A Practical Approach to Solving Machine Vibration Problems

By Victor Wowk, PE, Machine Dynamics, Inc.

The vibration analyst is first a strategist, then a mechanic. This defines the troubleshooting task into two journeys—the diagnostic journey and the remedial journey.[1] From this perspective, it is readily apparent that diagnosing the problem is only half the task. The vibration analyst without a "fixit" toolbox is like a bird with clipped wings. It may have the knowledge of flight, but it's not going anywhere with that knowledge. Likewise, the analyst often will have someone else actually fix the vibration, and they will usually diagnose the problem themselves anyway because they don't trust the original diagnosis.

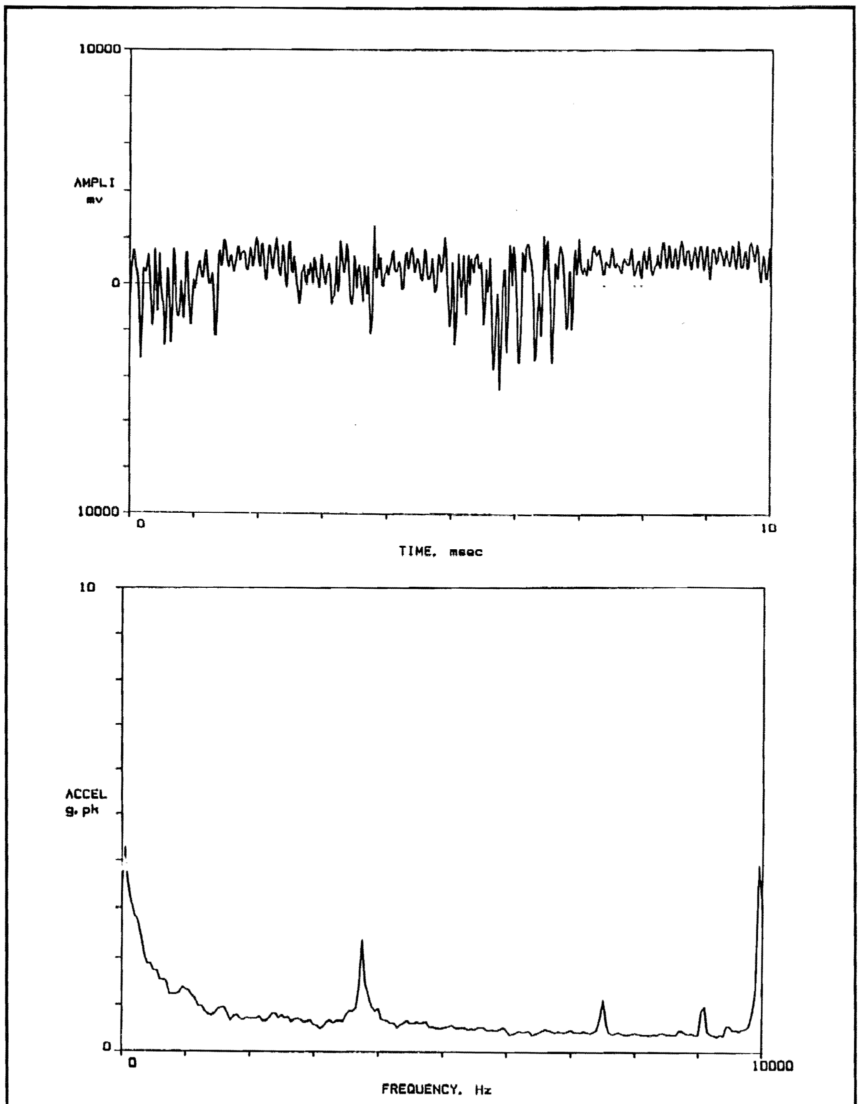

Figure 1 -
Loose connection to an accelerometer mounted on a gearbox. (a) Time domain view is non-symmetric about the zero axis. (b) Frequency domain view has high amplitude approaching zero frequency and elevated baseband.

Victor Wowk will be teaching the Level II Intermediate Vibration Analysis Course at the Predictive Maintenance Technology National Conference, December 4-6, 1995, in Indianapolis.

What good, then, is a spectrum of vibration? The answer is—It is only useful if something is done with that information. The acquisition of data and interpretation of that data is only half the diagnostic journey. The first part of this article will address general strategies for approaching a machine vibration problem. The second part will cover specific strategies for examining a machine in detail. The objective is to understand how it works under load and with thermal expansion.

Approaching a Vibration Problem

A big assumption must be made that the measurement system is not lying to us. By definition, a measurement system cannot lie because it is incapable of intentional deception, but it can present false information. It is up to the analyst to determine if the data is valid. Needless to say, every measurement system should be periodically calibrated. ANSI S2.11 recommends 6 month or 3 month periodic calibration for vibration measuring systems. This interval is O.K. for periodic health monitoring programs, but insufficient for acceptance testing.

For acceptance testing, I recommend calibrating the instruments shortly before, or shortly after the measurement. This could be a simple comparison check with another vibration instrument. This will add confidence to your data. The reason for this simple check is that acceptance testing is highly political. Whoever stands to lose a great deal of money will become very defensive. One typical defense posture is to attack the data, and if that is unsuccessful, to attack the interpretation of that data, i.e., the analyst. It continues to amaze me how some defenders have the audacity to challenge another's measurements when they come empty handed to the table with no contradictory measurements of their own. For those situations, I recommend cutting off discussions until the opposing side either brings their own measurements, or accepts yours. Challenge them to make an effort and play ball.

