnostic abilities of these newer expert systems, but their current fundamental design encompasses a much broader scope of tasks to perform: not only are data evaluated for a wide array of anomalous conditions, but the need for special test data can be recommended to improve and/or refine the analysis.

The normal analysis scenario is as follows: a portable data collector acquires vibration and performance data on a routine data collection route, the information is downloaded to a portable computer, and the plant machinery analyst initiates the diagnostic software packages that analyze the data. At least one predictive maintenance organization markets a continuous data acquisition system that could also be integrated with an expert system. After the machines that exhibit alert or alarm level conditions are identified (usually by automatic data scanning software), the expert system is systematically initiated on each machine. After processing the data and correlating the possible conclusions, the expert program identifies the feasible causes of the anomalous machinery conditions and usually assigns probabilities with each possible fault.

An expert system contains data from various machinery models and uses them as a reference in identifying anomalous conditions in equipment behavior. Vibration and acoustical data, pressure, temperature, and other available parameters are evaluated by a component know-ledge base that contains a complex array of logic-type rules to ascertain machinery condition. Presently the types of components that can be analyzed by expert systems include electric motors, pumps, fans, turbines, compressors, and gear boxes.

The strength of an expert system is that it is logical, impartial, and objective; it will not overlook a fault condition that an individual may mistakenly ignore or simply not be familiar with. It will provide a list of possible anomalies, a worksheet of sorts, that provides the machinery analyst with a starting point to evaluate machinery misbehavior. More frequently than not, anomalous operating conditions result from more than one fault, and it is difficult even for the experienced analyst to separate and identify numerous fault patterns that can exist in a single spectral signature. Such is the case of an expert system diagnosis of a fan system problem at an incineration plant.[23] Continuous, excessive vibration levels were causing repeated structural failures to the attaching ductwork. Diagnostics were performed using an expert system that identified a rare condition described as a "rotating stall at the impeller." Further investigations showed that the fan's inlet guide vanes, which were installed to increase efficiency by directing the paths of air, were installed backward. The system had been having operational problems for more than a year, and it was not until automated vibration analysis was performed on the system that the real root cause of the problem was solved.

One weakness of the expert analysis system is that it could be confusing and misleading to those who may use it. Conclusions drawn by the artificial expert may be unreasonable to the human expert, or just plain wrong. The software decision logic tree is only as good as its designer, and the pattern recognition analysis techniques are extremely dependent upon the accuracy of the data.

As the knowledge bases of various components improve, the skepticism of expert systems will eventually fade inasmuch as the wariness of computer usage has disappeared. Maintenance departments can benefit from routinely consulting the knowledge base of an expert system. Executing the automated diagnostic system on all routine data prior to any other analysis techniques has proven to be beneficial to increasing the accuracy of the machine diagnosis and reducing the time required to complete the analysis process. One organization reported an average labor savings of 20% among nine analysts who incorporated automated diagnostic techniques into their routine analysis methods.[23] The same study also reported that the use of an automated diagnostic system gave rise to more questions about the accuracy of manual analysis methods, not necessarily the accuracy of the expert system itself. A different study at a pulp and paper production facility[24] attributes a 75% savings of analytical time to an expert system that enabled its mechanics to perform the "run-of-the-mill stuff" and relinquished the maintenance engineer's time for solving the "tough problems." The most recent international development of expert system technology for specific application to nuclear plant pumps is DIAPO (a French acronym for DIAgnostic des POmpes—pump diagnostics), which is being deployed by the Electricite de France (EDF) in their pressurized water reactor (PWR) coolant pumps.[25] DIAPO is part of an overall diagnostic assistance system, PSAD (a French acronym for Poste de Surveillance et d'Aide au Diagnostic), that automatically monitors and examines vibration and operational data from major nuclear plant components. The twofold purpose of DIAPO is to detect failures at incipient stages and to identify the root causes of failures after they have occurred. This particular expert system is further discussed in Sect. 4.6.2.

The implementation of automated diagnostic systems may be one solution to reducing maintenance-related labor costs and personnel exposures in nuclear plants. When pump locations make routine manual data acquisition "dose prohibitive," the monitoring needs of these components may be best served by installing permanently mounted transducers that routinely supply spectral data to remote data acquisition systems. Although the equipment costs associated with a remote monitoring and diagnostic system in a nuclear plant may be an affordable expense, the real challenge is the labor costs associated with permanent installation and qualification of the instrumentation cabling.

One organization has proposed the use of wireless radio frequency (RF) data communications for predictive maintenance applications in nuclear plants.[26] The architecture of this type of plantwide monitoring is shown in Fig. 3.3. The intelligent data collector acquires data from various inputs and transmits data through RF communication to an intelligent satellite receiver/diagnostic unit, which is mounted adjacent to an existing plant telephone line. If data are received that exceed alert or alarm conditions, then the satellite unit contacts (by telephone) the host unit to report the anomalous condition. The host unit makes a report to the analyst who then has the option of requesting more diagnostic information from the satellite unit or personally investigating the source of trouble. Similar data communications architecture has been employed in emergency signal service for elderly and disabled persons. EPRI is sponsoring a research project (RP-3444) at selected utilities to investigate the feasibility of wireless monitoring and to verify that the RFs used for this type of communication would not interfere with the operation of plant equipment.

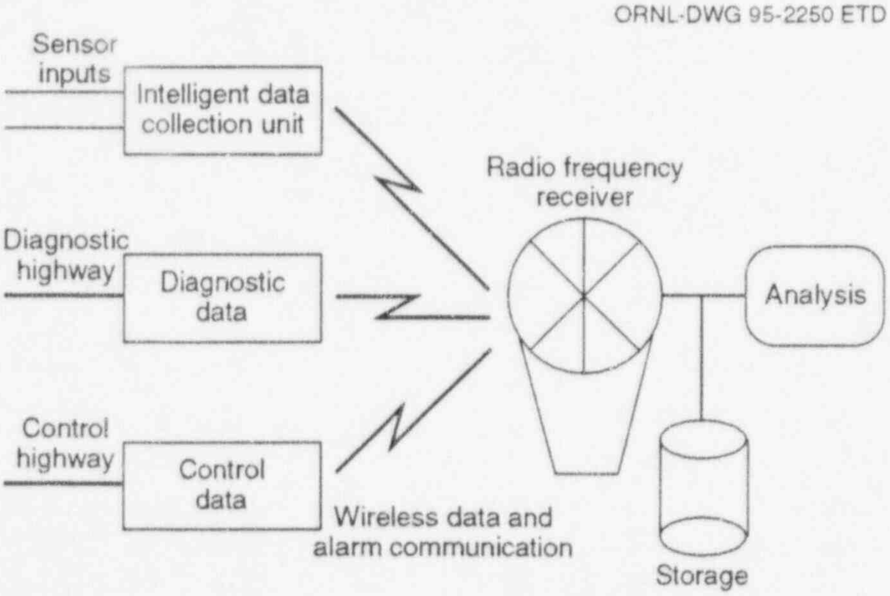


Figure 3.3 Schematic of wireless computer-based monitoring. Redrawn with permission from R. Gopal and S. Kannan, "The Global Database Server and Plantwide Predictive Maintenance," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992

4 Existing Pump Monitoring Programs

Many nuclear facilities have permanently installed vibration monitoring instrumentation for critical components such as reactor circulating pumps, main turbine generator systems, and some main feed and condensate pumps. The control rooms at these plants continuously display these data (usually as an overall vibration displacement reading) on a panel indicator or a strip chart recorder. In many cases, the machine's power consumption or motor root-mean-square (rms) current is also monitored. However, an increasing number of nuclear utilities are using vibration spectral signature analysis in predictive maintenance programs. These efforts routinely monitor a wide range of rotating equipment that usually includes pumps (and their drivers) for a variety of different services. The equipment having the easiest accessibility in low-dose areas is frequently the first to be monitored to test a new program, since portable data collectors with hand-held vibration transducers are prevalent in most beginning programs. The more mature programs may have added permanently mounted sensors on selected equipment and/or incorporated "expert" analysis systems to assist with the facility's diagnostic efforts.

Before establishing an effective equipment monitoring program, the machinery analyst is faced with numerous decisions that include which machines to monitor, the locations of the measurement sites on each machine, types of data to collect and analyze, and how often to acquire data. As an individual becomes more informed and acquainted with the practice of machinery monitoring and analysis, the program usually expands and evolves into a more customized program geared to the needs, requirements, and maintenance goals of the organization.

In the past decade there has been exponential growth in the industry that markets predictive maintenance technology. As a result of widespread consumer interest, numerous articles, books, and courses have been developed to train and equip maintenance organizations in the planning and execution of their predictive maintenance programs. Many of these programs are based on similar fundamental principles that can be applied in any industry. The following sections discuss features associated with the best industrial programs and illustrate how some of their characteristics might be applied in a nuclear plant.

4.1 Basic Monitoring Program Characteristics

According to K. R. Guy's paper on "Pump Monitoring and Analysis,"[27] the first item of business is to develop an equipment file package for each pump to be monitored. Figure 4.1 illustrates examples of the type of data that are usually included in this file. For instance, specific design information that is useful in vibration spectral analysis includes the number of impeller vanes (for vane passing frequency calculations) and the types of bearings (used to acquire bearing frequency data). Each equipment file should contain a copy of the vibration baseline spectrum that is believed to be representative of satisfactory pump operation at a nominal speed and load. A new baseline spectrum should always be acquired after major maintenance, such as a pump overhaul or rebuilding. Subsequent pump spectra will be compared to this baseline for abnormal deviations. The manufacturer's pump head curve is another important part of this equipment information package.

A list of the same model and/or same service pumps located at the facility is also beneficial. Often, an indicator of abnormal operation for a machine may not be an excessive amount of vibration but a spectral signature that exhibits unexpectedly different features when compared to the spectral signatures of pumps of the same design in system loops with similar service conditions.

Most programs prioritize the equipment to be monitored according to its importance to plant operation. For the example of nuclear applications, the first group would probably include any safety-related pumps, such as the reactor coolant pumps, auxiliary feed pumps, residual heat removal pumps; that is, any pump whose loss would compromise plant shutdown capability. In addition, nonsafety-related pumps, such as feedwater, condensate, and circulating water pumps merit consideration.* Any pump that is routinely monitored to comply with plant technical specifications would be best placed in this first group. These stipulations mandate that the monitoring of pumps in the first group should take precedence over monitoring pumps in other groups. Although some industrial and commercial facilities attest to monitoring their highest priority group of pumps every 2 weeks, the more prevalent practice is to monitor on a monthly basis.

The second monitoring group collects data monthly-to-quarterly and usually comprises those pumps that have redundant backups. The last group of pumps simply

*Although there are no ASME Code requirements for this latter group of pumps, they are more likely to be monitored (in a spectral vibration mode) than are safety-related pumps since the loss of one of these pumps can result in required plant shutdown and, hence, power production.

Existing

ORNL-DWG 95-2251 ETD

MOTOR:
NAMEPLATE DATA:
VENDOR: ____
MOTOR TYPE: ____
HP: ____ S.F.: ____
RPM: ____ FRAME: ____
AMPS: ____ POLES: ____
ROTOR BARS: ____ STATOR SLOTS: ____
END SHIELD SUPPORTED BEARINGS: ____
BEARING HOUSING OVERHUNG: ____
VENTILATION: ____
TYPE OF ENCLOSURE: ____
BEARINGS: LUBRICATION:
OUTBOARD: ____ OUTBOARD: ____
INBOARD: ____ INBOARD: ____
PUMP:
TYPE: ____ (DIFFUSER/VOLUTE)
MOUNTING: ____
FLUID: ____
IMPELLER DIAMETERS:
OUTSIDE DIAMETER: ____
INSIDE DIAMETER: ____
DIFFUSER/VOLUTE DIAMETERS:
OUTSIDE DIAMETER: ____
INSIDE DIAMETER: ____
BEARINGS OUTBOARD: ____
INBOARD: ____
LUBRICATION:
OUTBOARD: ____
INBOARD: ____
BLADES:
INLET:
IMPELLERS: ____
DIFFUSER: ____
OUTLET:
IMPELLERS: ____
DIFFUSER: ____
STAGES: ____
NPSH: ____
SUCTION PRESSURE: ____ FLOW: ____
FLUID TEMPERATURE: ____
PIPING ATTACHMENT: ____
SNUBBERS IN DISCHARGE PIPE: ____
BELTS:
NUMBER: ____
PULLEY SIZE:
DRIVE: ____
DRIVEN: ____
C-LINE DIST: ____

Figure 4.1 **Recommended pump data to be included in equipment file package. Redrawn with permission from F. R. Guy, "Pump Monitoring and Analysis," pp. 95–114 in *Proceedings of the Second NRC/ASME Symposium on Pump and Valve Testing, Washington, DC, July 1992*, USNRC Conference Proceeding NUREG/CP-0123**

contains all the other plant pumps that do not fall into the previous two categories and are monitored on a quarterly or longer basis. If program constraints do not allow data to be collected on a particular pump, monitoring routes can be developed that allow a technician to at least walk past the pump and quickly examine it. When regularly performed, general observations have been and always will be a form of predictive and preventive maintenance. The eyes and ears of a good mechanic are still some of the best sensors available today. If any pump is exhibiting abnormal operational behavior, it should be monitored more frequently until the problem can be diagnosed and corrected. After major maintenance, some industrial facilities continue a more frequent data collection in subsequent months to ensure that the maintenance was effective, the pump performance remains acceptable, and the maintenance activities did not inadvertently introduce operational problems.

The next step recommended by pump experts is to categorize, within each group, the pumps according to their mounting configuration (i.e., vertically or horizontally mounted). Knowing the mounting and the foundation may provide insight into possible problems encountered when troubleshooting the pump. The horizontally mounted pumps should be further classified into centerhung or overhung. A graphical representation of each pump depicting its configuration and the location of each measurement point is another valuable addition to the equipment file (Fig. 4.2).

Measurement point identification and preparation are important to ensure accuracy and repeatability in data acquisition and storage. Each permanently mounted sensor should be physically identified on the pump with a label or instrument tag. If using a hand-held or magnetically mounted transducer, the monitoring site for each measurement location should be clean to the casing's metal surface and free from numerous layers of paint to prevent data corruption due to mounting resonances. Each site should be labeled with a unique identifier so there is no guessing among different personnel as to the exact location for data acquisition. In addition, studies have shown that when different technicians were involved in manual data collection, human error was inadvertently introduced from slightly different pressures applied to hand-held probes.[28] To augment data repeatability, some vibration experts also recommend to users of magnetically mounted transducers to paint a small line on the magnet itself in order to line it up with a reference mark on the measurement site so that the magnetic pole orientation will be the same for a series of trended measurements.

The number of measurement points for each pump will vary in accordance with the design and service of the pump. As a minimum, vibration data should be acquired in two orthogonal directions at each accessible bearing housing and also in the axial direction at the pump thrust bearing housing. Some vibration analysts believe it is critical to monitor vibration data in the load zone of rolling element bearings, that is, the probable location of maximum bearing deflection due to loading forces placed on the bearing during operation. Others assert that it is sufficient (and all agree necessary) to always monitor at the same location(s).

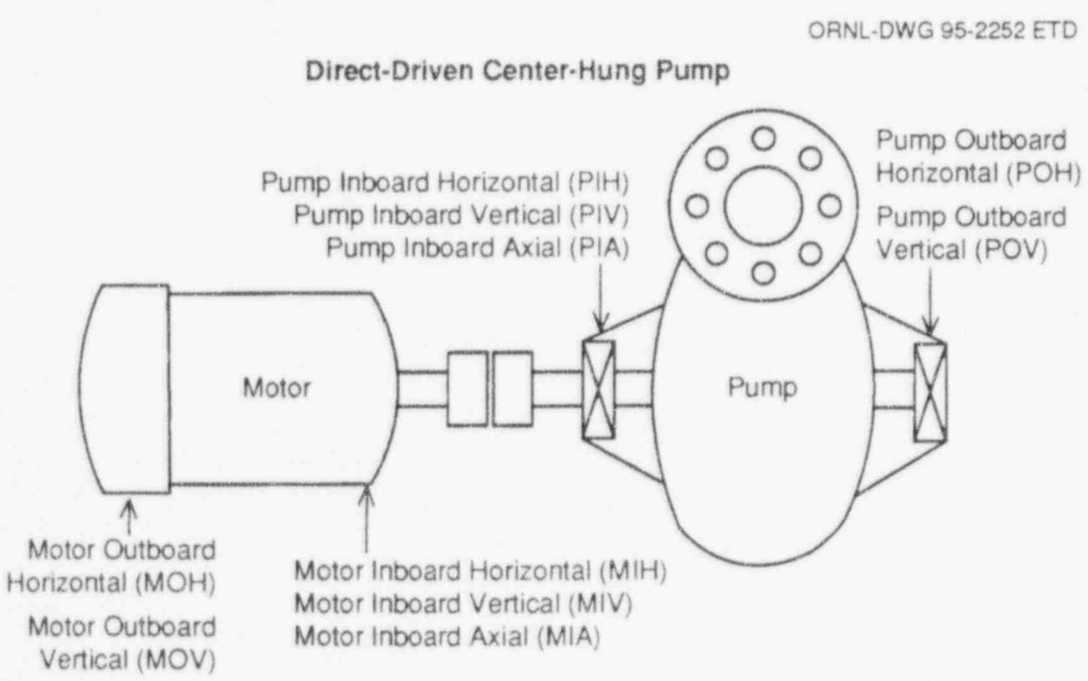


Figure 4.2 Schematic of measurement point locations on monitored pump

Existing

One resulting benefit from developing the equipment files for each pump is a good understanding for the types of problems that have been encountered during previous operational history (i.e., bearing problems, misalignment, etc.). The data collection and analysis instrumentation and sensors used for the pump monitoring program need to operate over frequency ranges that support the diagnosis of the plant's historical pump problems and other types of pump failures that warrant early detection. Almost all vibration sensors are adequate for problems that can be detected at "mid-band" frequencies (~100 Hz to 2000–3000 Hz). However, pump anomalies such as oil whirl, oil whip, and rotor rub are typically manifested at low frequencies that occur at 40 to 50% of the running speed. If low-frequency vibration amplitudes are measured near the electronic noise floor of the sensor and/or the monitoring instrument, signal corruption will occur and distort the accuracy of the acquired data. There are also high-frequency mechanical events that occur near and above 10,000 Hz that can be caused by pump cavitation, entrained gas (aeration or starvation), bearing impact noise, and high-pressure leaks. Faults on bearings, casings, rotors, and piping can create high-frequency vibrations that can overload an improperly sized sensor, cause low-frequency distortion in the data, and possibly obscure valuable subrotational and harmonic running speed information. To summarize, it is important to select the frequency response of the sensor so that bearing faults (and other high-frequency anomalies) on high-speed pumps will not go undetected.

Pump analysts should consider measurement scenarios that may require the simultaneous acquisition of data. Single-channel, portable fast Fourier transform (FFT) analyzers are ideal for the majority of data acquisition and analysis needs, but eventually the program may need the capability of acquiring two or more channels of data simultaneously. Such multichannel tape recorders and FFT analyzers may not always be needed for routine measurements but certainly provide a variety of options for analyzing pump operational data. Some nuclear plant emergency equipment is only operated quarterly for testing, and it is difficult to acquire data after the pump has been shut down in order to study problems that may have gone undetected during routine data collection. Data can be tape-recorded for any equipment that cannot easily be placed into service. When the data have been permanently stored or archived, the analyst can examine the data over various frequency ranges and perform FFTs at different spectral resolutions to determine the nature of impending pump problems. This practice allows the analyst to focus more time on data analysis since normal test runs may not last long enough to detect and/or resolve problems.

Many of the newer vibration analysis systems are designed to allow opportunity for expansion and growth in analysis capabilities. Numerous plants find it beneficial to keep a historical trend of operating temperatures in addition to vibration data, power consumption, lubrication analysis reports, and performance indices all in the same computer file for a specific pump. In addition, note-taking capabilities are also available from keyboards designed into several portable data collectors to store observational data such as unusual or excessive noise, leaking, strong odors, smoking, etc. Computer software specifically developed for plant monitoring programs has incorporated features that facilitate easy data storage, analysis, and rapid accessibility. Few equipment problems are manifested by a single type of variable; the more information the analyst can acquire and easily compare that is reflective of pump performance, the more accurately and promptly anomalous conditions can be understood.

Pump analysts often acquire and trend the results of pump performance data in their monitoring programs to examine the differences between present and previous operational indicators. Varying flow rates and loads will always change performance data, but unexpected changes in these parameters can act as precursors to pump degradation. Data windows that define a high alarm, a high alert, a low alert, and a low alarm can be determined to evaluate pump performance data. Some organizations apply statistical techniques that use standard deviation calculations to estimate acceptable values for steady-state data, but one simple and reasonable way to determine the acceptability of pump performance data is to review the pump's operating curve. Some programs set the high alarm for operating flow rates at 120% of the BEP to indicate the onset of cavitation. The low-end alarm is set at 65% of BEP to avoid damaging hydraulic instabilities. These limitations raise concerns regarding the use of 10 to 15% of the BEP during minimum flow testing. Since abnormal flow conditions can also excite pump bearing frequencies, it helps the analyst to know the historical trends of these pump performance parameters with respect to varying flow rates and loads.

Data management is another aspect of every monitoring program that influences its effectiveness in a facility. Every machinery monitoring program must analyze an enormous amount of real-time data in order to accurately determine true machinery performance. Information must be archived to maintain a historical trend of spectral and performance data that provides a faithful chronology of machinery behavior. A machinery monitoring program that yields so much data that diagnostics personnel are overwhelmed during evaluation is not likely to be effectively implemented in the facility.

Several data reduction techniques have been developed to facilitate analysis activities. One method uses the definition of trend parameters. Rather than storing every data point in a spectrum (such as 128, 256, 512, 1024, etc., points for increasingly detailed spectra), a single overall value is stored for the energy contained within selected frequency bands of interest that comprise the spectrum. There are known frequencies associated with multiples (or orders) of the machine's running speed that are indicative of anomalous behavior if they contain appreciable energy: the running speed itself, twice the running speed, blade pass frequencies (usually several multiples of the running speed), gear mesh frequencies, rolling element bearing frequencies, and critical speeds.

To simplify spectral analysis and the data storage requirements associated with many analysis techniques, a trend parameter is often defined to retain the overall vibration energy contained within a frequency window. As shown in Fig. 4.3, trend parameters representing three different multiples of the running speed are analyzed along with two different frequency bands to observe possible bearing defects. These frequency windows are user-defined for each machine in a frequency analysis program.

The benefits of using trend parameters (or similar methodologies) to reduce data are threefold: (1) storing trend parameters to characterize a spectral signature saves memory in portable data collectors and enables a larger data collection route to be developed for a fixed amount of memory; (2) since more machines can be monitored in a fixed amount of time with fewer points to collect and store, a monitoring program can routinely observe machines over a more expeditious time interval with, perhaps, fewer personnel requirements; and (3) the time required for a computer-based analysis of fewer data points usually allows faster production of the historical trend reports generated from these surveys.

The employment of trend parameter analysis techniques does not absolve an organization from acquiring spectral signatures on machinery. Most industrial programs that use this data acquisition method collect full spectral data on a quarterly basis and trend parameter data on a monthly basis. Some software programs allow the user to specify the automatic storage of full spectral data if any trend parameter value exceeds a user-defined limit. Rapid data evaluation is also facilitated by automated, computer-based analysis programs that scan observed parameter values and compare them with user-defined ranges. Computerized reports can also be automatically generated that detail measurement points on a collection route that exhibited

ORNL-DWG 95-2981 ETD

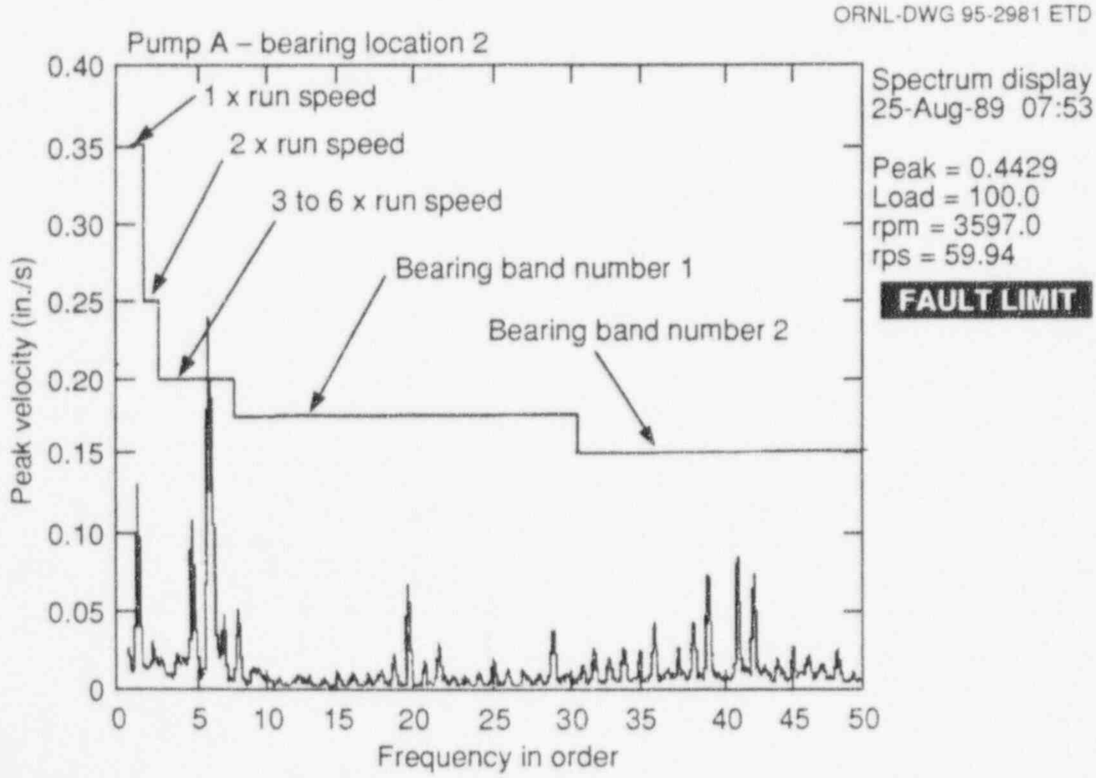


Figure 4.3 Example of frequency analysis based on multiples (orders) of pump running speed

anomalous behavior, thus pinpointing the analyst's attention to a machine that may require timely maintenance.

4.2 Predictive Maintenance Programs in U.S. Nuclear Plants

The participation and presentations by nuclear plant personnel at the past five Vibration Institute annual meetings indicate that numerous U.S. nuclear utilities have implemented state-of-the-art machinery monitoring and diagnostic technology at their facilities but have also accomplished this in a variety of ways due to respective differences in organizational goals. Most of these monitoring programs do not operate for the purpose of compliance with regulations or technical specifications, but as assistance and guidance to operations and maintenance organizations. Many of the aspects of the nuclear monitoring programs are similar to the characteristics of programs that have been successfully implemented for many years at fossil-fueled power plants.

Starting a predictive maintenance program in an industrial facility is a much simpler undertaking than for a nuclear plant; for industries it is limited to selecting and purchasing the equipment, training the personnel, and defining monitoring routes and machinery measurement requirements. As a minimum, nuclear plants are usually challenged with the additional process of procedure writing for machinery data acquisition, analysis, and results reporting. In some cases, written approval must be also obtained each time data are acquired, even if data acquisition is a nonintrusive activity. The high standards of documentation protocol normally associated with the nuclear industry may impede the prompt implementation of new measurement programs. According to open discussion during a panel session of the 1993 Vibration Institute Conference (St. Louis, Missouri), a nuclear maintenance manager representing a southwestern U. S. nuclear plant described to other attendees how his nuclear facility's quality assurance (QA) department had intervened in the new plant predictive maintenance program to the extent that it brought progress to a virtual standstill because the state-of-the-art monitoring equipment and computer analysis software being used was not "nuclear qualified."

Other nuclear plant personnel in attendance during the same panel session shared how they had successfully implemented their predictive maintenance programs by establishing and managing them separately from those groups collecting data for technical specifications and ASME Code requirements. This latter method used independent organizations, separate equipment, isolated offices, and different personnel to ensure that there was no interaction that might create confusion about the two program's different objectives.

Few maintenance organizations, particularly those struggling with declining budgets, are eager to accommodate the costs associated with the additional paperwork and procedural planning often required to execute new measurement programs. Management is frequently unwilling to support additional monitoring endeavors since the existing nuclear programs fulfill technical specification and ASME Code monitoring requirements. Unless personally familiar with the numerous benefits of predictive monitoring, there are maintenance personnel in all industries who believe enhanced monitoring programs are unwarranted. Even after they are implemented, numerous programs often lose initiative because management becomes too impatient in obtaining benefits.

Most of the generic problems associated with predictive maintenance program implementation that were discussed during the open panel session of the 1993 Vibration Institute Conference were attributed to labor shortages, not the accuracy or dependability of hardware or software. Utilities that operate several power plants have often found it very cost-effective to establish independent predictive maintenance groups that monitor equipment at all the different plants. When this is the case, regular reports of monitoring activities are directed to the plant managers as well as to the respective supervisors of the operations and maintenance departments.

4.3 In-Service Testing Requirements

Hydraulic and vibration in-service testing are required for ASME Class 1, 2, and 3 pumps that are provided with an emergency power source and have a specific function in shutting down a reactor or in mitigating the consequences of an accident. Plants are committed to specific editions of the code through their technical specifications. Since the code has evolved with time, not all plants operate under a single set of requirements. Table 4.1 provides the requirements of the 1983 ASME Sect. XI, Subsect. IWP, through the Summer 1983 Addenda.[29] Many plants are currently operating under these or similar requirements. Although not shown in Table 4.1, the IWP required vibration monitoring at only one point and orientation.

In 1987, the ASME issued, under the auspices of the Committee on Operation and Maintenance (OM) of Nuclear Power Plants, OM Part 6, Inservice Testing of Pumps in Light-Water Reactor Power Plants.[30] The Nuclear Regulatory Commission (NRC) has endorsed this standard

Table 4.1 Tabular synopsis of ASME Sect. XI pump test quantities (1983)

In-service test quantities

Quantity	Measure	Observe
Speed N (if variable speed)	x	
Inlet pressure, P_i	x[a]	
Differential pressure, ΔP	x	
Flow rate, Q	x	
Vibration amplitude, V	x	
Proper lubricant level or pressure		x
Bearing temperature, T_b	x	

Acceptable hydraulic and vibration ranges of test quantities[b]

		Alert range[c]		Required action range[b]	
Test quantity	**Acceptable range**	**Low values**	**High values**	**Low values**	**High values**
ΔP	0.93 to 1.02 ΔP_r	0.90 to < 0.93ΔP_r	1.02 to 1.03 ΔP_r	<0.90 ΔP_r	>1.03 ΔP_r
Q	0.94 to 1.02 Q_r	0.90 to 0.94 Q_r	1.02 to 1.03 Q_r	<0.90 Q_r	>1.03 Q_r
P_i[d]			(See notes)		
T_b[e]			(See notes)		
V, when $0 \leq V_r <$ 0.5 mils	0.0 to 1.0 mils	None	1.0 to 1.5 mils	None	>1.5 mils
V, when 0.5 mils < $V_r \leq$ 2.0 mils	0.0 to $2V_r$ mils	None	$2V_r$ mils to $3V_r$ mils	None	>$3V_r$ mils
V, when 2.0 mils < $V_r \leq$ 5.0 mils	0.0 to $(2 + V_r)$ mils	None	$(2 + V_r)$ mils to $(4 + V_r)$ mils	None	>$(4 + V_r)$ mils
V, when $V_r >$ 5.0 mils	0.0 to 1.4 V_r mils	None	1.4 V_r mils to 1.8 V_r mils	None	>1.8 V_r mils

[a]Measure before pump startup and during test.

[b]Paragraph IWP-3110 states the following:

"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 with new reference values established in accordance with IWP-3111 and IWP-3112. Reference values shall be determined from the results of the first in-service test run during power operation. Reference values shall be at points of operation readily duplicated during subsequence inservice testing."

Reference values are denoted by the subscripted "r."

[c]IWP-3230.

[d]P_i (inlet pressure) shall be within the limits specified by the owner in the record of tests (IWP-6000).

[e]T_b (bearing temperature) shall be within the limits specified by the owner in the record of tests (IWP-6000).

Source: Adapted from the ASME Boiler and Pressure Vessel Code, Sect. XI, IWP, Table IWP-3100-1, pp. 209–210 (1983).

(including the OMa-1988 Addenda) in 10 CFR 50.55a. Table 4.2 outlines the OM-6 test criteria per this edition of the code. In comparison to the earlier Sect. XI code, OM-6 allows a wider range of acceptance criteria for centrifugal pump hydraulic parameters. However, OM-6 requires that vibration be monitored in two orthogonal directions on each accessible pump bearing housing, as well as axially on accessible thrust bearing housings. OM-6 also provides for measurement of either vibration velocity or displacement, while IWP only specified displacement domain measurements. In addition, OM-6 also arranges for explicit upper limits on vibration, while IWP allowable amplitudes were based purely on baseline values (with the exception of very smooth-running pumps).

Because of a variety of issues, modifications to the ASME Code requirements have continued. One particular concern was that some pumps (most notably standby pumps at PWR plants) were being tested at minimum flow condition, which provided little useful information about the condition of the pump and at the same time was harmful to the pump.

To address these concerns, the OM Committee has approved several significant changes to the code, the current version of which is Subsect. ISTB of the ASME OM Code–1990, including the 1994 Addenda.[31] Tables 4.3 through 4.6, which are extracted from this version of the code, provide test requirements and acceptance criteria. Figure 4.4 illustrates the alert and action range vibration amplitudes as a function of pump speed. The Group A pumps are those pumps that are operated continuously or routinely during normal operation, cold shutdown, or refueling operations. Group B pumps are those pumps that are in standby systems that are not operated routinely except for testing.

Some of the important changes follow:

1. The quarterly Group B pump test requirements are changed to monitor only flow or differential pressure.

Table 4.2 OM-6 vibration and hydraulic test criteria

Pump type	Pump speed	Test parameter	Acceptable range	Alert range	Required action range
Centrifugal and vertical line shaft	<600 rpm	V_d or V_v	$\leq 2.5\ V_r$	>2.5 V_r to 6 V_r or >10.5 mils	>6 V_r or >22 mils
Centrifugal and vertical line shaft	≥600 rpm	V_v or V_d	$\leq 2.5\ V_r$	>2.5 V_r to 6 V_r or >0.325 in./sec	>6 V_r or >0.70 in./sec
Reciprocating		V_d or V_v	$\leq 2.5\ V_r$	>2.5 V_r to 6 V_r	>6 V_r

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 1.10ΔP_r	0.93 to <.95ΔP_r	...	<0.93ΔP_r	>1.10ΔP_r
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 1.10ΔP_r	...	...	<0.90ΔP_r	>1.10ΔP_r
Q (centrifugal pumps)	0.90 to 1.10Q_r	...	...	<0.90Q_r	>1.10Q_r

Notes: Vibration units are peak for velocity and peak-to-peak for displacement. Subscript *r* denotes reference value, *v* denotes velocity, and *d* denotes displacement.

Source: Adapted from the ASME OM Code–1987, including the 1988 Addenda.

Table 4.3 In-service test parameters and instrument accuracies

In-service Test Parameters

Quantity	Preservice test	Group A test	Group B test	Comprehensive test	Remarks
Speed, N	X	X	X	X	If variable speed
Differential pressure, ΔP	X	X	X [Note (1)]	X	Centrifugal pumps, including vertical line shaft pumps
Discharge pressure, P	X	X		X	Positive displacement pumps
Flow rate, Q	X	X	X [Note (1)]	X	
Vibration	X	X		X	Measure either V_d or V_v
Displacement, V_d					Peak-to-peak
Velocity, V_v					Peak

NOTE: (1) For positive displacement pumps, flow rate shall be measured or determined; for all other pumps, differential or flow rate shall be measured or determined.

Required Instrument Accuracy

Quantity	Group A and Group B tests	Comprehensive and preservice tests
Pressure	±2	±1/2
Flow rate	±2	±2
Speed	±2	±2
Vibration	±5	±5
Differential pressure	±2	±1/2

Source: ASME OM Code–1990, including 1994 Addenda.

