

# FALSE AND MISLEADING SOURCES OF VIBRATION

by

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## ABSTRACT

Many false sources of "vibration" in rotating machinery of all types are described and discussed. The focus is on the practical distinction between actual vibration and spurious indications of vibration caused by problems external to the equipment under study.

## INTRODUCTION

It is said that the great majority of vibration problems in rotating machinery are the results of a very few causes, chiefly unbalance, misalignment, and resonance. While this statement is unquestionably true, and most knowledgeable investigators concentrate on these areas, it overlooks one very important consideration, which is that all too often what is thought to be vibration actually is not.

A great deal has been written about vibration in turbomachinery and what it is. This presentation has a different approach.

The experienced vibration analyst knows there are many pitfalls along the road to solving vibration problems. And one large class of pitfalls consists of faults that are not "vibrations" at all, but false or inaccurate indications, or the result of some external influence.

A good principle to bear in mind is that "vibration" is not, in and of itself, a problem. Rather it is a *symptom*. That is, it is an *indicator* of a problem which has a cause and, therefore, a solution. A machine's response to a dynamic force is measured, not the force itself.

Another principle to remember is that vibration invariably is measured indirectly and is, therefore, subject to multiple sources of inaccuracy.

The experienced vibration analyst will approach every situation without any preconceptions that the vibration indications are either accurate or erroneous. Rather, the analyst should maintain a healthy skepticism as to the nature of the problem until it is well understood. Failure to do so often results in premature attempts to find "solutions" inappropriate to the real cause. In other words, be sure you know what you are dealing with before developing a solution.

## INSTRUMENTATION CONSIDERATIONS

The accuracy, frequency response, and dynamic range of modern vibration analysis instrumentation are remarkable and rarely present any serious limitation to the analyst. However, there are a number of ways in which an individual can "go wrong."

The principal types of vibration instruments used in an industrial environment are displacement probes (also called eddy current or proximity probes), which sense shaft position with respect to the housing, and the seismic (inertially referenced) devices, i.e., velocity pickups and accelerometers, which sense housing vibration (Figures 1, 2, and 3). Occasionally, vibration transducers are hand held, but for turbomachinery applications, permanent mounting is strongly recommended.

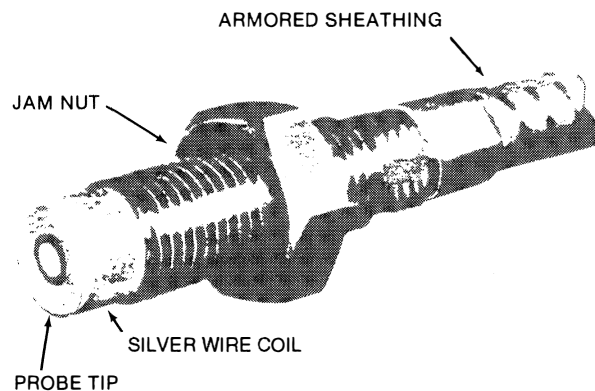


Figure 1. Typical Proximity Probe.

It is imperative to use the right type of instrument for the job. It should be common knowledge, but it bears emphasizing that one should choose with deliberation from among the available pickup types. Using an accelerometer to look for low frequency vibrations or using a displacement probe in search of high frequencies, such as gear mesh frequencies, will result in failure. Most authorities agree that the useful frequency limits for the various vibration pickups are shown in Table 1.

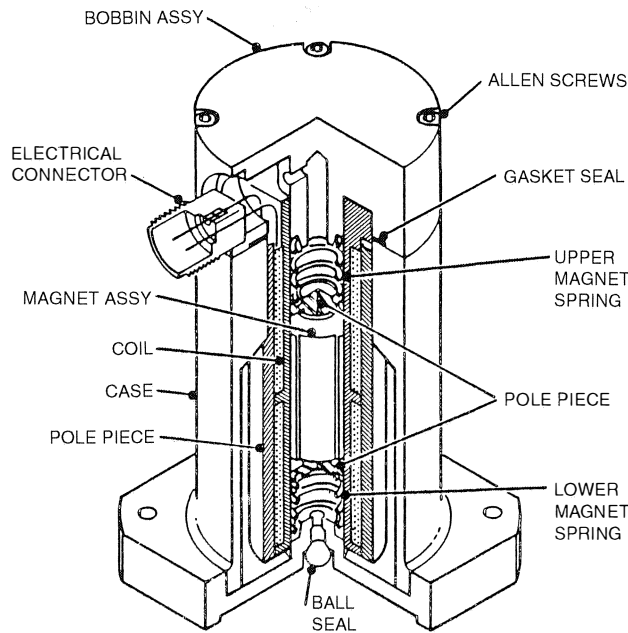


Figure 2. Typical Velocity Transducer.

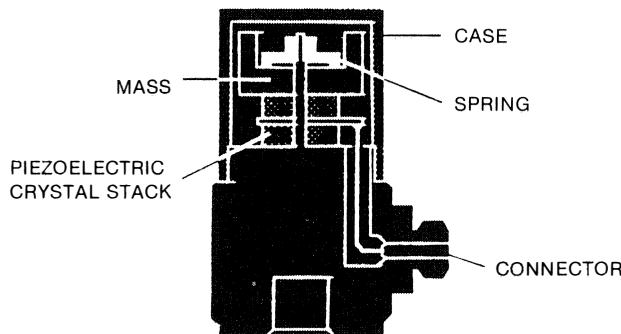


Figure 3. Typical Accelerometer.

Table 1. Transducer Recommended Frequency Ranges.

Displacement	0 to 1000 Hz
Velocity	15 to 2000 Hz
Accelerometer	50 to 6000 Hz (commercial grade)
	2 to 50,000 Hz (laboratory standard)

All types of vibration pickups are sensitive to proper mounting. They must not have any mounting bracket resonances within the frequency range in which they will be used.

Mechanically, an accelerometer is a lightly damped, high stiffness mass-spring system and has a resonant frequency at the high end of its useful frequency range. Its output is highly amplified at and approaching this resonance and so the accelerometer should not be used in this range. Since the output signal from a typical piezoelectric accelerometer begins to rise well

before the resonant frequency, the usual recommendation is to use 20 percent of resonance as an upper limit for a five percent deviation (Figure 4). This is accomplished by the use of a low pass cutoff filter. The deviations due to approaching accelerometer resonance are more than might be expected (Table 2). Although accelerometers are usually linear down to a frequency of a few Hertz, the output signal is too low to be useful at such levels. They are best for frequencies above 2000 Hz.

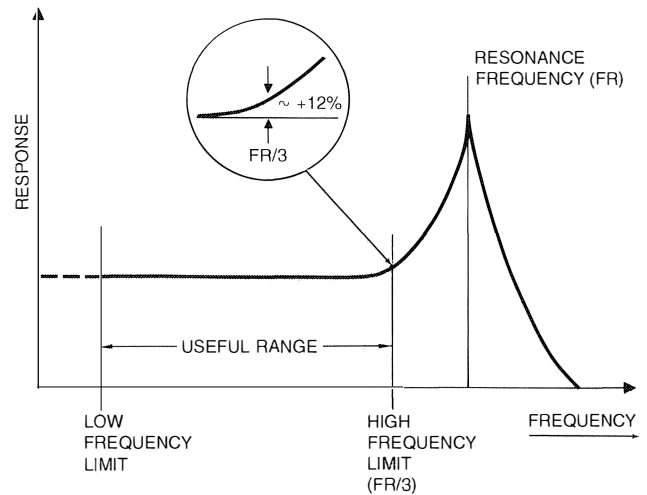


Figure 4. Frequency Response for an Accelerometer.

Table 2. Accelerometer Deviation Related to Resonant Frequency.

Let $f_n$ = accelerometer resonant frequency (hard mounted). Then:	
% $f_n$	% Deviation between measured and actual level
22	5
30	10
50	50% (+3dB)

The velocity pickup is also a mass spring system, but one which is well-damped and operates above its resonant or natural frequency. A case mounted coil moves in response to the external vibration. Inside the coil is a small permanent magnet suspended on soft springs. A voltage is induced in the coil proportional to the velocity of vibration. Resonance typically occurs at about 10 Hz and a high pass filter limits the low frequency response to 15 Hz. Contrast this with the accelerometer, the useful range of which extends only to a fraction of its natural frequency. Output of the velocity pickup remains quite linear up to about 2000 Hz, and rolls off gently thereafter (Figure 5). Alone among the types of vibration transducers, the velocity pickup can be hand held if there is no meaningful information above about 500 Hz.