Vibration measurements deal with very low-level AC voltages—typically on the order of 10 to 100 millivolts AC. These small signals are easily corrupted. Some typical signal contaminants are: intermittent connections; additional unwanted voltages (like 60-Hz noise); setup errors, like DC coupling, filters off or wrong window.

Intermittent connections are easily identified in either the time domain or frequency domain. Figure 1 shows an intermittent on the signal line. In the time domain view, Fig. 1(a), the signal is not symmetric about the zero voltage line. The signal is very different above the zero voltage line versus below it, which is characteristic of an intermittent connection. Also notice the high-amplitude swings of more than 2 volts. Machine vibration is rarely this large. In the frequency domain view, there is high-level baseband

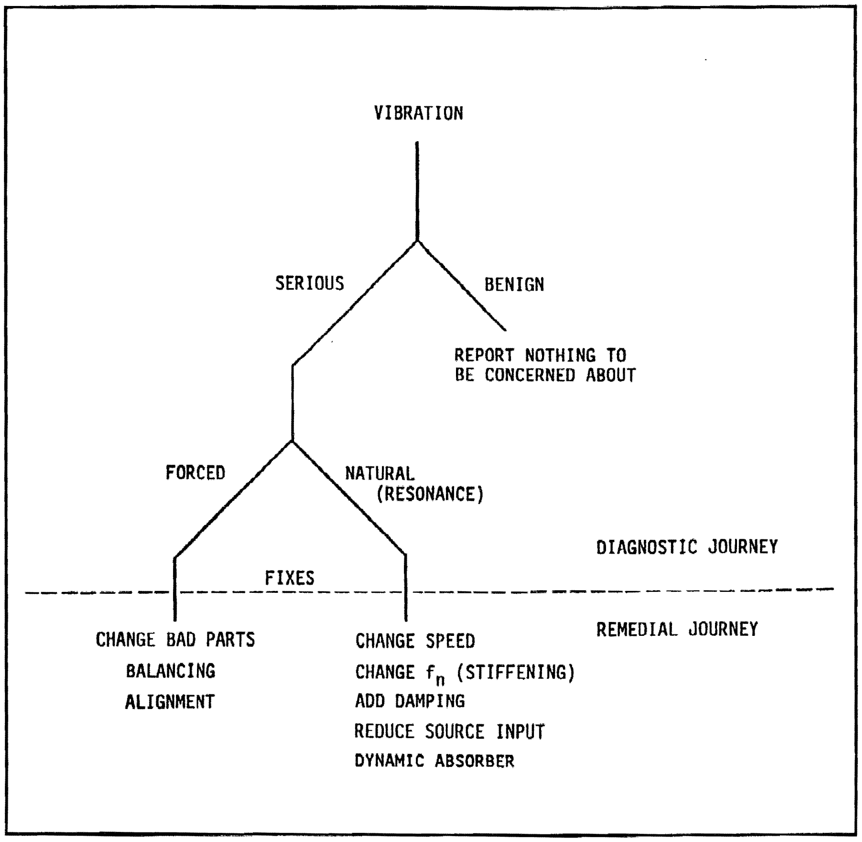

***Figure 2** -*
Flow chart for analysis.

Forced vibration problems will lead to fixes like changing bad parts, balancing, or alignment. The vibration measured at a point (i.e., a bearing) is a direct response to some defect. Correcting the defect will reduce the vibration at the measurement location. The vibration responds as expected and all is well with a happy ending. Forced vibration fixes are easier to deal with because we have technologies to handle them, and the result is predictable. Resonance problems paint a different, and uglier scene.

energy in addition to a steep rise at low frequency. Both of these are warning flags of signal corruption. In addition, 60Hz noise is the bane of every measurement system (or 50 Hz in some countries). A vibration analyst should always be suspicious of anything at 60 Hz.

A strategy to deal with possible signal contamination is to view the time signal. Vibration has a characteristic periodicity to it, with possible shock pulses. It is always symmetric about the zero voltage line. Another validity check is to note the overalls in different frequency spans. The overall energy in a larger span should always be larger than the overall in a smaller bandwidth, since it integrates a larger portion of the spectrum. It is possible that the vibration can change in the short time between measurements, but not likely. Mechanical systems have far more inertia than electrical measurement systems. Anything that changes within a fraction of a second, cannot be a mechanical change; it is probably a change in the measuring electronics.

Let us assume that the measuring system is not trying to deceive us, and that the data that is acquired is valid vibration. How do we deal with this data?

First of all, there are two kinds of machinery vibrations, benign and serious. Benign vibrations are characteristic of a machine's operation and will not cause any long term degradation or reduction in reliability. Examples of benign vibrations are blade passing, 120 Hz motor hums, high-frequency pure tones from motors, and broadband turbulence in some cases. The task for the analyst is to quickly identify a vibration as benign or serious, and then proceed accordingly.

A very strong indicator of serious vibration is almost anything significant (greater than 1.0 mils) at rotational speed. Other supporting information for serious vibrations are component failures, and cracks.

