Technical Paper

# Condition Monitoring of Turbomachinery – Tutorial 1

The advent of miniaturized electronics to the field of machinery condition monitoring has led to efficiency and avoidance of costly failures only imagined a decade ago. The "premium" for such insurance is becoming more and more justifiable.

The difference between an efficient profit generating machine and an expensive wreck may be only 0,25 mm – about the thickness of a fingernail. That is typical of the clearances between steam turbine blades and diaphragms that allow energy to be extracted from superheated, high-pressure steam to turn generators and produce condensed steam, which is returned to the boiler to start the cycle over. The smaller the internal clearances, the higher the efficiency, but the greater the risk of catastrophic contact between rotating and stationary parts. This risk can be worth over US $500,000 per day on a 600 MW unit, so the "premium" for insurance like machinery condition monitoring should be easy to justify.

## Rotor motion – required and undesirable

A perfect rotor would rotate around a pinpoint within its bearings, and there would be no:

- Lateral (radial) motion
- Axial (longitudinal) motion
- Torsional (circular pulsing) motion

These undesirable motions are always present because perfect rotors do not exist, but as long as they remain within tolerance, the machine operates within design efficiency.

## Radial vibration

Increasing radial vibration causes seal and bearing wear, reducing efficiency. Internal clearances are no longer as designed, so more energy is wasted. Increased bearing clearances allow more motion of the rotor relative to the machine case, ultimately allowing contact between rotor parts and casing. Non-contacting displacement probes mounted in, or adjacent to, the bearings can monitor this "shaft relative" motion precisely.

## Axial motion

Axial motion is typically controlled by balance pistons or drums, which essentially equalize pressures throughout a machine. Thrust bearings absorb normal axial loads, and non-contacting eddy current probes monitor the axial position of the rotor relative to the thrust bearing. Excess axial motion allows blading to strike diaphragms, and can cause the thrust bearing to fail. Thrust bearing failure can occur in seconds and cause a total loss of the machine.

## Torsional motion

If the combined load and excitation torque exceed the torsional stress limits of a shaft or coupling, the shaft can crack and "twist-off" in a very recognizable manner. Frequent electric motor start-ups can cause torsional failure. Motors generate torsional vibration at two and four times running speed.

## Other effects of rotor created vibration

Even if these undesirable shaft motions are below danger limits, they rob energy from the mechanical system. Additionally, the energy input to the mechanical system by rotor vibration can loosen bolts, crack foundations and casing, cause fatigue in piping and (by transmission through piping and foundations) even damage the bearings of stationary machines.

Transmitted vibration, which is the response of the system to input forces generated by shaft displacement, must be measured with velocity or acceleration transducers mounted on bearing caps or casings.

## Typical causes

The most common causes of radial vibration are changes in dynamic balance, coupling misalignment, bearing instabilities, loose rotor component and bearing parts. Slow changes in axial position can typically be attributed to thrust bearing and surging and other process changes such as flow reversal, water slugging, etc.

## Machinery condition monitoring

The number one requirement of any monitoring project is the correct placement of the proper transducer for the job. This requires a basic knowledge of the machine's construction and an understanding of mechanical "transmissibility".

The flow chart (→ **fig. 1**) shows that bearing type and casing-to-rotor weight ratios are the main controlling factors in the correct choice of transducers. The stiffer the bearing and the lower the damping, the higher the transmissibility. This dictates the use of "bearing cap" or "casing" measurements, which reflect the response of the mechanical system to input forces from the rotor. Such measurements are made with velocity pickups or accelerometers, depending on the frequency range to be covered. Velocity pickups typically are limited to vibration frequencies below 2 000 Hz.

Accelerometers are favored over velocity transducers because of their better frequency response, ruggedness and price, and are preferred by API (American Petroleum Institute) 678 standards. Rolling element bearings are a special case of the high transmissibility system, and accelerometers are useful in detecting small high frequency signals that are often early warning of internal damage (→ **fig. 2**).

On machines with plain bearings, high casing to rotor weight ratios and high damping, the non-contacting displacement probe measures the actual motion of the shaft relative to the bearing – the direction as well as the amount (→ **fig. 3**).