Both types of seismic devices are usually mounted directly on the machine housing without interposing brackets. Nonetheless, they are also very sensitive to proper mounting. The chief pitfalls are poor surface contact between pickup and housing (mill an accurate spotface for mounting), and failing to correctly

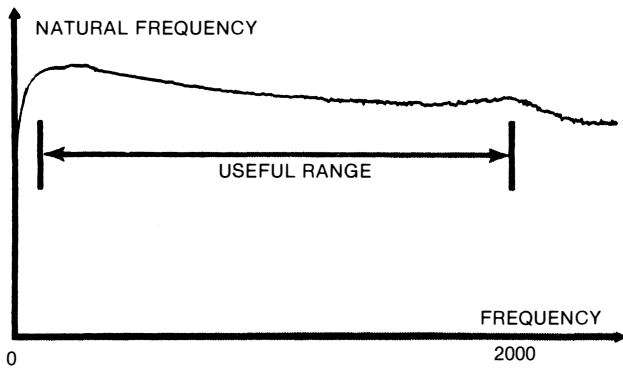


Figure 5. Frequency Response for a Velocity Transducer.

torque the accelerometer (use a torque wrench). A typical mounting detail is shown in Figure 6.

Where possible, accelerometers and velocity pickups should be located right at the bearing caps. The next best location is on a rigid part of the housing close to the bearing. The worst location is any place removed from the bearing area or on a thin or unsupported surface of the housing which may have its own resonance. Never mount on sheet metal.

It is very important to select a good location for the seismic transducer. By "good" is meant a place that is free of resonances or other amplifications, but provides a strong signal output. It is worth spending extra time trying different locations to establish the best one. This is especially true in the case of a series of identical units. Select a mounting position as close as possible to a bearing for best results. Sometimes a small shift in the pickup position will cause a significant change in the output signal. A good way to establish the preferred position is to mount several pickups with dental cement. This cement is very rigid and, while too brittle for a permanent mount, enables one to try many locations quickly. Use only catalytic or thermosetting cements. Do not attempt to use solvent drying or nonhardening adhesives. Once having established the best location, a permanent mounting spotface surface with tapped hole or stud should be provided. It is essential that the details and location of the mount be precisely duplicated on all units of a given design. Failure to do

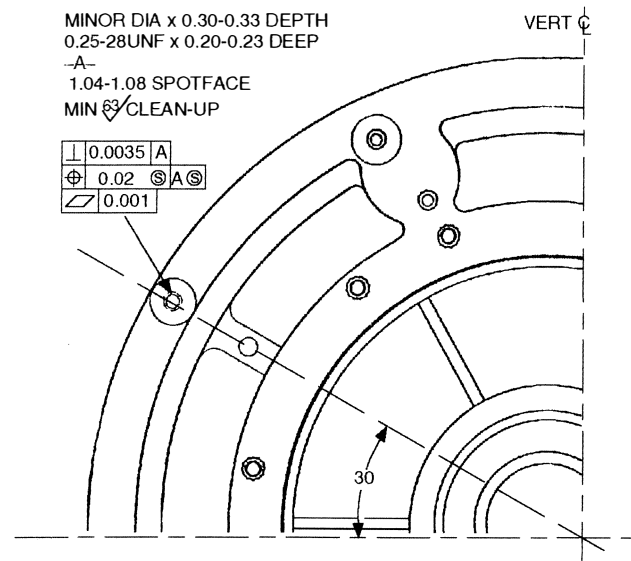


Figure 6. Accelerometer Mounting Detail.

so will prevent meaningful comparisons. Hand held or magnetically attached probes should be avoided for all but preliminary survey work. The popular accelerometer mounting techniques are shown in Figure 7 and the corresponding frequency response curves for a single accelerometer mounted by each method are shown in Figures 8, 9, 10, 11, and 12. These clearly illustrate the need for using the hard mount for any serious work.

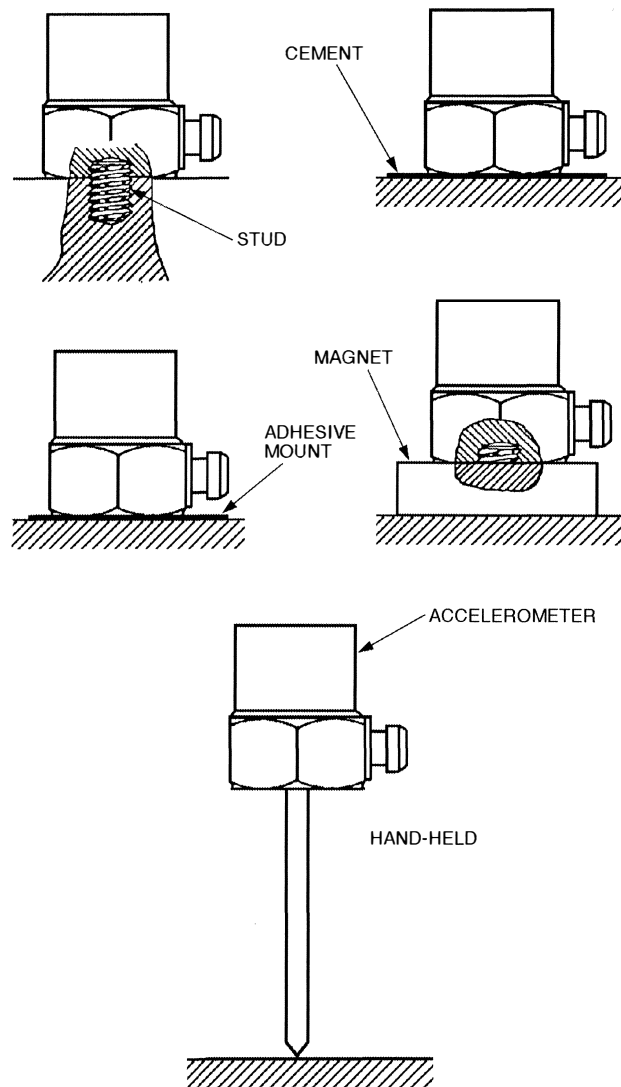


Figure 7. Accelerometer Mounting Methods.

CALIBRATION

All types of pickups can be checked for calibration using the proper equipment and techniques. Calibrators are now available that can handle all three types of devices (Figure 13). This calibrator can mechanically produce an accurate displacement (mils or microns), velocity (inches per second), or acceleration (g) level over a broad frequency range. Remember to duplicate the actual installation as nearly as possible. Some points worth mentioning regarding calibration are noted below:

Displacement Probe

Be sure the target material is the same as the shaft and is free of electromechanical disturbances (see section on Runout) be-

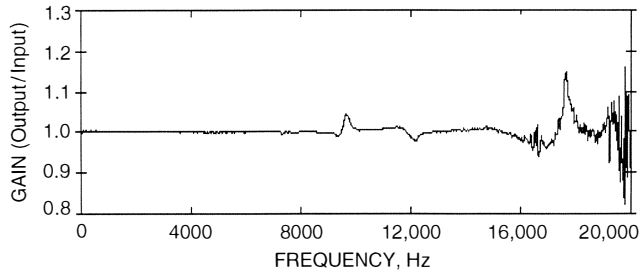


Figure 8. Accelerometer Response—Hard-Mounted.

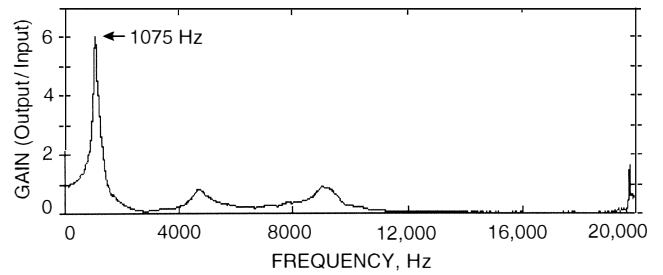


Figure 12. Accelerometer Response—Hand Held.

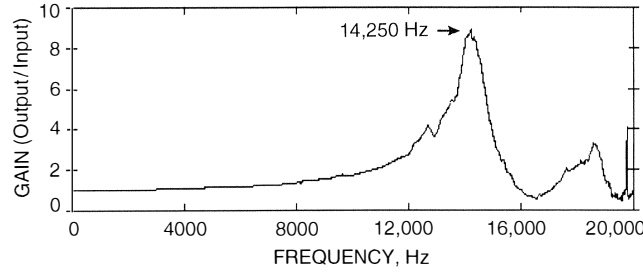


Figure 9. Accelerometer Response—Cement Mounted.

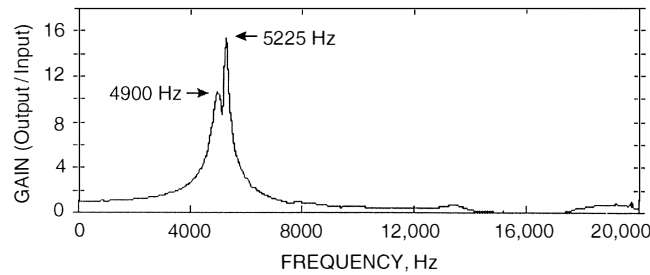


Figure 10. Accelerometer Response—Adhesive Mounted.