As soon as it is decided that the vibration is serious, and action will be taken, the analyst is at the second fork in the road. One fork leads down a forced vibration path. The other fork leads down a resonance path (Figure 2). The fixes for the two paths are very different. This is why it is important to identify the nature of the problem quickly.

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Resonance is a mechanical amplifier (Figure 3). It is non-linear, and behaves in an unpredictable manner. That is, a change in weight, or a change in alignment, does not produce a proportional change in response at the measurement point. This unexpected result is, sometimes, the first indication that resonance is a player. Other indications of resonance are:

•The vibration is much worse than typical. It is grossly bad.

•Vibration is directional, i.e., the amplitude is much higher in one direction compared to other directions.

•Speed sensitivity, i.e., it is worse at one speed

•An amplitude that varies. i.e., the needle swings back and forth, or the peak in a real time spectral display goes up and down

•Rumbles during run-up and coast-down

The verification of a resonance is an impact test to measure the natural frequencies when the machine is stopped.

Resonance problems lead to fixes very different than forced vibration problems. There are five known fixes for resonances:

- Change speed
- Change the natural frequency of the responding part
- Add damping
- Find the source vibration and reduce it. This may require unusually fine balancing or alignment. It may also require reducing what gets to the responding part by introducing some isolation in the path.
- Use a dynamic absorber

These fixes are different than the usual balancing, alignment, and changing parts for forced vibrations. Hence, it is important for the analyst to find the correct fork in the diagnostic journey quickly. The usual method of analysis is:

- If the machine is running, then do a typical vibration amplitude survey of 1X rpm at all bearing positions in the three orthogonal directions. Look for any other dominant amplitudes at other frequencies.
- Shut down the machine and observe the

amplitude during coast-down at the worst location.

• Do a few impact tests to measure the natural frequencies of the major components, i.e., driver and driven machines.

After acquiring this initial data, a picture begins to emerge of forced versus natural response. When resonance becomes evident, then the analyst should start thinking about clearing his schedule for the near future because this will not be a simple 4-hour balance job. The only easy fix for resonance is to change the speed, if that is an option. All of the other fixes are more difficult.

What's so special about resonance? First, it is a significant source of income for every independent vibration engineer. Without resonance, most of us would starve or be in a different line of work. The majority of income in my business has come from identifying and fixing resonance problems. The bottom line is that understanding resonance can be a moneymaker for anyone.

The second aspect of resonance is that it is not a simple concept to grasp. It is a structural effect. The component, or the structural path to the component, modifies the source vibration. It takes the small dynamic input force and amplifies it in a non-linear fashion. It doesn't just double the input, but can multiply it as much as 100 times depending on frequency (Figure 4). It does some other funny things also, like changing the direction and phase shifting. These do strange things to balance and alignment efforts. Notice in Figure 3 that just approaching the resonant frequency can cause significant amplitude and phase deviations.

The third aspect of resonance is that it is almost always destructive. It causes cracking and that is a dangerous condition. It doesn't just beat up bearings. It can cause a machine to come apart. As a result, I strongly encourage a safety strategy for the analyst: Use the longest cables you have and setup your instruments and yourself as far away as possible. Stay out of the machine's "line of fire"—the plane that a part will fly away along. (I have heard of some analysts who prefer to take the initial measurements from inside of their pickup trucks.) Resonance is definitely a serious vibration when it is active...it is never benign.

The fourth aspect of resonance is that it is a design fault. The fault is in the system and not necessarily the machine itself. The system is defined as the machine, its base, the foundation, and any attachments like piping or ducts. The design engineers do not usually have the luxury to run and test every possible system configuration. Even though resonance may be a system design fault, it is usually first detected on site during start-up testing. The default industry practice is to fix resonance problems on-site as the customer, or owner, observes. Every plant engineer feels that he/she deserves a smooth running machine for the money paid. Unfortunately, the machine represents only half of the equation. The structure is the other half (and dominant factor I might add) for resonance problems. It is possible to guess at the structural effect, but even the best guessers are only 80 percent right. The present state of the art is to design systems based on past favorable results, make only minor modifications, and hope for no resonances. If they do arise during start-up testing, the most practical and economical strategy is to call out an expert analyst, line up some welders, and fix it in place with some strategic stiffeners. The design is always finished when put into use.

