CHAPTER 16
CONDITION MONITORING OF MACHINERY

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INTRODUCTION

Condition monitoring of machinery is the measurement of various parameters related to the mechanical condition of the machinery (such as vibration, bearing temperature, oil pressure, oil debris, and performance), which makes it possible to determine whether the machinery is in good or bad mechanical condition. If the mechanical condition is bad, then condition monitoring makes it possible to determine the cause of the problem.[1,2]

Condition monitoring is used in conjunction with *predictive maintenance,* i.e., maintenance of machinery based on an indication that a problem is about to occur. In many plants predictive maintenance is replacing *run-to-breakdown maintenance* and *preventive maintenance* (in which mechanical parts are replaced periodically at fixed time intervals regardless of the machinery's mechanical condition). Predictive maintenance of machinery:

- Avoids unexpected catastrophic breakdowns with expensive or dangerous consequences.
- Reduces the number of overhauls on machines to a minimum, thereby reducing maintenance costs.
- Eliminates unnecessary interventions with the consequent risk of introducing faults on smoothly operating machines.
- Allows spare parts to be ordered in time and thus eliminates costly inventories.
- Reduces the intervention time, thereby minimizing production loss. Because the fault to be repaired is known in advance, overhauls can be scheduled when most convenient.

This chapter describes the use of vibration measurements for monitoring the condition of machinery. Vibration is the parameter which can be used to predict the broadest range of faults in machinery most successfully. This description includes:

- Selection of an appropriate type of monitoring system (permanent or periodic)
- Establishment of a condition monitoring program
- Fault detection
- Spectrum interpretation and fault diagnosis
- Special analysis techniques
- Trend analysis
- The use of computers in condition monitoring programs.

TYPES OF CONDITION MONITORING SYSTEMS

Condition monitoring systems are of two types: periodic and permanent. In a *periodic monitoring system* (also called an *off-line condition monitoring system*), machinery vibration is measured (or recorded and later analyzed) at selected time intervals in the field; then an analysis is made either in the field or in the laboratory. Advanced analysis techniques usually are required for fault diagnosis and trend analysis. Intermittent monitoring provides information at a very early stage about incipient failure and usually is used where (1) very early warning of faults is required, (2) advanced diagnostics are required, (3) measurements must be made at many locations on a machine, and (4) machines are complex.

In a *permanent monitoring system* (also called an *on-line condition monitoring system*), machinery vibration is measured continuously at selected points of the machine and is constantly compared with acceptable levels of vibration. The principal function of a permanent condition monitoring system is to protect one or more machines by providing a warning that the machine is operating improperly and/or to shut the machine down when a preset safety limit is exceeded, thereby avoiding catastrophic failure and destruction. The measurement system may be permanent (as in parallel acquisition systems where one transducer and one measurement chain are used for each measurement point), or it may be quasi-permanent (as in multiplexed systems where one transducer is used for each measurement point but the rest of the measurement chain is shared between a few points with a multiplexing interval of a few seconds).

In a permanent monitoring system, transducers are mounted permanently at each selected measurement point. For this reason, such a system can be very costly, so it is usually used only in critical applications where: (1) no personnel are available to perform measurements (offshore, remote pumping stations, etc.), (2) it is necessary to stop the machine before a breakdown occurs in order to avoid a catastrophic accident, (3) an instantaneous fault may occur that requires machine shutdown, and (4) the environment (explosive, toxic, or high-temperature) does not permit the human involvement required by intermittent measurements.

Before a permanent monitoring system is selected, preliminary measurements should be made periodically over a period of time to become acquainted with the vibration characteristics of the machine. This procedure will make it possible to select the most appropriate vibration measurement parameter, frequency range, and normal alarm and trip levels.

ESTABLISHING A CONDITION MONITORING PROGRAM

A condition monitoring program may be established to check the satisfactory operation of a single machine or, more usually, it is established to check the operation of a number of machines, perhaps all the machines in an entire plant. The following steps are usually considered in the establishment of such a program, depending on the type of machine and impact of failure of operation machines might have.

Step 1. *Determine the type of condition monitoring system,* described in the preceding section, that best meets the needs of the plant.

Step 2. *Make a list of all of the machines to be monitored* (see, for example, Table 16.1), based on the importance of these machines in the production line.

Step 3. *Tabulate the characteristics of the machines* that are important in conducting vibration analyses of the machines of step 2. These characteristics are associated with machine construction such as the natural frequencies of shafts, casings, and pedestals, and operational and defect responses. A tabulation of machine frequencies is important because fault analysis is conducted (Table 16.2) by matching machine frequencies to measured frequencies appearing in a spectrum. The following machine characteristics provide the necessary information for fault analysis.

- Shaft rotational speeds, bearing defect frequencies, number of teeth in gears, number of vanes and blades in pumps and fans, number of motor poles, and number of stator slots and rotor bars.
- Vibratory forces such as misalignment, mass unbalance, and reciprocating masses.
- Vibration responses due to process changes, such as temperature and pressure.
- Fault responses associated with specific machine types, such as motors, pumps, and fans.
- Sensitivity to instability in components, such as fluid film bearings and seals due to wear and clearance.
- Loads or changes in operating conditions.
- Effects of mass unbalance, misalignment, distortion, and other malfunction/defect excitations on vibration response.

TABLE 16.1 Machinery Classification for Monitoring

Machinery classification	Result of failure
Critical	Unexpected shutdown or failure causes significant production loss.
Interrupts production	Unexpected shutdown or failure causes minor interruptions in production.
Causes inconvenience	Inconvenience in operation, but no interruption in production.
Noncritical	Production is not affected by failure.

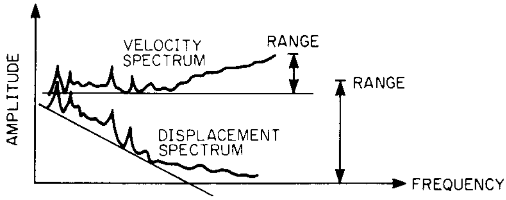


FIGURE 16.1 Displacement and velocity spectra obtained under identical conditions. The velocity spectrum requires a smaller dynamic range of the equipment which follows the transducer. Therefore, it is preferable.

Step 4. *Select the most appropriate vibration measurement parameter.* When an accelerometer is employed as the sensing device in a condition monitoring system, the resulting *acceleration* signal can be electronically integrated to obtain *velocity* or *displacement,* so any one of these three parameters may be used in measurements. The appropriate parameter may be selected by application of the following simple rule: *Use the parameter which provides the "flattest" spectrum.* The flattest spectrum requires the least dynamic range from the instrumentation which follows the transducer. For example, Fig. 16.1 shows a velocity spectrum and a displacement spectrum obtained under identical conditions. The dynamic range (i.e., the range from the highest to the lowest signal level) required to measure the displacement spectrum is much larger than the range for the velocity spectrum; it may even exceed the available dynamic range of the instrumentation. Therefore, according to this rule, velocity measurements should be selected.

The *flattest spectrum* rule applies only to the frequency range of interest. Therefore, the parameter selection, to some extent, depends on the type of machine and the type of faults considered.

Step 5. *Select one of the following vibration pickups that will best meet the requirements of step 4.*

Displacement Transducer. A displacement transducer is a transducer that converts an input mechanical displacement into an electrical output that is proportional to the input displacement. Displacement transducer of the eddy-current type (described in Chap. 12), which have noncontacting probes, are commonly used to measure the relative motion between a shaft and its bearings. This information can be related directly to physical values such as mechanical clearance or oil-film thickness, e.g., it can give an indication of incipient rubbing. Shaft vibration provides information about the current condition of a machine and is principally used in permanent monitoring systems, which immediately shut the machine down in the event of trouble. The use of displacement transducers is essential in machinery having journal bearings. However, proximity probe transducers (1) usually are difficult to calibrate absolutely, (2) have limited dynamic range because of the influence of electrical and mechanical runout on the shaft, and (3) have a limited high-frequency range.

