VIBRATION MONITORING OF TURBOMACHINERY

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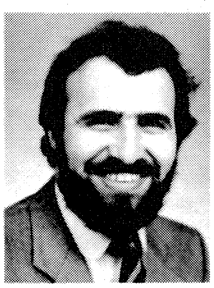

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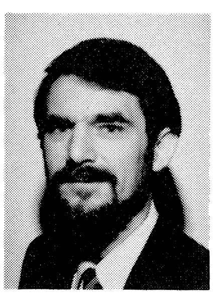

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ABSTRACT

The overall concept of vibration monitoring is presented based upon theory and the practical constraints of machinery construction, transducer characteristics and the cost/benefit ratio of available systems. General features of velocity, acceleration, and displacement measurements are considered. Machine malfunction characteristics, mechanical impedance ratio, and force versus restraints consideration are assessed with respect to selecting vibration transducers for measuring housing vibration, relative shaft vibration, and absolute shaft vibration. The construction, application, limitations, and comparison of available vibration transducers is discussed.

Typical vibration monitoring protection systems for various types of rotating machinery (turbines, compressors, pumps, fans, electrical motors and gears) will be discussed. Data acquisition, data management and analysis, cost, and general capabilities of periodic and computerized on-line vibration monitoring systems are analyzed. A presentation of vibration data in the form of overall vibration, vibration frequency spectrum, bode, waterfall and trend plots is addressed.

Description and identification of typical sources of vibration such as unbalance, misalignment, rubs, resonance, subsynchronous instabilities, and electrical problems is provided. Detection of blade related problems through the use of advanced techniques for defining vibration related machine malfunctions such as modal testing, strain gage, and radio telemetry are briefly discussed.

Vibration severity limits for measurements on both machine housings and shaft relative to the housing are presented; appropriate correction factors are introduced to accommodate different machine designs, installations, and vibration sources. The use of these limits, and examples of vibration monitoring successes, is supported by reviewing the actual field case histories. The significant concepts presented are supported by the actual case histories data.

INTRODUCTION

Modern monitoring systems require three major aspects in order to effectively protect the machinery from damaging dynamic actions. The system must bring together in an organized manner an accurate means for measuring the critical motions and forces within the machine indicative of its health; an effective means for collecting, analyzing, condensing, and organizing the signals acquired; and a realistic means for interpreting the collected, analyzed data.

Measuring the motions within turbomachinery requires a considerable extension of our natural senses. For example, if we only have X-Ray vision, we could see the rotor elements spinning inside as illustrated in figure 1. We could not, however, see the very minute motions that are sufficient to cause damage; motions which must be contained within the thickness of one page (about 4 mils) of this proceedings for most machinery!

Oldtimers often tell of balancing an "Indian Head Nickel" on edge to indicate a good running machine in years past. Although this remains a cute trick for demonstration purposes, science and technology have provided us with much better tools for properly measuring machinery vibration. Accelerometers and velocity transducers are now relatively inexpensive measurement tools for accurately sensing vibration of bearing housings and cases. Proximity probes or shaft riding velocity sensors have been readily available for 20 or more years to measure journal motions that critically affect the life of hydrodynamic bearings.

The journal excursions shown in Figure 1 are examples of the orbital motions that can be accurately sensed and recorded using a pair of proximity probes; in this case, gear box pinion orbits during a surge event of the compressor train shown in Figure 2. The pinion, spinning at about 6000 rpm, is acted upon by dynamic forces that are created when the surge momentarily unloads and then suddenly reloads the gear mesh; whirling outward the pinion journal momentarily touches the bearing. Fortunately, surge events are rarities in well controlled compressors and other forces predominate to stabilize journal rub.

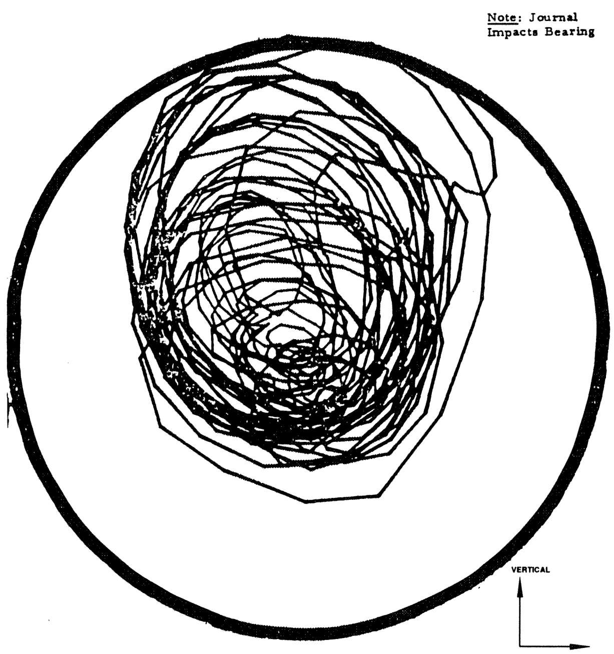

Figure 1. Pinion Lateral Response During Compressor Surge.

The recent rise in availability of microprocessors has had a profound effect on collecting and analyzing machinery vibration data for monitoring purposes. A great number of engineers now have PCs or equivalent on their desks with a variety of software tools for analyzing and presenting moderately large data bases. Monitoring systems have kept pace by developing hardware that either contains built in "PC" type processors or interfaces with the users PC. Acquisition of data varies from hand entry, to portable spectrum collectors, to dedicated transducer arrays.

The Fast Fourier Transform (FFT) is the universally adopted method for analyzing data as the frequency content of the vibration signal is most indicative of detrimental excitation sources. Significant hardware and software resources are required to

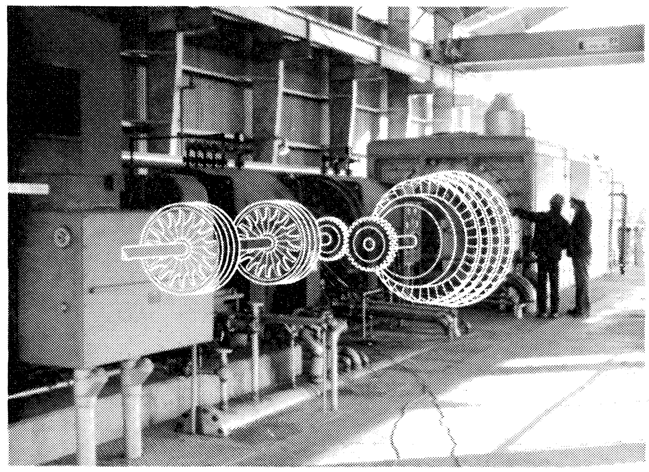

***Figure 2. Rotating Elements Inside a Typical Turbine Driven** Compressor Train.*

properly process vibration signals to assure adequate precision, freedom from errors, and rapid analyses.

Interpreting the data is almost always a matter of comparison of recent results and trends with historical data bases collected internally, with published criteria, or with expert opinion. Data bases developed, internally, on ones own machinery provides the real benefit of including the impact of plant operating and maintenance procedures and can be interpreted with plant production requirements in mind.

GENERAL VIBRATION CONCEPTS

It has become common practice to install vibration monitoring equipment on critical and costly industrial machines. The underlying objective, of course, is to detect an increase in vibration level when mechanical problems occur. Therefore, when selecting a vibration monitoring system, one of the first questions which must be answered is: "What vibration should be measured?"

- machine housing vibration,
- shaft vibration relative to bearing cap, or
- absolute shaft vibration

Each of these measurements is currently being used on a variety of industrial machines and each can claim success in instances where serious mechanical failure was averted by a warning of excessive vibration. To resolve this seemingly simple question, both the likely malfunctions of the machine and its mechanical impedance should be considered.

Machine Malfunction

Important factors determining the machine malfunction are mechanical design, machinery type, and the function the machine performs in a particular process.

Rotor related malfunctions include: mechanical or acoustic resonance, unbalance (from a multitude of causes), external preloads (e.g., misalignment between machines), internal preloads (e.g., misalignment of machine components), bearing failure, surge, cavitation, radial and axial rubs, thermally and mechanically bowed rotors, coupling lockup, gear wear, instability problems, lubrication loss, loose parts, and cracked shafts, blades, or impellers. Housing related malfunctions include bearing support failure, excessive piping forces, casing and foundation resonances, insufficient foundation support, loose parts, improper casing tiedown, foundation material failure, and thermal casing warpage.

Generally, rotor related malfunctions can be best detected by a vibration monitoring system measuring shaft vibration relative to the bearing cap with an exception of malfunctions related to high frequency vibration. On the other hand, housing and high frequency related malfunctions are best determined by a vibration monitoring system installed on the machine housing.

Mechanical Impedance

How much motion can be measured at the rotor and how much motion can be measured at the bearing housing is purely a function of the mechanical impedance characteristics of the machine. Mechanical impedance (Z) is the ratio of imposed force divided by velocity and is established by the stiffness, mass, and damping characteristics of the rotor, bearings, bearing support systems, machine casing, and foundation. There are two types of mechanical impedance which are most often associated with machine vibration: driving point impedance (Z_1) and transfer impedance (Z_2). Z_1 relates a forcing function generated by a machine rotor to absolute shaft velocity displacement, and Z_2 relates a forcing function to absolute machine housing velocity displacement, or more specifically:

$$Z_1 = F/V_{(shaft)}$$

$$Z_2 = F/V_{(housing)}$$

Where both $V_{(shaft)}$ and $V_{(housing)}$ are absolute motions of those components.

Mechanical impedance ratio T is then defined as:

$$T = Z_2/Z_1 = V_{(shaft)}/V_{(housing)}$$

Using this definition, we can say that machines with high mechanical impedance ratio will not readily transmit vibratory motion energy between shaft and housing; while machines with low mechanical impedance ratio easily transmit shaft motion to the bearing housings. Therefore, machines with high mechanical impedance ratio are usually monitored by transducers measuring relative shaft vibration, and the machines with low mechanical impedance ratio are usually monitored by transducers measuring machine housing vibration.

All machinery systems have dynamic properties which control their vibratory response during operation; chief among these is the system resonance effects. The mass, stiffness, and damping properties of the various machinery components control system resonance, as indicated by the amplification curves in Figure 3. Obviously the closer the operating excitation frequencies (ω) are to the machines natural frequencies (ω_r) the greater the amplification of excitation forces such an unbalance. Increasing the stiffness of bearings and supports or decreasing effective mass of rotating components are practical means of increasing machinery natural frequencies. Increasing system damping is an effective means of reducing amplification without significantly affecting the natural frequency.

Machines with high support stiffness and moderate stiffness journal bearings usually exhibit high damping and have a high impedance ratio. Conversely, machines with moderate support stiffness and high bearing stiffness (especially ball bearings) usually exhibit low damping and have low impedance ratio. Machines built with exceptionally flexible shafts will have very low damping resulting in very high amplification of excitation sources. Flexible shaft machines also may have nodal points near their bearings such that neither bearing mounted accelerometers nor journal sensing proximity probes adequately detect hazardous vibration levels (Figure 4).

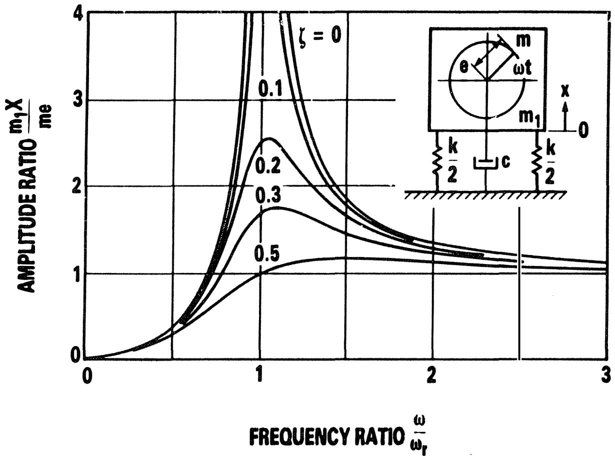

Figure 3. Effects of Resonance and Damping on Mechanical Vibration for a Simple System.