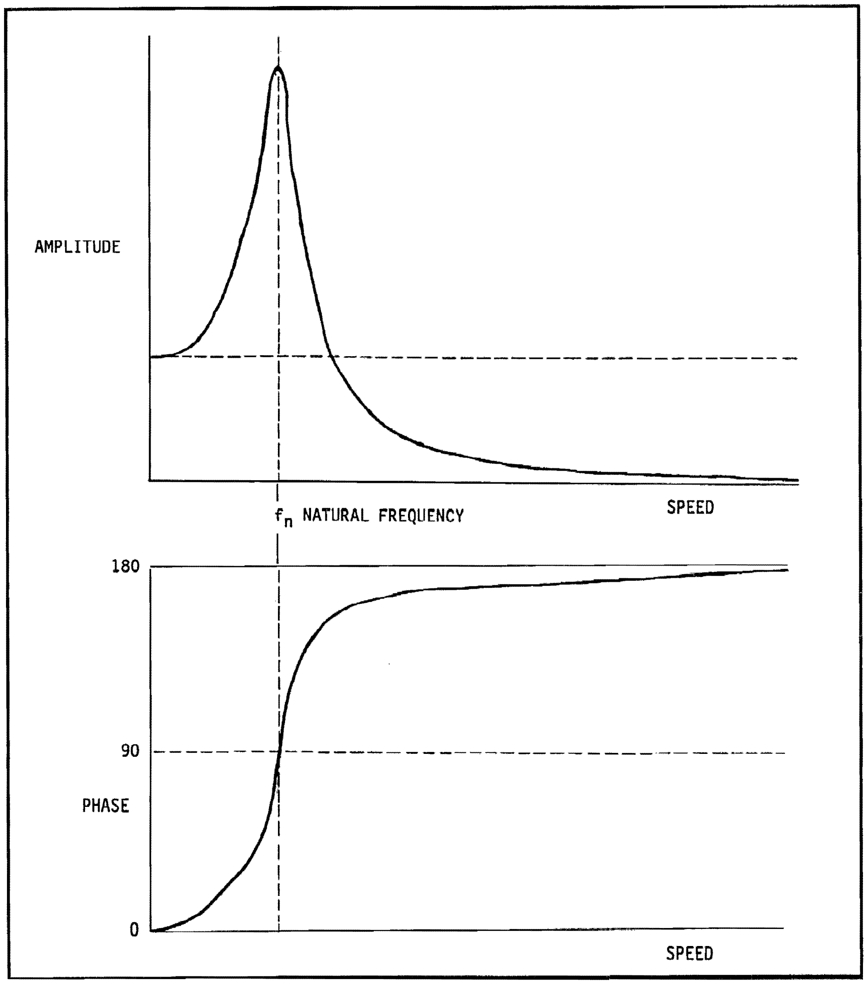
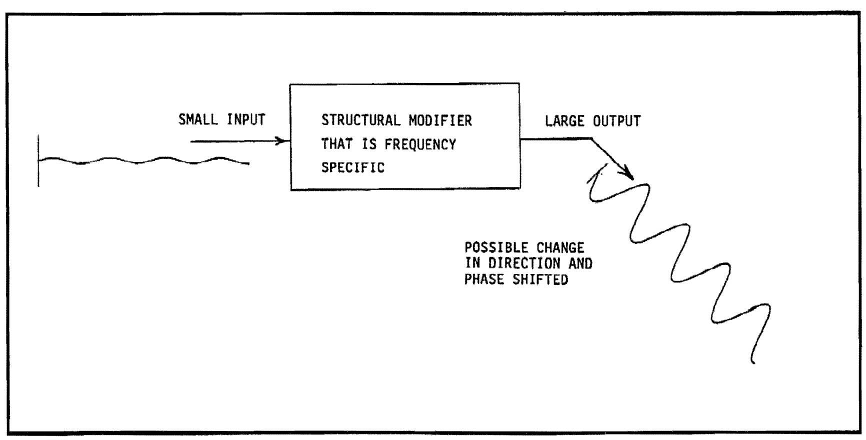


Figure 3 (Above) -
Resonance, the amplitude and phase response are non-linear.
Figure 4 (Below) -
Resonance is a structural modifier.



A system with no physical changes can also experience some dynamic changes over time. There is convincing evidence that all mechanical systems soften over time. This is due to: joints loosening; creep, or stress relaxation; microcracking; or other changes at the molecular, or granular level, caused by the long-term oscillation. Characterizing a system's natural frequencies when new, and trending afterwards, is a useful way to track these changes.

The good news is that every vibration problem has a technical solution. There may even be several good technical solutions, and the analyst is queried by the managers for the most cost effective fix. The only vibration problems that I have not been able to fix were the unfunded ones. That is, money was unavailable to execute the technical solution. That is a practical aspect of every vibration problem, i.e., is the fix affordable?

The fifth aspect of resonance is that it can change over time. Whenever the mass or stiffness of a system changes, then its natural frequencies will also change. Some examples of modifications that affect the natural frequencies are: cutting keyways in a shaft (softer stiffness shaft); modifying a shaft, i.e., machining new diameters (again softer stiffness shaft); adding more weight to a floor panel by bringing in more equipment. More dancers on a floor will change its dynamic behavior. (Lowers natural frequency); changing bearing types (affects mode shape); or using a different stiffness coupling (this affects the torsional resonant modes).

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A word about instruments. The particular brand or type of instrument is the often the least significant factor in the diagnosis. Most vibration measuring instruments today function and display results similarly, so that a competent analyst can use almost any instrument and arrive at the same diagnosis. Some older tuneable filter instruments can also be used very effectively for both analysis and balancing. It is important to note, however, that the FFT spectrum analyzer is useful in the hands of a competent user, but worse than nothing in the hands of a novice who does not have a good understanding of how to use it. I have seen more than one machine ordered to be rebuilt when nothing was wrong with it. The fault was not in the machine but in the measurement. Specifically, the calibration constant for the transducer was off by a factor of ten, making the current measurement appear ten times higher than the previous measurement. Other errors with filters, windows, and averaging can cause similar measurement deviations that can lead to costly judgements. An effective strategy to protect oneself from these blunders is to have someone else examine it with their instruments.