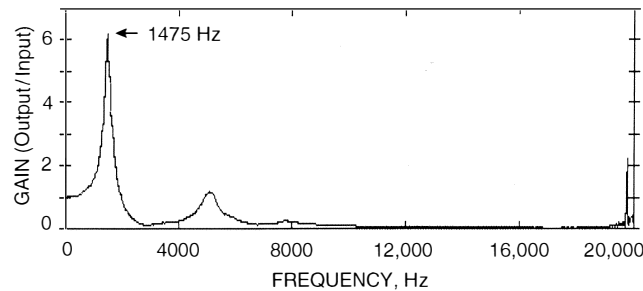


Figure 11. Accelerometer Response—Magnetic Mounted.

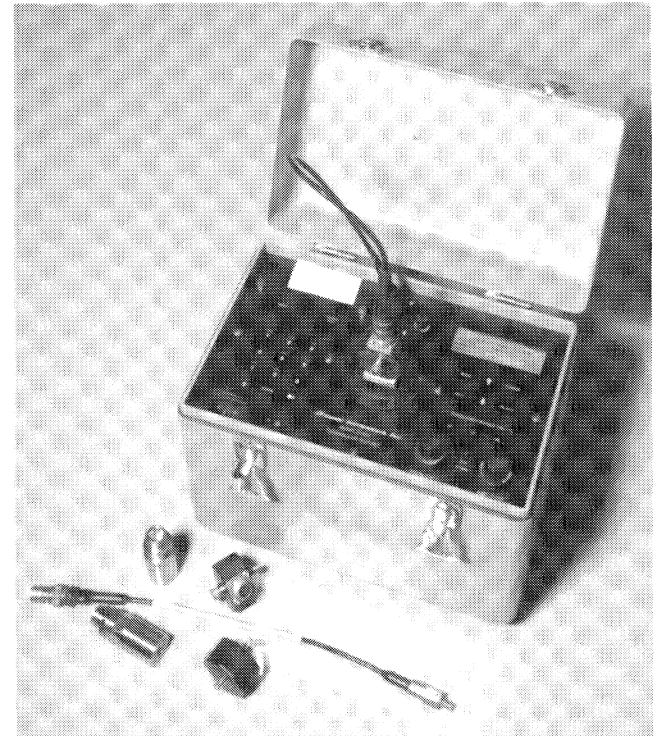


Figure 13. Vibration Transducer Calibrator.

cause differences in material properties can cause scale factor changes. The deliverable cables, connectors, and driver (also called oscillator-demodulator or proximator) should all be used, and it is desirable to use the deliverable power supply as well. The cable electrical length between probe and driver is especially critical as it directly affects the scale factor. While mounted on the calibrator, it is a good idea to check the setup for linearity over the gap range of interest. This can be done using plastic feeler gauges to set the gap between probe tip and target, and plotting the DC voltage vs gap. (Figure 14) Present day displace-

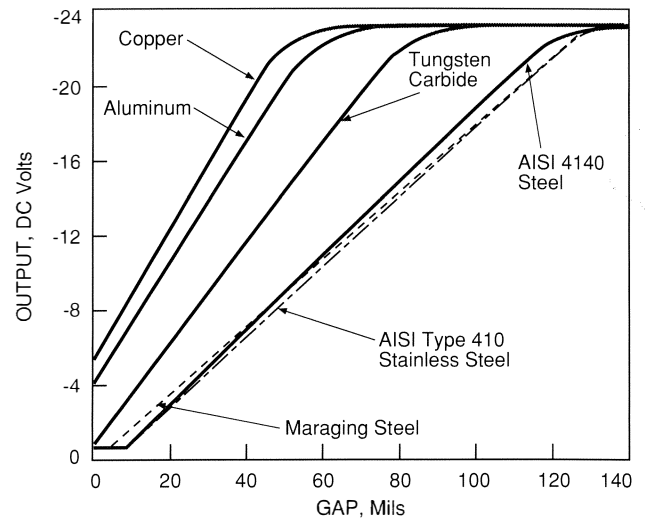


Figure 14. Typical Displacement Probe Calibration Curve.

ment probes are linear over at least 40 mils and should be installed at the center of this linear range.

*Velocity*

With the mechanical calibrator set to deliver a constant velocity signal, the output of the velocity pickup can be checked over a broad frequency range.

*Accelerometer*

The mechanical calibrator is again used, set to deliver a constant acceleration. As with the other pickup types, all associated equipment cables, charge amplifiers, etc., should be used during calibration. The accelerometer must be carefully mounted to achieve the full frequency response of which it is capable. Accelerometers generally come with a calibration chart, which should always be kept with the instrument.

**INSTRUMENT SELECTION**

Although displacement, velocity, and accelerometer devices all have their places, the displacement type is usually preferred for turbomachinery when appropriate for the frequency range of interest. All three types yield amplitude, frequency, and phase information, but the displacement probe also reveals key rotor bearing relationships such as the position and attitude angle of the journal within the bearing using the x and y gap voltages (possible because the output from this type of instrument extends to zero frequency) or by displaying a DC coupled orbit. Additionally, the phase is relatable to a fixed geometric feature on the rotor, usually a key or notch, which is useful for balancing. Also, the displacement probe indication is direct, that is, it detects the shaft motion with respect to the bearing. By contrast, velocity transducers and accelerometers measure resultant motion of the machine housing.

For high frequency analysis, typical for gear vibration problems, the accelerometer is the only choice. The same is true for rolling element bearing machines. These have essentially zero clearance, which negates the value of displacement probes, and when defective, can generate high frequency signals ideally suited to the accelerometer.

Each type of device has an amplitude range for which it is best suited. This is shown in Table 3.

Table 3. Transducer Dynamic Ranges.

Transducer	Lower Limit	Upper Limit	Dynamic Range
Displacement	0.01 mil	50 mil	74 dB
Velocity	0.02 ips	10 ips	54 dB
Accelerometer	0.01 g	50 g	74 dB

Magnetic bearings are now finding application in the industrial turbomachinery world. The position sensors provided as an integral part of the magnetic bearing system can provide essentially the same functionality as displacement probes. Moreover, the journal bearing response during levitation is easily measured with the position sensor leading to verification and calibration of the magnetic bearing control system.

It is good practice to keep the number of vibration instruments to a minimum to minimize the amount of electronic conditioning. Understanding is improved and solutions achieved when

using a limited amount of clean, reliable data vs a large quantity of questionable data.

The factors favoring one type of instrument over another are summarized in Table 4.

Table 4. Instrument Selection Considerations.

<p>Factors Favoring Displacement Type Pickups</p> <ul style="list-style-type: none"> <li>• Lower Frequency Range</li> <li>• Fluid Film Bearings</li> <li>• Low Rotor Mass to Housing Mass Ratio</li> </ul>
<p>Factors Favoring Velocity Type Pickups</p> <ul style="list-style-type: none"> <li>• Low to Medium Range Frequency</li> <li>• Light Support Structure</li> <li>• Medium to High Rotor Mass to Housing Mass Ratio</li> <li>• Need for Simple, Hand-held Device with Strong Signal</li> </ul>
<p>Factors Favoring Accelerometers</p> <ul style="list-style-type: none"> <li>• High Frequency</li> <li>• Anti-friction Bearings (zero running clearances)</li> <li>• Gears Present</li> <li>• Blades or Vanes Present</li> <li>• Medium to High Rotor Mass to Housing Mass Ratio</li> <li>• No Significant Low Frequency Information</li> </ul>

**SCALING**

Apart from using the correct pickup for the type of vibration to be measured, it is also important to choose a proper frequency scale when viewing either the waveform or the spectrum. When viewing the waveform on an oscilloscope, different things can be seen by changing the time axis scale. A short time scale, as in Figure 15, reveals details of the single cycle waveform. Using a very long scale, on the order of several seconds (Figure 16), can reveal long term changes, “beating” and the like. For spectrum analysis, both the frequency scale and the amplitude scaling are important. The careful analyst will routinely explore several frequency scales appropriate to the instrument chosen. Since the analysis time is inversely proportional to the chosen bandwidth, transient and unsteady events are better seen with a broader frequency scale. The “zoom” feature of a spectral analyzer allows any fraction of the complete frequency scale to be

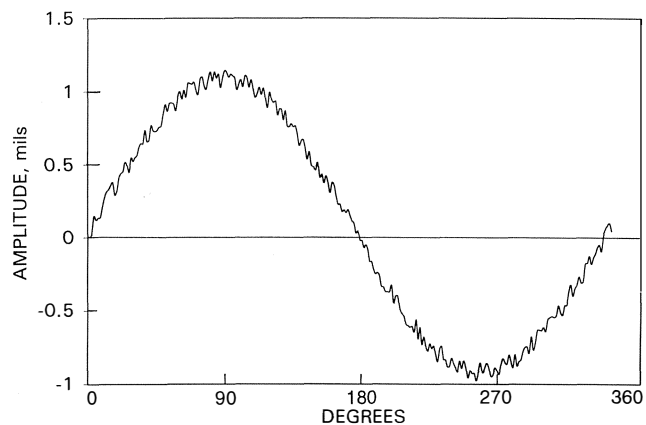


Figure 15. Short Time Axis Display.