In many cases, both types of transducers should be employed, because displacement probes respond better than accelerometers to forces generated by rotor vibrations close to running speed, whereas casing mounted accelerometers respond better than displacement probes to the high frequency forces generated by turbine blades, pump vanes, gear meshing, etc.

An important third case is that of shaft absolute measurements. When the displacement probe is mounted in a bearing, it must be assured the bearing is stable so the measured shaft relative motion is a fair assessment of the motion of the rotor relative to the casing. If the bearing pedestal is moving appreciably in response to forces from the rotor, there may not be much shaft relative motion to measure, but motion of the rotor relative to the casing could be very high. This is often the case with large turbine/generators and certain designs of boiler feed pumps, and the internal clearances must be protected by measuring "shaft absolute" motion. The preferred way of doing this is to use displacement probes to measure shaft relative motion and accelerometers to measure bearing motion – the two signals are normalized and algebraically summed up in a special monitor (→ **fig. 4**). "Shaft absolute" measurements were in fact the first vibration measurements taken, and were made using a "shaft rider" that was, in effect, a plunger riding on the shaft periphery with a velocity pickup mounted on the other end.

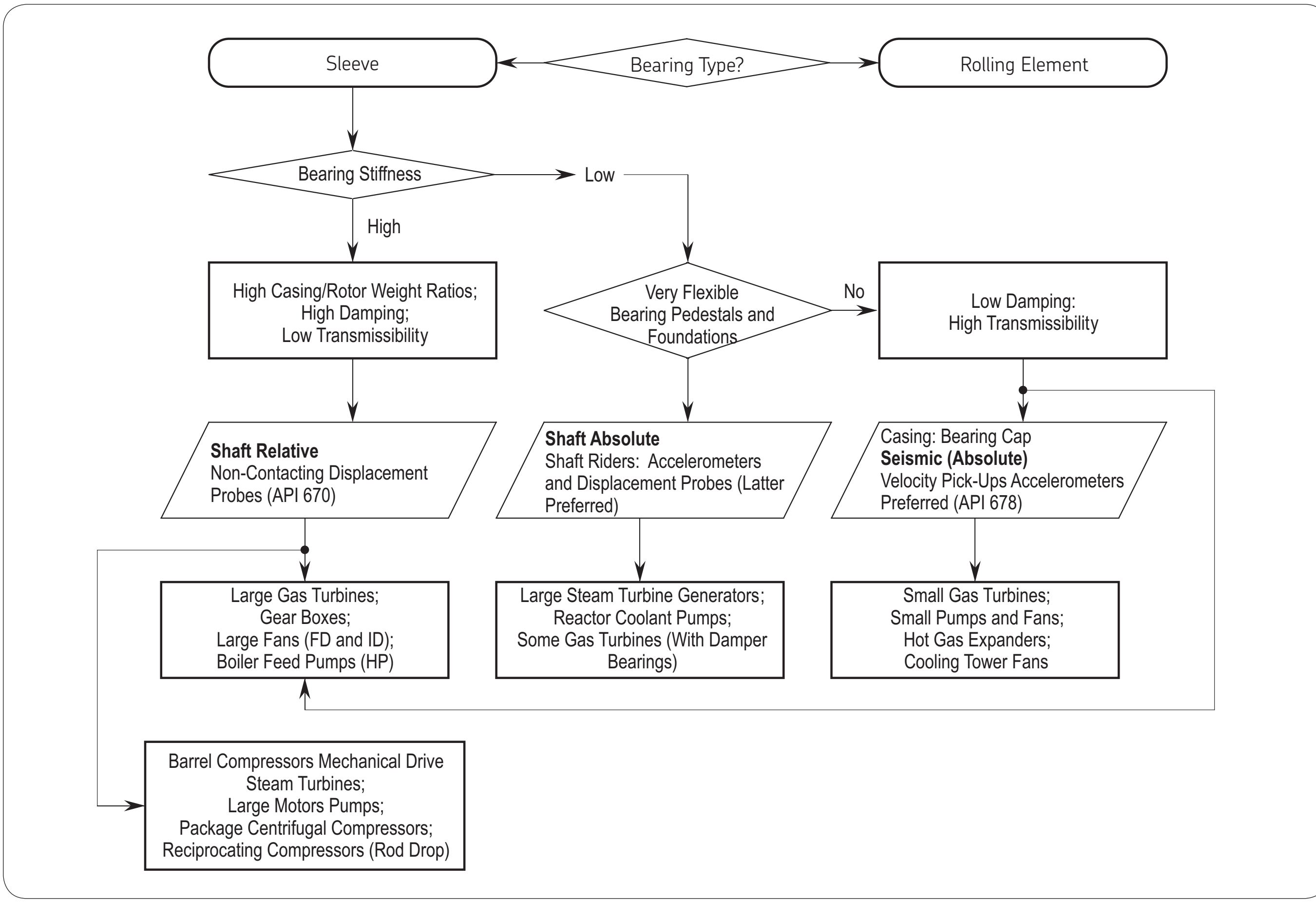


*Fig. 1. Machinery vibration transducer selection by machinery tape.*

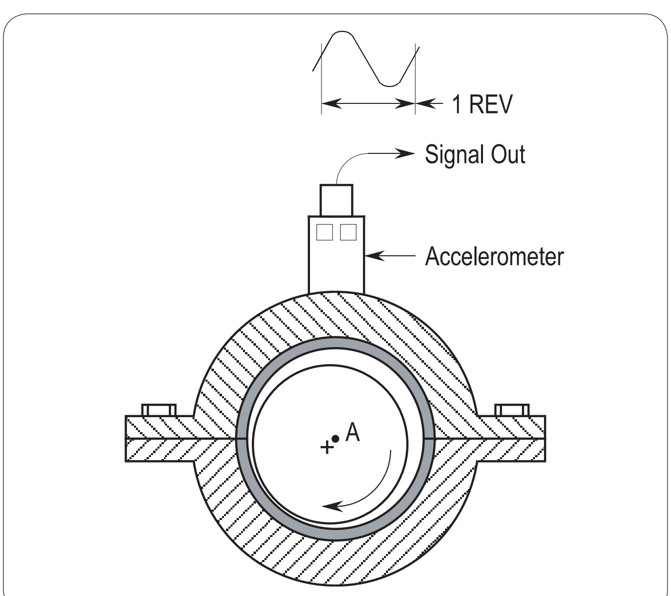


*Fig. 2. Accelerometer used to measure bearing cap motion.*

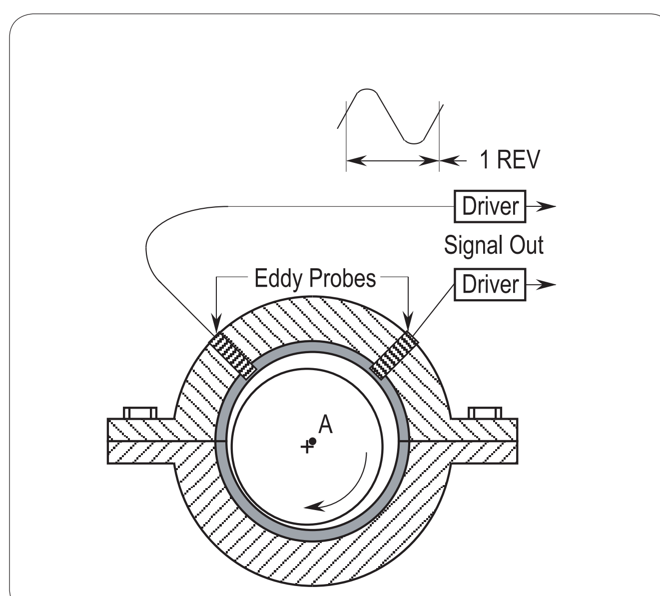


*Fig. 3. Eddy current probe used to measure shaft relative motion.*


*Fig. 4. SKF M800A monitor keeps track of turbine r/min, bearing clearances, shaft vibration, thrust, etc.*

# Special measurements for large steam turbine generators

For maximum efficiency, steam turbine internal clearances are in the order of 0,125 to 0,750 mm *(0.005 to 0.030 in.)* radial and 2,500 to 12,500 mm *(0.098 to 0.492 in.)* axial. Maintaining these clearances during the tremendous temperature changes of start-up and shutdown requires special monitoring circuits and transducers.

