Acoustics Based Condition Monitoring

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1. Introduction

Condition monitoring involves the continuous or periodic assessment of the condition of a plant or a machinery component while it is running. The use of conditional monitoring allows maintenance to be scheduled, or other actions to be taken to avoid the consequences of failure, before the failure occurs. It is typically much more cost effective than allowing the machinery to fail. Serviceable machinery includes rotating machines and stationary plant such as boilers and heat exchangers.

Basically condition monitoring is the process of monitoring some parameters from the machinery, such that a significant change in the parameter can give information about the health of the machinery. It involves the continuous or periodic assessment of the condition of a plant or a machinery component while it is running. With acoustics condition monitoring, a machine running in a good condition has a stable noise spectrum. Spectrum changes when the condition changes. Identification of noise sources and comparing their spectrum with that of a stable spectrum of a machine or plant in good condition can prove to be an important tool for condition monitoring. The main objectives for the condition monitoring can be listed as

a) Prediction of faults
b) Diagnosis of faults
c) More safety at work place
4) Less down time leading to more production and
5) Better inventory management for the spare parts

Although the main use for condition monitoring is to predict and hence assist in avoiding the unplanned equipment failures. However, there are other means in which condition monitoring can assist in improving maintenance during the Planned Maintenance phase. It helps in minimizing total equipment downtime by taking a holistic view of plant condition, and combining planned maintenance tasks, wherever possible, into a single equipment shutdown. It makes workplace safer by reducing the equipment failure. Information obtained during the condition monitoring also helps in optimizing the equipment performance. Different techniques for the condition monitoring are

1. Visual inspection
2. Vibration based analysis
3. Acoustics based analysis
4. Wear debris analysis
5. Performance analysis and
6. Thermography and other NDT techniques

The most basic form of condition monitoring is visual inspection by experienced operators and maintainers. Failure modes such as cracking, leaking, corrosion, etc can often be detected by visual inspection before failure is likely. Other forms of condition monitoring should generally augment, rather than replace, visual inspection. Wear debris detection analysis can be used rotating equipment such as gearbox's, bearings, turbines etc. Detecting ferrous and non-ferrous wear particles within the lubrication oil gives considerable information about the condition of the measured machinery. By creating and monitoring a trend of what debris is being generated it is possible to detect faults prior to catastrophic failure. Thermography can be used where heat is a indicative of failures such as electrical contacts and terminations. It can also be used to fine out defects in high-speed bearings, conveyor rollers, shafts and other metal parts.

Noise and vibration signal from machine can contain vital information about the internal process and can provide valuable information about a machine running condition. Noise signal are measured in a reason proximity to the external surface of the machine while vibration signals are measured on the surface of the machine. Most noise and vibration analysis instruments utilize a Fast Fourier Transform (FFT) which is a special case of the generalized Discrete Fourier Transform. It converts the vibration signal from time domain representation to its equivalent frequency domain representation. This Frequency domain representation of the time history is called frequency spectra. When a machine is in good condition, their noise and vibration frequency spectra will have some characteristic shape. As faults begin to develop, the spectra change. But in most of the cases these desired signals are mixed with some undesirable signal. Hence, analysis of these frequency spectra needs some specialized signal processing to relate it with the actual cause of fault in machineries.

In contrast to vibration signals, acoustic signals from a machine can be measured at distances sufficiently far from a vibrating machine surface with help of microphones. This allows the acoustics signal to be measured for machines working at a condition (extreme temperature and humidity) which is not suitable for vibration measuring sensors, like proximity transducers or accelerometers. In certain situations, it is advantageous to use single acoustics sensor to get superimposed acoustic signals from more than one machine elements and to correlate it later to know the health of the different machine elements which otherwise may not be possible with the vibration signals. But at the same time, acoustics transducers can pick up signal from other unwanted sources very easily. The limitation of the vibration monitoring method is that other conditions, such as uncertainties in the geometry of the test object, its surface conditions, and loading can also affect the vibration response, and it is necessary to distinguish the effects due to harmful conditions from those due to harmless ones. A basic of acoustics is presented in the next section.