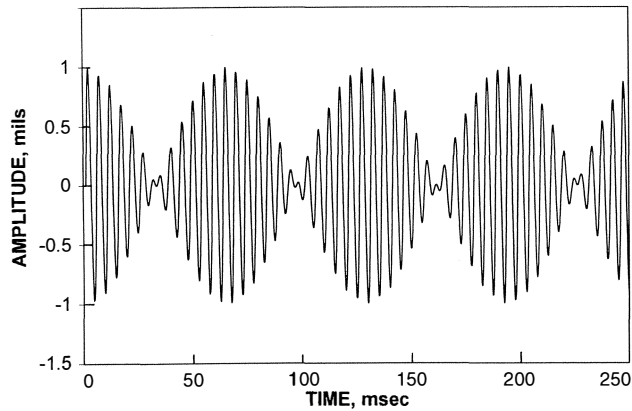


Figure 16. Long Time Axis Display with Beats.

enlarged for far greater detail. This is especially useful in separating closely spaced frequencies.

The choice of linear or logarithmic amplitude scales can also reveal different characteristics. While most people are more comfortable with linear scaling that shows things in a "real world" context, the use of the log scale takes advantage of the very large dynamic range of today's analyzers. Threshold conditions and incipient problems often can be seen on a log scale that would be invisible in the linear world. Compare Figures 17 and 18, which show the same signal displayed in linear and logarithmic formats, respectively.

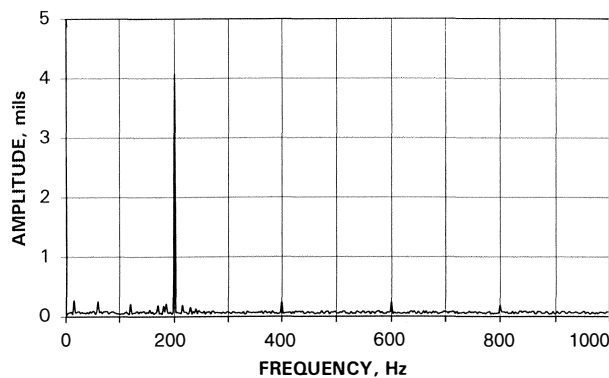


Figure 17. Linear Scaled Spectrum.

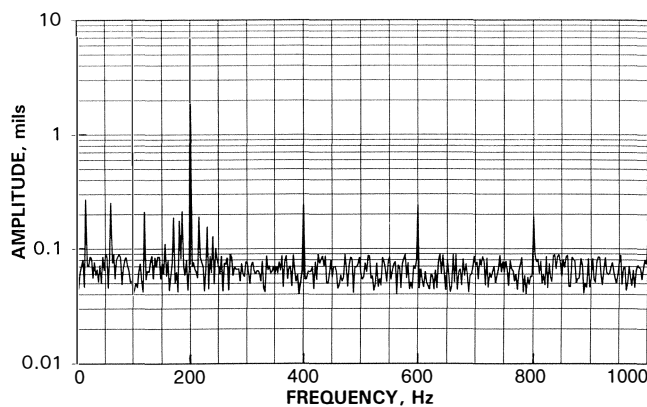


Figure 18. Logarithmic Scaled Spectrum.

The log scale is particularly useful in evaluating machinery with antifriction bearings, because the amplitudes of the characteristic bearing frequencies are usually very low with new equipment; hence, the "baseline" must be established using a very sensitive scale.

## INSTRUMENT ERROR SOURCES

### *Displacement Probe Systems*

While capable of amazing accuracy and detail of signal resolution, there are several common errors that must be avoided with the use of displacement probes.

First, the probe/cable/driver must be calibrated for the correct shaft material. The slope of the gap distance/output signal is usually standardized at 200 mV/mil (or 8.0 mV/micron). By its nature, the output varies with the conductivity of the shaft material. Substituting a different shaft or target material can change the scale factor considerably. Similarly, the length of the cable between the probe and driver is critical, in that a cable with an incorrect electrical length will change the effective slope of the curve, thus leading to erroneous level indications.

Care is needed in first setting up the probe. An adjustable probe should be carefully gapped with a digital voltmeter to ensure it is positioned midway in the linear range. It should be checked again when the unit is running at full load and completely heat-soaked, to see that the operating gap remains correct. A jittery or multiple phase mark may signal an improper gap setting.

Probes must be rigidly mounted to avoid any possibility of mount resonance. Sometimes a mounting bracket will have a natural frequency in the band of interest, and, thus, will resonate and give a false vibration indication. Before operating the machine, it is good practice to tap the mounting structure with the probe installed, but with the machinery not running, and examine the resulting spectrum for signs of resonance. A broad-band noise that quickly dampens out is normal. If one or more distinct peaks appear in the spectrum, probe mount resonance is suspected and the mounting arrangement must be improved to eliminate it. The best way to avoid this is to mount the transducer directly on the machine, without any interposing brackets.

Another setup error that can mislead the analyst is the phasing of the probes from end to end. Normally, the vertical and the horizontal probes lie at the same angular (clock) position from end to end. It is not necessary that they be in the true vertical and horizontal planes, but it is very desirable that they line up with each other from end to end, in the same clock position. When the probes are not at true vertical and horizontal orientations, API standards direct that the probe designated as "horizontal" be the one that lies to the left of vertical center when viewed from the driver end (Figure 19). Always check that the sense of the phase angles reported by the probes is consistent. One set (vertical or horizontal, as the case may be) should always "lead" the other by about 90 degrees. If this is the case for one end but the lead is about 270 degrees at the other end, then one pair of probes is most likely cross-wired and should be corrected to avoid confusion. (If the probes *are* connected correctly and there is a phase difference considerably more or less than 90 degrees, then a rotor, bearing or alignment problem may be present and must be corrected.)

Because intermittent signal and ground faults are particularly difficult to troubleshoot, all electrical (and mechanical) connections must be clean and secure. Interconnecting cables should be prevented from moving and electrical connectors must not be able to contact any metal surface. The connectors must be insulated. A single ground point is essential.

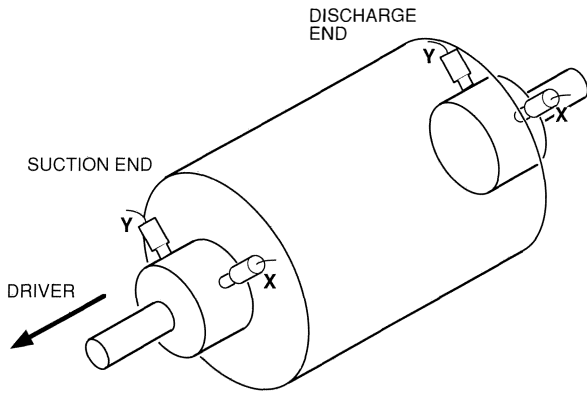


Figure 19. End to End Probe Phasing Convention.

In addition to being positioned at the correct radial gap, the displacement probe must also be at an appropriate axial location along the rotor length. "Adjacent to the bearing" is not always an adequate guideline. The probe position should be selected based on the modal shapes of the rotor as predicted by the rotordynamics analysis. If a probe should be inadvertently located at or close to a node, the output will not be meaningful. The worst case would be with both the probe and the bearing located right at a node or straddling it. Here, the bearing will be unable to provide viscous damping (because there is no motion and therefore no velocity), and the probe will be unable to report meaningful information.

**Accelerometers**

Most industrial accelerometers are of the piezoelectric type in which a minute electrical charge is produced in response to an applied stress. The output of a piezoelectric accelerometer tends to be nonlinear with respect to frequency. Output rolls off at the low end and the useful high end of the frequency range is constrained by transducer resonance. Refer again to Figure 4. Filters must be used to avoid distortion. Additionally, the instrument may be affected by temperature (although some have quite high temperature ratings) and, to some extent, by humidity. Temperature ranges suited to each type of transducer are shown in Table 5. The so-called "cross-axis" sensitivity means that some output will result from vibration not in the principal axis of the device. Lead length is critical. Not only do excessively long leads result in a loss of sensitivity and signal attenuation, they also are susceptible to picking up stray electrical noise. If the leads are not secured and are free to move, so-called tri-

Table 5. Transducer Temperature Operating Range.

