1

Introduction and Background

1.1 Introduction

Machine condition monitoring is an important part of condition-based maintenance (CBM), which is becoming recognized as the most efficient strategy for carrying out maintenance in a wide variety of industries. Machines were originally 'run to break', which ensured maximum operating time between shutdowns, but meant that breakdowns were occasionally catastrophic, with serious consequences for safety, production loss and repair cost. The first response was 'preventive maintenance', where maintenance is carried out at intervals such that there is a very small likelihood of failure between repairs. However, this results in much greater use of spare parts, as well as more maintenance work than necessary.

There is now a considerable body of evidence that CBM gives economic advantages in most industries. An excellent survey of the development of maintenance strategies is given by Rao in a keynote paper at a recent Comadem (Condition Monitoring and Diagnostic Engineering Management) conference [1]. Maintenance is often regarded as a cost centre in many companies, but Al-Najjar *et al.* [2, 3] have long promoted the idea that CBM can convert maintenance to a profit centre. Jardine *et al*. [4, 5] from the University of Toronto have documented a number of cases of savings given by the use of CBM. The case presented in [5], from the Canadian pulp and paper industry, is discussed further in Chapter 6, in connection with the authors' approach to prognostics.

To base maintenance on the perceived condition of operating machines (many of which are required to run continuously for 12 months or more) requires that methods are available to determine their internal condition while they are in operation. The two main ways of getting information from the inside to the outside of operating machines are vibration analysis and lubricant analysis, although a few other techniques are also useful.

This chapter includes a description of the background for and methods used in condition monitoring, while most of the rest of the book is devoted solely to the methods based on vibration analysis, which are the most important. This chapter describes the various types of vibration measurement used in condition monitoring, and the transducers used to provide the corresponding vibration signals. It also describes the basic problem in interpretation of vibration signals, in that they are always a compound of forcing function effects (the source)

Vibration-based Condition Monitoring: Industrial, Aerospace and Automotive Applications   Robert Bond Randall
© 2011 John Wiley & Sons, Ltd

and transfer function effects (the structural transmission path), and how the two effects may be separated.

1.2 Maintenance Strategies

As briefly mentioned above, the available maintenance strategies are broadly:

1. **Run-to-Break**. This is the traditional method where machines were simply run until they broke down. This in principle gives the longest time between shutdowns, but failure when it does occur can be catastrophic and result in severe consequential damage, for example of components other than the ones that failed, and also of connected machines. As a result, the time to repair can be greatly increased, including the time required to obtain replacement parts, some of which might be major items and take some time to produce. In such a case, the major cost in many industries would be production loss, this often being much greater than the cost of individual machines. There is still a place for run-to-break maintenance, in industries where there are large numbers of small machines, for example sewing machines, where the loss of one machine for a short time is not critical to production, and where failure is unlikely to be catastrophic.

2. **(Time-Based) Preventive Maintenance**. Maintenance is done at regular intervals which are shorter than the expected 'time between failures'. It is common to choose the intervals to be such that no more than 1–2% of machines will experience failure in that time. This does mean that the vast majority could have run longer by a factor of two or three [6]. The advantage of this method is that most maintenance can be planned well in advance and that catastrophic failure is greatly reduced. The disadvantages, in addition to the fact that a small number of unforeseen failures can still occur, are that too much maintenance is carried out and an excessive number of replacement components consumed. This approach has been known to cause reduced morale in maintenance workers (who are aware that most of the time they are replacing perfectly good parts) so that their work suffers and this can give rise to increased 'infant mortality' of the machines, by introducing faults which otherwise never would have happened. Time-based preventive maintenance is appropriate where the time to failure can be reasonably accurately predicted, such as where it is based on well-defined 'lifing' procedures, which can predict the fatigue life of crucial components on the basis of a given operational regime. Some components do tend to wear or fatigue at a reasonably predictable rate, but with others, such as rolling element bearings, there is a large statistical spread around the mean, leading to estimates such as the one given above, where the mean time to failure is two to three times the minimum [6].

3. **Condition-Based Maintenance (CBM)**. This is also called 'predictive maintenance' since the potential breakdown of a machine is predicted through regular condition monitoring and maintenance is carried out at the optimum time. It has obvious advantages compared with either run-to-break or preventive maintenance, but does require having access to reliable condition monitoring techniques, which not only are able to determine current condition, but also give reasonable predictions of remaining useful life. It has been used with some success for 30–40 years, and for example the above-mentioned report [6] by Neale and Woodley predicted back in 1978 that maintenance costs in British industry could be reduced by approximately 65% by appropriate implementation in a number of industries that they

identified. However, the range of monitoring techniques was initially quite limited, and not always correctly applied, so it is perhaps only in the last 15 years or so that it has become recognized as the best maintenance strategy in most cases. Initially the greatest successes were attained in industries where machines were required to run for long periods of time without shutting down, such as the power generation and (petro)chemical industries. The machines in such industries typically run at near constant speed, and with stable load, so the technical problems associated with the condition monitoring were considerably reduced. As more powerful diagnostic techniques have become available, it has been possible to extend condition monitoring to other industries in which the machines have widely varying speed and load, and are perhaps even mobile (such as ore trucks in the mining industry). The potential benefits given by CBM applied to hydroelectric power plants and wind turbines are discussed in [7, 8], respectively.

This book aims at explaining a wide range of techniques, based on vibration analysis, for all three phases of machine condition monitoring, namely fault detection, fault diagnosis and fault prognosis (prediction of remaining useful life).

1.3   Condition Monitoring Methods

Condition monitoring is based on being able to monitor the current condition and predict the future condition of machines while in operation. Thus it means that information must be obtained externally about internal effects while the machines are in operation.

The two main techniques for obtaining information about internal conditions are:

1. *Vibration Analysis*. A machine in standard condition has a certain vibration signature. Fault development changes that signature in a way that can be related to the fault. This has given rise to the term 'mechanical signature analysis' [9].
2. *Lubricant Analysis*. The lubricant also carries information from the inside to the outside of operating machines in the form of wear particles, chemical contaminants, and so on. Its use is mainly confined to circulating oil lubricating systems, although some analysis can be carried out on grease lubricants.

Each of these is discussed in a little more detail in the following, along with a couple of other methods, performance analysis and thermography, that have more specialized applications.

1.3.1   Vibration Analysis

Even in good condition, machines generate vibrations. Many such vibrations are directly linked to periodic events in the machine's operation, such as rotating shafts, meshing gearteeth, rotating electric fields, and so on. The frequency with which such events repeat often gives a direct indication of the source and thus many powerful diagnostic techniques are based on frequency analysis. Some vibrations are due to events that are not completely phase locked to shaft rotations, such as combustion in IC (internal combustion) engines, but where a fixed number of combustion events occur each engine cycle, even though not completely repeatable. As will be seen, this can even be an advantage, as it allows such phenomena to be separated

from perfectly periodic ones. Other vibrations are linked to fluid flow, as in pumps and gas turbines, and these also have particular, quite often unique, characteristics. The term 'vibration' can be interpreted in different ways, however, and one of the purposes of this chapter is to clarify the differences between them and the various transducers used to convert the vibration into electrical signals that can be recorded and analysed.

One immediate difference is between the absolute vibration of a machine housing and the relative vibration between a shaft and the housing, in particular where the bearing separating the two is a fluid film or journal bearing. Both types of vibration measurement are used extensively in machine condition monitoring, so it is important to understand the different information they provide.

Another type of vibration which carries diagnostic information is torsional vibration, that is, angular velocity fluctuations of the shafts and components such as gears and rotor discs.

All three types of vibration are discussed in this chapter, and the rest of the book is devoted to analysing the resulting vibration signals, though overwhelmingly from accelerometers (acceleration transducers) mounted on the machine casing.

It should perhaps be mentioned that a related technique, based on measurement of acoustic emission (AE), has received some attention and is still being studied. The name derives from high-frequency solid-borne rather than airborne acoustic signals from developing cracks and other permanent deformation, bursts of stress waves being emitted as the crack grows, but not necessarily otherwise. The frequency range for metallic components is typically 100 kHz to 1 MHz, this being detected by piezoelectric transducers attached to the surface.