## Eccentricity

First the shaft is so heavy that at rest it takes on a bend or “bow” that must straighten before certain speeds are reached, or radial rubs occur. The amount of this bow is monitored by an eddy current probe that measures the peak-to-peak motion of a small disc, mounted outboard of the number 1 bearing (→ **fig. 5**).

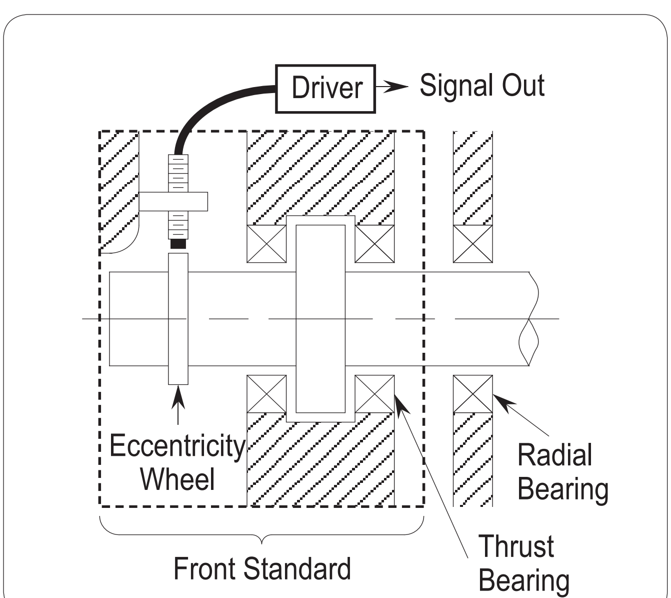


*Fig. 5. Measurement of shaft eccentricity – the amount of bow or “bend” in the shaft.*

# Casing expansion

Turbine casings are pinned to their foundations by the footing surrounding the exhaust hood. When superheated steam enters the casings, they expand by up to 25 mm *(1 in.)* or more. The front standard (including the thrust bearing and front bearing assembly) are keyed to the foundation on sliding feet to allow expansion without buckling the casing. To ensure this movement occurs correctly, an LVDT transducer is mounted to the foundations with its plunger fastened to the front standard (→ **fig. 6**).