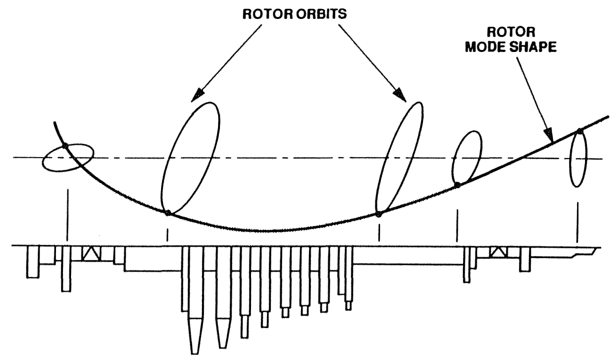

Figure 4. Additional Shaft Proximity Sensors Provide Enhanced Definition of Rotor Motion Near Shaft Critical Speeds.

A word of caution should be offered about establishing a mechanical impedance ratio (absolute shaft amplitude *vs* housing motion); the mechanical impedance ratio varies not only from one machine type to another, but can vary on machines of the same design due to the type of machine problem experienced especially if these problems are characterized by widely varying frequencies. It is also not uncommon to see a given machine undergo mechanical impedance changes due to variations in lube oil tempereature and pressure, alignment, bearing preload, and machine speed. When there is sufficient cause to believe that the machine to be monitored is in this middle ground, then installing both shaft and bearing housing sensors should be considered.

VIBRATION MONITORING SYSTEMS

Most of the vibration monitoring systems available on the market can be classified as simple vibration level monitoring for trip protection, periodic vibration monitoring, and continuous on line computerized vibration data acquisition.

The simplest vibration monitoring systems for trip protection respond to excessive levels of overall vibrations. These systems first "detect" the signal using some form of rectification and amplifying circuitry. The detected signal is then compared with setpoints for trip and alarm. Signal detection is the process of establishing the vibration level. Various circuits exists which provide the peak-to-peak, zero-to-peak, or RMS vibration level; the choice is mostly a function of convention for the sensor used.

Monitoring systems with basic trip protection are widely deployed and are essential equipment for large turbomachinery; however, the only diagnostic information considered is overall vibration level. Additional information such as spectral characteristics, phase relationships, and dependence of these features on operating parameters (speed, load, time) are often desirable for diagnostic purposes, trending, and warnings of incipient failure. This information usually requires a continuous online computerized system or, at least a periodic vibration monitoring.

The newest class of protection systems appearing in the marketplace uses digital electronics to give a level of protection, claiming to be similar to or better than the generally accepted analog circuitry, while adding the ease and control of digital programmability. In an analog protective system, most parameters such as signal detection circuit, trip and alarm levels, voting logic, delay times, etc. must be hardwired prior to installation. The benefit of digital based systems is that these parameters can be established on installation, and changed at a later time if conditions or experience warrant. With appropriate design and use of modularity and redundancy, it is usually possible to achieve the same degree of reliability presently associated with the analog protective systems. The potential for convenient integration of such digital protective systems with various types of predictive monitoring system is clear. Cost per channel of these new digital protective monitors is also reduced.

The discussion presented hereafter will guide users in selecting and assessing the capability of computerized and periodic vibration monitoring. Additional information can be found in references [1, 2, 3, 4, 5].

Computerized Online Vibration Monitoring System

The application of computers to vibration monitoring is a rapidly changing area. Computerization presents both opportunities and choices to users of turbomachinery who desire more complete and accurate information about the health of their equipment. For maintenance personnel, it presents the opportunity to change the nature of concern from reactive to anticipatory. When the normally installed basic shutdown protection system performs its function and trips the machine on high vibration, it generates questions that often require considerable action.

- What caused the trip?
- What action must we taken before we can go back on line?
- What is the impact on future operation?

A computerized monitoring system, however, when properly used will generate anticipatory questions well in advance of a trip.

- What is causing this change?
- What action can we take to avoid a trip?
- When can action be scheduled to minimize impact on operation?

There are a variety of monitoring systems now available. The choices include a number of systems which emphasize vibration together with general purpose monitoring and control, and systems which provide limited capability to handle vibration information as one of a broad range of functions. This discussion will concentrate on systems whose primary emphasis is on vibration.

Major functional elements of a computerized vibration monitoring systems, system performance, and considerations for interactions and inherent tradeoffs are issues to be addressed. Also identified are the level of capability available in system products currently in the marketplace. The full list of capabilities so identified may not be available from any single system, but the list should assist the reader to identify those features which are most important and to form a basis for evaluating systems which are available. The discussion is not a product comparison, but rather an attempt to present some of the considerations which should influence a choice between products for a particular application.

An important step in the process of selecting a computerized monitoring capability is to set functional goals for the system. Goals may be adjusted as specific products are reviewed, and budget realities are faced, but the following is offered as a useful starting point in initial goal setting. Major functional areas and associated features are shown in Table 1, along with the detailed discussion following the table.

Table 1. Overview of Computerized Vibration Monitoring Functions.

FUNCTIONS	FEATURES	
DATA ACQUISITION	• Signal Conditioning • Multiplexing • A → D Conversion • Buffering	• Precision • Sampling Rate • Speed Increments • Parallelism
DATA MANAGEMENT & ANALYSIS	• FFT • Programmability • Diagnostics • Management of Conflicts • Distributed Processing	• Frequency • Alarm Driven • Data Frozen at Trip • Multi-Tiered Storage
DATA RETRIEVAL & DISPLAY	• Spectrum • Waveform • Orbit • Bode • Polar • Comparison	• Waterfall • Trend • Bar Chart • Tabular • Operator Displays • Resonant Parameters
INTERFACES	• User Access • Protection System Interfaces • Data Collector Interface • General Purpose Data Base Interface	

Data Acquisition. Before a signal is acquired by the system, it usually requires some degree of conditioning. Common signal conditioning includes low pass filtering, high pass filtering, attenuation or amplification of the signal. To avoid aliasing errors created by signals above the frequencies of interest, low pass filtering is universally applied to cutoff signals above 0.4 times the sampling rate. Likewise, high pass filtering is used to strip off DC bias or eliminate very low frequency signal components, leaving only the time varying portion of the signal which can then be sampled with much higher resolution.

When either high or low pass filtering is used, the effects should be understood. For example, low pass filtering may have undesirable effects when the time domain signal or when details of the orbit are of interest. Similarly, high pass filtering may not be desirable if the absolute level of a signal is of interest—for example, thrust position signals, shaft radial position signals, or thermal growth signals, since the DC signal level is lost.

The number of bits in the analog to digital (A to D) converter (precision) has a significant influence on the resolution of small signals, particularly when superimposed on a large signal, and on the detection of changes in these small signals. Attenuation and amplification of the signal are techniques used to bring the maximum range of the signal within full scale for the A to D converter in order to maximize resolution. Some systems provide autoranging, which automatically attenuates or amplifies the signal to bring its range near to full scale.

The filtering situation is complicated if the user desires to record the DC level of eddy current probes with the AC component superimposed. For example, most eddy current probes have a nominal gap setting of about 50 mils. To handle this, full scale must correspond to at least 80 mils. Therefore, with a 12-bit A to D, a vibration component or change of less than 0.02 mils is lost. With an 11-bit A to D, anything less than 0.04 mils is lost and with an 8-bit A to D, anything less than 0.32 mils is lost. Because of the potential dynamic ranging problem with A to D conversion of eddy current probes, one choice is to treat the DC gap signal and the vibration signal as separate channels handled separately by the system if both items of information are of interest.

Although systems can provide up to 100 KHz sampling rate on any individual channel, 20 kHz is more widely available. In addition, most systems provide some user control over the sampling rate for each channel. Sampling rate influences the highest vibration frequency which can be meaningfully covered by a frequency spectrum (typically 0.4× sampling frequency). With a 20 kHz sampling rate, an assessment to about 130× running frequency of a 3600 RPM machine is possible. Thus, a vibration at blade passing frequency on a 70 blade compressor stage should be detectable.

An important consideration during machine startup or shutdown is the speed increment between successive data sets. The factors influencing this increment are the speed of the machine, the number of channels to be acquired, the number of parallel data acquisition channels available, settling and autoranging time after switching, and FFT processing time.

Data Management and Analysis. One of the most important functions of data management and analysis is to transform data from the time domain to the frequency domain. Almost all systems perform this operation on the digitized data using the Cooley-Tukey FFT algorithm. The function may be perf●rmed in specially designed hardware (sometimes called firmware), in a general purpose parallel processor, or in software.

To establish the time required to acquire, process and store spectra data, users must consider storage volume, speed of data acquisition and computation. For example, with 500 channels, requiring as much as six seconds per channel to switch, acquire, and analyze, the minimum time to for all channels is close to one hour if each channel is acquired sequentially; parallel processing can greatly reduce this time.

The volume of data for long term monitoring is usually conserved by increasing the interval between data sets preserved beyond the most recent 24 hours. Schemes for greatly reducing data storage volume, while maintaining equivalent information may consist of condensing the data to overall vibration, static (DC) levels, one or more spectral lines (e.g., 1×, 2×, 3× machine orders), and relevant process variables (speed, flow, etc).

The benefits of more frequent data acquisition for a machine indicating a potential problem are clear. This feature is quite widespread among available systems although differences may exist in the frequency of data acquisition after an alarm.

Some systems are capable of capturing and holding samples of data obtained for a short period of time before a trip occurred. This capability is very useful as it allows analysis of events leading to the trip. Some form of continuous acquisition and discarding (or overwriting) of data is required during normal operation so that, when the alarm sounds, the discarding operation is disabled and all of the last data is held.

Some systems include trend extrapolation which will forecast the time to alarm or trip, or can alarm on a specified rate of increase along with absolute vibratin levels. A further aid to incipient problem detection is the availability of different alert and alarm levels which correspond to different frequency bands. With this capability, the user can program the system to indicate specific problems corresponding to running speed orders (½×, 1×, 2×, etc.), blade passing frequency, etc. A technique available in some systems is a programmable diagnostic matrix into which the inhouse machinery expert can introduce likely causes of measured symptoms; causing appropriate messages to appear if these symptoms arise.

All the major systems on the market today have some degree of distributed handling and processing of the data. This may consist of a series of satellite data acquisition systems which according to some schedule acquire and store a series of sets of time series data, for subsequent transmission to a host computer and FFT analysis. The benefits of distributed processing generally include more flexibility, more throughput, and more responsiveness. Some large systems allow for data acquisition capability at remote sites, with modem transmission to a central computerized monitoring system.

Presentation of Vibration Data. Useful displays of vibration data greatly enhance a monitoring systems value for defining machinery problems quickly. Some of the options available are as follows:

Spectrum Display showing amplitude as a function of frequency (Figure 5 [6, p. 224]). Typically, this is a 400 line display, though some systems provide for fewer spectral lines with beneficial reduction in system cost.