One of the first applications to machine diagnostics was to detection of cracks in rotor components (shafts and blades) in steam turbines, initiated by the Electric Power Research Institute (EPRI) in the USA [10]. Even though EPRI claimed some success in detecting such faults on the external housing of fluid film bearings, the application does not appear to have been developed further. AE monitoring of gear fault development was reported in [11], where it was compared with vibration monitoring. The conclusion was that indications of crack initiation were occasionally detected a day earlier (in a 14 day test) than symptoms in the vibration signals, but the latter persisted because they were due to the presence of actual spalls, while the AE was only present during crack growth. Because of the extremely high sampling rate required for AE, huge amounts of data would have to be collected to capture the rare burst events, unless recording were based on event triggering. In [12], AE signals are compared with vibration signals (and oil analysis) for gear fault diagnostics and prognostics, but the AE sensors had to be mounted on the rotating components and signals extracted via slip rings. Because of the difficulty of application of AE monitoring to machine condition monitoring, it is not discussed further in this book, although new developments may change the situation.

1.3.2   Oil Analysis

This can once again be divided into a number of different categories:

1. *Chip Detectors*. Filters and magnetic plugs are designed to retain chips and other debris in circulating lubricant systems and these are analysed for quantity, type, shape, size, and so on. Alternatively, suspended particles can be detected in flow past a window.

2. ***Spectrographic Oil Analysis Procedures (SOAP)***. Here, the lubricant is sampled at regular intervals and subjected to spectrographic chemical analysis. Detection of trace elements can tell of wear of special materials such as alloying elements in special steels, white metal or bronze bearings, and so on. Another case applies to oil from engine crankcases, where the presence of water leaks can be indicated by a growth in NaCl or other chemicals coming from the cooling water. Oil analysis also includes analysis of wear debris, contaminants and additives, and measurement of viscosity and degradation. Simpler devices measure total iron content.

3. ***Ferrography***. This represents the microscopic investigation and analysis of debris retained magnetically (hence the name) but which can contain non-magnetic particles caught up with the magnetic ones. Quantity, shape and size of the wear particles are all important factors in pointing to the type and location of failure.

Successful use of oil analysis requires that oil sampling, changing and top-up procedures are all well defined and documented. It is much more difficult to apply lubricant analysis to grease lubricated machines, but grease sampling kits are now available to make the process more reliable.

1.3.3 Performance Analysis

With certain types of machines, performance analysis (e.g. stage efficiency) is an effective way of determining whether a machine is functioning correctly.

One example is given by reciprocating compressors, where changes in suction pressure can point to filter blockage, valve leakage could cause reductions in volumetric efficiency, and so on. Another is in gas turbine engines, where there are many permanently mounted transducers for process parameters such as temperatures, pressures and flowrates, and it is possible to calculate various efficiencies and compare them with the normal condition, so-called 'flow path analysis'.

With modern IC engine control systems, for example for diesel locos, electronic injection control means that the fuel supply to a particular cylinder can be cut off and the resulting drop in power compared with the theoretical.

1.3.4 Thermography

Sensitive instruments are now available for remotely measuring even small temperature changes, in particular in comparison with a standard condition. At this time, thermography is used principally in quasi-static situations, such as with electrical switchboards, to detect local hot spots, and to detect faulty refractory linings in containers for hot fluids such as molten metal.

So-called 'hot box detectors' have been used to detect faulty bearings in rail vehicles, by measuring the temperature of bearings on trains passing the wayside monitoring point. These are not very efficient, as they must not be separated by more than 50 km or so, because a substantial rise in temperature of a bearing only occurs in the last stages of life, essentially when 'rolling' elements are sliding. Monitoring based on vibration and/or acoustic measurements appears to give much more advance warning of impending failure.

1.4   Types and Benefits of Vibration Analysis

1.4.1   Benefits Compared with Other Methods

Vibration analysis is by far the most prevalent method for machine condition monitoring because it has a number of advantages compared with the other methods. It reacts immediately to change and can therefore be used for permanent as well as intermittent monitoring. With oil analysis for example, several days often elapse between the collection of samples and their analysis, although some online systems do exist. Also in comparison with oil analysis, vibration analysis is more likely to point to the actual faulty component, as many bearings, for example, will contain metals with the same chemical composition, whereas only the faulty one will exhibit increased vibration.

Most importantly, many powerful signal processing techniques can be applied to vibration signals to extract even very weak fault indications from noise and other masking signals. Most of this book is concerned with these issues.

1.4.2   Permanent vs Intermittent Monitoring

Critical machines often have permanently mounted vibration transducers and are continuously monitored so that they can be shut down very rapidly in the case of sudden changes which might be a precursor to catastrophic failure. Even though automatic shutdown will almost certainly disrupt production, the consequential damage that could occur from catastrophic failure would usually result in much longer shutdowns and more costly damage to the machines themselves. Critical machines are often 'spared', so that the reserve machines can be started up immediately to continue production with a minimum of disruption. Most critical high-speed turbomachines, in for example power generation plants and petrochemical plants, have built-in proximity probes (Section 1.5.2) which continuously monitor relative shaft vibration, and the associated monitoring systems often have automatic shutdown capability. Where the machines have gears and rolling element bearings, or to detect blade faults, the permanently mounted transducers should also include accelerometers, as explained below in Section 1.5.4.

The **advantages** of permanent monitoring are:

- It reacts very quickly to sudden change and gives the best potential for protecting critical and expensive equipment.
- It is the best form of protection for sudden faults that cannot be predicted. An example is the sudden unbalance that can occur on fans handling dirty gas, where there is generally a build-up of deposits on the blades over time. This is normally uniformly distributed, but can result in sudden massive unbalance when sections of the deposits are dislodged.

The **disadvantages** of permanent monitoring are:

- The cost of having permanently mounted transducers is very high and so they can only be applied to the most critical machines in a plant.
- Where the transducers are proximity probes, they virtually have to be built into the machine at the design stage, as modification of existing machines would often be prohibitive.

- Since the reaction has to be very quick, permanent monitoring is normally based on relatively simple parameters, such as overall RMS or peak vibration level and the phase of low harmonics of shaft speed relative to a 'key phasor', a once-per-rev pulse at a known rotation angle of the shaft. In general such simple parameters do not give much advance warning of impending failure; it is likely to be hours or days, as opposed to the weeks or months lead time that can be given by the advanced diagnostic techniques detailed in later chapters of this book.

Of course, if transducers are mounted permanently, it is still possible to analyse the signals in more detail, just not continuously. This gives the advantage that intermittent monitoring can be carried out in parallel with the permanent monitoring and updated at much more frequent intervals, typically once per day instead of once per week or once per month, to give the best of both worlds. In order to take advantage of the powerful diagnostic techniques, the permanently mounted transducers would have to include accelerometers, for the reasons discussed below in Section 1.5.4.

For the vast majority of machines in a plant it is not economically justified to equip them with permanently attached transducers or permanent monitoring systems. On the other hand, since the major economic benefit from condition monitoring is the potential to predict incipient failure weeks or months in advance, so as to be able to plan maintenance to give the minimum disruption of production, acquire replacement parts, and so on, it is not so important to do the monitoring continuously. The intervals must just be sufficiently shorter than the minimum required lead times for maintenance and production planning purposes. A procedure for determining the optimum intervals is described in [13]. A very large number of machines can then be monitored intermittently with a single transducer and data logger and the data downloaded to a monitoring system capable of carrying out detailed analysis.

The **advantages** of intermittent monitoring are:

- Much lower cost of monitoring equipment.
- The potential (through detailed analysis) to get much more advance warning of impending failure and thus plan maintenance work and production to maximize availability of equipment.
- It is thus applied primarily where the cost of lost production from failure of the machine completely outweighs the cost of the machine itself.

The **disadvantages** of intermittent monitoring are:

- Sudden rapid breakdown may be missed and in fact where failure is completely unpredictable this technique should not be used. On the other hand, the reliability of detection and diagnostic techniques for predictable faults is increasing all the time and can now be said to be very good, in that considerable economic benefit is given statistically by correct application of the most up-to-date condition monitoring techniques [2–5].
- The lead time to failure may not be as long as possible if the monitoring intervals are too long for economic reasons. This is in fact an economic question, balancing the benefits of increased lead time against the extra cost of monitoring more frequently [13].

