Robert Bond Randall

Vibration-based Condition Monitoring

Industrial, Automotive and Aerospace Applications

Second Edition

WILEY

VIBRATION-BASED CONDITION MONITORING

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INDUSTRIAL, AUTOMOTIVE AND AEROSPACE APPLICATIONS

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Dedicated to the memory of Professor Simon Braun, 1933–2020.

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Foreword

About 10 years ago, when the first edition of Prof Randall's *Vibration-based Condition Monitoring* was published, I enthusiastically welcomed it, because a book was finally published that presented a systematic approach to a subject, 'Condition-Based Monitoring', which had grown in complexity over the years, with contributions from many important scholars, but in a non-systematic way.

There were already books that dealt with partial aspects (e.g. *Vibration and Acoustic Measurement Handbook* (1972) by Michael P. Blake and William S. Mitchell, *A Practical Vibration Primer* (1979) by Charles Jackson and *Fundamentals of Noise and Vibration Analysis for Engineers* (1989) by Michael P. Norton), or they dealt with specific fields (including, among the many, Maurice L. Adams Jr.'s *Rotating Machinery Vibration from Analysis to Troubleshooting* in 2001). There was already a reference journal (i.e. *Mechanical Systems and Signal Processing*, founded in 1987 with foresight by the late and recently deceased Simon Braun), but it lacked an organic work, a book that presented a systematic approach and introduced both methods and techniques.

Certainly, in twenty-first century society, one could reflect and debate for a long time whether ***a book*** still represents 'the instrument' for the transmission of knowledge. Today we have different ***media***, but I remain personally convinced that, in the scientific field, ***the book*** is still a fundamental tool; what has certainly changed is the way it is used: probably many readers of ***this book*** are not doing it, right now, on paper media, but on a digital medium.

The scientific community and engineers were lucky because this book was written by Prof Randall. I do not think there is any need to present him, because he has a long career of research, of development of signal processing techniques, of case study analysis and of teaching. I was lucky enough to meet him in person 20 years ago, and my esteem for him has always grown, as a monotonic function, not only for its scientific aspects, but also human.

Now, this second edition fills in some inevitable gaps (when writing for the first time a wide-ranging work like this, it is impossible to delve into all the topics or not neglect some, which appeared secondary at that time), but above all introduces and deepens new methods and techniques that have been fully developed in the last ten years, such as tacho-less techniques.

Why is Condition Monitoring so important in engineering and, more generally, in today's world? Prof Randall explains very well the reasons in the introduction of this book: it is a fundamental component for some of the so-called pillars of the technology paradigm of Industry 4.0, at least for IoT ('Internet of Things'), but also for Big Data analytics. Condition Monitoring allows the full implementation of Condition-Based Maintenance (CBM), with remarkable economic advantages, from a single machine to entire plants and industrial facilities, from manufacturing, to services and utilities. Finally, Condition Monitoring is the basis of a predictive – i.e. prognostic – approach to determining the residual useful life (RUL) of a component or a system.

It is certainly ambitious to define what the purpose of science is, and illustrious minds have applied themselves to this: from Greek philosophers to Galileo, from Descartes to Gödel, from Cantor to Popper. If we limit ourselves to the narrow sphere of Engineering and to its purpose, it is not possible to fail to recognise that it must explain *'how'* one does something and *'why'* it is done. In this case, in his book, Prof Randall explains very clearly the *'how'*, that is, the most well-established methods and techniques for condition monitoring are analysed in detail and implemented. To do this, he uses one of the most natural signals generated by mechanical systems: vibrations, inextricably linked to the dynamic behaviour of the mechanical systems themselves. Prof Randall also explains in detail *'why'* applying Condition Monitoring is so important.

There is also, however, another interesting '*why'* to analyse, by limiting the scope to the foreword to a book: it concerns Condition Monitoring's rapid development in recent years and its pervasiveness in the modern world. Condition Monitoring is certainly a technological innovation, and as such, its genesis and evolution can be analysed by means of the mechanisms of generating innovation, starting from the more traditional ones, such as the 'Technology-Push', theorised by Joseph A. Schumpeter way back in 1911, and the 'Demand-Pull' most recently introduced by Jacob Schmookler in 1966.

Certainly, the 'epic' and 'primordial' phase of Condition Monitoring (we could call it the 'Proto-Condition Monitoring') was governed by technology-push: without the microprocessors (introduced in the Cold War, not so much for the space race as it is commonly believed, but for the guidance and control of intercontinental missiles), without the invention of miniaturised and reliable sensors (the switch from the strain-gage to the piezoelectric accelerometer happened between the 1940s and the 1950s, with the starting up of manufacturers such as Brüel & Kjær, Columbia Research Laboratories, Endevco, Gulton Manufacturing and Kistler Instruments – some of which are still firmly on the market – or the introduction of the eddy-current proximity probe in 1961 for rotary machines by Bently Nevada), without the personal computers and without the low-cost storage systems, Condition Monitoring would have remained confined to laboratories. Very often, the hardware manufacturer also produced the necessary software and supplied the brainware: for example, minicomputers to collect data and run signal processing methods and rule-based systems for the implementation of condition monitoring. Think, for example, to Hewlett-Packard and Sohre's tables of 1968 or to Bently Nevada and their ADRE systems, and the signal processing methods developed by Donald Bently himself and Agnes Muszynska.