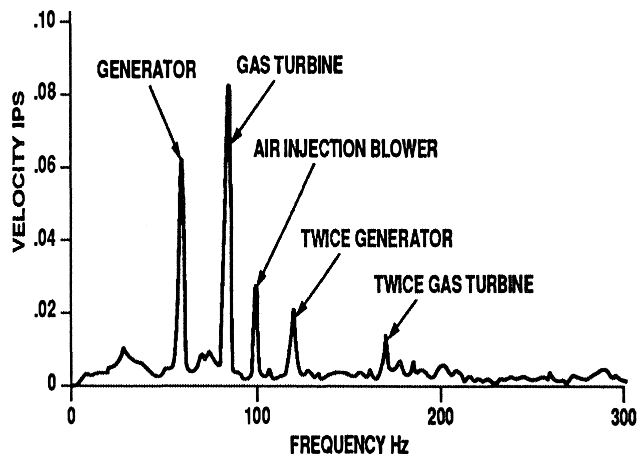

Figure 5. Spectrum Display of Machine Casing Velocity Signature [6, p. 224].

Waveform Display in which vibration is presented as a function of time (Figure 6).

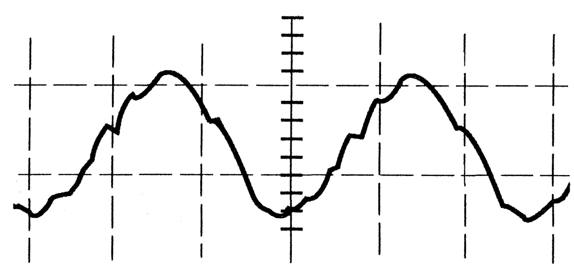

Figure 6. Waveform Display of Vibration.

To support orbit display, it is necessary to have X-Y displacement probes installed and to have the time-series data stored or recreatable. Orbits may be filtered (synchronous) or unfiltered (Figure 7). Some systems can show the change in filtered orbits during startup or coastdown.

A Bodé plot generally shows synchronous amplitude (and often phase) as a function of speed during startup or coastdown (Figure 8).

A Nyquist plot (also referred to as Polar plot) shows synchronous amplitude *vs* phase, during startup or coastdown, in polar form (Figure 9).

A waterfall plot (also referred to as raster or cascade) shows changes in vibration frequency spectra as a function of either rotor speed, load, or time (Figure 10).

Trend plots show variation in some measure of vibration as a function of time covering short, medium, or long time scales (Figure 11).

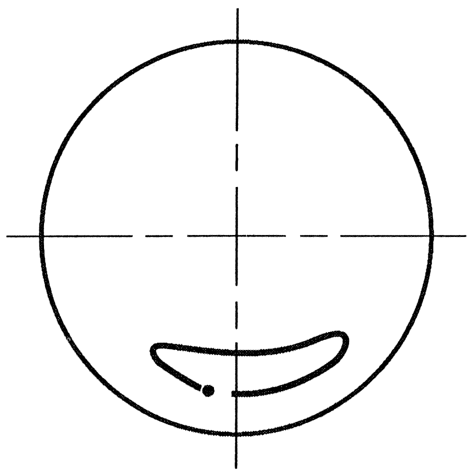

Figure 7. Unfiltered Orbit Display Typical of Misalignment.

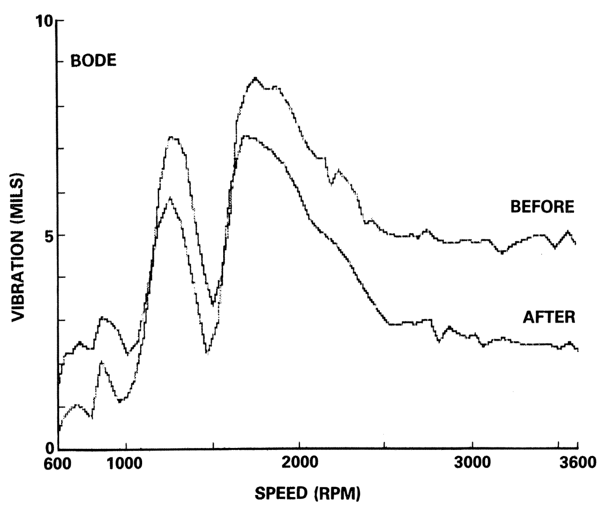

Figure 8. Bod/le Plot Showing Amplitude and Phase During Startup.

Bar charts show single vibration parameters (e.g., overall or 1× vibration) at a number of points on machine train (Figure 12).

Tabular data provides specific magnitudes of selected vibration parameters for plant or machine train.

Displays designed for operators including:

• Machine train configuration showing vibration sensor locations (Figure 13).

• Simulated analog meters which can be viewed at a glance for conformance with middle of the range nominal values (Figure 14).

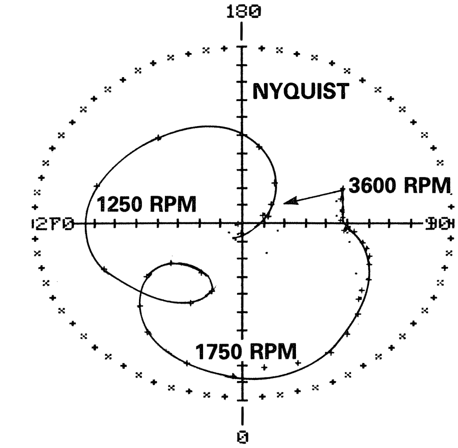

Figure 9. Polar Plot Showing Amplitude, Phase, and RPM During Startup.

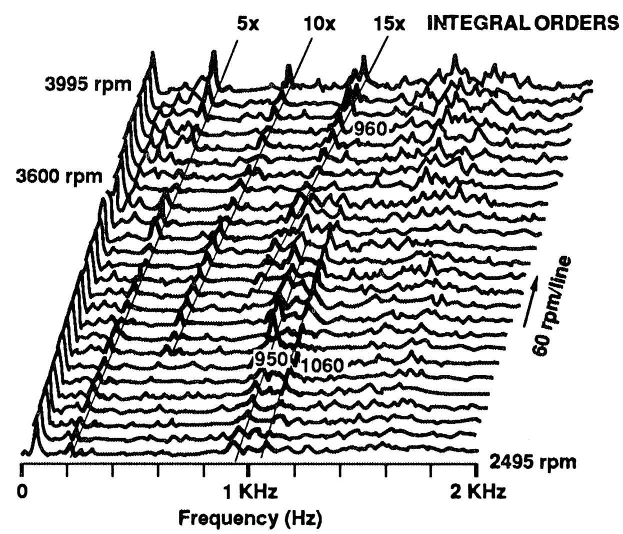

Figure 10. Waterfall Plot Used for Analysis of Blade Resonance Problem.

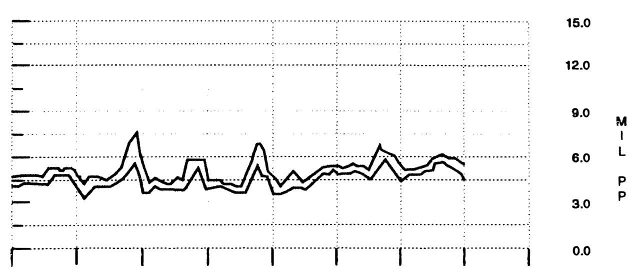

Figure 11. Trend Plot Showing Overall Vibration for One Week Period on a Shaft Rotor in Vertical (Upper) and Horizontal (Lower) Direction [2].

• Normalized bar charts for selected critical parameters where full scale is set to trip or alarm level for that channel.

Periodic Portable Vibration Monitoring

Some machinery often falls into the category for which it is not possible to justify the installation of a costly, permanently installed, computerized vibration monitoring system. In this case,

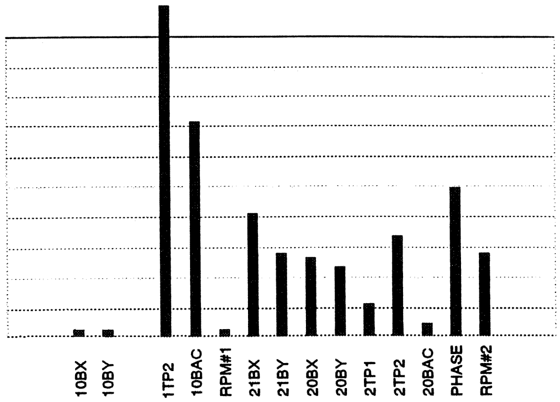

Figure 12. Bar Charts Showing Amplitudes of Various Train Vibration Parameters [4].

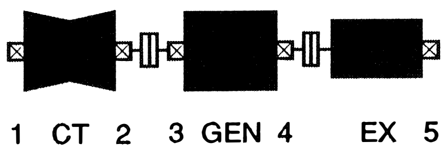

Figure 13. Machine Train Configuration.

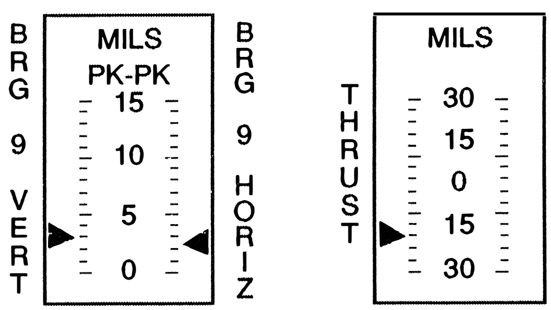

Figure 14. Simulated Analog Meters.

manual data collection using portable data collectors or electronic note pads (Figure 15) is an attractive alternative method.

The data collectors offered by several major manufacturers of vibration instrumentation equipment are becoming a practical tool in many predictive maintenance programs. One advantage

Figure 15. Typical Data Collector.

of a portable data acquisition system is that they eliminate the need for maintenance personnel to generate charts of data, thus automating the labor intensive process of monitoring.

Data collectors accept a signal from temporarily or permanently installed vibration transducers. This data is stored in the data collector and transferred into a host computer at the end of the monitoring cycle for storage, analysis, and reporting of data using predictive maintenance software. The principal purpose of a periodic vibration monitoring program is to detect changes in vibration parameters which indicate the onset of problems, and analyze vibration signals if a problem arises. Vibration levels are monitored when the machine is in good condition to provide a benchmark against future changes.

Some systems are designed to detect the general condition of a machine (overall vibration, trending), while others can offer more comprehensive information for diagnosis of a particular vibration-related machine fault (orbit display, frequency spectrum).

Portable data acquisition systems are undergoing continuous, rapid, development. Therefore, in the future, the user would face an increasing amount of choices available to him. Some current features which are important in selecting a data collector and associated software used with a host computer are presented forthwith.

All collectors provide overall vibration readings often required for detection of a gross machine malfunction. More advanced collectors can average overall vibration levels obtained on the machine, thus reducing the possibility of errors associated with unsteady effects influencing the quality of the vibration signal.

Some collectors can show the strength of vibration signal *vs* frequency and even provide averaged spectrum information, which are important features for establishing the source of vibration problems.

Simultaneous recording of two vibration signals and phase is a useful feature for displaying orbits and performing machine balancing. This feature is especially useful if the machine is equipped with X-Y eddy current probes or dual probes measuring shaft and bearing housing displacement simultaneously.

Some data collectors can accept input from various transducer types, for example, eddy current probes measuring shaft vibration, and accelerometers measuring bearing housing vibration. This feature is especially important if a plant has machines equipped with different transducer types.

Other features which should be considered for evaluation of data collectors include: the amount of data which can be stored before downloading to a host computer (especially important for complex systems with several hundred entry points), screen visibility, ease of entering data, weight, size, and operating temperature limits.