Forces

Every vibration has an oscillating force as its source. The vibration is just the symptom of this force. It is the forces inside of machines that we really want to understand, since they lead to the bad part or to the right corrective action. The exterior oscillating motion does no harm, it only creates waves in the air. However, some significant forces inside are causing this motion, and causing internal damage.

Humans are always looking for an easy solution, then take the rest of the day off. Unfortunately, I see no other way than to understanding the internal forces, and this comes only after understanding how a machine operates. The most valuable vibration analyst is a prior mechanic who has the benefit of visual references to the internal workings of machines. While different classes of machines generate different cyclical force patterns, they all have some things in common. One common component is bearings. These are the force couplers between the moving parts and the stationary world, and the components most prone to wear and failure.

The purpose of bearings is to transfer forces between the moving and stationary parts, and at the same time to avoid contact. The objective is to reduce friction and abrasive, or rubbing wear. On machines with oil or grease lubricated bearings, this means no metal-to-metal contact. The question then becomes, how do we detect metal-to-metal contact? The answer is shock pulses.

Whenever two objects make contact, a stress wave originates at the points of contact and travels through both objects at the speed of sound supported by those materials. This is a shock pulse. It generally has a frequency from 1,000 to 20,000 Hz, depending on the material density and hardness. The speed of sound in steel at normal room temperature is 16,600 feet per second. With a normal machine component of 2 feet in length, this will lead to a supported standing wave in the part of 8,300 Hz. Other physical phenomena lead to shock-pulse energy in the same frequency band of 1 to 20 KHz. Typical machine components, like bearing parts, ring at those natural frequencies. The time duration of the actual metal-to-metal contact creates a stress wave with frequencies in the 1 to 20 KHz band, sometimes also higher.

This shock pulse is easily detected by an accelerometer on the outside of the bearing, or any other location that has a good structural path to one of the bearing surfaces. The pulse is non-directional. It bounces around inside the material, and can be detected anywhere on the surface, in any direction. Some of this shock-pulse energy couples to the air and becomes noise.

Plain bearings should be supported on an oil film and there should be no shock pulses during steady-state operation. There could be a dry friction rub during start-up before the shaft rides up on the oil wedge. Figure 5 shows the time-domain view of a well-operating plain bearing. There are no shock pulses, as it should be. Figure 6 shows a time-domain view of a plain bearing that has a rub. Notice the high-frequency shock pulses. This bearing operated for about 2 weeks after this data was acquired.

The time-domain view for shock pulses should be normalized for the speed. Higher-speed machines will naturally generate more viscous friction in the lubricant, and more turbulence in the vicinity. The two different machines in Figure 5 (a generator) and Figure 6 (1,000-hp mill fan) were both operating at 1,800 rpm.

The other common type of bearing is the rolling element bearing—commonly called ball bearings. The rolling elements could be balls, cylindrical rollers, tapered rollers, or spherical rollers, with appropriate matching races. Ideally, the rollers are supported on a grease or oil film, and there should be no metal-to-metal contact. Practically, perfect geometry in rolling elements in not achievable; and, as a result, there are imperfections in the rollers and the races, and size variances. Also, the rollers sometimes skid during changing thrust conditions, and the edges of the rollers contact the races. Finally, the cage, or retainer that separates the rollers, is the least perfect component, and the rollers make contact with it. The bottom line is that all rolling element bearings have metal-to-metal contact. The better bearings will generate less shock pulses. Poor quality, or damaged bearings, will generate more shock pulses. Higher speeds will always cause more shock pulses in any rolling-element bearing.

Shock pulses, in a rolling-element bearing, indicate one of three things: lack of lubrication; a damaged, or poor quality, bearing; or excessive dynamic stresses in the machine.

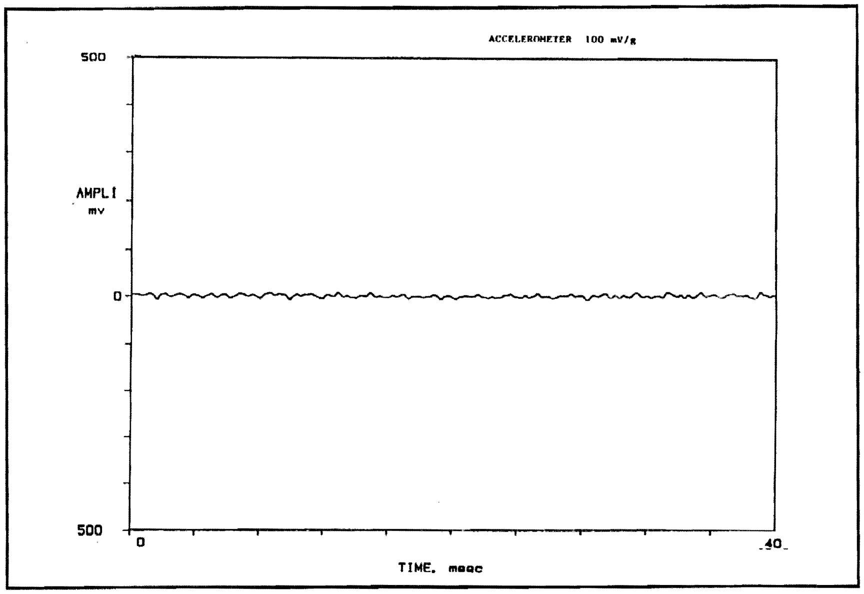
The first corrective action to take is always to verify good lubrication. Inject some grease and see if the shock pulses go away. If they do, then it was just a matter of poor lubrication, and it's time to discuss the lubrication schedule with the mechanics.