This phase was followed by a 'maturity' phase, governed mainly by the demand-pull, which we could call the 'Meso-Condition Monitoring', during which some large players immediately realised the benefits of Condition Monitoring and implemented it within a CBM approach, as an economic driver for cost reductions. At this stage, the leading roles were big companies and operators of 'big fleets', both in a physical and figurative sense, in various sectors: from the military (think the US Navy) to the transport and aerospace (as in the case of NASA), from the energy (first of all GE and Siemens) to the manufacturing.

However, the two traditional technology-push and demand-pull models do not explain, as it is often the case in technology, why what we might call the 'Neo-Condition Monitoring' is growing so rapidly, in more recent years. The explanation, from a technological innovation point of view, is given by an interactive vision: on the one hand, technological evolution introduces new tools (hardware in the broadest sense: sensors, wireless systems, computers and memory) and new signal processing techniques are proposed, with a frequency if not weekly, at least monthly. On the other hand, as we said, the condition monitoring market, thanks in part to the IoT, has become immense.

In light of these considerations, it is clear how important it is that we have a reference and authoritative text for ‘Vibration-based Condition Monitoring’ and all of us who work in science and technology should be grateful to Bob Randall (I now allow myself to move to a more confidential tone) for writing this second updated and expanded edition, which will certainly become a new milestone.

Paolo Pennacchi
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October 2020

About the Author

Bob Randall is a visiting Emeritus Professor in the School of Mechanical and Manufacturing Engineering at the University of New South Wales (UNSW), Sydney, Australia, which he joined as a Senior Lecturer in 1988, and where he is still active in research. Prior to that, he worked for the Danish company Brüel and Kjær for 17 years, after 10 years' experience in the chemical and rubber industries in Australia, Canada and Sweden. His research and publication record while with Brüel and Kjær was treated as PhD equivalent when he joined UNSW. He was promoted to Associate Professor in 1996 and to Professor in 2001, retiring in 2008. He has degrees in Mechanical Engineering and Arts (Mathematics, Swedish) from the Universities of Adelaide and Melbourne, respectively. He is the invited author of chapters on vibration measurement and analysis in a number of handbooks and encyclopedias, including *Shock and Vibration Handbook* (McGraw-Hill) and *Handbook of Noise and Vibration* Control (Wiley). He is a member of the Advisory Board of *Mechanical Systems and Signal Processing* and of the Editorial Board of *Trans. IMechE Part C*. He is the author of more than 350 cited papers in the fields of vibration analysis and machine diagnostics, and has supervised seventeen PhD and three Master's projects to completion in those and related areas. From 1996 to 2012, he was Director of the Defence Science and Technology Organisation (DSTO) Centre of Expertise in Helicopter Structures and Diagnostics at UNSW. He has an interest in languages and has lectured in English, Danish, Swedish, French, German, and Norwegian.

Preface to The Second Edition

In the 10 years since the first edition was published, there have been a very large number of developments in vibration-based machine condition monitoring, not only in the techniques applied, but also in its much wider acceptance in industry, as the most efficient basis for maintenance. This has, for example, changed the relative importance of intermittent manual monitoring and permanent online monitoring. The former approach covered, and still does cover, a much larger number of (simpler) machines, but the latter now represents a much larger investment in monitoring equipment and systems.

Where online monitoring was previously confined largely to expensive and critical machines in say a chemical or power generation plant, which often ran at constant speed and almost constant base load, the huge growth in development of renewable energy sources in the last decade, in particular wind turbines, first onshore, and then offshore, has prompted the development of techniques that can cope with widely dispersed smaller machines, often multiple units in large wind farms, with widely varying speed and load. Such techniques could then also be applied to other machines with difficult access, and variable speed and load, such as automated machines in manufacturing plants, and mobile equipment in mines, etc, the latter often being driverless and remotely controlled. Industry 4.0 and the associated Internet of Things (IoT) recognise the importance of including more transducers in autonomous machines, not only for automatic control, but also for optimised condition-based maintenance (CBM), and the economic benefits of CBM.

The new edition contains nine chapters instead of the original six. Chapters 1, 2, and 4 only have minor updates. Because of the need to process signals with varying speed, considerable advances have been made in order tracking, involving resampling time signals at uniform spacings in rotation angle of a machine. The accuracy of doing this has been greatly improved, allowing multiple transformations back and forth between the time and angle domains. This was found very useful to cope with the fact that shaft speed related signals, such as gearmesh frequencies, are made more periodic in the angle domain, while structural response properties, such as impulse responses, retain constant time scales in the time domain, independent of speed. It is now often possible to extract the speed information from the response signal itself, avoiding the need for a tachometer signal. A related topic is the accurate determination of the machine speed, often from the vibration signals as well. A new Chapter 5, called *Some special signal processing techniques* has thus been added to address this, but it also includes carry-over of some updated topics from the original Chapter 3, *Basic signal processing techniques*. Significant new material has been added to the remaining topics in the updated Chapter 3.