Host computer and associated software is the key to successful implementation of vibration monitoring programs, especially if a large amount of equipment is covered under the program. It organizes and automates the gathering, analyzing, storing, and reporting of critical vibration data.

Currently, almost all software available on the market can plot trends. More comprehensive software can compare vibrations at particular frequencies or produce waterfall display for viewing historical spectral trends, which is an important feature for detecting and analyzing a particular machine fault. Some also permit direct comparison of trends between several locations, thus allowing for comparison of vibration levels on similar machines.

All software have some capabilities to set machine points and routes, which may include route repositioning, recalling, deleting or adding selected points into the route, as well as route schedule, instructions, and statistics.

Cost of Vibration Monitoring Programs

To justify a vibration predictive maintenance program for machinery, it is necessary to demonstrate that the potential savings of the program offset the cost of the program.

The potential savings of the vibration monitoring program include: prevention of a catastrophic failure, reduced maintenance and downtime cost due to early problem identification, severity assessment of a vibration-related problem, orderly backtracking after a trip or failure, and reduced insurance costs.

The costs associated with the program include: purchase price and installation of the vibration monitoring equipment; labor and material cost for the vibration system maintenance and data handling; lost production due to nuisance machine shutdowns attributable to vibration monitoring equipment malfunctions.

It should be noted that the validity of the numbers associated with the cost/benefits analysis of vibration monitoring program is often questioned due to variability of the assumptions in their computation procedure. Some companies rely on several scenarios to establish the range of potential savings/loss associated with the implementation of a vibration monitoring system. In general, the level of vibration monitoring protection needed will depend on the type and severity of the service, criticality of equipment, number of units at one site, availability and qualification of maintenance personnel, and machinery type and age.

Some typical equipment costs are shown in Table 2 of vibration monitoring systems, broken into three categories: basic vibration monitoring for trip protection, periodic vibration monitoring, and on-line computerized continuous vibration monitoring.

Table 2. Typical Vibration Monitoring System Costs.

TYPE OF MONITORING SYSTEM	COST	COMMENTS
Basic Vibration Monitoring for Trip Protection	\$600 to \$900 per channel	Continuous analog protection, based on overall vibration level. Sensor cost not included.
Periodic Vibration Monitoring	\$20,000 to \$25,000 for a single data collector and a PC based data management system	Vibration spectrum data recorded at regular intervals-typically monthly comparison with baseline trends of data.
On-Line Computerized Continuous Vibra f Monitoring	40 channel system: \$1500 to \$2000 per channel 500 channel system: \$300 to \$700 per channel	Regular storage of spectral information and trending; variety of displays including orbits, Nyquist, Bode, Waterfall. Cost does not include trip protection, monitor, and sensors.

The costs represent recent estimates and are presented as guidelines; accurate costs can only be developed when quotations are obtained from a supplier for a specific application.

VIBRATION TRANSDUCERS

The installation of vibration measuring devices has become common practice on critical and costly industrial machines. The underlying objective is to detect an increase in vibration level before mechanical problems occur. To achieve this, it is necessary to select a transducer type that will measure the vibration (machine housing or shaft) most likely to reveal the expected failure characteristics [7, 8]. Shaft measurements are recommended for rotor related malfunctions which include: imbalance, misalignment, shaft bow, and fluid film bearing instability. Velocity pickups or accelerometers, installed on machine casings or bearing housings, are to detect housing high frequency rotor related malfunctions which include: ball bearing fatigue, support looseness, casing or foundation resonance, loose parts, and blade or gear problems (Figure 16).

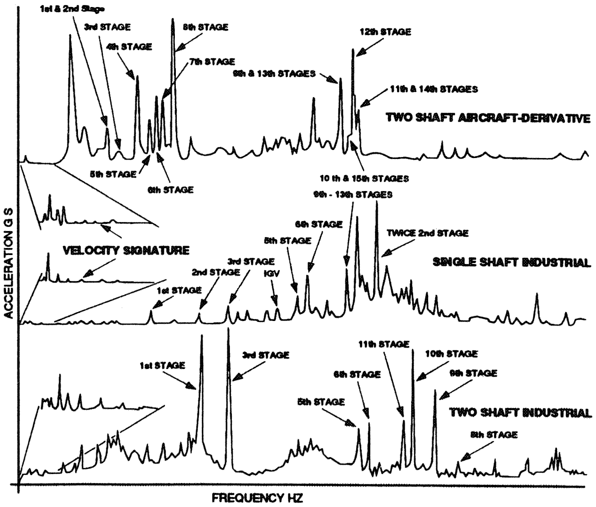

Figure 16. Casing Acceleration and Velocity Vibrations Signatures Recorded on the Compressor Section of Three Gas Turbines [6, p. 145].

Since every transducer has its merits along with its limitations, no single type can be relied upon to meet all application requirements. Therefore, many critical installations require more than one transducer type for complete protection. A summary of advantages, disadvantages and application of the various transducer types is presented in Table 3, followed by a specific description of each transducer type and its application.

Eddy Current Probes—Measurements of Shaft Relative Vibration and Position

Keyphase measurements and measurements of shaft vibration or shaft position relative to the bearing clearance and axial shaft movement, are normally made using eddy current probes. For shaft vibration and position measurements the transducer is installed in a threaded hole in the bearing (Figure 17) or on a

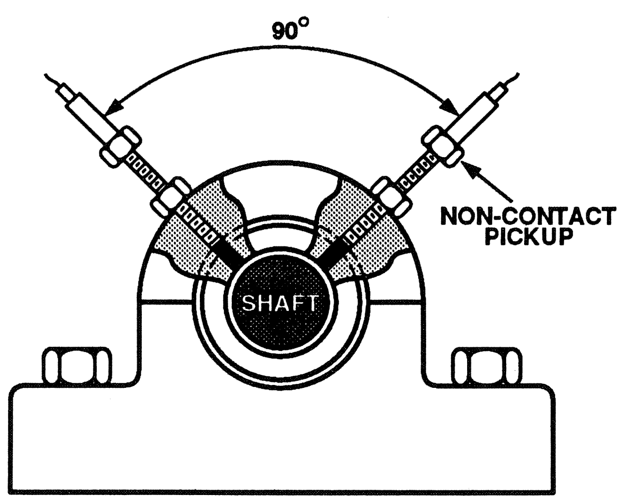

Figure 17. X-Y Eddy Current Probe Installation.

Table 3. Comparison of Vibration Transducers.

TRANSDUCER TYPE	ADVANTAGES	INSTALLATION	DISADVANTAGES
EDDY CURRENT NONCONTACT DISPLACEMENT PROBES	•Can measure static and dynamic shaft motion. •Accurate low frequency response. •No wear. •Small size. •Measures directly in displacement units. •Easy to calibrate.	•Mechanical or electrical runout noise. •Limited high frequency sensitivity. •Calibration sensitive to shaft materials. •Difficult to install or replace. •Requires external power source.	Popular on machines with fluid film bearings. Usually one pair of X-Y probes 90° apart per bearing.
VELOCITY PICK-UPS	•Simple to install with space available. •No external power source. •Strong response in the mid-frequency range. •Can be installed on a temporary basis.	•Transducer resonance noise and phase shift. •Cross-axis noise. •Can be affected by magnetic fields. •Performance degradation due to wear. •Difficult to calibrate.	Typical on older machines. Mounted on machine casing or bearing housing.
ACCELER-OMETERS	•Simple to install. •Good high frequency response. •Some models suitable for high temperature environment. •Small size. •Relatively good reliability.	•A remote charge amplifier required for high temperature environment (approximately above 260° F). •Possibility of low frequency noise due to: integration to displacement, loose connections, etc.	Typical on aircraft derivative gas turbines, fans, axial compressors, and small pumps. Sometimes added to or replaced older velocity pick-ups. Occasionally added to existing vibration probes for improved diagnostics.
DUAL PROBES	•Combined advantages of eddy current probes and velocity pick-ups or accelerometers. •Measures shaft and housing vibration separately, and can measure absolute shaft displacement.	•Combined disadvantages of eddy current probes and velocity pick-ups or accelerometers.	Common on some large industrial gas turbines. Usually one probe per bearing, installed in vertical direction. Sometimes two probes 90° apart per bearing.

rigid bracket so that the tip of the pickup can be brought close to the shaft. The eddy current probe assembly consists of a transducer, extension cable, connector, and oscillator-demodulator (Figure 18). An external oscillator provides a high frequency carrier signal to a coil in the pickup tip, producing a magnetic field radiating from the tip of the displacement probe on to the shaft. The shaft of the machine creates an impedance to carrier signal amplitude, and this impedance varies inversely with the gap between the shaft and pickup tip. As the shaft approaches the probe tip the magnetic field strength reduces due to generation of the currents on the shaft with the carrier signal amplitude modulated proportionally to the shaft displacement. Additional normalization of the carrier signal adjusts its output to a specific sensitivity usually 100 or 200 mv/mil. The typical linear probe range extends from 10 mils to 80 mils gap. The signal DC component represents the average gap between the probe tip and the

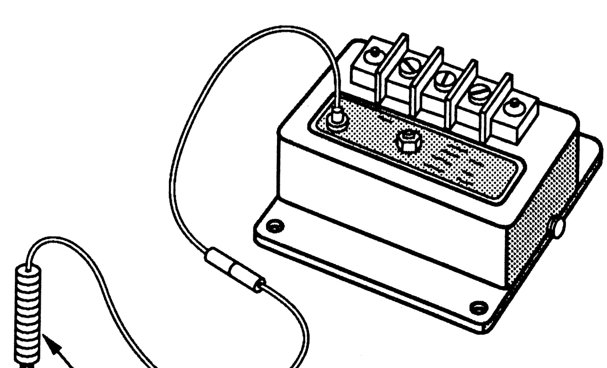

Figure 18. Typical Eddy Current Assembly.

shaft, and its AC component represents the dynamic shaft displacement (vibration). Shaft material, changes in cable length, and sometimes temperature, influence the probe sensitivity of voltage *vs* gap.

To monitor shaft vibration, ideally, two eddy current probes are installed 90 degrees apart on each gas turbine bearing. The two probes together can provide valuable information on vibration amplitude levels, shaft orbit, and shaft position. Commonly used noncontact high temperature probes are rated up to 350°F (180°C). For higher temperatures, specific probe environmental limits should be reviewed. The typical probe tip diameter is either 0.190 in (5 mm) or 0.300 in (8 mm). By adding two X-Y redundant probes at locations which are difficult to access, and operate in a harsh environment, the reliability of the monitoring system can be enhanced.

Knowledge of the vibration phase angle enhances diagnosis of a number of vibration problems. An additional eddy current transducer observing a once per revolution event, such as a notch on the shaft, reliably provides the needed phase reference. For temporary installations, a piece of reflective tape, attached to the shaft surface, observed by an optic sensor, can be sufficient for a phase angle measurement.

One of the important parameters obtained with eddy current probes is a steady state position of the rotor with respect to the bearing clearance. However, this type of measurement should be viewed with caution when applied to high temperature environment. Even high temperature probes, which minimize the influence of temperature on voltage *vs* displacement curves, can introduce some measurement errors. If the probe gap is set initially with shaft stationary at ambient temperature, the effect of increasing temperature on these probes makes the apparent gap less than the true gap. For probe and driver systems which have been optimized for high temperature operation the uncertainty in gap introduced by increasing temperature can be limited to 1.0 to 2.0 mils at 200°F, and 2.0 to 5.0 mils at 350°F. These temperature optimized probes tend to be of larger diameter (0.300 in or 8 mm as opposed to 0.190 in or 5.0 mm). Available calibration data indicates that the uncertainty can reach 10 mils at 200°F if the probes installed are not optimized in high temperature operation. Because of this temperature sensitivity shaft absolute position signals must be interpreted with considerable care and with awareness of the effect of temperature on the DC signal of the probe in question. To minimize the uncertainty, calibration curves should be obtained for the specific probes to be installed at different temperatures encompassing the anticipated range; the bearing metal and oil temperature should be measured; and the effect of temperature on both level and slope of the curves should be considered. As an example of the uncertainty introduced by temperature sensitivity, under normal loading conditions, the shaft can appear to be running in the top half of the bearing. At the same time, properly calibrated and interpreted probes can give reliable information about shaft position.

