Improve Turbomachinery Reliability by Taking Corrective Procedures

Age old problems of unbalance, distortion, misalignment, critical speed, looseness and vibration due to stator-rotor contact still head the list as major causes of turbomachinery outages. Here is what to look for, where to measure, and what to do.

High speed turbomachinery is inherently reliable, having few moving parts, but on occasion, even these machines break down. However, if units are properly installed, maintained and operated, service interruptions can be minimized. Problems arising in the field during operation of high speed turbomachinery, as well as their causes, are identified and the solutions are discussed.

Fig. 1 identifies common vibration problems. Unbalance, one such problem, always shows a strong “one cycle per revolution” response over the entire speed range, with strongly pronounced rotor flexural critical speeds, especially at the higher critical’s. This response, however, depends on the location of the unbalance. Serious unbalance problems can occur with machines running above 1.8 times the first flexural critical, because of the necessity of balancing in more than three planes.

There have also been many serious problems with long, thin rotors running at a high critical speed ratio, especially at ratios of 2.5 and higher. To correct such problems, disassemble the rotor and balance each component very precisely on a balancing arbour. Also, repeat balancing after each component is assembled on the shaft. In this way, the balance corrections are made close to where the unbalances occur, and internal couples will not result.

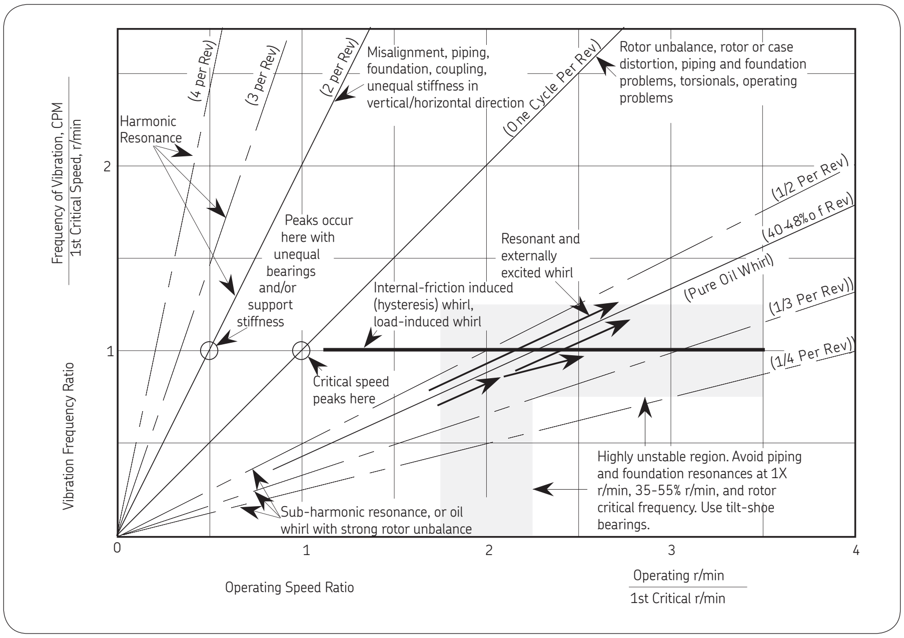


Fig. 1. Frequency versus speed is plotted for common vibration problems.

Real problems occur if the rebalancing is done before the rotor has completely and uniformly cooled after the assembly of each component. Cooling can take up to 24 hours in an environment that should be free of drafts and heat radiation. Also, turn the rotor occasionally to promote even temperature distribution. Any attempt to rush matters may result in a rough-running rotor, requiring another disassembly and rebalance from scratch.

The same basic problem exists with integral disc rotors. The balancing procedure here relies heavily on statistical methods and laws of probability.

The obvious solution to the multi-plane balancing problem is, of course, not to run at such high operating speed/critical speed ratios. However, this solution increases cost and requires more bodies in a given compressor train. Recently, several companies have installed full speed balancing facilities that can be used to check and correct difficult rotors.