Shortly after the publication of the first edition there was a breakthrough in cepstrum analysis, which meant that time signals could be obtained by editing the cepstrum of continuous signals. This was not previously thought to be possible, since the complex cepstrum, which can be reversed back to time signals, requires continuous phase spectra, which are a property of transients only. However, there are many applications where the real cepstrum can be edited, to obtain an edited log amplitude spectrum, which can then be combined with the original non-continuous phase spectrum, to give edited time signals with very little error. Just one example of this is where the editing removes families of harmonics from the spectrum (and time signals) by notching in the cepstrum. The phase will be in error at the frequencies of the removed components, but these have been greatly reduced (to the same level as adjacent noise components) and are widely spaced, so the effect of the residual phase errors is usually negligible. This new possibility gives rise to so many new applications, including extraction of separated gear and bearing signals under varying speed conditions, and pre-processing of machine vibration signals as a precursor to operational modal analysis (OMA), that a new Chapter 6, devoted to *Cepstrum analysis applied to machine diagnostics*, has been added. It should be mentioned that OMA is now recognised as being an important part of advanced condition monitoring, to allow the extraction of force signals from measured vibration responses.

The new Chapter 7, *Diagnostic Techniques for particular applications*, is a substantially updated version of the old Chapter 5, *Diagnostic techniques*, even though many of the headings are the same. Quite recently, it has been discovered that the analysis of gear transmission error (TE) is much more powerful as a diagnostic tool than originally thought, and though introduced in the first edition, it was then thought that the required mounting of shaft encoders would restrict it to being a laboratory tool. However, with the rapid progress of Industry 4.0 and the IoT, it is becoming more common to build such transducers into machines at the design phase, also because of their benefits in control, in particular of variable speed machines. This topic has thus been considerably expanded, and important new applications developed. There have also been many other improvements in the diagnostics of gears and bearings, not least under variable speed and load conditions. While still being a very important tool, the main approach to bearing diagnostics in the first edition, based on spectral kurtosis and the kurtogram, has been shown to sometimes fail, for example when impulsive signals from other sources, such as EMI (electromagnetic interference) dominate the kurtosis, so a number of alternative approaches can now be chosen, which have benefits in many situations. Diagnostics of internal combustion (IC) engines has also been greatly improved, for example to include mechanical faults such as piston slap and bearing knock.

Another area which has become much more practicable is the ability to make realistic detailed simulation models of machine components, and even complete complex machines, using CAD, FEM, and multi-body dynamics. This has become a standard part of the machine development process, greatly reducing the number of intermediate prototypes. It has led to the adaptation of such models to individual machines (known as 'digital twins'), for example to compensate for tolerance variations, and even changes in performance with time, in particular in the now much more common situation where manufacturers are responsible for the whole-life performance and maintenance of machines. For the latter application, it would be advantageous to simulate faults in components, using substructuring techniques to incorporate them in the overall models, in place of the original healthy components. A new Chapter 8 has thus been added, entitled *Fault simulation*, giving typical approaches for the cases of gears, bearings and IC engines, largely developed since the publication of the first edition. It is particularly with dynamic machine models that OMA is required to update the simulation models, not only because they vary more with operating conditions than static structures, but also because the forcing functions are much more complex in general. Extraction of these forces, using the modal models, is also much more important in machine condition monitoring than in structures, where the condition is indicated primarily by modal properties.

Simulation is also now a very important part of prognostics, because of the impossibility of experiencing fully documented failures in sufficient quantities to use purely data driven methods, so the new Chapter 9, replacing the old Chapter 6 on *Fault trending and prognostics*, has been considerably updated, even though the ideal solution to this problem has still not been found. The sometimes blind faith in ‘Big data’ as a future solution is misplaced, although such techniques will play a large part in compensating for the effects of wide variations in operating conditions, for machines in normal, or near normal, condition, for which big data does exist. This will greatly increase the reliability of detection of departures from normal condition, and at least aid prediction of developments into the near future. The first advances will most likely be made in fleets of similar machines, e.g. wind turbines and helicopter gearboxes, where increasing numbers of documented cases will be incorporated into predictive systems, based initially on physics-based fault development models, including simulation of various degrees of sophistication, right up to digital twins. Watch this space.

Because of the large amount of new material, a large number of additional acknowledgements are required, not only for new work, but also because some earlier work has now increased in importance for machine condition monitoring. In that category must now be added Dr Yujin Gao, whose pioneering work on cepstral methods of modal analysis, forms the basis of much of the added material of Chapter 6, and Dr Yuejin Li, whose work, still not widely published, on the weighting of responses at a fixed point on the casing to faults in rotating planet bearings led to the first application of the log envelope of a bearing signal, and will almost certainly assist in the diagnostics of planet gears and their bearings in the future.

I would like to acknowledge the contributions of all the many co-authors of the large number of new papers by our group at UNSW, from 2011 to the present, but in particular would like to thank my colleagues in that period including Dr Wade Smith, Dr Pietro Borghesani, and Prof. Zhongxiao Peng, as well as my former PhD students Dr Jian Chen, Dr Lav Deshpande, and Dr Michael Coats, whose doctoral works make substantial contributions to the new edition. I am very sad to report that Dr Chen tragically passed away in July 2019, at a very young age, after a short illness. It is also with great sadness that I have to report the passing in June 2015, of Dr Peter McFadden, many of whose contributions are reported in both editions.