Transducer	Low	High
Displacement	0°F	350°F
Velocity		
Standard	-40°F	500°F
High Temperature	-40°F	700°F
Accelerometer		
Internal Charge Amp	-60°F	250°F
External Charge Amp	-60°F	700°F

boelectric noise can be generated. It is good practice to keep to a minimum the signal cable lengths for *all* types of vibration equipment and to avoid placing instrument signal lines in the same conduit or cable runs with ac power lines (Figures 20, 21, and 22).

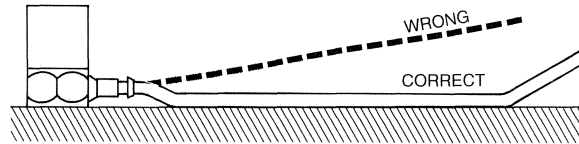


Figure 20. Proper Cable Securing.

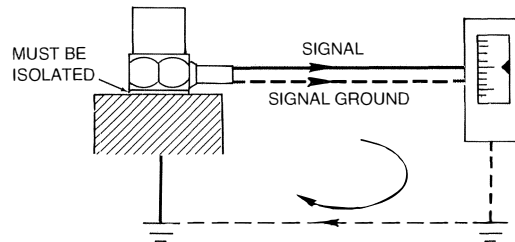


Figure 21. Ground Loop.

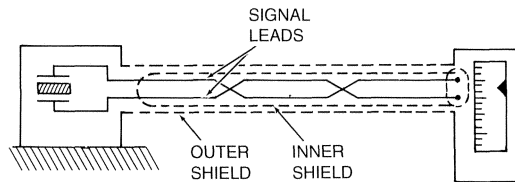


Figure 22. Double Shielded Cable.

When using intrinsically safe barriers in conjunction with a vibration probe, do not drive more than the rated capacity of probes with one ISB. It is a low power unit and exceeding the rated capacity causes a loss of amplitude linearity.

**SIGNAL DISTORTION**

The precise, detailed and often low-level electrical signals used in vibration analysis are subject to numerous distortions. A few of these are discussed in this section.

Electrical cross talk arises from signal leakage into neighboring systems or channels. It can be discriminated by observing the response to a change in the level of signal applied to another channel.

Tape recorders are very useful for minimizing machinery run time and for permanent records. Head alignment is critical especially if phase measurements will be studied. Amplitude modulation (AM or "direct") recorders are limited in low frequency response. Frequency modulation (FM) recorders are usable down to dc but require high tape transport speed if high frequency response is important. Both types exhibit reduced dynamic range (signal to noise ratio) compared to the original signal.

Signal integrators, sometimes used to shift from acceleration to velocity or from velocity to displacement, are useful for some work but can be sources of noise and distortion. Double integra-

tion to derive a displacement signal from an accelerometer is not recommended.

A special case of signal distortion is the clipped waveform. Here, the maximum (and/or minimum) peaks of the waveform is shaved off. Importantly, this can be an actual machinery characteristic or an electrical fault.

The clipped waveform can result from a vibration level which approaches some mechanical limit, such as a bearing or seal radial clearance. Or, it can result from an overloaded electrical circuit. The distinction is important because the resulting signal can be very much the same. The characteristic spectrum of a clipped signal, whether from a mechanical or electrical overload, contains a series of harmonics. Recall that the Fourier transform of a square wave is a series including all the odd harmonics. The square wave is the "worst case" situation; usually the distorted waveform looks more like Figure 23 with corresponding spectrum, as in Figure 24.

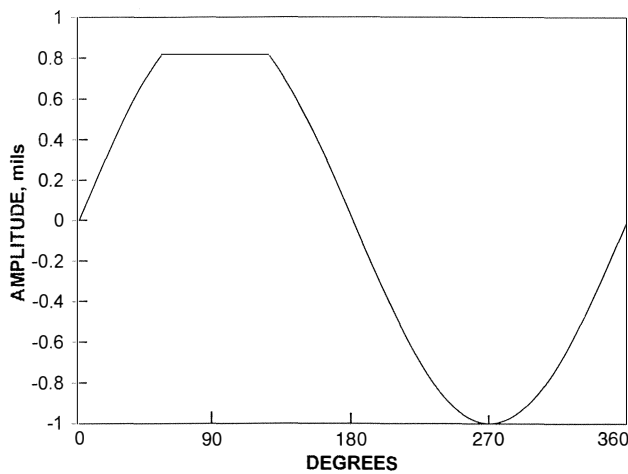


Figure 23. Clipped Waveform.

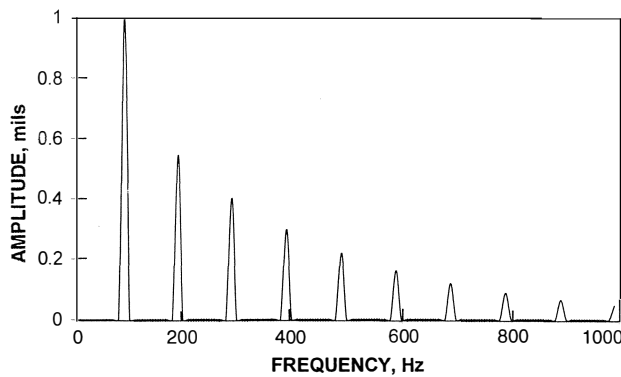


Figure 24. Spectrum of Clipped Waveform.

**SIDEBAND FREQUENCIES**

Analysis of sideband frequencies in the vibration spectrum can reveal the nature of several classes of mechanical faults. Good references exist for using this technique to distinguish such faults as misalignment and tooth errors in gears, looseness in tiedown bolts and bearing caps, and the beating of two machines running at slightly different speeds. In a manner similar to the generation of harmonics discussed in the preceding section, sidebands can also be generated by overload of the

electrical measuring circuit. A spectrum plot is presented in Figure 25 showing sideband frequencies centered around the mesh frequency of a speed increasing gear.

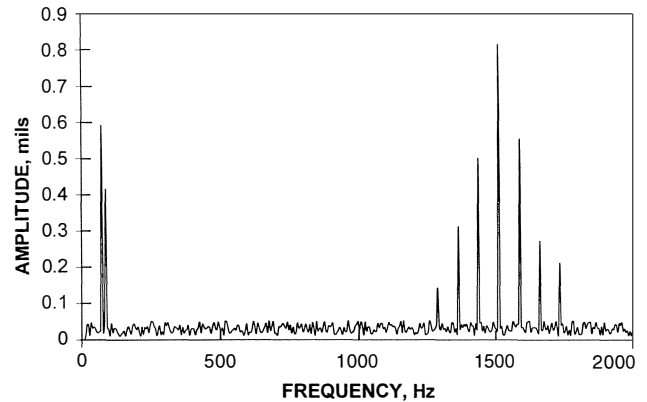


Figure 25. Sideband Frequencies.

**ROTOR BOW**

Rotor thermal bow is a temporary condition that is sometimes caused by uneven heating or, more often, uneven cooling following equipment shutdown. It is classed here as a nonfault because it is usually entirely correctable with no more effort than careful operation. When a piece of high-temperature turbomachinery comes to rest, the rotor cools rather slowly in the absence of the flow of fluid through the casing. Since the upper area of the casing tends to remain hotter, the rotor can take a slight bow upward in the center (the opposite of a gravity sag) (Figure 26). This is a temporary condition; no permanent deformation of the rotor has occurred.



Figure 26. Rotor Bow Due to Uneven Cooling.

Behavior of a bowed rotor is best illustrated with the Bodé plot. Such a plot is shown in Figure 27 for a normal rotor. The significant features are a distinct peak in amplitude as the first critical speed is traversed and an accompanying 180 degree (approximately) shift in the phase angle. With a bowed rotor, when the unit is restarted, there can be violent vibration, since a small bow can equate to a very large mass center eccentricity. The increased vibration will be evident almost from zero speed, but will be especially severe approaching the first critical (translatory whirl) where the modal unbalance aggravates the response to this critical. Ironically, the vibration amplitude at the displacement probe may actually go through a *minimum* at the critical speed (Figure 28). Also note how the phase angle continues to change after the critical speed has been passed.

