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MONITORING AND DIAGNOSIS OF VIBRATION IN ROTATING MACHINERY

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1.0 SCOPE

This data sheet provides guidelines for periodic and continuous monitoring programs for rotating machinery considered important to plant operations.

Information on vibration monitoring of reciprocating machinery can be found in Data Sheet 13-26, *Internal Combustion Engines.*

1.1 Changes

January 2000. This revision of the document has been reorganized to provide a consistent format.

2.0 LOSS PREVENTION RECOMMENDATIONS

2.1 Operation and Maintenance

The recommended approach to vibration monitoring is as follows:

2.1.1 Immediately after installation of a turbomachine or a new rotor, baseline traverses from zero to maximum speed should be made for each bearing. These traverses should be done using tracking filters so that plots of amplitude vs. speed can be obtained. They should be done, at least, with bearing cap instruments mounted horizontally. Traverses with the instruments mounted vertically and with the shaft proximity probes may be taken if desired. The purpose of the latter would be to confirm that cap instrumentation is more sensitive to unbalance.

2.1.2 At some convenient later time, similar traverses should be taken during a deceleration.

2.1.3 When an increase in vibration occurs, especially if it is sudden, a diagnostic vibration analysis (including vibration frequency analysis) should be made. The increase of concern would be 0.5 mils for machines running at 0.5 mil or less, or 1.0 mil for machines operating at over 0.5 mils. (One mil equals 25 microns.)

The diagnostic analysis should consist of:

1. A frequency analysis of the vibration signal at the operating speed by means of an analyzer, which produces a spectrum of vibration amplitude vs. frequency component, and

2. A vibration scan (preferably with bearing-cap instrumentation) during a deceleration from, and acceleration to, operating speed. The scanning instrument should have a tracking filter to produce a record of vibration over the speed range, and to analyze such record into its individual frequency/amplitude spectrum.

The vibration scan referred to above is of most value when there is a baseline scan available as part of the machine record. Such a baseline would pinpoint bearing criticals and their amplitudes, as well as indicate—by vibration buildup as operating speed is approached—the proximity of the first rotor mode to the operating speed.

2.1.4 Evaluation of the diagnostic traverses should involve comparing amplitudes with baseline signatures at maximum speed, and as the machine passes through its critical speeds. A significant increase in amplitude at a critical speed implies that an unbalance has developed in the rotor.

It would be expected that the increase in amplitude would occur without a change in the critical speed. If a shift in critical speed has occurred, the cause should be sought in the bearing pedestals or supports. Weakened or broken bearing supports lower the critical speed.

If there is any indication that the rate of buildup of amplitude at operating speed has changed, a shift in the rotor-mode critical speed should be suspected. The most likely cause of this would be a severely cracked shaft If a shaft proximity probe or a shaft rider is installed, a scan of its response should be obtained for confirmation.

In evaluating the data from diagnostic traverses, the more sensitive instrumentation—bearing cap or shaft monitoring—should be used. Only parallel traverses can indicate with certainty which type is more sensitive for a given type of machine.

Table 1 is a guide for interpretation of vibration readings and trends and provides guidelines for action. Bearing vibration is unpredictable. The table presents the most likely causes of a given symptom and suggests the most efficient approaches for investigation.