One of the greatest losses to the machine diagnostic community was the passing, in March 2020, of Professor Simon Braun, founder of the journal *Mechanical Systems and Signal Processing*. I would like to reiterate my acknowledgement, from the first edition, of his continued support and mentorship over many years. It is perhaps worth mentioning that over 50% of the references in the new material of the new edition were published in MSSP.

A special thanks is again due to Professor Jérôme Antoni, who moved to INSA Lyon, France, several years ago, and who has continued to be a steady inspiration and collaborator in many of the new developments.

To the list of international academic institutions, with whom collaboration has enhanced much of the new material, must be added Vrije Universiteit Brussel (VUB), Belgium, on OMA and wind turbine monitoring, and Professor Paolo Pennacchi’s group at Politecnico di Milano, on bearing diagnostics, initiated by the visit to UNSW in 2011 of Pietro Borghesani, during his PhD studies. SpectraQuest Inc., Richmond USA, supported work on bearing diagnostics and system modelling, while the former Leuven Measurement Systems (LMS, now Siemens), Belgium, supported the groundbreaking PhD project of Dr Jian Chen.

Once again, I would like to acknowledge the support of my wife, Helen, who encouraged me to keep writing at all times, even when it threw a greater load on her.

About the Companion Website

This book is accompanied by a companion website.

www.wiley.com/go/randall/vibration

This website includes:

Exercises and tutorial questions

1

Introduction and Background

1.1 Introduction

Machine Condition Monitoring is an important part of Condition-based Maintenance (CBM), which is becoming recognised 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.

Even at the time of the first edition of this book, there was a considerable body of evidence that CBM gave economic advantages in most industries. An excellent survey of the development of maintenance strategies was given by Rao in a Keynote paper at a Comadem (Condition Monitoring and Diagnostic Engineering Management) conference in 2009 [1]. Maintenance is often regarded as a Cost Centre in many companies, but Al-Najjar [2–4] has long promoted the idea that CBM can convert maintenance to a Profit Centre. Jardine et al. [5, 6] from the University of Toronto documented a number of cases of savings given by the use of CBM. The case presented in [6], from the Canadian pulp and paper industry, is discussed further in Chapter 9, in connection with their approach to prognostics.

Since the first edition of this book was written, the value of CBM has been even more accepted, and for example a recent report [7] indicates that the global value of the machine condition monitoring market will rise from USD 2.6 to 3.9 billion from 2019 to 2025. This is in part due to the increasing general acceptance of Industry 4.0, or the Fourth Industrial Revolution, which according to Wikipedia 'is the trend towards automation and data exchange in manufacturing technologies and processes which include cyber-physical systems (CPSs), the internet of things (IoT), industrial internet of things (IIoT), cloud computing, cognitive computing and artificial intelligence'. CBM is expected to be part of the 'Smart factory', and this is evidence that it is starting to be more extensively applied in manufacturing plants, for example on complex automated machine tools, whose operations have rapid changes in speed and load. This was just not possible in the early days of condition monitoring, where the monitored machines tended to run for long periods at constant speed and loads, for example in power generation and chemical plants. Some of the newer techniques to

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cope with variable speed and loads have been developed in the intervening 10 years and are described in this new edition.

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 whilst 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 primarily 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) 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 can 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, e.g. 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. It does mean that the vast majority could have run longer by a factor of two or three [8]. 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. It 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 [8].

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 are not only able to determine current condition, but also give reasonable predictions of remaining useful life. It has been used with some success for 35–45 years, and for example the abovementioned report [8] 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 the last 20 years or so that it has become recognised 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). Refs. [9, 10] discuss the potential benefits given by CBM applied to hydroelectric power plants and wind turbines, 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 condition 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' [11].
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 etc. 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 along with a couple of other methods, performance analysis and thermography, that have more specialised 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 gear teeth, rotating electric fields, etc. 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, i.e. 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 mainly from accelerometers (acceleration transducers) mounted on the machine casing.

It should perhaps be mentioned that a related technique, based on the 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–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 [12]. Even though they 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 [13], 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 recordings were based on event triggering. In [14], 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.

There is some evidence that AE measurements on rolling element bearings can detect very small incipient faults a little earlier than vibration measurements (e.g. [15]), but it is almost certain that the frequency range of the excitation will fall below 100 kHz as the faults become physically larger, after which the AE transducers could not be relied upon to follow the fault development. In the author's opinion, other methods which can detect resonant responses in the frequency range 40–100 kHz, are likely to detect bearing faults almost as early as AE transducers, and would still give sufficient advance warning to allow prognostics of the fault development as the frequency range of the excitation reduces with increase in fault size.

One such method is the proprietary 'PeakVue' method incorporated in Emerson CSI analysers [16], where the signal used to generate the envelope is down-sampled from the original rate of 102.4 kHz in such a way that envelope information is not lost. If the sample rate is to be reduced by a factor of 50, for example, to give an envelope spectrum range of about 1 kHz, instead of simply retaining every 50th sample (which might completely miss short high frequency pulses) the absolute peak value in every 50 samples is retained. Because the signal is not lowpass filtered before down-sampling, this of course gives aliasing (see Chapter 3), but only of the high frequency carrier, which does not contain diagnostic information. The important information about the repetition frequency of the response pulses (from a bearing fault, for example), is contained in the envelope signal, which is retained, as explained in [17]. The down-sampling approach in [15] is based on the same principle. The high frequency signals (up to the original lowpass filter frequency of about 40 kHz) containing the bearing fault pulses, are almost certainly dominated by the accelerometer mounting resonance, which as pointed out in [16] varies considerably based on the mounting method. It is therefore unlikely that the PeakVue method will give repeatably scaled measurements, suitable for trending, but it will often give a very early detection and diagnosis of a bearing fault. The same accelerometer as used for the PeakVue method is also used for conventional vibration analysis in the lower frequency range where the accelerometer response is linear and more repeatable.