To summarize, **permanent monitoring** is used to shut machines down in response to sudden change and is thus primarily used on critical and expensive machines to avoid catastrophic failure. It is based on monitoring relatively simple parameters that react quickly to change and typically uses proximity probes and/or accelerometers. **Intermittent monitoring** is used to give long-term advance warning of developing faults and is used on much greater numbers of machines and where production loss is the prime economic factor rather than the cost of the machines themselves. It is usually based on analysis of acceleration signals from accelerometers, which can be moved from one measurement point to another.

1.5   Vibration Transducers

Transducers exist for measuring all three of the parameters in which lateral vibration can be expressed, namely displacement, velocity and acceleration. However, the only practical (condition monitoring) transducers for measuring displacement, proximity probes, measure relative displacement rather than absolute displacement, whereas the most common velocity and acceleration transducers measure absolute motion. This is illustrated in Figure 1.1, which shows a bearing pedestal equipped with one horizontal accelerometer and two proximity probes at 90° to each other. The latter, even though termed vertical and horizontal, would normally be located at ±45° to the vertical so as not to interfere with the usual bolted flange in the horizontal diametral plane of the bearing (Figure 1.2).

1.5.1   Absolute vs Relative Vibration Measurement

Proximity probes measure the relative motion between a shaft and casing or bearing housing (as illustrated in Figure 1.1). It is important to realize that this gives very different information from the absolute motion of the bearing housing, as measured by a so-called 'seismic transducer' as exemplified by an accelerometer. These two parameters are probably as different as the temperature and pressure of steam, even though sometimes related.

The relative motion, in particular for fluid film bearings, is most closely related to oil film thickness, and thus to oil film pressure distribution, as calculated using Reynolds' equation [14]. It is thus also very important in rotor dynamics calculations, as these are greatly influenced
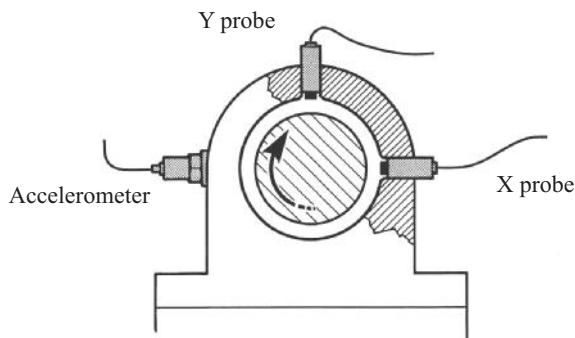

Figure 1.1   Illustration of absolute against relative vibration (Courtesy Brüel & Kjær).
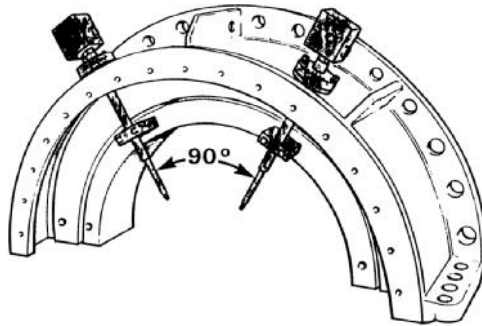
Figure 1.2 Proximity probes installed in a turbine bearing cap.

by the bearing properties. These questions are discussed in more detail in Chapter 2, which gives further references on fluid film bearings and rotor dynamics. However, a fluid film bearing is a very nonlinear spring, and therefore the amplitude of relative vibration does not give a direct measure of the forces between the shaft and its bearing. An increase in static load, for example, causes the oil film to become thinner, and the bearing stiffer, with reduced vibration amplitude, even though the higher load might be more likely to cause failure.

The absolute motion of the bearing housing, on the other hand, responds directly to the force applied by the shaft on the bearing (these being the same since the inertia of the oil film is negligible), and since the machine structure tends to have linear elastic properties, the vibration amplitude will be directly proportional to the force variation, independent of the static load.

In other words, the journal bearing stiffness and damping properties, and thus the dynamic bearing forces, are most directly related to the relative position and motion of the shaft in the bearing, but the response to these forces is most directly indicated by the absolute motion of the housing. An advantage of proximity probes is that they can measure both the absolute position of the shaft in the bearing and the vibrations around the mean position. Direct current (DC) accelerometers do exist, but are rarely used in machine monitoring, since it is still not possible to integrate the signals directly to total velocity and displacement because of the lack of constants of integration. Accelerometers are thus used to measure fluctuations in acceleration around a mean value of zero. This can be integrated to absolute velocity and displacement (fluctuations), but excluding zero frequency.

Other comparisons between the different types of transducers depend on the technical specifications for dynamic range, frequency range, and so on, so each type will be discussed in turn.

1.5.2 Proximity Probes

Proximity probes give a measure of the relative distance between the probe tip and another surface. They can be based on the capacitive or magnetic properties of the circuit including the gap to be measured, but by far the most ubiquitous proximity probes are those based on the changes in electrical inductance of a circuit brought about by changes in the gap. Such probes were pioneered by the company Bently Nevada, now owned by GE, and are very widely

used for machine monitoring [15]. Figure 1.2 shows typical proximity probes installed in the bearing cap of a turbine.

The medium in the gap must have a high dielectric value, but can be air or another gas, or for example the oil in fluid film bearings. The surface whose distance from the probe tip is being measured must be electrically conducting, so as to allow the generation of eddy currents by induction. A signal is generated by a 'proximitor' (oscillator/demodulator) at a high frequency and its amplitude is directly dependent on the size of the gap between the probe and the measurement surface. Amplitude demodulation techniques are used to retrieve the signal. A typical probe can measure reasonably linearly in the gap range from 0.25 to 2.3 mm with a maximum deviation from linearity of 0.025 mm (1.1% of full scale) with a sensitivity of 200 mV/mil (7.87 V/mm). Thus, in the sense of the ratio of maximum to minimum value, the dynamic range is less than 20 dB, but in the sense of the ratio of the maximum to minimum component in a spectrum, this would be limited by the nonlinearity to at best 40 dB.

Linearity is not the only factor limiting the dynamic range of valid measurement. By far the biggest limitation is given by runout, called 'glitch' by Bently Nevada [16]. Runout is the signal measured in the absence of actual vibration and is composed of 'mechanical runout' and 'electrical runout'. Mechanical runout is due to mechanical deviations of the shaft surface from a true circle, concentric with the rotation axis, and these include low-frequency components such as eccentricity, shaft bow and out-of-roundness, and shorter components from scratches, burrs and other local damage. Electrical runout is due to variations in the local surface electrical and magnetic properties and can be affected by residual magnetism, and even residual stresses, as well as subsurface imperfections. Much can be done to minimize runout before a shaft goes into service [16], but in general it is unlikely that the dynamic range from the highest measured component to the highest runout component would be more than 30 dB. It is possible to use 'runout subtraction' to compensate to some extent for runout, but the benefits are very limited. In principle, the runout, both mechanical and electrical, can be measured under 'slow roll' conditions (<10% of normal operating speed), when it can be assumed that the vibration is negligible, and then subtracted from measurements at higher speed. This is most valid for the first harmonic (fundamental frequency) of rotation and can often be done by the monitoring system by vector subtraction. It is unlikely to be valid above critical speed for measurements made below critical speed, at least where the runout is due to shaft bow. Another reason why the runout subtraction might not be valid is that, on large machines, thermal expansion from low to high load/speed means that the section of shaft on which the slow roll measurements are made is different from that aligned with the probes under normal operating conditions. Some machines are required to run without shutdown for one or more years, and the monitoring position on the shaft is also subject to change through wear of thrust bearings. Proximity probes are in fact used in axial position monitoring of rotors.

Interestingly, the dominant standard for shaft vibration monitoring, the American Petroleum Institute's API 670 [15], states that no correction is to be made for runout in indicated vibration levels. It also states that the total runout should not exceed 25% of the maximum allowed peak-to-peak vibration amplitude. This corresponds to just −12 dB and is the best indicator of the valid dynamic range of a proximity probe measurement. Even where runout subtraction can be carried out successfully, it is unlikely the improvement in dynamic range would be more than 10 dB, say from 30 to 40 dB, and that primarily at low harmonics. The higher the harmonic, the shorter the wavelength, and thus the greater the likelihood that measured runout would be affected by small axial displacement due to thermal expansion or wear.