If the shock pulses initially go away, but return in a few minutes, then further greasing won't help. If the bearing is known to be good, that is, it was just installed, then it is time to look for excessive dynamic stresses. These can be caused by misalignment, gross unbalance, or excessive loads. If the alignment and balance are known to be good, then it is time to question the design. Maybe this is not the right bearing for the application.

Shock pulses are an immediate warning flag of impending trouble and therefore the best indicator to diagnose bearing condition. Every other indicator; bearing frequencies, enveloping, temperature, oil analysis, are not as reliable as shock pulses. The only complication I have seen is that pump cavitation can also generate shock pulses that look identical to metal-to-metal contact. The best approach is to ignore shock pulses from a pump that is known to be cavitating.

The recommended strategy is to examine all rolling element bearings in a consistent manner. The virtue of this approach is that someday you may be called upon to judge a bearing that you have no previous history on. The consistent manner of examination is to use an accelerometer, and set up a time display window of 40 milliseconds and plus and minus 500 millivolts. This corresponds to a frequency span of 5 KHz. If the accelerometer has a sensitivity of 100 mv/g, then 500 millivolts full scale corresponds to plus and minus 5 g's. Figure 7 shows good and bad ball bearings using this view.

The speed of the shaft needs to be considered when judging bearings in this manner. All bearings, even the very slowest ones turning at 1 revolution in 9 minutes (1/9 rpm), will generate a "clunk" when a bad portion of the bearing comes into the load zone. However, high-speed bearings will generate almost continuous shock pulses, and this is how they normally run. The value of this analysis strategy is for comparison of bearings at the same speed. With enough experience, the analyst can immediately judge even an unknown bearing, in just a few seconds of observing the time display.



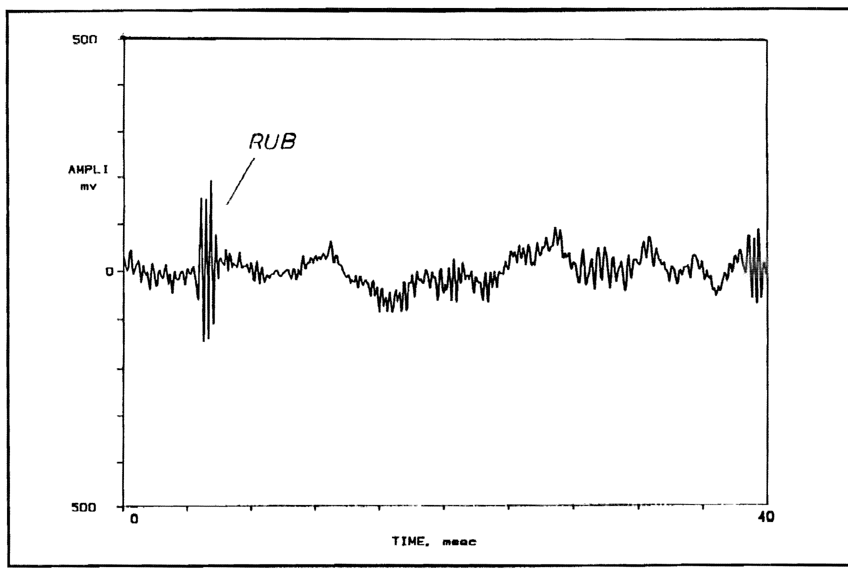


Figure 5 (Top) -
Time signature of a well-operating plain bearing with no rubs.

Figure 6 (Bottom) -
A plain thrust bearing rubbing.

From this perspective of bearing analysis, the best analysis instrument is a digital storage oscilloscope with an accelerometer. For a quantitative measure, a gated counter with an adjustable threshold level works very well. Who needs an FFT analyzer?

Bearing misalignment has been underestimated as an installation fault. Plain bearings will simply rub themselves a clearance if misaligned. It is hoped that the clearance opens up sufficiently before the temperature rises to a high enough level to soften the babbitt material and push it around. Friction and temperature rise are the bad effects that will destroy a plain bearing. The friction is detectable as a shock-pulse rub.

All rolling element bearings have some internal clearance engineered into the design. The purpose for