Another somewhat similar approach, the Shock Pulse Method (SPM), specifically uses the accelerometer resonance to carry information about high frequency fault pulses, such as from bearings, and uses the demodulated signal for a range of diagnostic purposes, including envelope spectrum analysis. In this case the accelerometer mounted resonance is specified as 32 kHz, and is ensured by tight control of the mounting method, including the use of steel rod wave guides to carry the signal from the machine casing to where it can more easily be measured. The conventional SPM techniques use only this resonant response, but from 2015, a new ‘DuoTech’ transducer has been made available which can cover the conventional lower frequency vibration range as well ([18]).

Because of the difficulty of application of AE monitoring to machine condition monitoring there are only limited further discussions 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, etc. Alternatively, suspended particles can be detected in flow past a window.
2. **Spectrographic Oil Analysis Procedures (SOAPs).** 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, etc. 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.

Since the first edition of this book was published, more advanced online techniques have been developed ([19, 20]). Ref. [19] describes online measurement of oil viscosity and other parameters such as different particle concentration levels, while Ref. [20] uses online image analysis to obtain particle quantity, size, shape, and colour. This does help to remove some of the problems involved in sending out oil samples to external organisations for analysis, including much more extensive sampling, and greatly reduces the time involved.

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, etc. 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, e.g. 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 point in time, thermography is still 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. There is some development towards the combined use of wear debris analysis and vibration analysis, the first indicating the type and total amount of wear, and the second the detailed distribution of the wear, but this book concentrates on the vibration analysis part.

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 turbo-machines in, for example, power generation plants and petro-chemical 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 in Section 1.5.4.

Note that ‘permanent’ or ‘online’ monitoring is not synonymous with ‘real-time’ monitoring, which is rarely required in machine monitoring, as compared, for example, with automatic control. Real-time processing implies causal signal processing, which has severe disadvantages compared with non-causal processing, as typified by the use of the Fast Fourier Transform (FFT), which is perhaps the dominant signal processing tool in machine monitoring. As shown in Chapter 3, use of the FFT involves batch processing of time records selected out of a continuous record, but individually treated as though repeated periodically. This means that the second half of each time record is implicitly treated as negative time (thus non-causal), but gives huge advantages in that it allows for almost ideal filters, with no phase distortion, which can completely exclude adjacent strong frequency components, not possible with causal filters, which have a much more gentle ‘roll-off’, and introduce phase shifts over relatively wide frequency ranges. The delay involved in the non-causal processing is just the processing time of one such time record, and would rarely be more than a second or so, this normally being negligible compared with the time constants associated even with real-time processing. Even if a large turbomachine were automatically tripped (that would be rare because a human would normally have to make that decision) the speed (which is normally reduced over a period of many hours) would not change significantly in the few seconds difference between causal and non-causal processing. Another example where real-time processing is of no advantage is if the machine being monitored were, say, an aircraft engine, what would be the difference between the warning ‘this engine will self-destruct in two seconds’, compared with two milliseconds? Neither would be of much use.

The **advantages** of permanent, or online, 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.

- It is sometimes more economical to have permanently mounted transducers on widely distributed and difficult-to-access machines, such as wind turbines, and automated manufacturing machines, and then the additional cost of transmitting the collected signals back to a centralised monitoring system is economically justified. This approach is now being applied also for mobile equipment, such as mining trucks and machines, many of which are autonomous.

The **disadvantages** of permanent monitoring are:

- The cost of having permanently mounted transducers is very high, so previously could only be justified for the most critical machines in a plant, or where it is difficult for operators to access the machines. However, the cost of transducers such as accelerometers is continuously being reduced, and with the increased development of autonomous machines, more in-built transducers are required anyway, in conformity with the 'Internet of Things'. Because of the increasing realisation of the benefits of CBM, online monitoring is being extended to more and more machines.
- Where the transducers are proximity probes, they virtually have to be built in to 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 in this book.

Of course, if transducers are mounted permanently, it is still possible to analyse the signals in more detail, just not continuously. It 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 many plants, it is not economically justified to have them equipped 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 and acquire replacement parts etc., it is not always 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 [21]. 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 maximise 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 [2–6].
- 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 [18].

To summarise, **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, viz. 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).

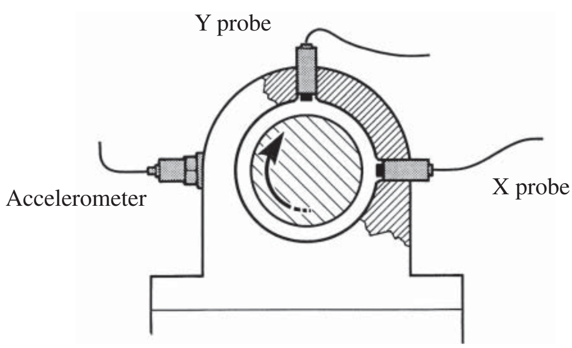

Figure 1.1 Illustration of absolute vs relative vibration. Source: Courtesy Brüel & Kjær.