Table 1. Guidelines for the analysis of bearing vibration trends

A. Abrupt Increase:	0.5 mil (12.5 microns) for units operating at 0.5 mil (12.5 microns) or less; 1.0 mil (25 microns) for units operating over 0.5 mil (12.5 microns); 2.0 mil (50 microns) for units operating over 1.0 mil (25 microns).
Action:	Perform vibration frequency analysis and accel/decel scans.
Diagnosis:	1/rev. vibration: Possible fracture of blade, shroud and/or other rotor part. Possible cracked disk or spindle. A dismantle and internal inspection should be done immediately. Other possible causes are (1) a bowed or bent shaft, and (2) disks loose on the shaft Significant components of 2/rev. or 3/rev.: Possible pedestal looseness; check and correct.
B. Pronounced Increase over a period of 4–5 days:	
	0.5 mil (12.5 microns) for units operating initially at 0.5 mil (12.5 microns) or less; 1.0 mil (25 microns) for units initially operating over 0.5 mil (12.5 microns); 2.0 mil (50 microns) for units initially operating over 1.0 mil (25 microns).
Action:	Perform vibration frequency analysis and accel/decel scans using both bearing-cap and shaft-monitoring instrumentation.
Diagnosis:	1/rev., possibly with components of 2/rev. and 3/rev.: Possible crack progression in shaft Investigate at once.
C. Gradual Increase over a period of 3-6 months:	
	0.75 mil (19 microns) for units operating initially at 0.5 mil (12.5 microns) or less; 1.5 mil (37.5 microns) for units initially operating over 0.5 mil (12.5 microns); 3.0 mil (75 microns) for units initially operating over 1.0 mil (25 microns).
Action:	Perform vibration frequency analysis and accel/decel scans.
Diagnosis:	0.4-0.5/rev. vibration: Possible oil whip in hydrodynamically lubricated bearings. Special investigation necessary; internal bearing clearance or oil viscosity may have changed. 1/rev. vibration: Action depends on object, as follows: *Steam Turbine:* Possible buildup of deposits on blades. Wash turbine with wet steam. Review efficiency records. If cleaning does not work there is a possibility that disks may be loose on the shaft A dismantle and internal inspection should be performed within 3 months. *Gas Turbine:* Possible buildup of deposits on compressor or turbine blades. Review performance to see if location of buildup can be identified. Clean compressor. If cleaning does not reduce the vibration, blade ## erosion may be the cause. Schedule an internal inspection within 3 months. *Dynamic Compressor:* Possible buildup of deposits on blades. Clean compressor rotor. Review efficiency records. If cleaning does not correct the vibration, schedule a dismantle and internal inspection at a convenient time. *Fans and Blowers:* Possible buildup of deposit on blades. Review flow and efficiency records.. Reduction in flow capacity implies deposits. If cleaning equipment is installed, the blades should be cleaned. If efficiency only is affected, excessive erosion may be the cause. An internal inspection should be scheduled to determine the problem and make the necessary repairs. *All Units:* Possible buildup of hard deposits in splined, lubricated coupling, leading to misalignment of coupling. Inspect and clean coupling. *High Frequency:* Rolling-element bearing or gearbox damage, depending on frequency. Inspect and replace bearing or gears, as necessary.

3.0 SUPPORT FOR RECOMMENDATIONS

3.1 Loss History

The following table is a five year loss history where vibration was a factor. The machinery involved was predominantly rotating machinery such as turbines, compressors, fans, pumps, etc. Included in the statistics are losses due to lack of monitoring, inadequate monitoring, failure to follow up when poor conditions were obvious and failure to calibrate equipment.

Year	*No. Losses*
94	8
95	8
96	11
97	7
98	4

4.0 REFERENCES

4.1 FM Global

Data Sheet 5-12, *Electric Generators.*

Data Sheet 5-18/14-15, *Protection of Electrical Equipment.*

Data Sheet 7-100/13-16, *Dynamic Compressors.*

Data Sheet 12-60, *Suction Press Rolls.*

Data Sheet 12-61, *Mechanical Refrigeration.*

Data Sheet 13-3, *Utility Steam Turbine-Generators*.

Data Sheet 13-17, *Gas Turbines.*

Data Sheet 13-26, *Internal Combustion Engines.*

APPENDIX A GLOSSARY OF TERMS

Following are definitions of some of the terms used in connection with vibration monitoring, with brief comments on their significance.

Vibration: an alternating movement of some part of a machine, consisting of repeated uniform cycles of motion from a neutral position to a peak in one direction, back through the neutral position to a peak in the opposite direction, and then back to the neutral position.

Frequency: the number of complete cycles of vibration (as defined) per unit time. The usual units are cycles per second.

Displacement: the peak-to-peak range of motion in each cycle, usually expressed in mils or microns. A mil is one-thousandth of an inch (25.4 microns). Displacement is measured by a displacement sensor, such as an eddy-current proximity probe.

Velocity: as the vibrating part passes through the neutral position; its velocity is a maximum. This velocity is measured by a seismic velocity pickup. The unit of velocity is expressed in in/sec. (m/sec.).