The valid frequency range of proximity probes is typically 10 kHz, but this is misleading, as the actual limit is likely to be given by a certain number of harmonics of the shaft speed, because of the dynamic range limitation. As explained in Chapter 4, Section 4.2.1, mechanical vibrations tend to have roughly uniform spectra in terms of velocity and thus reduce in terms of displacement as $1/f$, where f is the frequency. It is unlikely that more than 10 or so harmonics would be within the valid dynamic range as restricted by runout. This severely restricts the diagnostic capabilities of proximity probes, in particular for long-term advance warning of incipient failure, and is the main reason why the major part of this book is devoted to analysis of accelerometer signals, which have much larger dynamic and frequency ranges, as explained in Section 1.5.4.

A typical example of the restricted frequency range of proximity probe measurements is given in Figure 1.3, which compares spectra of signals from a proximity probe and an accelerometer on the same machine at the same time. The signals were recorded by the author from the monitoring system of a centrifugal compressor in a Canadian chemical plant.

The spectrum of the proximity probe signal (Figure 1.3(a)) is completely dominated by harmonics of the shaft speed (133 Hz). However, only the first two or three are presumably valid, as the higher harmonics are quite uniform. In the spectrum of the accelerometer signal (Figure 1.3(b)), which has been integrated to (absolute) displacement for easier comparison with the (relative) displacement of the 'prox. probe' signal, the first two or three harmonics protrude above the noise. However, at higher frequencies there are four harmonics of the vane pass frequency (11 vanes) visible, which could be useful for diagnostic purposes (see
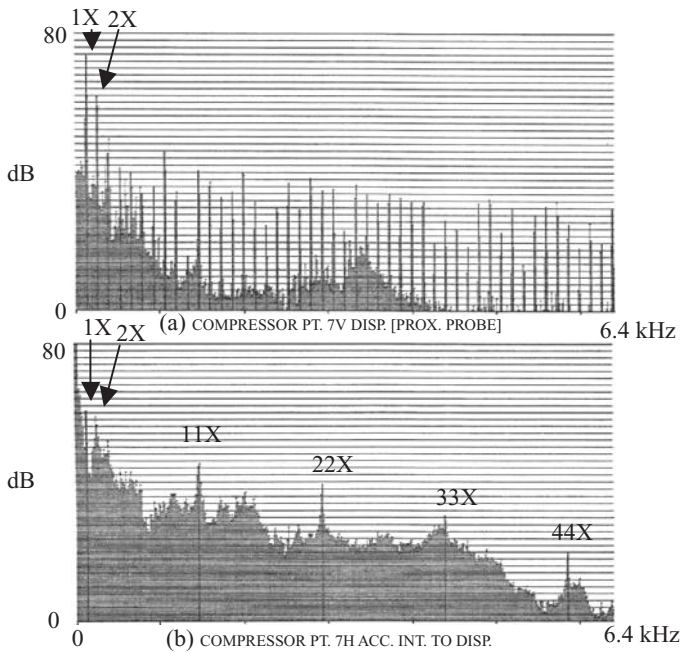

Figure 1.3 Comparison of spectra measured on a centrifugal compressor: (a) proximity probe; (b) accelerometer signal (integrated to displacement).
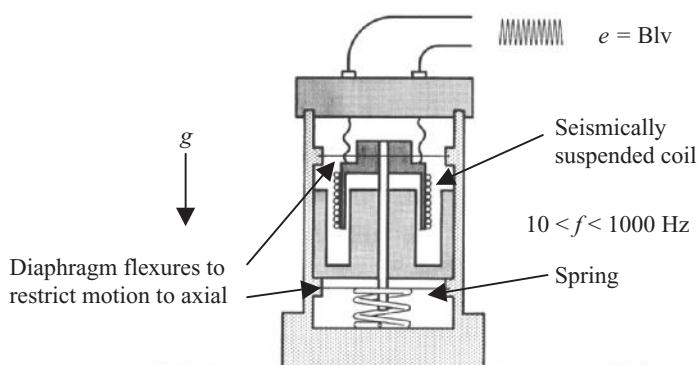
Figure 1.4 Schematic diagram of one realization of a velocity pickup (Courtesy Brüel & Kjær).

Section 2.2.4 of Chapter 2). There is nothing remarkable about the same harmonics in the 'prox. probe' spectrum. Note that even though the accelerometer signal contains much more noise from gas flow, the latter can be removed by synchronous averaging (see Section 3.6.2 of Chapter 3), exposing the harmonics of shaft speed. On the other hand, this cannot be used to remove the runout effects from 'prox. probe' signals, since they are perfectly periodic with the rotation speed of the shaft.

1.5.3 Velocity Transducers

Transducers do exist which give a signal proportional to absolute velocity. They are effectively a loudspeaker coil in reverse and typically have a seismically suspended coil in the magnetic field of a permanent magnet attached to the housing of the transducer (as in Figure 1.4) or the inverse, where the coil is rigidly attached to the housing and the magnet seismically suspended. A body is said to be seismically suspended when it is attached to another by a spring such that when the second body is vibrated, the first will move with it at low frequencies, but when the excitation frequency exceeds the natural frequency of the suspended mass on its spring, it will remain fixed in space, and the second body will move around it. When the housing of the transducer (or pickup) is attached to a vibrating object, the relative motion between it and the seismically mounted component (for frequencies above the suspension resonance) is equal to the absolute motion of the object in space. To avoid problems with excessive response to excitation in the vicinity of the resonance frequency, the damping of the suspension is usually quite high, typically of the order of 70% of critical damping, and this also means that the amplitude response of the transducer is reasonably uniform almost down to the resonance frequency.

Figure 1.5 (from [17]) shows the frequency response of a generalized vibration transducer of the type described, for different values of damping, against frequency ratio with respect to the natural frequency of the suspension. It is seen that for critical damping ratio $\zeta = 0.7$, the frequency range for amplitude ratio close to one (i.e. output equals input) is as wide as possible. On the other hand, for this value of damping, the phase deviation from $180°$ (the ideal asymptotic value) extends to quite a high frequency. This means that the amplitude spectrum of signals captured with such a transducer will be quite accurate, but waveforms will not
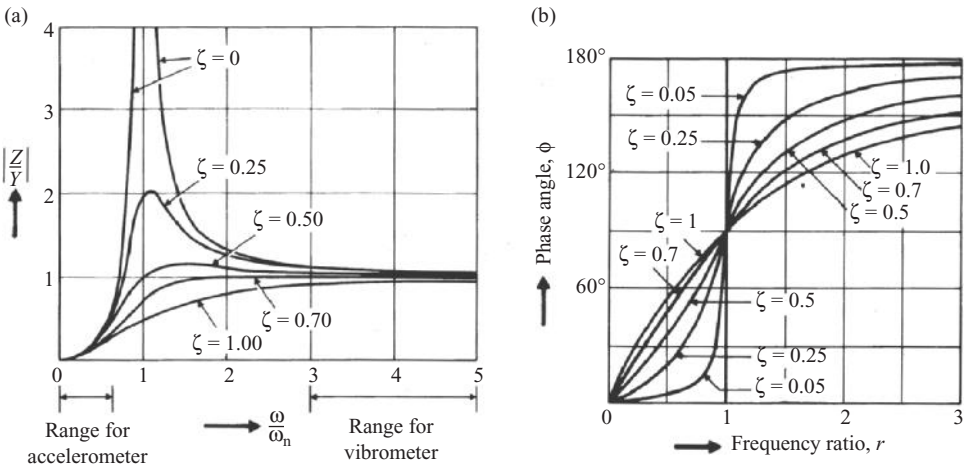
Figure 1.5   Frequency response of a seismically suspended vibrometer [17]: (a) amplitude characteristic; (b) phase characteristic. ζ = critical damping ratio (Reproduced by permission of Pearson Education South Asia Pte Ltd).

necessarily follow the original. As will be seen from the Fourier analysis theory of Chapter 3, to reproduce repetitive impulses, for example, all harmonics must be in phase at the time of occurrence, and this would not be the case if they were measured with such a transducer if the low harmonics were in the range of phase distortion.