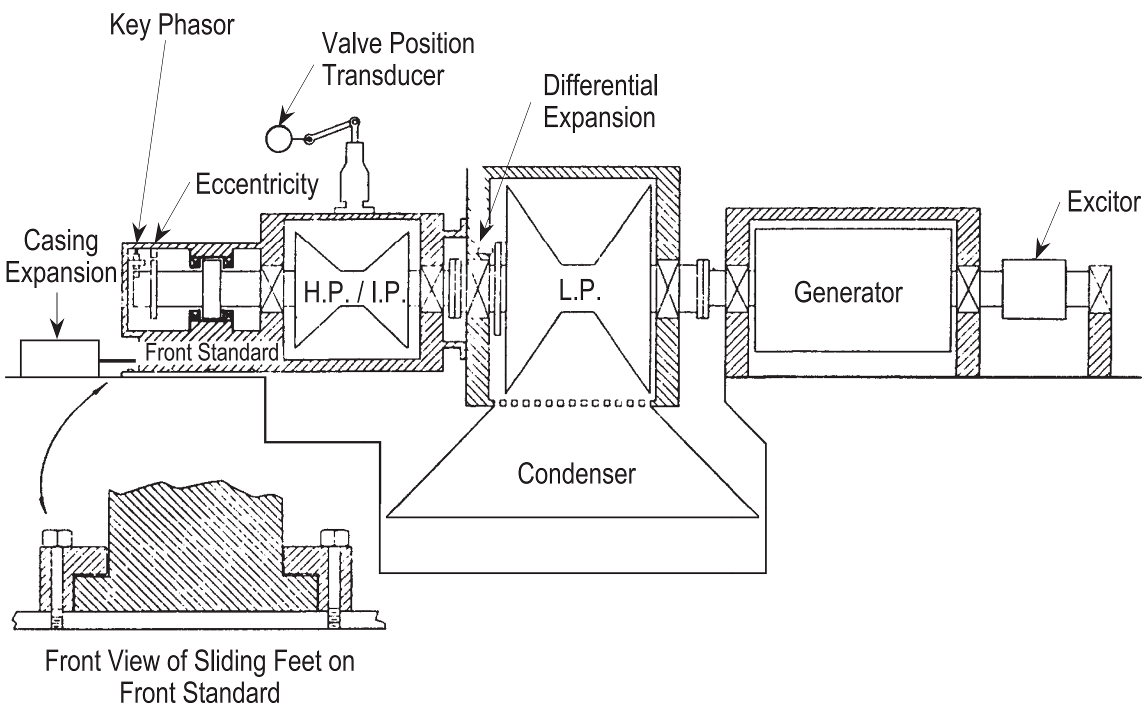


***Fig. 6.** Typical block diagram of a steam turbine showing turbine supervisory transducer positions.*

## Differential expansion

During start up, the rotor must expand at the same rate as the casing to prevent axial rubs. Depending on the design of the turbine, there are currently three ways to monitor differential expansion, all using eddy current probes:

- **Fig. 7a** shows a long-range probe observing a perpendicular collar on the shaft.
- **Fig. 7b** shows two "normal" range probes (25mm *(1 in.)*) observing a perpendicular target so that it is never out of range of one or the other probes (typical of General Electric (USA) turbines).
- **Fig. 7c** shows the ramp method (typical of Westinghouse turbines) where the range of the probes is artificially amplified by the ramp.

All of these methods have some serious drawbacks, and other methods are being pursued so that accuracy will not suffer so much from temperature, environment, radial shaft movement, etc.

## Valve position indication

To determine how much steam is being fed to the turbine, a rotary potentiometer is used to measure the position of the cam that operates the steam control valve (on GE units). Westinghouse units usually require individual transducers for each governor valve (up to eight), and the output of each potentiometer is gauged with the others to provide a percentage of total valve openings.

## Speed, phase and rate of change of speed

Speed is best measured by an eddy current probe that observes a notch on the shaft, usually referred to as a "one-per-rev event" (**→ fig. 8**). A speed monitor automatically converts this to rev/min and the same signal serves as a fixed reference point on the shaft to measure the "phase" or direction of the vibration. This is essential for dynamically balancing turbines, and also assists in the diagnosis of mechanical problems. Rate of change of speed is also derived from the same signal and is necessary to maintain the maximum permissible start-up rates and provide smooth transmission from soak-point to soak-point.

## Other necessary measurements

Bearing metal temperatures, thrust position, lube oil pressures and temperatures (absolute and differential) are advisable on all machinery and no less so on large steam turbines. The value of these measurements is multiplied synergistically when combined with vibration, phase, etc., for troubleshooting and control.

## Data acquisition, analysis and display

Once the correct transducers are in place, the choice of information usage and display become part of the operating philosophy of the plant. If simple manual control by the operator is all that is required, then a rack of standard monitors is sufficient. This monitor rack provides clear visual indication of present status and can give audible and visual alarms, or even automatic shutdown, when acceptable levels are exceeded. Tape recorders, balance analyzers and spectrum analyzers can be easily connected to the signals via buffered front panel outputs.

Due to the shortage of control panel space, and the fact that individual meters are expensive and only used on rare occasions, more advanced microprocessor controlled devices that print out records of alarms, store history and compute trends are coming into general usage. Such devices typically manage multiple channels of mixed inputs in a very compact volume and are much less expensive on a per channel basis than standard monitors.