If velocity and frequency are known, displacement can be obtained as follows:

$$\text{Displacement (mils)} = (318.3) \bullet \frac{\text{(Velocity)}}{\text{(Frequency)}^2} \quad \text{(English Units)}$$

$$\text{Displacement (microns)} = (318{,}309) \bullet \frac{\text{(Velocity)}}{\text{(Frequency)}^2} \quad \text{(Metric Units)}$$

Acceleration: as the direction of motion of the vibrating part reverses (at peak motion), the acceleration is a maximum. Acceleration is expressed in g-units (i.e., the number of accelerations of gravity experienced). It is measured by a piezoelectric accelerometer.

One g-unit = 32.17 ft/sec^2 (English Units)
= 9.81 m/sec^2 (Metric Units)

If acceleration (in g-units) and frequency are known, displacement can be determined as follows:

$$\text{Displacement (mils)} \quad (19{,}573) \bullet \frac{\text{(Acceleration)}}{\text{(Frequency)}^2} \quad \text{(English Units)}$$

$$\text{Displacement (microns)} \quad (457{,}158) \bullet \frac{\text{(Acceleration)}}{\text{(Frequency)}^2} \quad \text{(Metric Units)}$$

Amplitude: the magnitude of the vibrating motion, whether expressed as displacement, velocity or acceleration.

APPENDIX B DOCUMENT REVISION HISTORY

Original — Apr. 1985

Revised — Jan. 1988

Revised — Sep. 1998

APPENDIX C DISCUSSION

C.1 Scope

The principal value of vibration monitoring of rotating machinery is in protecting bearings, foundations, oil lines and stationary components from the effects of fatigue due to unbalance. It can also warn that fracture of a rotating component, such as a compressor or turbine blade, has occurred, or that an extensive crack has developed in the shaft or rotor. However, vibration monitoring cannot help prevent the initiation of such fractures or cracks.

This discussion applies to normal operating conditions, not to the extreme conditions that may occur subsequent to rotor damage of some type.

C.2 Value of Vibration Monitoring

1. Vibration instrumentation can help protect the bearings from high alternating loading due to rotor unbalance, angular misalignment of rotor coupling flanges, or rotor bow. This alternating loading could lead to fatigue of the bearings.

2. Vibration instrumentation can help protect components mounted on or attached to the stationary structure (in particular, oil lines to the bearings) from fatigue due to vibration. Vibration is excited by an unbalance in the rotor, or by an angularly misaligned or bowed rotor.

3. Vibration instrumentation can help protect the rotor from radial rubbing, either at blade tips or at the interstage seals or packing, provided the vibration limits are properly established.

4. Vibration instrumentation can detect gear vibration, which may lead to fatigue of gear teeth.

5. Vibration instrumentation can detect deterioration of rolling element bearings and help forestall further disintegration.

6. In some cases, permanently installed vibration instrumentation can produce an alarm, or even trip a machine directly, in time to prevent substantial consequent damage after failure in the rotor. This is not a consistent function, however.

7. Vibration instrumentation can detect oil whip in sliding bearings, and subsynchronous resonance.

8. The phenomena that produce the lower frequency (1, 2, 3 × rotational speed) vibration of bearings are:

a) rotor unbalance due to broken, excessively worn, or loose rotating parts;

b) rotor bow;

c) angular misalignment at couplings;

d) bearing support system looseness; and

e) electrical faults in the rotors of electrical machinery, gear damage or wear.

These phenomena do not produce alternating stresses in the rotors of turbomachinery, and, therefore, do not lead to fatigue of these rotors. They do produce alternating loads on the bearings, however, and this could lead to damage to bearings or stationary components.

C.3 Areas where Vibration Monitoring is of Little or No Value

1. Vibration instrumentation cannot protect against high-frequency fatigue of rotating blading. Bearing vibration is neither a cause nor a concomitant of blade vibration.

2. Vibration instrumentation cannot protect against high-frequency fatigue of wheels or disks. It can, in some cases, warn that a crack is developing in time to prevent consequent damage.

3. Vibration instrumentation cannot protect against high-cycle fatigue of a shaft due to bearing misalignment (a condition that arises when one or more bearings in a train of three or more is not lined up with the others, either due to installation error, or to foundation settlement).

4. Vibration instrumentation cannot protect against the following hazards to rotating machinery:

a) Overspeed;

b) Water induction;

c) Lubrication failure;

d) Shaft fractures;

e) Blade and disk cracking due to high-frequency fatigue;

f) Thrust bearing wiping (not related to lubrication).