Table 4.4 In-service test frequency

Pump group	Group A test	Group B test	Comprehensive test
Group A	Quarterly	NA	Biennially
Group B	NA	Quarterly	Biennially

General Note: NA = Not applicable.
Source: ASME OM Code–1990, including 1994 Addenda.

2. The new comprehensive pump test, performed biennially, requires the monitoring of speed, differential pressure, discharge pressure, flow, and vibration amplitude, with the pump operating within 20% of the design point.

3. The hydraulic acceptance criteria for the comprehensive pump test are tighter for centrifugal pumps.

4. More accurate instrumentation is required for the comprehensive test.

It should be noted that the NRC has not yet endorsed this most recent version of the ASME Code.

At present, the ASME Code does not require the use of spectral vibration monitoring. A proposed modification to the code is in the process of review; it would *allow* the use of spectral vibration monitoring in lieu of increased monitoring frequency for pumps that are in the alert range.

4.4 Industrial and International Standards

Most industries that perform vibration monitoring on their equipment do not have to legally comply with any specific vibration standards for equipment. The motivating force for

Existing

Table 4.5 Group A and comprehensive tests vibration acceptance criteria[1]

Pump Type	Pump Speed	Test Parameter	Acceptable Range	Alert Range	Required Action Range
Centrifugal and vertical line shaft [Notes (2) and (3)]	<600 rpm	V_d or V_v	$\leq 2.5\ V_r$	$>2.5V_r$ to $6V_r$ or >10.5 to 22 mils	$>6V_r$ or >22 mils
Centrifugal and vertical line shaft [Notes (2) and (3)]	≥ 600 rpm	V_v or V_d	$\leq 2.5\ V_r$	$>2.5V_r$ to $6V_r$ or >0.325 to 0.7 in./sec	$>6V_r$ or >0.7 in./sec
Reciprocating		V_d or V_v	$\leq 2.5\ V_r$	$>2.5V_r$ to $6V_r$	$>6V_r$

GENERAL NOTE: The subscript r denotes reference value.

NOTES:
(1) Vibration parameter is per Table ISTB 4.1-1. V_r is vibration reference value in the selected unit.
(2) Refer to Fig. ISTB 5.2-1 to establish displacement limits for pumps with speeds ≥ 600 rpm or velocity limits for pumps with speeds <600 rpm.
(3) Including positive displacement pumps except reciprocating.

Source: ASME OM Code–1990, including 1994 Addenda.

Table 4.6 In-service test hydraulic acceptance criteria

GROUP A TEST HYDRAULIC ACCEPTANCE CRITERIA

Test Parameter	Acceptable Range	Alert Range	Required Action Range	
			Low	High
P (Positive displacement pumps)	0.93 to $1.10P_r$	0.90 to $<0.93P_r$	$<0.90P_r$	$>1.10P_r$
ΔP (Vertical line shaft pumps)	0.95 to $1.10\Delta P_r$	0.93 to $<0.95\Delta P_r$	$<0.93\Delta P_r$	$>1.10\Delta P_r$
Q (Positive displacement and vertical line shaft pumps)	0.95 to $1.10Q_r$	0.93 to $<0.95Q_r$	$<0.93Q_r$	$>1.10Q_r$
ΔP (Centrifugal pumps)	0.90 to $1.10\Delta P_r$	none	$<0.90\Delta P_r$	$>1.10\Delta P_r$
Q (Centrifugal pumps)	0.90 to $1.10Q_r$	none	$<0.90Q_r$	$>1.10Q_r$

GENERAL NOTE: The subscript r denotes reference value.

GROUP B TEST HYDRAULIC ACCEPTANCE CRITERIA

Test Parameter	Acceptable Range	Required Action Range	
		Low	High
ΔP (Centrifugal pumps including vertical line shaft pumps), or	0.90 to $1.10\Delta P_r$	$<0.90\Delta P_r$	$>1.10\Delta P_r$
Q (All pump types) [See Note (1)]	0.90 to $1.10Q_r$	$<0.90Q_r$	$>1.10Q_r$

NOTE:
(1) Measure Q for positive displacement pumps.

GENERAL NOTE: The subscript r denotes reference value.

Table 4.6 (Continued)

COMPREHENSIVE TEST HYDRAULIC ACCEPTANCE CRITERIA

Test Parameter	Acceptable Range	Alert Range	Required Action Range	
			Low	High
P (Positive displacement pumps)	0.93 to $1.03P_r$	0.90 to $<0.93P_r$	$<0.90P_r$	$>1.03P_r$
Δ (Vertical line shaft pumps)	0.95 to 1.03 ΔP_r	0.93 to $<0.95\Delta P_r$	$<0.93\Delta P_r$	$>1.03\Delta P_r$
Q (Positive displacement and vertical line shaft pumps)	0.95 to $1.03Q_r$	0.93 to $<0.95Q_r$	$<0.93Q_r$	$>1.03Q_r$
ΔP (Centrifugal pumps)	0.93 to 1.03 ΔP_r	0.90 to $<0.93\Delta P_r$	$<0.90\Delta P_r$	$>1.03\Delta P_r$
Q (Centrifugal pumps)	0.94 to $1.03Q_r$	0.90 to $<0.94Q_r$	$<0.90Q_r$	$>1.03Q_r$

GENERAL NOTE: The subscript r denotes reference value.

Source: ASME OM Code–1990, including 1994 Addenda.

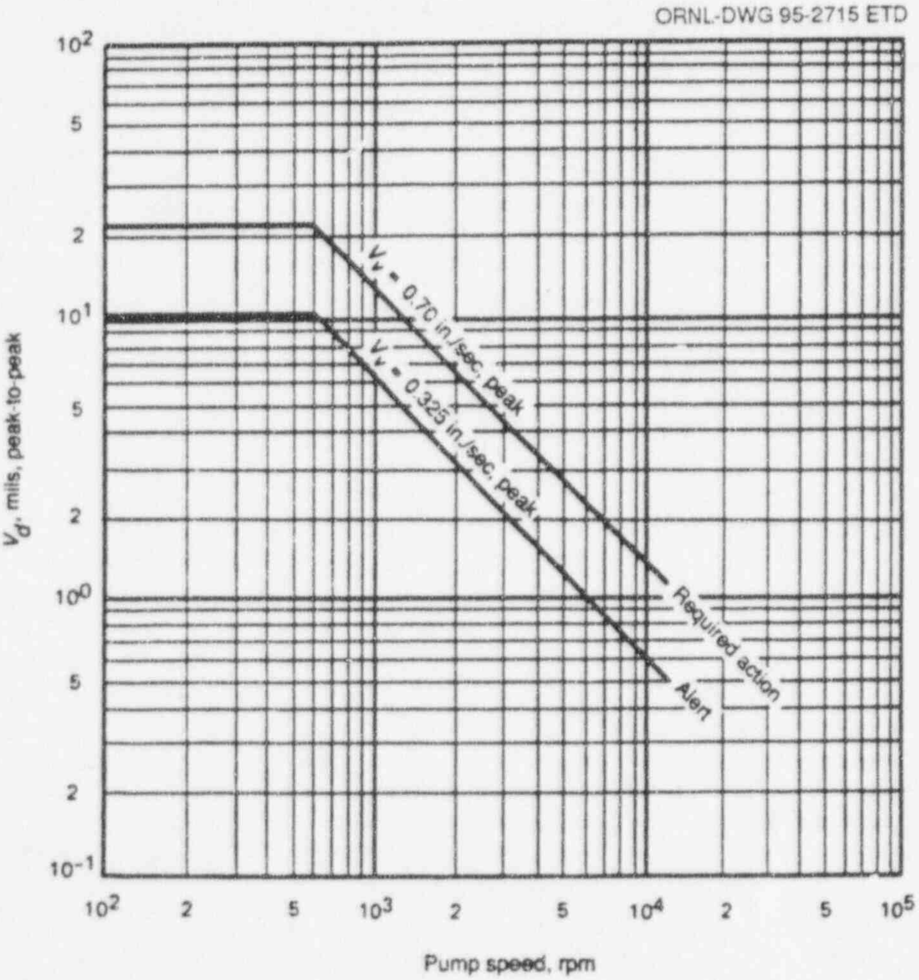


Figure 4.4 In-service test vibration limits. ***Source:*** **ASME OM Code–1990, including 1994 Addenda**

performing dynamic analyses in industry is money: the application of predictive maintenance technologies can augment performance and confirm the dependability of rotating machinery that, consequently, yield increased productivity and reduced downtime. Safety is certainly an added benefit but not the motivating force. One widely accepted and utilized standard for a variety of industries is the American Petroleum Institute (API) Standard 610,[32] which is now in its seventh edition. API standards are published as an aid to procurement of standardized equipment and materials; the standards are not intended to inhibit purchasers and producers from purchasing or producing products made to specifications other than API; the information is simply provided as a guideline. Nevertheless, the long-term and widespread use of API-610 as an acceptance standard is a testimony to its accuracy, its applicability, and general success in providing specifications to measure pump performance.

API-610 specifies allowable vibration amplitudes for both unfiltered (overall) and specific spectral frequencies. Figure 4.5 shows the allowable amplitudes for both antifriction and sleeve bearings. The more restrictive of the velocity or displacement reading is to be applied. The standard requires that vibration be measured on the bearing housing for antifriction-bearing pumps and on the shaft for sleeve-bearing pumps. The allowable amplitudes are for operation within 10% of rated speed and capacity. Running-speed and vane-pass frequencies are specifically called out for spectral domain (filtered) amplitude consideration.

The API-610 standard also calls for five-point performance tests, with suggested flow rates to include shutoff, minimum continuous stable flow, midway between minimum and rated flow, rated flow, and 110% of rated flow (Sect. 4.3.3.2.1). Different hydraulic criteria are applied for shutoff and rated flow points and also vary with rated differential head.

The eighth edition of the API-610 standard is expected to be published in the spring of 1995. Preliminary indications are that the standard will be revised to reduce the overall allowable amplitudes of both the filtered and unfiltered vibration. In addition, the eighth edition of the standard will no longer refer to "peak" velocity, but will rather be based on rms amplitude. The reason for the switch from so-called "peak" to rms is that, in reality, the historical use of "peak" in the vibration community has been $\sqrt{2}$ times the rms value (i.e., based on an assumed sinusoidal waveform) rather than true peak. By switching to the rms-based criteria, API hopes to eliminate this historical source of confusion. A third anticipated change is provision for increasing the allowable amplitude of vibration for situations where operation is outside of the preferred operating range (70 to 120%). Preliminary indications are that a 30% increase in the allowed vibration amplitude will be specified for off-design operation.

Again, it should be emphasized that the guidelines outlined in API 610 are recommendations for procurement specifications and not in-situ testing. However, many industries still use these guidelines as the foundation of their acceptance criteria employed in the evaluation of pumps in monitoring programs or in performance testing after major maintenance. As industrial predictive maintenance programs have matured, extensive historical trends evolving from large data bases can warrant various modifications to acceptance criteria for selected components based on unique differences in foundation, piping arrangements, or any other special type of physical influence that affects spectral signatures and overall vibration levels.

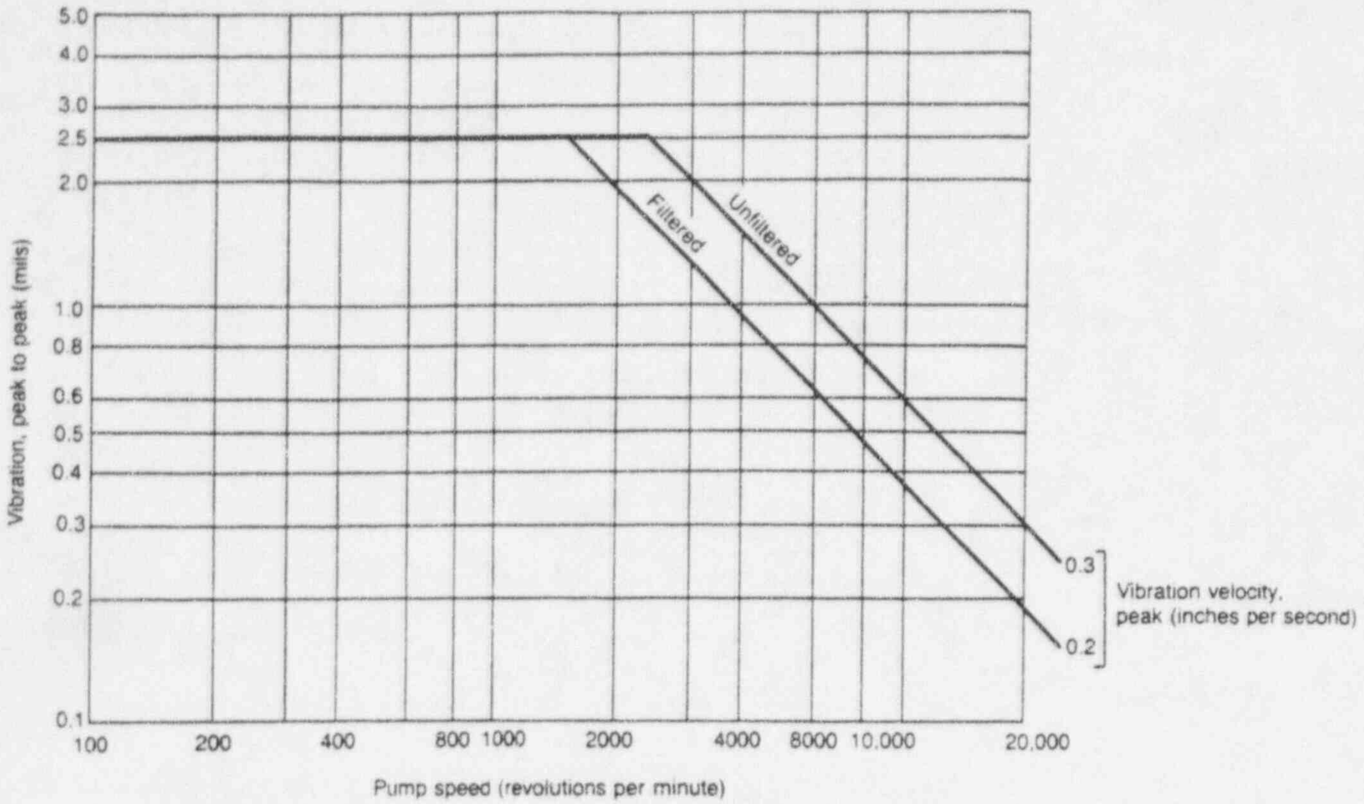
In addition to API-610, numerous industries use the vibration severity chart developed by T. C. Rathbone in 1939. This chart, shown in Fig. 4.6, illustrates a common feature which should be readily apparent in any vibration severity chart: in terms of displacement, tolerable vibration amplitudes decrease with increasing frequency. As previously discussed in Sect. 3.1, the relationship between force, displacement, and frequency requires that an increasing contribution from velocity and acceleration with increasing frequency produces a decreasing displacement in order to maintain a given level of force. Because velocity is probably the best single dynamic parameter describing the overall force, the boundaries between categories of machine condition in the Rathbone chart are lines designating constant velocity.

The International Standards Organization (ISO) publishes two vibration severity criteria that are used by numerous European industries, including several nuclear plants: ISO Standards 2372 and 3945. ISO 2372 was designed primarily for shop testing, and ISO 3945 was developed for the in-situ evaluation of vibration of larger machinery. Both standards, shown as a composite in Fig. 4.7, apply to machinery operating within the speed range from 10 to 200 Hz, 600 to 12,000 rpm, and specify a measurement limited to a frequency band of 10 to 1,000 Hz. Both standards make a distinction between flexible support and rigid support machines. A flexible support is defined as having its first natural frequency below the main frequency of excitation, presumably the machine's running speed. Conversely, a rigid support is one in which the first natural frequency of the support structure is higher than the main excitation frequency.

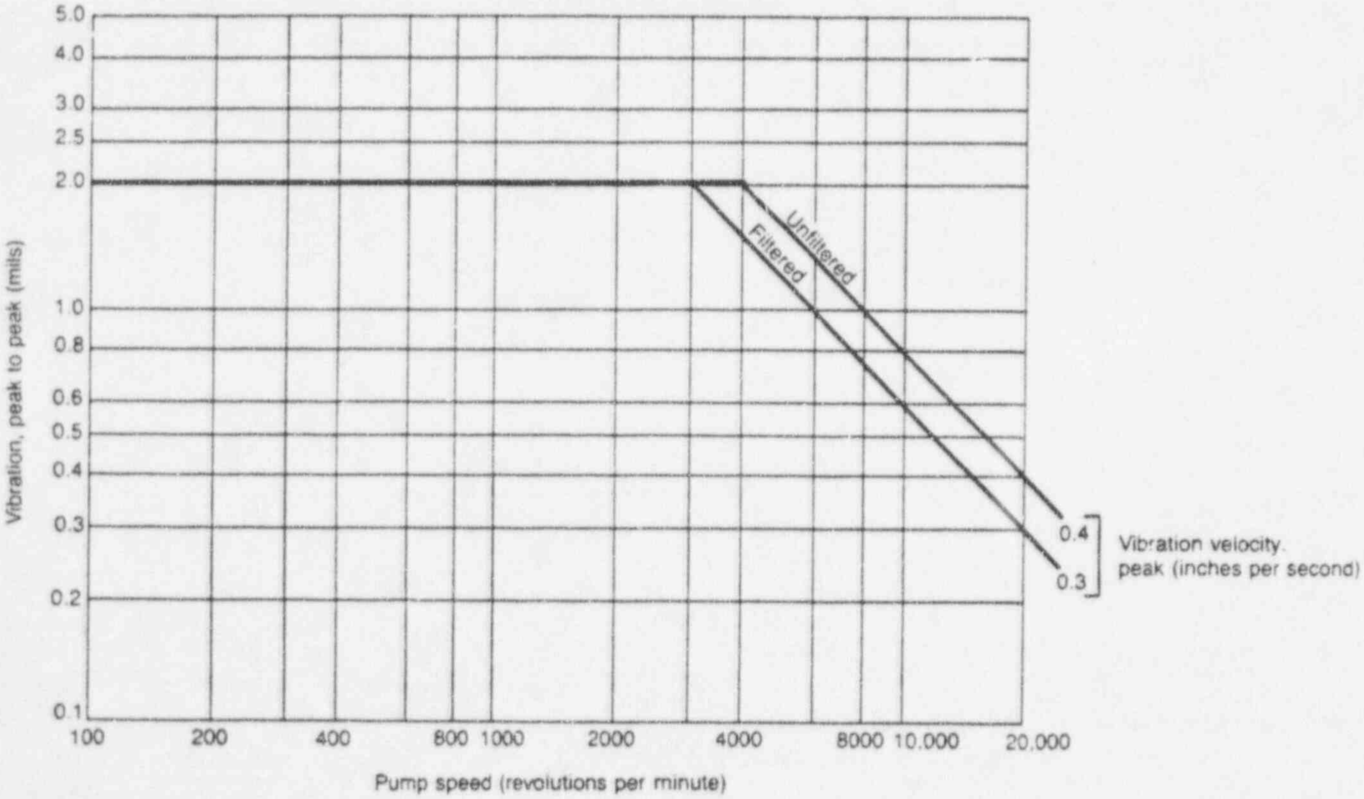
4.5 High Flux Isotope Reactor Monitoring Program

The High Flux Isotope Reactor (HFIR) is a research reactor constructed in the 1960s and is currently managed by Martin Marietta Energy Systems, Inc., for DOE. Like other nuclear reactors in the United States, the HFIR is

ORNL-DWG 95-2714 ETD



Bearing-Housing Vibration Limits (Antifriction Bearings)



Shaft Vibration Limits (Sleeve Bearings)

Figure 4.5 API-610 vibration limits for pumps. Redrawn with permission from "Centrifugal Pumps for General Refinery Service," ANSI/API STD 610-1990, 7th Edition, American Petroleum Institute, November 1990

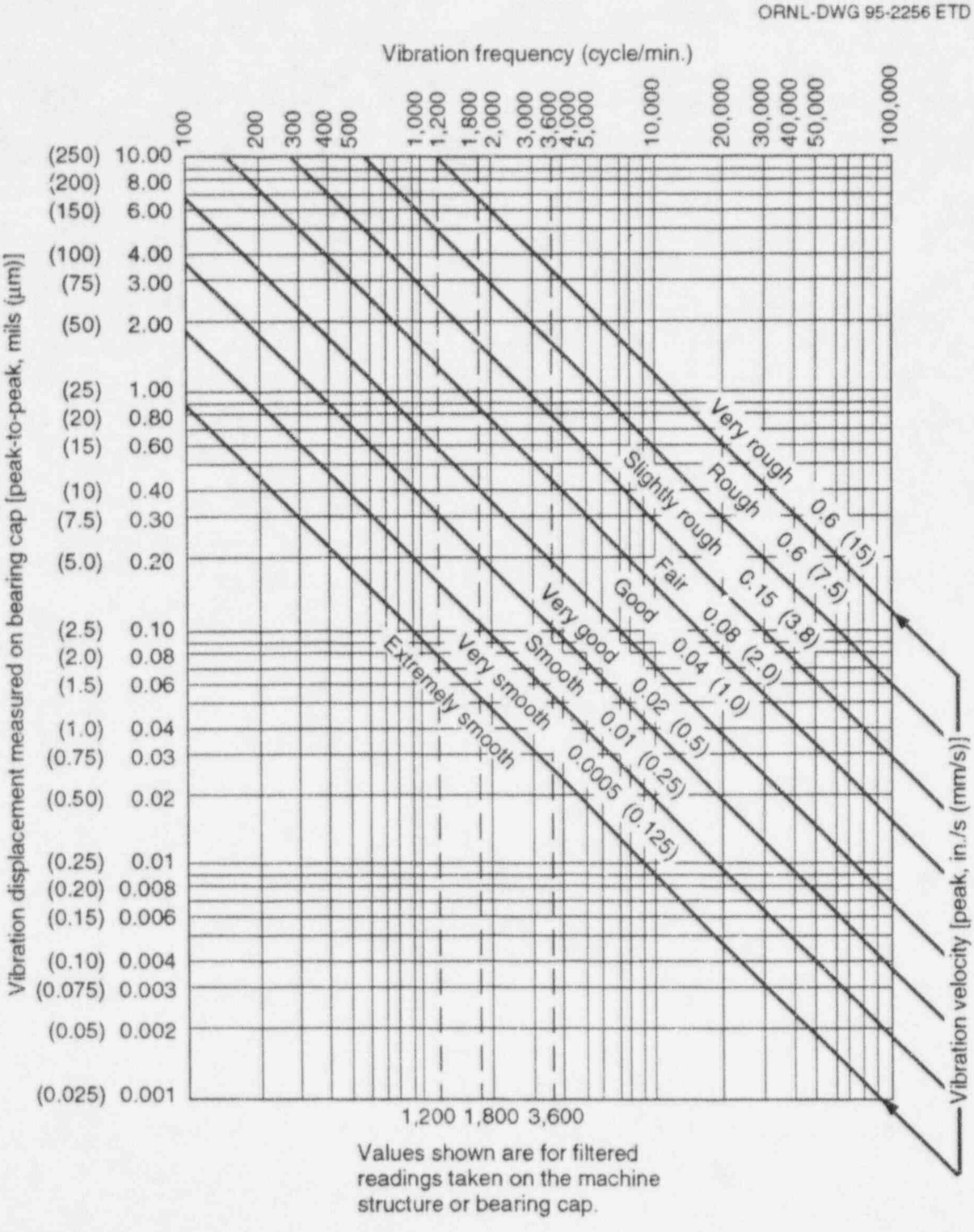


Figure 4.6 Rathbone casing vibration severity chart (at bearing cap). Adapted with permission from J. S. Mitchell, *Machinery Analysis and Monitoring*, Penn Well Publishing Company, Tulsa, Oklahoma, 1981

ORNL-DWG 95-2257 ETD

Ranges of radial vibration severity			Quality judgment for separate classes of machines			
Range	rms velocity In the range 10-1000 Hz at the range limits mm/s	in./s	Class I	Class II	Class III	Class IV
0.28	0.28	0.011	A	A	A	A
0.45	0.45	0.018	A	A	A	A
0.71	0.71	0.028	A	A	A	A
1.12	1.12	0.044	B	A	A	A
1.8	1.8	0.071	B	B	A	A
2.8	2.8	0.11	C	B	B	A
4.5	4.5	0.18	C	C	B	B
7.1	7.1	0.28	D	C	C	B
11.2	11.2	0.44	D	D	C	C
18	18	0.71	D	D	D	C
28	28	1.1	D	D	D	D
45	45	1.8	D	D	D	D
71			D	D	D	D

Machine Classes

Class I — Small machines to 20 hp
Class II — Medium size machines 20 to 100 hp
Class III — Large machines 10-200 rev/s, 400 hp and larger, mounted on rigid supports
Class IV — Large machines 10-200 rev/s, 400 hp and larger, mounted on flexible supports

Acceptance Classes

A Good C Unsatisfactory

B Satisfactory D Unacceptable

Figure 4.7 International standards ISO 2372 and ISO 3945. Adapted with permission from J. S. Mitchell, *Machinery Analysis and Monitoring*, Penn Well Publishing Company, Tulsa, Oklahoma, 1981

undergoing detailed equipment and system studies to maximize operational life and to evaluate opportunities for plant life extension. As part of these studies, a component monitoring program was implemented to accurately assess the operational behavior of plant machinery as well as scrutinize problems that have been identified on selected reactor system support equipment.

The monitoring program employs vibration spectral and waveform analysis techniques on data that are periodically collected, stored, trended, and analyzed on a personal computer by using vibration data base management software. The program has the capability to expand surveillance practices to include OA, IR scanning, and electrical data.

Several different systems are included in the monitoring program: exhaust stack fan system, cooling tower pumps and fans, pressurizer pumps, cleanup system pumps, building cooling system, building compressed air, and the primary coolant pump system. In addition to pumps, other components monitored in the program include motors, fans, blowers, gearboxes, refrigeration compressors, and air compressors. Monitoring periods vary slightly for the various types of equipment. The safety-related pumps and motors are monitored every 2 weeks to 30 d, depending on immediate need and testing schedules. Essential units are monitored every 60 d, and nonessential equipment is monitored every 90 d.

Existing

Interaction with operations has developed so well that it is a routine procedure to acquire the data and store it in a personal computer data base. Presently, a site vibration group collects and reviews the data and makes any necessary recommendations for maintenance actions. When a unit exhibits abnormal vibration readings, advanced field testing is usually performed by a centralized vibration engineering group who supports analysis activities at various DOE plant sites in the Oak Ridge area.

For example, when pressurizer pump data are collected during routine monitoring, additional surveillance is also performed to comply with the Standard Test Procedure (STP), which is required for all safety-related equipment. The STP test results are compared to ASME Sect. XI criteria for determination of operational readiness. In the near future, overall parameter values will be trended and provided to the operations staff to assist with their decision-making capabilities.

Maintenance is utilized when standard repairs are required or field support is needed for additional testing. Notification of incipient or imminent problems is made to the maintenance group to enable them to better schedule repairs for the particular system.

Several areas are monitored where background radiation is high. Portable vibration monitoring equipment is used in these areas in order to acquire a large amount of spectral information in a very short period of time. After returning to the office complex, this information can be down-loaded to the personal computer and extensively evaluated to check the condition of the machine. Using the results of the machinery analysis, personnel can determine existing and future maintenance needs and order any necessary components that may be needed for replacement.

There have been numerous "saves" attributed to this program. For example, during routine monitoring, a deteriorating coupling was identified on a cleanup pump/motor set located in a high background radiation zone. Maintenance was alerted to the probable source of the problem, and the coupling was replaced. Without spectral monitoring, this component might have caused unwarranted damage to the pump and motor that would be more expensive to repair in terms of both the maintenance costs and personnel radiation dose required to return them to service.

Developing looseness is a common problem with aging equipment. Several machines have been diagnosed with this anomaly through routine spectral monitoring. Although some of these machines required advanced analysis techniques to isolate the source of the problem, many problems were resolved by simply tightening various hold-down bolts. Resonance problems can also result from looseness or broken, degraded structures. This problem is often manifested as a gradual rise in running speed vibration and can be characterized by using advanced analysis techniques.

Although the maintenance department has acted upon most recommendations in a timely manner, there is one case history where an identified problem was ignored for almost 2 years and resulted in a major overhaul of a pump. The original problem was a coupling misalignment that was introduced from maintenance activity on the unit. The situation was reported to maintenance, but due to work overloads and other demands, the pump's condition was ignored. The final result was extreme system looseness, arising from deteriorated pump bearings, bearing shaft adapters, and bearing support housings. This problem created a costly overhaul that could have been avoided if the original recommendation had been acted upon in an appropriate period of time.

The primary coolant pumps are very important units that are essential to operation of the HFIR. Ongoing routine monitoring of these units over a 3-year period showed no trends in overall vibration levels that were of any concern. However, when the bearing model numbers were identified and programmed into the vibration data base program, some very small bearing fault frequency peaks became identifiable where none before were distinguishable. The spectral trends were evaluated over the 3 years of stored data. This study indicated that the faults had existed since the beginning of the monitoring program and had not grown at all. These faults were visible on two of the four pumps, exhibited very small amplitudes, and were probably due to installation damage. They are considered to be stable at present but will be carefully watched for future upward trends.

This monitoring program has resulted in a greater awareness of the condition of most of the HFIR's equipment and a higher confidence level of what may be expected of the equipment in the near term. Numerous problems have been solved using the routine vibration spectral monitoring and advanced analysis techniques, some of which resulted in preventing catastrophic failures of equipment. In addition, the machinery monitoring program has fostered proper maintenance scheduling by enabling the staff to accurately prioritize maintenance needs and avoid unforeseen equipment problems.

4.6 Selected International Monitoring Programs

A literature survey was conducted to gather information about European pump monitoring programs. When information was available on pump monitoring, it was usually part of a description of an overall predictive maintenance program implemented at a particular nuclear plant that monitored numerous types of machinery. Although the information that was obtained is categorized by country, this arrangement does not imply that all monitoring in a particular country is the same or similar among its respective nuclear facilities.

Most of the pumps discussed in this section are non-safety related, but are critical to power production. Clearly, the focus of predictive monitoring programs at foreign nuclear facilities is similar to that for domestic nuclear and non-nuclear facilities, namely on the equipment whose failure would be most disruptive to power production.

4.6.1 Czechoslovakia

In 1983, the Dukovany and Mochovce PWR nuclear plants retrofitted modest surveillance systems in their respective reactor units to monitor the condition of primary system components. Each multipurpose system not only acquires vibration data, but also loose parts data and selected process parameters. The various types of data are collected through a multiplexer arrangement and can be stored on a tape recorder. One accelerometer is mounted in a vertically oriented position on each main reactor circulating pump flange near the bearing and has a frequency response range of 500 to 10,000 Hz.

The vibration analysis consists of an evaluation of the overall amplitude (in rms units), trended data comparisons, and assessments of spectral bandwidths. Pump anomalies have been discovered by (1) scrutinizing the step change trends of overall vibration amplitudes (2) comparing the spectral signatures of different main circulating pumps of identical design functioning in the same service, (3) evaluating before and after maintenance signatures on a specific pump, and (4) spectral feature analysis techniques. The latter method identifies specific frequencies with the pump operating speed, impeller blade rotation, and associates a certain bandwidth of frequencies with pump gland system operational characteristics. The users of this monitoring system believe it has successfully provided a means to evaluate the integrity of the main circulating pump's gland system in order to ensure reliable operation of the pump for future periods.[33]

4.6.2 France

In order to minimize maintenance and outage costs, EDF is currently testing an integrated monitoring and diagnostic assistance system: PSAD. PSAD is devoted to on-line monitoring and diagnostics of the main components of nuclear power plants such as the turbine generator, primary circuit, and reactor coolant pumps. In this framework, each monitored component is connected to an ad hoc surveillance system via existing instrumentation. A two-level hierarchy is used for the reporting of events. Initially, local stations exhibit an on-line alarm display and automatically generate a diagnosis. Both local and national analysis stations can access the plant's data base via a computer network to perform high-level diagnostic tasks.

In addition, EDF's pump expert system, DIAPO, is also being evaluated for anomaly analysis. DIAPO is an expert system for main coolant pumps that is included as part of a general monitoring framework whose goal is to improve predictive maintenance capabilities for nuclear power plants. Jeumont Industries, a pump manufacturer, is participating with EDF in DIAPO development. Figure 4.8 displays the major surveillance parameters and monitoring processes provided by main coolant pump instrumentation: bearing temperatures, relative shaft position, pump seal injection flow, and vibration information that includes synchronous, spectral, shock, and resonant frequency data. Besides these parameters, various measurements related to the primary system circuit are also used by the system: temperatures, flow, pressure, and power. Altogether, DIAPO uses about 200 monitoring parameters to perform a main coolant pump diagnostic.[25]

A first version of the DIAPO expert system is currently being implemented to test the knowledge model and validate expertise. This version encompasses about 70 different diagnostic hypotheses and is able to diagnose 90 initial causes of abnormal behaviors. Ongoing DIAPO tasks include the improvement of diagnostic strategies, a better treatment of diagnostic parameter temporal evolution, and the upgrading of the knowledge base.

4.6.3 Germany

Like other European nuclear plants, Germany's diagnostic focus has been primarily directed to the main coolant pumps. There have been two distinct efforts developed to monitor main coolant pump behavior. First, the condition monitoring system (COMOS) was implemented in the Grafenrheinfeld KWU PWR and yielded valuable data when a main coolant pump shaft ruptured in December 1986. COMOS was primarily developed for main coolant pump shaft vibration monitoring, but it has also been applied to general vibration surveillance of passive primary components. The monitoring procedure is based on the

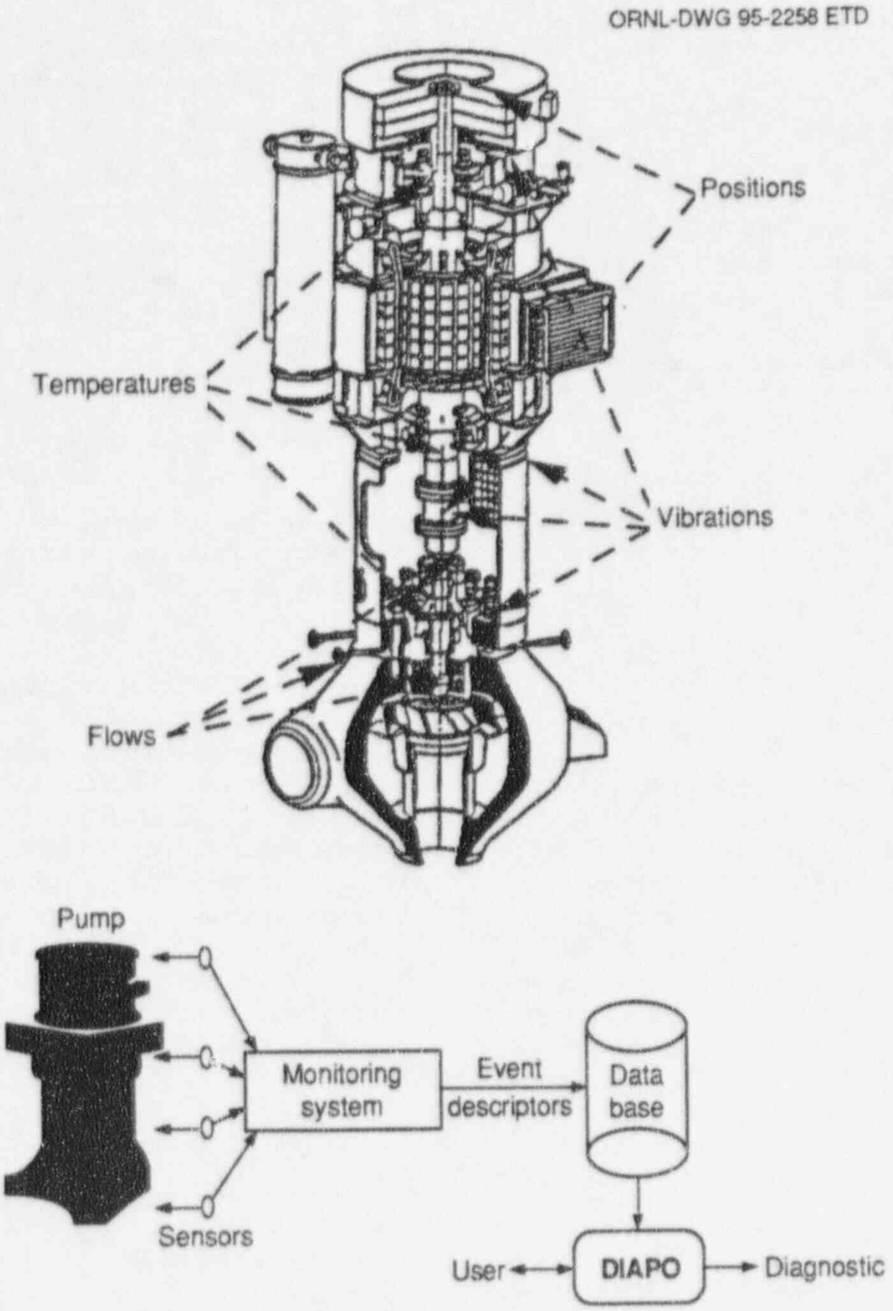


Figure 4.8 DIAPO pump monitoring data and diagnostic process. Redrawn with permission from M. Porcheron and B. Ricard, "DIAPO, an Expert System to Support Reactor Coolant Pump Monitoring and Diagnostics," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992

detection of deviations from reference signatures. For this purpose, statistical discrimination techniques were developed to provide increased sensitivity to changes in spectral features.

