Vibes scope machine health

Short of a complete tear down, vibration analysis is one of the best ways for assessing the condition of rotating machinery.

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An engineer checks a machine with a handheld vibration analyzer.

A vibration analysis can quickly pinpoint problems with bearings, alignment, and balance in rotating machinery such as pumps, fans, and compressors. There are two basic approaches to vibration analysis: continuous monitoring systems (CMS) and portable data collection. CMS, as the name implies, continuously monitors vibration levels of a machine or small groups of machines. Vibration sensors permanently mount on the machines and transmit data to a PC through an a/d converter. The sensors connect to the converter by direct wire, wireless link, or Ethernet. The PC typically outputs to a PLC with alarm or machine-trip capabilities, or both. Continuous monitoring is used primarily for critical machinery that has high downtime costs.

The portable data-collection method uses a battery-powered, handheld vibration analyzer. The analyzer wires directly to an accelerometer, which typically has a magnetic base for attachment to a machine-bearing housing. A bearing housing offers the best transmission path from a rotating member to the sensor. Vibration data downloads from the analyzer to a PC for review. Analyzers store various preprogrammed data sets chosen to match a machine's parameters including rpm and overall vibration level. While data sets vary from machine to machine, the same set is used consistently for a given machine. This allows an apples-to-apples comparison of vibration data taken over time.

Such a trend analysis that compares historical to current vibration levels is preferred to taking a single vibration measurement. For example, a machine located on the second floor of a building will vibrate more than an identical machine sitting at ground level on a concrete base. An engineer taking a single measurement from

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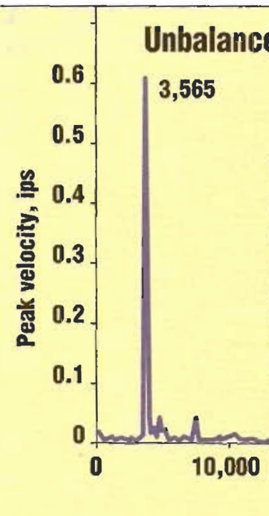
each machine may conclude that there is a defect with the machine on the second floor when, in fact, it is because the machine mounting location is less stiff. A trend analysis establishes a baseline for vibration levels that accounts for machine mounting. When a trend is not available, an overall peak velocity reading of 0.25 ips or less is acceptable for typical machinery. Equipment used in precision machining operations may tolerate substantially less vibration. The ISO Standard 10816 and the **General Motors** Vibration Standard (under development) list acceptable vibration levels for various machines.

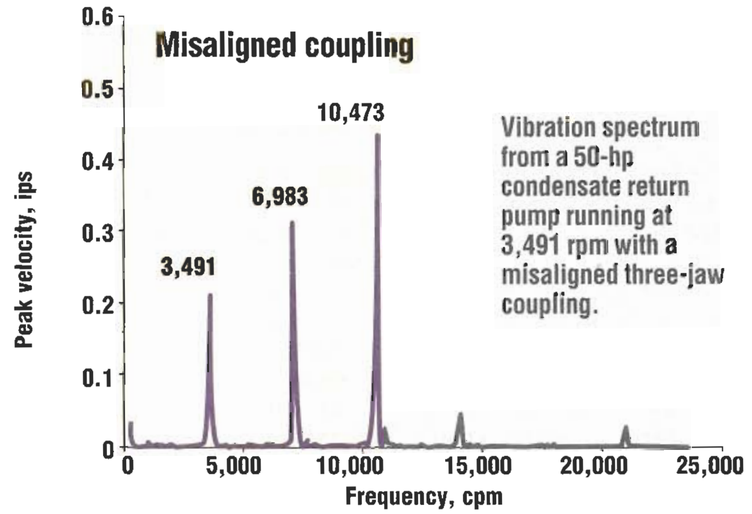
Commonly used units in vibration analysis are displacement (mils), velocity (ips) and acceleration (*g*). Displacement gives poor signal response at mid-to-high frequencies and therefore is used primarily for balancing. This is because unbalance problems show up at running speed, which is one of the lower frequencies of interest in a vibration spectrum. Displacement-signal response decays at higher frequencies, partially filtering out unwanted contributions from, say, looseness or misalignment, and leaving only the unbalance signal.

Velocity is best for evaluating vibration problems up to 1 kHz, such as looseness and coupling issues. Acceleration gets the nod for investigating short-duration (high-frequency) events above 1 kHz. including faulty bearings and gears, and motor/electrical issues such as damaged rotor bars, loose stator coils, and malfunctioning dc drives.

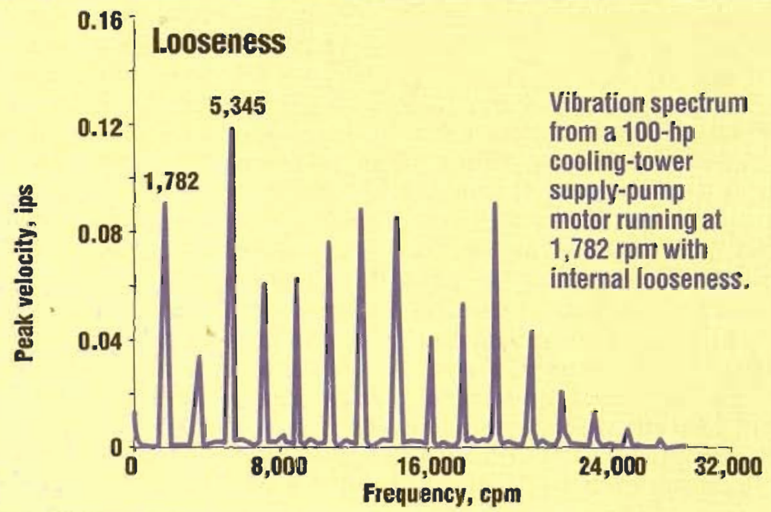
For example, a machine bearing with a spall on one of its races will see an impact as each rolling element passes over the spall, similar to a car tire hitting a pothole. Impacts are of short dura-