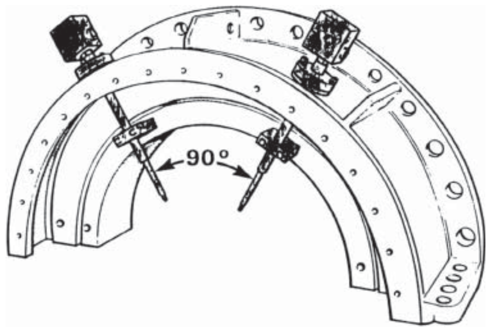

Figure 1.2 Proximity probes installed in a turbine bearing cap.

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 realise 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 [22]. It is thus also very important in rotor dynamics calculations, as these are greatly influenced 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 also the vibrations around the mean position. 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. It should be kept in mind that if the zero frequency expected values of acceleration or velocity were different from zero, the machine would not be staying in the same place.

Other comparisons between the different types of transducers depend on the technical specifications for dynamic range, frequency range, etc., 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, becoming a standard for the American Petroleum Institute (API) [23]. 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 0.25–2.3 mm with a maximum deviation from linearity of 0.025 mm (1.1% of full scale) with a sensitivity of 200 mV mil^{-1} (7.87 V mm^{-1}). Thus, in the sense of the ratio of maximum to minimum value, the dynamic range is <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 [24]. 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 minimise runout before a shaft goes into service [24], 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 [23], 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 Section 4.2.1, mechanical vibrations tend to have roughly uniform spectra in terms of velocity, and thus reducing 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 1.3a) 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 1.3b), 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

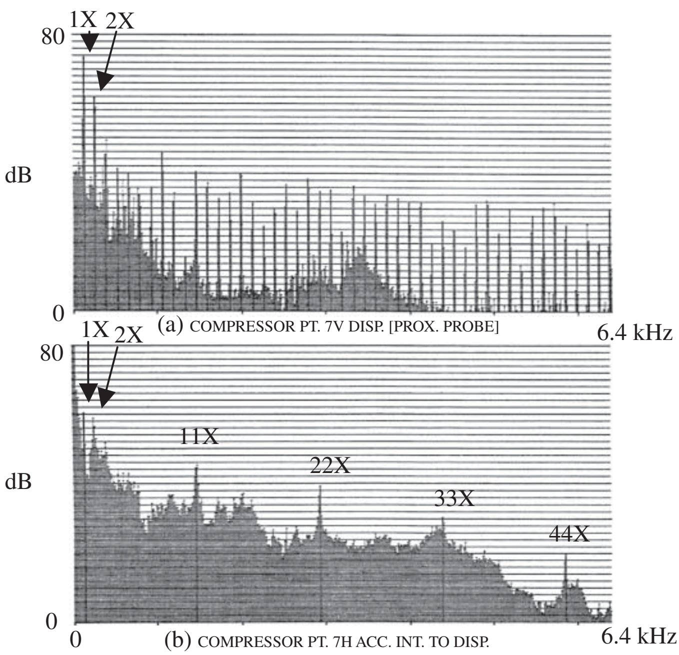

Figure 1.3 Comparison of spectra measured on a centrifugal compressor (a) Proximity probe (b) Accelerometer signal (integrated to displacement).

be useful for diagnostic purposes (see Section 2.2.4). 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 5.3.1) 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 first 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 [25]) shows the frequency response of a generalised 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 1 (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 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.

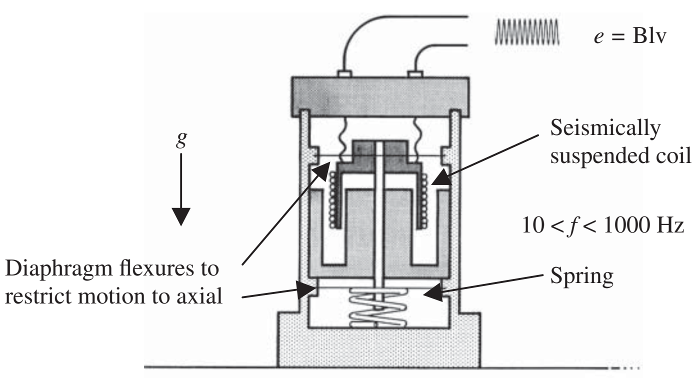

Figure 1.4 Schematic diagram of one realisation of a velocity pickup. Source: Courtesy Brüel & Kjær.