In the case of a velocity pickup, the relative motion of the magnet in the coil gives a voltage signal proportional to velocity (and thus the absolute velocity of the housing). The dynamic range (ratio of largest to smallest measurable signal) of such a transducer is about 60 dB. The lower frequency limit is typically set (by adjustment of the suspension resonance frequency) to 10 Hz, while the highest measurable frequency is limited by the resonances of internal components to about 1–2 kHz. Much of the data for the VDI 2056 and ISO 2372 standards (Section 4.2.1) was gained with velocity transducers of this kind and that is the main reason why the frequency limits in those and later standards are 10 Hz to 1 kHz.

Relative to accelerometers, velocity pickups are much heavier and bulkier. It will be shown in the next section that an accelerometer plus integrator is a much better velocity transducer.

1.5.4   Accelerometers

Accelerometers are transducers which produce a signal proportional to acceleration. By far the most common types for use in machine condition monitoring are piezoelectric accelerometers, which make use of the piezoelectric properties of certain crystals and ceramics. Such piezoelectric elements generate an electric charge proportional to strain. In a typical design as shown in Figure 1.6(a), a so-called 'compression' type, the piezoelectric elements are sandwiched between a mass and the base, the whole assembly being clamped in compression via a spring. This arrangement can also be considered as one representation of the general vibration transducer whose frequency characteristics are shown in Figure 1.5, except that it is designed to operate below the natural frequency of the suspended element (in the 'Range for
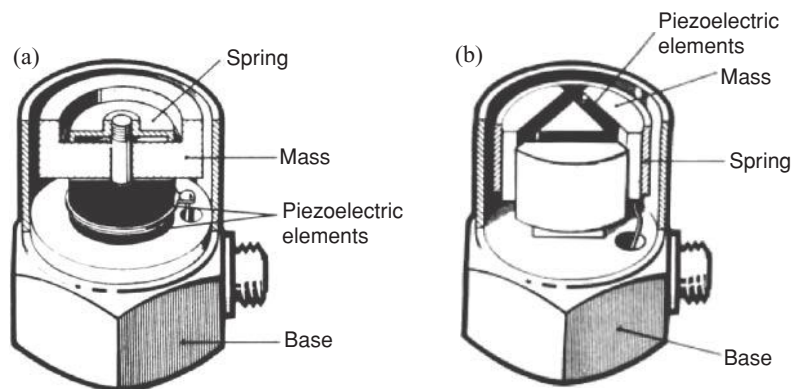
Figure 1.6 Typical accelerometer designs: (a) compression type; (b) shear type (Courtesy Brüel & Kjær).

accelerometer' in Figure 1.5). When the base of the accelerometer is connected to a vibrating object, the mass is forced to follow the motion of the base by the piezoelectric elements, which act as a very stiff spring. The varying inertial force of the mass causes the piezoelectric elements to deform slightly, giving a strain proportional to the variation in acceleration. They then produce an electric charge proportional to this acceleration, and so their sensitivity is quoted in picocoulombs per metre per second squared, $pC/(m\,s^2)$. As discussed below, this must be converted to a voltage by a charge amplifier. The clamping spring, while very stiff, is much less stiff than the piezoelectric elements, so its force remains effectively constant, but it is required to maintain a positive compression force on the assembly.

Figure 1.6(b) shows an alternative design where the piezoelectric elements deform in shear (they must be polarized so as to produce a charge proportional to shear rather than compressive strain). This particular design is the patented 'delta shear'$^{\circledR}$, design, by Brüel & Kjær, where the centre post to which the assembly is clamped has an equilateral triangular or delta (Δ) cross-section, meaning that the mechanical properties of the assembly are isotropic, with no preferential direction. The spring in this case is a cylindrical clamping spring, once again to maintain positive compressive forces between the masses, the elements and the centre post. Other shear designs exist, where the piezoelectric elements are clamped in one direction against a rectangular centre post, but this has the disadvantage of different transverse resonance frequencies in different directions. Another isotropic design uses cylindrical elements and masses, but these must then be cemented together, giving lower structural integrity and temperature limitations.

The electric circuit including the piezoelectric elements has very high impedance and is subject to a number of problems, such as pickup of signals from electromagnetic radiation. The latter is minimized by using coaxial cables with an outer braided wire shield. The type of cable connector shown in Figure 1.6, a so-called 'microdot' connector, gives the best results for laboratory measurements, but the associated standard microdot cables are not very practical for regular measurements in the field. A more robust double-shielded microdot cable solves some of these problems, but other types of cables with more robust TNC connectors or equivalent may be found preferable for regular monitoring, even if the repeatability and frequency range are degraded slightly.

If the transducers are connected directly to a voltmeter, the voltage corresponding to the generated charge is directly affected by the impedance of the circuit, and in particular the capacitance of the accelerometer cable, which varies with its length. For this reason it is generally necessary to use accelerometers together with charge amplifiers, which convert a given charge at the high-impedance input side to a proportional voltage on the low-impedance output side. A typical design converts 1 pC at the input to 1 mV at the output. Cables can be very long on the output side without effect on sensitivity and with negligible noise pickup. Another problem with the high-impedance circuit is the sensitivity to 'triboelectric noise', or generation of a static electric charge by rubbing between the inner conductor and its sheath. Triboelectric noise is often minimized by using a low-friction PTFE sheath and graphite lubricant between the inner conductor and the sheath.

The problems with the high-impedance circuit can be reduced to a minimum by building the charge amplifier into the transducer. Miniaturization of electronic circuits has now made this possible, and it does solve many of the practical problems associated with special cables, electrical interference, and so on. It does have one disadvantage, and that is that it is more difficult to detect overload of the input circuit. Separate charge amplifiers often have an overload indicator and the possibility to change the gain either before or after filtration, integration, and so on. It should always be kept in mind that the input circuit has to cope with the full signal generated by the transducer, even that part outside the final frequency range selected by high- and low-pass filters. Piezoelectric accelerometers have low internal damping, and it is quite common for the resonant gain in Figure 1.5(a) to be 30 dB above the linear value in the operating range. The resonance frequency is typically 30 kHz, but in some machines, such as gas turbines, there can be significant excitation at that frequency. This might cause overload of the input amplifier, even if the signal is low-pass filtered at 10–20 kHz (the maximum valid frequency of measurement) in the amplifier.

The resonant frequency for transverse motion of the accelerometer is often lower than that in the main measurement direction, in particular for compression-type accelerometers, and even though the transverse sensitivity is only a small percentage, the signals can become distorted if the transverse resonance is excited. The best way to solve this problem (and the excitation of the main axial resonance), where there can be strong excitation at such high frequencies, is to mount the accelerometer on a 'mechanical filter'. This contains an elastomeric layer having a spring constant such that in combination with the mass of the transducer the mounted resonance is, say, one-third that of the transducer itself, at the same time providing good damping to reduce the resonance peak to between +3 and +5 dB. As opposed to scientific and laboratory measurements, for condition monitoring purposes it is most important that the frequency response of the transducer system is repeatable, rather than strictly linear, as the measurement in any case represents an external measurement of internal events, and the response of the transducer can be considered part of the response of the machine itself at that measurement point. Thus, it is quite common to use accelerometer signals up to 50–65% of the transducer resonance, where the deviation from linearity might be as high as 5 dB, even though the recommended range for linear measurements is typically one-third the resonance frequency. In the same way, the resonance frequency of the mechanical filter can be within the measurement range, as long as it is repeatable. In this connection it should be kept in mind that the stiffness properties of the elastomer will vary with temperature, so care should be taken if the temperature of the mounting point is subject to variation.

1.5.4.1 Frequency and Dynamic Ranges

One of the main advantages of accelerometers is the extremely wide range of both amplitude and frequency that they provide. The typical dynamic range of an accelerometer is 160 dB ($10^8 : 1$), although in conjunction with an amplifier this might be reduced to 120 dB ($10^6 : 1$) for a particular gain setting on the amplifier. As mentioned in the preceding section, a typical upper frequency limit for condition monitoring purposes is 10–20 kHz, while the lowest valid frequency is below 1 Hz.