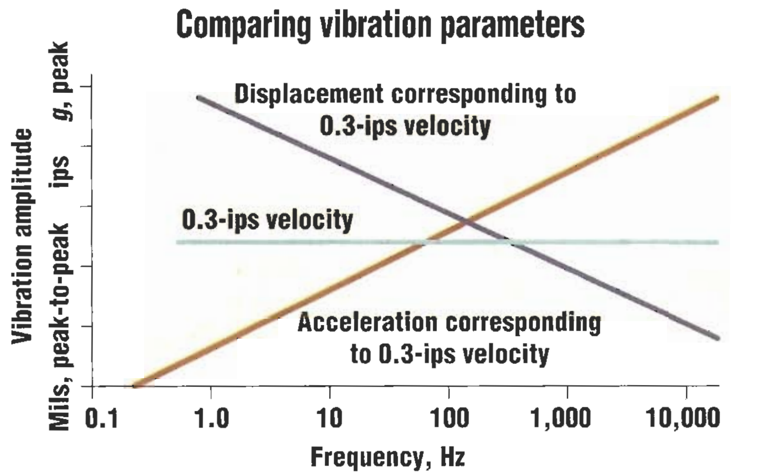
Vibration spectrum from a 100-hp furnace combustion air blower with an unbalance condition. The blower is supported by rolling-element bearings and turns at 3,565 rpm, the same frequency as the velocity peak. Note that a bent shaft, which has a center of gravity offset from its center line, gives the same type of vibration signature as an unbalance condition.



Vibration spectrum from a 50-hp condensate return pump running at 3,491 rpm with a misaligned three-jaw coupling.



Vibration spectrum from a 100-hp cooling-tower supply-pump motor running at 1,782 rpm with internal looseness.



A vibration signal of 0.3 ips displayed in displacement, velocity, and acceleration parameters. Under 5 Hz, low acceleration values would be nearly indistinguishable from signal noise. However, above 1 kHz, acceleration provides a better response signal than velocity or displacement.

tion so the bearing housing barely moves. An analyzer looking at displacement or velocity would not be able to detect spalling damage. However, such impacts can impart a substantial amount of force to a machine frame. Measuring acceleration of the bearing housing permits the calculation of impact force from Newton's Second Law ($F = ma$).

Next, consider a vibration reading from an unbalanced fan. The unbalance causes the bearing housing to sinusoidally rock left to right at a frequency equal to the fan turning speed. In practice, such waveforms contain many sine waves of different frequencies. It would be impossible to recognize a machine defect from this complex waveform. That is why the vibration analyzer performs a Fast Fourier Transform (FFT) on the waveform and converts the data to a frequency-versus-amplitude plot (spectrum). Most OEMs and machine rebuilders understand the importance of a good balance job, so unbalance is uncommon in new or rebuilt units. In service, however, fans commonly see blade erosion and dirt buildup, necessitating periodic cleaning and field balancing. Uneven buildup of dirt over several months would cause fan vibration amplitude to trend up, as expected.

Another source of machine vibration is the problem of misaligned shaft couplings. Consider the case of an otherwise balanced fan with an angularly misaligned coupling between the fan and motor. Rotating the shaft 180° pushes the fan and motor in opposite directions, out of phase. Conversely, two components simultaneously moving in the same direction would be in phase. A spectrum shows misaligned couplings as a vibration peak at twice the running speed, sometimes accompanied by peaks at one and 3× running speed.

Yet another source of vibration is external or internal looseness. Examples of external looseness include loose bearing-housing bolts or base bolts, or a weak or deteriorated machine base. A vibration spectrum indicates looseness as a peak at running speed and possibly several of its multiples. For instance, a machine running at 3,560 rpm would show a vibration peak in the spectrum at 3,560 cpm, and possibly 7,120 cpm (2 × 3,560), and 10,680 cpm (3 × 3,560). Internal looseness is loss of proper fit such as from a loose locking mechanism on a bearing ID, excessive clearance between a bearing housing and bearing OD, or a loose impeller on a shaft. The vibration spectrum of a machine with internal looseness also shows a peak at running speed along with many multiples.

All rotating machinery resonates at a particular speed or resonant frequency. A machine allowed to operate at its resonant frequency can sustain serious damage, even catastrophic failure. Designers of rotating machinery typically use finite-element analysis to predict resonant frequencies and avoid having them close to operating speeds.

But resonance problems can arise when equipment originally designed to run at a steady speed has been updated with a variable-frequency drive. For example, a pump with a resonant frequency of 900 cpm built to operate at 1,200 rpm may now run at speeds between 800 and 1,200 rpm, which is within its resonant frequency. Fixing the problem can be complicated, and may involve modifying the machine and support structure. Typically, the most economical fix is to program the drive so the machine quickly passes through its resonant speed. **MD**

MAKE CONTACT

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