Unbalance problems caused by thermal distortion of the rotor are quite common. Naturally, turbines are more affected than compressors and motors, but even with these it is by no means rare to run into thermally unstable shaft forgings, which take a bow when they are heated to an elevated temperature. This bow will not straighten out with time unless the rotor is allowed to cool off. Turbine rotor forgings are tested for heat stability for this reason, being rotated slowly in an oven while temperature is increased and decreased several times. Runout is recorded at several places along the rotor. Maximum allowable runout, at 38 °C *(100 °F)* above operating temperature, is usually about 0.1 mil/ft. of bearing span, but not to exceed 1 mil.

The recent upsurge in problems caused by unstable rotor forgings may well be the product of lower quality standards of the forgings in an effort to reduce cost. Some turbine manufacturers, for example, no longer have a heat stability test for their rotor forgings unless the customer specifically calls for it. Also, the final test on the finished machined rotor is only rarely performed. Rotors for compressors, pumps and electric motors are usually not heat stability tested at all.

The first time a rotor is heated it will take a temporary bow until the entire surface of the rotor is evenly oxidized, at which time the bow should disappear. This phenomenon can also be observed during the first start of a turbine. Therefore, behavior of a machine during the first start and even during the first run cannot be considered to be representative, and it makes no sense to take a vibration signature or corrective action unless the machine has been repeatedly started and loaded. Also, turbine rotors take a bow with a rapid change of steam or gas temperature. A rule of thumb is that steam-temperature fluctuations should not exceed a rate of approximately 4 °C *(8 °F)* minimum.

With gear-type couplings, problems with excessive radial clearance are common, 7 mils is an average. Certain designs also open up another 3 to 6 mils under centrifugal stress. Check radial clearance by mounting indicators on shaft and outside diameter, and by jacking up at the spacer. Radial clearance should not exceed 1 mil as a general rule, and even less for high speed machines.

Watch for distortion

Distortion causes vibration in an indirect way, either by generating misalignment between driver and driven machine, or by causing internal rubs or uneven bearing contact. These, in turn, transmit forces to the rotor, inducing it to generate forces of its own, such as unbalance and a wide variety of oil film and friction induced stresses.

There is no single distinguishing frequency for distortion, but one cycle per revolution is the most common. Also, loads on casing supports can shift, setting off a series of resonance problems. Piping forces and foundation distortion often cause this type of difficulty, with the casing feet sometimes lifting clear off the foundation or soleplates. Misalignment may be caused by improper grouting or rust under soleplates, especially where non-shrink grout is used. Under no circumstances should carbon steel shims be used.

Foundations and baseplates that are exposed to the weather are notorious for thermal distortion. Remedy: protect the foundation from the weather by providing a roof, shades and windbreakers. This is a relatively inexpensive way to get a very significant improvement in plant reliability.

A similar type of problem occurs where the foundation or base is locally heated by steam pipes or by poor insulation of the machine's hot parts. Here, hot soleplates may buckle and break loose from the grout, causing serious vibration problems, especially if they are long or large.

Steel bases can warp badly from both thermal stress and pipe forces. If steel baseplates are not stress relieved after welding, there will be a gradual redistribution of stresses within the base over several years of operation, and alignment of the unit will be continuously changing.

Another cause of distortion is binding of sliding supports because of inadequate lubrication. Do not underestimate the importance of lubrication under the sliding feet of compressors and at the exhaust of condensing turbines. There are some very large compressors that do not allow for any thermal expansion of the casing. They have neither axial nor transverse keys; the feet are simply bolted to the pedestal and usually are not even dowelled.

At present, there appears to be no satisfactory method of keeping these compressors in alignment. Piping forces push them all over the place if the pedestal has enough flexibility to absorb the thermal growth. If not, the pedestal may be so badly strained that it will crack all around, usually on the inside of the support pillars. The best way to keep some semblance of stability is to dowel all four feet, using 3/4 in. dowels. This, however, requires redowelling after each realignment. Further, sliding feet should be properly lubricated by using either solid-lubricant impregnated shims or Teflon shims, so that the center keys are not severely strained, and the machine cannot move to one side as a result of one foot binding.