It should be noted that such a low minimum frequency is an advantage of the shear design, since a major limiting factor at low frequency is given by the fact that the sensitivity of piezoelectric materials varies to some extent with temperature. Because of the thermal inertia of the transducer, its rate of temperature variation is limited, and there is no problem above a certain frequency. As will be seen in Figure 1.6, in the compression design the piezoelectric elements are in direct contact with the base and can respond more rapidly to temperature change, whereas in the shear design, the elements are more isolated from the base. Another reason for generation of noise at low frequencies is 'base strain', where the larger bending deflections corresponding to a given acceleration at low frequency can distort the piezoelectric elements, but much less in the shear design, once again because the triangular centre post is more isolated from the base. Thus, the lower limiting frequency of a compression-type accelerometer might be as high as 5–10 Hz.

Figure 1.7 compares typical dynamic and frequency ranges for the three main types of transducers. It is immediately evident that the accelerometer has much wider ranges than the other two. The dynamic range is shown in terms of the measured parameter, that is acceleration for an accelerometer, velocity for a velocity pickup and relative displacement for a proximity probe. Superimposed on the original diagram are two possible ranges for an accelerometer and integrator which produces a signal proportional to velocity. The diagram assumes that the accelerometer has a dynamic range of 120 dB for one gain setting, but this can be moved by a further 30–40 dB by gain adjustment. Even though the integrator represents a low-pass filter, with slope −20 dB per frequency decade, the dynamic range of the combination is still greater than 60 dB over a frequency range of three decades. It is illustrated how this three-decade range can be simply switched between (for example) 10 Hz to 10 kHz and 1 Hz to 1 kHz, even with the same accelerometer and (external) amplifier. This is obviously much more flexible than the fixed approximate two-decade range of the velocity pickup. The technical specifications of the combined accelerometer/integrator are also better; for example, the phase distortion can be made negligible within a factor of three above the integration cutoff frequency.

Using an electronic integrator, such as built into some external charge amplifiers, was very valuable when the typical dynamic range of recorders and analysis systems was 50–80 dB, since this meant that the limited range could accommodate the velocity signal, which inherently occupied the minimum dynamic range (as explained above). With modern data acquisition systems, the dynamic range of amplifiers and analogue-to-digital (AD) converters is more typically 120 dB, and so signals recorded as acceleration can be integrated numerically to velocity if desired. This has the advantage that it can be done with non-causal integrators with zero phase distortion within the measurement range.

Velocity signals can be further integrated to displacement, either electronically or numerically, whereas the inverse operation of differentiating displacement signals to velocity, and velocity signals to acceleration, tends to introduce problems by amplification of high-frequency
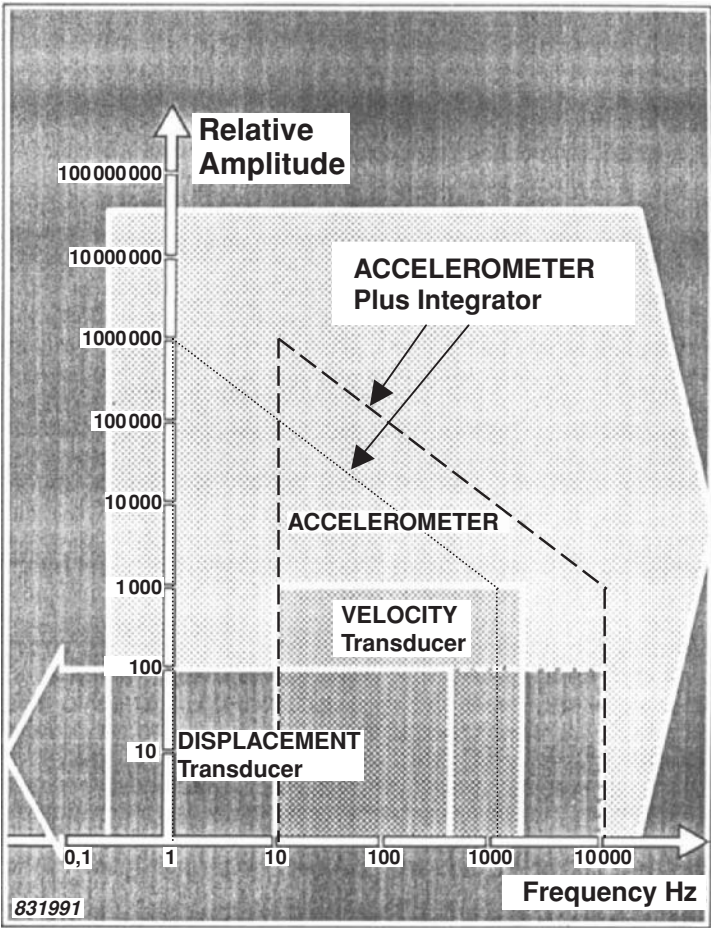
Figure 1.7 Typical frequency and dynamic ranges for the three main transducer types with superimposed ranges for an accelerometer and integrator (Courtesy Brüel & Kjær).

noise. This can be done successfully numerically, as long as the frequency components above the highest valid frequency (not dominated by noise) are first removed by low-pass filtering.

1.5.5   Dual Vibration Probes

Shaft vibration is normally measured by proximity probes, but this gives the motion relative to the housing. To obtain the absolute motion of the shaft, it is necessary to add this relative motion to the absolute motion of the housing, and so-called 'dual probes' are designed to do this. They contain both a proximity probe and a seismic probe to measure the absolute motion of the housing. The seismic probe can be either a velocity transducer (with signal integrated to absolute displacement) or an accelerometer (with signal double integrated to absolute displacement). The overall frequency and dynamic ranges of the combination would

normally be limited by the proximity probe, so it could be said that the accelerometer gives no particular advantage over the velocity probe, but on the other hand the accelerometer signal could be separately analysed in its own right, in which case it could give some advantage.

The ratio of relative to absolute vibration varies widely from machine to machine, and so where it is important to know how the shafts of adjacent machines are vibrating (because they have to be connected by a coupling, for example) then this has to be on the basis of overall absolute motion, as produced by a dual probe.

1.5.6 Laser Vibrometers

In recent years there has been a rapid development of vibration transducers based on the laser Doppler principle. In this technique, a coherent laser beam is reflected from a vibrating surface and is frequency shifted according to the absolute velocity of the surface (in the direction of the beam) by the Doppler effect. The frequency shift is measured by an interferometer and converted to velocity. Note that because the frequency shift occurs at the reflection, the result is virtually independent of the motion of the transmitter/receiver; in other words, it measures absolute rather than relative motion.

Laser vibrometers have the big advantage that they do not load the measurement object, and the measurement point can be changed easily and rapidly by deflecting the light beam. This is useful for making repeatable measurements over a grid in the minimum time possible. For this reason, they are now used extensively for modal analysis measurements and perhaps to a lesser extent for operational deflection shape (ODS) measurements. The latter can be very useful for diagnostic purposes, even though not discussed explicitly in this book, but because a scanning laser vibrometer system is so expensive (up to hundreds of thousands of dollars) it would only have a very limited application in machine monitoring. Even without the scanning system, the vibrometers are quite bulky and difficult to move around, so they could not at present be used for intermittent monitoring. It is possible that in the future they will be miniaturized to such an extent that they could be used for portable field measurements. The author has heard a presentation where the presenter mused that in the future vibrometers could be built into a hard hat, and the operator would just have to look in the direction of a machine, utter the ID of the machine into a microphone, and the laser and imaging system would locate the machine and take measurements at a prescribed number of monitoring points on it. Currently, however, they are not really a viable option for regular condition monitoring, even though they are used for example in production quality control measurements [18].

1.6 Torsional Vibration Transducers

Some failures in machines occur because of excessive torsional vibration. When the machine has only one compound shaft, for example a motor-driven pump or a turbine-driven centrifugal compressor, there is very little coupling between torsional vibrations and lateral vibrations as measured by either accelerometers or proximity probes, and so it is desirable to measure the torsional vibrations directly. When there is a gearbox in the train, there is some coupling, because input and output torques, and torque fluctuations, are different on either side of the gearbox, and the differences have to be supported by the housing and foundation, giving rise to lateral vibrations. However, it can still be an advantage to measure the torsional vibrations directly, in part to separate them from purely lateral vibrations from other sources.