Accelerometers and Velocity Pickup. Pickups of this type, described in Chap. 12, are usually lightweight and rugged. They are always used for detecting faults which occur at high frequencies (say, above 1000 Hz), for example, to detect rolling-element bearing deterioration or gearbox wear. Acceleration measurements of bearing vibration will provide very early warning of incipient faults in a machine.

Step 6. *Select the measurement locations.* When a periodic (off-line) monitoring system is employed, the number of points at which measurements are made is limited only by the requirement for keeping measurement time to a minimum. As a general rule, bearing vibration measurements are made in the radial direction on each accessible bearing, and in the axial direction on thrust bearings. It is not usually necessary to measure bearing vibration in both the horizontal *and* the vertical direction, since both measurements give the same information regarding the forces within the machine; this information is merely transmitted through two different transmission paths. This applies for *detecting* developing faults. It will later be seen, however, that in order subsequently to *diagnose* the origin of the impending fault, measurements in both the horizontal and the vertical direction may give valuable information. When measuring shaft vibrations with permanently mounted proximity transducers, it is convenient to use two probes on each bearing, located at 90° from each other, thereby providing an indication of the orbit of the shaft within the bearing. Axial displacement transducers, programmed to shut the machine down on preset levels, are mounted where a thrust measurement will protect the machine rotating parts, such as blades, from rubbing the stationary casing due to fault-induced axial forces.

When a permanent (on-line) monitoring system is employed using a seismic pickup, the number of measurement points usually is minimized for reasons of economy. Selection must be made following a study of the vibration spectra of different bearings in order to locate those points where all significant components related to the different expected faults are transmitted at measurable vibration levels if full spectrum comparison is performed. If only broadband measurements are monitored, then a further requirement is that all frequency components related to the expected faults must be of approximately the same level within the selected frequency range. Otherwise, measurements must be made in selected frequency bands.

Step 7. *Select the time interval between measurements.* The selection of the time interval between measurements requires knowledge of the specific machine. Some machines develop faults quickly, and others run trouble-free for years. A compromise must be found between the safety of the system and the time taken for measurements and analysis. Measurements should be made frequently in the initial stages of a condition monitoring program to ensure that the vibration levels measured are stable and that no fault is already developing. When a significant change is detected, the time interval between measurements should be reduced sufficiently so as not to risk a breakdown before the next measurement. The trend curve will help in determining when the next measurement should be performed.

Step 8. *Establish an optimum sequence of data acquisition.* The sequence in which data acquired in a condition monitoring program must be planned so that the data are acquired efficiently. For example, the data collection may be planned on the basis of plant layout, on the type of data required, or on the sequence of components in the machine train, from driver to driven components.

FAULT DETECTION IN ROTATING MACHINERY

It is highly desirable to be able to detect all types of faults likely to occur during the operation of rotating machinery. Such faults range from vibrations at very low frequencies (subsynchronous components indicating looseness, oil whirl, faulty belt drive, etc.) to vibrations at very high frequencies (tooth-meshing frequencies, blade-passing frequencies, frequencies of structural resonances excited by faulty rolling-element bearings, etc.). Such detection should be applicable to the complete range of machines in a plant, which operate from very low to very high speed. This requires the selection of equipment and analysis techniques which cover a very broad frequency range.

Measurements of *absolute* vibration levels of bearings provide no indication of the machine's condition, since they are influenced by the transmission path between the force and the measurement point, which may amplify some frequencies and attenuate others. Bearing vibration levels change from one measurement point to another on a given machine, since the transmission paths are different; they also change for the same reason from machine to machine for measurements made at the same measurement point.[3] Therefore, in estimating the condition of a machine, it is essential to monitor *changes* in vibration from a reference value established when the machine was known to be in good condition. Changes are expressed as a ratio or, more commonly, as a *change of level,* i.e., the logarithm of a ratio, in decibels.

The objective of condition monitoring of a machine is to predict a fault well in advance of its occurrence. Therefore, a measurement of the overall vibration level will not provide successful prediction because the highest vibration component within the overall frequency range will dominate the measurement. This is illustrated in Fig. 16.2, which shows an example where overall measurements of the vibration velocity resulted in an incorrect prediction with an overestimate of the lead time. The early detection of faults in machinery can be made successfully only by comparison with a *reference spectrum.* This section compares types of spectrum analysis for this purpose.

Condition monitoring techniques employed during transient operating conditions of the machine (i.e., when the machine is running up to full speed or slowing down from full speed) differ significantly from the techniques employed during steady-state operating conditions. Therefore it is essential that a careful investigation be carried out to ensure that the condition monitoring technique selected is appropriate for the conditions of measurement.

FALSE ALARMS

Changes in machinery vibration may result from a number of causes which are not necessarily related to the deterioration of the machine. For example, a change in speed of the machine or a change in the load on the machine usually greatly modifies the relative amplitudes of the different components of vibration at a fixed transducer location or modifies the relative pattern of vibration at different locations. Depending on the criteria used for fault detection, such changes may result in a false indication of deterioration of the machine. Appropriate selection of the technique employed can avoid such false alarms.

HOW SPECTRUM CHANGES ARE RELATED TO THE CONDITION OF A MACHINE

To obtain information about changes in condition of a machine, vibration spectra should be compared only for similar operating conditions. The influence of operat-

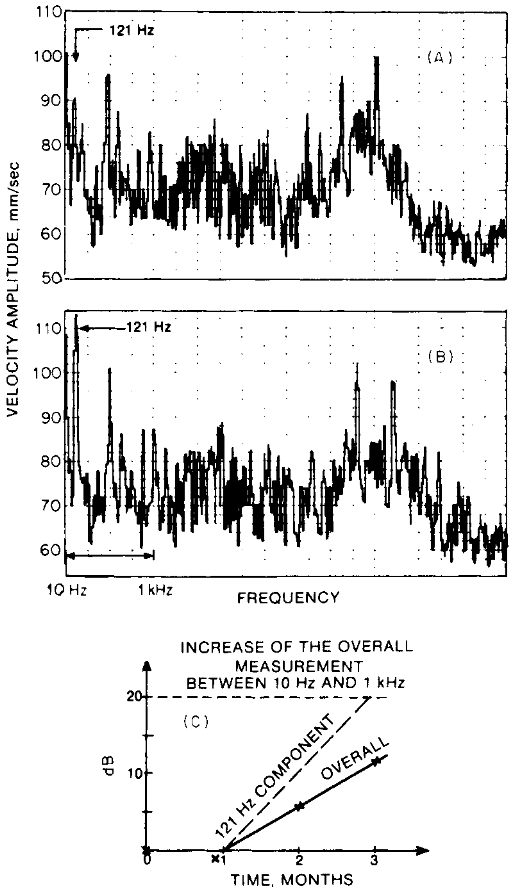


FIGURE 16.2 Trend analysis performed on an overall measurement and on an individual component. (*A*) The velocity spectrum of vibration measured on a gearbox after installation. Note the high amplitude of the 480-Hz component, dominating the reference spectrum. (*B*) The velocity spectrum 3 months later. Note the dramatic increase in the 121-Hz component, which corresponds to the output shaft speed of the gearbox. (*C*) Curves comparing the increase in the 121-Hz component in the velocity spectrum; the increase in overall velocity in the band from 10 to 1000 Hz indicates a developing fault.

ing condition of the machine (such as machine speed, load, and temperature) on the vibration parameter being measured varies greatly for different types of machines. Speed changes of up to 10 per cent usually can be compensated for, and spectra can be compared. If the speed changes are greater than this value, the operating condition of the machine should be considered to be different and a new reference spectrum used as a basis of comparison. The reference spectrum need not be measured when the machine is new (after allowing for a run-in period). The reference spectrum can be determined at any time during the life of a machine provided the vibrations are stable, since a stable spectrum is a sign of stable operation of the machine. The principal difficulty is to establish when changes in the spectrum are sufficiently large to warrant stopping the machine.