Coupling misalignment common

Misalignment causes friction or deflection forces in the coupling, and this in turn brings about the rotor and bearing system to deflect and to amplify these forces. Secondary phenomena, such as harmonic resonances, which can become severe may also be created. For gear couplings, vibration frequencies are characteristically two times the running speed for gear couplings, but frequencies at running speed are also often observed. Another strong frequency is sometimes the number of coupling teeth times the running speed. Failures may occur in the seals, couplings, bearings, or shaft ends.

Coupling misalignment can be caused by thermal expansion of the casing support structure, by settling or thermal distortion of foundation and baseplate, or by piping forces that deflect the casing and its support. Foundation problems occur where units are mounted on structural steel baseplates that are usually quite flexible, at least in the horizontal direction, and that generally have insufficient mass and unpredictable resonant frequencies. The high thermal conductivity of steel adds considerably to the problem.

Individual footers, common in building design, should never be used for the support of rotating equipment. New foundation design should include a top slab of proper mass and rigidity – a heavy continuous mat supported on piles if necessary. Also, each unit should be mounted on an individual foundation to control resonant frequencies and to prevent vibration transmission. The columns should have adequate rigidity in both vertical and horizontal directions, and the resonant frequencies of the structure should be tuned so there is no interference with operation.

Do not be afraid of critical speed

Critical speed is the speed at which a shaft buckles, because even minute unbalances cause large shaft deflections due to centrifugal forces. This differs from resonant vibrations in that the shaft does not vibrate back and forth, but rotates with an ever increasing bow. Consequently, if deflections are not controlled, the shaft would ultimately bend rather than fail in fatigue as is the case with resonance.

The difference between critical speed and resonance cannot be overemphasized, because there are still many rotor vibration problems that could be avoided by properly distinguishing between the two. If a designer regards a critical as a resonance, the bearings may be too small in diameter and of the wrong design and the bearing support structure may lack mass and rigidity (→ **figs. 2** and **3**). The remedy for resonance (internal damping) is totally ineffective or very harmful, and can cause a friction induced whirl for a rotor with a critical speed problem.

The end result could be that critical speeds show up at the wrong speed, that they are excessively rough and that other instabilities of the rotating system may occur. All this makes the machine difficult to balance at operating speed and may require extensive redesign to correct the resulting problems.

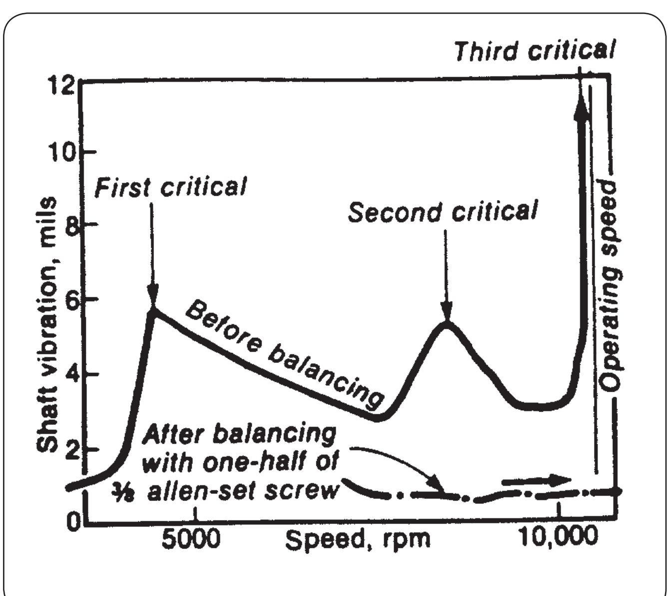


Fig. 2. Vibration plot of a turbine, uncoupled, exhaust end. Turbine exhibits excessive amplification at criticals due to insufficient rigidity of bearing case mounting.