Modern machinery monitoring systems, such as the powerful SKF M800A system, have standard computer interfaces that allow connection to process control systems and/or desktop computers for machinery information, analysis, and reporting. The M800A system also contains embedded spectrum modules for fast fourier transform monitoring and alarm windowing. Combined with the desktop-based machinery diagnostic workstations, this capability provides the most powerful condition monitoring package available.

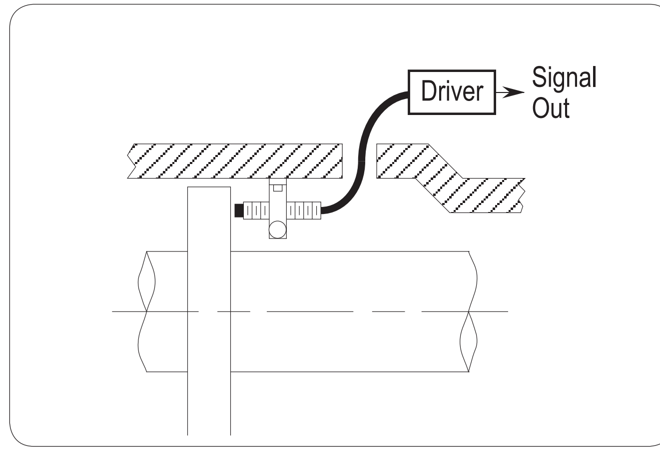


*Fig. 7a. Measurement of differential expansion: A perpendicular collar on the shaft is observed by a long-range probe.*

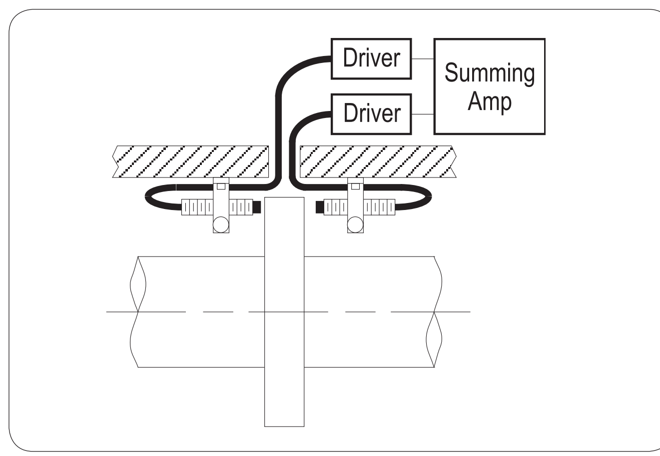


*Fig. 7b. Measurement of differential expansion: Complementary probes method favored by General Electric (USA).*

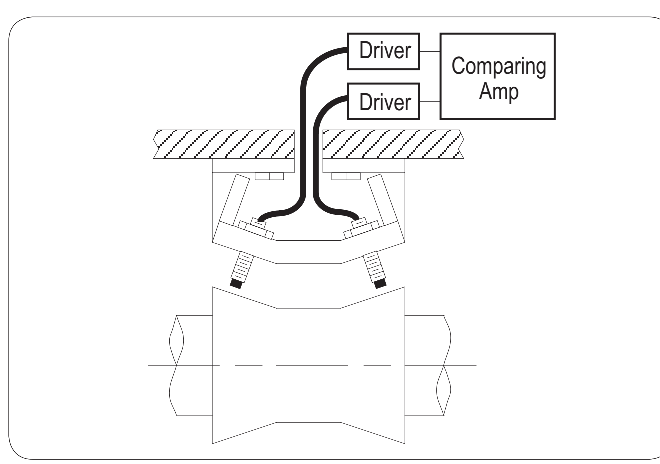


*Fig. 7c. Measurement of differential expansion: Ramp method favored by Westinghouse.*