Most national and international standards for the measurement of bearing vibration do not consider frequency spectra; instead, they give values for vibration changes of the rms value of the velocity amplitude from 10 to 1000 Hz (or 10,000 Hz) for machines in good and bad condition. These ratios have successfully been transposed to characteristic components in the vibration spectrum such as unbalance or frequency. Usually, a change in the bearing vibration amplitude (measured in terms of acceleration, velocity, or displacement) on any characteristic component from the spectrum by a factor of 2 to 2.5 (6 to 8 dB in vibration level) is considered significant; a change by a factor of 8 to 10 (18 to 20 dB in vibration level) is considered critical, unless specified otherwise by the manufacturer. Limits for shaft vibration measurements, giving the relative motion of the shaft inside the bearing, directly relate to physical bearing clearance in the machine. The required time interval between measurements varies greatly from one machine to another and depends directly on the expected mean time between failure and the deterioration rate of the expected failures; therefore, measurements should be made more frequently as soon as incipient deterioration is noticed.

Successful fault detection in machinery is the first step toward a successful condition monitoring program. Early recognition of deterioration is the key to valuable fault diagnosis and efficient trend analysis. Consequently, this phase of condition monitoring should not be neglected, although sometimes it may seem tedious.

SPECTRUM INTERPRETATION AND FAULT DIAGNOSIS

Commercially available fast Fourier transform analyzers provide a suitable tool for spectrum interpretation. They provide constant bandwidth (on a linear frequency scale), and, by means of zoom or extended lines of resolution, they also provide very high resolution in any frequency range of interest. This permits (1) early recognition and separation of harmonic patterns or sideband patterns and (2) separation of closely spaced individual components. Fast Fourier transform analyzers also may provide diagnostic tools such as synchronous time averaging, cepstrum analysis, and/or use of the Hilbert transform for amplitude and phase demodulation (see Chap. 13).

Table 16.2 classifies different types of faults and indicates at which frequency the faults are displayed in a vibration spectrum. Although such a table is of considerable help in spectrum interpretation, any such simplified presentation must be used with care, as illustrated by the examples considered below. The various faults can be classified according to their spectral components, as follows.

SUBSYNCHRONOUS COMPONENTS

Subsynchronous components of vibration (at frequencies below the rotational speed of the machine) usually occur where sleeve bearings are used. The most common are the vibrations due to oil whirl, hysteresis whirl, resonant whirl, or mechanical looseness. These types of instability and nonlinear behavior are described in detail in Ref. 4. Figure 16.3 shows a spectrum measured on the journal bearing of a centrifugal compressor with mechanical looseness. A characteristic pattern of half-order harmonics of rotation speed can be clearly seen. Figure 16.4 shows a spectrum of the journal bearing of a pump in which a developing oil whirl shows up clearly at 21 Hz (42 percent of the rotation speed) and its second harmonic.

Both examples clearly indicate how the use of a linear frequency scale facilitates the diagnosis of the fault by providing a clear indication of the different types of harmonic patterns. High resolution is required to separate a half-order harmonic component due to looseness (exactly 50 percent of rotation speed) from a component due to oil whirl (42 to 48 percent of rotation speed).

LOW HARMONICS OF ROTATIONAL SPEED

Low harmonics of the rotational speed are generated by shaft unbalance, misalignment, and eccentricity, as well as cracks in shafts and bent shafts. These various faults may be difficult to distinguish, since they are mechanically related. A bad coupling may result in misalignment. A bent shaft results in unbalance. Even a well-known and well-defined fault such as unbalance may give misleading vibration components. The exciting fault due to eccentric masses is a centrifugal force (thus radial) rotating at the shaft speed and is therefore expected to result in a component in the vibration spectrum at the machine speed and in the radial direction. However, dynamic unbalance may also result in a rocking motion and consequently in vibration in both radial and axial directions. In the same way, if there is a nonlinear transmission path from the point where the force is applied to the point of measurement, a rise in the harmonics of the rotation speed can be observed in the vibration spectrum, due to distortion of the signal.

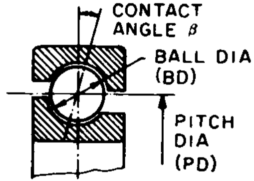
The phase relationship between bearings provides essential information for differentiating these various types of faults. As an example, unbalance will generate a rotating force, and therefore the phase relationship between bearings can be expected to be identical in both horizontal and vertical directions (in the absence of resonances). For mass unbalance, the phase difference between a vertical and horizontal transducer is 90° on the same bearing. Misalignment, however, does not create a rotating force, and thus the phase relationship between bearings in both vertical and horizontal directions can be vastly different.

HARMONICS OF THE POWER LINE FREQUENCY

Vibrational components, which are related to the frequency of the power line or variable frequency drive, or to the difference between the synchronous frequency and the rotational speed, occur in electric machines such as induction motors or generators. These vibrations are due to electromagnetically induced forces. These forces, which occur in the case of a malfunction in the electric machine, are related to the air gap between the rotor and the stator and to the current. The faults on the electric machine are due either to the stator (called *stationary faults*) or to the rotor (called *rotating faults*). They may originate from either a variation in the air gap or a variation in the current. Table 16.3 summarizes how these various faults show up in the low-frequency range of the vibration spectrum.[5]

Figure 16.5 shows a vibration signal measured on the rolling-element bearing of an asynchronous electric motor. By zooming in the region of the high-level 100-Hz

TABLE 16.2 A Vibration Troubleshooting Chart

Nature of fault	Frequency of dominant vibration, Hz = rpm/60	Direction	Remarks
Rotating members out of balance	$1 \times$ rpm	Radial	A common cause of excess vibration in machinery
Misalignment and bent shaft	Usually $1 \times$ rpm Often $2 \times$ rpm Sometimes 3 and $4 \times$ rpm	Radial and axial	A common fault
Damaged rolling element bearings (ball, roller, etc.)	Impact rates for the individual bearing component Also vibrations at high frequencies (2 to 60 kHz) often related to radial resonances in bearings	Radial and axial	Uneven vibration levels, often with shocks  Impact Rates f (Hz): For Outer Race Defect $f(\text{Hz}) = \frac{n}{2} f_r \left(1 - \frac{\text{BD}}{\text{PD}} \cos \beta\right)$ For Inner Race Defect $f(\text{Hz}) = \frac{n}{2} f_r \left(1 + \frac{\text{BD}}{\text{PD}} \cos \beta\right)$ For Ball Defect $f(\text{Hz}) = \frac{\text{PD}}{\text{BD}} f_r \left[1 - \left(\frac{\text{BD}}{\text{PD}} \cos \beta\right)^2\right]$ n = number of balls or rollers f_r = relative rps between inner and outer races
Journal bearings loose in housing	Subharmonics of shaft rpm, exactly ½ or ⅓ × rpm	Primarily radial	Looseness may only develop at operating speed and temperature (e.g., turbomachines)