The sensors used for the overall system include inductive absolute displacement sensors on top of the reactor pressure vessel, relative displacement sensors at the pump housing and on hot leg positions near the steam generators, neutron noise sensors of the external core safety instrumentation, and piezoelectric pressure sensors located at inlet and outlet pipes. This sensor array has been used for periodic measurements of primary system components and to analyze the signals for undue deviations. The vibration analysis is based on power spectral densities and correlation functions of selected pump signal pairs. In order to achieve a complete understanding of these spectra within a frequency range of interest, systematic measuring campaigns were conducted during pre-operational and operation plant conditions.

Germany's Nuclear Safety Standard Commission requires periodic measurements of the vibration behavior of the primary pumps at least three times per fuel cycle. At the Grafenrheinfeld plant, additional measurements had been made in order to get a better understanding of the long-term spectral characteristics, particularly under modified operational conditions. Pump shaft vibration amplitudes were being monitored on a daily basis via two perpendicular eddy current sensors located nearby the gasket group of the main coolant pump. On November 30, vibration engineers recorded vibration amplitudes in the "x" and "y" directions at 50 and 100 μm, respectively. By December 4, the vibration amplitudes had exponentially grown in the x and y directions to 450 and 500 μm, respectively. In the final stages, the crack propagation of this particular shaft demonstrated a rapid failure development; the vibration amplitude exponentially increased over 150% during the last 48 h prior to complete rupture.

Plant engineers concluded that it was possible to identify the presence of a crack in a radially loaded vertical main coolant pump by means of vibration analysis techniques. Experience at Grafenrheinfeld regarding shaft rupture at main coolant pumps was summarized as follows:

1. Monitoring is based on amplitude trend analysis of rotation specific frequency components.

2. Crack propagation has been observed to develop in the time span of several hours to several months.

3. Changes of operational parameters (e.g., primary pressure, primary temperature, sealing water pressure, sealing water temperature) may influence the shaft vibration signals in a strong way.[34]

Another monitoring technique being tested at German nuclear plants uses acoustic signatures from loose parts monitoring systems (LPMSs) for component condition monitoring and on-line degradation diagnosis on the reactor's primary system. In the last several years, plant engineers determined that impact-induced events that were detected had not only been caused by foreign bodies in natural collection areas, but also by components whose operational status deviated from normal conditions.

The basic requirement for this type of monitoring is the qualified analysis of acoustic signals with detailed burst data interpretation and trending. Acoustic signals collected from measurements and analyses of 12 German nuclear plants, containing more than 6700 bursts, established the data base for the identification of qualified acoustic signal patterns associated with primary system components.

Degradation has been detected in two different types of primary system pumps using these acoustical pattern comparison techniques.

1. Rattle noise indications from LPMS sensors in one PWR plant gave hints that irregularities existed at a main coolant pump. Inspection of the dismantled pump during a subsequent outage showed that the fixing screw of the impeller had detached and the impeller cap was loose.

2. Due to damage resulting from the construction phase, the weld of a cover ring of an internal axial pump of a boiling water reactor (BWR) plant was broken. The ring performed fluid-induced vibrations and hit the pump settlement. The failure could be detected by means of acoustic signal monitoring. During reactor start-up activities, unusual acoustic noise occurred at higher pump speeds and burst signals could be identified in the corresponding sensors. Systematic shutdown of single internal axial pumps identified the affected pump and enabled repair of the component. A complete break off of the cover ring would have resulted in major pump damage.

4.6.4 Sweden

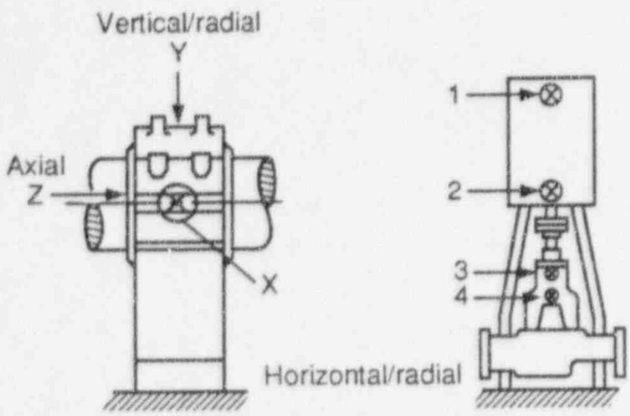
Currently, there are no regulations governing the monitoring of nuclear plant pumps in Sweden. However in 1989, a vibration monitoring program was implemented at the Barsebaeck (BWR) Nuclear Plant. Table 4.7 provides a synopsis of the pumps and machines that are monitored as part of the condition monitoring program at this facility. The monitoring program uses a portable FFT analyzer that collects and downloads data to a personal computer.

The criteria for the selection of accessible monitoring locations consisted of one horizontal, one vertical, and one axial measurement point on each bearing for a total of at least 12 measurement points per pump and motor set (refer to Fig. 4.9). Although bearing proximity mandated the location of most measurement sites, several exceptions were given: the secondary coolant pumps (due to operational needs), the feedwater pumps, and the main condensate pumps. These pumps have measurement points partly on the pump body and also on the closest connecting pipe for the purpose of determining cavitation. Permanently installed velocity transducers are mounted on the main recirculation pumps, the feedwater pumps, the shutdown cooling pumps, the main condensate pumps, and the main cooling water (saltwater) pumps. These latter pumps provide an alarm to the central control room at an early warning limit, but the signals are also accessible by a portable vibration data collection device.

Table 4.7 Barsebaeck's machinery monitoring program

Each	Name of pumps	The measuring collection includes: Electric motor	Pump	Fan	Hydraulic gear unit	Generator compr.	Measuring points per unit each	Security related: Yes	No	Operation pressure bars
3	Feed water pumps	X	X		X		63	X		80,7
4	Main recirculation pumps	X	X				8	X		70,0
4	Frequency changer assembly	X			X	X	96	X		–
2	Reactor water shutdown cooling pumps	X	X				4	X		72,0
3	Containment spray pumps	X	X				45	X		15,0
2	Core spray pumps	X	X				42	X		20,0
2	Auxiliary feed water pumps	X	X				24	X		100,0
3	Main condensate pumps	X	X				24	X		30,0
2	Auxiliary condensate pumps	X	X				18	X		30,0
3	Control oil turbine pumps	X	X				27		X	20,0
2	Generator cooling system pumps	X	X				24		X	24,0
4	Main cooling water pumps	X	X				36	X		1,5
4	Gas turbine lubricating oil pumps	X	X				36	X		5,0
3	Service water pumps	X	X				36		X	5,0
3	Closed circuit cooling pumps for startup and shutdown	X	X				36	X		10,0
3	Closed circuit cooling pumps for plant operation	X	X				36	X		10,0
3	Distribution pumps for demineralized water	X	X				36	X		5,0
2	Vent fans for active areas	X		X			24	X		–
2	Fans for generator bar cooling	X		X			12		X	–
2	Air compressor for manoeuvre air	X				X	20	X		9,0
2	Pressurized nitrogen gas system compressor	X				X	18	X		27,0
Summary on each plant		58	46	4	4	8	685			

ORNL-DWG 95-2259 ETD



Barsebaeck monitors most motors and pumps that are easily accessible. Measurement sites usually include one horizontal, one vertical, and one axial measurement point at each bearing.

Figure 4.9 Barsebaeck's pump monitoring locations

Each measurement point is identified with an identification tag that is attached with high-temperature tape. In addition to overall amplitudes, spectral data are acquired at each point and characterized by up to seven frequency bands that are subsequently trended. Other process parameter data that are collected, stored, trended, and analyzed include pressure, flow, and temperature. Machines are monitored on a monthly basis. Vibration amplitude limits are established according to ISO 2372. When this standard is not applicable, limits are created by the agreement of the machine's operating history and the input from the plant predictive maintenance project group. This group comprises representatives from the operation department, mechanical maintenance department, and the electrical maintenance department.

Although no cost benefit analysis has been performed on the program, Barsebaeck greatly values its monitoring program and supplied a case history of cavitation detection in a main condensate pump. The inducer mounted in the bottom of this pump is supposed to prevent cavitation in the first stage. During routine data collection measurements that included this particular pump, the spectral data revealed features indicative of numerous loose materials, nonharmonic frequency components, and a high vane pass frequency (refer to Fig. 4.10). Cavitation damage was suspected. During the subsequent outage, the pump was inspected, and the diagnosis was confirmed. Material had become loose from the inducer, and cavitation pitting was prevalent on the first stage impeller. The pump had to be rebuilt, and permanent velocity transducers were installed on the refurbished unit to facilitate control room monitoring.

4.6.5 Japan

In January 1989, TEPCO suffered a major casualty in a reactor circulating water pump in one of its BWR plants. In the aftermath of that casualty, TEPCO has undertaken numerous measures to learn more about condition monitoring methods and systems and has sponsored research similar to that documented in the EPRI-NMAC report entitled the "Predictive Maintenance Primer."

One action resulting from this research was the installation of eddy current shaft position sensors on all of TEPCO's reactor circulating pumps to measure any shaft deflection from the normal rotating axis on a continuing basis. The loss of the pump-end hydraulic bearing and subsequent shaft seizure in the failed pump was caused by a defective weld in the bearing. As an additional precaution, TEPCO mounted an accelerometer on the pump casing to monitor impeller movement as indicated by cavitation.

TEPCO scheduled, in July 1993, the implementation of a new monitoring program for the safety-related pumps in one of their nuclear stations. The purpose of this program is to evaluate the operational health of pumps by performing system diagnosis such as spectral analysis of vibration data and process information trending. The program will collect, analyze, and trend vibration spectral data, acoustic data, and process information data on at least a monthly basis. Figure 4.11 depicts the data acquisition and analysis methodology utilized by the Toshiba Corporation to accomplish TEPCO's program goals: (1) a data collection route (defining various machines and their respective data points) is loaded via a memory card into a portable data analyzer, (2) measurements are acquired and stored in the memory of the device, and (3) the information is downloaded into a personal or small mainframe computer. The types of analyses that can be conducted include time waveform analysis, frequency analysis, cascade spectral analysis (e.g., waterfall plots), and single or multiple parameter historical trending and comparison. Tables 4.8 and 4.9 provide a thorough synopsis of the new program that details the components being monitored, the location of their measurement points, the various types of measurements, the collection device, and the analysis items.

Toshiba has also developed an off-line prototype expert system named the Maintenance Support Expert System for Rotating Machines (MAINS). Initial development has been based on model pump fault simulations.[35] Although it has not been expressly documented, MAINS software probably interfaces with the software developed for the Toshiba

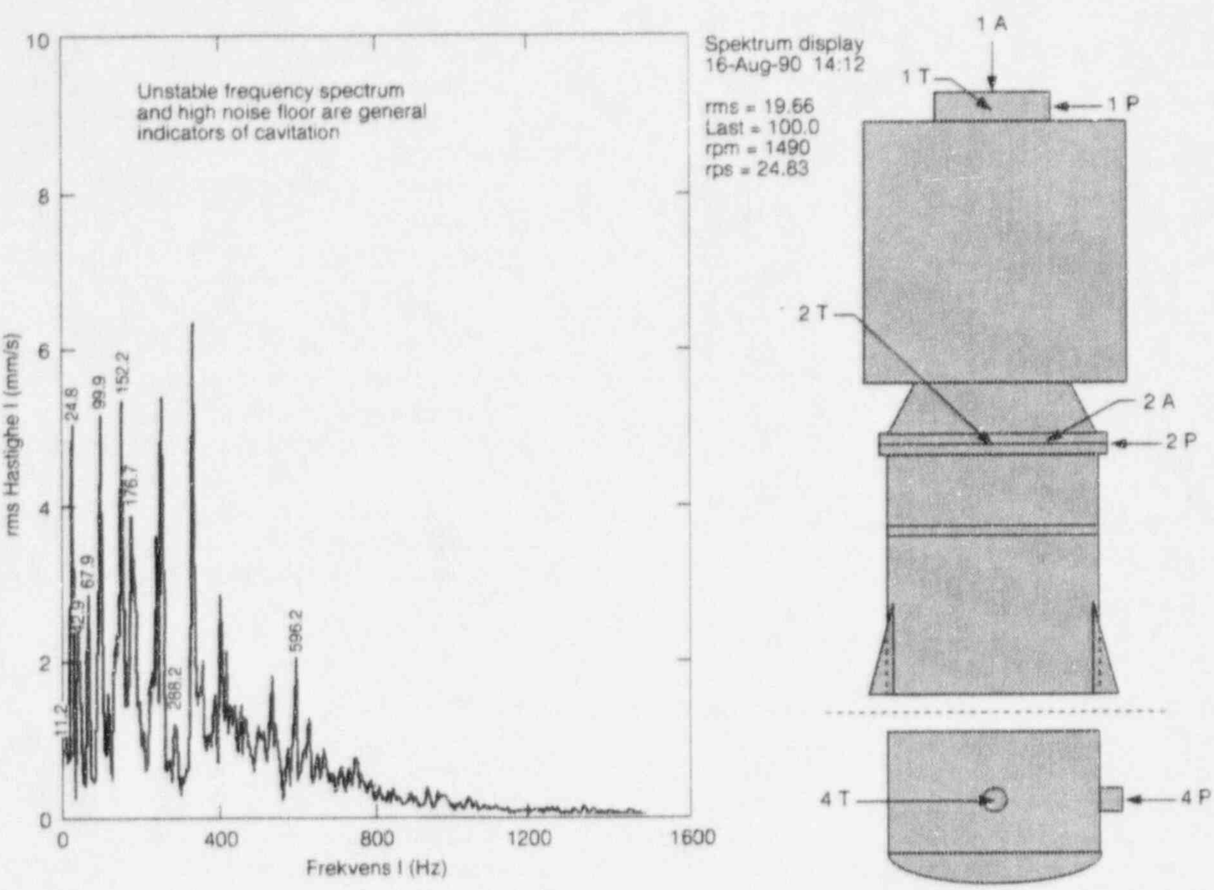


Figure 4.10 Barsebaeck main condensate pump frequency spectra displaying cavitation

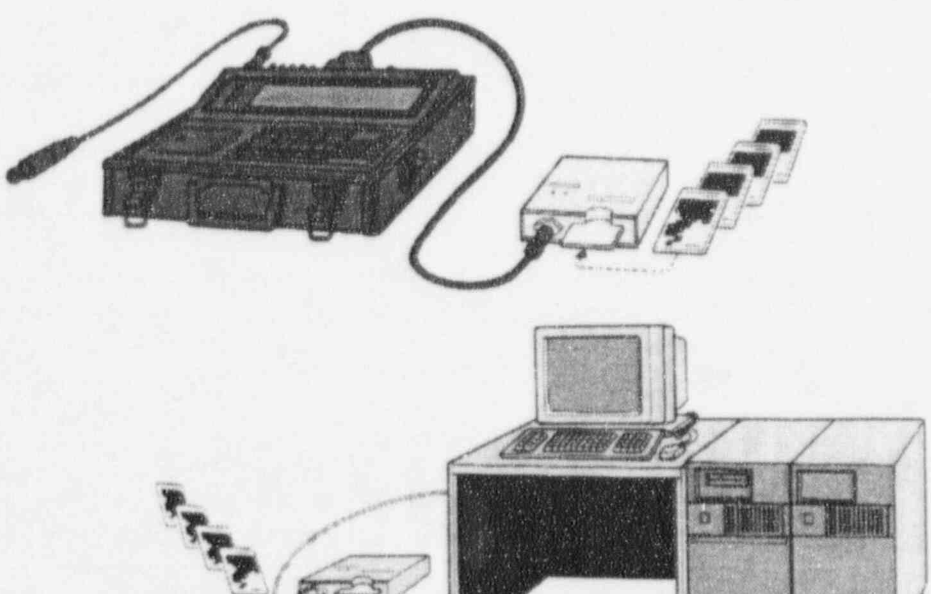


Figure 4.11 Toshiba Maintenance Support Expert System for Rotating Machines (MAINS)

Table 4.8 TEPCO measurement data and check analysis items

Pump name	Measurement item	Detector used in measurements	Check and analysis items
PLR pump	1. Pump shaft vibration (x, y axes direction)	Shaft vibration detector	1. Vibration wave (trending). Check on vibration wave against rotating speed, flow rate, electrical power, etc. 2. Vibration spectral analysis. Perform vibration spectral analysis on vibration signal against rotating speed, flow rate, electrical power, etc.
	1. Vibration at the top of motor	Vibration detector	1. Vibration wave (trending). Check on vibration wave against rotating speed, flow rate, electrical power, etc. 2. Vibration spectral analysis. Perform vibration spectral analysis on vibration signal against rotating speed, flow rate, electrical power, etc.
	3. Process data[a]	Process instruments	Check plant condition and other related components' conditions under PLR pump operation condition.
	4. Pump casing sound	Sound monitor	1. Sound wave (trending). Check on sound wave against rotating speed, flow rate, electrical power, etc. 2. Sound spectral analysis. Perform sound spectral analysis on sound signal against rotating speed, flow rate, electrical power, etc.
TDRFP (A), (B) MDRFP (A), (B) CP (or HPCP) (A), (B), (C) RHR (A), (B)	1. Pump motor vibration[b]	Permanent vibration detector or temporally portable vibration detector	1. Vibration wave (trending). Check on vibration wave against rotating speed, flow rate, electrical power, etc. 2. Vibration spectral analysis. Perform vibration spectral analysis on vibration signal against rotating speed, flow rate, electrical power, etc.
	3. Process data[c]	Process instruments	Check plant condition and other related components' conditions under pump operation condition.

[a]Measurement items: reactor core pressure, reactor core flow rate, electrical power, reactor thermal power, PLR loop flow rate, motor generator (MG) voltage, MG current, PLR motor shaft bearing temperature, pressure difference (between inlet and outlet) of pump, pump seal room pressure, pump seal room temperature, MG speed, jet pump flow rate.

[b]Measurement points (location)

- TDRFP (turbine drive reactor feed pump) (A), (B) pump coupling side, noncoupling side up-down direction by temporally vibration detector (Vertical direction by permanent vibration detector)
- MDRFP (motor drive reactor feed pump) (A), (B) Same as above.
- HPCP (A), (B), (C) [Two operating pumps of three HPCPs] Same as above.
- CP (A), (B), (C) [Two operating pumps of three CPs] X and Y directions on the top of motor
- RHR (residual heat removal) (A), (B) X and Y directions on the top of motor

[c]Measurement items: Reactor core pressure, reactor core flow rate, electrical power, reactor thermal power, pump outlet pressure, pump suction pressure, pump rotating speed, pump flow rate, etc.

Table 4.9 TEPCO data collection period and number of measurements

Pump name	Number of pumps	Number of data collection				
		Start-up	Operation pattern change	Rated power operation	Shutdown	PLR parameter change
PLR pump	2	5	3	1/month[a]	3	1
TDRFP	2	3	3	1/month[a]	2	1
MDRFP	2	1			1	
CP or HPCP	3	4	3	1/month[a]	3	1
RHR pump	2	1[b]		1/month[c]	[b]	

[a]If there is no start-up, shutdown, or operation pattern change during this month.
[b]Collect data for pumps that are operated under reactor cooling mode.
[c]Collect data for pumps that are operated under surveillance mode.

pump monitoring system that TEPCO is using. When an anomalous condition is detected by "plant patrollers," who are the trained personnel performing routine data acquisition, MAINS is provided pertinent data obtained from the affected machine. A "cause-consequence" matrix is generated to assist with pump problem identification. In addition, historical trend data analysis evaluates data tendencies via curve fitting techniques and estimates when the data will exceed a predetermined limitation value. As a result, the trend monitoring enables plant staff to detect a possible anomalous machine condition in an early stage. Furthermore, recommendations are presented that detail a reaction guide for plant operations.

4.6.6 Korea

Another expert system has been developed and implemented at the Korea Electric Power Corporation's Kori-2 PWR nuclear power plant. It is used for the diagnosis of three main systems—the rod control system, the reactor coolant pumps, and the pressurizer. The system diagnoses malfunctions quickly and offers appropriate guidance to reactor operators. The method of inference applied to this system is rule-based deduction with certainty factor operation. The diagnostic symptoms include alarms, indication lamps, parameter values, and valve lineup that can be acquired at the main control room.

The reactor coolant pump was selected as one of the target expert system domains because of its importance to safe reactor operation and due to the frequent failures that have been reported. The pump is divided into three domains: the motor, the seal, and hydraulic systems. Nearly 30 control room alarms can be related to domains inherent to the reactor coolant pump. When anomalous symptoms initiate alarms to the control room, the expert system seeks to determine a primary causal alarm among existing multiple alarms, automatic actions, and probable failure modes; these are classified as obvious symptoms. The nonobvious symptoms are instrument readings, parameter trends, plant computer signals, and valve lineup. All symptoms are evaluated to diagnose probable cause(s) and to develop a guide to the operator's emergency actions and follow-up treatment. The classification of reactor coolant pump malfunctions is shown in Fig. 4.12. The general failure modes include seal failure, vibration, loss-of-seal injection flow, loss of component cooling water, electrical bus failure, and high bearing temperature.[36]

The primary difference between the Korean expert system and other expert systems evaluated in this report is the intended user. This system is used to guide the reactor operations personnel, not maintenance staff. The goal of other expert systems is to gain enough forewarning of abnormal component operation so that the component and/or system does not degrade to a mechanical condition or state that warrants reactor operator response or intervention.

ORNL-DWG 95-2762 ETD

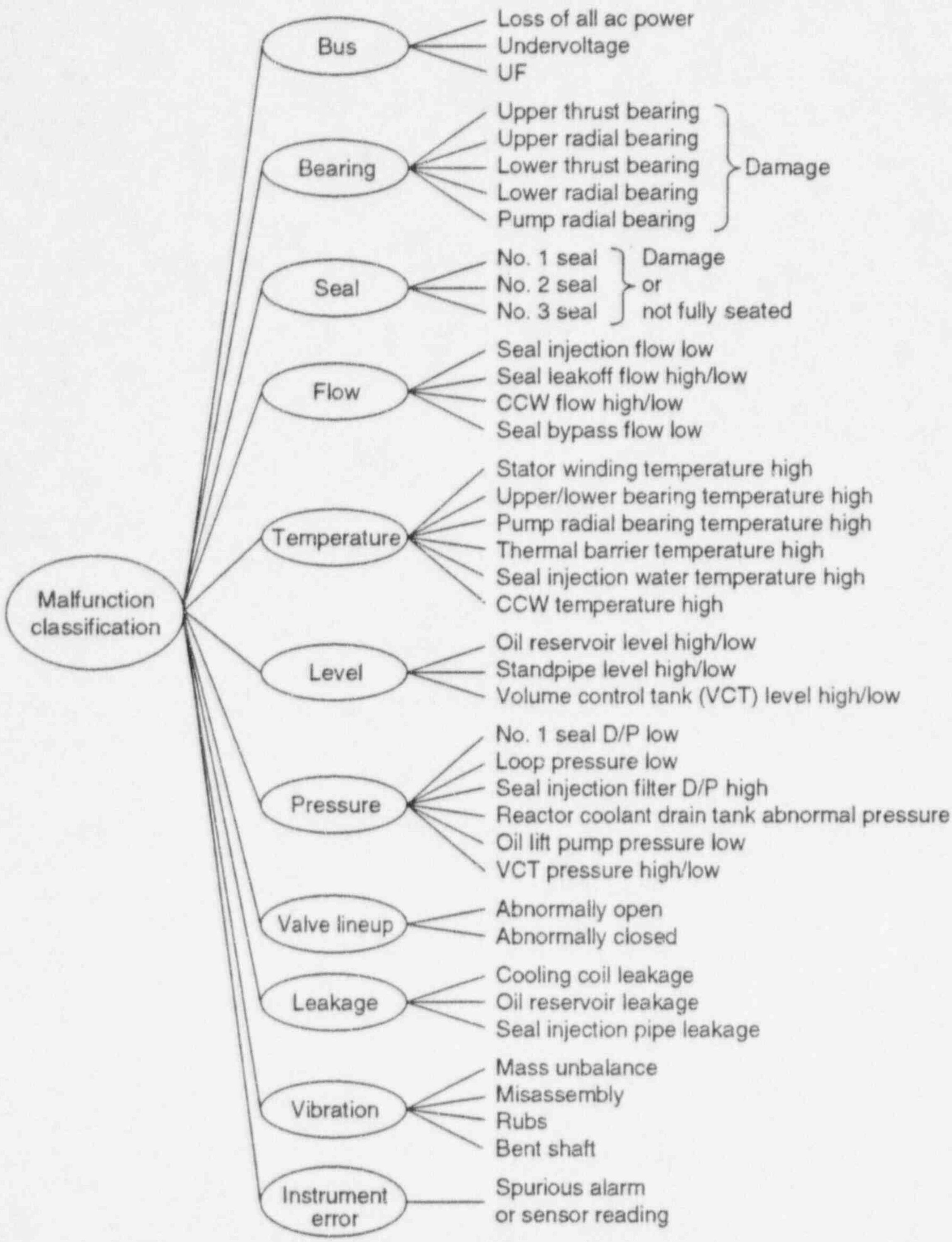


Figure 4.12 Kori-2 reactor coolant pump expert system malfunction classification. Redrawn with permission from S. W. Cheon et al., "Development of an Expert System for Failure Diagnosis of Primary Side Systems," *Nuclear Technology*, 97 (1), 1–15 (January 1992)

5 Detection of Motor Degradation

5.1 Common Commercially Employed Motor Diagnostic Techniques

Most pumps used in both nuclear and non-nuclear applications are motor-driven. A preliminary review of recent pump and pump motor failure data from the Nuclear Plant Reliability Data System (NPRDS) (managed by the Institute for Nuclear Power Operations) indicates that a significant fraction (>25%) of pump and pump motor failures* occurs with the motor.

This chapter discusses methods used to diagnose motor conditions. Although the motor is obviously a separate component, it is important to recognize that the pump and motor are not only directly coupled, mechanically speaking, but the reliability of the pump is inherently linked to the reliability of the motor.

An EPRI study[37] indicated (based on a utility survey) the following sources of motor failures:

- Bearing-related—41%
- Stator-related—37%
- Rotor-related—10%
- Other—12%

It is important to recognize that bearing-related faults are, in many cases, not directly the result of inherent bearing defects, but rather damage caused by loads present during operation. It is likely that a significant number of the motor bearing-related failures that have been experienced are at least partly attributable to rotor and stator degradation-related factors. Potential vibration sources for motors are numerous, including vibration that directly results from the motor itself, as well as from the device to which it is coupled. The motor-generated vibration categories have been classified as follows:[38]

- Magnetic,
- Thermal,
- Electrical,
- Mechanical,
- Fatigue and Deterioration,
- Torsional, and
- Aerodynamic.

Off-line and on-line monitoring of motors can often identify the existence of motor-related faults in the incipient stage before the fault progresses. For example, significant magnetic unbalance (which can result from a variety of causes) will eventually result in damage to other components, such as bearings. However, the existence of magnetic unbalance can be detected by both off-line and on-line diagnostic methods and corrected before additional damage is incurred.

Historically, off-line testing of motors has been more frequently used than has on-line testing. Some commonly applied off-line tests include the following:[39–42]

1. Measurement of resistance to ground with the application of a high dc voltage ("megger") tests;
2. High potential (or "hi-pot") tests, in which higher voltages are applied and leakage current to ground is monitored;
3. Surge tests, in which dc voltage pulses are applied to two phases simultaneously, and the ring down from the pulses is compared;
4. Phase-to-phase resistance measurements; and
5. Phase-to-phase inductance measurements.

The first three of these off-line tests are oriented toward monitoring the condition of the stator, with primary focus on stator insulation (resistance and dielectric strength) condition. Megger and hi-pot tests are used to detect the condition of the insulation relative to ground, while surge tests can indicate the presence of interphase or interturn faults as well as degraded resistance to ground. Meggering is the most commonly used technique; discussions with utility personnel indicated that it is a common practice to megger motors and measure the insulation polarization index (defined in IEEE Standard 43-1974[43]) on approximately a refueling frequency basis. At present, hi-pot testing and surge testing appear to be less commonly used and are often invoked only if there is already some question about motor condition. The use of hi-pot and surge testing has met resistance from some users, since insulation failures can occur during the tests themselves. Examples have been cited[39,41] where megger testing indicated adequate insulation, only to have subsequent surge testing show (or result in) insulation failure.

---

*The term "failures," as used here, refers to failure records in the NPRDS data base. Not all such *failures* truly involve failure of the pump to be capable of delivering flow; ~40% of the motor failures did involve sufficient degradation that the motor was either not capable of performing at all (e.g., cracked shaft) or was in severe distress (e.g., overheated bearing).

The failure data considered were for several safety-related systems at BWR and PWR plants. The period considered was 1990–1993. A detailed study of the failure data, including a characterization of each reported event, is planned, and a future report will document the results.

The fourth technique, phase-to-phase resistance measurement, is primarily useful in detecting short-circuited or open-circuited turns in the windings. The last test, inductance measurement, provides a useful indication of inductive coupling between the stator and rotor, as well as degraded stator windings. Proper implementation of this type of test can provide indication of rotor-related problems such as static or dynamic eccentricity as well as an indication of rotor bar condition. It can also indicate stator-related problems, such as phase-to-phase or turn-to-turn shorts. Available commercial systems typically combine two or more of these (as well as other) test methods in order to more fully evaluate motor condition.

It is important to recognize that these off-line tests are often conducted after the motor has cooled down. As a result, measurements of temperature-dependent conditions, such as rotor bow, insulation integrity, and rotor bar resistance observed during such tests may not always reflect the conditions that exist during motor operation. Suppliers of test systems (particularly those that measure inductance) often encourage the use of the test equipment immediately after motor shutdown in order to provide the closest representation of running conditions. Unfortunately, it is often difficult to accommodate this need, particularly in the case of nuclear plants where safety-related equipment is involved, due to clearance tagout procedures, technical specification operational requirements, and other administrative factors. Example inductance measurement data are presented in Sect. 5.2.

On-line testing methods using motor current analysis techniques have been employed in recent years. The available commercial systems are primarily focused on rotor condition monitoring and generally rely on spectral analysis of the motor current. The current is normally monitored by use of clamp-on current transformers or hall-effect probes either directly on one of the phase power cables or indirectly by clamping on a permanently installed current transformer's secondary-side lead. In general, these methods rely on the comparison of slip-pole sideband amplitudes to the line frequency amplitude. Additional discussion on this approach is provided in Sect. 5.3.

Monitoring of all three phases of current can be used to detect unbalanced load distribution among the motor phases. It is important to recognize that an unbalance in current amplitudes can result from a variety of sources, including inductive or resistive unbalance of the motor phases, variations in the resistance in the leads and connections to the motor, and unbalance (amplitude and/or phase angle) in the power supply voltage. A simple summation of the three phases' current waveforms can also provide an indication of insulation breakdown (the sum of the instantaneous currents will equal zero if there is no current flow to ground).

Motor current and/or power spectral analysis has also been found to provide a useful indication of driven device conditions. Motor current signature analysis, which was originally developed at Oak Ridge National Laboratory (ORNL) (reference NUREG/CR-4234, Vol. 2[44]), has been used in diagnosing the condition of devices driven by motors. The technique is based on the recognition that load fluctuations will result in fluctuations in the current to the motor; time and/or spectral domain analysis of the current fluctuations can help understand the sources and amplitudes of the load fluctuations. A straightforward extension of this technology is the application to spectral power monitoring. Chapter 6 provides results of monitored pumps using power measurements.

Vibration analysis of motors provides useful information about motor mechanical conditions that is not readily available from current or power analysis. This is particularly true for antifriction bearings, which must be significantly degraded (near the point of catastrophic failure) before degradation can be detected from motor current/power signals. However, it is important to point out that overall vibration amplitudes are also somewhat insensitive to bearing conditions, since overall vibration amplitude is dominated by relatively low-frequency components, such as mechanical unbalance or misalignment. In addition, spectral vibration analysis will not always provide an accurate indication of motor electrical/magnetic conditions.

Off-line and on-line monitoring methods have complementary strengths and weaknesses. For example, insulation degradation that has not developed to the point of affecting motor performance (and therefore has not been observed during normal operation) may be detected by off-line test methods such as megohm, hi-pot, or surge tests. Alternatively, in some cases, an insulation defect may exist at operating conditions (specifically at elevated temperatures), but not be detected at ambient temperature conditions even when a higher than normal voltage is applied.

Another example of the complementary nature of off-line and on-line monitoring is for rotor dynamic eccentricity and rotor bar crack detection. Off-line test systems have proven capable of accurately detecting these conditions if the condition exists when the testing is performed. However, because of the difficulty in performing these tests with the motor at operating temperature conditions, actual operating problems may not be detected off-line if the defect is sensitive to motor temperature (which can certainly be the case for cracked rotor bars due to unequal axial expansion during operation).

Other diagnostic methods not discussed herein, such as thermography and OA, can provide indication of degradation that will not be as readily detected by the technologies discussed above. In summary, it is important to recognize that no single monitoring technique will identify all potential motor degradations prior to failure; however, a combination of off-line and on-line monitoring can be very effective in maintaining highly reliable motors.

5.2 Example of Off-Line Data

Measurement of phase-to-phase inductance provides an indication of the balance of inductive coupling between the rotor and the stator for the individual phases. It can further provide an indication of dynamic and static eccentricity that may exist.

Figures 5.1 and 5.2 chart the measurement of inductance of individual phases of a pump motor as the rotor was manually turned through one full rotation. (It should be noted that the actual measured data was phase-to-phase inductance, from which the individual phase inductances shown in the figures are calculated). The data shown in Fig. 5.1 were gathered with the motor at room temperature conditions, while the Fig. 5.2 data were acquired after operating the pump driven by the motor at full load conditions for about one-half hour, thereby elevating the motor temperature. In Fig. 5.1, the approximately sinusoidally varying amplitude for each of the three phases results from mechanical and/or magnetic rotor eccentricity. The eccentricity could be attributable, for example, to rotor bowing or out-of-roundness; alternatively, it could result from magnetic eccentricity caused by nonhomogenous permeability of the rotor.

Figure 5.2 shows the results of measurements made after the motor was warmed by running. The inductance for each of the three phases increased; the average inductance increase was about 8%. The cause of this increase in inductance (which is measured with an inductance meter, using a 1-kHz signal) is attributed to the overall reduction in the gap between the rotor and stator that occurs due to rotor heating (stator heating also occurs, but not as greatly as the rotor, since this motor is a totally enclosed, fan-cooled motor). The angles of peak inductance for all three phases shifted by ~25° (advanced, according to the arbitrary rotational angles selected) after running the motor. This shift is attributable to a change in the orientation of the slight mechanical bow of the rotor.