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–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 type 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 1.6a, 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 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 pico-Coulombs per metre per sec^2, (pC/ms^{-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

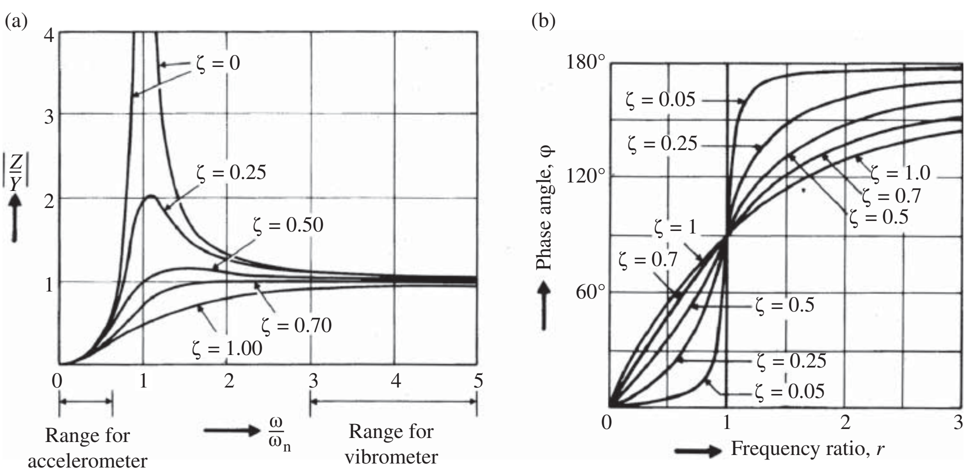

Figure 1.5 Frequency response of a seismically suspended vibrometer [25]. (a) Amplitude characteristicy (b) Phase characteristic. ζ = critical damping ratio.

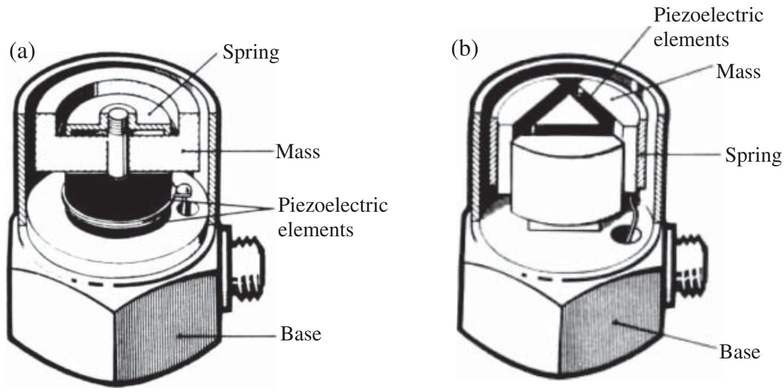

Figure 1.6 Typical accelerometer designs (a) Compression type (b) Shear type. Source: Courtesy Brüel & Kjær.

the piezoelectric elements, so its force remains effectively constant, but it is required to maintain a positive compression force on the assembly.

Figure 1.6b shows an alternative design where the piezoelectric elements deform in shear (they must be polarised so as to produce a charge proportional to shear rather than compressive strain). This particular design is the patented 'delta shear®' design, by Bruel & Kjaer, 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 electrical 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 minimised 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 minimised 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 having the charge amplifier built into the transducer. Miniaturisation of electronic circuits has now made this possible, and it does solve many of the practical problems associated with special cables, electrical interference, etc. 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 etc. 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 1.5a 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 lowpass 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 few percent, the signals can become distorted if the transverse resonance is excited. One 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 1/3 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 1/3 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, and 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, i.e. 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. It 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 lowpass filter, with slope – 20 dB per frequency decade, the dynamic range of the combination is still >60 dB over a frequency range of three decades. It is illustrated how this three decade range can be simply switched from (for example) 10 Hz–10 kHz to 1 Hz–1 kHz, even with the same accelerometer and (external) amplifier. This is obviously much more flexible than

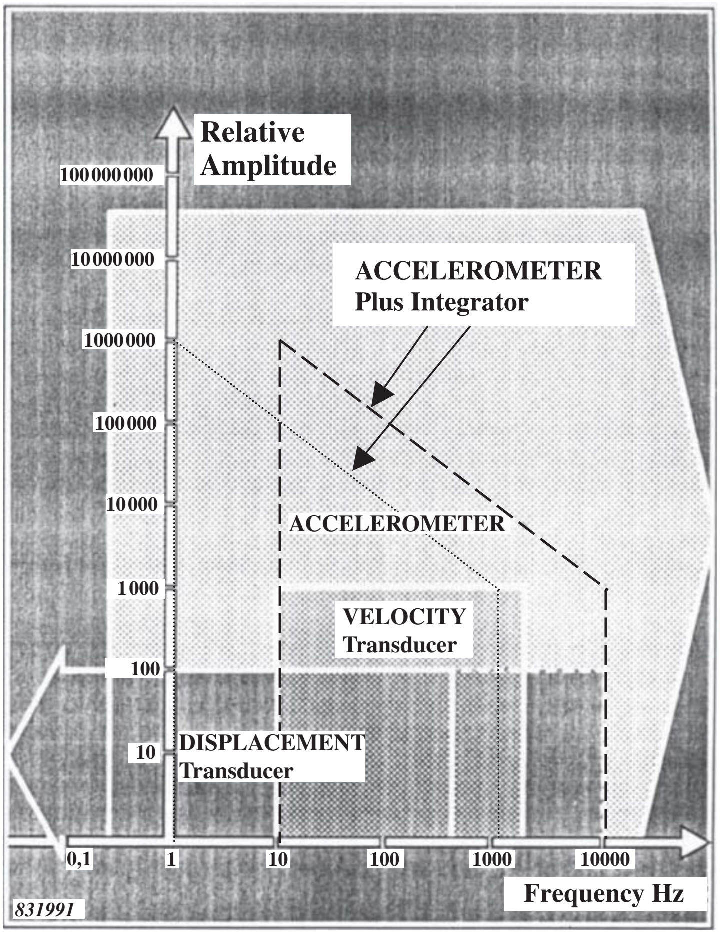

Figure 1.7 Typical frequency and dynamic ranges for the three main transducer types with superimposed ranges for an accelerometer and integrator. Source: Courtesy Brüel & Kjær.