2. Introduction to Acoustics

2.1 Sound

Sound may be defined as a time-varying disturbance of the density of a fluid from its equilibrium value, which is accompanied by a proportional disturbance of sound pressure and is associated with small oscillatory movements of fluid particles. Acoustic disturbance propagates at a characteristic speed, the ‘speed of sound’. The most commonly measured acoustical quantity is sound pressure. Pressure is measured because (1) the mechanical input to the auditory system is pressure; (2) It is the easiest acoustical quantity to measure; (3) mechanical structures respond to pressure.
Because of the electronic characteristics involved in most of the sound measurements, the term sound pressure is conventionally understood to be effective pressure or root mean square (rms) pressure. It is defined as

$$P^2 = \lim_{T \to \infty} \frac{1}{T} \int_{-T/2}^{T/2} p^2(t)dt \tag{1.1}$$

where T is the time duration over which the mean or average is taken and $p(t)$ is the pressure**.**

Sound waves transport energy in the form of kinetic energy of particle motion and potential energy of elastic strain. The measure of the magnitude and direction of the local rate of energy transport is termed the sound intensity and is donated by $\boldsymbol{I}$. Unit of this vector quantity is W/m^2. It is defined as

$$I = \frac{1}{T} \int_{-T/2}^{T/2} P(t) u(t) dt \qquad (1.2)$$

where T is the period of one cycle of harmonic motion and $\boldsymbol{u}$ is the particle velocity.

The output of sound source is generally quantified in terms of the radiated sound power W and its unit is the watt. This quantity is largely independent of the environment of the source, unlike sound pressure, which varies with distances and directional location.

The total power output of a source is given by integrating the intensity traveling across a surface which totally encloses the source. The sound power output is defined by

$$W = \int_{s} \boldsymbol{I}.\boldsymbol{n} dS \qquad (1.3)$$

where $\boldsymbol{n}$ is the unit vector pointing outwards from the closed surface S.

Human perception of sound ranges from a lower limit of 20 μPa to an upper limit of about 200 Pa. This represents a considerable range – about 10^7. Because of this, it is more convenient to firstly work with relative measurement scales rather than absolute measurement scales, and secondly to logarithmically compress them. Hence logarithmic equivalents of P_{rms}^2, I and W are conventionally employed. Their common unit is the decibel (dB). The logarithmic measures are defined as,

Sound pressure level, $L_P = 10log_{10}\left[\frac{P_{rms}^2}{P_{ref}^2}\right]$ dB; (1.4)

Sound intensity level, $L_I = 10log_{10}\left[\frac{I}{I_{ref}}\right]$ dB; (1.5)

Sound power level, $L_w = 10log_{10}\left[\frac{W}{W_{ref}}\right]$ dB; (1.6)

Sound pressure level and Sound intensity level at a point in far-field

For a plane wave traveling in the positive x-direction, omitting the vector notation,

$$u(t) = \frac{p(t)}{\rho_o c}, \qquad (1.7)$$

where ρ_o is the density of the medium and c is the velocity of sound in that medium. The mean sound intensity is obtained by substituting the value of $u(t)$ in equation (1.2),

$$I = \frac{1}{T} \int_{-T/2}^{T/2} \frac{P^2(t)}{\rho_o c} dt, \qquad (1.8)$$

or,

$$I = \frac{1}{\rho_o c} \frac{1}{T} \int_{-T/2}^{T/2} p^2(t) dt, \qquad (1.9)$$

or, $$I = \frac{P^2}{\rho_o c} \,, \tag{1.10}$$

Now the sound intensity level L_I is given by,

$$L_I = 10log_{10}\left[\frac{I}{I_{ref}}\right] \text{dB}\,, \tag{1.11}$$

Substituting the value if I in terms of p, in the above expression,

$$L_I = 10log_{10}\left[\frac{P^2}{\rho_o c} \times \frac{1}{I_{ref}}\right] \text{dB}\,, \tag{1.12}$$

or, $$L_I = 10log_{10}\left[\frac{P^2}{P_{ref}^2} \times \frac{(2 \times 10^{-5})^2}{\rho_o c \times 10^{-12}}\right] \text{dB}\,, \tag{1.13}$$

or, $$L_I = L_P + 10log_{10}\left[\frac{(2 \times 10^{-5})^2}{\rho_o c \times 10^{-12}}\right] \text{dB}\,. \tag{1.14}$$

Value of $\rho_o c$ in the above equation is pressure dependent. At 20^{o}C and 1 atmospheric pressure it is 0.16 dB. Hence for all intents and purpose, $L_I \approx L_P$.

Relationship between sound pressure levels at two points in free-field sound propagation for point sources

A sound source can generally be modeled as a point spherical sound source if its diameter is small compared with the wavelength that it generated, or if the measurement position is at a large distance away from the source.