High-frequency fatigue failures of rotating components are due to forced vibration of that component. This is almost always due to resonance. The exciting force is due to disturbances in the flowpath caused by some stationary geometric feature. The periodicity of the disturbance is generated by the rotation of the rotor. Vibration instrumentation mounted on the stationary structure can see only the stationary disturbance at best; it can detect the vibration only if it is mounted on the vibrating component, i.e., the rotor.

A similar argument holds for high-cycle fatigue of shafts due to bearing misalignment. The loads on the bearings resulting from this type of misalignment are stationary in space. Therefore vibration instrumentation on a bearing cap can sense no alternating forces. The alternating stresses in the shaft as it rotates into and out of the stationary fields can be picked up only by a strain gage mounted on the shaft

C.4 Vibration Monitoring

Vibration monitoring may be either continuous or intermittent. In the former case, accelerometers, velocity pickups, or eddy-current proximitors are permanently installed on a machine (usually at the bearings), and connected to a continuous readout monitor or to a plotting device. An alarm may be annunciated when the vibration reaches a preset level. In some cases, protective circuitry can trip the machine at a higher, dangerous vibration level.

Intermittent monitoring usually involves hand-held vibration instruments. These may vary from a simple vibration meter, which merely indicates the overall vibration level, to a vibration spectrum analyzer, which can analyze the signal, separating it into individual frequencies and corresponding amplitudes. Intermittent monitoring should be performed at intervals sufficiently frequent that a sudden change in vibration at a bearing can be detected in a reasonable time after its occurrence.

Vibration monitoring addresses changes in vibration level rather than level itself (unless the latter has reached a dangerous level). A sudden change in vibration of as low as 0.5 mil (13 microns), or a gradual change of 1 mil (25 microns) over a period of six months should warrant a vibration analysis. Evaluations of changes in vibration level should be performed at similar operating conditions, whenever possible.

C.5 Critical Speeds

Following is a fundamental definition of critical speed: A critical speed is one of a set of discrete rotational speeds at which the rotor of a turbomachine can whirl under the action of centrifugal loads due to unbalance. This is a somewhat abstruse definition, and it may be illustrated as shown in Figure 1.