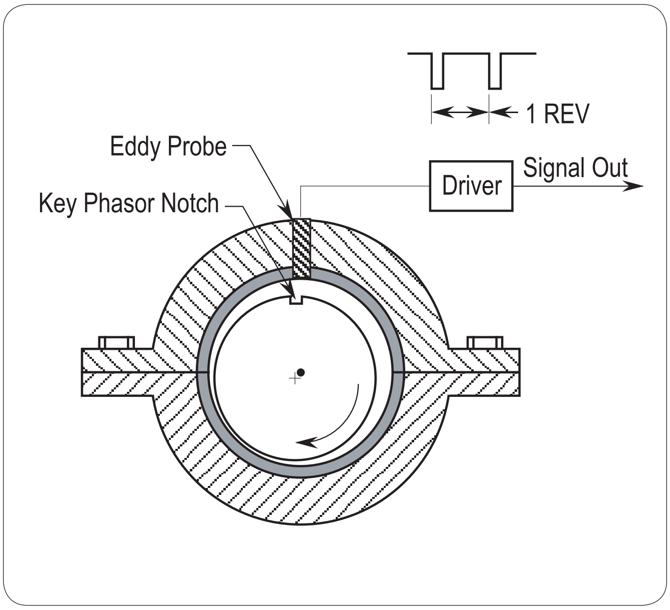


***Fig. 8. Key phasor signal generation; an eddy probe observes the key phasor notch on the shaft to measure speed, rate of change of speed and phase.***

A typical system provides protection, analysis and diagnosis over 1 000 channels. The computer and spectrum monitoring modules work together to perform high-speed on-line spectrum monitoring, comparison and analysis.

Every 5 to 12 seconds, the vibration signals are analyzed, compared to baseline, and a warning issued if the spectrum has changed. This is a very sensitive method of detecting mechanical problems, sometimes months before they become serious. Valuable start-up and coast-down data can also be taken automatically by such systems, and the data post-processed for a variety of different displays that provide different insights. Such systems can be viewed as “resident machinery analysts” in plant control rooms, 24 hours a day, every day.

# The future

Although the origins of machinery condition monitoring can be traced back several decades, it really only started as a viable technology about 30 years ago, with the introduction of reliable transducers, especially non-contacting displacement probes. API 670 (the first industrial standard for vibration monitoring) was published in 1976 and laid down basic standards for transducer and monitor accuracies and reliabilities, installation practice and “voting logic” considerations so that nuisance alarms were reduced to a minimum and industry could entrust automatic shutdown of expensive processes to this relatively unfamiliar technology.

Since the advent of microprocessors and electronic memories, machinery condition monitoring can encompass mechanical vibration, process parameters, efficiency calculations and diagnostic capability that was only imagined a decade ago.

As with most problems in a competitive free market system, machinery manufacturers are self-regulating and must cure the problems to stay in business. To assist the process, organizations such as the API (American Petroleum Institute) write standards not only for machinery condition monitoring equipment, but for various types of machinery. For example, API 617 deals with centrifugal compressor design and sets standards for rotor unbalance and other vibration. More importantly, API 617 sets limits on the rotor “Amplification Factor” at critical speeds. This in effect sets limits on maximum response, or conversely minimum damping in the rotor bearing system.

This is a very significant step, because less damping means higher efficiency – the goal of the designer. But the user’s goal is also reliability, and API 617 (and its steam turbine equivalent API 612) is essentially telling designers that efficiency must be compromised in the interests of reliability.

The Electrical Power Research Institute (EPRI), a nonprofit organization in Palo Alto, California, funded by utilities in the USA, has a charter to finance and manage research projects on behalf of the power generation industry. An increasing number of their projects are now concerned with machinery condition monitoring and design improvements that can result from these studies. Recommendations on monitoring boiler feed pumps, study programs on hydroturbines and vertical pumps, evaluation of computerized vibration analysis and diagnosis systems, and most recently, a study on how to detect cracks in shafts, are some of the condition monitoring–related programs that have been funded. Insurance companies such as Travelers and Hartford Steam Boiler not only have a vested interest in vibration technology, but use it themselves for surveying clients' machinery.

The interest of these organizations is vital considering most power generation plants are well over 30 years old and run at 65% availability, compared to over 90% in the hydrocarbon processing industries. Their efforts to improve and ensure machinery availability constitutes not only an investment in vibration measurement, but an affirmation that the technology does work and has a measurably healthy return on investment – by protecting those few hundredths of a millimeter between turbine blades whirling at transonic speeds and a motionless casing.

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