The most common source of noise found in eddy current probes is shaft electrical or mechanical runout. Often runout errors are created when a journal is flame sprayed or plated to rebuild worn surfaces. To prevent the probe signals from becoming useless after such an operation, the shaft surface sensed by the probe should be properly masked. Surface grinding and rolling or shot peening are treatments that improve uncoated shafts. Alternately, special analysis equipment can subtract the runout signal. A runout subtractor first records the phase and the shaft displacement at slow roll when all motion is presumed to be due to runout and then vectorially subtracts this displacement from the displacement readings obtained during machine operation.

Accelerometers—Measurements of Housing Vibration

An accelerometer is an electromechanical transducer which produces voltage proportional to the acceleration to which it is subjected. Among many different types of accelerometers, the piezoelectric ones are the most common in turbomachinery application. The piezoelectric accelerometers can be divided into two major groups: compression and shear types (Figure 19 and Figure 20).

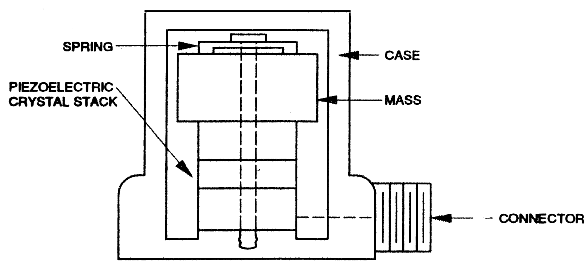

Figure 19. Compression Type Piezoelectric Accelerometer [6, p. 37].

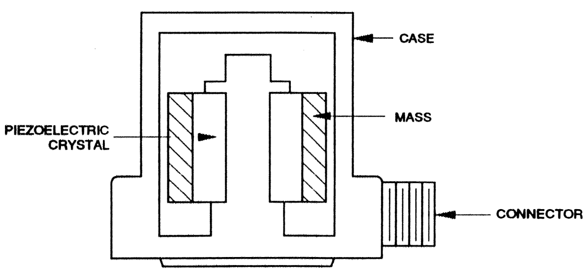

Figure 20. Shear Type Piezoelectric Accelerometer [6, p. 41].

A typical piezoelectric transducer consists of one or several piezoelectric crystal elements which are loaded by a mass or masses and preloaded with a spring. The piezoelectric crystal produce an electrical charge proportional to the applied force when loaded in compression (compression type accelerometers) or shear (shear type accelerometers). Converting a piezoelectric crystal generated signal to a signal suited for use with conventional vibration monitoring devices requires internal or external electronics. An accelerometer with built-in electronics can only withstand temperatures up to 260°F (125°C). Therefore, remotely located charged amplifiers are required for high temperature accelerometers which are rated up to 1380°F (750°C). The signal strength amplitude of a piezoelectric accelerometer is affected by changes inn temperature. This becomes especially important when the accelerometer is subject to varying operating conditions or temperature gradient across the accelerometer surface; gas flowing over the accelerometer can be of significantly different temperature from the temperature of the mounting surface. With a remote charge amplifier, the connecting cable can be a source of additional noise due to loose connections, ground loops, triboelectric noise (usually used by separation of the conductor and shield due to vibration,) and electromagnetic noise due to nearby electric fields.

Some special considerations also apply in mounting accelerometers to the vibrating surface. Accelerometers are usually mounted on a flat, smooth surface by a threaded stud (Figure 21). This mounting technique offers significant reliability and repeatability than alternative mounting techniques such as epoxy or magnet attachments.

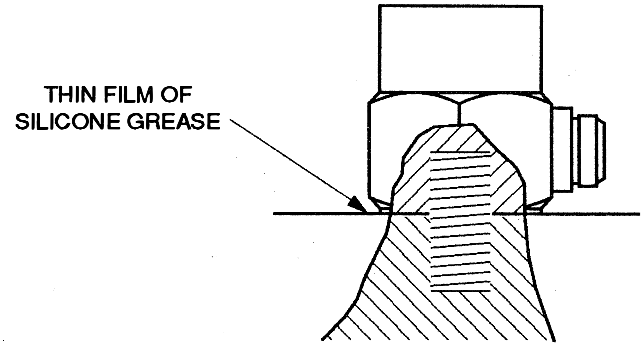

Figure 21. Installation of a Thread Mounted Accelerometer.

Velocity Transducers—Measurements of Housing Vibration

Seismic velocity transducers often measure bearing housings or machine casing vibration. A typical velocity pickup is shown schematically in Figure 22 [6, p. 35]. A cylindrical coil in the pickup is attached to a permanent magnet suspended on springs. The signal is generated by the motion of the coil in the magnetic field of the magnet. The spring suspension system is designed to have a very low natural frequency so that the magnet remains stationary in pace at frequencies above 8 to 10 Hz. A damping medium, typically a synthetic oil, is generally added to damp critically the natural frequency of the spring mass system and roll off its response characteristics below approximately 10 Hz. For permanent monitoring the transducer is usually bolted or stud mounted directly to the bearing housing or the machine casing. For temporary monitoring or probing surveys, the pickups can be hand held or magnetically secured on the machine casing.

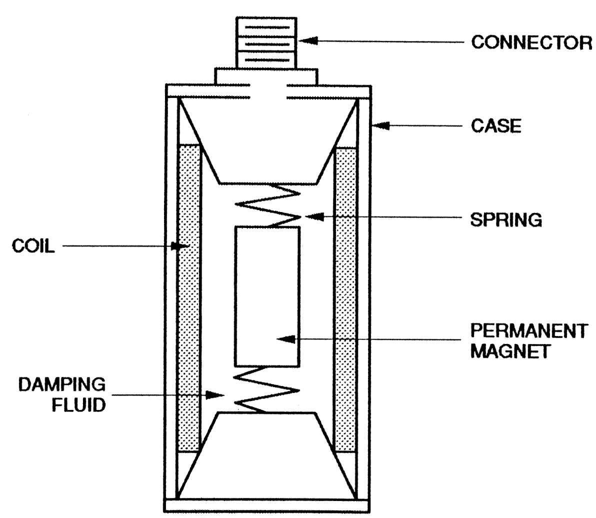

Figure 22. Typical Velocity Pickup [6, p. 35].

Although some velocity pickups are available with linear amplitude response from 1 to 5000 Hz, velocity pickups are generally used to measure vibration displacement at low and intermediate vibration frequencies, typically 10 to 1500 Hz.

High temperature velocity pickups are usually rated up to 600°F to 700°F (315°C to 400°C), which is generally adequate for most installations. Some velocity pickups have a limited longevity and often require replacement after one to two years of operation.

Dual Probes—Measurements of Absolute and Relative Shaft Vibrations

Some large industrial rotating machinery installations have a dual probe (Figure 23) which combines an eddy current probe and a seismic transducer (usually a velocity pickup). With this combination, signals are available for detecting relative shaft vibration, absolute shaft vibration, or absolute bearing housing vibration. Since the signal from an eddy current probe is proportional to dynamic shaft displacement relative to the bearing and the signal from a seismic probe signal is integrated to the appropriate displacement units and added to the shaft relative displacement signal to obtain shaft absolute displacement.

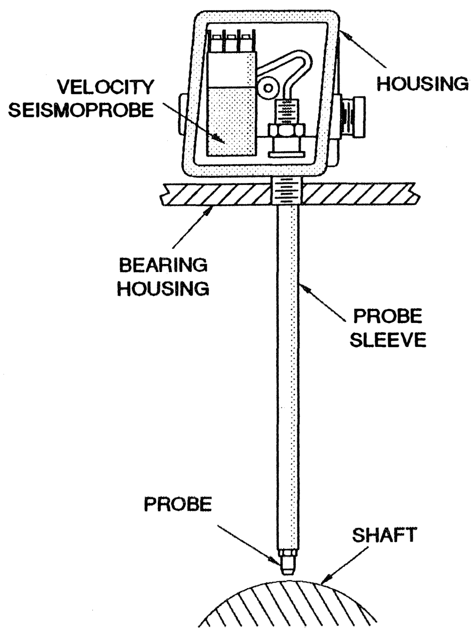

Figure 23. Typical Dual Probe [2].

Although the use of a single vertical dual probe is quite common two dual probes on each bearing 90 degrees apart are desirable (Figure 24). Some companies rely on the measurements of absolute shaft vibration, i.e., vibration readings from the eddy current probe added to the vibration reading from the seismic probe. Others use the relative shaft displacement data for detecting a vibration problem and utilize the vibration information from the seismic probes to assist in identifying the root cause of the problem.

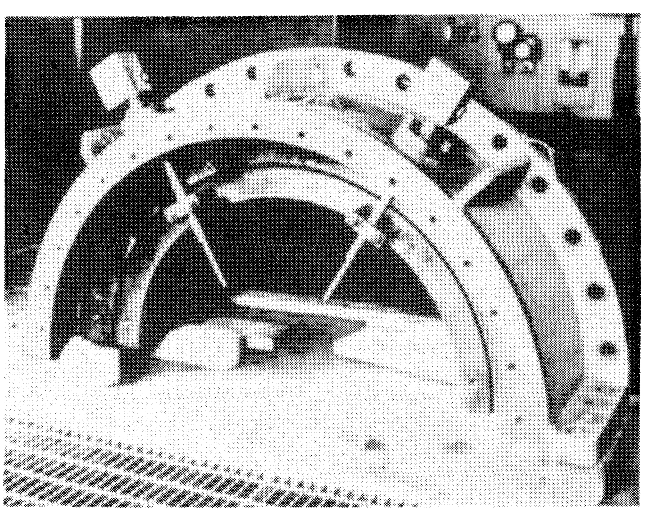

Figure 24. X-Y Dual Probe Installation [2].

ADVANCED METHODS FOR DETECTING BLADE VIBRATION AND TIP CLEARANCE

Several independent industry surveys show that blade related problems are the leading cause of turbomachinery equipment failures. The consequences of a lost blade are usually severe, and can cause complete destruction of the unit. Several techniques are now emerging which show promise for diagnosis of blade failure causes and for detecting incipient blade failure. The promising techniques include strain gage telemetry to monitor dynamic strain, accelerometers to detect the proximity of blade resonances to excitation frequencies, impact tests to define static blade natural frequencies, and noncontact sensors (optical, eddy current, capacitive, magnetic, and acoustic-doppler) to monitor, tip clearance, blade position, and time of arrival. Some of these techniques are discussed later in more detail.

Strain Gage Measurement

Advanced strain gage technology have resulted in several experiments to measure cyclic stress of blades and impellers. Slip rings and telemetry systems may serve as the link between the strain gage and the data analysis system. Slip rings have relatively short life span and are prone to signal noise during rotation. In telemetry systems, signals from the sensors are encoded onto a radio frequency carrier by transmitters. The transmitted signals are received by a stationary antenna and receiver, which decodes the signal. Strain gages are advertised to survive temperatures up to 2100°F and centrifugal force fields in excess of 20,000 gs. Installation of strain gages is a laborious process requiring considerable preparation such as surface degreasing, bead blasting, and the use of special bonding compounds.