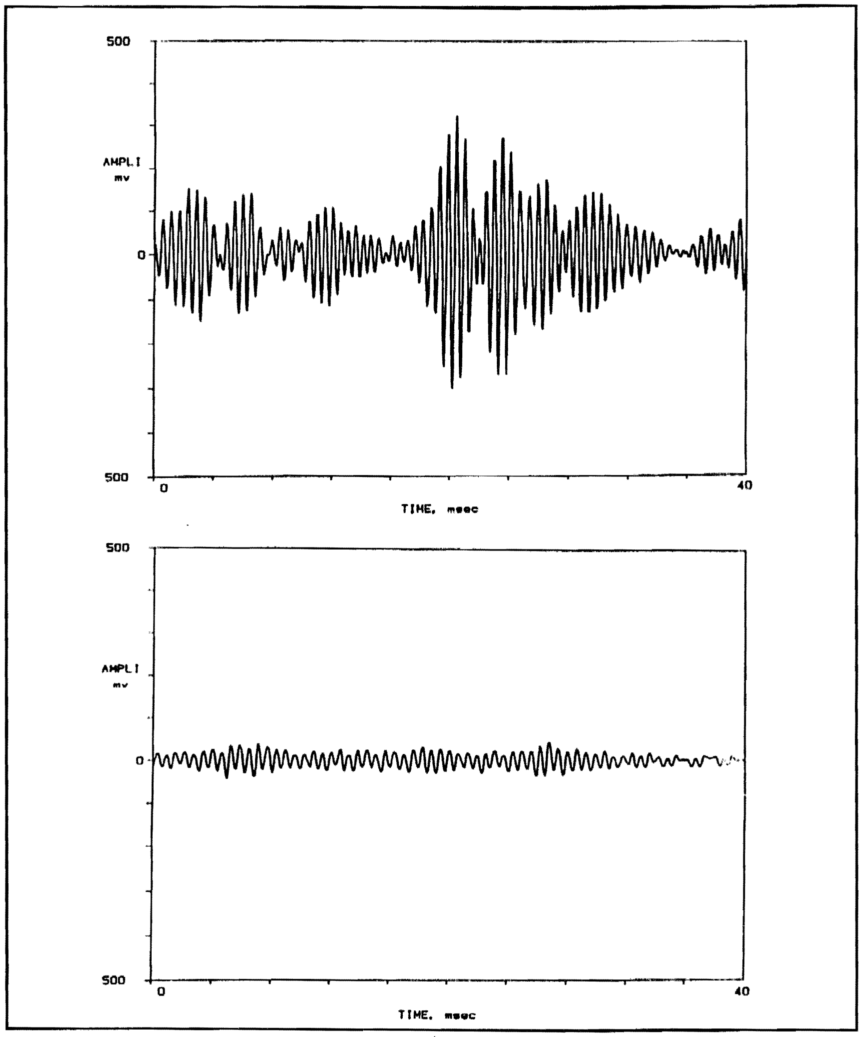
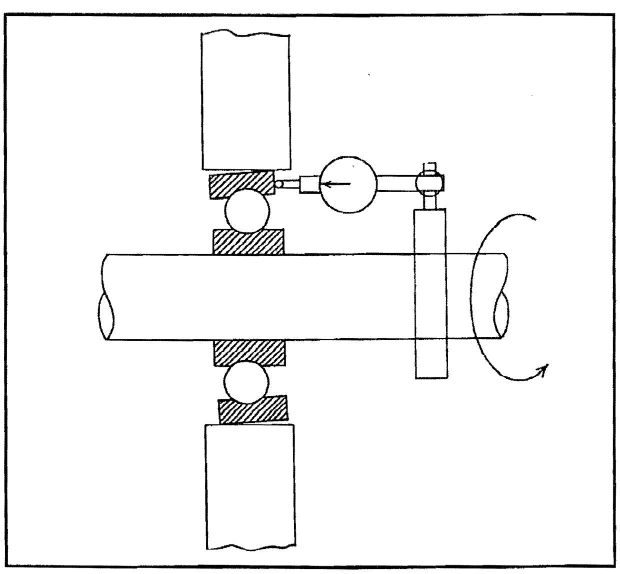


Figure 7 (Above) -
Vertical acceleration from two identical motors. (a) Bad bearing with shock pulses. (b) Good bearing.
Figure 8 (Below) -
Bearing misalignment and measuring perpendicularity of the outer race to the shaft rotating axis.



this internal clearance is to accommodate some expansion due to press fits and thermal expansion. This internal clearance was not intended to accommodate misalignment. The misalignment considered here is a non-perpendicularity between the shaft axis and the outer race. Ideally, the two should be perfectly perpendicular and all of the internal clearance is available for expansion, as intended. When some non-perpendicularity exists, then the bearing has less clearance available for expansion and it is forced to operate in a narrower temperature band before problems occur.

If there is no internal clearance due to misalignment during installation, then bearing problems are immediate. They are noisy, generating both bearing frequencies and shock pulses. Failure comes quickly—hours to days, to weeks, depending on the severity of the high dynamic stresses. The best solution is to measure for bearing alignment with a simple fixture as shown in Figure 8. This should be done for all new bearing installations with the exception of true self-aligning bearings. The true self-aligning bearings are the spherical roller type for radial bearings, and the tilting-pad type for thrust bearings. All other bearings require a bearing alignment check. Allowable tolerances are:[2]

0.001"/inch	Commercial applications
0.0005"/inch	High-speed applications
0.0002"/inch	Precision bearings

Shaft-to-shaft misalignment creates problems and symptoms identical to bearing misalignment. The best diagnostic procedure is a process of elimination. Once proper shaft alignment has been verified, if misalignment symptoms are still present, then it is time to suspect bearing misalignment. Bearings can be aligned in place with a hammer, a punch, and a measuring system.

Machines

The most common machines are electric motors. Fans are machines that consume the most analysis and balancing time. Reciprocating machines are the most difficult to analyze. Turbomachinery is the most expensive to repair when it fails. Different machine classes present different types of analysis problems. It helps to understand how they work, but there is some common ground to begin analysis from. The common vibrations are: rotating speed, 1X rpm, or harmonics; shock pulses at bearings, and other contact points; and resonances. Generally, any of the common vibrations are serious.