Oil-film whirl or whip in journal bearings	Slightly less than half shaft speed (42 to 48 percent)	Primarily radial	Applicable to high-speed (e.g., turbo) machines
Hysteresis whirl	Shaft critical speed	Primarily radial	Vibrations excited when passing through critical shaft speed are maintained at higher shaft speeds. Can sometimes be cured by tightening the rotor components.
Damaged or worn gears	Tooth-meshing frequencies (shaft rpm $\times$ number of teeth) and harmonics	Radial and axial	Sidebands around tooth-meshing frequencies indicate modulation (e.g., eccentricity) at frequency corresponding to sideband spacings. Normally only detectable with very narrow-band analysis and cepstrum analysis.
Mechanical looseness	$2 \times$ rpm		Also sub- and interharmonics, as for loose journal bearings
Faulty belt drive	1, 2, 3, and $4 \times$ rpm of belt	Radial	The precise problem can usually be identified visually with the help of a stroboscope
Unbalanced reciprocating forces and couples	$1 \times$ rpm and/or multiples for higher-order unbalance	Primarily radial	
Increased turbulence	Blade & vane passing frequencies and harmonics	Radial and axial	An increased level indicates increased turbulence
Electrically induced vibrations	$1 \times$ rpm or 2 times line frequency	Radial and axial	Should disappear when power turned off

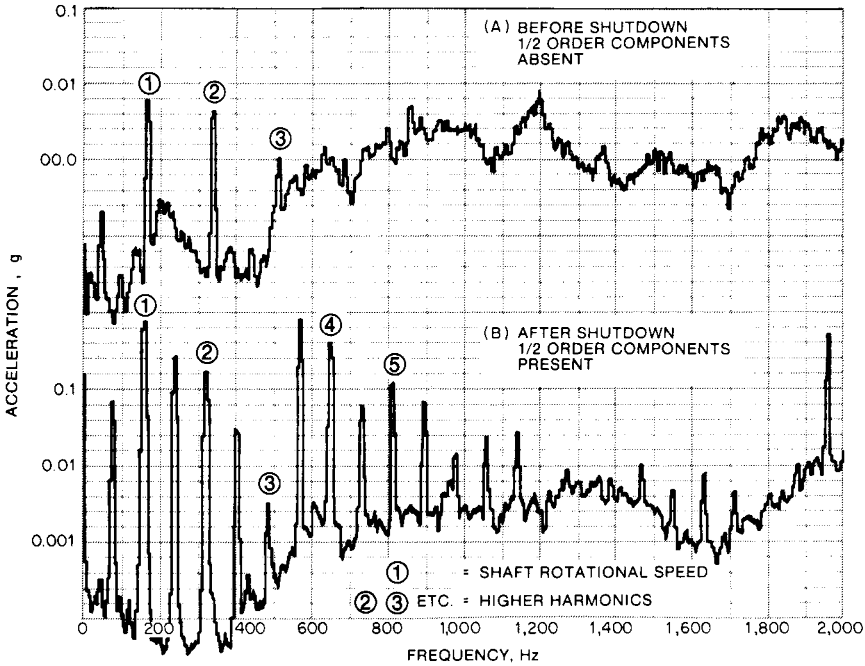


FIGURE 16.3 Acceleration spectra of a journal bearing on a centrifugal compressor. (*A*) Compressor in *good condition.* Before shutdown, the vibration pattern is normal with few harmonics of the compressor's rotation speed and broadband noise at higher frequencies due to inherent turbulences. (*B*) Compressor *with looseness* in the journal bearing. After shutdown, the higher-order harmonics have an increased amplitude, and the presence of half-order harmonics can be observed.

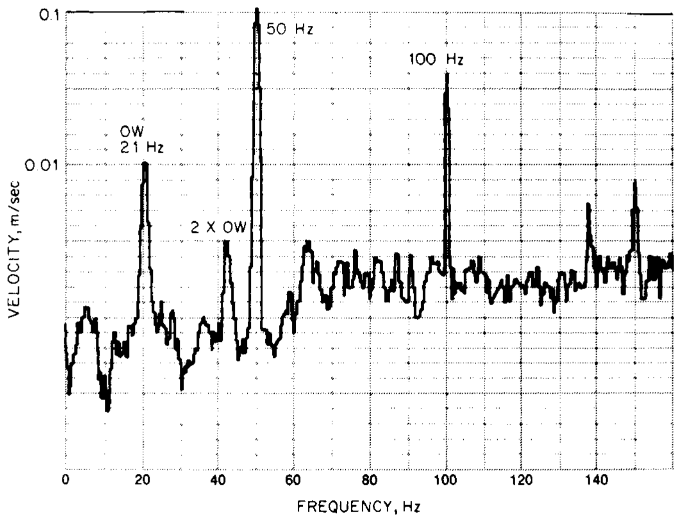


FIGURE 16.4 Spectrum analysis showing component due to oil whirl at 42 percent of the rotation speed measured on the journal bearing of a pump.

TABLE 16.3 "Rotating" and "Stationary" Magnetically Induced Vibrations in Induction Motors

Type of problem	Symptomatic frequency of vibration	Typical cause	
		Air-gap variations	Current variations
Stationary	2 × line frequency	Static eccentricity, weakness of stator support	Stator winding faults
Rotating	1 × rpm with 2 × slip-frequency sidebands	Dynamic eccentricity, bent rotor, loose rotor bar(s)	Broken or cracked rotor bar(s) or shorted rotor laminations

component (i.e., twice the line frequency in Europe), this component can be diagnosed as the pole-passing frequency of 100 Hz and not the 2 × rotation speed at 99.6 Hz which could have been an indication of a faulty alignment. This demonstrates the value of being able to zoom to the frequency region containing the component of interest. The zoom or extended lines of resolution provides sufficient resolution to separate closely spaced components. It is of no help in analyzing synchronous machines or generators, since the rotation speed and the line (mains) frequency are identical. In such a case, the machine should be permitted to coast to a stop. When the power is cut, electrically induced components of vibration disappear, and the harmonics of the rotation speed gradually decrease in frequency and amplitude.

Vibration forces resulting from an effective variation of the reluctance in the magnetic circuit as a function of the rate of the stator and rotor slot passing will be

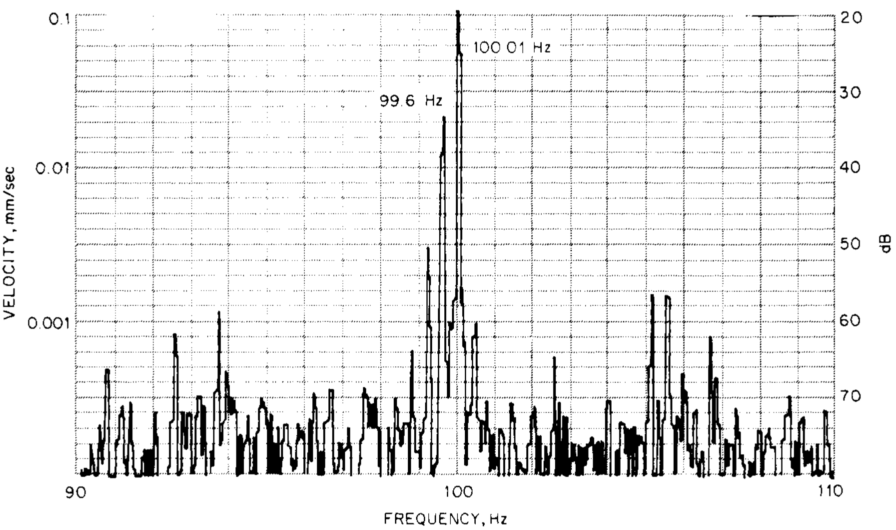


FIGURE 16.5 Spectrum analysis employing zoom frequency analysis around the 100-Hz component, measured on the rolling-element bearing of an asynchronous electric motor. Note the simultaneous presence of the pole-passing frequency (100 Hz) and the second harmonic of rotation speed (99.6 Hz). A lesser resolution would not permit a separation.

present even in a motor which is in good condition. These vibrations occur at the slot harmonics given by the following equation:

$$f_{\text{slot}} = R_s f_{\text{rot}} \pm k f_{\text{line}}$$

where f_{slot} = slot passing frequency
R_s = number of rotor slots
f_{rot} = rotating speed
k = zero or even number
f_{line} = power line frequency

The vibration components at low frequency differentiate between stator problems and rotor problems. They do not, however, indicate whether the faults originate from variations in air gaps or current. The components at the slot harmonics, on the other hand, will behave differently depending on whether the fault originates from an air gap or current variation as indicated in Table 16.4.