ORNL-DWG 95-2263 ETD

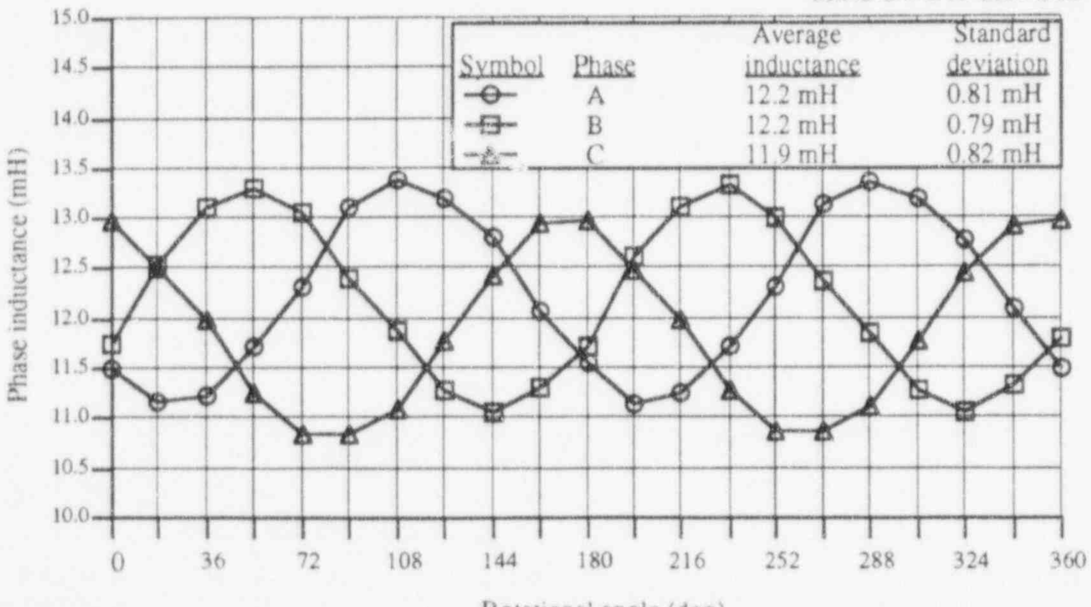


Figure 5.1 Individual phase inductance for 7.5-hp pump motor at room temperature

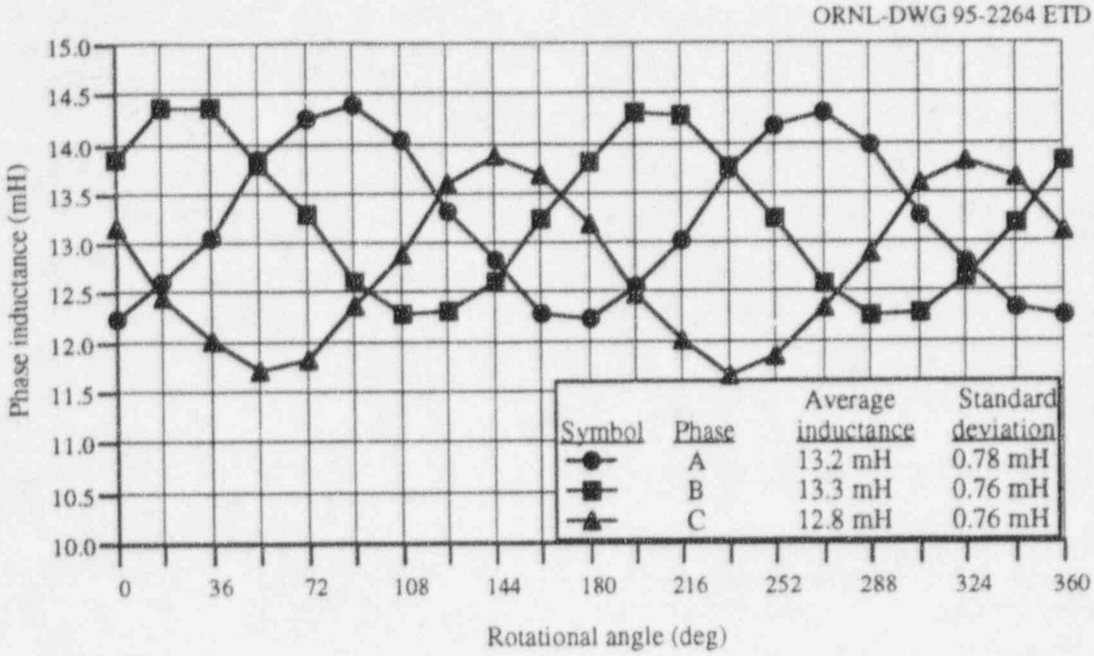


Figure 5.2 Individual phase inductance for 7.5-hp pump motor immediately after motor shutdown

In both the cold and hot conditions, the inductive imbalance (maximum deviation from average) at any angle is slightly over 10%. The average inductive imbalance, based on the overall average of the full rotation, is 1.5% cold and 2.4% hot. Neither of these imbalance measures is considered excessive, based on available historical data from a variety of motors. One manufacturer (motor user) that has employed inductive imbalance measuring on a consistent basis has found that a phase-to-phase imbalance of 10% (equivalent to a 20% imbalance for the individual phases) is a useful threshold for determining acceptability for reliable operation for installed motors. A recent trade journal article[45] suggests that new motors should have a maximum inductive imbalance of 5%.

5.3 Examples of On-Line Data

In order to provide an example of the types of results that can be achieved from on-line motor monitoring, some sample test results will be provided. The results presented are intended to illustrate both the capabilities and some difficulties associated with on-line monitoring.

5.3.1 Unconditioned Motor Current Spectra—Some Example Data

The most frequently monitored on-line parameter for motors is current. The extent of monitoring can range from a simple observation of an analog or digital current indicator to spectral analysis of the motor current waveform. This section explores the usefulness of spectral analysis of current for motor condition monitoring based on examples of motor current data that have been collected at ORNL over the last couple of years.

5.3.1.1 Degradation Testing of a 38-hp Motor Rotor

A three-phase, four-pole, 38-hp motor was tested at several controlled degradation states in a test conducted by a utility with assistance provided by ORNL through a DOE industrial support program. The motor load was connected to a generator that was used to control the motor load. A variety of parameters were monitored, including current, voltage, and vibration at several conditions:

- Original condition
- Partially broken rotor bar (~75%)
- One fully broken rotor bar
- Two fully broken rotor bars
- Three fully broken rotor bars
- Partially cracked end ring
- Fully cracked end ring.

The purpose of the testing was to evaluate the capability of several current monitoring diagnostic systems to correctly diagnose motor rotor conditions and to compare the indications available from motor current to vibration spectra. The motor current monitoring systems employed primarily rely upon the amplitude of certain characteristic spectral peaks in the motor current spectrum. The raw motor current spectrum (around 60 Hz) for the original motor condition is shown in Fig. 5.3.

The annotated spectral peak in Fig. 5.3 occurs at what will herein be referred to as a slip-pole sideband of the line frequency (60 Hz). Published literature on slip-related phenomena have often interchanged slip, slip frequency, slip-poles, and other terms. The definitions of terms used in this report and recommended for more general application follow:

$$\text{Slip (rpm)} = n_s - n \ , \tag{5.1}$$

$$\text{Slip (dimensionless)} = \frac{n_s - n}{n_s} \ , \tag{5.2}$$

$$\text{Slip frequency (Hz)} = 60 \cdot \text{Slip (dimensionless)}$$

$$= \frac{\#\ \text{poles} \cdot \text{slip (rpm)}}{120} \ , \tag{5.3}$$

$$\text{Slip-poles (Hz)} = 2 \cdot \text{slip frequency (Hz)}$$

$$= \frac{\text{poles} \cdot \text{slip (rpm)}}{(60\ \text{rpm/Hz})} \ , \tag{5.4}$$

where n_s is the synchronous speed for the motor, and n is the actual speed (both speeds expressed in rpm).

It should be noted that the frequency of the current in the *rotor* is slip frequency [Eq. (5.3)]. Also note that these equations apply to applications where line frequency is 60 Hz.

For the Fig. 5.3 annotated peak, the frequency is 57.7 Hz. Since this is a slip-poles sideband of 60 Hz, slip-poles are 2.3 Hz. The slip (rpm) from Eq. (5.4) is calculated to be

$$2.3\ \text{Hz} \cdot 60\ \text{rpm/Hz} \div 4\ \text{poles} = 34.5\ \text{rpm} \ ,$$

indicating a motor speed (n) of 1765.5 rpm, since synchronous speed for a four-pole motor is 1800 rpm.

Some practitioners of motor current monitoring technology would automatically assume that a slip-pole sideband of a magnitude as high as −39 dB would indicate multiple broken rotor bars and/or other significant rotor asymmetry. A rule of thumb that has been frequently applied is that a sideband of greater than −45 to −50 dB (or even less) suggests one or more failed rotor bars.[46,47] Others have been somewhat more cautious in the explicit characterization of the level of degradation.[10] The Fig. 5.3 data were acquired for a rotor with no faulty bars. Clearance measurements with the motor in the installed condition (this motor had open ends that allowed the mechanics to take crude measurements with plastic feeler gauges) indicated some static eccentricity (i.e., the clearance between the rotor and stator was not circumferentially uniform). A runout check on the rotor indicated that there was only very minor dynamic eccentricity. Both mechanical measurements were made with a cool motor.

Figures 5.4 through 5.6 show the raw current spectrum for one, two, and three broken rotor bars (the original rotor had 73 rotor bars). A progressive increase in the amplitude of the slip-pole sidebands (both negative and positive sides) was found as motor bars were broken. It is important to recognize that the change in amplitude observed was likely a function of the exact bars that were broken.

ORNL-DWG 95-2265 ETD

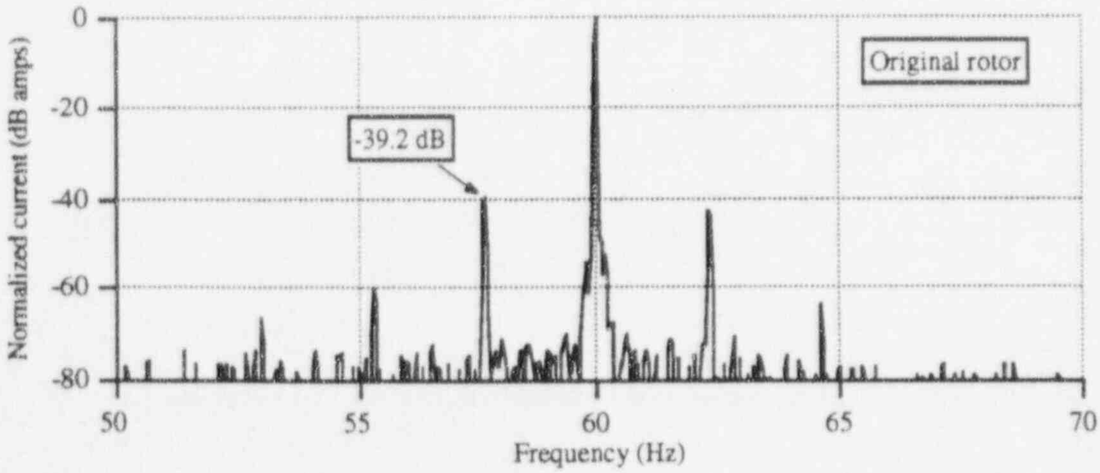


Figure 5.3 Normalized motor current spectrum for the original rotor

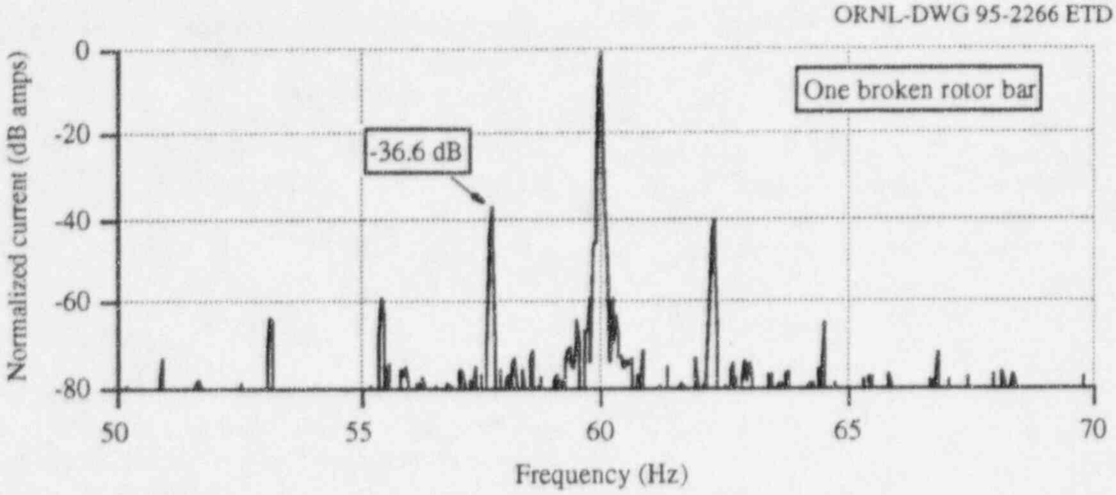


Figure 5.4 Normalized motor current spectrum for one broken rotor bar

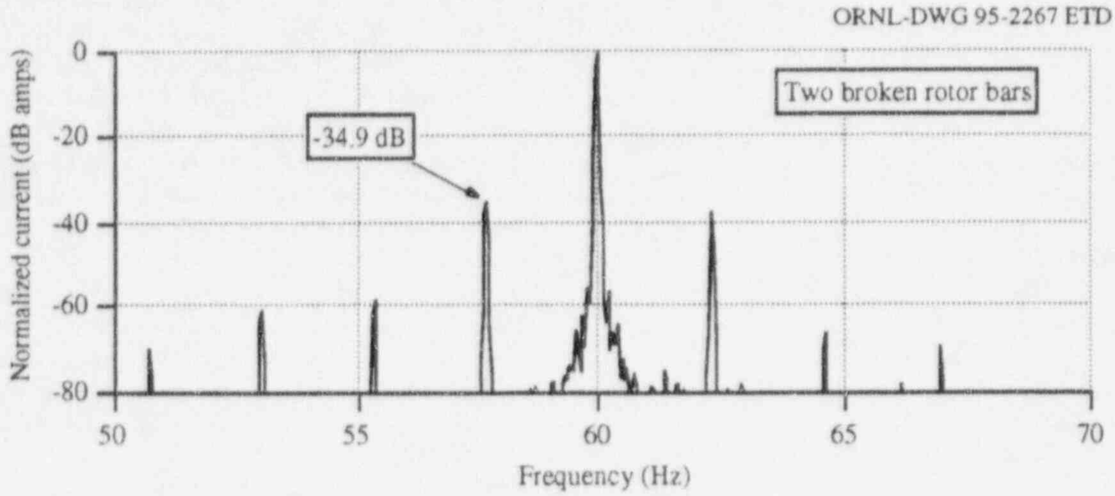


Figure 5.5 Normalized motor current spectrum for two broken rotor bars

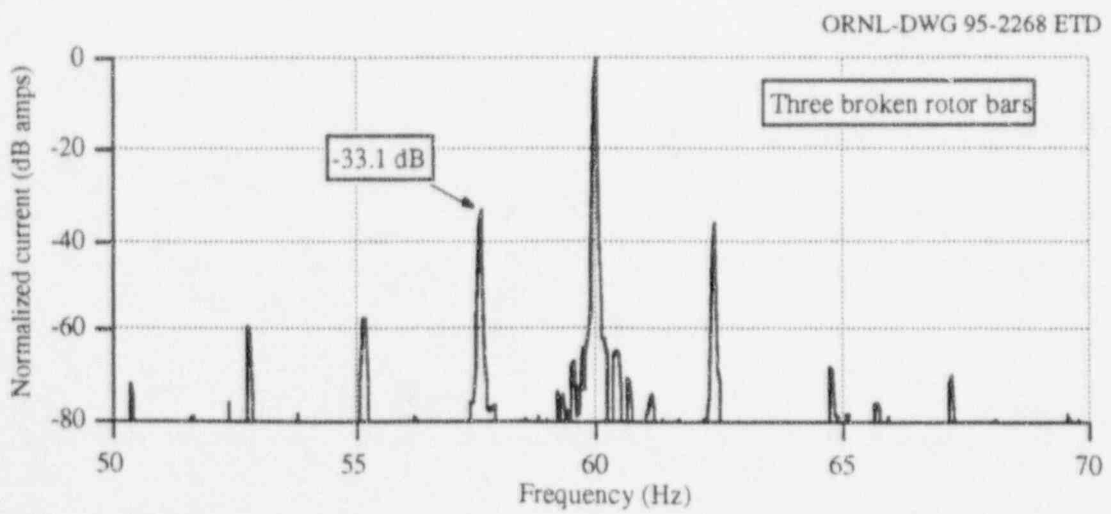


Figure 5.6 Normalized motor current spectrum for three broken rotor bars

The first two bars broken were adjacent; the third bar broken was diametrically opposite the first. It is likely that somewhat different patterns would have emerged had different bars been broken. Although an increase in sideband amplitude with increased rotor degradation was observed, the application of generic amplitude criteria may be difficult when dealing with a broad variety of motors, since the baseline condition obviously can significantly affect the results.

5.3.1.2 Field Monitoring of Pump Motors

"Good" motors often have slip-pole sidebands that are significantly lower than the amplitudes that existed for the base case shown in Fig. 5.3. For example, the unconditioned current spectrum for a 75-hp pump motor is shown in Fig. 5.7. Note that the spectral amplitude at the slip-pole sideband is significantly lower in amplitude and is essentially at the noise floor for this data. The noise floor for the Fig. 5.7 spectrum is higher than for Figs. 5.3–5.5. The principal reason for the higher noise level is pump flow-related load fluctuations that result in broadband noise in the current spectrum. Figure 5.8 shows the unconditioned current spectrum for the same pump motor at an operating condition where the pump is more hydraulically unstable, and as a result, the current spectrum has a higher noise floor. Figures 5.7 and 5.8 clearly indicate that (1) pump as well as motor conditions can significantly influence the current spectrum and (2) a higher level of pump instability could significantly obscure motor-related spectral conditions.

Figures 5.9 and 5.10 show unconditioned current spectra for two identical pumps at a coal-fired facility* that are inherently more hydraulically stable than the test facility pump shown in Figs. 5.7 and 5.8. Data taken on the P7 pump (Fig. 5.9) indicate a slip-pole sideband of approximately the same amplitude as that seen for the motor that was destructively tested. The modulation of the current was sufficient that a current indicator on the motor control center could be visibly seen to "beat" (i.e., the needle swung slightly at the slip-pole frequency). Note that the amplitude of the slip-pole sideband for P8 (Fig. 5.10) is only slightly greater than that in Figs. 5.7 and 5.8, but the frequency is clearly evident in the spectrum due to the lower level of pump-related hydraulic noise.

The motors for both P7 and P8 were being operated below rated load conditions and were not causing other problems for the utility; as a result, the utility chose to leave the P7 motor in service. This type of analysis had not been used by the utility previously, and the engineers were naturally reluctant to take corrective action in the absence of an operational problem.

5.3.1.3 Small Single-Phase Fan Motor Rotor Degradation Test Data

Destructive testing on a small, 1/10-hp fan motor was performed to evaluate the capability of detecting rotor

*The utility operating the plant where the pumps shown in Figs. 5.9 and 5.10 are used is not the same utility that conducted the motor testing shown in Figs. 5.3–5.6.

ORNL-DWG 95-2269 ETD

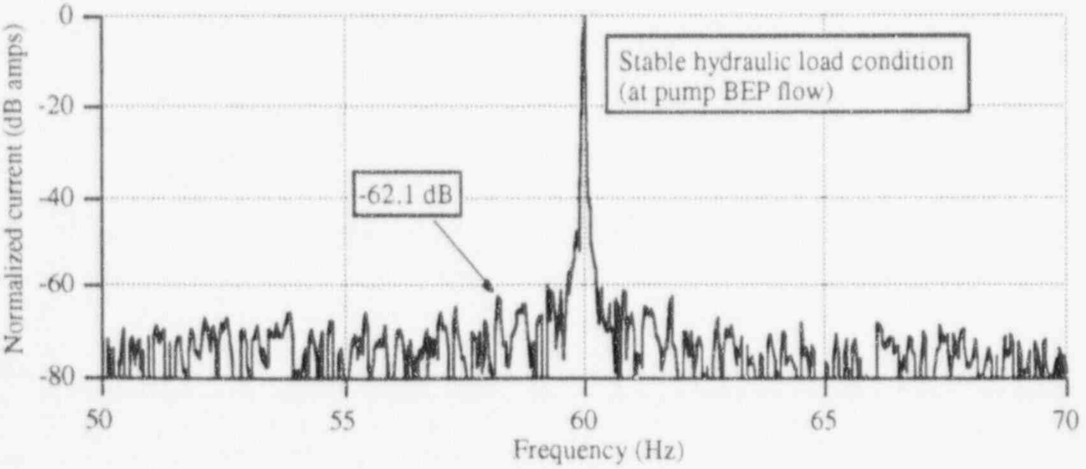


Figure 5.7 Normalized current spectrum for pump motor at rated load conditions for test facility pump

ORNL-DWG 95-2270 ETD

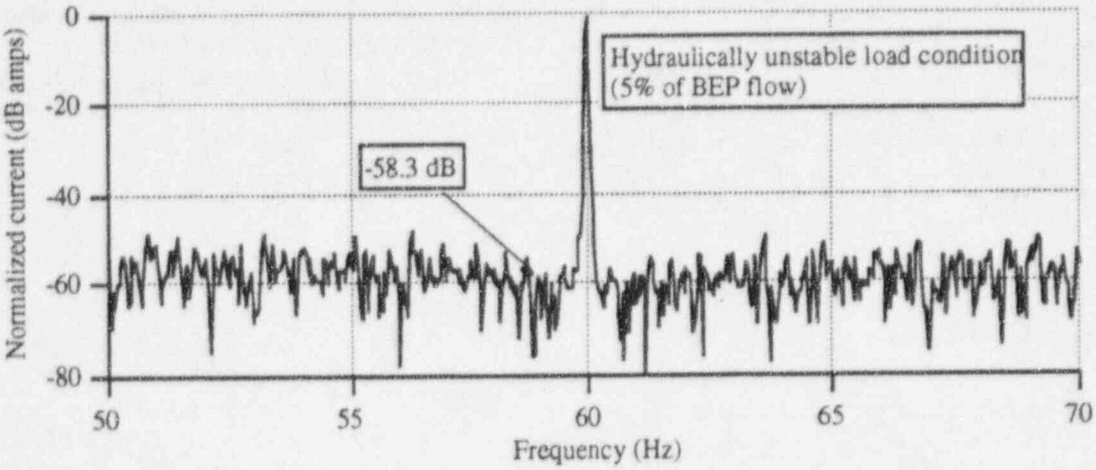


Figure 5.8 Normalized current spectrum with test facility pump at more hydraulically unstable conditions

ORNL-DWG 95-2271 ETD

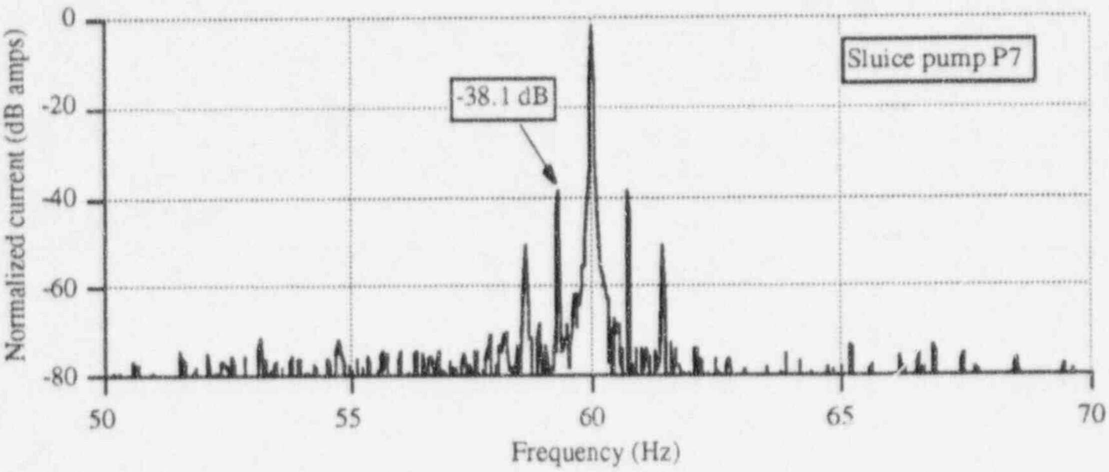


Figure 5.9 Fly ash sluice pump P7 motor current spectrum

ORNL-DWG 95-2272 ETD

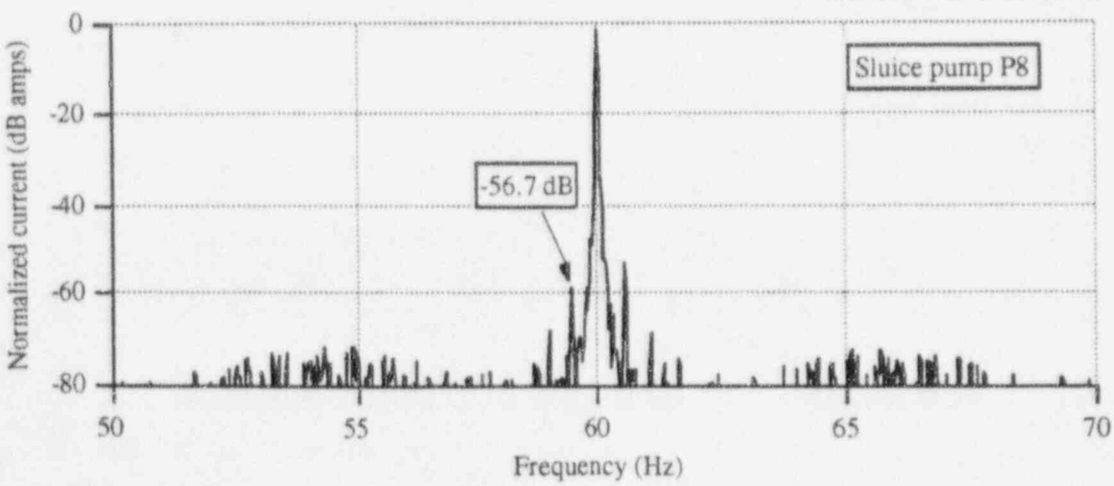


Figure 5.10 Fly ash sluice pump P8 motor current spectrum

degradation from the unconditioned motor current spectrum. Figures 5.11 and 5.12 show the current spectrum in the vicinity of 60 Hz for two motor conditions: without artificial degradation in one case and with two broken rotor bars in the other. The lower slip-pole sideband amplitudes are annotated.

The fan motor had 22 rotor bars; thus, almost 10% of the rotor conductors were broken, yet there was actually a decrease in the slip-pole sideband amplitude (although note that the running speed was lower, as evidenced by higher slip).

5.3.2 Other Means of Understanding Rotor Condition from the Current Spectra

In the degradation testing discussed in Sect. 5.3.1, the utility considered not only the amplitude of the fundamental slip-pole sideband but also sidebands of harmonics of the line frequency. A method that has been developed at ORNL focuses on modulations of higher frequency components in the motor current spectrum. As noted previously, hydraulically or mechanically related noise (particularly for pumps) in the current spectrum surrounding the line frequency can significantly affect the ability to observe trends in the slip-pole sideband amplitude. The combination of the circuit associated with the motor stator to rotor coupling and the rotating inertia of the rotor and the pump tend to minimize the effect of high-frequency hydraulic and mechanical load-related fluctuations on the motor current/power. Electrical/magnetic field coupling effects between the stator and rotor are not so dramatically filtered, and the high-frequency portion of the current tends to be more influenced by these components. As a result, the high-frequency components tend to be more sensitive to motor conditions, while the lower frequency components (i.e., the spectra around 60 Hz) tend to be more sensitive to mechanical loads. In addition, the higher frequency portion of the current signal can be processed so as to provide a more precise indication of motor speed.

ORNL-DWG 95-2273 ETD

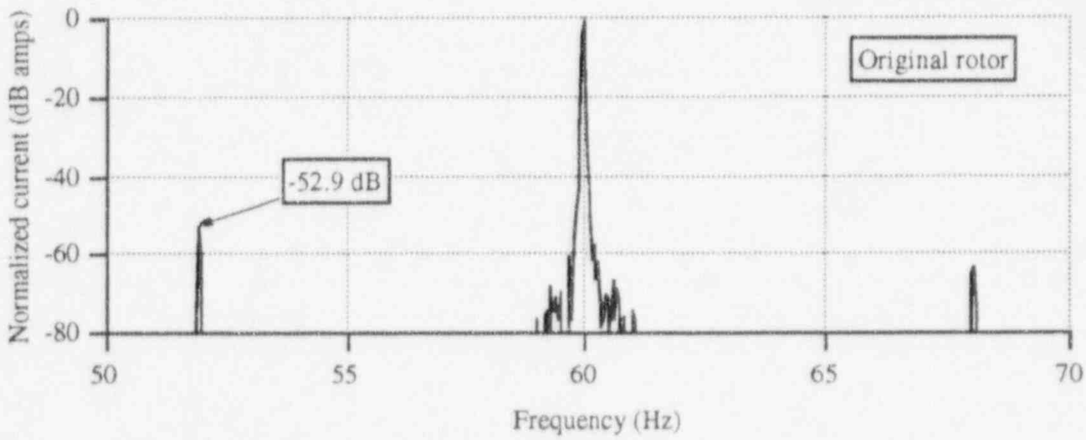


Figure 5.11 Small fan motor current spectrum with no rotor degradation

ORNL-DWG 95-2274 ETD

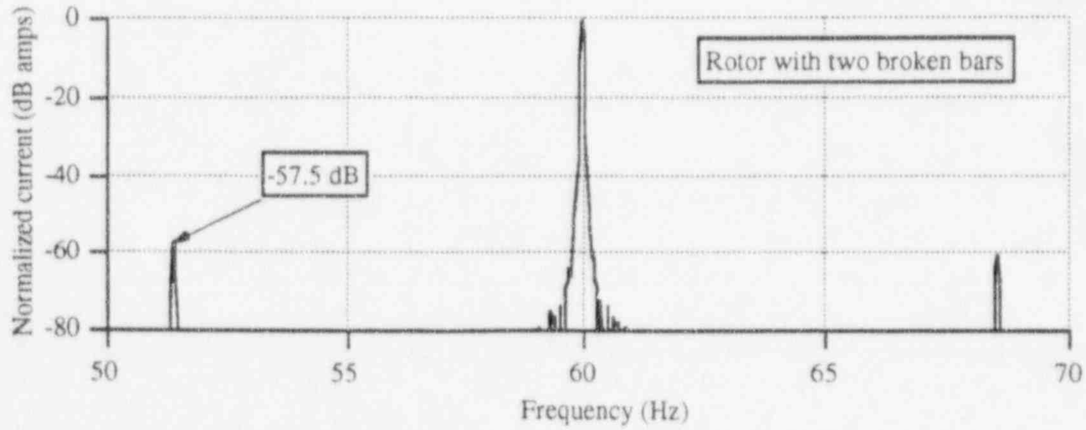


Figure 5.12 Small fan motor current spectrum with artificial rotor degradation

Figures 5.13–5.15 show the portion of the motor current spectrum in the vicinity of 60 Hz below the stator slot pass frequency for the 38-hp motor discussed earlier. The amplitude of the principal peak, as well as slip-pole sidebands, grows with increasing rotor degradation. The stator slot-related frequency is one of several higher frequency components that has been observed to be responsive of rotor degradation (others include the rotor bar pass frequency and complex sidebands of stator slot and rotor bar pass frequencies). The data shown here were high pass filtered and amplified before digitization, thereby reducing the noise floor level.

The change in amplitudes and extent of slip-pole side banding are clear as the rotor degradation proceeded to one and two broken bars. The change in pattern from two to three broken bars is not clear.

The portion of the spectrum between 1690 and 1720 Hz was digitally amplitude demodulated for the conditions shown in Figs. 5.13–5.16. Figure 5.17 shows the time domain data for this portion of the spectrum. The change in pattern with progressive degradation is actually somewhat clearer here than in the spectral domain. For all four conditions, the principal frequency component occurs at slip-poles. The data suggest that the original rotor had some level of electrical eccentricity (perhaps a combination of static and dynamic eccentricity and variations in the electrical properties of the rotor conductors). As the rotor was damaged, the amplitude of both the ac and dc components of the modulations changed, as well as the general characteristic shape.

While the higher frequency region of the motor current spectrum appears to show a higher level of sensitivity to

ORNL-DWG 95-2275 ETD

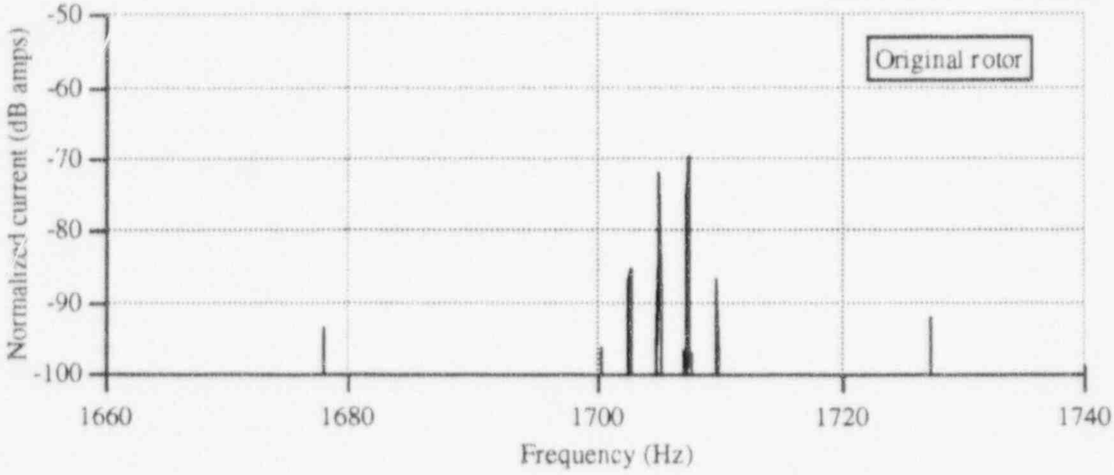


Figure 5.13 Stator slot pass frequency lower sideband—original rotor

ORNL-DWG 95-2276 ETD

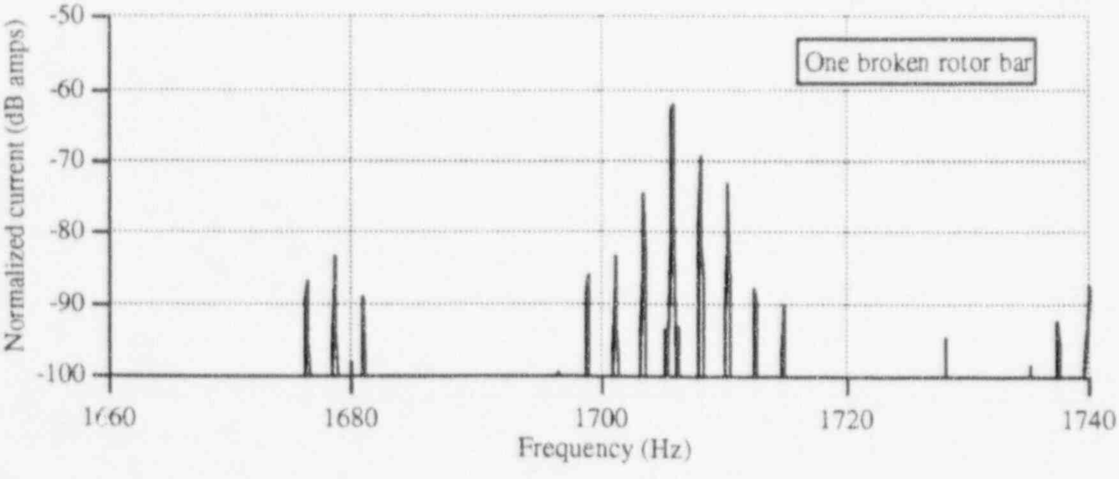


Figure 5.14 Stator slot pass frequency lower sideband—one broken rotor bar

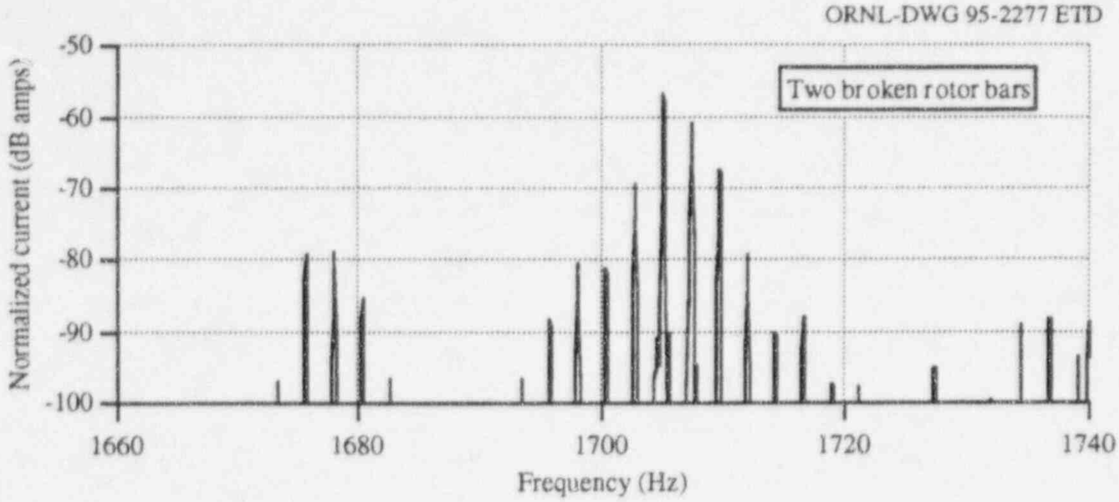


Figure 5.15 Stator slot pass frequency lower sideband—two broken rotor bars

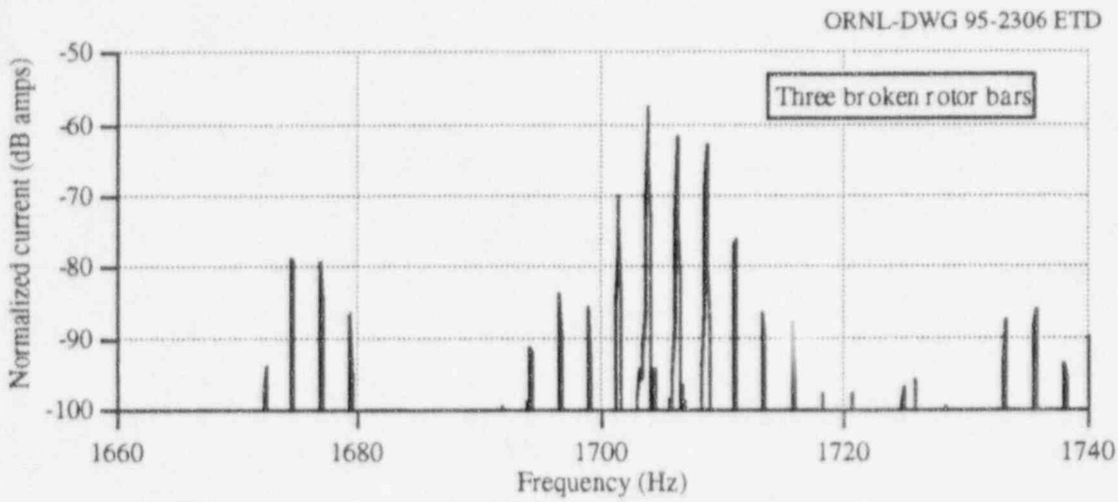


Figure 5.16 Stator slot pass frequency lower sideband—three broken rotor bars

rotor degradation than does the overall, unconditioned spectrum, the amount of historically available data of this type is extremely limited, and like the 60-Hz sideband amplitude method, such results are at present more useful for trending than application in an absolute sense.