For a sound source of power W, sound intensity at distances R_1 and R_2 is given by $I_{R_1} = \frac{W}{4\Pi R_1^2}$ and $I_{R_2} = \frac{W}{4\Pi R_2^2}$ respectively. Hence,

$$L_{I_{R1}} = 10log_{10}\left[\frac{W}{4\Pi R_1^2} \times \frac{1}{I_{ref}}\right], \text{ and} \tag{1.15}$$

$$L_{I_{R2}} = 10log_{10}\left[\frac{W}{4\Pi R_1^2} \times \frac{1}{I_{ref}}\right], \tag{1.16}$$

Rearranging $L_{I_{R2}}$,

$$L_{I_{R2}} = L_{I_{R1}} - 20log_{10}\left[\frac{R_2}{R_1}\right], \tag{1.17}$$

Hence for far-field conditions, assuming $L_I \approx L_P$,

$$L_{P_{R2}} = L_{P_{R1}} - 20log_{10}\left[\frac{R_2}{R_1}\right] \tag{1.18}$$

Equal Loudness Contours and Frequency Weightings

Loudness level is the sound pressure level at which a 1 kHz tone is judged as loud as another sound. The unit of loudness level is decibel, but to distinguish it from ordinary decibels of sound pressure level, it is called the phon [3]. Equal loudness curves are determined by asking the listeners to adjust the level of variable frequency pure tone, until it matches the subjective loudness of a reference pure tone, normally at 1 kHz. A complete set of iso-loudness curves are then plotted out for the different levels of the reference pure tone.

A weighted frequency scale is a table of correlations applied to sound pressure levels on a power basis as a function of frequency. Over the years, dozens of frequency weighting scales have been developed. Currently, for community and industrial noise measurements, A, B, and C frequency weightings are used. These three frequency weightings are standardized so as to roughly match the assumed frequency response of the ear as implied by the standard equal loudness contours. Three different frequency weightings are required because equal loudness contours become flatter at increasing sound level. The A frequency weighting was intended for use at 40 dB sound level, the B frequency weighting at around 70 dB sound level and C frequency weighting at around 100 dB sound level. But due to some advantages during measurements, when taking A weighted measurements and the reasonable correlations obtained between A frequency weighted measures of sound level and reported annoyance or fatigue, etc., the A frequency weighting is in wide use.

In the next section, we have presented relations between different faults in the machine elements and their effect on the acoustic signals.

2. Acoustics for Condition Monitoring

Both the time domain and the frequency domain signals are used for the fault identification. Sound pressure level in the frequency domain is the most used tool for the fault identification in machine. Fault detection for machines having moving parts such as gears, rotors and shafts, rolling element bearings, journal bearings, and flexible coupling have some common features. They are (i) Rotational speed of the moving part, (ii) Background noise, (iii) Location of the transducer, (iv) The dynamic interaction between the item and the other items in contact with it (v) Dynamic interaction between the vibrating surface and the surrounding fluid.

The factor which is common to all items involving rotational motion is rotational motion itself - the dominant noise and/or vibration frequency is always related in some manner to it. For example, in gears, the main factors would be the rotational speed, location of microphones and the interaction of two gears.

2.1 Frequency Analysis of Gears

The main frequency at which gearing induced vibrations will be generated is the gear meshing frequency, f_m given by

$$f_m = nt \times rpm/60\,. \tag{2.1}$$

where, nt is the number of teeth, and rpm is the rotational speed of the gear. The dominant source of noise and vibration in gears is the interaction of the gear teeth. The discrete, impulsive noise is associated with the various meshing impact processes, and the broadband noise is associated with friction, fluid flow, and general gear system structural vibration and noise radiation. Even when there are no faults present, the dynamic forces that are generated produce both impulsive and broadband noise. The expulsion of fluid from meshing gear teeth can also sometimes produce shock waves. The increase in noise level at the gear meshing frequency and the various harmonics is associated with wear. The most common gear fault is a discrete gear tooth irregularity such as a broken or chipped tooth. With a single discrete fault, high noise and vibration level scan be expected at the shaft rotational frequency and its associated harmonics.

2.2 Frequency Analysis of Rotors and Shafts

The two most common faults associated with rotating shafts are misalignment and unbalance. With misalignment, the vibration is both radial and axial, and the increase in vibration is at the rotational frequency and the first few harmonics. With unbalance, the vibration is generally radial and increase in vibration is at rotational frequency.

2.3 Frequency Analysis of Bearings

The primary noise and vibration mechanism for rolling contact bearings is the impact process between the rolling elements and the bearing races. The primary noise and vibration mechanism for the sliding-contact bearings is the friction and rubbing that occurs when there is inadequate or improper lubrication.

With inadequate or improper lubrication, journal film can break down producing a 'stick-slip' excitation of the shaft and other connected machine components. Another common source of noise and vibration in journal bearings is oil whirl. Oil whirl can be identified as noise or vibration at a frequency which is approximately half the shaft rotation speed. It occurs at half the shaft rotational speed because the oil film next to the shaft rotates at the shaft speed and the oil film next to bearing is stationary; hence the average oil velocity is half the shaft speed. For rolling contact bearings, the vibration comes from four main sources. They are (i) Bearing element rotations, (ii) Resonance of the bearing elements and attached structural supports, (iii) Acoustic emission, and (iv) Intrusive vibrations.