Figure 16.6 shows that by using a zoom around slot harmonics, sidebands can be observed at twice the slip frequency, thereby permitting the diagnosis of broken rotor bars. For a four-pole motor, sidebands occur at four times the slip frequency.

As an alternative to using signal analysis of vibration, signal analysis of the motor current may be used to monitor certain types of problems. It is a more direct measurement for all electrical problems and, with the help of algorithms, makes it possible, for example, to determine with a certain amount of accuracy the number of broken rotor bars. Reference 6 mentions that mechanical phenomena such as worn gears, tooth wear, and steam packing degradation (in motor-operated valves) can be detected as well. It also mentions the applicability of this technique to dc motors.

HIGHER HARMONICS OF THE ROTATIONAL SPEED

Higher harmonics of the rotational speed typically occur where characteristic frequencies are an integral multiple of the rotational speed of the machine, for example, in the case of gearboxes, compressors, and turbines, where vibration occurs in multiples of the number of teeth, blades, lobes, etc. An increase in components, such as tooth-meshing frequencies or blade-passing frequencies, indicates deterioration acting on all teeth or blades, e.g., as uniform wear or increased turbulences, respectively.

"Ghost components" sometimes are observed in vibration spectra obtained from measurements on gearboxes; these components appear as tooth-meshing frequencies, but at frequencies where no gear in the gearbox has the corresponding number of teeth. Such components arise from faults on the gear-cutting equipment which have been transmitted to the new gear. Being geometrical faults, they are not load-sensitive, nor do they increase with wear; rather, as the gear's surface wears, they tend to decrease with time. The frequencies of the components are an integral multiple of the number of teeth on the index wheel and therefore appear as harmonics of the speed of rotation of the faulty gear.

SIDEBAND PATTERNS DUE TO MODULATION

Modulations, frequently seen in vibration measurements on gearboxes, are caused by eccentricities, varying gear-tooth spacing, pitch errors, varying load, etc. Such modulations manifest themselves as families of sidebands around the gear-tooth-meshing frequency with a frequency spacing equal to the modulating frequency

TABLE 16.4 Troubleshooting Guide of Induction Motor Vibrations

Static eccentricity	$2 \times$ line frequency and components at $\omega \times [nR_s(1-s)/p \pm k_1]$	Radial	Can result from poor internal alignment, bearing wear, or from local stator heating (vibration worsens as motor heats up).
Weakness/looseness of stator support, unbalanced phase resistance or coil sides Shorted stator laminations/turns	$2 \times$ line frequency	Radial	Referred to as "loose iron." Difficult to differentiate between this group using only vibration analysis, but they will also be apparent at no load as well as on load.
Loose stator laminations	$2 \times$ line frequency and components spaced by $2 \times$ line frequency at around 1 kHz	Radial	Can have high amplitude but not usually destructive. The high-frequency components may be similar to static eccentricity.
Dynamic eccentricity	$1 \times$ rpm with $2 \times$ slip-frequency sidebands and components at $\omega \times [((nR_s \pm k_e) \times (1-s)/p) \pm k_1]$	Radial	Can result from rotor bow, rotor runout, or from local rotor heating (vibration worsens as motor heats up).
Broken or cracked rotor bar Loose rotor bar Shorted rotor laminations Poor end-ring joints	$1 \times$ rpm with $2 \times$ slip-frequency sidebands and components similar to those given above for dynamic eccentricity with addition of $2 \times$ slip-frequency sidebands around slot harmonics	Radial	The slip sidebands may be low level, requiring a large dynamic range as well as frequency selectivity in measuring instrumentation. Typical spectra show that these components in the region of the principal vibration slot harmonics also have slip-frequency sidebands.

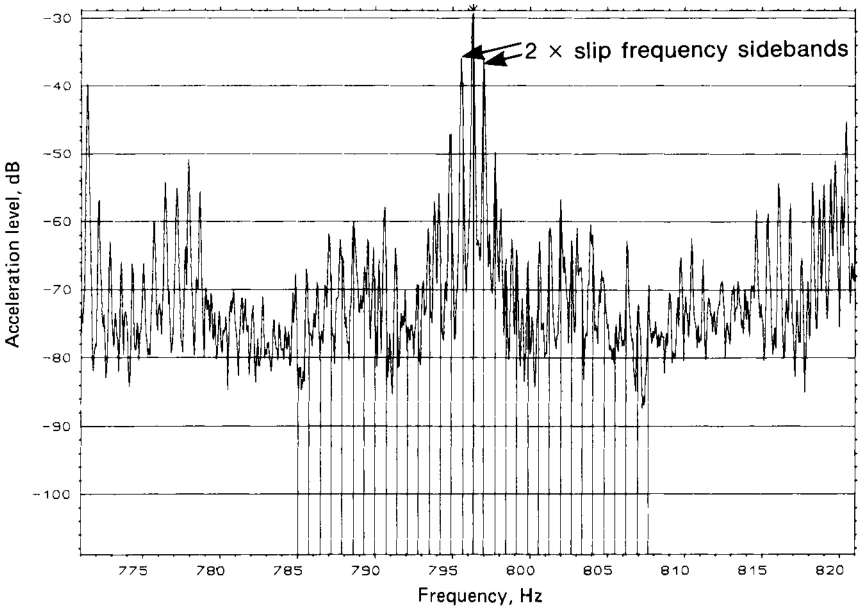


FIGURE 16.6 Zoom spectrum centered around the second principal vibration slot harmonic, showing 2 × slip-frequency sidebands on the component at this frequency.

(e.g., the rotation speed of the faulty gear in the case of an eccentric gear). Figure 16.7*A* shows the distribution of the sidebands for such a condition. Any gear in a gearbox can be a source of modulation. In order to distinguish all possible sidebands, the analysis must be carried out with sufficient resolution to detect sidebands with a spacing equal to even the lowest rotational speed inside the gearbox, and therefore the zoom feature is indispensable.

Local faults, such as cracked or broken gear teeth, also appear as a family of sidebands with a spacing equal to the rotation speed of the faulty gear, as this induces a change in tooth deflection, during meshing, once per revolution. The sidebands shown in Fig. 16.7*B* are low in level and cover a broad frequency range. Very often the influence of the transmission path will modify the shape of the sideband pattern and does not permit a precise diagnosis.[7] Local faults are best detected in the time waveform of Fig. 16.7*B*.[9] Similarly, sidebands at the rotational speed and slip frequency are quite common in patterns for asynchronous machines.