If the speed of the unit can be controlled, it is usually possible to "roll out" the bow by idling at low speed for a few minutes. For a gas turbine, which is started with a separate driver, the normal



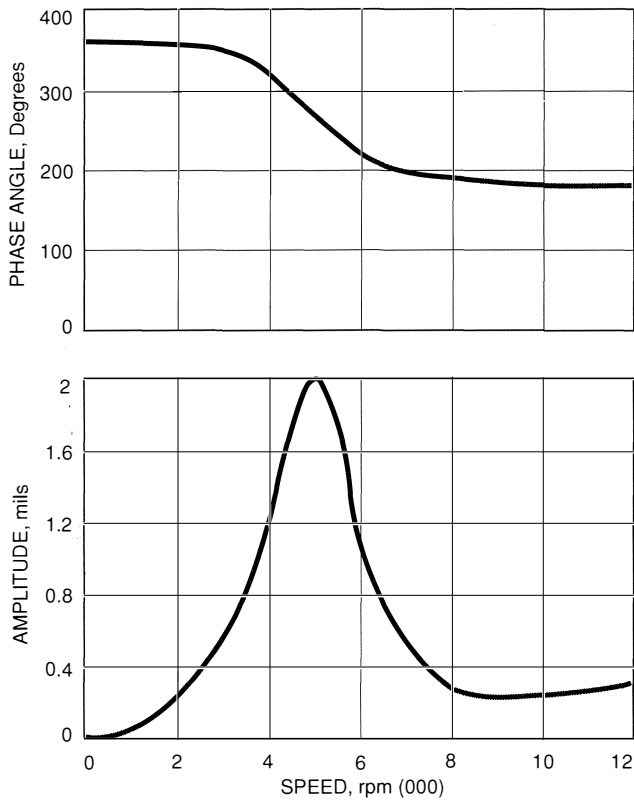


Figure 27. Normal Bodé Plot.

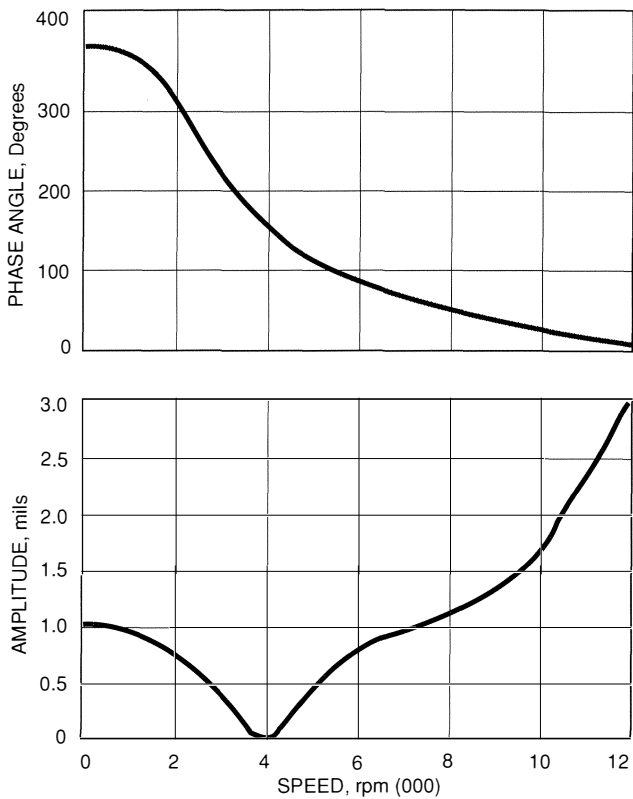


Figure 28. Start-up Bodé Plot with Bowed Rotor.

(or somewhat extended) crank cycle is often sufficient to remove any bow. The vibration readings will steadily decrease as the bow is corrected and the rotor behavior should return to normal. It is important not to hurry this process. Remember that due to the mode shape, amplitudes will be much higher at the rotor center than at the bearings where the vibration is usually measured. The risk is that a heavy rub may occur leading to serious and permanent mechanical damage which results in added unbalance or wear of seals that would lower aerodynamic performance.

It often is characteristic for vibration to be normal when the unit is restarted quickly after shutdown, because there will have been insufficient time for the bow to develop. It is also normal to experience normal behavior when the unit is restarted after a long cool-down period. The problem of rotor bow, thus has a lower and an upper time threshold, and these will vary from machine to machine depending on design and operating temperatures.

Rotor thermal bow can be avoided by cooling the machine by maintaining slow rotation (using a turning gear or intermittent means of rotating the shaft) or by simply waiting for the entire machine to equilibrate.

It should be noted that not all rotor bows are correctable in the above manner. It is possible that a *permanent* sag or mechanical bow may occur and that must be corrected before attempting a restart.

**RUNOUT**

A problem encountered rather frequently with displacement probes (but not seismic pickups) is journal runout. Runout is mechanical out-of-roundness or electrical nonuniformities in the journal or often a combination of the two. It hampers accurate analysis, because it adds to (or subtracts from!) the true dynamic shaft motion and, thus, can obscure the real behavior of the rotor.

*Mechanical runout* can arise from mismachining or mishandling the shaft in manufacture or assembly. A mismachined shaft may have a slight eccentricity, ellipticity or even multiple lobes. Eccentricity of the probe target surface to the running axis (i.e., bearing centers) will produce a once-per-turn false signal. An elliptic, or an egg-shaped surface, will produce predominantly twice-per-turn false signals while multiple lobes (as from poor grinding) will produce multiple harmonics. A once-per-turn event such as a “ding” or scratch will produce a harmonic series in the spectrum. This is summarized in Table 6. Lack of care when installing the probes is a source of surface damage that shows up as runout. Seeing two disturbances in the orbit at 90 degree separation is a clue to this condition. The amount of runout does not have to be large to interfere with accurate

Table 6. Sources of Mechanical Runout and Corresponding Spectral Characteristics.

Cause	Spectral Characteristics
Eccentricity	1 x Running Speed
Ellipticity	2 x Running Speed
Multi (n)-lobed	n x Running Speed
Localized Damage	Harmonic Series

analysis. A mishandled shaft could have numerous small “dings” or scratches on the surface which the probe picks up and dutifully reports as “vibration.” Mechanical runout can be detected by careful measurement before the rotor is installed. The best preventive measures are extreme care in machining and especially in handling.

*Electrical runout* arises from imperfections in the shaft material, as opposed to the shaft geometry. Residual magnetism can result from processing (principally the use of magnetic chucks or improper magnetic particle inspection), concentrations of stress from cold working, or metallurgical nonhomogeneity of the shaft material. The precipitation hardening alloys are particularly susceptible to electrical runout. Physical examination will not reveal this condition, which may be slightly subsurface (the displacement probe “sees” some distance below the shaft surface).

It is important to realize that runout does not always add to the overall level of vibration. Depending on the phase relationship between the once-per-turn component of runout and the true dynamic once-per-turn motion, the runout can actually *reduce* the perceived level. Runout almost always “mushes over” the phase shift while traversing a critical and may mask the severity. One typical runout situation is illustrated in Figure 29. Also note that it is entirely possible (although eminently undesirable) to “balance out” runout. Balancing at a fixed speed can reduce the overall once-per-turn level, which is the composite of runout and dynamic motion. As noted, this is a very bad idea, since it falsely reduces the level and can well interfere with interpretation of future changes in vibration level.

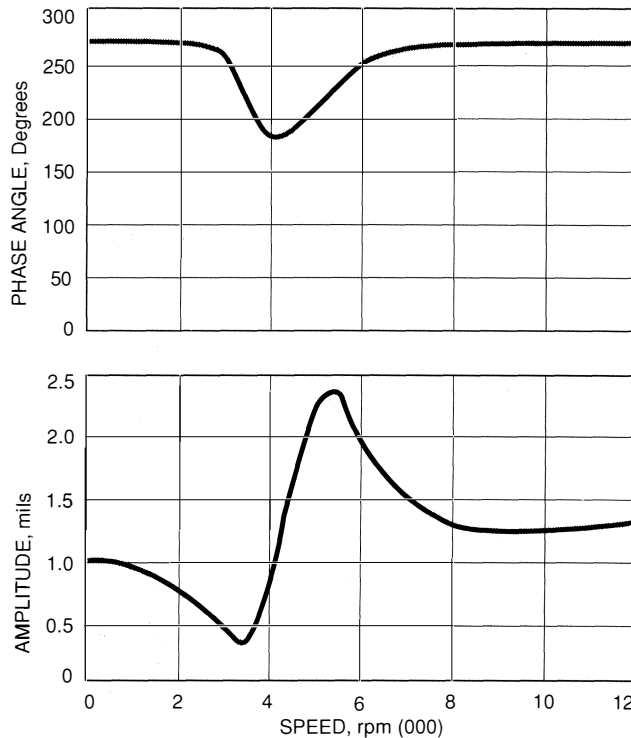


Figure 29. Bodé Plot with Runout.

Various methods are available for reducing runout. For magnetic electrical runout, these include degaussing and micropeening. Mechanical runout may be improved by burnishing the shaft or, in extreme cases, by sleeving or metal coating the affected area. Material nonhomogeneities cannot be easily removed. If excessive, the shaft should be discarded.