5.3.3 Vibration Analysis for Rotor Condition Monitoring

Vibration data were also acquired during the motor degradation test. Figure 5.18 shows the vibration velocity spectra for the horizontal, motor outboard bearing location for four rotor conditions. There are some interesting patterns that emerge. First, the amplitude of all harmonics of running speed (other than the fundamental) decreased when the first rotor bar was broken. For the two broken bar case, there were not only significant increases in amplitude but also the development of considerable slip-pole sideband modulation of running speed and running speed harmonics. However, the third broken bar (diametrically opposite the first two bars) actually returned the overall vibration pattern to one very similar to that observed with one broken bar. Thus, neither the overall vibration amplitude nor sideband modulation patterns provided a clear indication of the level of rotor damage.

5.4 Summary

On-line and off-line diagnostics are available to allow effective trending of motors for degradation. These diagnostics can also be applied in an absolute sense (i.e., without prior historical data), but it is important to recognize that there are much greater uncertainties associated with untrended data.

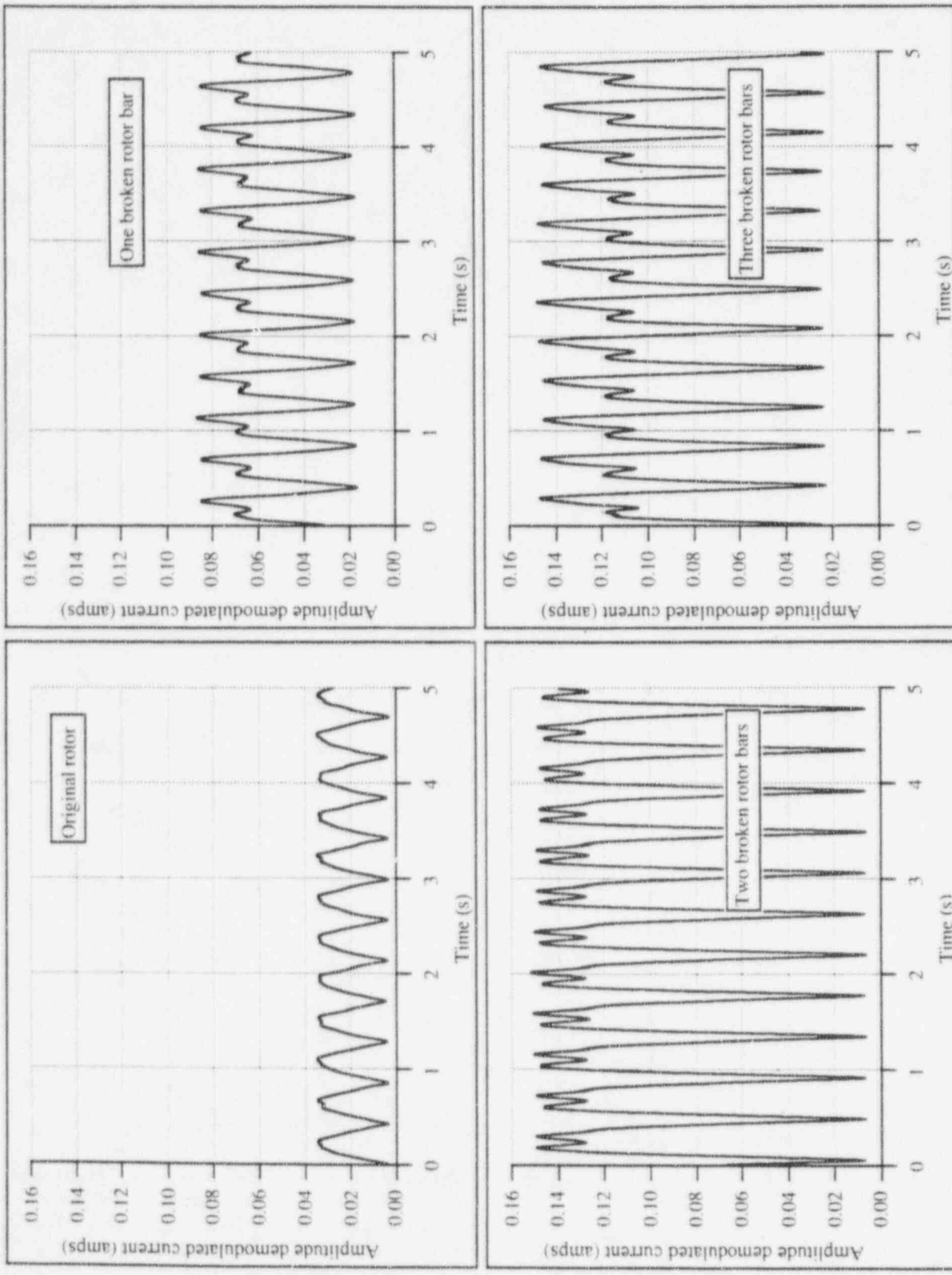


Figure 5.17 Amplitude demodulated stator slot pass frequency-related current signal in time domain

ORNL-DWG 95-2279 ETD

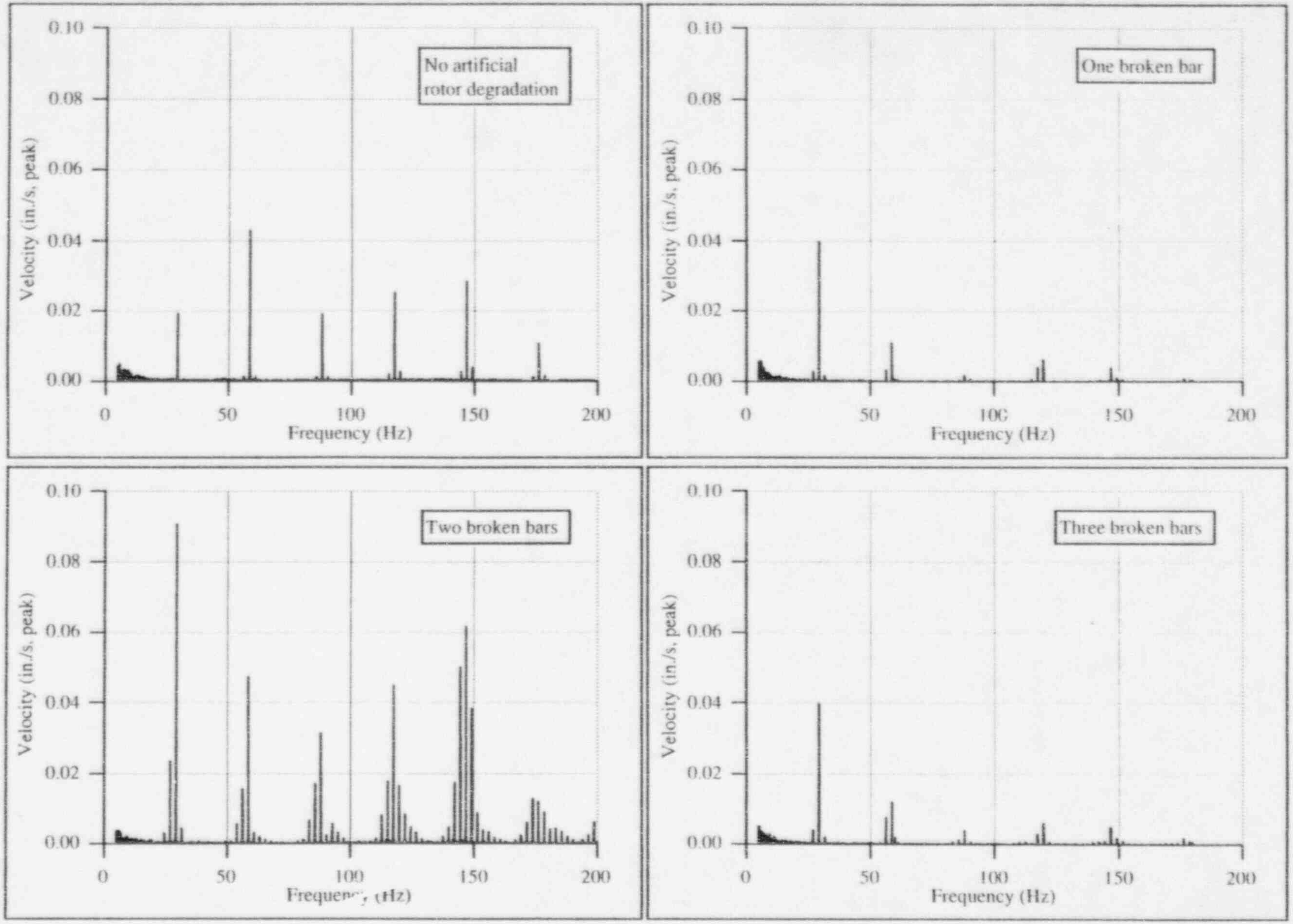


Figure 5.18 Vibration spectra for four rotor conditions

Some off-line test methods, such as meggering and hi-pot testing, have been used for many years, and sufficient data have been acquired to permit the development of industry standards specifying acceptance criteria.[43,48] Other off-line tests, such as inductance measurement have become more popular in recent years; for such techniques, industry consensus standards have not yet been developed. Available techniques have been found to have merit, but experience has shown that none are foolproof. There are inherent risks in the use of some of these methods.

On-line test methods, specifically spectral current monitoring, have also been growing in usage in recent years; there are likewise no consensus standards available to guide in the establishment of acceptance criteria. Controlled test data have shown that, generally speaking, the amplitude of the slip-pole sidebands of 60 Hz (and harmonics) do trend upward with increased rotor degradation (although very small motors may be less sensitive, as noted in Sect. 5.3.1). The unconditioned current spectral data presented in this report indicate that there are several difficulties in the use of the slip-pole sideband method with generic, absolute criteria:

1. While a significant amplitude slip-pole sideband of the line frequency and harmonics does appear to normally indicate an electromagnetically associated modulation, it appears that an absolute determination of the extent of rotor degradation cannot as of yet be made. Furthermore, a motor may successfully operate for years with the existence of relatively high sideband amplitudes (i.e., some source of magnetic asymmetry), depending upon the environment, motor load, type of motor, and other factors.
2. In motors used to drive pumps with significant hydraulic noise (or other equipment that experiences relatively high amplitude, nondiscrete, low-frequency torque fluctuations), the sideband noise level can reduce the ability to clearly distinguish the slip-pole sideband, particularly for trending purposes.
3. At least for some motors, significant rotor degradation can exist without significant changes in the current spectra.
4. The methodology has not been widely employed, and as a result, a significant data base has not been accumulated and published.

Other portions of the current spectrum, such as those associated with rotor bar and stator slot pass frequencies, show the potential for clearer indication of rotor condition. There is even less available historical data for these spectral components, however.

Given the current absence of systematically and independently assessed results on a broad range of motors, it would appear prudent to rely upon motor current data primarily for trending until a technical basis is developed to provide absolute criteria.

Spectral vibration data analysis may or may not help identify motor electrical problems; it can certainly be used to detect bearing-related problems, however, and should be considered one of the cornerstones of a predictive maintenance program for motors.

A program that employs a combination of off-line and on-line techniques will provide both complementary and confirmatory indications of motor health, as well as providing a higher level of assurance that faults are detected at the incipient stage.

6 Detection of Pump Stressors

6.1 Classification of Pump Stressors

In a study performed for the NRC on the aging and service wear of auxiliary feedwater (AFW) pumps[49] (hereinafter referred to as the "pump study"), pump stressors were classified as belonging to the following categories:

- Mechanical
- Hydraulic
- Tribological
- Chemical
- Low-relevance factors

The study was based largely on experience with similar pumps used in boiler feed service in fossil applications. It should also be noted that at the time the study was conducted, there were minimal available failure data.

The scope of the pump study did not include the pump driver. A subsequent study of the AFW system[50] (hereinafter referred to as the "system study") indicated that pump driver-related failures were a principal source of system degradation. By including the pump driver, an alternative categorization is as follows:

- Mechanical
- Hydraulic
- Tribological
- Chemical
- Other (including those associated with the pump driver)

The pump study indicated that the principal relevant chemical stressor for AFW pumps, oxidation of 400 series stainless steels through chemical reaction with stagnant water, was in effect a tribological factor since its relevance was primarily related to abrasive oxides. As a result, the pump study dealt with the effects of the first three stressor categories (mechanical, hydraulic, and tribological) on the various parts of the pump.

Some specific stressors within these three categories are identified in Table 6.1. The specific pump components that are affected by such stressors were identified in the pump study to include the items listed in Table 6.2.*

*Other components identified were the pressure-containment casing and the pump support. Because the stressors identified were not judged to be high-importance factors for these components, they are not included in Table 6.2.

Table 6.1 Principal stressors of influence for AFW pumps

Mechanical	Hydraulic	Tribological
Torque transmitted loads	Hydraulic loads	Rubbing between rotating and nonrotating members
Rotor-dynamic loads	Fluid impingement	Bearing lubricant breakdown
Piping forces	Internal pressure	Surface fatigue
Seismic loads	Cavitation	Lubricant contamination and degradation
Vibration (all sources)		Starts and stops
		Fretting
		Surface oxide formation

Table 6.2 Primary components influenced by identified stressors

Rotating elements	Nonrotating internals	Mechanical subsystems
Shaft	Diffusers/volutes	Thrust bearings
Impellers	Return channels	Radial bearings
Thrust runners	Wear surfaces	Shaft seals
Fasteners	Fasteners	Thrust balancer
Miscellaneous spacers		Coupling
		Fasteners

Hydraulic and tribological factors, primarily associated with dynamic loads from hydraulic forces and cavitation, were identified as the principal stressors of significance for both rotating elements and nonrotating internals components. Tribologically related factors, which are more diverse in nature, were identified as being the primary influence for the mechanical subsystem components. The tribological factors associated with the mechanical subsystem components identified in the pump study ranged from axial and radial hydraulic loads (both static and dynamic) to breakdown or loss of lubricant in gear couplings. Mechanical factors, such as overtightening of packing, and rotor dynamic loads were also indicated as principal stressors of influence.

The pump study also identified the operation of AFW pumps at low-flow rates as a particularly important factor in accelerated aging. This factor, as well as the inability to gain useful hydraulic data on the operating condition of the pump when tested at low flow, has led to changes in ASME Code requirements.[51,52]

The system study, which by its nature investigated historical system failure data (as opposed to focusing on individual component stressors), found that problems with turbine drives were the largest contributor to overall system degradation. Over 80% of the turbine drive problems were associated with either the turbine's control subsystem, the trip and throttle valve, or the governor valve. Likewise, over 80% of pump motor drive problems were related to either the instrumentation and control subsystem or the motor breaker. Thus, a significant portion of the drive-related failures were attributable to control-related aspects of the drivers. While the AFW system may not be representative of most nuclear plant systems in regard to driver problems (few pumps use turbine drives) and failures of the control components are beyond the scope of this study, the contribution of instrumentation and control problems to overall pump reliability should obviously not be ignored.

A study of the dynamic loading of pumps[53] that are similar to AFW pumps indicated that for pumps that are balanced according to American Petroleum Institute Standard API 610,[32] the hydraulic unbalance force is "clearly the biggest dynamic load acting on a pump rotor during normal operating conditions." The study did note that the tight mechanical unbalance limits obtained originally (in accordance with API 610) may be difficult to maintain. The study also noted that at low-flow rates, "broad band vibrations can become predominant, and can be higher than any other vibrations."

While these stressors were identified in studies on AFW pumps and systems specifically (or on pumps similar to AFW pumps in the case of Ref. 53), they are generally applicable to the broader spectrum of pumps. There are other factors involved in certain applications, and the principal stressors of influence may vary with the specific design and use, but the general observations remain valid.

6.2 Historically Applied Monitoring Practices

Probably the most commonly applied method for monitoring rotating equipment condition in general, and pumps in particular, has been vibration analysis. The measurement of vibration has been employed for many years. It has evolved from strictly analog devices such as vibrating reeds to digitally based expert systems. Current expert systems compare the vibration spectra to pre-established limits and previously acquired data to (1) indicate whether the machine is operating within the desired range, (2) indicate potential sources of problems if not, and (3) project remaining life before repair is necessary. Other commonly monitored pump parameters include head and flow, bearing temperatures, and lube oil condition.

The ASME Code[31,*] requires that pump head, flow, and vibration be monitored periodically. The frequency and exact requirements depend upon the type of pump and the specific test being conducted. The vibration monitoring requirement is for a broadband, unfiltered (i.e., non-spectral) amplitude, and the required frequency response range is from one-third of pump minimum shaft rotating speed to at least 1000 Hz.

Another parameter that is seldom used for routine field monitoring, but is often used in troubleshooting and in special test programs (particularly when addressing low-flow conditions or other problem pump conditions such as cavitation), is pressure pulsation.[54–56] Pump discharge and suction, as well as various points in the pump casing have been monitored for pressure pulsation analysis.

6.3 The Detection of Hydraulically Induced Vibration

The original AFW pump study indicated that effects of low-flow-related forces could be a principal contributor to pump degradation. Subsequent failure of multiple AFW

*It should be noted that the NRC has not yet, as of this writing, endorsed the 1990 version of the ASME Code.

pumps[57] as well as independent studies on similar pumps[53] support the validity of this observation. Utility personnel involved in reviewing the Ref. 57 pump failures indicated that neither the ASME Code monitored vibration nor spectral vibration monitoring indicated that a problem existed.

Many vibration guides are available that typically address mechanically induced vibration components, such as unbalance, misalignment, defective bearings, worn gears, etc. Unfortunately, it is often the case that such guides or tables fail to address *pump*-related components. This could well account for the failure to observe indications of degradation or sources of degradation.

The literature suggests that the effects of vibration resulting from undesired pump operational conditions may be manifested in various frequency regimes. Table 6.3 lists several frequency regions that have been found with field experience. Note that Table 6.3 is not intended to be a comprehensive listing of all vibration frequencies that may be found; specifically, there are no mechanically related frequencies.

6.4 A Comparison of Monitoring and Analysis Methods

The frequency regimes identified in Table 6.3 are primarily based on vibration monitoring experience. Vibration monitoring is certainly the most commonly applied field diagnostic, so it is a natural result that hydraulic, as well as mechanical, stressors and degradations would be most often identified in that context. However, it is important to recognize that bearing or pump casing vibration will not necessarily represent all sources of unstable conditions. For example, significant torsional loading variations can exist without being manifested at normally monitored vibration locations.

Test data were acquired on three pumps to compare the results of vibration, motor input power, and pressure pulsations (the latter for only one pump) with each other as well as the spectral components identified in Table 6.3. It is important to note that the pumps tested are relatively low-energy pumps. This is an important factor, since low-energy pumps are less significantly affected by hydraulic forces than are high-energy pumps.[53] Generally speaking, the pumps used in nuclear applications are significantly larger, higher energy density pumps, some of which are multistage; thus the results from the tests conducted are not intended to be representative of these pumps. However, the data presented are useful in showing general trends and relationships.

Nominal pump parameters for the tested pumps are provided in Table 6.4. All three pumps are single-stage, double-volute type; Pumps A and B are single suction (overhung), while Pump C is double suction (between bearings). From these general design characteristic, certain tendencies could be noted. Generally speaking, higher suction specific speed pumps have been observed to be more likely to experience low-flow-related instabilities; therefore, Pump A would be expected to be more sensitive to off-design operation. Likewise, Pump C might be expected to experience a higher level of hydraulic unbalance, due to the increased difficulty in casting perfectly symmetrical impellers for double-suction pumps. It is

Table 6.3 Some hydraulically induced vibration components

Item	Frequency	Source of vibration	References
A	0–10 Hz (broadband)	Suction recirculation (Ref. 13 relates this to Gap "A," which is the gap between impeller and diffuser/volute sidewalls)	58,59,*a*
B	<15 Hz (broadband)	Unsteady flows due to recirculation or turbulence (similar to item A)	53
C	0 to 1.5 × running speed, broadband	Recirculation	53
D	0.5 to 0.95 × running speed	Rotating stall	53
E	0.6 to 0.9 × running speed	Hydraulic instability	58,59
F	1 × running speed	Hydraulic unbalance	53,58,59,*a*
G	0.5–10 kHz (broadband)	Cavitation	53
H	Vane pass frequency	Impeller vane to diffuser vane gap (Gap "B")	53,58,59,*a*
I	Relatively distinct peak, 1–15 Hz	Surge or system instability	53

[a]E. Makay, "Trouble-Shooting High Energy Input Power Plant Boiler Feed Pumps," 1989 EPRI/MVI Centrifugal Pump Short Course Lecture Notes.

Table 6.4 Nominal pump parameters for test pumps

Parameter	Pump A	Pump B	Pump C
General style	Horizontal, overhung (single suction)	Horizontal, overhung (single suction)	Horizontal, between bearings (double suction)
BEP flow rate (gpm)	2,000	200	1,000
Nominal speed (rpm)	1,780	3,500	1,765
Best efficiency head (ft)	137	100	170
Specific speed (unitless)	1,990	1,570	1,185
Suction specific speed (unitless)	12,750	10,700	5,760
Motor power rating (hp)	75	7.5	50
Number of impeller vanes	6	5	8

important to note that these general propensities can be significantly offset (or exacerbated) by other factors, such as piping layout, individual pump dimensions, etc. It is not feasible to develop simple rules of thumb for complex machinery like pumps that apply to every pump. Therefore, these general tendencies of certain pump classes can be used only as a first, crude indicator of expected performance.

6.4.1 Pump A Vibration Data Analysis

Vibration data were acquired at various flow rates for Pump A, ranging from shutoff (0-gpm) conditions to greater than best efficiency point (BEP) flow. The vibration data to be discussed were acquired with accelerometers mounted on (1) the pump casing, in an axial direction, and (2) the housing of the radial bearing closest to the pump, in the horizontal, radial direction.

Figure 6.1 provides pump casing axial and bearing housing radial vibration velocity spectral data for Pump A at four flow rates ranging from shutoff to 105% of BEP. For the axial vibration data, the running speed peak generally increases with increasing flow, while the vane pass frequency amplitude decreases with flow, suggesting increased hydraulic imbalance with increasing flow (however, see discussion under Sect. 6.4.4 relative to hydraulic imbalance). These patterns are not duplicated in the radial horizontal data. For both monitored locations, the spectral region below 100 Hz is somewhat noisier at the low-flow conditions. Note that with the exception of a small peak occurring at around 850 Hz in the radial signal, there is very little spectral energy in the velocity domain above 400 Hz. The spectral displays above 1000 Hz are not shown since the velocity domain spectra above 1000 Hz are insignificant compared to the lower frequency components.

Figure 6.2 provides the same vibration source data as Fig. 6.1, except that the vibration is presented in the acceleration domain and displayed up to 5000 Hz, since there is obviously considerable energy present. There are multiple sources of the higher frequency energy including cavitation and bearing fault frequencies. For the axial location (left column), the significantly higher amplitude of broadband noise in the 3000- to 5000-Hz range is attributable to cavitation. The radial location peak at about 850 Hz that is barely visible in the velocity domain is one of the more dominant peaks in the acceleration domain. A review of the bearing fault frequencies indicated that all four fault-related frequencies (bearing inner race, outer race, cage, and ball frequencies) for the pump thrust bearing have harmonics that converge at about 850 Hz.

Figure 6.3 is the same velocity data as shown in Fig. 6.1, but zoomed to reveal more details of the spectra from 0 to 200 Hz. For Fig. 6.3, the graph tick mark labels are at harmonics of running speed. The nonharmonic peak that occurs at about 140 Hz in the radial data (right column) appears to be related to the inner race frequency of the thrust bearing. Figure 6.3 more clearly shows the higher level of broadband noise that exists at reduced flow conditions from about half of running speed to around four times the running speed. Note that the principal areas of higher noise are comparatively different at the two locations. For the axial location (left column), most of the noise appears as sidebands of the running speed, while at the radially monitored location, the noise bandwidth is broader, extending up to four times running speed.

By comparing the data observed in Figs. 6.1–6.3 with the components reported in Table 6.3, it can be concluded that the pump depicts some evidence of cavitation (broadband noise over a fairly wide frequency band) that is stronger at low-flow conditions, as well as indication of recirculation

ORNL-DWG 95-2280 ETD

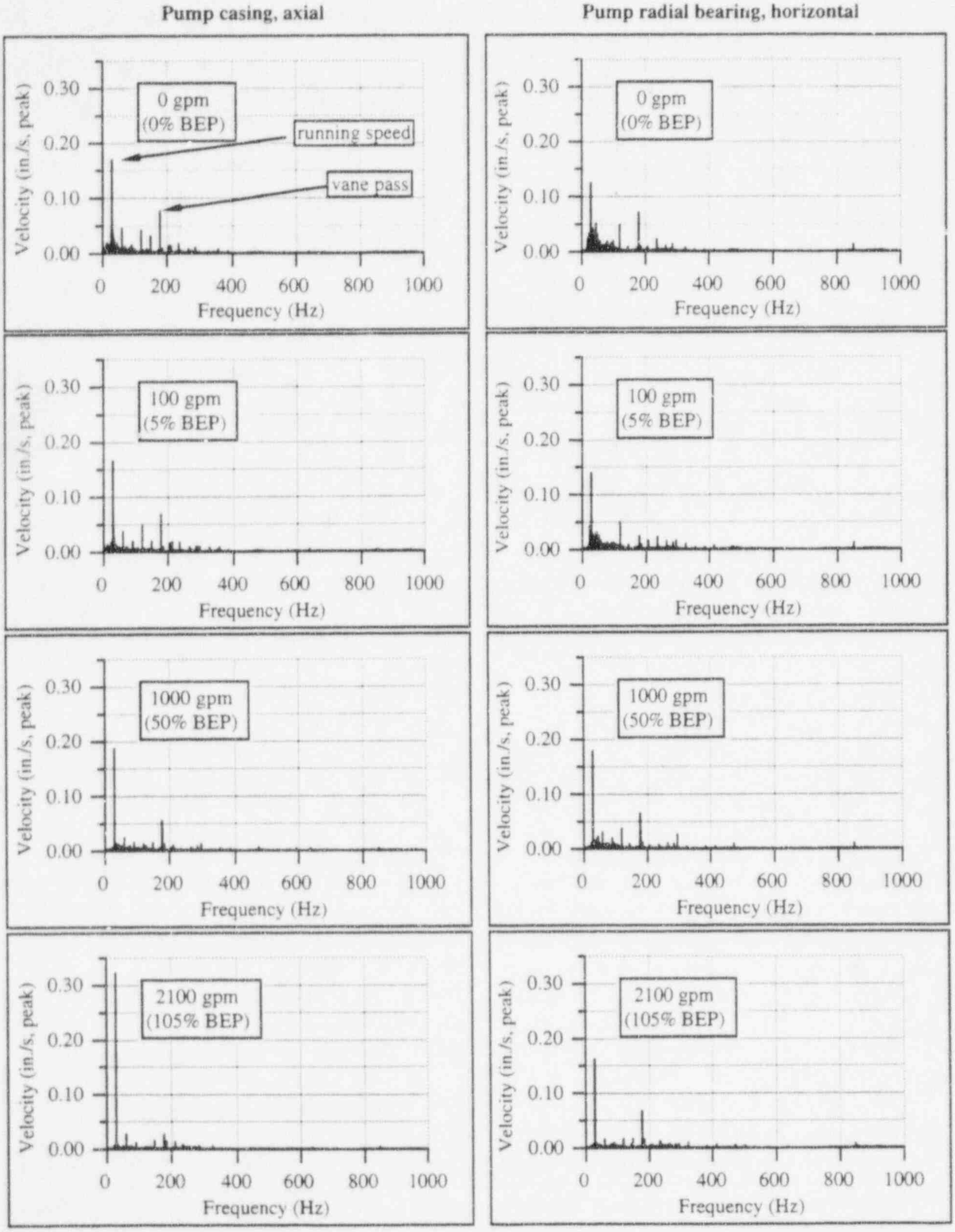


Figure 6.1 Pump A vibration spectra in velocity domain

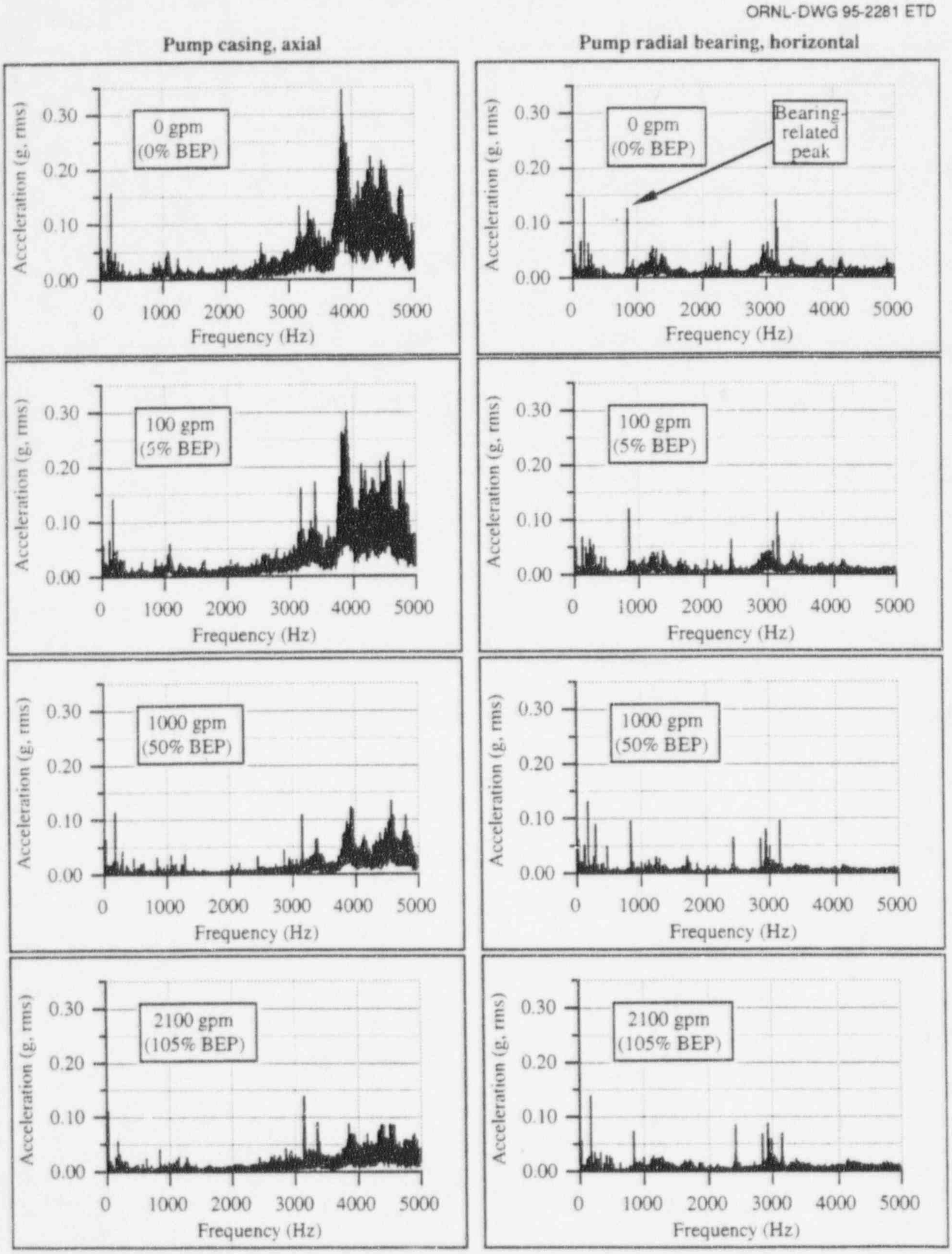


Figure 6.2 Pump A vibration spectra in acceleration domain

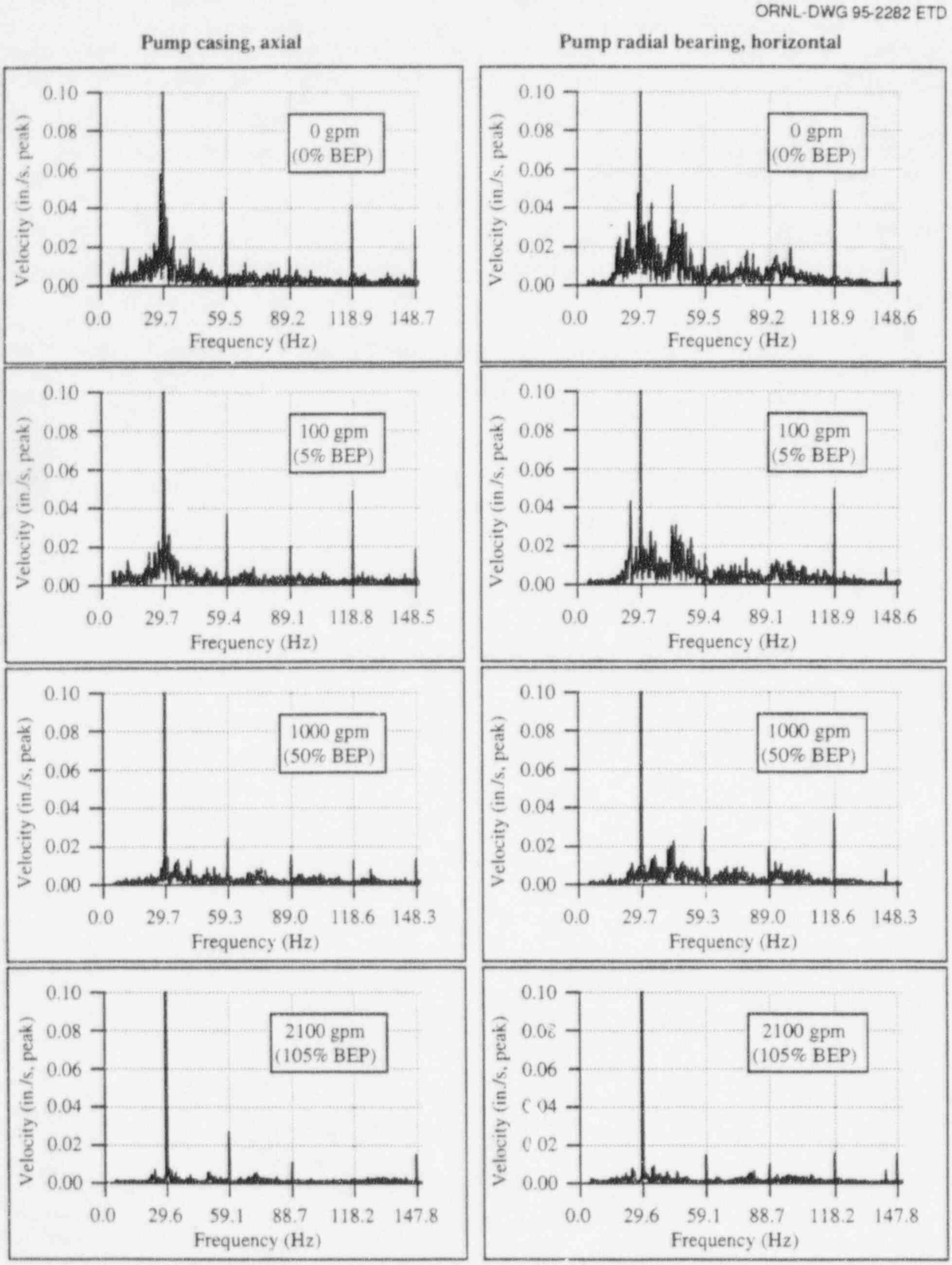


Figure 6.3 Pump A vibration spectra in velocity domain (zoomed)

at low flow (although the range of the broadband noise extends beyond the 1.5 times rotating speed suggested in Table 6.3). Independently, there was audible cavitation noise when the pump was being operated at low-flow conditions.