Laser-Optical Blade Tip Clearance Measurement System

Laser-optics systems offer the means to make precision dimensional measurements of rapidly moving rotating objects. One of the laser-optics system designs currently employed by one of the jet engine manufacturers uses a laser generated light source, which is focused on a single optical fiber and is imaged on the target blade tip through a lens and sapphire prism (Figure 25). The reflected light returns through the prism and lens and is focussed on the output coherent fiber optic bundle. With air cooling the probe can function up to 2000°F and has been used in jet engine tests for measuring average blade tip clearance. With appropriate signal processing, this probe appears capable of measuring not only rotor motion and individual blade tip clearances, but blade vibration and rotor torsional vibration. Typical accuracy of this probe is 0.003 in or less. The probe usually provides from 20 to 50 hours of continuous operation before being removed for cleaning or other repairs.

Capacitance and Eddy Current Proximity Sensor Tip Clearance Measurements

Capacitance proximity sensors determine the distance between the sensor and the target (blade tip) by measuring the capacitance reactance of the sensor/target system. A typical capacitance sensor assembly consists of a sensor surrounded by an insulated cylindrical guard (Figure 26). The guard ring is connected to the shield and the sensor is wired to the coaxial cable electrode. Capacitance sensors are linear over a range of 0.4 sensor diameters and are accurate to 0.003 in for typical applications. Capacitance probes have been reportedly used in tem-

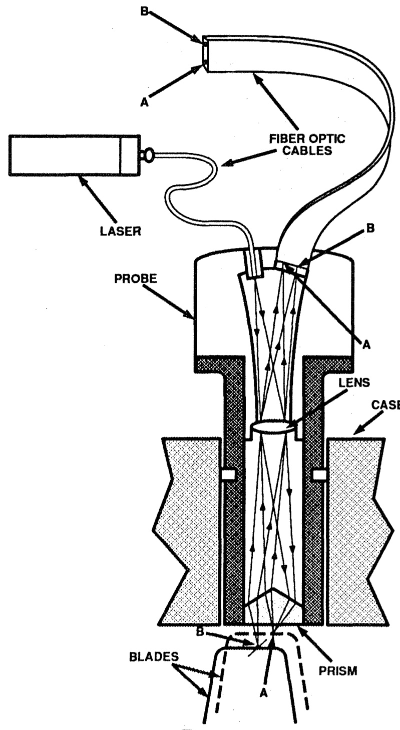

Figure 25. Laser Optical Sensor System.

perature environments in excess of 2000°F without external cooling. Capacitance probes calibration has been typically temperature sensitive and the probe reliability has been hampered by the presence of moisture and other contaminate, but recent improvements have resulted in significant reductions to these problems.

Manufacturers of eddy current probes advertise recent advancements that allow uncooled operation to 1100°F with accuracy of 0.002 in. Inductive interference requires that the turbine casing be pierced by a hole at least six times the desired measurement range.

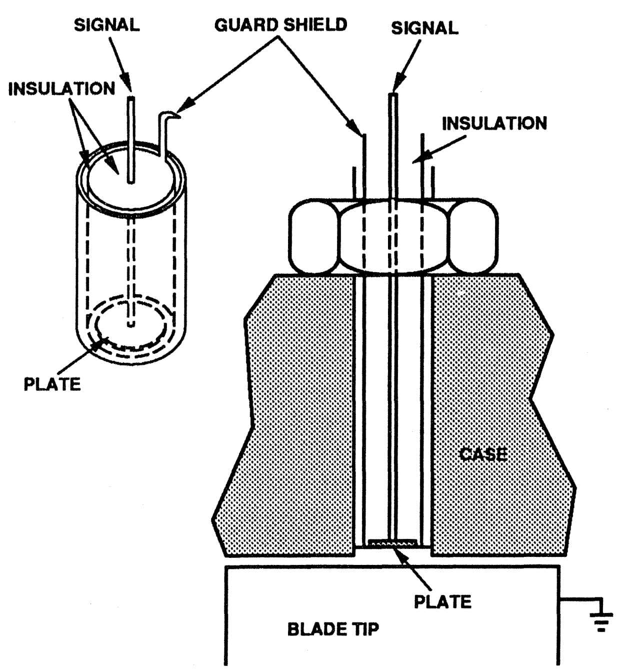

Figure 26. Capacitance Proximity Probe Installation.

Both capacitance and eddy current probes have degraded range capabilities for thinly tipped blades. Special tuned electronics are required to provide measurements at high blade passage frequencies. There is also a need for enhanced signal processing software to expand the ability of the probes to monitor individual blade clearance and vibration.

MORE COMPREHENSIVE VIBRATION LIMITS AND TEST CASES

Due to the large variation in machine designs, vibration limits are based on statistical or consensus evaluations of vibration and failure data from many operating units. As such, there are no absolute limits that will assure successful longterm operation nor firm upper limit that will cause failure for any specific machine. The best to be hoped for is to increase the probability of successful long-term operation and minimize unscheduled outages due to equipment failure. Using statistical criteria, there will always be unexpected failures and amazing vibration endurance of some machines. In general, high vibrations are bad, low vibrations are good, and the line separating the two is somewhat uncertain. The authors know of several plants where a vibration reduction program resulted in considerable savings due to reduced machine outages. Thus, lowering the vibration to reasonably obtainable levels should be encouraged.

Vibration in rotating machinery may be the result of several different phenomena and affect various machine parts. Therefore, reliable vibration measurements and their assessment should be based on each particular machine type, its installation, vibration sources, and failures that are likely to occur.

Most vibration failures can be classified according to the following consequences:

- Structural fracture due to fatigue or dynamic overload.

- Wear, fretting, or surface fatigue of bearings, gears, couplings, etc.

- Performance loss due to internal machine clearance rubs (seals, blades, impellers).

An important factor to recognize in applying vibration limits is growth of vibration amplitudes with time. All machines, regardless of their health, exhibit some form of vibration; however, on machines where a mechanical defect has deteriorated to an unacceptable level, rapid growth in vibration and consequent failure can be expected. In addition to vibration severity criteria, machine resonance characteristics should be controlled. The margin between the rotor bearing resonance (critical speed) of a machine and its operating speed range has a very strong influence on the resultant vibration sensitivity to unbalance. Only a slight change in this margin can have a dramatic effect on vibration level. This is also true for such sources as blade resonance problems on fans and turbines.

During the past several years, there have been many vibration severity guidelines proposed by standards organizations [9, 10], technical societies [11, 12, 13, 14], and equipment manufacturers [15, 16, 17], along with experienced individuals [18]. Using these standards as a base and adding the experience accumulated by Southwest Research Institute with various types of rotating equipment, vibration limits have been established for machine housing vibration (Figure 27) and for shaft vibration relative to bearing housing (Figures 28, 29). These vibration limits cover a wide range of equipment, installation types, and machine malfunction characteristics.

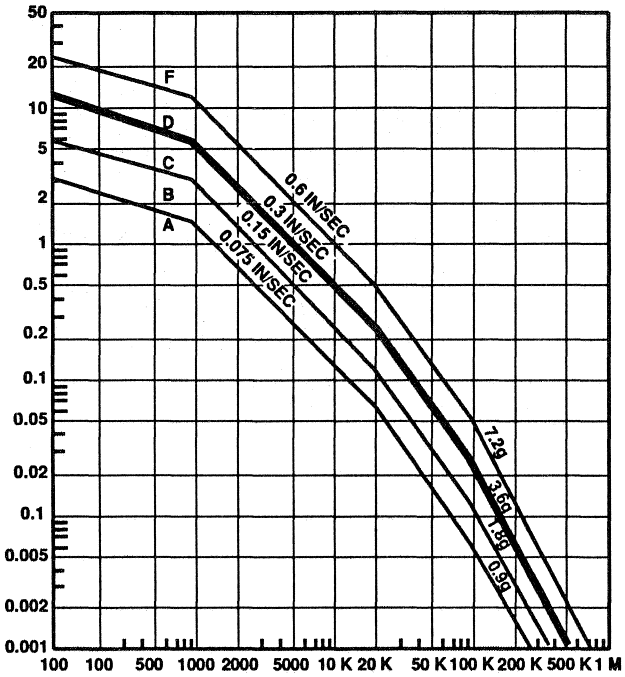

Figure 27. Housing Vibration Limits.

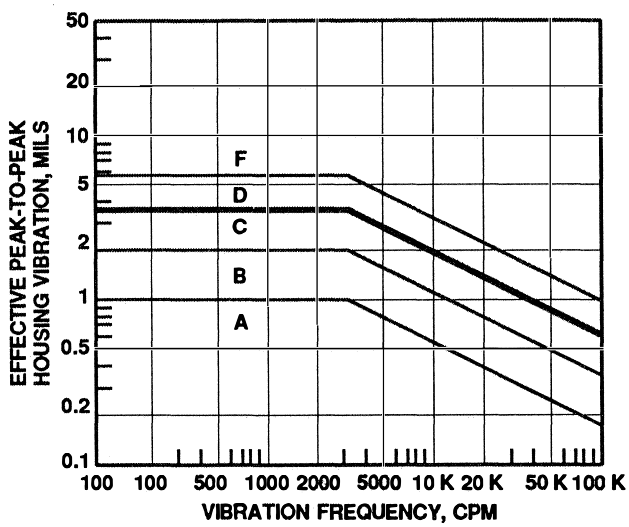

Figure 28. Shaft Relative Motion Vibration Limits.

Of course, any vibration criteria utilized should be compared with the manufacturer's recommendations and user's experience with the particular type of machinery. Obviously, the manufacturer's vibration limits should be used, if they are more stringent and have a clear basis.

The housing vibration chart (Figure 27) corresponds to vibration, which is measured on the machine casing or bearing housing utilizing velocity pickups or accelerometers. For more convenient analysis of vibration problems, this chart is divided into regions where velocity or acceleration measurements are most appropriate.

It is important to realize that some subjective judgement is required in conjunction with these charts. To equitably accommodate different machine designs, installations, and vibration problems, the charts for housing vibration and shaft relative vibration should be used with a table of appropriate correction factors (Table 4).

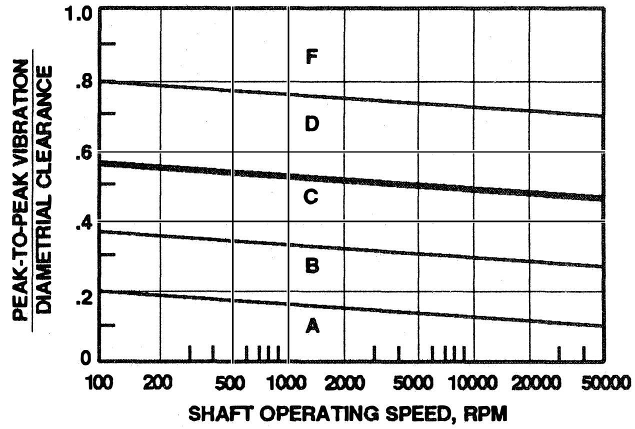

Figure 29. Shaft Vibration Limits Related to Bearing Clearance.

Relative shaft vibration is measured by proximity probes. Two vibration severity charts are provided for relative shaft vibration. The first chart (Figure 28) utilizes only actual relative shaft displacement and requires usage of appropriate correction factors. The second chart (Figure 29) establishes journal severity limits with respect to bearing clearance. The probability of bearing metal-to-metal contact and unacceptably high bearing load increases with the ratio of maximum journal motion to bearing clearance. It should be noted that this chart does not require the application of any correction factors.