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 3 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 it meant that this 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 noise. It can be done successfully numerically, as long as the frequency components above the highest valid frequency (not dominated by noise) are first removed by lowpass filtration.

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 either be 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 does measure 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) they 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 miniaturised 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 they 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 [26].

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 7.4.2. 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 7.2.2.

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, 7.2.2, and 7.4.2. 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 differentiation 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 (typ. 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 5.1) to compensate for slow speed changes. Both methods are discussed and compared in [27].

1.6.2 Torsional Laser Vibrometers

Torsional laser vibrometers also exist, which have two laser beams directed at the surface of a rotating shaft [28]. 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 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 8.4.1, 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:

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

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

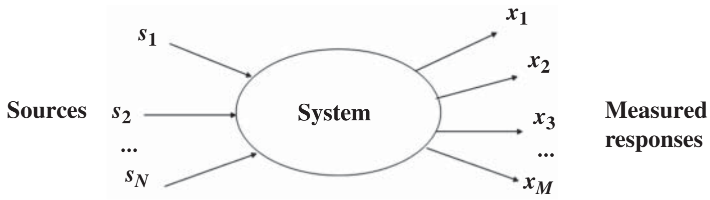

Figure 1.8 Illustration of a number of measured responses of a system due to a number of excitation sources.

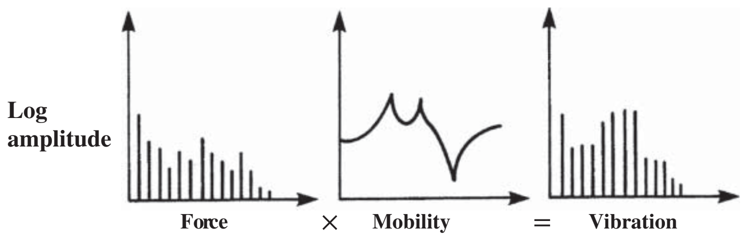

Figure 1.9 Combination of forcing function and transfer path to give response vibration for one source. Source: Courtesy Brüel & Kjær.

where the upper-case letters in Eq. (1.2) represent the Fourier transforms of the lower-case symbols, and the asterisk represents the convolution operation (Section 3.2.6).

For an individual source (forcing function) in Eq. (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 real part, and phase as 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 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 recognisably 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 [400, 800, 1200] Hz with a narrowband resonance at 1000 Hz excited by white noise (averaged over 1000 spectra), and using Hanning weighting. The difference cannot be seen on linear amplitude scales.

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.

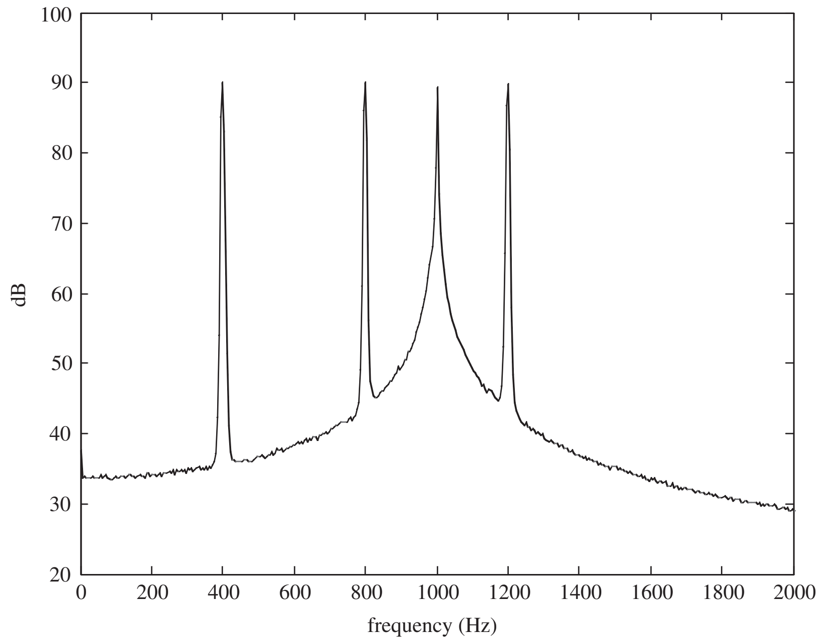

Figure 1.10 Comparison of the spectra of discrete frequency components at 400, 800, and 1200 Hz with that of a narrow band resonance at 1000 Hz.

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 7.2.4), 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', i.e. opening and closing with 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 generalisation 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 non-linearities, 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 7.1, the presence of harmonics (perhaps only visible on a logarithmic amplitude scale) allows a much more accurate measurement of the fundamental frequency, by fitting a finely tuneable ‘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 a 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 Ref. [29].

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, and there is a special series of conferences, IOMAC (International Operational Modal Analysis Conference) specifically devoted to it. It is used for updating of simulation models of machines and faults in Chapter 8.

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