Formerly, electronic runout subtraction devices were available which employed a programmable read-only memory (PROM) device that stored a representation of the runout waveform and subtracted it from the composite signal with the result that only the true dynamic signal remained. While quite effective, they appear to have been discontinued due to lack of industry acceptance. Overall, it is far better to avoid runout than to correct it.

Runout that has escaped notice in the preassembly stages can usually be detected by observing a rotor during coast-down. On a Bodé plot (Figure 29 again), the amplitude does not fall off toward zero as the rotor comes to rest, but persists at some level. If no other dynamic effects, such as a locked-up coupling, are present during the coastdown, the residual level can be interpreted as the runout. This is a good time to record the amplitude and the phase angle of the runout on all channels for later use in balancing. On a spectrum chart, the same is true except that runout, such as that caused by a scratch on the shaft, will reveal itself as a harmonic series, the amplitudes of which do not decay until the rotor comes to rest.

With respect to its effects on the running speed component of vibration, runout can be thought of as a fixed vector superimposed on the true, dynamic vibration vector.

As an interesting aside, higher order critical speeds can sometimes be detected by carefully watching the behavior of the second harmonic of the runout spectrum as the rotor slows. An “interference” may occur as the unit speed passes one half of the higher order critical.

## EXTERNAL RESONANCE

Any turbomachine can be mathematically modelled as a combination of masses, springs and dampers representing the rotor, bearings, casing, and structural supports. Analysis of the overall system will reveal a number of resonances or critical speeds corresponding to the number of degrees of freedom modelled. Sometimes, external resonances of the supports, baseplate or foundation can interact with the turbomachine to produce vibrations that are excited by, but not the fault of, the machinery. For several such sources, refer to Figure 30.

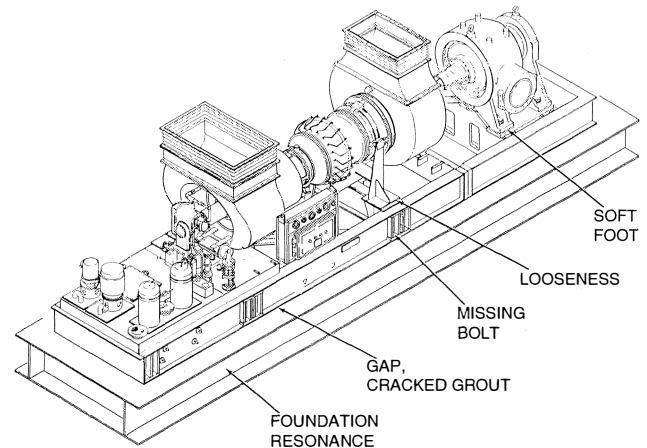


Figure 30. Sources of External Resonance.

These can be the result of poor design or improper installation. Concrete foundations offer good damping and are seldom involved in external resonance. On the other hand, steel supports, which are common on offshore platforms, are much more lively and can definitely have resonances within the machine operating

speed range if not carefully designed. Perhaps the most common installation faults causing structural external resonances are cracked or discontinuous grouting, "soft foot" condition, and loose or missing tiedown bolts. These can leave sections of the baseplate unsupported and able to vibrate vertically at a resonant frequency corresponding to the unsupported length. An effective way to determine if this problem exists is to take (hand held) vertical vibration readings at the machine foot, at the baseplate, and at the foundation. If everything is securely fastened, the amplitude and phase readings will be similar at all locations. With looseness, there will generally be a distinct difference. (Ignore anything above about 500 Hz when using a hand held pickup of any type.) Fretting, the formation of iron oxides due to micromotion, observed at the foot-to-base interface is another clue to looseness.

At first glance, an external resonance can be mistaken for a rotor critical speed. The amplitude will reach a peak at some speed and decrease thereafter, as it would for a rotor critical speed. However, there are some clues to guide the analyst to the correct diagnosis in cases of external resonance. Normally, the bearing characteristics will be approximately isotropic (the same stiffness and damping in the vertical and horizontal planes), so nearly circular orbits are expected. However, with external resonance, vibration amplitude will usually be stronger in one distinct plane. This is often, but not always, the vertical direction. With displacement probes, this will be revealed as a strongly elliptical orbit. Another telling clue is a phase angle plot that fails to complete an approximately 180 degree shift as amplitude peaks, but that instead recovers as shown in Figure 31.

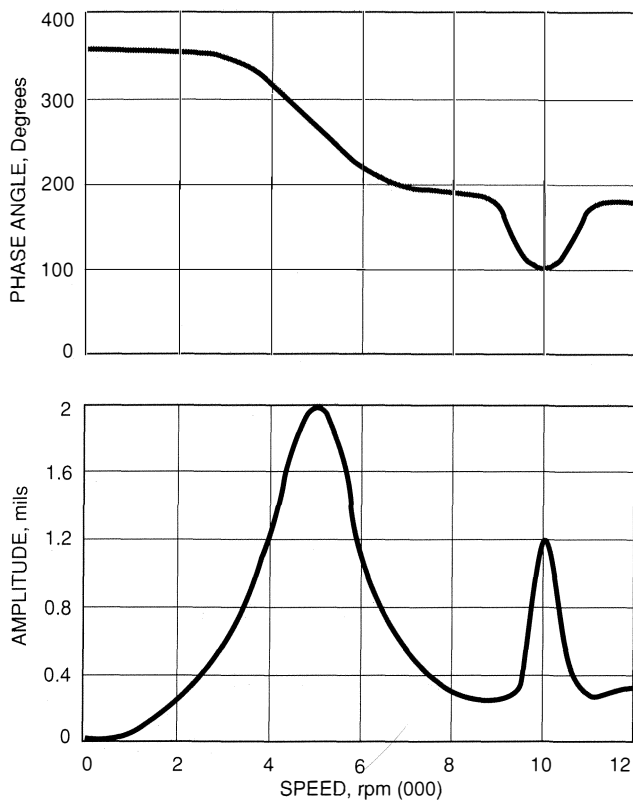


Figure 31. Bodé Plot with External Resonance.

Other sources of external mechanical resonances are piping systems, ductwork and expansion joints that can resonate and feed back into the rotor/bearing system.

External resonances can be aerodynamic/acoustic in origin as well as mechanical. Watch for the effects of pressure pulsations (as from reciprocating compressors) and unusual acoustic phenomena in piping systems. In general, any unnatural, periodic noise is a candidate for causing trouble and should be investigated.

## EXAMPLES

Following are several actual examples of false vibration in the form of condensed case studies.

- A new centrifugal compressor in natural gas transmission service exhibited much higher vibration on all channels in the field as compared to factory running tests. It was operating in the alarm region irrespective of the speed or operating pressures. The manufacturer's service representative arrived prepared to rebalance the rotor, but discovered that there was a component in the spectrum at 60 Hz of almost 1.0 mil. This was determined to be electrical noise and was traced to the fact that the displacement probe cables were run in a common tray with 60 Hz ac power lines. The problem was solved by separating the lines, thus eliminating the induced electrical signal.

- A back pressure steam turbine driving an integrally geared (four-poster type) air compressor ran at low vibration but could not be restarted after a brief shutdown to repair an oil leak. On restart attempts, vibration increased rapidly with speed. A service representative arrived the following day and was able to start the unit without incident. It was then manually shut down and a hot restart attempted. Again, the vibration was very high, but a restart a few hours later was successful. A thermal bow had developed while the hot rotor was at rest for a short time. Consideration was given to adding a turning gear to slow roll the train but, instead, a hot start procedure was developed that eliminated the problem. Using this procedure, the turbine was brought to a low idle speed and held there for about 20 minutes, so the bow could "roll out" after which the rotor could be accelerated to operating speed with no adverse effects.

- After routine maintenance, a centrifugal compressor operated at much higher vibration than previously but only at the suction end. Alignment was carefully rechecked and found to be correct. A spectral analysis showed a series of harmonics that resembled looseness. However, on coastdown, the harmonics were seen to remain high until the rotor came to rest. The problem was traced to a marred shaft under the displacement probe caused by improperly reinstalling the probe. The probe had been forced into contact with the shaft and produced a localized defect. It was necessary to disassemble the unit to polish the shaft and eliminate the runout condition.

- A 4.0 MW gas turbine driving an electrical generator through a speed reducing gear on an offshore platform exhibited high vibration in the vertical plane but normal levels in the horizontal direction. In carrying out a vibration survey of the base and substructure, it was found that the steelwork supporting the train was driven into resonance by the running speed of the turbine. The structure was stiffened by adding members that raised the natural frequency above the maximum running speed. With the resonant frequency raised above the operating speed range, vibration in both planes remained low and acceptable.

- A speed increasing gear located between an electric motor and a centrifugal compressor ran with a very high acceleration level under all conditions although it had recently passed a thorough factory test. After considerable investigation, it was found that the wrong cutoff filter was installed and this allowed the accelerometer resonant frequency response to be transmitted to the readout device. A lower frequency low pass filter was installed and the indicated vibration level returned to normal.