Bearing element rotations generate vibrational excitation at a series of discrete frequencies, which are a function of the bearing geometry and the rotational speed. These are the frequencies which provide information about the condition of the inner race, outer race and rolling elements of a bearing. Several discrete frequencies can be expected from rolling contact bearings and are a function of bearing geometry and the rotational speed. They are

1. Shaft rotational frequency, $f_s = N/60$
2. the rolling element pass frequency on the outer race, which is associated with outer race defects;
3. the rolling element pass frequency on the inner race, which is associated with inner race defects; and
4. the rolling element spin frequency, which is associated with ball or ball cage defects.

All these defects initially manifest themselves as narrowband spikes at the respective frequencies. As the size of a bearing defect increases, the bandwidth of the narrowband spike increases and it eventually becomes broadband and the overall vibration energy associated with the defect increases.

2.4 Frequency Analysis of Fans and Blowers

The two main types of fans are centrifugal and axial. As a general rule, axial fans are noisier than radial fans because they require higher pressures.The main sources of discrete noise in centrifugal fans are:

1. pressure fluctuations that are generated as the blades pass a fixed point in the space and 2. pressure fluctuations that are generated as the blades pass the scroll cut off point.

They generate a family of discrete tones with the blade passing frequency being the fundamental. This blade passing frequency (and its associated harmonics) is given by

$$F_b = \left(rpm \times N_b \times n\right)/60\,, \tag{2.2}$$

where N_b is the number of blades, and $n = 1, 2, 3$, etc. In addition to this harmonic family, there is significant broadband aerodynamic noise associated with vortex shedding, turbulence, etc.

Axial flow fans generate more noise than centrifugal fans. In addition to the discrete blade noise and broadband aerodynamic noise, there are several other mechanisms which result from non-linear interactions between the blades and the fluid and the blade and the wake. The dominant interaction noise components are associated with rotating unsteady pressure patterns, the resultant noise of which is also at the blade passing frequency and its associated harmonics.

2.5 Frequency Analysis of Pumps

Typical examples hydraulic pumps include centrifugal, reciprocating, screw and gear pumps. The most common vibration problems associated with pumps are unbalance, misalignment, defective bearings and resonance. They manifest themselves as a result of (i) hydraulic forces, (ii) cavitations and (iii) recirculation.

Hydraulic forces manifest themselves as discrete frequency noise and/or vibration at frequencies corresponding to the total number of compression or pumping events per revolution multiplied by shaft r.p.m. and its harmonics, i.e.

$$F_p = \left(n \times rpm \times N_p\right)/60, \tag{2.3}$$

where, N_p is number of compression or pumping events per revolution and $n = 1, 2, 3$, etc. For instance, the total number of pumping events associated with centrifugal fans is related to the number of impeller vanes – the hydraulic forces associated with pressure pulsations within the pump, which are generated as an impeller vane passes a stationary diffuser.

In addition to the above mentioned discrete frequency noise and vibration, pumps also display broadband noise characteristics due to turbulence, cavitations and recirculation.

2.6 Frequency Analysis of Electrical Equipment

Most electrical equipment is a source of noise and vibration. Some typical examples include transformers, electric motors, generators and alternators. The noise and vibration from electric motors, generators and alternators is particularly useful as a diagnostic tool.

Transformer noise is generally associated with the vibration of the core and the windings due to magnetomotive forces. The noise associated with these vibrations is a discrete frequency hum at twice the supply frequency and at its associated harmonics.

Noise and vibration sources in electric motors are of three types:

1. mechanical, 2. aerodynamic, and 3. electromagnetic.

Mechanical problems are generally associated with defective bearings, unbalance, looseness, misalignments, and winding damage due to mechanical shock, impact or fretting, etc. Aerodynamic problems are generally associated with ventilation fans and include such items as discrete blade passing frequencies, resonant volume excitations within the motor housing, broadband turbulence, etc.

Some basic relationships for electric motor bearing vibration signal are:

1. 1 $\times$ shaft rotation frequency and associated harmonics- mechanical unbalance;
2. 2 $\times$ shaft rotation frequency and its associated harmonics- misalignment between the motor and the driven load;

The electrical supply frequency, f_e, is related to the shaft rotational frequency, f_s, by $f_e = \left(f_s \times p\right)/2$ where, p is the number of magnetic poles.