Even when not representing potential failure in torsion, torsional vibrations sometimes carry significant diagnostic information as to machine condition, such as with reciprocating machines for example, where variations in torsional vibration indicate non-uniform torque inputs from different cylinders. This is explained in more detail in Sections 2.3.2, 4.3.3 and 5.6.2.2. Yet another example where it can be advantageous to measure torsional vibrations is in connection with gears, where the dynamic transmission error is effectively the difference in (scaled) torsional vibration on the input and output sides. The reason for the scaling is that the error is actually a linear displacement along the line of action of the meshing of the two gears and thus represents a different rotational angle on each if the gear ratio is not 1 : 1. The use of transmission error as a diagnostic tool is explained in Section 5.4.2.

Thus, the various means of measuring torsional vibrations are now discussed.

1.6.1 Shaft encoders

Shaft encoders are not a torsional vibration transducer as such, but information about torsional vibration (i.e. angular velocity variations) can be obtained by analysing shaft encoder signals. Shaft encoders give out a series of pulses at equal angular intervals, with typically 1024 per revolution. They are sometimes attached to the free end of a shaft (with the housing attached to the housing of the machine), but 'through-shaft' encoders also exist, which can be placed elsewhere on the shaft, possibly even between bearings. Mounting on the free end of a shaft would often be via a flexible coupling, which restricts the range of frequencies that can be transmitted to the encoder, but which means that low harmonics of shaft speed are faithfully reproduced. When flexible couplings are not used, even slight misalignment can give some distortion of low-harmonic components, but this may not be a problem if information is primarily desired about higher harmonics (e.g. toothmesh harmonics of gears).

There are two methods that can be used to extract torsional vibration information from shaft encoder signals. The first is to use phase and/or frequency demodulation of the encoder pulse frequency, as described in Sections 3.3.2, 4.3.3, 5.4.2 and 5.6.2.2. Phase demodulation obtains the torsional vibration information in terms of angular displacement, while its time derivative, frequency demodulation, expresses it in terms of angular velocity. It is possible to take a further derivative to express it in terms of angular acceleration, but there is no equivalent modulation term. The second method is to use a very high-frequency clock (typically 80 MHz) to measure the time intervals between pulses from the encoder. The reciprocal of this can be scaled in terms of average angular velocity in the interval. One advantage is that a fixed number of samples is obtained per revolution, so there is no need to perform order tracking (see Section 3.6.1) to compensate for slow speed changes. Both methods are discussed and compared in [19].

1.6.2 Torsional Laser Vibrometers

Torsional laser vibrometers also exist and have two laser beams directed at the surface of a rotating shaft [20]. The reflected signals are processed in such a way that everything is cancelled except the torsional motion expressed as angular velocity. It can be shown that this applies for arbitrary shape of the cross-section of the shaft.

As for laser vibrometers which measure lateral motion, the major advantage is that the measurement is non-contact (though a section of the shaft must be exposed to view), but
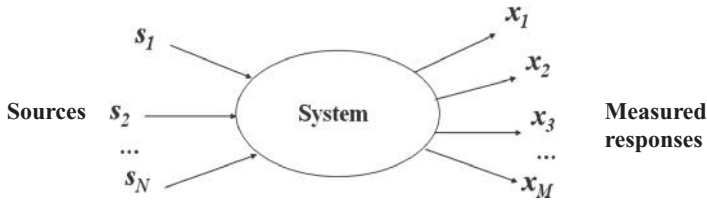
Figure 1.8 Illustration of a number of measured responses of a system due to a number of excitation sources.

because of the dual beam they are even more expensive than single beam vibrometers. An example of the use of a torsional laser vibrometer to measure the speed fluctuations of the crankshaft of a large diesel engine is given in Section 5.6.2.2, but it is pointed out that equivalent results were obtained using a much cheaper shaft encoder. There was a slight difference in that the laser vibrometer measured the overall angular motion of the crankshaft, including rocking of the engine block, whereas the encoder measured the torsional fluctuations relative to the block, which in that case were more relevant, but the differences were small.

1.7   Condition Monitoring – the Basic Problem

It is worth discussing the way in which condition information can be extracted from vibration signals. Measured vibration signals are always a combination of source effects and transmission path effects. In general, as illustrated in Figure 1.8, a measurement at one point will be a sum of responses from a number of sources. Such a system is known as a multiple input, multiple output (MIMO) system.

   The contribution to the response at one measurement point from one source, in the time domain, is a convolution of the force signal with the impulse response function (IRF) of the transmission path from the source to the measurement point (see Section 3.2.6). In the frequency domain (and Laplace domain) this simplifies to a product of their respective spectra, the spectrum of the IRF being equal to the corresponding frequency response function (FRF). This can be represented symbolically as

$$x_i = \sum_j s_j * h_{ij} \tag{1.1}$$

$$X_i = \sum_j S_j H_{ij} \tag{1.2}$$

where the upper case letters in Equation (1.2) represent the Fourier transforms of the lower case symbols, and the asterisk represents the convolution operation (Section 3.2.6.2).

   For an individual source (forcing function) in Equation (1.2) the product is even further simplified to a sum by taking the logarithm. Since the FRF H_{ij} is complex, its logarithm has log amplitude ratio (log gain) as the real part, and phase as the imaginary part, meaning that the log amplitude of the response is the sum of the log amplitudes of the source and FRF, and the phase is the sum of the phases. This is illustrated in Figure 1.9 for the log amplitudes
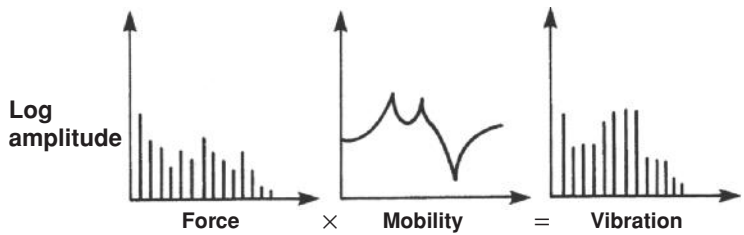
Figure 1.9 Combination of forcing function and transfer path to give response vibration for one source (Courtesy Brüel & Kjær).

only. The 'mobility' is the FRF corresponding to force input and vibration velocity output. Note that the indicated multiplicative relationship is actually depicted as additive on log amplitude scales.

In Figure 1.9 the forcing function is depicted as consisting of discrete frequency components, which is typical for many machines running at constant speed. It illustrates that resonance frequencies do not appear in response spectra in such cases unless directly excited by a forcing frequency. For broadband noise excitation the response spectra will have peaks at resonance frequencies, but not discrete frequency peaks. These are usually recognizably different from discrete frequency components because of the broadness of the base, at least on log amplitude scales, and using a window function such as Hanning (Section 3.2.8.2). Figure 1.10 shows a numerically generated example combining three discrete frequency components at
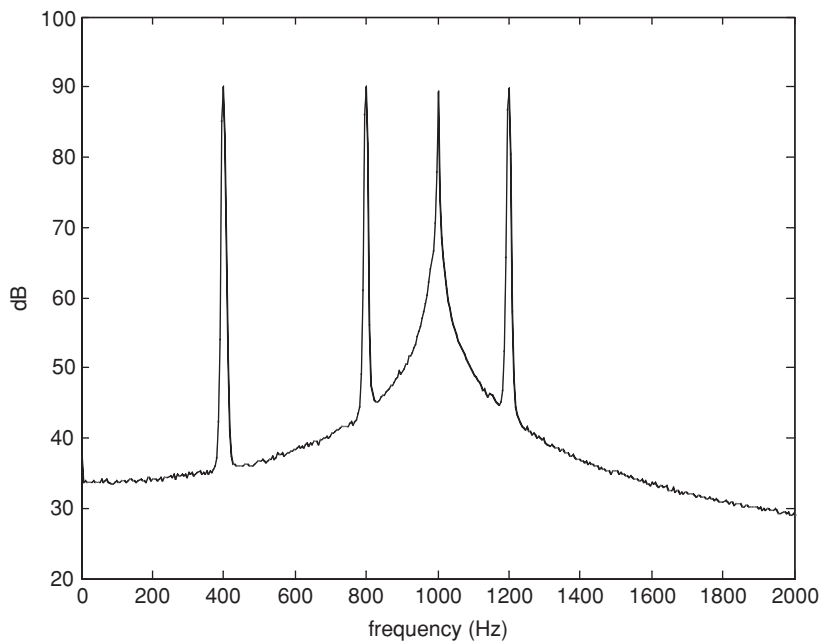

Figure 1.10 Comparison of the spectra of discrete frequency components at 400, 800 and 1200 Hz with that of a narrow band resonance at 1000 Hz.