HARMONIC PATTERNS NOT HARMONICALLY RELATED TO THE ROTATIONAL SPEED

Harmonic patterns which are not harmonically related to the speed of rotation typically appear where there are local faults in rolling-element bearings.[9,10] A local fault produces an impulse having a repetition rate equal to the characteristic frequencies of the bearing: ball-passing frequency for the outer raceway, ball-passing frequency for the inner raceway, and twice the ball-spin frequency (see Table 16.2). Such faults appear as a series of harmonics separated by the impact frequency with an ampli-

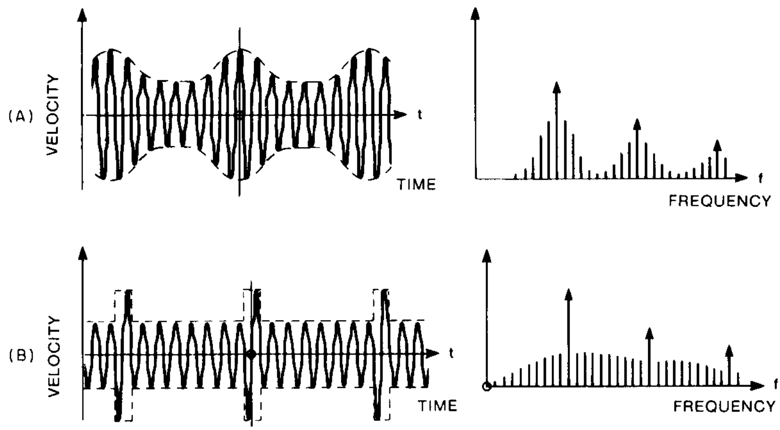


FIGURE 16.7 Distribution of sideband patterns for distributed and local faults on a gear. (*A*) *In the case of a distributed fault,* sidebands have a high level and are grouped around the tooth-meshing frequency and harmonics with a spacing equal to the speed of the faulty gear. The time waveform (lower) shows bursts of energy where the fault passes through the gear meshing area. (*B*) *In the case of a local fault* (such as a cracked or broken tooth), the sidebands have a low level and expand widely over a large frequency range. However, the time waveform shows definite evidence of a chipped tooth. (*Eshleman.*[8])

tude proportional to the spectrum of the single impulse. As illustrated in Fig. 16.8, such an impact tends to excite bearing defect frequencies or excite structural resonances in the frequency range covered, and the harmonic patterns around these resonances thus are emphasized. This provides two methods of detecting rolling-element bearing faults: (1) by finding the fundamental of the impact rate in the low-frequency range; and (2) by finding the harmonic pattern at the impact frequency in the high-frequency range, where resonances are excited; this may be difficult because speed fluctuations tend to smear these components.

SPECIAL ANALYSIS TECHNIQUES

Table 16.5 summarizes the applications of the various analysis techniques described below.

ENVELOPE DETECTION

Envelope detection (envelope detectors are discussed in Chap. 13) is particularly useful for fault diagnosis in machinery, since it permits elimination of the signal resulting from background vibration and concentrates the analysis in the frequency range placing the greatest emphasis on the harmonic pattern of the impact frequency—a resonance of the structure excited by the impulse. This can be done by either analog or digital means.[11] Figure 16.9 illustrates the analog process. The signal is first bandpass-filtered around the frequency range where a significant broadband increase has been detected, as illustrated in Fig. 16.9*B* and *D* (usually one or more resonances between 2 and 20,000 Hz have been excited). The filtered signal (which

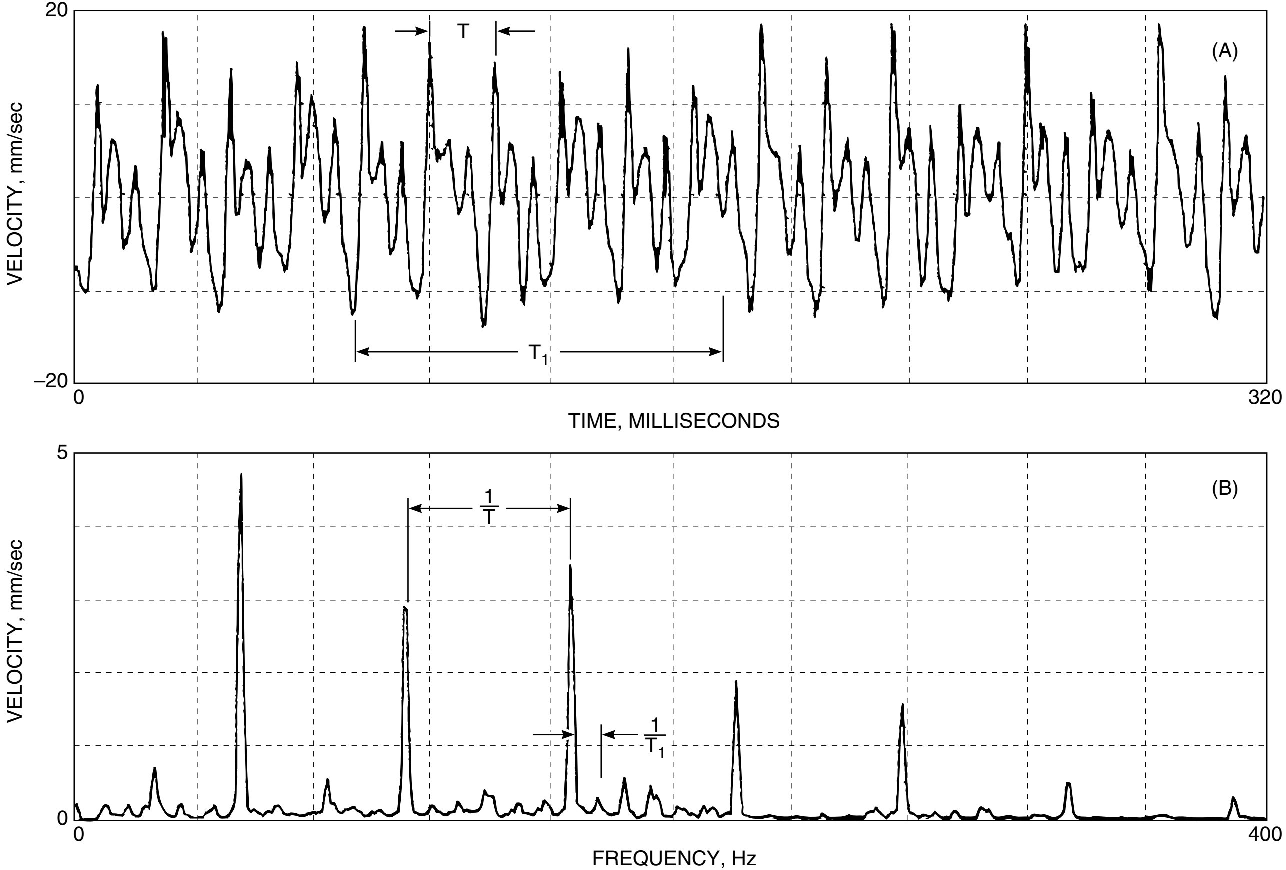


FIGURE 16.8 Effect of a local fault in a rolling-element bearing. (*A*) *In the time domain,* there is a repeated impact having a period *T.* (*B*) *In the frequency domain,* the repeated impact results in a line spectrum containing multiple bearing frequencies with impact rate ($1/T_1$) sidebands, where ($1/T_1$) is the operating speed of the unit. (*Eshleman.*[8])

TABLE 16.5 Typical Applications of the Various Analysis Techniques

Technique	Application	Fault/machine
Zoom	Separation of closely spaced components Improvement of signal-to-noise ratio, separation of resonances from pure tones	Electrical machines, gearboxes, turbines
Phase	Operational deflection shapes Detection of developing cracks in shafts Balancing	
Time signal	Waveform visualization for identification of distortion	Rubbing, impacts, clipping, cracked teeth
Cepstrum	Identification and separation of families of harmonics Identification and separation of families of sidebands	Rolling elements bearing, bladed machines, gearboxes
Envelope analysis	Amplitude demodulation Observation of a low-frequency amplitude modulation happening at high frequency	Rolling element bearing, electrical machines, gearboxes
Dynamic crest factor	Calculation of high-pass filtered signals	Faults in low-speed machines
Synchronous time averaging	Improving signal-to-noise ratio Waveform analysis Separating effects of adjacent machines Separating effects of different shafts Separating electrically and mechanically induced vibrations	Electrical machines, reciprocating machines, gearboxes, etc.
Impact testing	Resonance testing	Foundations, bearings, couplings, gears
Scan analysis	Analysis of nonstationary signals	Fast run-up/coast down

now contains only the ringing of the selected resonance excited by the repetitive impacts, Fig. 16.9*C*) is rectified and analyzed once again in a low-frequency range in order to determine the repetition frequency of the impacts, as shown in Fig. 16.9*E* and *F*.