Table 4. Vibration Service Factors.

SERVICE FACTORS	APPLICATION
K_1 = 1.0	NORMAL FACTOR FOR FILTERED VIBRATION NEAR RUNNING SPEED
K_2 = 0.85	UNFILTERED VIBRATION FOR NEAR RUNNING SPEED SOURCES
K_3 = 3.3	ANY SELF-EXCITED OR UNSTABLE VIBRATION SOURCE
K_4 = 1.4	EQUIPMENT RATED LESS THAN 300 HP
K_5 = 0.7	RIGID ROTOR MACHINES
K_6 = 0.6	SOFT FOUNDATION MACHINES
K_7 = 3.5-10	HIGH CASING-TO-ROTOR MACHINES
K_8 = 3.5	HIGH FREQUENCY BLADE VIBRATION SOURCES
K_9 = 0.35	HIGH FREQUENCY GEAR OR ROLLER BEARING SOURCES

Five quality grades for measured vibration are defined for these charts:

A No faults (typical new equipment).

B Acceptable (correction is not necessary).

C Marginal (correction is recommended to save on future maintenance).

D Failure probable (watch closely for changes and prepare to shut down or change operating conditions to reduce vibration).

F Danger of immediate failure.

Some important points should be emphasized to assure that these vibration charts (Figures 27, 28) are properly applied when machine vibration measurements are evaluated:

• Vibration limits for machine housing vibration should be considered separately from relative shaft motion.

• An effective vibration value must be obtained by multiplying the measured vibration value by one or more applicable correction factors listed in Table 4. Correction factors that apply to housing vibration include: rotor rigidity, foundation type, power rating, casing to rotor weight ratio, high frequency vibration, instabilities, and unfiltered vibration. Shaft vibration service factors differentiate running speed vibration from rotor instabilities, power rating, and unfiltered data.

• If vibration frequencies below running speed are apparent, then one must determine if the source is an instability or forced excitation. If it is an instability, then the factor of 3.3 must be applied to the measured data and entered in the chart at the running speed frequency.

• The severity criteria based on unfiltered vibration readings is only applicable for excitations near running speed frequency. In this case, an additional correction factor of 0.85 is applied and the resultant effective values entered on the charts at shaft rotational speed.

• To convert filtered vibration displacement in mils to velocity in /sec, and acceleration in "g," the following formulas should be used:

$$V\text{ peak} = 5.23 \times 10^{-5}\,\text{Dp-p} \times N$$

$$A\text{ peak} = 1.41 \times 10^{-8}\,\text{Dp-p} \times N^2$$

where Dp-p - peak-to-peak displacement measurement, mils
V peak - 0 to peak, velocity measurement, in/sec
A peak - 0 to peak, acceleration measurement, g
N - frequency, cpm

The application of these vibration severity criteria are clarified by reviewing the results obtained from field studies of actual plant equipment.

• The first example involved a troubleshooting investigating and balancing of two boiler feed pump turbines operating at 4000 to 5800 rpm that had experienced bearing failure. Turbine rotor resonances were found at 2200 and 5800 rpm. For one turbine, the apparent filtered shaft relative vibration was about 3.5 mils near top speed, but further investigation revealed that 1.4 mils of the signal was due to shaft runout. For the remaining 2.1 mils of actual shaft vibration, a correction factor (K_1) of 1 would apply. Comparison of this effective vibration with Figure 28 indicates the level was marginal.

On the adjacent turbine, a filtered bearing housing vibration of 3.1 mils was measured at maximum running speed. The structure was considered soft mounted since a turbine pedestal resonance of 3400 rpm was present below running speed. Applying a correction factor (K_6) of 0.6 yields an effective vibration of 1.8 mils. Comparison of this effective vibration with Figure 27 indicates that failure is probable, which was consistent with experience on the turbines.

Balancing reduced vibrations to less than 1.0 mil on housing and shaft. Thus, after applying the appropriate correction factors, final vibrations were acceptable for both turbines.

• Draft fan blade failures are especially dangerous, as the housing cannot restrain the massive parts that might be released from the impeller. In the second example, an investigation of the original design blade vibrations on a 900 rpm fan revealed a blade resonance of only about six percent above three times running speed when the fan was cold. Changes in operational temperatures every time the unit was started would cause this resonance to match with the excitation frequency at three times running speed for brief periods.

A filtered bearing housing vibration of 1.2 mils was measured during the resonance period. Multiplying this vibration by the appropriate correction factor (1.2×3.5) yields 4.2 mils, which is in the "F" or immediate failure region of Figure 27 for 2700 cpm. After several days of operation and several startups, the vibrations increased suddenly and the fan was tripped off. Subsequent inspection revealed blade cracks near the welds, which indicated that blade modifications were necessary to avoid excitation of transient resonances.

• In the third example, bearing housing measurement indicated 14.5 mils filtered vibration on a 900 rpm induced draft fan. As the rotor critical speed and foundation resonance were found by testing to be 1100 cpm and 720 cpm, respectively, then correction factors for rigid rotors (K_5) and soft foundations (K_6) can be applied. The effective vibration is calculated by multiplying these factors by the measured vibration ($14.6 \times 0.7 \times 0.6$) which yields 6.1 mils. Entering this effective vibration on the housing severity chart (Figure 26) indicates the severity level is between marginal and probable failure, which is undesirable for longterm operation. Correction of the thermal bow problem and rebalancing reduced the vibration to 2.0 mils, or an effective vibration of 0.8 mils, which is acceptable.

• A lube oil pump motor for an auxiliary turbine had experienced repeated bearing failures prior to commissioning. An investigation of the system dynamics found that 13 mils filtered casing vibration occurred at the running speed frequency of 3600 cpm and the applicable correction factors, K_4 and K_5, essentially cancel each other and the effective vibration is well into the immediate danger region so the pump was shutdown. Further investigation yielded that the motor shaft had excessive misalignment and the motor casing resonance existed only 1.5 percent below running speed. Once the alignment problem was fixed and the motor support was stiffened to move the case resonance, vibrations were reduced to acceptable level and no further problems were reported.

• An investigation of a 3000 hp steam turbine was conducted to determine if it was safe to operate the unit. Maximum vibration readings of 1.5 in/sec bearing housing motion and 1.6 mils shaft relative motion were found at running speed frequency of 11,400 cpm. In evaluating the vibration severity, a service factor of 1.0 was applied as no other factors were appropriate. It was decided to shut the turbine down immediately as the bearing housing was in the "F," or danger, region and shaft vibrations were in the "D," or probable failure, region. Inspection revealed some seal rub damage and excessive unbalance, which was 40 times of specified value. Further investigation indicated that high unbalance was due to the previous shaft repair which involved sleeving one of the journals causing shift of the rotor mass center. Rebalancing at high speed (flexible shaft) using multiplane balancing techniques and adding weights on both sides of the sleeve journal brought the vibration down to the no fault region, "A;" housing vibration was reduced to 0.025 in/sec and shaft vibration was reduced to .5 mils. Subsequently no problems have been reported for this turbine.

To summarize major points for applying vibration severity criteria to assess machinery condition:

• First hand experience with a variety of equipment has shown that the simplified vibration criteria used in the past is less than adequate for many actual rotating components.

• To equitably accommodate different machine designs, installations, and vibration problems, an effective vibration value must be obtained by multiplying the measured vibration value by one or more applicable correction factors listed in Table 4.

• There are no absolute limits that will assure successful longterm operation, nor firm upper limit that will cause failure for any specific machine. As a result, some subjective judgement is always required in application of any vibration limits.

• Any vibration criteria utilized should be compared with the manufacturer's recommendations and user's experience with the particular type of machinery.

• An important factor to recognize in applying vibration limits is growth of vibration amplitudes with time. Machines where a mechanical defect has deteriorated to an unacceptable level, rapid growth in vibration and consequent failure can be expected.

• To properly assess the severity of existent problem, it is necessary to establish the source of machine vibration and selected appropriate transducer to measure the vibration (machine housing or shaft) most likely to reveal the expected failure mechanism.

SOURCES OF VIBRATION

Establishing the proper source of excessive vibration is probably the most difficult task of machinery analysis. After the source of vibration is determined, it is generally possible to assess the severity of the existing problem by comparing the vibration level against a specified vibration standard. As the sources of the various excitation forces often occur at different frequencies, it is very useful to characterize measured vibration signals with respect to frequency. Thus, spectral analysis is a recommended procedure for solving vibration problems and evaluating the reliability of equipment.

Vibration amplitudes obtained from spectral analysis should be investigated in relation to machine design, installation, and vibration source. Judicious placement of vibration transducers so that a vibration source can be separately identified is necessary to gain maximum use of the vibration criteria. Separation of gear mesh excitation from turbine blade vibrations, for example, might be determined by analyzing signals from a few trial locations and applying judgement based on knowledge of the machine's design and exact frequencies of each source. An overview of several most common vibration sources follows.

Resonance

Resonance is probably the most common cause of high vibration and most vibration related failures. Very often user vibration specifications ignore resonances completely and depend upon the manufacturer for such criteria. If resonant margins are not specified and verified by testing, the component could be very precisely balanced or adjusted to meet the acceptable vibration limits at the time of purchase. However, longterm use of such a machine may result in high maintenance requirements as slight unbalances or other distortions accumulate causing vibrations to increase to unacceptable levels. General guidelines on critical speed margins for lateral shaft vibration are provided by API standards for several different types of rotating equipment. For example, compressors and pumps are required to have lateral critical speed margins of 20 percent above maximum operating speed or 15 percent below minimum operating speed and amplification factors should not exceed eight while going through criticals (Figure 30). If the measured critical speed falls within the excluded range, then the manufacturer must demonstrate that the vibrations at the critical speed are within acceptable vibration limits with considerably more unbalance than would normally be expected.

Special attention should be given to blade related problems. Very little can be done to minimize blade vibrations after they have occurred as a problem in the field. The blade design should be investigated thoroughly prior to actual manufacturing of the unit. The fundamental principle in dealing with blade vibrations is to avoid resonance. The blade natural frequency should not be within 10 percent of any identifiable excitation frequencies at any point in the normal operating range, including multiples of blade passage frequency and multiples of running speed. The aerodynamic phenomena are too complex to be predictable at the present time and vary greatly from machine to machine. As a result, any attempt to live with an excitation of the lower four or five blade modes is likely to cause blade loss failure regardless of excitation force predictions. If the frequency of the driving forces cannot be changed, the blade design should be changed to detune it from resonance. The calculation of the natural frequencies of a blade are quite involved and should include the effect of centrifugal forces combined with a knowledge of the root fixity factor. Simplified calculations, however, can give sufficient accuracy for the lower modes to determine if a blade vibration can be a problem.

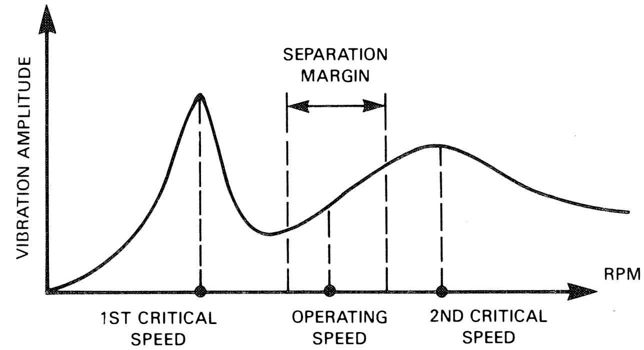

Figure 30. Rotor Critical Speed Effects on Vibration Response.