• A pipeline booster natural gas compressor consisted of a single overhung impeller on a two bearing rotor. During factory test, vibration at the coupling end was found to be out of specification. Trial weights were added to the coupling hub, but resulted in almost no change in the high vibration level. Instead, the vibration at the impeller end, which was initially satisfactory, increased beyond the acceptable limit. The calculations showed that, to eliminate the unbalance, a very large correction weight would be required. It was impractical to add this much weight, and it would have driven the impeller end vibration far above the limit. The recommendation was made to tear down the compressor and rebalance the rotor. However, just before doing this, the test engineer took the time to trace the vibration probe leads and found that the impeller end and coupling end probes were reversed. The high vibration was actually at the impeller end. This explained why the trial weights were ineffective in reducing the vibration level. Now, small trim balance weights were added at the impeller and the rotor balance was improved enough to allow full speed operation within specification.

Thinking back, the clues to this situation were present, if not initially recognized. Initial high vibration was thought to be at the coupling end of the rotor, although a small fraction of the overall rotor weight was supported by the coupling end bearing. It is much more likely that the heavy impeller would have the large unbalance. Also, efforts to trim balance the rotor were ineffective and produced a larger change at (what was thought to be) the opposite end.

## CLOSURE

Vibration problems in high-speed, high-power-density turbomachinery can be the result of dozens of causes. Refer to the APPENDIX for a listing of many of these causes; undoubtedly, there are others. Because the potential sources are so numerous, the temptation to “do something” quickly is strong and can overpower the more rational process in which positive identification of the cause(s) of vibration precedes the corrective action. Many vibration problems are truly problems within the machinery. However, the analyst should maintain a healthy skepticism, always bearing in mind that numerous “faults” can be caused by external events. It is both frustrating and uneconomical to chase the wrong gremlin. First, find out what is wrong. Then, fix the real problem.

## APPENDIX

### *Vibration Sources in Turbomachinery*

#### *Rotor Related*

#### Unbalance:

- static
- dynamic
- built in couples

#### Proximity to critical speed

Rotor instability—resonant subsynchronous vibration  
(reexcitation of first critical speed)

Bent shaft (permanent rotor bow)

Thermal bow (temporary—hot restart)

Cracked shaft

Foreign object left in rotor cavity

Trapped liquid in rotor

Noncircular shaft effects (anisotropic shaft stiffness)

Torsional critical speed—startup or running

Runout

- electrical

- mechanical

- combined

### *Journal Bearings—Fluid Film*

#### Bearing instability

- oil whirl @ 41 to 43 percent of running speed
- oil whip @ rotor first critical speed

Low bearing stiffness/poor damping due to excessive clearance or wear

Oil flooded bearing, gear or coupling guard

Bearing oil starvation—dry whirl

Radial bearing overload

Radial bearing low eccentricity and stiffness due to insufficient load

Tilting pad journal bearing pad flutter (rare)

Instability induced by bearing anisotropy (vertical/horizontal stiffness difference)

Bearing nonlinearity due to nonuniform tilting pad assembled diameters

Bearing preload/bore misalignment effects

Bearing preload—external forces (piping distortion, fluid)

Edge loading effects

Pivot wear or fretting

Oil retainer lockup instability

Babbitt fatigue

Failure of babbitt bond

Incorrect TPJ pad bore diameter

TPJ load deformation of pad, pivot, or support ring

Miscellaneous bearing mismachining (chamfers, oil supply holes, etc.)

Eccentric pivot TPJ installed reversed

Bearing clogged incorrectly

Bearing nonlinearity (sum and difference frequencies)

Bearing damage due to arc welding

Shipping damage (brinelling or fretting)

Materials incompatibility (wire wool effect)

Inadequate crush on split bearing

### *Antifriction Bearings*

Inner race defect

Outer race defect

Ball(s) defect

Cage defect

Inadequate preload

Skidding of rolling elements

Misapplication

Inadequate (or excessive) lubrication

### *Seals*

Seal ring lockup (oil seal not centering)

Oil seal ring rotation

Instability induced by labyrinth seal preswirl

Gas compressor dry gas seal radial shift

### *Lubrication System*

Excessive oil pressure

Oil foaming/poor drainage

Aerated oil supply—inadequate reservoir residence time

Trash in oil—bearing damage

Inadequate or trapped drain line

Wrong oil—type, viscosity or additives

Oil additives depleted (antifoam, antioxidation or VI improver)

Oil supply temperature too high or too low

Excessive seal air in oil drain line

*Gear*

Gear tooth error—mesh frequency or multiples (high load related)  
 “Float” in gear bearing under light load—subsynchronous  
 Coupling PD eccentricity, ellipticity  
 Gear defect—hunting tooth related  
 Gear nonhunting ratio missreassembly  
 Interaction of gear rotor critical speed and gear meshing  
 Double helical gear apex runout  
 Epicyclic gear excessive ring to planet clearance (“hula hoop”)  
 Inadequate or excessive gear backlash  
 Broken or chipped tooth  
 Gear eccentric PD  
 Eccentric mount  
 Gear dipping into oil in housing

*Couplings*

Gear type coupling lockup  
 Misclocked coupling after wear in  
 Missing or inadequate lubrication  
 Coupling tooth wear, fretting  
 Gear coupling improper axial engagement  
 Gear coupling inadequate root/crown clearance  
 Gear coupling excessive root/crown clearance  
 Gear coupling lack of proper crown - binding  
 Coupling to coupling guard inadequate clearance  
 Low speed coupling improper limited end float setup  
 Coupling lateral critical speed  
 Coupling torque lockup  
 Gear coupling lube grease void unbalance  
 Flexible disk (dry) coupling axial resonance  
 Flexible disk coupling incorrect axial setup

*Instrumentation*

Probe support resonance  
 Seismic probe resonance  
 Electrical noise—line frequency and multiples  
 Seismic probe at “lively” case location  
 Improper probe/transducer calibration  
 Poor seismic probe mount  
 Intermittent electrical grounding  
 Probe linearity (gap out of range)  
 Noisy monitor power supply (60, 180 Hz)  
 Signal overrange (clipping, sum, and difference effects)  
 Transducer crosstalk  
 Transducer crossaxis noise

*Fluid Dynamic or Acoustic*

Fluid induced subsynchronous vibration  
 Internal leakage or bypass  
 Centrifugal compressor surge or incipient surge  
 Rotating stall in gas compressor  
 Instability induced by labyrinth seal eccentricity  
 Cavitation in bearing or driven pump  
 Resonance/pulsation in connected gas piping  
 Blade, splitter or vane passing frequency disturbance  
 Gas pipe flow turbulence  
 Choke/turbinizing  
 Eddy shedding frequency (Oaelian harp)  
 Helmholtz resonator effect  
 Organ pipe effect—standing acoustic wave  
 Strouhal effect at pipe branch  
 Bypass line choke or dynamics

*Installation or Misoperation*

Misalignment—offset or angular due to process piping thermal growth/pressure effects

Nonlinear support stiffness (cracked grout, gap under skid)  
 Soft foot casing distortion  
 Loose hold down bolts  
 Liquid carryover or slugging  
 Foundation sag or distortion  
 Consequence of reverse rotation (damage to noncentral pivot bearing)  
 Compressor fouling, dirt or polymer buildup  
 Damage due to foreign object ingestion

*Accessory or External*

“Noisy” accessory—gear, pump, etc.  
 Induced from driver  
 Induced from driven equipment  
 Housing resonance  
 Pressure pulsations (as from parallel reciprocating compressors)  
 Exhaust duct or other resonance/drumming  
 Proximity to structural resonance

*Build Fault*

Rub (radial or axial; steady or intermittent)  
 Loose rotating component  
 Induction motor rotor/stator eccentricity (“hunting”)  
 Bearing loose mount or low axial retaining force in housing  
 Rotor low axial preload (poor stack)  
 Rotor element misclocked  
 Poor journal surface finish  
 Rotor inadequate shrink fits  
 Friction induced whirl—bad fit

*Miscellaneous Design Faults*

Heat (or cold) soakback into bearing area or along shaft  
 Asymmetrical thermal distortion of bearing housing  
 Impeller disk or blade resonance  
 Thermal stress relief of bearing housing  
 Thermal transient on start; bearing clearance change  
 Bearings located at nodal point—inadequate damping  
 Turbine combustor rumble  
 Poorly damped stretch bolt

*Miscellaneous*

Friction induced whirl (as from loose shrink fits)  
 Torquewhirl  
 Broken rotor bar—electric motor  
 Turn-to-turn short—electric motor  
 Cavitation—liquid pump or bearing  
 Electrostatic discharge bearing damage  
 Thermal relaxation of locked in stresses

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