Since the ASME Code does not require spectral monitoring of vibration, the vibration waveforms were also converted to the velocity domain and analyzed. The analysis of the data went well beyond ASME requirements by not only evaluating the overall amplitude, but also determining fluctuations in the overall amplitude during the ~14 s over which data were acquired.

Figure 6.4 provides the results of the analysis. Note that the two upper plots present the nonspectral or overall amplitude as rms amplitude × 1.414. The reason for providing rms × 1.414 rather than true peak is that historical data that have been labeled "peak" have, in most cases, been actually rms × 1.414. The average pump axial amplitude slightly increases at the upper end of the flow rates monitored, while the radial vibration slightly decreases at the higher flow rates. Both locations are somewhat higher at the shutoff condition than at the intermediate flow conditions. There is little useful information that can otherwise be distilled from the average values. The presentation of the data as rms × 1.414 rather than true peak is an important distinction, since the true peak is often several times rms × 1.414. The lower left plot of Fig. 6.4 shows the ratio of true peak to rms × 1.414 for both monitored locations as a function of flow rate.

The maximum and minimum values shown in the two upper plots of Fig. 6.4 are based on the calculation of rms values for 0.5-s snapshots of the vibration data over a total sample time of 14 s. When the fluctuations in measured amplitude are considered, as opposed to simply the average rms × 1.414, a somewhat more revealing pattern emerges than the simple average rms. The lower right-hand plot of Fig. 6.4 shows the ratio of maximum to minimum values of the 0.5-s sets. At the lowest flow rates, there were more fluctuations in short-duration measurements, which indicate less stable overall conditions. This is expected and is qualitatively consistent with the generally noisier spectra observed at the low-flow conditions.

The effect of changes in the bearing-related flaws component noted above was also explored. Using the 0-gpm radial vibration data as a source, the signal was modified in two ways. First, the accelerometer signal was digitally filtered to remove the principal frequencies that appear to be bearing-related, as well as the broadband noise above 850 Hz. Second, the bearing flaw-related energy around 850 Hz was digitally amplified by a factor of 10. Figure 6.5 shows the unmodified velocity domain spectrum, the spectrum with the bearing fault frequencies removed, and the spectrum with the 850-Hz component increased. While the differences are clearly evident in the spectral domain, there is minimal effect on the overall, nonspectral value. For the case where the 850-Hz bearing flaw component is increased by a factor of 10, the overall amplitude increased by less than 20%.

A real degradation in bearing condition would certainly manifest itself differently than the simple digital model used above. For instance, harmonics and/or running speed sidebands of defect frequencies might be expected as the bearing degraded. However, the point to be made is that relatively large increases in high-frequency components have minimal effect on the nonspectral overall vibration velocity amplitude.

6.4.2 Pump B Vibration Data Analysis

Vibration data were acquired at various flow rates for Pump B, ranging from shutoff (0-gpm) conditions to approximately twice BEP flow. The vibration data were acquired with accelerometers mounted on (1) the pump casing, in an axial direction, and (2) the housing of the inboard radial bearing.

Figures 6.6–6.8 provide velocity and acceleration domain vibration spectra for both monitored locations. Some observations on the data in these figures follow.

- The running speed axial velocity amplitude for Pump B (Fig. 6.6) is relatively constant with flow rate, unlike Pump A, where the axial running speed amplitude at the highest flow rate is approximately twice that at shutoff (Fig. 6.1).
- There is a significant level of broadband noise at both locations in the acceleration domain (Fig. 6.7) signals at the highest flow rate (200% BEP), indicating cavitation. Although the pump and system generate a generally higher level of noise at this flow rate, cavitation is not audibly detectable.
- There are a series of relatively well defined peaks in the range of 1600 to 2200 Hz and also in the range of 3100 to 3400 Hz in the acceleration domain horizontal radial signal. More detailed analysis of the spectra indicated that these peaks originate from a combination of the rotor bar pass frequency (and second harmonic), characteristic bearing frequencies (including a harmonic convergence similar to that noted for Pump A above), and running speed and 120-Hz sidebands of these components. It is important to note that the motor used to drive this pump is a high-efficiency motor. High-efficiency motors have been observed to be somewhat more likely

ORNL-DWG 95-2283 ETD

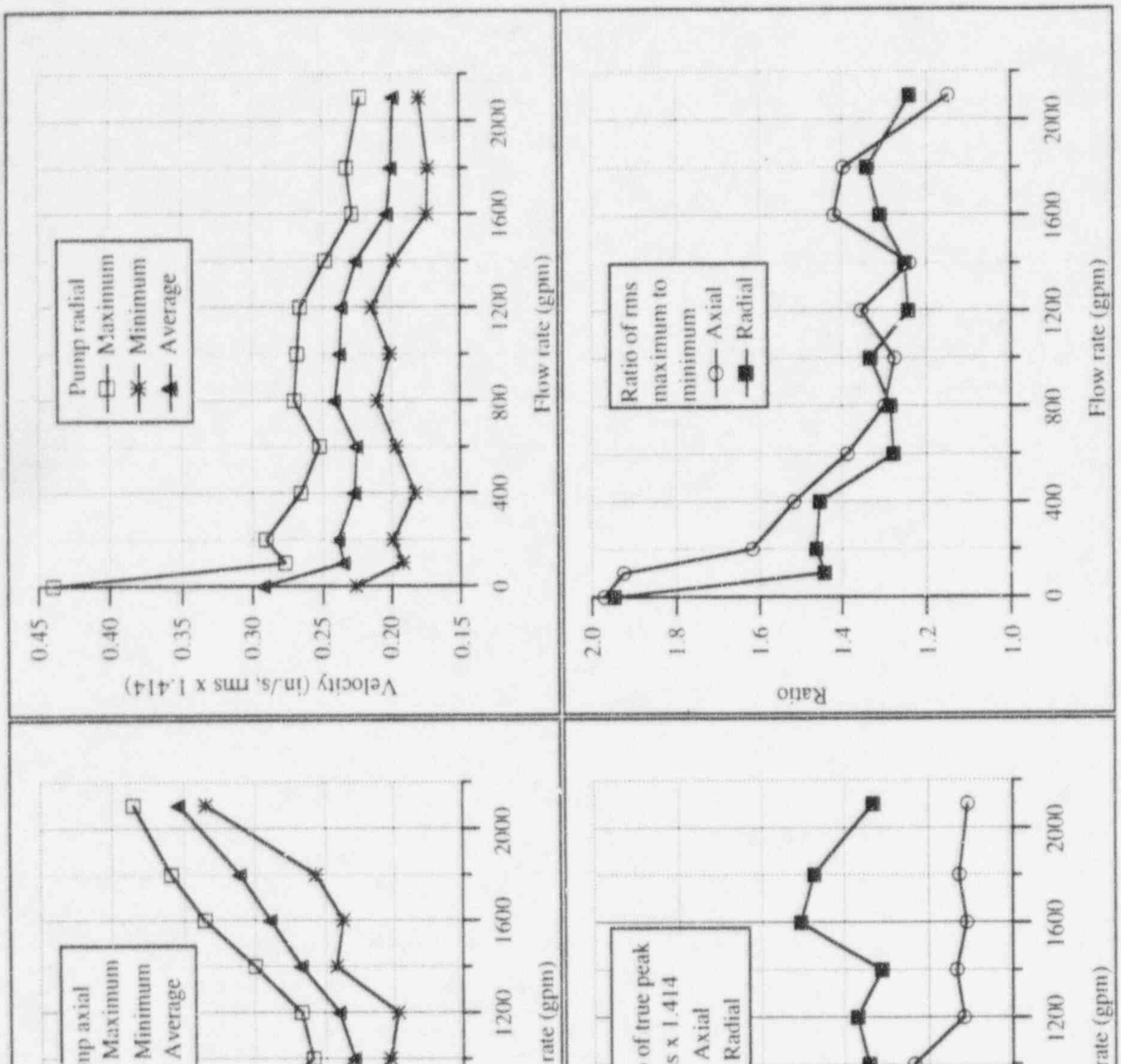


Figure 6.4 Summary rms vibration data for Pump A

ORNL-DWG 95-2284 ETD

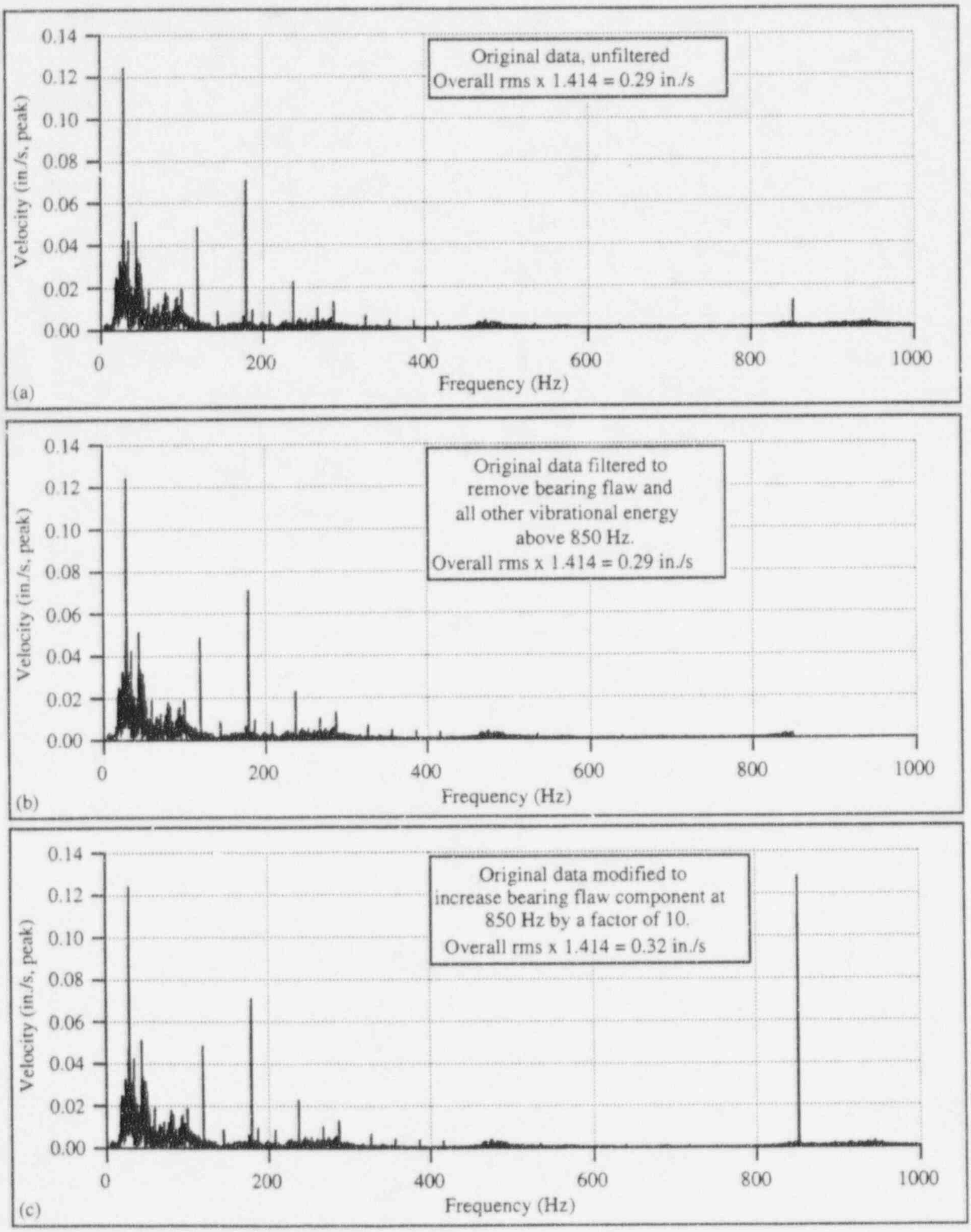


Figure 6.5 Pump A horizontal radial velocity spectra at 0 gpm

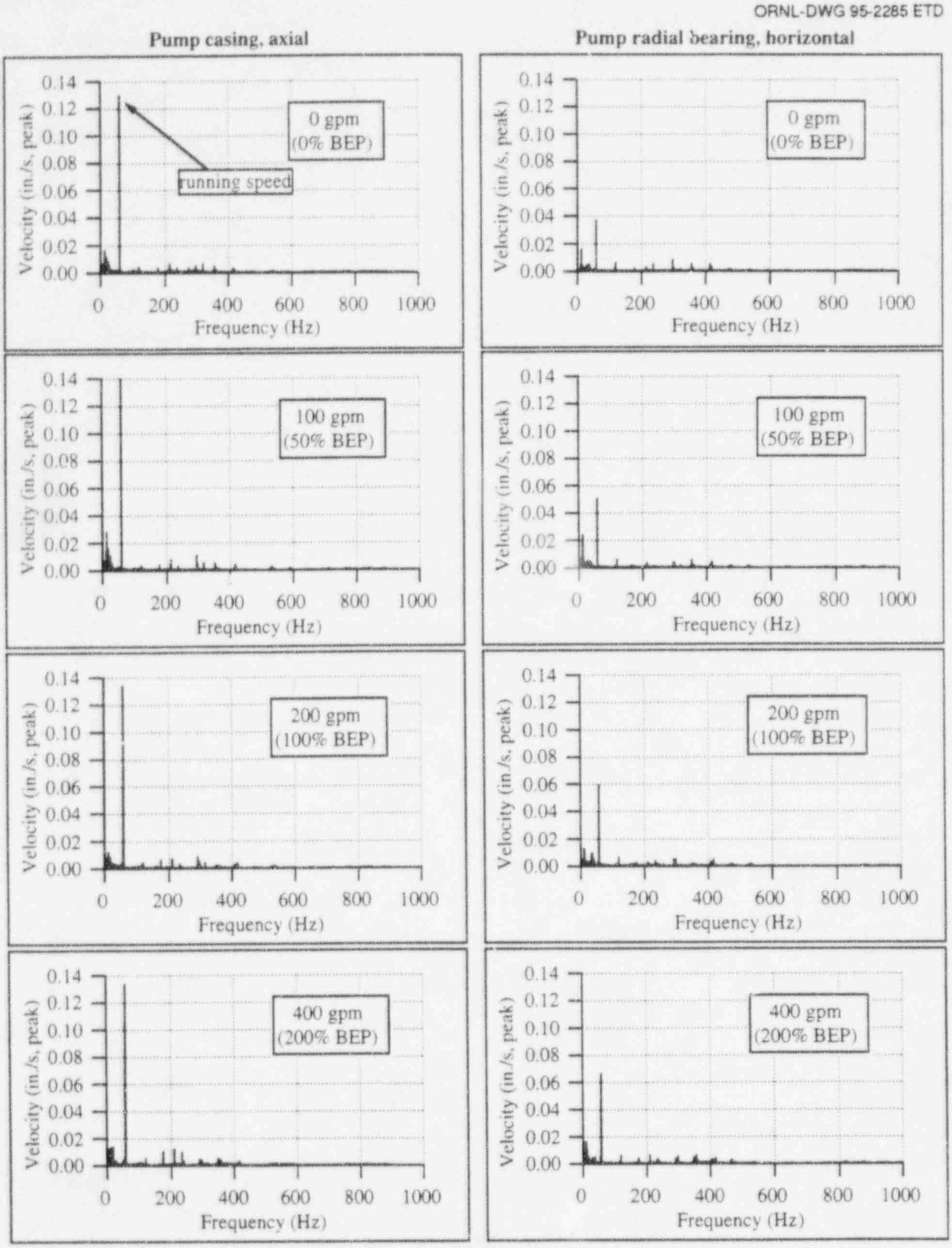


Figure 6.6 Pump B vibration spectra in velocity domain

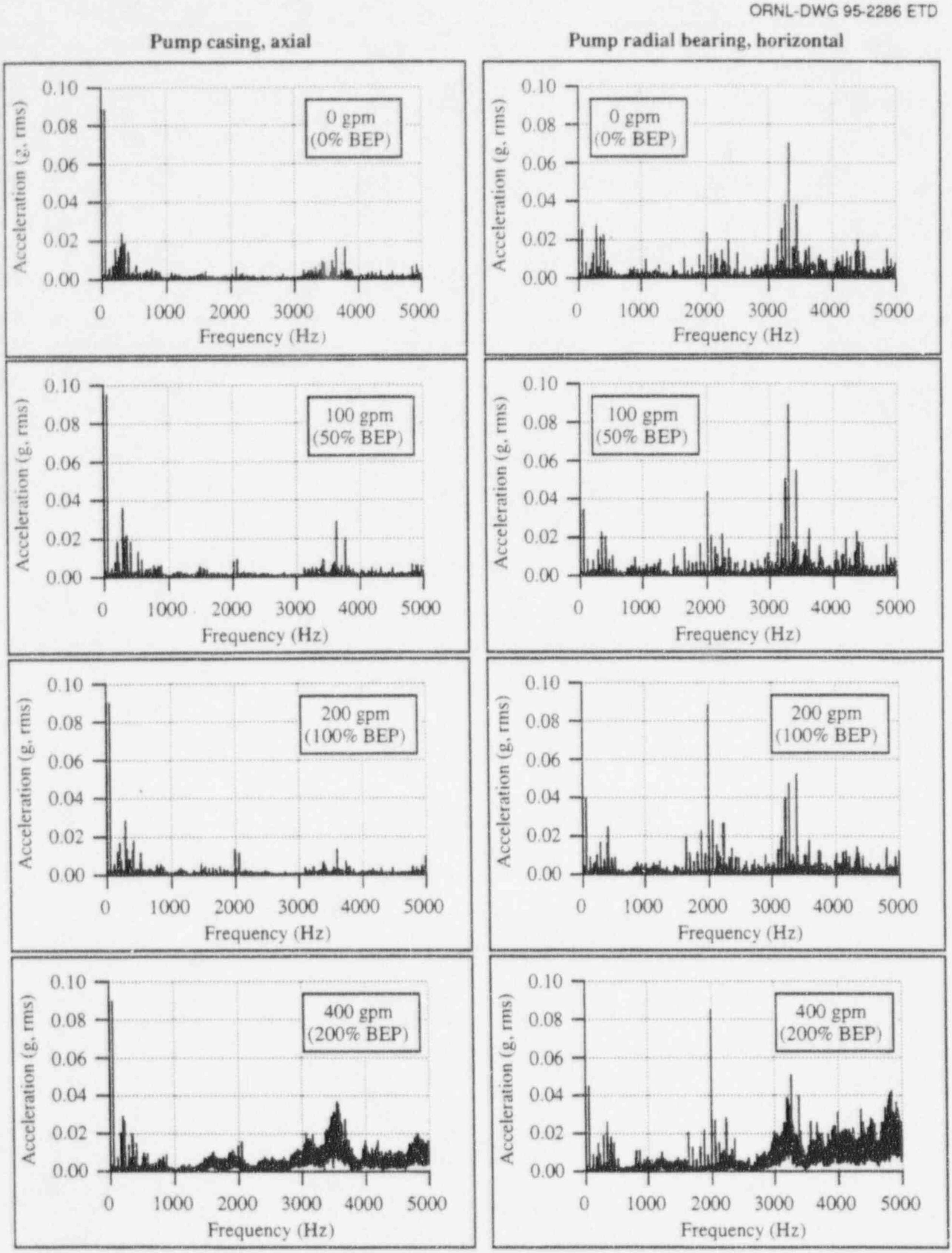


Figure 6.7 Pump B vibration spectra in acceleration domain

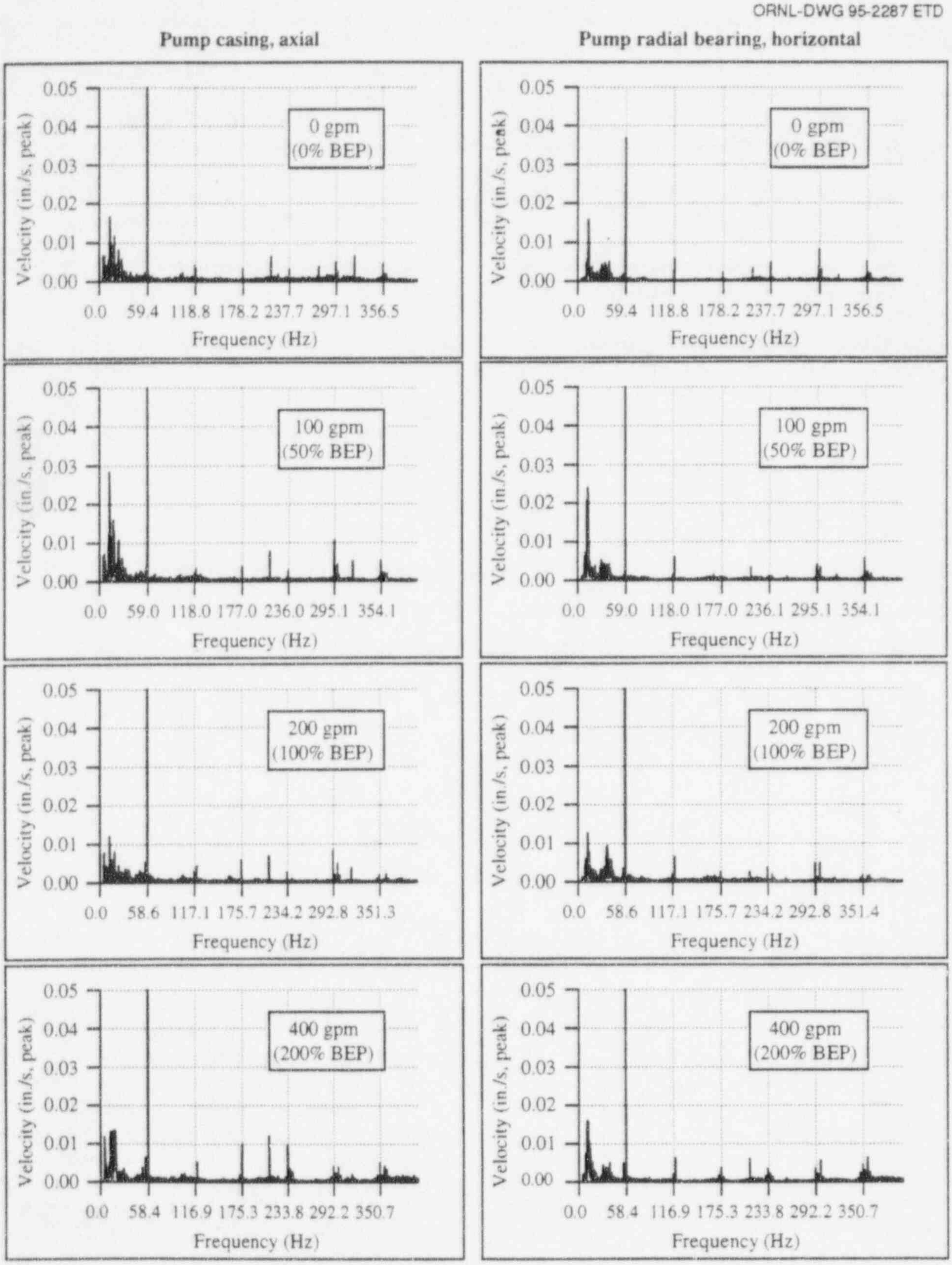


Figure 6.8 Pump B vibration spectra in velocity domain (zoomed)

to generate electrically related vibration (such as rotor bar pass and 120-Hz components) than conventional motors.

- The vane pass peak is least well-defined in the axial vibration for the lowest and highest flow cases. In Fig. 6.8, for which the x-axes are laid out in orders of running speed, the fifth harmonic (vane pass) of running speed can be seen to be better defined for the 50% and 100% of BEP cases than for the shutoff and 200% case.

Figure 6.9 presents summary, nonspectral rms vibration data for the flow rates previously discussed. For this pump, the general trend is that radial vibration generally increases with flow, while there is no consistent trend with flow for the axial vibration. The overall amplitudes, and the level of fluctuation at all flow rates, are less than that for Pump A.

Clearly, there is important information in the higher frequency region of the vibration spectrum; its effect is minimally seen in the overall, nonspectral rms amplitude, as the acceleration frequency spectra for these pumps have shown. Figure 6.10 shows the waveforms of the same vibration signal, with different digital filters applied. The upper plot shows the signal with a 6000-Hz digital low-pass filter applied to the signal (which was also analog low-pass filtered at 6000-Hz nominal cutoff frequency for anti-aliasing before digitization). The lower plot of Fig. 6.10 shows the same signal after having been low-pass filtered at 1000 Hz. The overall rms amplitude is only marginally changed. The filtering at 1000 Hz almost totally eliminates the effects of cavitation, rotor bar pass, and higher frequency bearing-generated vibration.

The significance of the 1000-Hz filter is that the ASME Code requires the frequency span to be included in vibration monitoring range from one-third of running speed to 1000 Hz. Many vibration analyzers allow the user to select the bandwidth and then apply anti-aliasing filters to limit signals outside of the selected bandwidth. Thus, an analyzer set up to cover the exact range required by the ASME Code would also filter out the higher frequency components previously noted.

6.4.3 Pump C Vibration Data Analysis

Figures 6.11 and 6.12 provide velocity and acceleration spectral vibration data for Pump C, which was monitored at the pump inboard and outboard bearings. The acceleration spectra for both bearings indicate multiple harmonics of a bearing-related frequency. Particularly in the 900- to 1500-Hz range, the equally spaced peaks (which are successive harmonics of a fundamental frequency that is just over 90 Hz) are indicative of a bearing flaw. Figure 6.13 shows enlarged results for the 100-gpm case, indicating the extent to which the bearing harmonics dominate the higher frequency portion of the acceleration spectrum. It is important to note that the overall amplitudes seen here are very low; they nevertheless provide an early indication of bearing degradation that, if trended through succeeding levels of degradation (which may also be characterized by relatively low amplitudes, but changing spectral signatures), would allow corrective action to be performed prior to bearing failure. The spectra and the noted overall amplitudes shown in Fig. 6.14 again demonstrate the lack of sensitivity of the overall amplitude value to bearing-related components of the spectrum.

6.4.4 Pressure Pulsation Analysis

Pressure pulsations originate from various sources, including impeller vanes passing diffuser vanes or volute cutwaters, nonsymmetric loading at rotational frequency, and other hydraulically borne load variations. The vibration components listed in Table 6.3 are associated with hydraulic conditions, and vibration is a secondary response to the hydraulic loadings. Careful pressure analysis should be an inherently better means of monitoring the sources of hydraulically induced vibration. Of course, pressure monitoring is somewhat intrusive in nature and is therefore not a practical tool to use as a routine field monitor.

Pressure pulsation data for Pump B were acquired and spectrally analyzed. Two pressure transducers were used—one located about 6 in. upstream of the pump suction nozzle, and the other located about 2 ft from the pump discharge nozzle. Both analog pressure signals were digitized. The suction pressure signal was subtracted from the discharge pressure signal to yield a pump head signal.

Figure 6.15 provides the pressure spectrum for the pulsations in the pump head at the same four flow conditions for which vibration data presented in Figs. 6.6–6.8 were acquired. The pulsation amplitude is expressed in rms terms. The pump vane pass frequency was, for all except the highest flow case, the dominant component but trended down with increasing flow rate. This is not surprising, since the overall head decreases with increasing flow rate; however, note that the decay is decidedly nonlinear with average head. At shutoff conditions, the rms pressure pulsation amplitude at vane pass frequency is over 1% of the total head. At the intermediate-flow conditions, the vane pass pulsation amplitude is less than 0.5% of the head. At the highest flow condition, it is again over 1% of pump head.

One particularly interesting result is the dramatic increase at four times running speed at the highest flow rate,

ORNL-DWG 95-2288 ETD

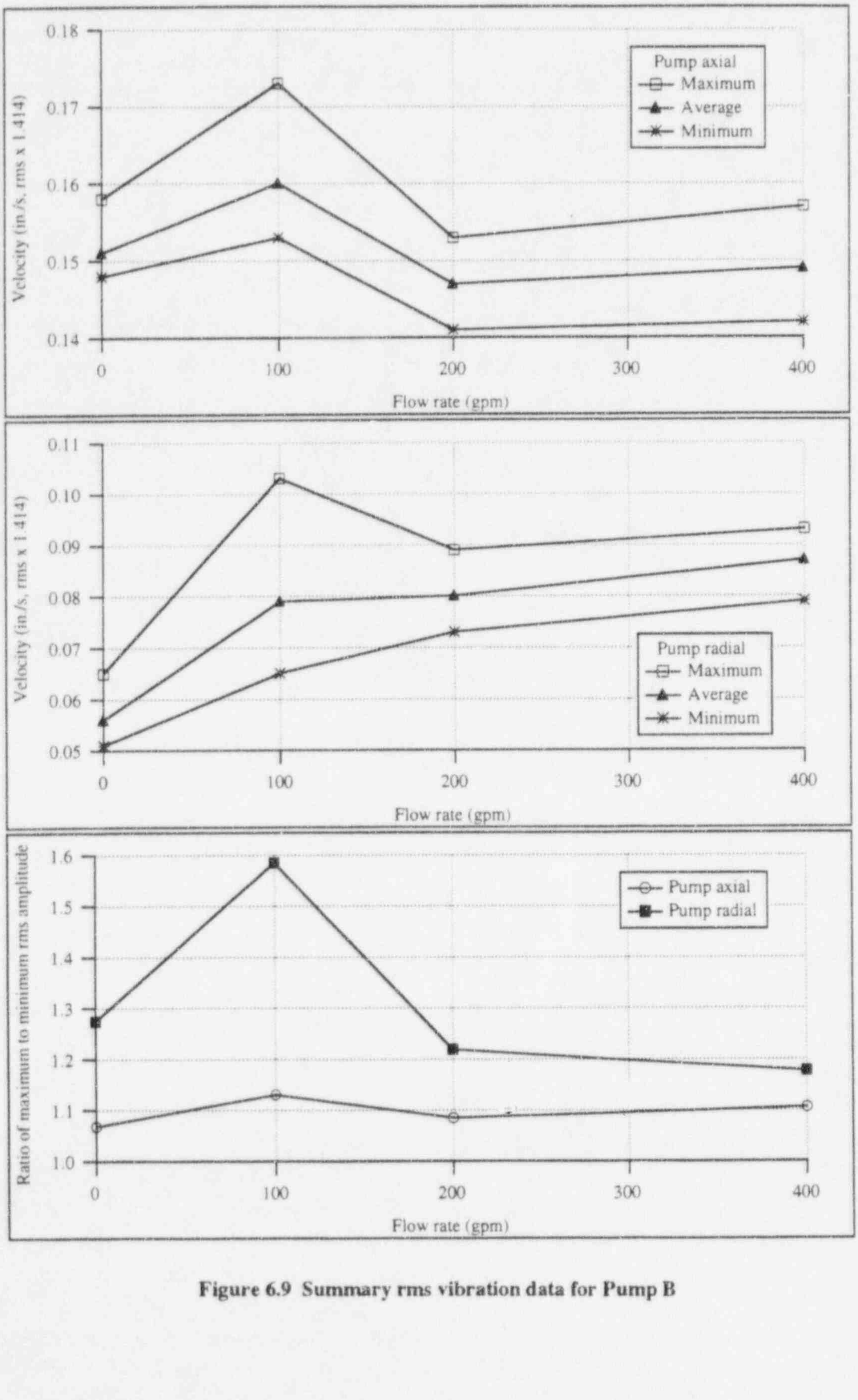


Figure 6.9 Summary rms vibration data for Pump B

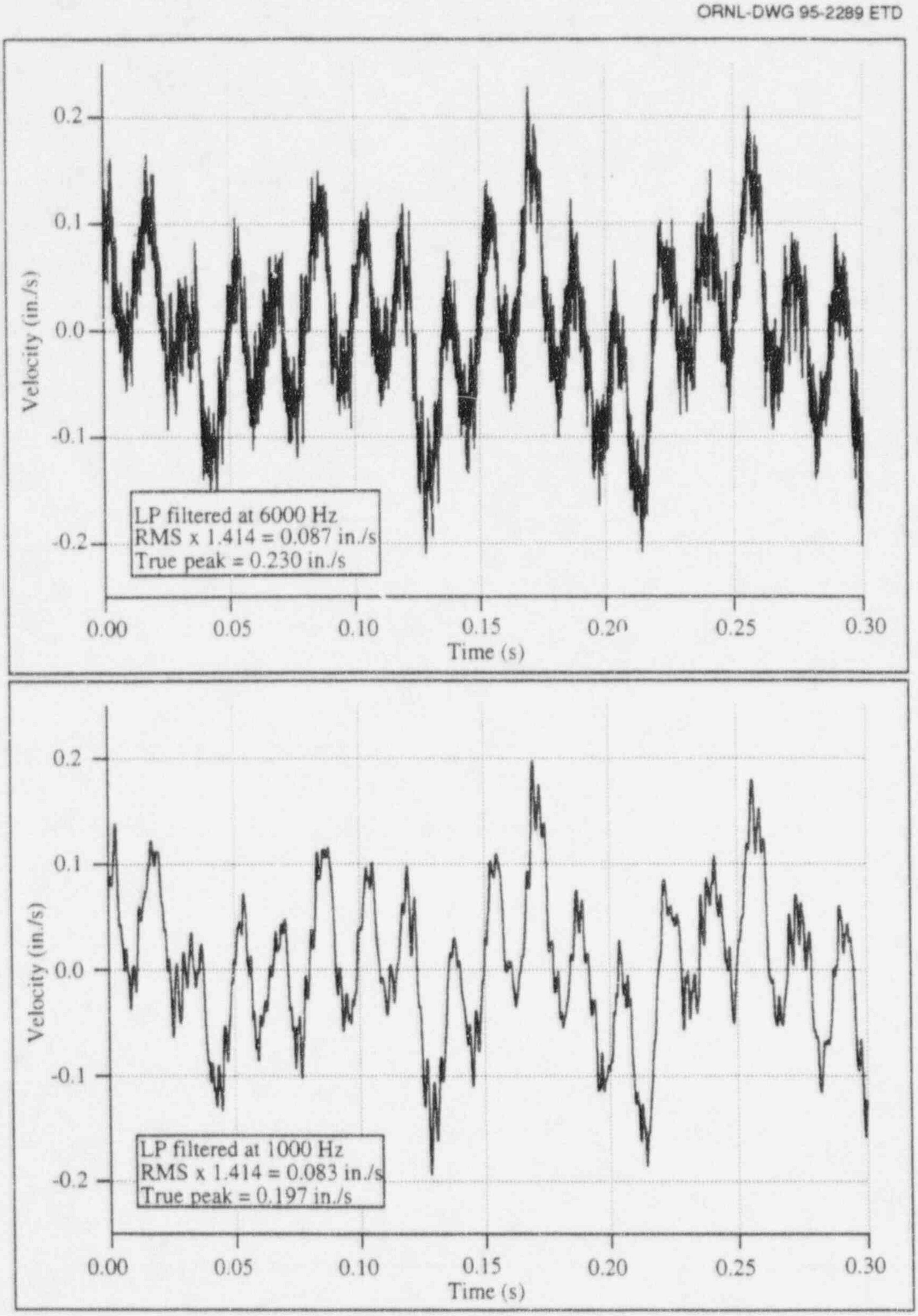


Figure 6.10 Radial vibration velocity waveforms for Pump B at 400 gpm for two digital low-pass filter applications