The advantages of envelope detection are as follows:

1. The use of bandpass filters eliminates background noise resulting from other vibration sources (for example, from unbalance or gear vibration). All that remains is the repetition rate of the impacts exciting the structural resonance, possibly amplitude-modulated.
2. High-frequency analysis is not required, since only the envelope of the signal is of importance, not the signal itself, which can extend upward to hundreds of kilohertz.
3. Diagnosis is possible, since the impact frequencies are determined and can be related to a specific source (ball-passing frequency for the outer raceway, ball-

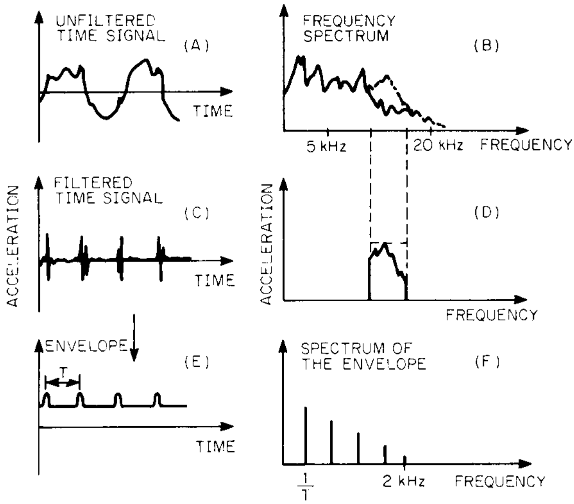


FIGURE 16.9 Principle of analog envelope detection applied to the analysis of impacts due to rolling-element bearing faults. (*A*) *Unfiltered time signal.* (*B*) *The corresponding spectrum in the frequency domain.* (The dotted spectrum represents the reference spectrum before the fault developed. Note the broadband increase due to excitation of a resonance by the bearing fault.) (*C*) *Frequency spectrum after application of a bandpass filter* in range where the change caused by a ball-bearing fault has been detected. (*D*) *Time signal which corresponds with C;* contains ringing of a resonance which is excited periodically. (*E*) Envelope of time signal from *D.* (*F*) *Low-frequency analysis of the envelope from E,* yielding the impact rate due to the fault.

passing frequency for the inner raceway, ball-spin frequency, fundamental train frequency, or some other source of repetitive impacts, for example, a cracked gear tooth).

Figure 16.10*A* and *B* shows the acceleration spectra from 0 to 25 kHz of a good bearing and a faulty bearing. Note that the spectrum is noticeably higher on the good bearing than on the faulty one, which confirms that comparative measurements should not be made between different measurement points or different machines. Absolute vibration levels do not provide a satisfactory indication of the condition of a machine; only changes in level are relevant. Any simple method of bearing fault detection such as shock pulse measurement, spike energy, kurtosis, or crest factor was difficult to use on this specific machine, because a forced-lubrication system gave repetitive pulses at a frequency of 5.4 Hz, independent of the rotating speed, which dominated the whole vibration signal. Figure 16.10*C* and *D* shows the analysis of the envelopes on the good and the faulty bearings obtained after zooming around 5400 Hz with an 800-Hz frequency span. The only noticeable pattern on the good bearing comes from the forced lubrication system. In contrast, the result of the envelope analysis on the faulty bearings shows a complex pattern, and frequency information is absolutely necessary to confirm whether or not there is a ball-bearing fault. The following frequencies appear: 5.4 Hz (the repetition rate of the forced

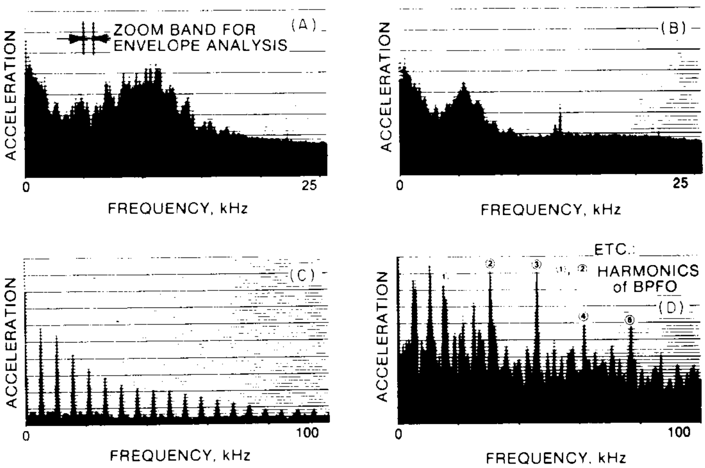


FIGURE 16.10 (*A*) Acceleration spectrum of a good bearing in the frequency range from 0 to 25 kHz. (*B*) Acceleration spectrum of a faulty bearing in the frequency range from 0 to 25 kHz. (*C*) Envelope spectrum in 100-Hz range of the signal in *A;* the zoom analysis is centered at 5408 Hz and has a frequency span of 800 Hz. (*D*) Envelope spectrum in 100-Hz range of the signal in *B;* the zoom analysis is centered at 5408 Hz and has a frequency span of 800 Hz.

lubrication system on the actual bearing, and its harmonics), 6.4 Hz (the repetition range of the forced lubrication system on adjacent bearings, and its harmonics), and 15.43 Hz (the ball-passing frequency for an outer raceway defect, and its harmonics).

APPLICATION OF CEPSTRUM ANALYSIS

The use of cepstrum analysis (explained in more detail in Chap. 13) is particularly advantageous for detecting periodicities in the power spectrum (e.g., harmonics and sideband patterns), since it provides a precise measure of the frequency spacing between components.[10,11] Figure 16.11 shows the spectrum and the corresponding cepstrum analysis of a measurement made on an auxiliary gearbox driving a generator on a gas-turbine-driven oil pump. As a fault on one of the bearing develops, the first rahmonic appears and then increases at a quefrency equal to the reciprocal of the spacing in the frequency spectrum which corresponds to an outer raceway defect in one of the bearings. Another advantage of cepstrum analysis is that one component in the cepstrum represents the global "power" content of a whole family of harmonics or sidebands, and this value is practically independent of extraneous factors such as machine-load condition, selection of measurement location, and phasing between amplitude and phase modulation.

Figure 16.12 shows the evolution in terms of time of two different frequency components in the spectra of Fig. 16.11. Figure 16.12*A* represents the evolution of the harmonic component at 7640 Hz, which shows a clear ascending slope, and also the evolution of the harmonic component at 5620 Hz, which shows a rather horizontal slope. Figure 16.12*B* represents the changes on the first rahmonic in the cepstrum and shows the effective evolution of the fault corresponding to a steep increase followed by stabilization as is typical for a spall in a rolling-element bearing.