Subsynchronous Instabilities

Subsynchronous instabilities are another possible source of vibration in plant operating equipment. "Instabilities" is the term given to those vibration sources that are self excited by some mechanism where fluid, aerodynamic, or frictional forces interact with the rotor. Instabilities usually exhibit vibration frequencies at about ½ running speed (Figure 31) and have a tendency to suddenly increase in amplitude with disastrous results. Rotating stall of draft fan air foils at low flow is another source of subsynchronous rotor vibration, but generally is not as dangerous because the excitations are not unstable, but are simply the result of unbalanced flow forces. The following items increase the susceptibility of a rotor to subsynchronous instabilities:

• Rotors operating above the first critical speed.

• Compressors with high outlet pressure and high molecular weight gas.

• Lightly loaded bearings.

• Presence of high pressure pulsations and acoustical resonances.

• Stiff radial bearings relative to the shaft flexural stiffness.

• Inadequate seal design.

• Sudden shock loads.

• Presence of rotor frictional sources such as light shrink fits over a long section.

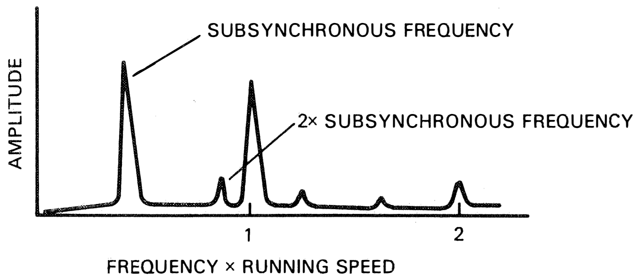

Figure 31. Spectral Characteristics of Subsynchronous Instabilities.

Most common types of instabilities include:

• *Fluid film bearing instability.* Self excited rotor vibration due to fluid film bearing instability, known as oil whirl and oil whip, have been an area of concern for a long time. This type of instability can often be eliminated by a change in bearing configuration to more stable design. Bearing designs which have the highest order of stability are tilting pad and squeeze film bearings.

• *Floating ring seal instability.* Floating ring seals can act in a similar manner as journal bearings and thus potentially contribute to self excited vibration. The analog to a circular short bearing is quite obvious and the corresponding theories allow the stiffness and damping properties in the locked up condition to be estimated. In lockup, floating ring deals are held radially by friction forces which do not allow them to follow the shaft motion. Their destabilizing effect may be reduced by balancing the floating ring seal axial force in such a way that the friction forces are as small as possible. Another possibility is to optimize the clearance geometry in order to obtain favorable dynamic stiffness properties.

• *Labyrinth seal instability.* One of the dominant self excitation mechanisms in high pressure compressors has been found to have its origin in the labyrinth seals. Labyrinth seal instability problems depend heavily on the labyrinth seal geometry and machine operating conditions; more specifically: labyrinth radius, strip height, rotational speed, internal pressures, Mach number, and inlet swirl velocity. Generally, the best way to reduce labyrinth seal instability is by suppressing the inlet swirl velocity of the leakage flow entering the seal.

• *Aerodynamic whirl instability.* Aerodynamic whirl instability is related to lateral aerodynamic forces acting on the compressor or turbine blades. These forces are generated by varying tip clearances when the rotor is displaced from the center position. It has been determined that susceptibility of the rotor to this type of instability increases with an increase in gas density and rotational speed of the machine.

• *Hysteretic instability.* Hysteretic (frictional) instability usually occurs if rotors operate above the first critical speed and internal forces that lag the bending displacements are allowed to occur. The source of excitation is normally the frictionally suppressed movement in the shrink fits of wide discs or impellers shrunk to the shaft. To reduce this type of instability, the axial contact length of the shrink fit should be as short and as tight as possible without exceeding the yield strength of the material.

Unbalance

Unbalance is a common source of machinery vibrations. Unbalance is always characterized by radial vibration of the shaft or casing which is in exact synchronization with rotor speed (Figure 32). Unavoidable geometry imperfections due to fabrication or material variation will result in rotor unbalance such that virtually all new high speed machines will require balancing at the factory. Some very sensitive machines will also require rebalancing after installation. Vibration should be expected to gradually change with time as the effects of erosion, wear, and particle adhesion act to change the unbalance. Thus, rebalancing will eventually be necessary on most machinery. Rotor parts such as thrust collars, cooling fans and coupling hubs may be significant sources of unbalance, but are often overlooked because of their small size.

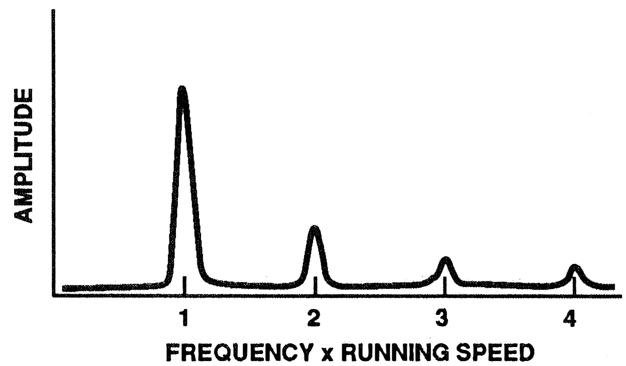

Figure 32. Spectral Characteristics of Imbalance.

Shaft bow is a type of unbalance caused by gravity sag or shaft thermal gradients. This bow will "roll out" or change with time after the rotor is started. The application of turning gear and roll-out procedures to minimize shaft bow is well known in the industry.

Shaft bow due to thermal distortion, which has been observed on ID fans, is a more difficult problem to identify and solve than normal shaft bow. The root cause of the thermal distortion is an apparent variation of thermal coefficient of expansion or Young's modulus throughout the shaft or impeller. A bimetallic strip effect occurs as the rotor is heated, causing the shaft's center of mass to be moved away from the center of rotation. Careful correlation of vibration with shaft thermal growth is necessary to identify this source of variable unbalance.

Magnetic Unbalance

Magnetic unbalance in motors is an excitation source that is often mistaken for mass unbalance. Forces due to magnetic asymmetry usually rotate at electrical synchronous speed and will beat with mass unbalance on induction machines. A test to distinguish magnetic unbalance from mass unbalance is to observe the real time vibration spectra as power to the machine is cut off. If a significant vibration reduction appears to occur instantaneously, then that portion was due to magnetic unbalance and the remainder due to mass unbalance.

Misalignment

Excessive misalignment of rotating elements driven through flexible couplings is usually indicated by a large second order vibration component (Figure 33). Occasionally, large first order vibrations are also observed. High axial vibration is another indication that misalignment is likely. Shaft operating misalignment is affected by relative thermal growth, static forces applied by piping or condenser attachments, deterioration of support grouting, etc. Vibration due to first order misalignment can be differentiated from unbalance by recording vibration in relation to speed. Unbalance vibration will increase with speed squared, while misalignment vibration will not change if resonances are not involved. For machines which cannot be conveniently shut down, it is recommended that a record be made of the vibration spectrum when the machine is first started up and in good alignment. This can be used for later comparison to determine if the alignment of the machine is still satisfactory.

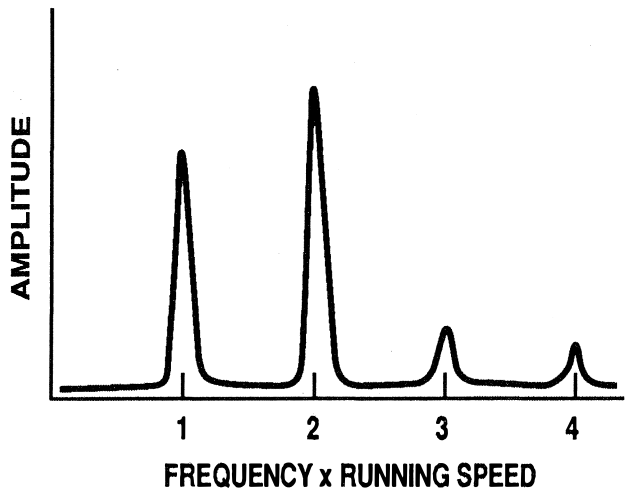

Figure 33. Spectral Characteristics of Misalignment.

Radial Rubs

Radial rubs between a rotor and close clearance stationary components can cause damage to seals and blade tips. Rubbing is usually undesirable and sometimes catastrophic. The reported characteristics of radial rub induced vibration include fixed rotor subharmonics, subsynchronous vibration at a natural frequency, supersynchronous vibration at a natural frequency, time varying synchronous vibrations, and vibration at multiples of running speed. Thus, it is difficult to make general statements about radial rubs, and the symptoms of rubbing are influenced by the materials in contact, the impedance to motion of the seal when contacted, location of natural frequencies, etc. Radial rubs of stainless steel rotors appear particularly damaging and are known to cause rapid permanent rotor deformation. High performance steam turbines, on the other hand, are designed with very tight labyrinth clearances and are almost expected to rub mildly during the early stages of commissioning. These temporary rubs can cause localized heating and a temporary bow in the rotor, with associated high unbalance, particularly when the rotor is running below its first critical speed. The symptoms of this particular rub induced vibration are a slow time variation of synchronous amplitude and phase, and often a square like orbit (Figure 34).

Axial Rubs

Axial rubs between stationary and rotating components can occur when relative axial motion between the two is sufficient to eliminate operating clearance. This problem is known to occur in large industrial gas turbines where the blades migrates in their slots due to breakage or deterioration of the blade locking mechanism (the problem has occurred in both compressor and turbine sections), and relaxation or creep of the vane or nozzle under load at high temperature. The consequences can range from progressive gradual "machining" of the contacting parts to catastrophic failure [19].

There has been some limited success in detecting blade migration as a result of change in vibration at the bearing mounted

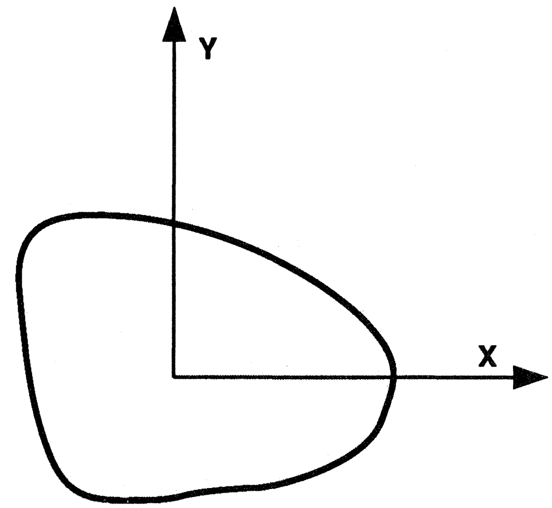

Figure 34. Shaft Orbit for Rub Induced Synchronous Vibration.

vibration sensor. In one example, a two bearing industrial gas turbine exhibited increasingly higher vibration measured by displacement probes during startup [19]. Initially, the gas turbine alarmed regularly and tripped occasionally. Inspection showed that a significant number of compressor blades had moved axially in the third stage, causing rubbing against the adjacent diaphragm. The fact that simple alarm and trip limits on vibration level have detected blade migration suggests that more critical trend monitoring of vibration vector changes (accounting for both amplitude and phase) should increase the detectability of blade migration problems should they occur.

Foreign Object Damage

Foreign object damage occurs when a loose object passes through a machine. Small amounts of damage may go undetected. Partial or complete loss of blade or blade distortion is likely to show up with sensitive vibration monitoring as a vector change in vibration or sometimes a trip. Again, close monitoring of vibration vectors should detect severe foreign object damage in the event that no other operational symptoms are detected.

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