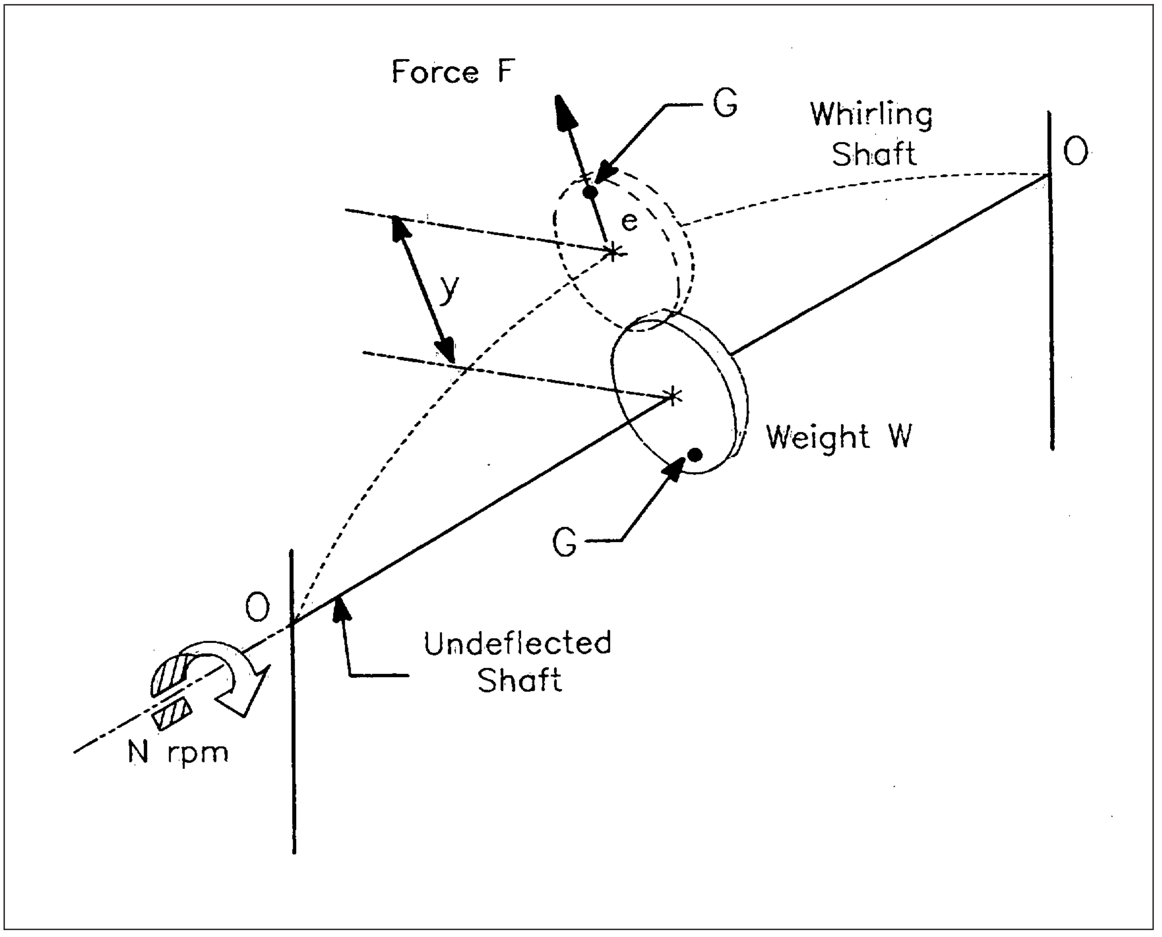

Fig. 1. Whirling of a rotor with a single unbalanced mass.

In Figure 1 the center of gravity G of the mass is eccentric with respect to the centerline of the bearings (0-0) by an amount e. The product w•e is the unbalance U. As the rotor is accelerated from rest, the mass exerts a centrifugal force on the shaft equal to

$$F = K \times N^2 \times w \times e = K \times N^2 \times U$$

where N is the instantaneous rotational speed, and K is a constant which makes the dimensions consistent, whether in English or in metric units.

The force F is proportional to the square of the rotational speed N. As N is increased the deflection (or whirl y) of the shaft builds up to a very high value; after a certain speed is passed, the whirl dies out again. The speed at which the whirl is a maximum is the critical speed. In the general case a rotor has many critical speeds; whirl can only occur at such a speed (or near it as the whirl builds up or down in the vicinity of a critical).

Each critical speed of a rotating machine has a distinct mode shape. Normally, only two or three are of interest. There are one or two criticals below the maximum operating speed, and the mode shapes of these are characterized by a relatively undeflected rotor whirling on flexible bearings (Fig. 2(a)). These are known as ''bearing'' modes (or ''rigid-rotor'' modes).