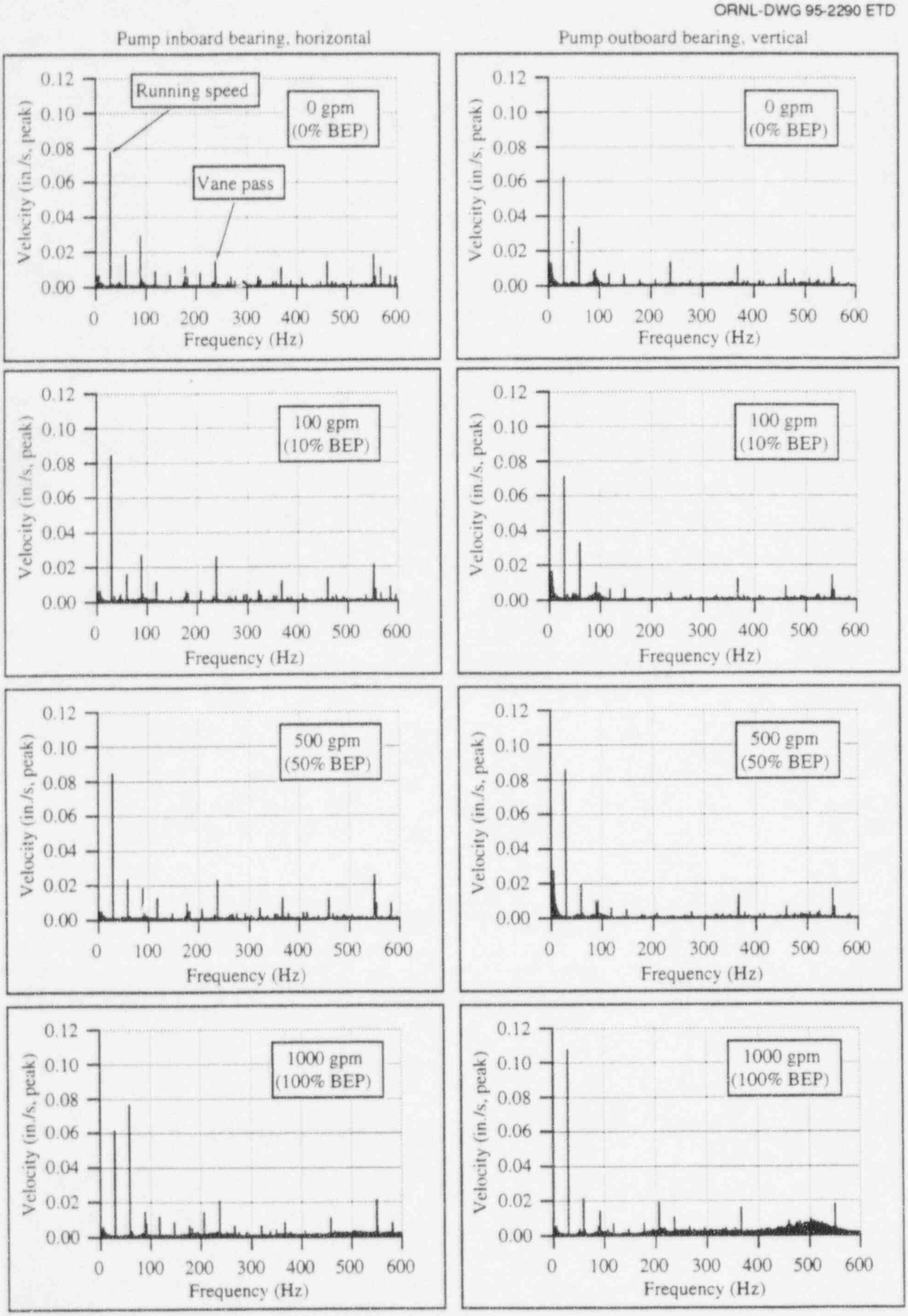


Figure 6.11 Pump C vibration spectra in velocity domain

ORNL-DWG 95-2291 ETD

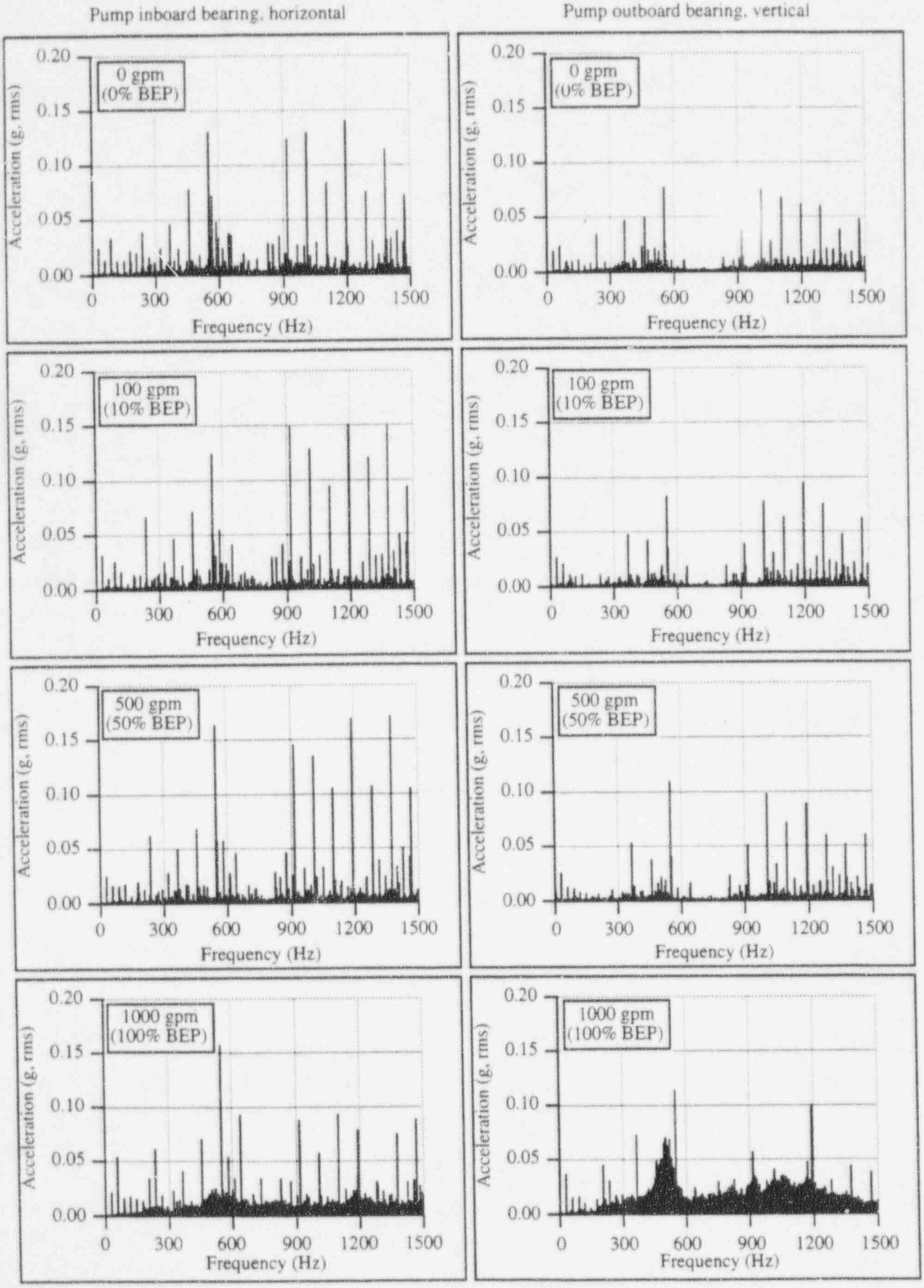


Figure 6.12 Pump C vibration spectra in acceleration domain

ORNL-DWG 95-2292 ETD

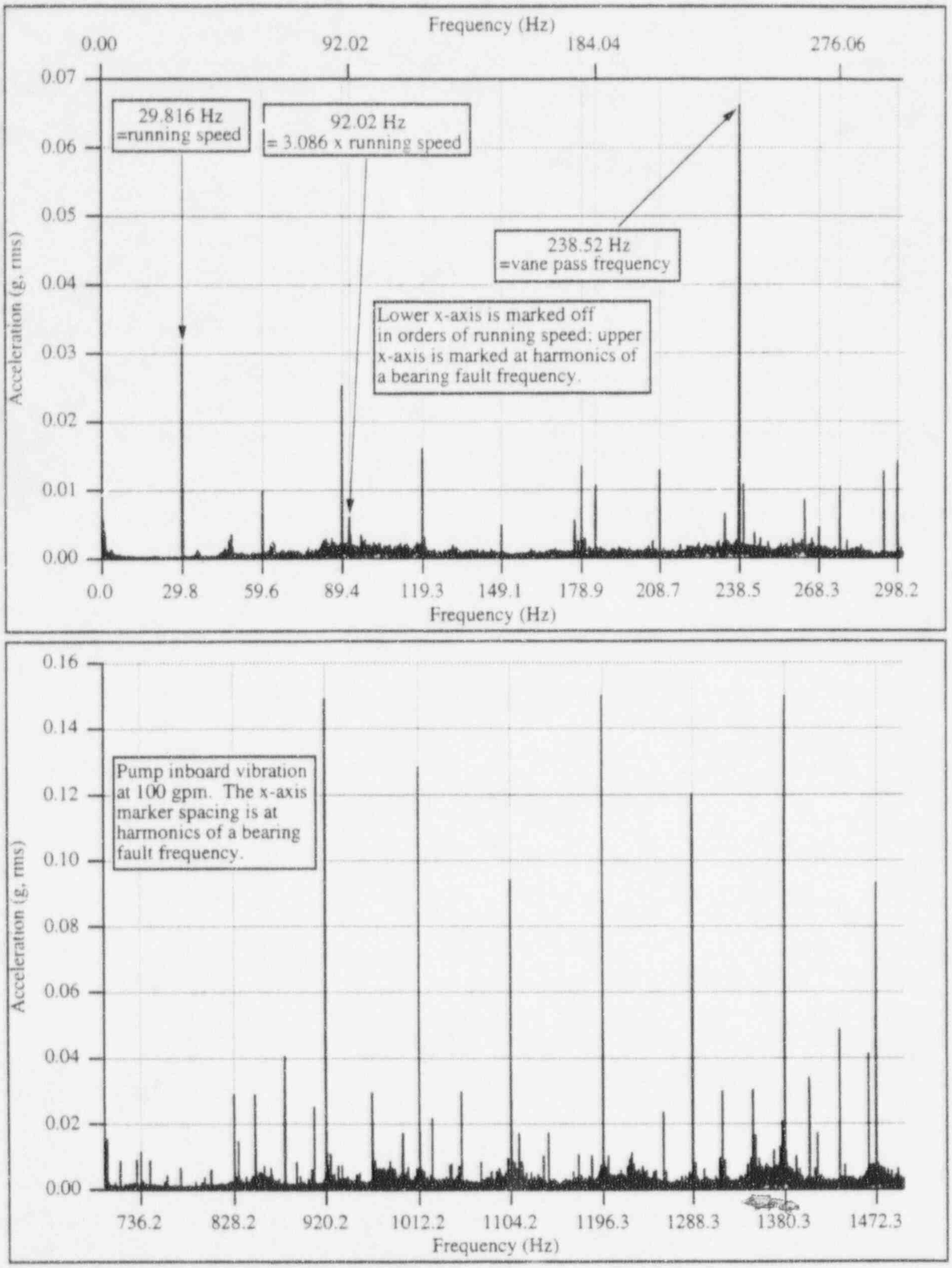


Figure 6.13 Pump C vibration spectra showing hydraulic and bearing-related fault frequency peaks

ORNL-DWG 95-2293 ETD

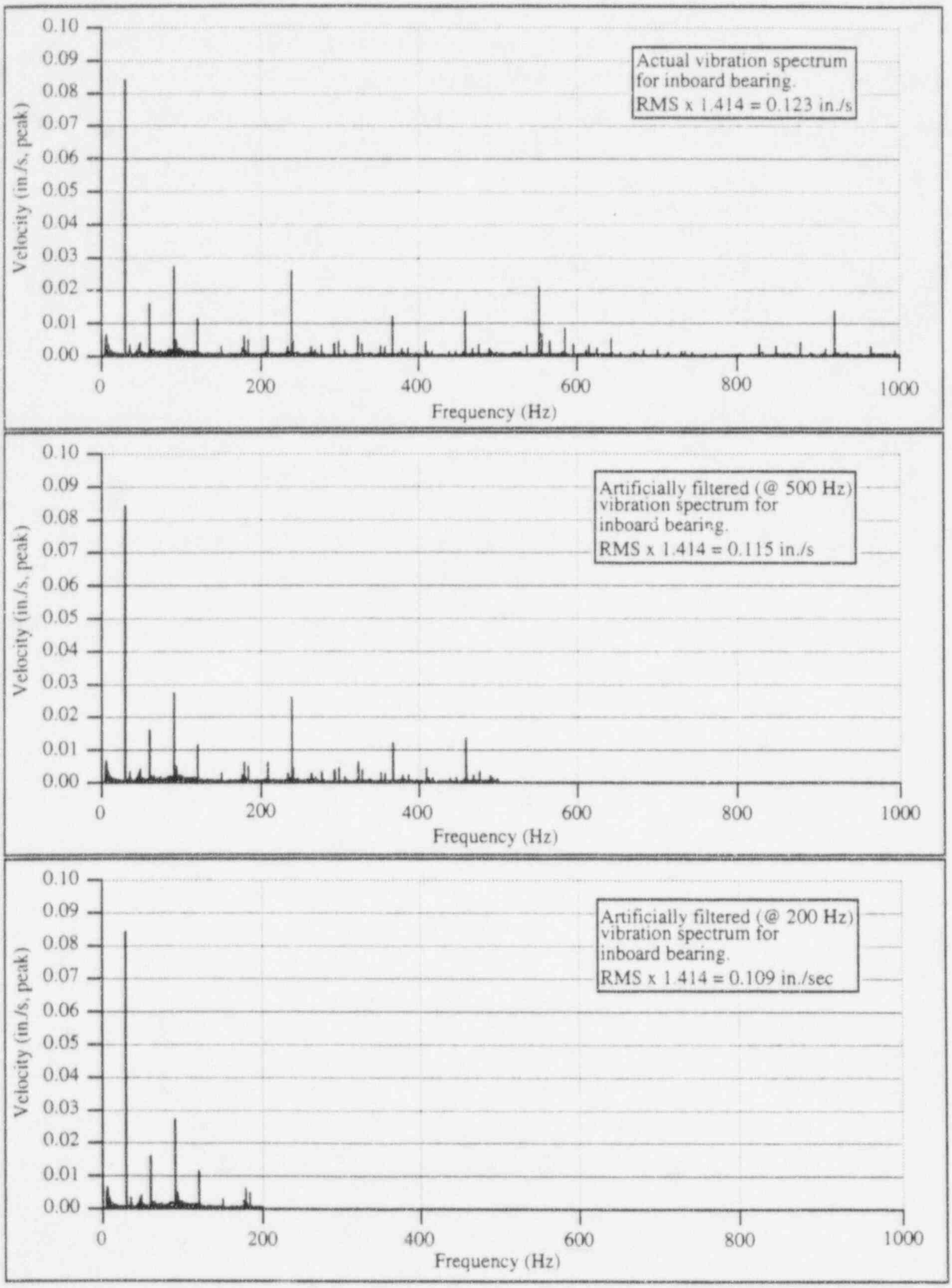


Figure 6.14 Pump C vibration velocity spectra: as measured and with artificial filtering

ORNL-DWG 95-2294 ETD

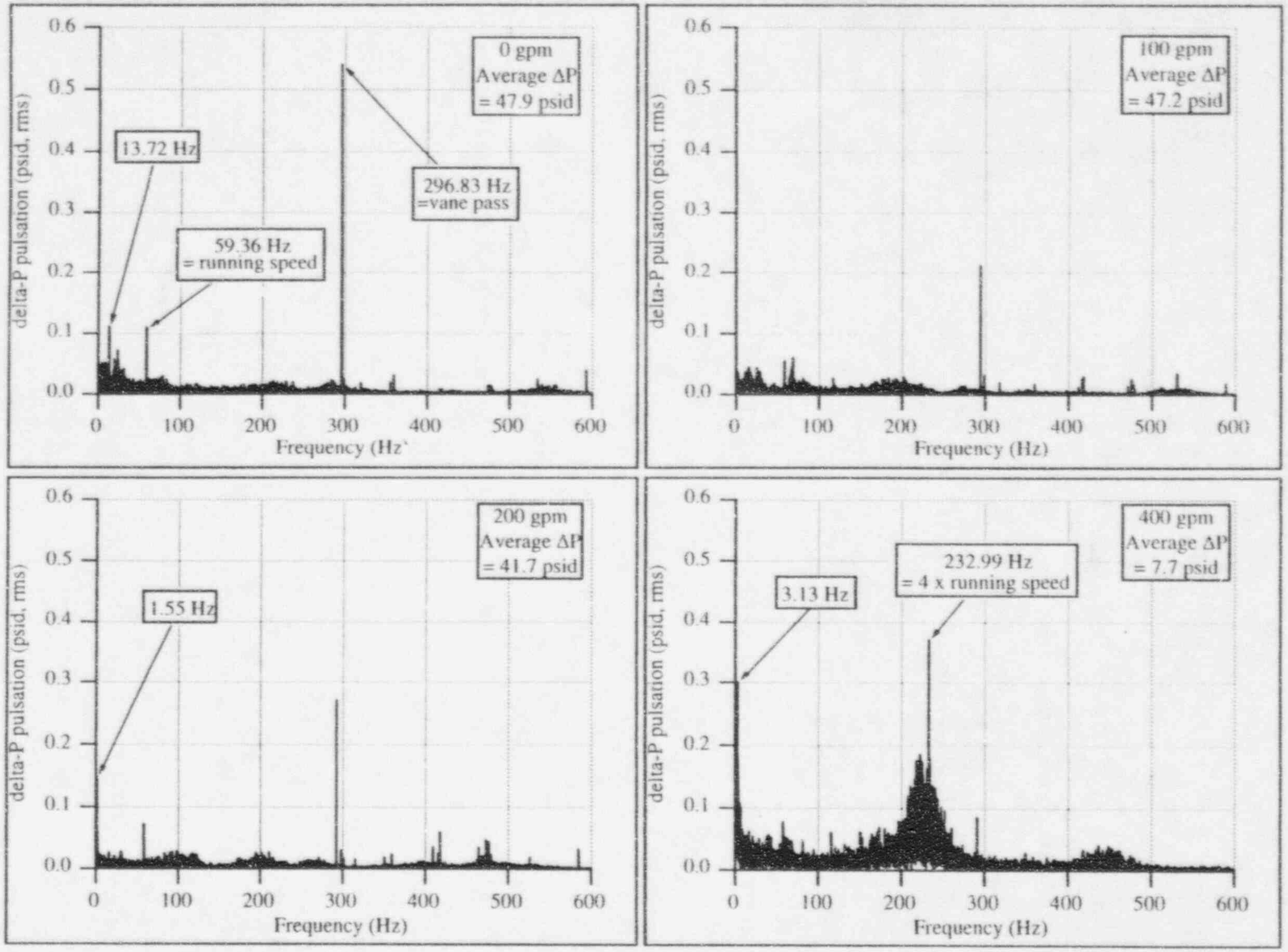


Figure 6.15 Pump ΔP pulsation spectra—Pump B

accompanied by a general broadband noise in the spectrum in this vicinity.

Pump suction and discharge pressure pulsation spectra are shown in Fig. 6.16. The subsynchronous suction and discharge spectra are shown in more detail in Fig. 6.17. By comparing the total head, the suction, and the discharge spectra, certain spectral components can be tied to specific sources. For example, the dramatic change in total head at four times running speed for the highest flow case (Fig. 6.15) is not present in the suction pressure spectrum, but is present in the discharge spectrum (Fig. 6.16), indicating that this is a discharge-related phenomenon. Another example feature of interest can be seen at 15.3 Hz in both the suction and discharge pressure spectra at 100 gpm but is not present in the total head spectrum. This suggests that an external influence, such as a resonant frequency in the loop, affected both signals individually, and is probably not a pump-generated component. Other spectral features such as the spectral peaks at 1.55 and 3.13 Hz for the 200- and 400-gpm cases, respectively, are common to all three signals. It is of interest that the ratio of these two frequencies is the same as the flow ratio.

6.4.5 Motor Power Analysis

Motor input power was monitored on all three pumps. All three phases of current were monitored using clamp-on current transformers. All three phases of voltage were monitored using dropping resistor networks. The total input power was measured by summing the products of the individual phase currents and voltages (using both analog and digital multiplication).

Motor power acts as a transducer of the motor load. Obviously, as the motor load increases, the motor input power increases to accommodate the load; thus it is sensitive to changes in pump hydraulic and mechanical loads. For most motors, the motor power is relatively linear with load, so it is a parameter that gives a reasonably representative indication of load fluctuations. Motor power does have some inherent frequency limitations when monitoring pumps due to the rotating inertia of the motor and pump, as well as the pumped fluid. In order for the motor power to change in an induction motor, the rotating speed must either increase or decrease (as the result of a load change). High-frequency components, such as vane pass frequency, create load fluctuations at such a rate that the relatively large inertia of the rotating equipment and the pumped fluid essentially allow the rotating equipment to pass through these events with little or no change in speed.* However, for lower frequency load components, particularly at synchronous speed and below, the motor input power can be a relatively informative tool.

It is important to recognize that the motor power spectra are merely reflective of the *stator* power input. In order to more closely relate the stator input power to rotor power fluctuations and fundamentally to the torque fluctuations, it would be necessary to model the motor and to account for the overall system inertia. Nevertheless, the motor power input spectra provide some unique insights into load fluctuations, as will be shown.

It might be noted here that motor current also responds to load changes and has been historically used to monitor motor loads.[44] Motor current is somewhat nonlinear with load for most motors (being most responsive at high loads) however; its use to understand load fluctuations at different load conditions requires the application of a current vs load adjustment. Current as a transducer of pump loads was not analyzed here for that reason. The ease of acquiring current (vs power) makes it an inherently attractive parameter to monitor, and it is likely that with correction factors applied to account for nonlinearities, current could also be effectively employed as a tool for understanding pump conditions.

Figures 6.18 through 6.20 provide normalized power spectra for Pumps A, B, and C.† The normalization is performed by dividing the power spectra by the average power for the particular flow condition. For Pump A, it can be clearly seen that at the two low-flow conditions, the low-frequency end of the power spectrum (up to about half of running speed) is much noisier than at the higher flow rates. Also, note that the spectral peak just below 30 Hz (corresponding to running speed) decreases with increasing flow, which is exactly opposite to the pattern noted in the axial vibration data for Pump A. One possible interpretation of this apparent inconsistency emerges when it is recognized that both mechanical and hydraulic unbalance are manifested at running speed, but the phase angles of the mechanical and hydraulic unbalance components are not necessarily aligned. Since the pump motor input power is not particularly sensitive to mechanical unbalance (except at significant levels of unbalance, in our experience) but is

*It might be noted that vane pass frequency has been observed in spectral analysis of motor power, however. The results to date show some promise, but have not been fully explored.

†Note that the four flow conditions for Pump B power are not identical to those shown for vibration and pressure pulsation previously. Instead, the selected flow rates for the three pumps are roughly equivalent, in terms of percent of BEP.

ORNL-DWG 95-2295 ETD

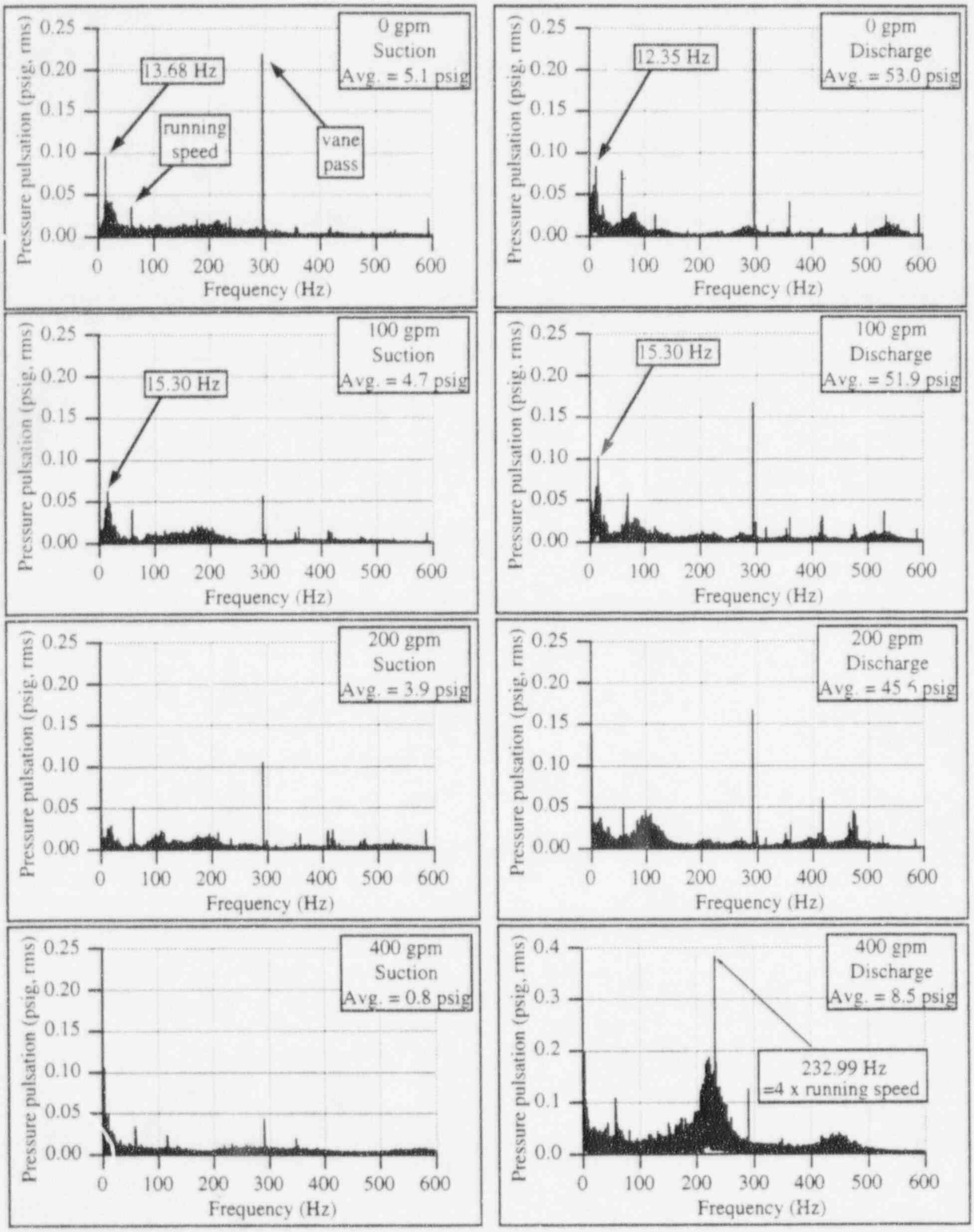


Figure 6.16 Pump suction and discharge pressure pulsation spectra—Pump B

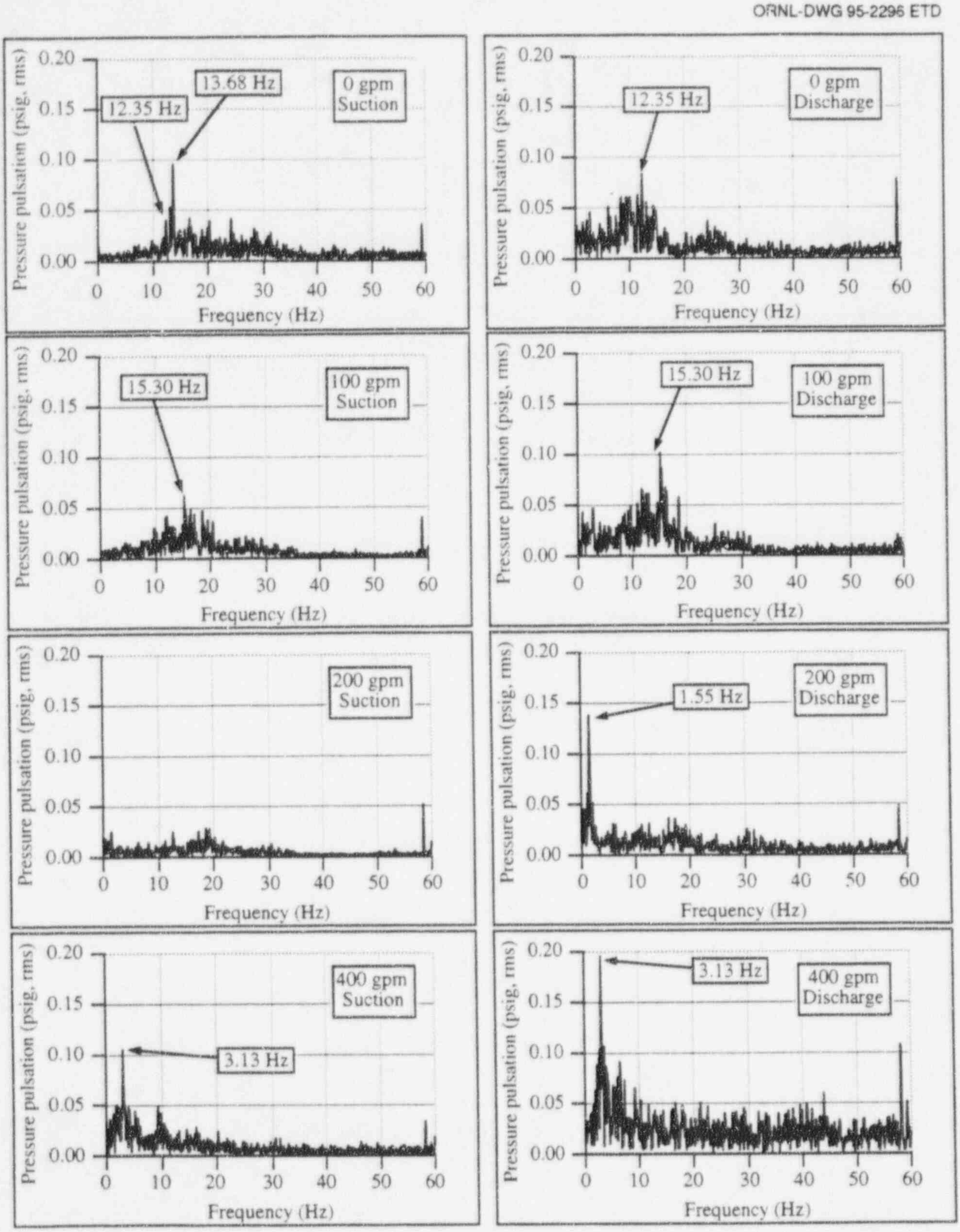


Figure 6.17 Pump B suction and discharge pressure low-frequency pulsation spectra