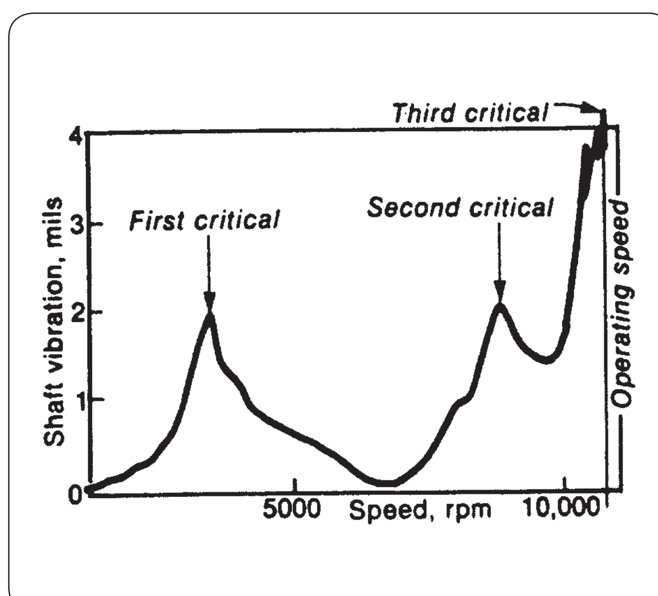


Fig. 3. Vibration plot of 11-stage centrifugal compressor shows excessive amplification at criticals due to excessive bearing span resulting in a critical speed ratio of 3:9.

Most large, high speed machines with a wide operating range will have one or more criticals within operating speed range, because the interaction of rotor, bearings and foundation results in many critically that cannot all possibly be avoided. The designer must make sure that amplitudes remain within normal limits, even with unbalances that occur during normal operation. The designer must also make sure that no unsafe amplitudes can develop from upsets or emergencies (→ **fig. 4**). The new API specs for turbines and compressors (API 612 and API 617) include detailed procedures (→ **fig. 5**).

It can be perfectly safe and reliable to operate on certain well damped criticals, as indeed many machines do – intentionally or otherwise. If vibration levels are normal and there is no pronounced sensitivity to unbalance, then there is little reason to worry.

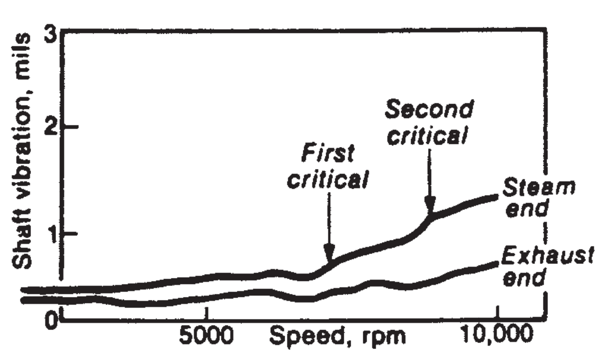
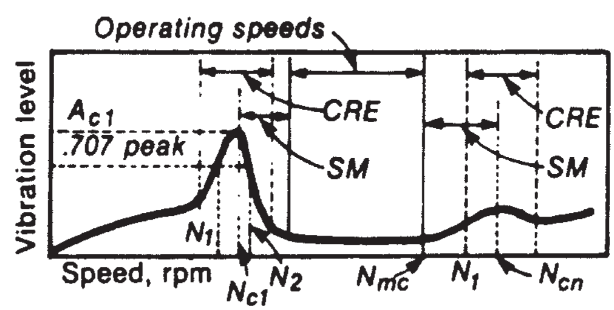


Fig. 4. Unbalance test run of turbine with artificial unbalance at coupling overhang. Amplification at criticals is very low because machine has large rigid bearings, large shaft diameter and short bearing span. Criticals are within operating speed range and safe for operation.



N_{c1} = Rotor first critical center frequency cycles/min
N_{cn} = Critical speed
N_{mc} = Maximum continuous speed, 105%
N_1 = Initial (lesser) speed at 0.707 × peak amplitude (critical)
N_2 = Final (greater) speed at 0.707 × peak amplitude (critical)
SM = Separation margin
CRE = Critical response envelope
A_{c1} = Amplitude at N_{c1}

Fig. 5. Rotor response plot from API specs 612 and 617.

Looseness disguised problem

There have probably been more problems due to looseness than to any other single cause. The higher the speed, the more critical tight assembly becomes. Often these vibrations are misinterpreted for some other type of malady, and long shutdowns result because the problem does not respond to the treatment prescribed.