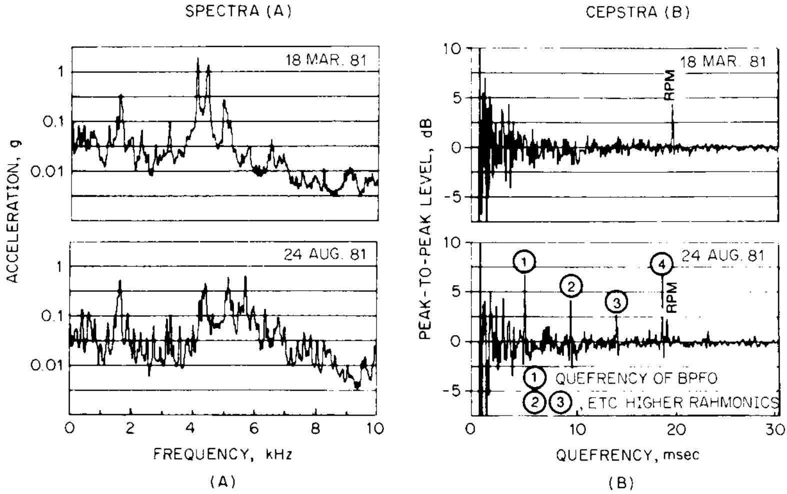


FIGURE 16.11 Analyses of vibration of an auxiliary gearbox before and after the development of a fault on one of the bearings.[10] (*A*) Spectrum analysis; (*B*) the corresponding cepstrum analysis.

Envelope detection and cepstrum analysis make it possible to determine the frequencies involved precisely and are thus useful for condition monitoring of industrial machines. They permit a reliable diagnosis of defects. Also, cepstrum analysis appears to be an invaluable condition-related parameter for following fault development (e.g., those that develop in rolling-element bearings) that shows up as a family of harmonics or sidebands.

The use of post-processing techniques makes it possible to manipulate the cepstrum, for instance, by removing selected parts of the cepstrum and then transforming the cepstrum back to the frequency and/or time domain. For example, such manipulations make it possible (1) to evaluate the size of a spall in a rolling element bearing and (2) to reconstitute the transfer function between the input force and the vibration response without measuring the input function.[11,12]

APPLICATION OF GATED VIBRATION ANALYSIS ON RECIPROCATING MACHINES

Vibration signals from reciprocating machines (such as diesel engines, reciprocating compressors, hydraulic pumps, and gas engines) differ from those of rotating machines in that they are not stationary. Instead, they consist of short impulses which occur at different points in time for different events (valves opening and closing, piston slap, combustion, etc.) and are repeated with the same timing for each new machine cycle. If these signals are averaged over a longer period of time, as is common practice in the analysis of rotating machines, these individual events would be averaged out so that changes would go undetected.

In reciprocating machines, different events will excite different resonances of a structure; the resulting frequencies that are generated provide valuable diagnostic information. *Timing* provides equally valuable information because the time when an event occurs may be related to what is actually happening in the cycle of the engine.

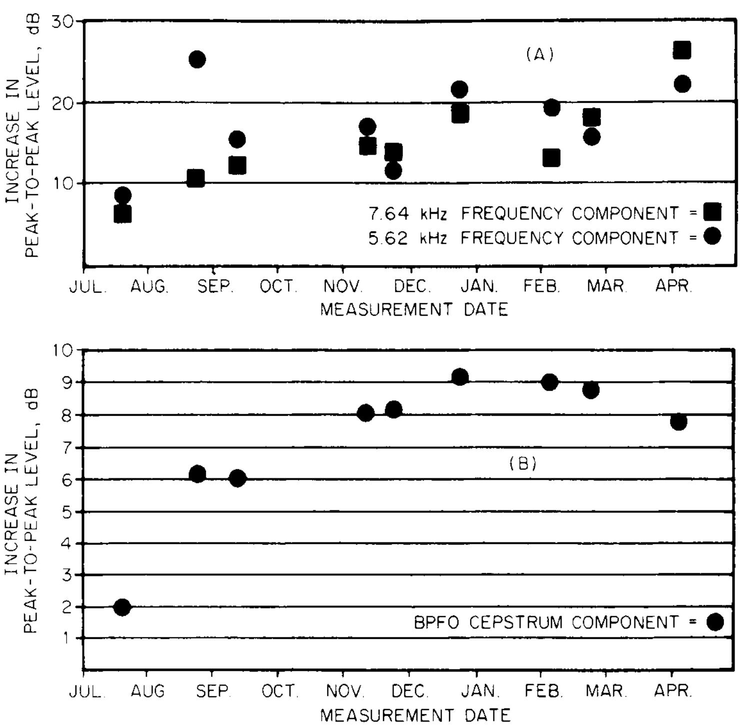


FIGURE 16.12 (*A*) Development of the fault illustrated in Fig. 16.13, using the harmonic at 7640 Hz and the harmonic at 5620 Hz as the parameter. (*B*) The corresponding trend curve using the cepstrum component as the parameter, showing a smooth evolution.[10]

In gated vibration analysis, the vibration signal is analyzed at various angles of the crankshaft in order to cover a complete cycle of the machine in a three-dimensional plot.[13] The analyzer is triggered by a once-per-cycle trigger signal; then the delay after triggering is shifted to provide adequate overlap; this procedure continues until a complete cycle is covered. Note that each spectrum represents actually an average over many machine cycles for one time delay. This process averages any differences between machine cycles.

An alternative to gated vibration analysis is gated sound intensity analysis, which is more suited to noise control and quality checks.[14]

TREND ANALYSIS

Trend analysis makes use of graphs of a condition-related parameter versus time (date or running hours) to determine when the parameter is likely to exceed a given limit. The goal of a successful condition monitoring program is to predict the time of an expected breakdown well in advance of its occurrence in order to shut down the machine in ample time, to order spare parts, and thereby to minimize the shutdown time. Since all vibration criteria indicate that equal changes on a log scale correspond to equal changes in severity, data for a trend analysis should be plotted on a logarithmic scale in decibels. A linear trend on a logarithmic scale is found occasionally, but the actual trend may follow another course; for example, when the fault feeds back on the rate of deterioration (e.g., gear wear), the trend, when plotted on a logarithmic scale, may then be exponential. In some cases the fault changes suddenly in finite steps (for example, a spall caused by gradual subsurface fatigue), making it very difficult to extrapolate to determine the date of the shutdown. To ensure accurate trend analysis, the following precautions should be taken:

1. Determine a trend based on measurements of a parameter directly related to a specific type of fault—not on measurements of overall levels.
2. Diagnose faults *before* attempting to interpret a trend curve in order to (*a*) select the appropriate parameter for the type of fault which is being monitored (for example, the parameter may be the level of an individual component, of a cepstrum component, or of a selected frequency range) and (*b*) observe critically the results of the trend analysis so as to determine if the linear or exponential interpolation is adequate.
3. Keep in mind that the best estimate of the lead time will be obtained by employing a trend of the most recent measurements.

USE OF COMPUTERS IN CONDITION MONITORING PROGRAMS

Computers can be of great help in a condition monitoring program in handling, filing, and storing data and in performing tedious computations such as spectrum comparison and trend analysis. A condition monitoring system which incorporates a computer includes:

1. A *recording device* for storing the analog or digital time signals or frequency spectra. In a permanently installed monitoring system, the analog time signal is directly connected to the following items.
2. An *analyzer* with both fast Fourier transform (FFT) narrowband analysis and advanced diagnostic techniques (zoom, cepstrum) for diagnostics.
3. A *computer and appropriate software* which provide (*a*) management of the measurement program, including route mapping, storage of reference spectra/cepstra, and new spectra/cepstra; (*b*) a comparison of spectra and a printout of significant changes; and (*c*) trend analysis of any chosen parameter (individual component or overall level in a given frequency range). In a permanent monitoring system, the complete process (i.e., a new analysis) is performed automatically at a predetermined rate, which is adjusted as the fault develops.

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