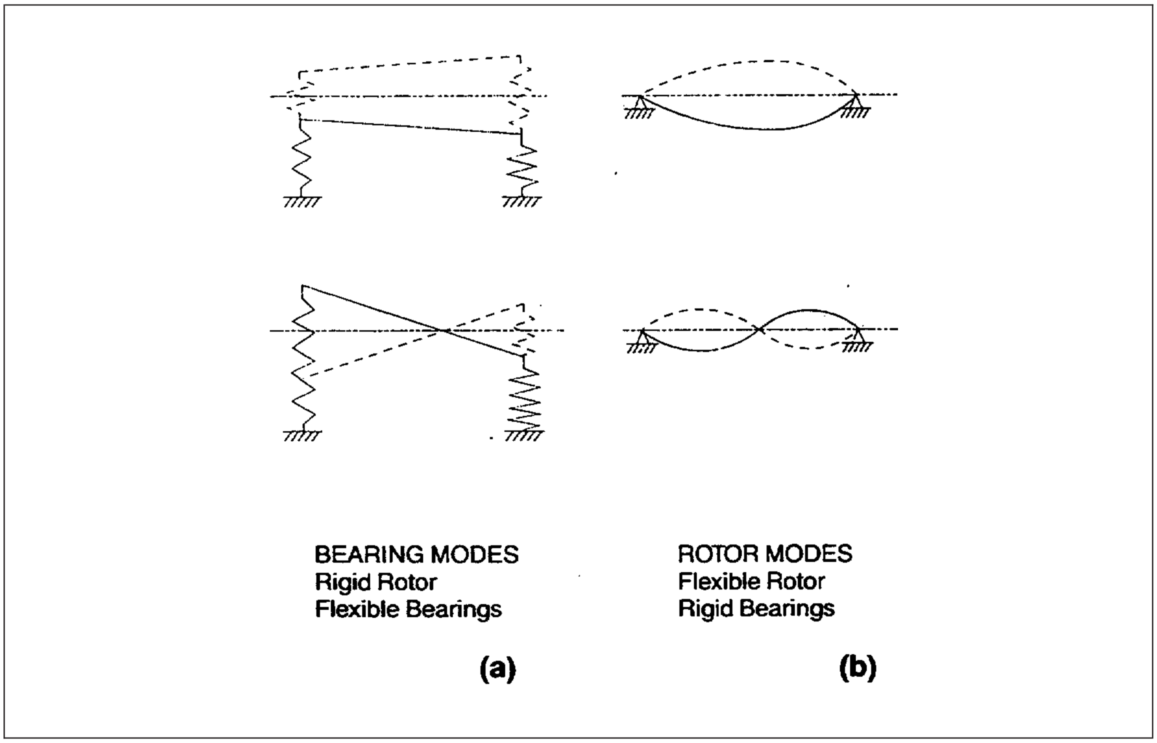

Fig. 2. Typical mode shapes.

Their characteristics are governed by the stiffnesses (spring rates) of the bearing pedestals. Since these stiffnesses usually differ in vertical and horizontal directions, the critical speeds and their sensitivities to unbalance will also be different in the two directions. The sensitivity of a pickup to unbalance will then depend on whether it is mounted vertically or horizontally.

A critical speed, a small percentage above the maximum operating speed, may also be of interest, since it may produce some near-resonant vibration in the operating speed range. This mode is characterized by a deflected rotor whirling on rigid bearings, i.e., with relatively little deflection of the bearing supports (Fig. 2(b)). This is known as a ''rotor'' mode, and its characteristics are governed by the stiffness of the rotor.

Some machines, such as some small pumps and expanders, are designed with flexible rotors. Their rotor modes may be within the operating range. Such machines require very careful balancing if they are to run smoothly. In some cases ''high-speed'' balancing must be used.

C.6 Type of Measurement

The type of vibration pickup that should be used depends on the particular information desired. Velocity pickups or accelerometers mounted on the bearing caps are usually adequate for the detection of rotor unbalance (which may be due to broken blades or rotor parts), rotor bow, angular misalignment at couplings, bearing pedestal looseness, or electrical unbalance.

Velocity pickups are very sensitive over the range of frequencies of interest in critical speed analysis. However, they have moving parts which may bind or fail after an extended period of operation in a permanent installation mode. Accelerometers are more durable, and, because they have a wider frequency of response, they are more suitable for installations having gearboxes and/or rolling element bearings, which generally produce high-frequency vibration.

Many steam turbines, particularly the larger units, are equipped with shaft-riders or eddy-current proximitors. These are designed to measure the motion of the shaft at the bearing relative to the bearing cap or housing. This particular measurement may not be responsive to unbalance in the rotor, since the spring-rate of the oil film between the journal and bearing may be very high relative to the bearing pedestal spring-rate, and the shaft may simply move along with the pedestals. In one notable case in which rotor blades and shrouds had fractured, a hand-held vibrometer proved to be much more sensitive than the installed proximity probe. Therefore, it is advisable, where such instrumentation is installed, to complement the automatic monitoring by intermittent monitoring of bearing-cap vibration.

The principal value of such shaft-riders and proximitors may be in detecting shaft cracking before complete fracture occurs. When a crack has progressed extensively through a shaft, the latter's stiffness is reduced. This should result in a lowering of the rotor mode to bring it closer to, or into, the operating range. This would be accompanied by an increased whirling of the shaft at operating speed, which should be most detectable as a movement relative to the bearing. Since unbalance would not have changed, the bearing modes would not be affected, so bearing-cap pickups would detect little, if any, change.

C.7 Axial Vibration

Axial vibration pickups are sometimes mounted at thrust bearings. They may pick up axial hunting of the rotor due to some performance instability such as compressor stall, or due to cavitation, in the case of pumps. The principal value of such pickups may be in support of the radial and horizontal instrumentation in identifying rotor bow or angular misalignment in cases where the thrust bearing happens to be in a location where the slope of the whirling shaft is high. Axial vibration of the thrust bearing sometimes supplements $2 \times$ rpm and $3 \times$ rpm vibration of the journal bearings in these cases.

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