Typical symptoms of a loose assembly of bearings or bearing liners look exactly like either an oil whirl (frequencies approximately 43 to 50%) or like an hysteresis whirl (frequency equals rotor critical speed). Even a unit with tilt shoe bearings can experience these whirls, if the bearing is not mounted in the casing with an interference fit of at least 2 mils total, diametral.

Often the looseness required to produce even the most violent vibrations is microscopic, sometimes only a fraction of $1/1\,000$ of an inch. Remember, everything that is not absolutely tight in the bearing assembly with a positive interference fit is suspect on a high speed machine. A loose fit may work fine nine times out of ten, but the tenth case could be one not easily forgotten.

Ball seat bearings are notorious for this type of problem, unless the ball seat is securely locked after bearing alignment. The belief is that ball seats are self-adjusting, which they definitely are not. To verify this, check the friction vectors. Machines with ball seat bearings require a thorough checkout of the actual bearing position before the bearing is "locked" in place. This includes blueing of the bearing, as well as checking of seat contact and clearances.

The bearing mounting should be checked as the very first move in case there is a sub-synchronous vibration. The normal procedure to check bearing crush is to put approximately 5 mils of shimstock between the bearing case split, and plastigage (or small pieces of lead wire) all around the bearing. Then tighten the bearing cap bolting. The crush is determined by subtracting the shim thickness from the plastigage reading. Diametral crush should be 1 to 2 mils minimum. For large bearings, use approximately 0.5 mil/in.

Shims on the bearing should only be used in an emergency, and the shim must be placed and secured to prevent blockage of the oil supply. If the bearing bracket bore is too large, remove an adequate amount of material at the bearing bracket split.

A different type of problem occurs with looseness of the casing on its supports. Check all bolts in the support structure for tightness, including casing hold down bolts and soleplate bolting. Also, make a thorough check for any gaps between feet and other mounting surfaces using a feeler gauge, giving special attention to clearances under sliding casing feet, cracks in the foundation, clearances in guide keys, etc. Mounting indicators around the machine to observe thermal expansion during warm up and cool off is also good practice.

Many potential causes for trouble can be observed in this manner. Listening to the machine with a listening rod and feeling with the fingertips for differential vibration at mating surfaces are other simple and effective methods for finding looseness in the structure.

The condition of the insulation, lagging and heat shielding is always important to note when troubleshooting. Few people realize how important it is to prevent heat radiation to supports and foundation, and to maintain uniform casing temperature to prevent warping.

Prevent rubs at all cost

To prevent rubs is the purpose of all shaft vibration limits. No real harm is done until the rotor metal contacts the stator metal. Of course, rubs are not the cause of the problem, but rather the result of some other form of difficulty, such as piping and foundation problems, thermal shock and many others.

For rubs, frequencies can be almost anything, often including the rotor or stator resonant frequencies, though these are not nearly as clear and strong as in the case of looseness or friction excited whirls. Characteristics is the presence of many harmonic and resonant frequencies all over the spectrum.

A type of rub deserving mention is the so-called dry whirl. In this case, the rotor just rolls around inside the stator clearance, like a planetary gear, with an enormously high frequency, often ultrasonic because of the small difference of diameter of rotor and stator. This can happen, for example, where inducer vanes of open impellers, or shrouds of closed impellers, radially strike the stationary shroud, if the rotor and/or bearing flexibility allows this kind of orbiting.

The dry whirl is more apt to occur while the rotor is passing through a critical, when the shaft has no bending stiffness. An enormous amount of energy is consumed in this process, and the unit may actually slow down momentarily. There may be no sound because the frequencies are often above hearing range. But parts such as bolts and nuts, and sometimes the whole bearing case, may fall off. Here, the appearance of the rubbed parts is characteristically mushroomed rather than wiped, indicating rolling contact rather than a sliding or wiping one.

To avoid this, allow enough radial clearance, especially seal clearance, while taking into account abnormal deflections at criticals and during upsets. Sharpening the tips of vanes and/or making the stationary member of a soft or an abradable material, such as honeycomb, felt metal, carbon, sinter aluminum, etc., will also prevent dry whirl, or at least lessen its severity.

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