In addition, machine-specific vibration patterns are: blade-passing frequencies, gear-mesh frequencies, 120-Hz motor hums, broadband turbulence, belt frequencies, noise, and high-frequency tones from electrical machines.

Generally, the machine-specific vibration patterns are benign when not accompanied by any of the common vibrations. This is an over generalization, and if this was all we needed to know, then there would be no need to delve into the inner workings of machines. Unfortunately, the dynamic world of machines is not always a simple cause and effect relationship with repetitious forces and linear responses. There are compounding, and confounding, effects of temperature and load. This is where the knowledge of internal forces within a machine can clarify a complicated situation.

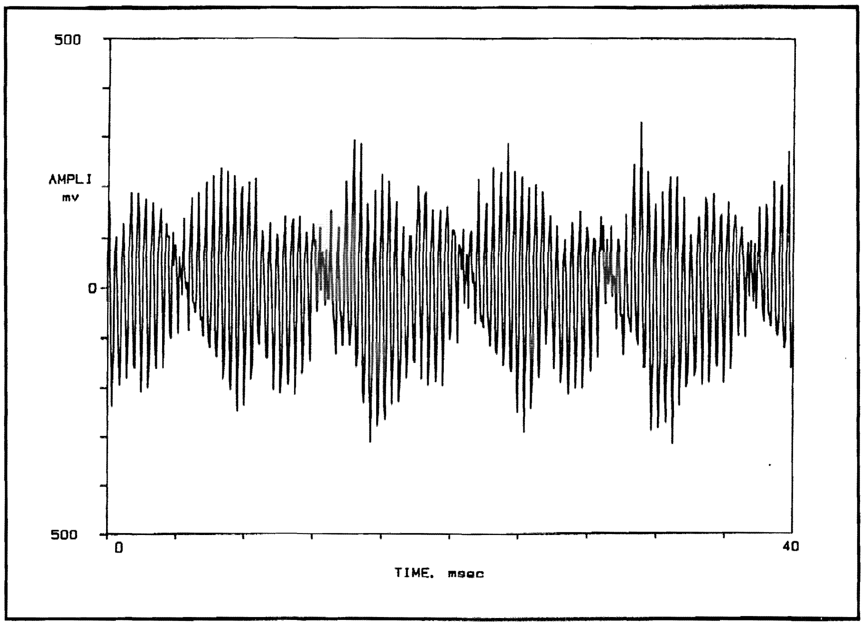
The machine-specific vibration patterns are load related. They are usually benign and are an indicator of normal machine operation, with some exceptions. Gear-mesh vibrations are always present whenever gears mesh. The amplitude of gear mesh, its harmonics, and its sidebands is an indicator of how well or poorly they mesh, and an indicator of wear. The 120-Hz motor hum is normal for all electrical machines and is a measure of the quality of construction. It can, however, also be an indicator of single phasing and a possible loose connection. This is a serious condition because it precedes an electrical fire. High frequency (700 to 5,000 Hz) pure tones from electric motors is normal. They are rotor bar/stator slot-passing frequencies and are not related to life. High-frequency shock pulses, however, are bad news. They indicate metal-to-metal contact. The best way to differentiate the two is viewing the waveform in the time domain. The rotor bar/stator slot phenomena is a repetitious sine wave that varies in amplitude (Figure 9). It looks very similar to a beat, and the wave pattern repeats at regular intervals.

Load changes will cause changes in the amplitudes of the machine-specific vibration patterns. Small amplitude changes at these frequencies have nothing to do with defects or impending failure. They are simply reflecting the changes in internal forces with load. Even large amplitude changes can be load related. How do we react to these? I suggest do nothing unless the change in amplitude can be demonstrated to be non-load related.

Thermal expansion has been underestimated as a cause of machine breakdown. Nothing can constrain metal from expanding when the temperature rises. When the thermal expansion reaches physical limitations, then distortion occurs. Vibration, in the form of rubs, or 1X-rpm unbalance, begins dramatically. This is where a knowledge of internal machine construction is useful. A visual reference of the bearings, seals, thrust limits, and how the machine was assembled, is most useful. There is one telltale indicator of thermal expansion problems: the vibration gradually changes over time as the machine warms up. This is a clear indicator of a temperature effect. Any vibration symptom that begins immediately when the machine reaches full speed is a mechanical rotating effect, not a gradual temperature effect. However, the delta change from start-up to plus 60 minutes of running time, is a temperature effect (assuming the load is constant).

The failures on machines today are of three types—manufacturing variability, excessive stresses, and design faults. Vibration analysis has the capability to detect all three of these undesirable conditions, and the vibration signature provides a footprint to begin the hunt. I wish you good hunting, and remember that some go down easy, some go down hard, and some take more than one shot.

For more information on vibration measurement and analysis contact Victor Wowk, Machine Dynamics, Inc., 10117 Trevino Loop NW, Albuquerque, NM 87114; (505) 898-2094.



References -
1 J. M. Juran, Quality Control Handbook, McGraw-Hill.
2 Source Machinery's Handbook, twenty-first edition, Industrial Press, page 638.

Figure 9 -
High-frequency motor vibrations. These are normal, especially on large motors.