[400, 800, 1200] Hz with a narrowband resonance at 1000 Hz excited by white noise (averaged over 1000 spectra) and using Hanning weighting.

Even in the MIMO case, where a general response spectrum is no longer the product of a single source spectrum and single FRF but a sum of these, the appearance of the spectrum is still very similar, in particular on log amplitude scales. This is partly because the same discrete frequency components tend to appear at all measurement points with different strengths, and resonance frequencies are global properties of a structure, so tend to appear in all FRFs, once again with different strengths, so that one or two paths will tend to dominate for a particular measurement point and for a particular resonance peak.

The basic problem in diagnosing the reason for changes in response vibrations is to decide whether the change has occurred at the source(s) or in the structural transmission path.

Quite often, a change in condition results from a change at a source, such as an increase in unbalance force, or a change in the force between meshing gears. On the other hand, other types of faults may primarily result in changes in the structural response, such as a developing crack in a machine casing. Sometimes the two effects couple with each other, with the change in structural response giving a change in the forcing function. In one such case, a developing tooth root crack obviously affects the local structural properties, in particular the stiffness of that particular tooth (as discussed in Section 5.4.4), but in terms of responses at the bearings, over much of the frequency range this can primarily be interpreted as a change in the forcing function at the toothmesh. In the case of a crack in a shaft, as discussed in Section 2.2.1.2, if the crack is 'breathing', that is opening and closing every revolution of the shaft, the character of the forcing function changes, giving rise to responses at the odd harmonics of the shaft speed, in contrast to the even harmonics primarily generated when the crack is permanently open.

Apart from a number of coupled cases as just mentioned, in a broad generalization it can be said that for machines running at constant speed, sinusoidal components in response signals result from sinusoidal forcing functions at the same fundamental frequency, although because of structural nonlinearities, the responses will usually be distorted from sinusoidal and thus contain some level of harmonics of the fundamental frequency, even if the forcing function is relatively pure, such as a simple unbalance. This is the reason why frequency analysis is so powerful as a diagnostic tool, since families of harmonics with a given frequency spacing almost certainly result from a forcing function at that frequency. As discussed in Section 5.1, the presence of harmonics (perhaps only visible on a logarithmic frequency scale) allows a much more accurate measurement of the fundamental frequency, by fitting a finely tunable 'harmonic cursor' to the family. In a similar way, some effects modulate other frequencies at a lower rate and give rise to families of sidebands around the harmonics of the 'carrier' frequency, and a sideband cursor can find these modulating frequencies very accurately. An example is given by vibration signals from meshing gearteeth, where the harmonics of the toothmesh frequency (gear rotational speed times the number of teeth) are often modulated by the shaft speeds of the two gears in mesh (Section 2.2.2.2). Even where the carrier frequency is a random signal, but modulated by discrete frequency components, the latter can still be detected because of the so-called 'cyclostationary' properties of such a signal. A number of examples relating to bearing and engine diagnostics are given throughout the book.

The problem of deciding whether a change in a response signal is due to a change at the source or in the transmission path is one example of the more general problem of 'blind source

separation' (BSS), and the related topic 'blind system identification'. These are both areas of current research, which undoubtedly will have considerable impact on machine diagnostics in the future, but it is a little early to include this topic in a book such as this. However, the interested reader may like to view the special issue on mechanical applications of BSS in [21].

A mechanical application of blind system identification is the topic of 'operational modal analysis', where inherent dynamic properties of structures are deduced from response measurements only. This is now a regular topic at conferences on more general modal analysis.

References

1. Rao, B.K.N. (2009) Advances in diagnostic and prognostic strategies and technologies for failure-free maintenance of industrial assets. Comadem 2009, San Sebastian, Spain, 9–11 June.
2. Al-Najjar, B. and Alsyouf, I. (2004) Enhancing a company's profitability and competitiveness using integrated vibration-based maintenance: a case study. *Journal of European Operation Research*, **157**, 643–657.
3. Al-Najjar, B. (2009) A maintenance model for identification, quantification and elimination of losses in companies profitability: an application example. Comadem 2009, San Sebastian, Spain, 9–11 June.
4. Vlok, P.J., Coetzee, J.L., Banjevic, D. *et al.* (2002) Optimal component replacement decisions using vibration monitoring and the PHM. *Journal of the OR Society*, **53**, 193–202.
5. Sundin, P.O., Montgomery, N. and Jardine, A.K.S. (2007) Pulp mill on-site implementation of CBM decision support software. Proceedings of International Conference of Maintenance Societies, Melbourne, Australia.
6. Neale, M.J. and Woodley, B.J. (1978) A guide to the condition monitoring of machinery. *Report TRD 223* for the British Department of Industry.
7. IEEE draft standard P1438/D1.5 (1999) *Guide for Applications of Plant Condition Monitoring for Hydroelectric Facilities*. Section 4.3 'Potential Benefits'.
8. McMillan, D. and Ault, G.W. (2008) Condition monitoring benefit for onshore wind turbines: sensitivity to operational parameters. *IET Renewable Power Generation*, **2** (1), 60–72.
9. Braun, S. (ed.) (1986) *Mechanical Signature Analysis*, Academic Press, London.
10. Armor, A.F. (1983) On-line diagnostics for fossil power plants: the promise and the reality. Proceedings of EPRI 1982 Conference and Workshop, Hartford, CT, USA, 25–27 August, EPRI CS-2920, pp. 1-5–1-26.
11. Raad, A., Zhang, F. and Randall, B. (2003) On the comparison of the use of AE and vibration analysis for early gear fault detection. The Eighth Western Pacific Acoustics Conference, Melbourne, Australia.
12. Tana, C.K., Irving, P. and Mba, D. (2007) A comparative experimental study on the diagnostic and prognostic capabilities of acoustics emission, vibration and spectrometric oil analysis for spur gears. *Mechanical Systems and Signal Processing*, **21** (1), 208–233.
13. Sherwin, D.J. and Al-Najjar, B. (1999) Practical models for condition monitoring inspection intervals. *Journal of Quality in Maintenance Engineering*, **5** (3), 203–220.
14. Childs, D. (1993) *Turbomachinery Rotordynamics*, John Wiley & Son, Inc., New York.
15. American Petroleum Institute (2000) *API Standard 670, Machinery Protection Systems*, 4th edn, API Publishing Services, Washington, DC.
16. Bently Nevada Application Note 'GLITCH' Definition, Sources, and Methods of Correcting. Available via http://www.gepower.com/prod_serv/products/oc/en/bently_nevada/application_notes.htm (accessed 7 September 2010).
17. Rao, S.S. (2005) *Mechanical Vibrations*, Prentice Hall, Englewood Cliffs, NJ.
18. Vass, J., Šmíd, R., Randall, R.B. *et al.* (2008) Avoidance of speckle noise in laser vibrometry by the use of kurtosis ratio: application to mechanical fault diagnostics. *Mechanical Systems and Signal Processing*, **22** (3), 647–671.
19. Sweeney, P.J. and Randall, R.B. (1996) Gear transmission error measurement using phase demodulation. *Proceedings of the IMechE, Part C: Journal of Mechanical Engineering Science*, **210** (C3), 201–213.
20. Polytec, Basics of Rotational Vibration Sensing. Available via http://www.polytec.com/vib-university (accessed 8 October 2010).
21. Antoni, J. and Braun, S. (eds) (2005) Blind source separation. *Mechanical Systems and Signal Processing*, **19** (6) (Special issue).