ORNL-DWG 95-2297 ETD

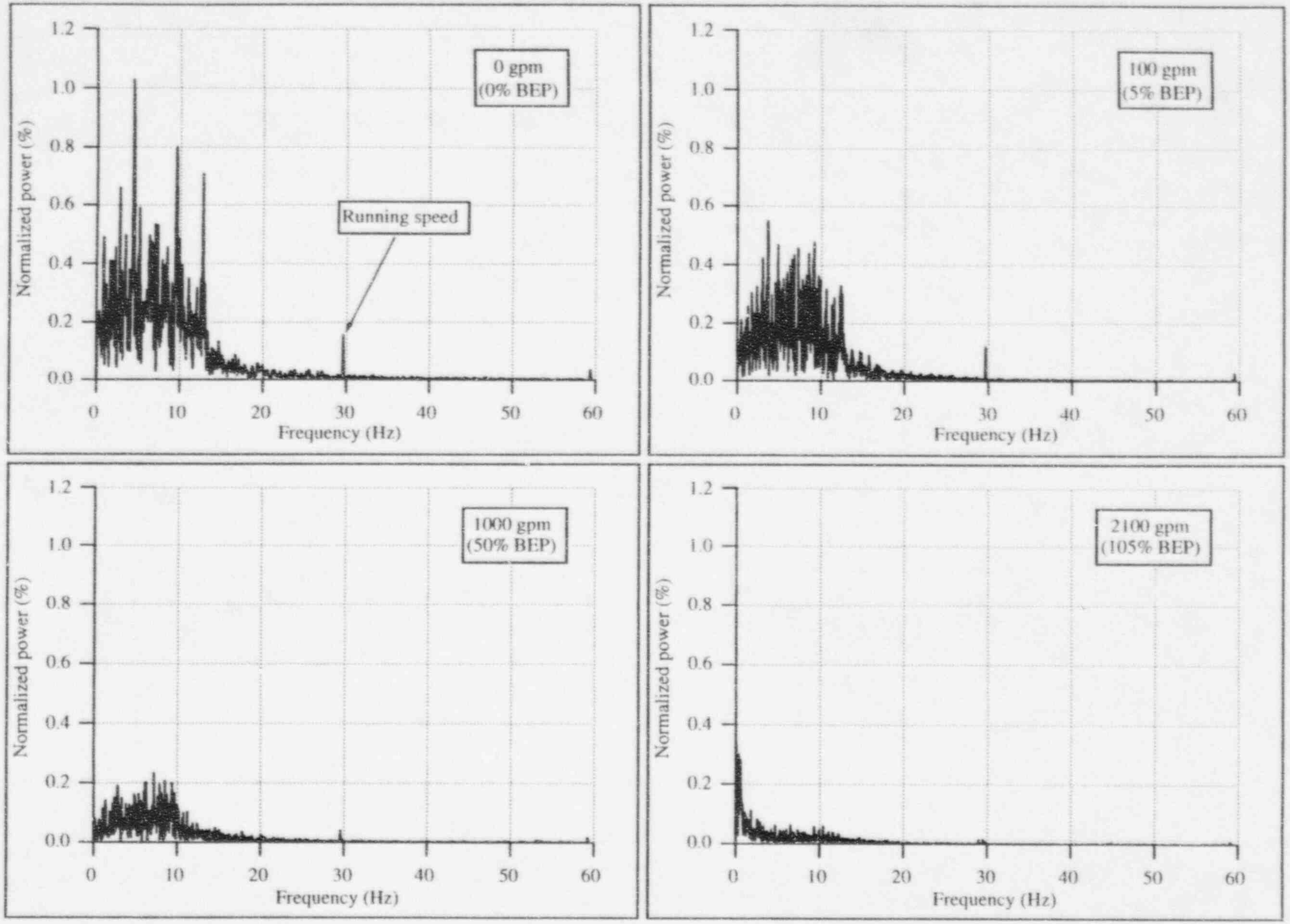


Figure 6.18 Selected power spectra for Pump A

ORNL-DWG 95-2298 ETD

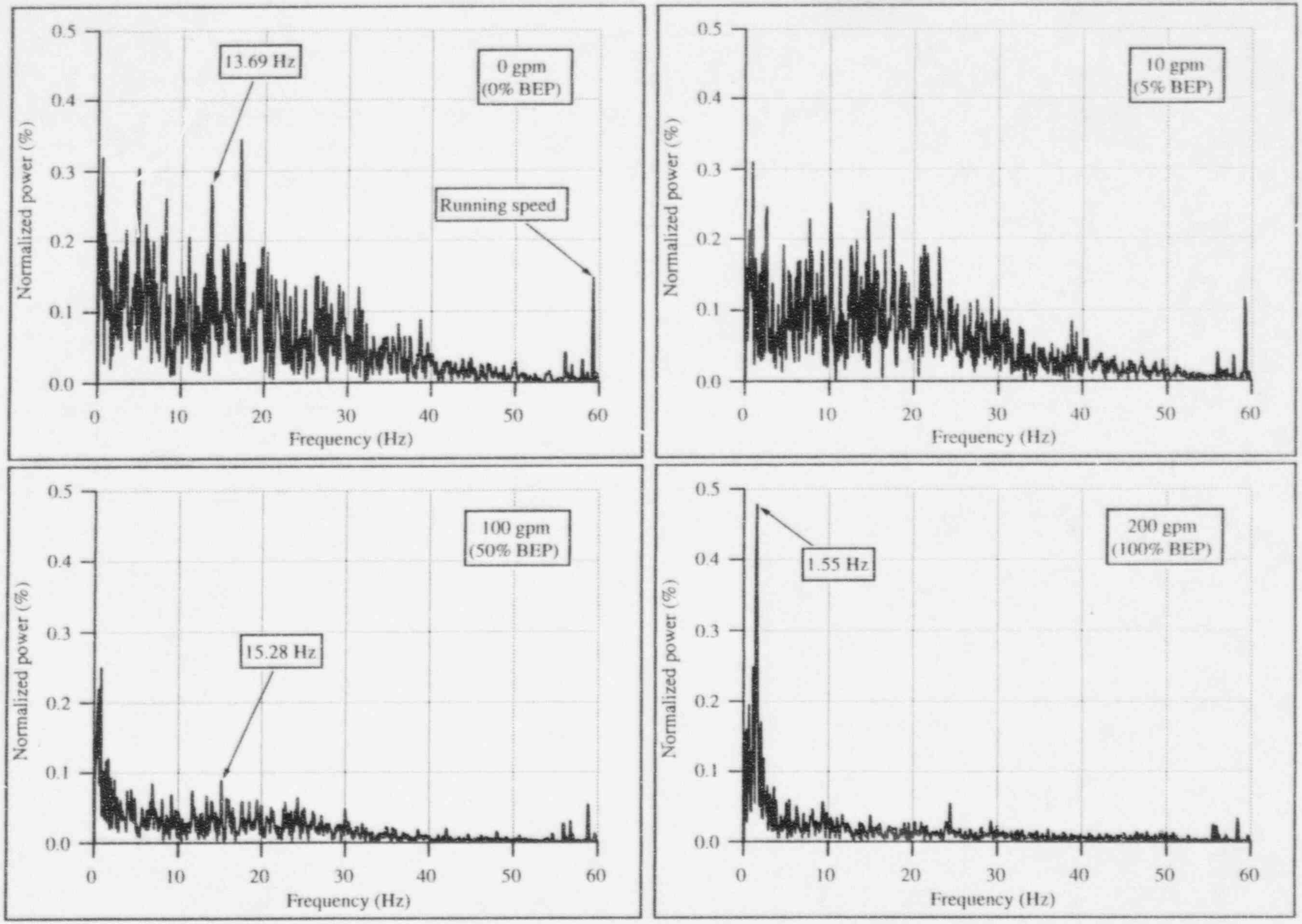


Figure 6.19 Selected power spectra for Pump B

ORNL-DWG 95-2299 ETD

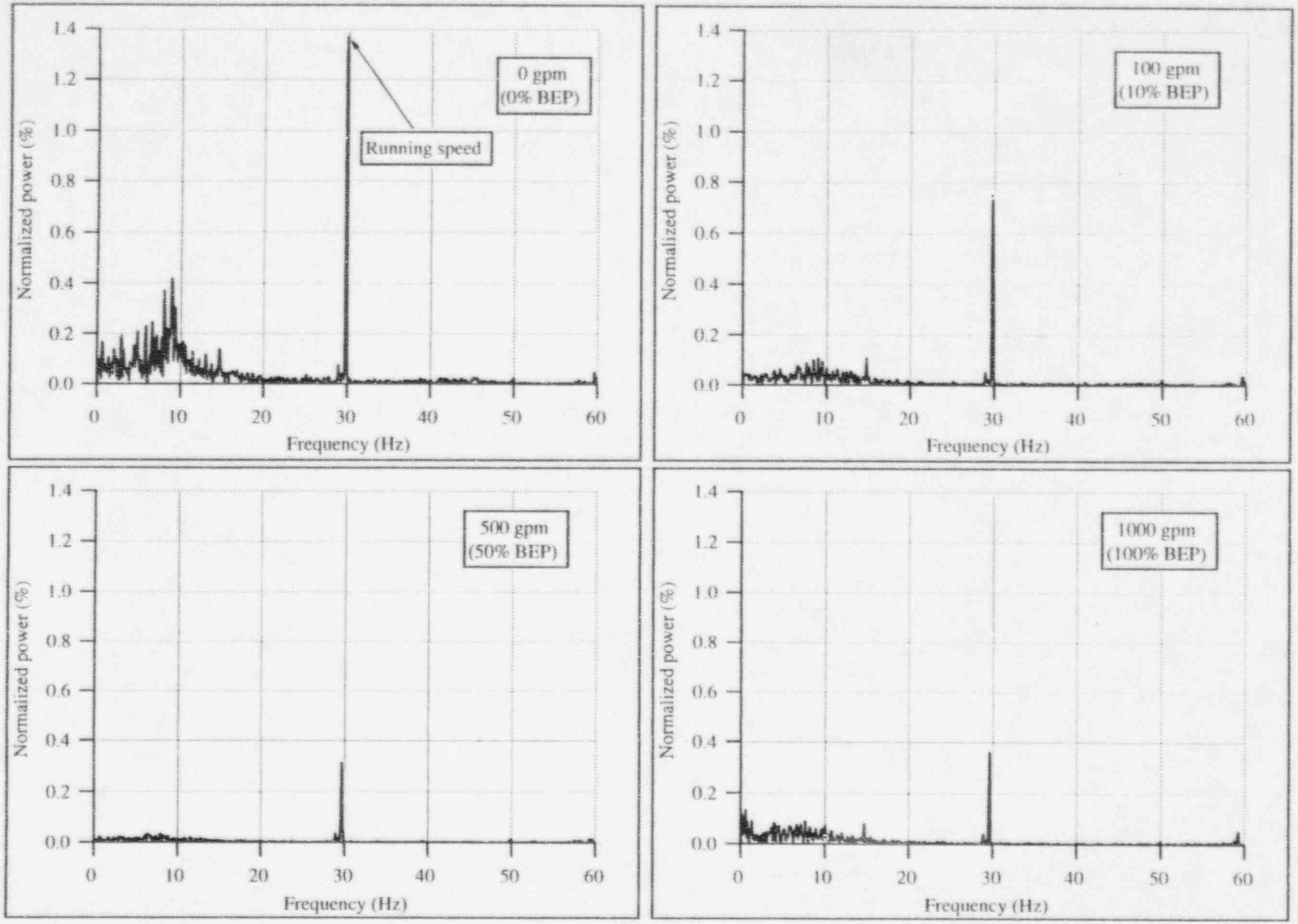


Figure 6.20 Selected power spectra for Pump C

sensitive to *torsional* load fluctuations, the data suggest that there is increased hydraulic unbalance at low flow. However, this hydraulic unbalance is apparently out of phase, to some extent, with the existing mechanical unbalance, resulting in decreased overall running speed vibration at low-flow conditions.

Pump B also has a level of low-flow-related broadband noise at low frequencies (again, up to about half of running speed), although at a significantly lower absolute level than Pump A. The amplitude of the running speed peak (at just less than 60 Hz) also diminishes with increasing flow. Pump C also exhibits low-frequency broadband noise for the shutoff flow condition. The running speed amplitude decreases with increasing flow, as was observed for both other pumps. More interestingly, the running speed amplitude for Pump C is almost an order of magnitude greater than that for Pump A (which is more directly comparable, since it operates at about the same speed). This is particularly significant since the vibration data for Pump C showed considerably lower running speed amplitudes than did Pump A. Also, note that the Pump C outboard bearing vibration amplitude at running speed increased slightly with increasing flow rate. Thus both the Pump A and the Pump C motor load fluctuation spectra provide insights into torsional load fluctuations that are not at all available in the vibration spectrum. The high level of running speed load fluctuation in Pump C is, at least in part, attributable to the fact that this is a double-suction pump; difficulties in casting symmetrical double-suction pumps cause them to be more likely to experience hydraulic unbalance (at 1 × running speed).

As flow rate increases, there is a general trend for all three pumps to develop some very low frequency (<5 Hz) energy. This can be seen for Pump A at 2100 gpm (105% BEP); in Pump B at both 200 and 400 gpm (100% and 200% BEP, respectively); and in Pump C at 1000 gpm (100% BEP). For Pump B, the energy is manifested as a more discrete peak.

Figure 6.21 presents a summary plot of power instability as a function of flow, including more flow conditions than have been discussed above. The motor power instability is calculated by dividing the standard deviation of the motor power by the average motor power and is thus normalized for the different flow rates of a particular pump, as well as providing a common parameter against which to compare different pumps. The load of Pump A clearly fluctuates more at low-flow conditions than either Pump B or C. The pumps have instability ratios that are comparable near the BEP, however. The test loop for Pump B is a relatively short loop that can be operated with minimal flow restrictions, thus allowing the pump to be run at flow rates significantly beyond its design condition. At the highest flow rate, the motor power exhibits more instability than at even the shutoff flow condition. Note that for all three pumps, the point of minimum instability is between 70 and 90% of BEP. Logic would suggest that the point of least instability would be at BEP. One possible explanation for this apparent inconsistency is system configuration. Since the BEP (taken from vendor pump curve) is not based on as-installed configurations, but on pump vendor test facility configurations, it is possible that the piping configuration in the field was not optimum. For example, Pump A has an

ORNL-DWG 95-2431 ETD

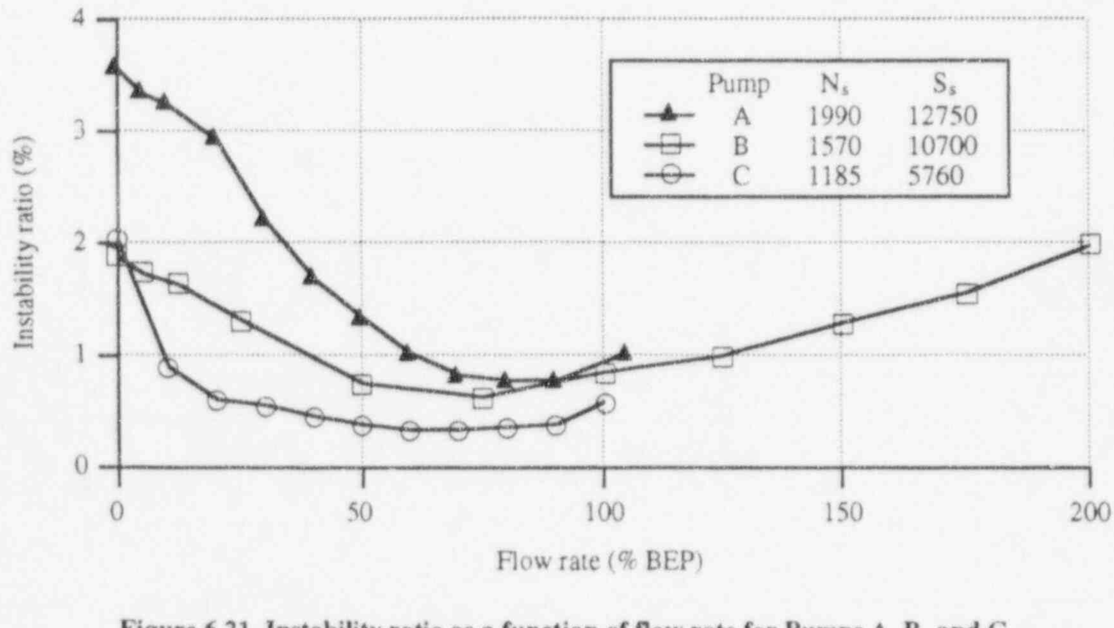


Figure 6.21 Instability ratio as a function of flow rate for Pumps A, B, and C

elbow installed at the pump suction, while a reducer, gate valve, and elbow are immediately upstream of the Pump C suction.

6.5 Comparing Results

Vibration and motor power data were recorded for Pump A. The results of the individual parameters were discussed previously. The overall vibration velocity spectra (Fig. 6.1) tend to be dominated by running speed and harmonics, making it more difficult to see the broader patterns in the acceleration domain (Fig. 6.2). The acceleration domain spectra for Pump A (particularly axial vibration) show a considerably higher level of broadband noise at low-flow rates (Fig. 6.2). The lower frequency spectra of the vibration velocity show a marginal increase in low-flow-related broadband noise from below running speed to about four times running speed. While these particular spectral components are not directly observable in the motor power signal, the results of the broadband noise are apparent in that the unstable flow-induced noise also causes lower frequency load variations that are manifested as input power fluctuations. It is important to note that the overall rms vibration velocity signals, measured in accordance with ASME Code requirements (Fig. 6.4), do not show the tendency toward hydraulically unstable conditions at low flow (although the variations in the rms amplitude do shed some insight).

Vibration, pressure pulsation, and motor power data were recorded for Pump B. The overall vibration amplitude was minimally affected by operation at either low or very high flow rates. The vibration acceleration spectra did show a significant increase in broadband, higher frequency noise at the highest flow rate (Fig. 6.7).

The Pump B pressure pulsation data indicated that vane-pass frequency amplitude dropped with increasing flow rate. However, at the highest flow rate, a significant level of four times running speed activity was noted in the pump discharge pressure. Similar patterns were not found to exist in the vibration spectra. Some significant very low frequency peaks in the pressure pulsation data at the higher flow rates were observed. The amplitude of the running speed peak in the pump suction pressure was found to be largest at the lowest and highest flow rates.

For Pump B, as for Pump A, the motor power spectra do not show the same spectral features as vibration. However, there were identical spectral peaks in the motor power and pressure pulsation spectra at the low-frequency range. For example, note the presence of spectral peaks at 1.55 Hz in both the pressure and power spectra for the 200-gpm case. Although the power spectrum for the 400-gpm case is not shown, it also had a spectral peak in common with pressure pulsation (at 3.13 Hz). The motor power instability was also consistent with the amplitudes at running speed in the pressure data. Likewise, for the highest flow condition, the high level of noise at four times running speed was manifested as an increased power fluctuation.

For Pump C, vibration and motor power data were recorded and analyzed. The most notable feature of the Pump C analysis results was the high amplitude of the running speed frequency in the motor power spectra, even though the bearing housing vibration was relatively low.

6.6 Pump Stressors Summary

Previous studies indicate that the principal sources of service wear and degradation of internals for AFW pumps are hydraulic- and cavitation-related loads. Available literature indicates that various hydraulic-related stressors occur at frequencies ranging from as low as 0–10 Hz to 10 kHz (and beyond).

The spectral vibration data collected during this study generally agreed with the literature (although it should be noted that the pumps tested were relatively low power pumps and would not be expected to experience the extent of hydraulic loading that much of the literature is based on). A comparison of the spectral vibration data with overall, nonspectral amplitude indicated that the overall amplitude is not particularly sensitive to either hydraulic load variations or to bearing condition.

Motor power data were observed to have some features that are similar to pressure pulsation data; however, pressure transducers are not as limited in frequency response as the motor power. The principal drawback to the use of pressure transducers is their intrusive nature. Motor power load fluctuations appear to provide some useful insights into the general hydraulic stability of pumps that is complementary to that available from vibration spectra. Since radially and axially mounted vibration transducers will not necessarily be able to observe torsional fluctuations, motor power may be able to detect effects that cannot be detected by vibration (likewise, other sources of degradation, such as mechanical imbalance, may not be observed in motor power). Further study of how to best quantify and relate motor power to other parameters is needed before it can be applied more broadly.

It is important to note that the principal merits of motor power or current monitoring of pumps are in the spectral and noise characteristics of the data, *not in the simple overall amplitude*. In many ways, this is analogous to vibration data, where likewise the overall amplitude provides minimal diagnostic insights.

7 Conclusions

The development of new and informative measurement and diagnostic technologies in the past two decades has enabled machinery analysts to comprehend many of the cause-and-effect relationships that manifest specific pump operation and behavior. More accurately than ever before, pump performance can be quantified by a host of various parameters that are easily and precisely measured. Because of these technological advances, pump degradation is more preventable and controllable than ever before.

Vibration spectral analysis is widely accepted as an extremely powerful tool in the machinery analyst's diagnostic arsenal. Abnormalities such as misalignment, unbalance, looseness, and bearing faults are readily discernible with spectral analysis. In the 1970s, when most nuclear power plants were either under construction or on the drawing board, this science was in its infancy; the 1980s saw this technology grow and mature; and in the 1990s, vibration spectral analysis has become a cornerstone of predictive maintenance programs throughout the world.

However, the collection and trending of overall vibration amplitudes is not vibration *spectral* analysis. As discussed in this report, the overall vibration displacement (and more recently, overall velocity) measurements mandated by the existing ASME Code contribute limited information value to the pump analyst. In particular, bearing degradation, known as a leading cause of pump degradation and failure, is often not evident in overall vibration amplitude measurements until the bearing faults are so severe that there is little remaining bearing life in which to plan and conduct maintenance. When degradation that normally occurs over a long period of time remains undetected, other pump components can be unnecessarily affected and/or damaged. Machinery analysts know many causes of pump degradation and, with today's technology, also know how to detect and prevent it.

The historical trending of lubricant analysis data can verify normal and abnormal machinery wear and degradation, can verify the presence of moisture and other contaminants, and can detect bearing wear at its earliest stages. The availability of on-site analysis packages has made this technology more practically useable for a wider variety of equipment.

Advances in thermographic measurement equipment have enabled machinery analysts to detect degradation not readily identified through other conventional diagnostic techniques. Thermography is nonintrusive and can be used to scan for hot spots that are indicative of impending degradation. For example, misaligned couplings that create excessive friction are easily seen through the lenses of the thermographic camera. Other abnormalities such as overheated bearings, misaligned and rubbing shafts, and components exhibiting inadequate lubrication have all been detected through thermographic evaluation.

The performance of a pump is dependent upon the operability of the pump itself, but also its driver. There have been numerous advances in electric motor diagnostics in the past decade. As highlighted in this report, nonintrusive motor current signature analysis and other motor monitoring techniques are making new inroads into the diagnostics of motor degradation. Since one purpose of required safety-related pump testing is to ensure the performance of the pump during an emergency situation, then it would be advisable that diagnostic evaluations be extended to also include the motor that drives the pump, and how the pump and motor perform as a system. Furthermore, monitoring of the motor parameters such as current and power can provide valuable insights into *pump* operational condition, which may not be as readily or as nonintrusively obtained from other diagnostic methods.

Expert diagnostic systems have been available for the past decade; time has seen their abilities expand and their costs decline. Most expert systems marketed in the United States today do have a good diagnostic success rate. They provide an excellent starting point for diagnosing equipment problems, can provide guidance and insight regarding a variety of different machinery anomalies, and can save valuable time when identifying incipient failures. The use of these artificial expert systems on the more routine and mundane tasks can also relinquish time for human experts to study the more time-intensive problems. With an increasing number of nuclear plants entering into their third decade of operation, maintenance organizations that merge state-of-the-art expert systems with their normal diagnostic procedures may increase their capability to detect and correct machinery problems and, therefore, increase the probability of continued safe plant operation.

The use of spectral vibration analysis, oil analysis, thermography, motor current and power analysis, and other advanced techniques provides the pump diagnostician with a powerful set of tools with which to monitor the operational health of the pump. For many of these techniques, there are no consensus standards governing acceptable limits; in fact, the limits are somewhat machine dependent. Nevertheless, the *trending* of such parameters provides excellent insights into many of the degradation mechanisms that may act on pumps. A careful implementation of such techniques, even in the absence of hard and fast criteria, can provide much more useful information about pump health than has historically been available through existing ASME Code requirements. While it is beyond the scope of this document to suggest specific implementation methodology, serious consideration of less explicit, more flexible requirements appears warranted.

References

1. W. E. Nelson, "Addressing Pump Vibrations: Part I," *Pumps and Systems*, 1(2) (March 1993).*

2. T. L. Doersher, "Optimum Utilization of Predictive Maintenance and Performance Monitoring Programs," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.†

3. G. Zwingelstein, "The Role of Predictive Maintenance in the Reliability Centered Maintenance (RCM) Program of Electricite de France," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.†

4. C. Jackson, *The Practical Vibration Primer*, Houston: Gulf Publishing Co., 1979.

5. J. R. Nicholas, Jr., and R. K. Young, "Status of Condition Monitoring and Predictive Methods and Systems in United States and Canadian Nuclear Utilities as of Early 1991," pp. 278–292 in *Proceedings of the ANS International Meeting on Nuclear Power Plant and Facility Maintenance, Salt Lake City, Utah, April 7–10, 1991*, Vol. 1, June 1991.

6. A. R. Bush, "Calculate Temperature Rise through Boiler-Feed Pumps," *Power* 107(4), 69 (April 1963).*

7. I. J. Karassik et al., Eds., *Pump Handbook*, 2nd Edition. New York: McGraw-Hill, 1986, pp. 2, 281.

8. J. R. Nicholas, Jr., "Predictive Condition Monitoring of Electric Motors," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.†

9. J. D. Kueck, "Assessment of Valve Actuator Motor Rotor Degradation by Fourier Analysis of Current Waveform," Paper No. 92 WM079-4EC presented at the IEEE/Power Engineering Society Winter Meeting, January 26, 1992, *IEEE Transactions* (1992).*

10. J. S. Mitchell, *Machinery Analysis and Monitoring*, Penn Well Publishing Company, Tulsa, Oklahoma, 1981.

11. T. J. Holroyd and N. Randall, "Use of Acoustic Emission for Machine Condition Monitoring," *British Journal of Non-Destructive Testing*, 35 (2), 75–78 (February 1993).*

12. B. L. Agee, X. Zeng, L. D. Mitchell, and M. I. Schiefer, "An Automated System for Acquiring Modal Analysis Data," *Sound and Vibration*, 14–21 (June 1992).*

13. T. Spettel and R. Garvey, "On-site Oil Analysis: A New Tool for the Vibration Analyst," pp. 49–54 in *Proceedings of the 17th Annual Meeting of the Vibration Institute, Willowbrook, Ill., June 8–10, 1993*, 1993.

14. Computational Systems, Inc., "PC-Based Integration of Spectrographic, Ferrographic and Vibration Analysis Data," *P/PM Technology* 4(1), 21–23 (Jan.–Feb. 1991).*

15. J. Mountain, "Lubricant and Mechanical Condition Analysis," *Nuclear Plant Journal*, 11(3), 72–90 (May–June 1993).*

16. R. Wurzbach, "Thermographic Monitoring of Mechanical Components in Rotating Equipment," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.†

17. D. Hosmer, "Boiler Feed Pump Program," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.†

18. N. B. Stockton, "Considerations for Reference Pump Curves," pp. 115–127 in *Proceedings of the Second NRC/ASME Symposium on Pump and Valve Testing*, NUREG/CP-0123, July 1992.†

19. T. F. Hoyle, "The Comprehensive Test: A New Approach to Nuclear Pump Testing," EPRI Power Plant Pumps Symposium, Tampa, Fla., June 1991.†

20. S. Yedidiah, "Diagnosing Problems of Centrifugal Pumps, Part II," *Fluid Movers: Pumps, Compressors, Fans, and Blowers*, pp. 187–193, McGraw-Hill Publications Co., New York, 1980.

References

21. J. W. Allen and J. S. Bohanick, "Development and Demonstration of Surveillance and Diagnostics of Rotating Machinery for Reducing Radiation Exposure to Nuclear Power Plant Personnel—Final Report," DOE Report DOE-ET-34002-3, April 1988.

22. J. L. Frarey et al., "An Expert System for On-Line Machinery Diagnostics," EPRI NP-3652, August 1984.

23. K. Piety and G. Mascoro, "Practical Experience Using an Automated Diagnostic System in Field Service Programs," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.†

24. I. Liddle and S. Reilly, "Diagnosing Vibration Problems with an Expert System," *Mech. Eng.*, 115(4), 54–55 (April 1993).*

25. M. Porcheron and B. Ricard, "DIAPO, an Expert System to Support Reactor Coolant Pump Monitoring and Diagnostics," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.†

26. R. Gopal and S. Kannan, "The Global Database Server and Plantwide Predictive Maintenance," EPRI 5th Predictive Maintenance Conference, Knoxville, Tenn., September 1992.†

27. K. R. Guy, "Pump Monitoring and Analysis," pp. 95–114 in *Proceedings of the Second NRC/ASME Symposium on Pump and Valve Testing, Washington, DC, July 21–23, 1992*, USNRC Conference Proceeding NUREG/CP-0123, July 1992.†

28. S. V. Bowers, K. R. Piety, and R. W. Piety, "Real-World Mounting of Accelerometers for Machinery Monitoring," *Sound and Vibration*, 25(2) (February 1991).*

29. ASME Boiler and Pressure Vessel Code, Sect. XI, Subsect. IWP, including Summer 1983 Addenda, American Society of Mechanical Engineers, 1983.‡

30. ASME/ANSI OMa-1988 Addenda to ASME/ANSI OM-1987, Operation and Maintenance of Nuclear Power Plants, Part 6, Inservice Testing of Pumps in Light-Water Reactor Power Plants.‡

31. "Code for Operation and Maintenance of Nuclear Power Plants," ASME OM Code–1990, including 1994 Addenda, American Society of Mechanical Engineers.‡

32. "Centrifugal Pumps for General Refinery Service," ANSI/API STD 610-1981, 7th Edition, American Petroleum Institute, January 1990.‡

33. J. Kott, L. Haniger, Z. Hubacek, and J. Chladek, "Systems of Operational Diagnostics on Czechoslovak Nuclear Power Stations," Skoda Works Information Center, Report ZJE-279, 1988, Plzen, Czechoslovakia.

34. F. van Nierkerk and R. Sunder, "COMOS—An On-line System for Problem-Oriented Vibration Monitoring," *Prog. Nucl. Energy* 21, 155–171 (1988).*

35. K. Naruse, S. Ogawa, Y. Watanabe, and O. Ozaki, "Off-line Rotating Machinery Diagnosis System," pp. 250–261 in *Proceedings of the ANS Nuclear Power Plant and Facility Maintenance Meeting, April 1991*, Vol. 1, June 1991.

36. S. W. Cheon et al., "Development of an Expert System for Failure Diagnosis of Primary Side Systems," *Nucl. Tech.*, 97 (1), 1–15 (January 1992).*

37. R. M. McCoy et al., "Improved Motors for Utility Applications: Volume I: Industry Assessment Study Update and Analysis." EPRI EL-4286 Vol. 1 (RP-1763-2), September 1985.

38. P. I. Nippes, "Detecting and Identifying Vibration Problems in Rotating Electrical Machinery," *Proceedings of the 16th Annual Meeting of the Vibration Institute, June 1992*, Vibration Institute, 1992.

39. M. Subudhi et al., "Improving Motor Reliability in Nuclear Power Plants," USNRC Report NUREG/CR-4939, Vol. 2, November 1987.†

40. J. R. Nicholas, "Predictive Condition Monitoring of Electric Motors," *P/PM Technology* 6(4), 28–32 (August 1993).*

41. D. E. Schump, "Motor Insulation Predictive Maintenance Testing," *Plant Engineering* 45(2), 47–49 (January 24, 1991).*

42. I. M. Culbert, H. Dhirani, and G. C. Stone, "Handbook to Assess the Insulation Condition of Large Rotating Machines," Electric Power Research Institute Power Plant Electrical Reference Series, Vol. 16 (1989).

43. "IEEE Recommended Practice for Testing Insulation Resistance of Rotating Machinery," IEEE Standard 43-1974.‡

44. H. D. Haynes, Martin Marietta Energy Systems, Inc., Oak Ridge National Laboratory, "Aging and Service Wear of Electric Motor-Operated Valves Used in Engineered Safety-Feature Systems of Nuclear Power Plants," USNRC Report NUREG/CR-4234, August 1989.†

45. R. L. Kratowicz, "Quality: The Key to a Smart Motor Purchase," *Plant Services*, p. 89–93, September 1994.*

46. G. B. Kliman and J. Stein, "Induction Motor Fault Detection via Passive Current Monitoring—a Brief Survey," *Proceedings of the 44th Meeting of the Mechanical Failures Prevention Group, April 1990*, Vibration Institute, 1990.

47. J. Reason, "Pinpoint induction-motor faults by analyzing load current," *Power* 31(10), 87–88 (October 1987).*

48. IEEE Recommended Practice for Insulation Testing of Large AC Rotating Machinery with High Direct Voltage, IEEE Standard 95-1977.‡

49. M. L. Adams and E. Makay, "Aging and Service Wear of Auxiliary Feedwater Pumps for PWR Nuclear Power Plants," USNRC Report NUREG/CR-4597, Vol. 1, July 1986.†

50. D. A. Casada, Martin Marietta Energy Systems, Inc., Oak Ridge National Laboratory, "Auxiliary Feedwater System Aging Study," USNRC Report NUREG/CR-5404, Vol. 1, March 1990.†

51. W. Greenstreet, "Low-Flow Operation and Testing of Pumps in Nuclear Plants," *Proceedings of the Symposium on Inservice Testing of Pumps and Valves, August 1–3, 1984, Washington, DC*, USNRC Conference Proceeding NUREG/CP-0111, October 1990.†

52. R. S. Hartley, "Description of Comprehensive Pump Test Change to ASME OM Code, Subsection ISTB," pp. 493–509 in *Proceedings of the Third NRC/ASME Symposium on Valve and Pump Testing, July 18–21, 1994*, July 1994.

53. S. Florjancic and A. Frei, "Dynamic Loading on Pumps—Causes for Vibrations," pp. 171–184 in *Proceedings of the Tenth International Pump Users Symposium, Houston, Texas, March 9–11, 1993*, March 1993.

54. D. P. Sloteman, P. Cooper, and J. L. Dussourd, "Control of Backflow at the Inlets of Centrifugal Pumps and Inducers," pp. 9–22 in *Proceedings of the First International Pump Symposium, Houston, Texas, May 22–24, 1984*, May 1984.

55. E. Makay, P. Cooper, D. P. Sloteman, and R. Gibson, "Investigation of Pressure Pulsations Arising from Impeller/Diffuser Interaction in a Large Centrifugal Pump," *Proceedings of the Rotating Machinery Conference and Exposition '93, Nov. 10–12, 1993, Somerset, N.J.* Vol. 1.**

56. S. S. Florjancic, A. D. Clother, and F. J. L. Chavez, "A Case History—Improved Hydraulic Design Lowers Cavitation Erosion and Vibrations of a Water Transport Pump," pp. 81–90 in *Proceedings of the Tenth International Pump Users Symposium, Houston, Texas, March 9–11, 1993*, March 1993.

57. 10CFR21 Reportability of a Potential Safety Hazard from Broken Cast Iron Diffusor Pieces in Auxiliary Feed Water Pumps, letter to R. Fuhrmeister (USNRC) from G. Young (Ingersoll-Rand), September 19, 1991.

58. W. E. Nelson and J. W. Dufour, "Pump Vibrations," pp. 137–147 in *Proceedings of the Ninth International Pump Users Symposium, Houston, Texas, March 3–5, 1992*, March 1992.

References

59. E. Makay and J. A. Barrett, "Changes in Hydraulic Component Geometries Greatly Increased Power Plant Availability and Reduced Maintenance Cost: Case Histories," pp. 85–99 in *Proceedings of the First International Pump Symposium, Houston, Texas, May 22–24, 1984*, May 1984.

---

*Available in public technical libraries.

†Available for purchase from National Technical Information Service, Springfield, VA 22161.

‡Available from American National Standards Institute, 1430 Broadway, New York, NY 10018, Copyrighted.

**Available from New Jersey Institute of Technology, University Heights, Newark, NJ 07102.

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

U.S. NUCLEAR REGULATORY COMMISSION

BIBLIOGRAPHIC DATA SHEET

(See instructions on the reverse)

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

NUREG/CR-6089
ORNL-6765

2. TITLE AND SUBTITLE

Detection of Pump Degradation

3. DATE REPORT PUBLISHED

MONTH	YEAR
August	1995

4. FIN OR GRANT NUMBER

B0828

5. AUTHOR(S)

R. H. Greene, D. A. Casada

6. TYPE OF REPORT

Technical

7. PERIOD COVERED *(Inclusive Dates)*

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

Oak Ridge National Laboratory
Oak Ridge, TN 37831-8038

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

Division of Engineering Technology
Office of Nuclear Regulatory Research
U.S. Nuclear Regulatory Commission
Washington, DC 20555-0001

10. SUPPLEMENTARY NOTES

11. ABSTRACT *(200 words or less)*

This study examines the methods of detecting pump degradation that are employed in domestic and overseas nuclear facilities. This report evaluates criteria mandated by required pump testing at U.S. nuclear plants and compares them to state-of-the-art diagnostic programs and practices implemented by other major industries. Since the working condition of the pump driver is critical to pump operability, a review of new applications of motor diagnostics is also provided that highlights developments in this technology. Vibration spectral analysis is a powerful diagnostic tool for the pump analyst. The routine collection and analysis of spectral data is superior to other technologies in its ability to accurately detect numerous types and causes of pump degradation. Existing ASME testing criteria do not require the evaluation of pump vibration spectra but instead overall vibration amplitude. The mechanical information discernible from vibration amplitude is limited, and several pump failures in the nuclear power industry were not detected by vibration amplitude monitoring. Since spectral analysis provides pertinent information concerning the mechanical condition of rotating machinery, its incorporation into ASME testing criteria may merit a relaxation in the monthly-to-quarterly testing schedules that seek to verify pump operability.

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

aging, motor current signature analysis, power fluctuations, reliability, NPAR, nuclear power plants, operating experience, MCSA, pump, vibration analysis, spectral analysis, low-flow, pump failure, rotor degradation, pump degradation, predictive maintenance, diagnostics, cavitation, motor degradation

13. AVAILABILITY STATEMENT

Unlimited

14. SECURITY CLASSIFICATION

(This Page) Unclassified

(This Report) Unclassified

15. NUMBER OF PAGES

16. PRICE

Printed on recycled paper

Federal Recycling Program

UNITED STATES
NUCLEAR REGULATORY COMMISSION
WASHINGTON, D.C. 20555-0001

OFFICIAL BUSINESS
PENALTY FOR PRIVATE USE, $300

SPECIAL FOURTH-CLASS RATE
POSTAGE AND FEES PAID
USNRC
PERMIT NO. G-67

120555139531 1 1AN1RV
US NRC-OADM
DIV FOIA & PUBLICATIONS SVCS
TPS-PDR-NUREG
2WFN-6E7
WASHINGTON DC 20555