2.3.3 CENTRIFUGAL PUMP MECHANICAL PERFORMANCE, INSTRUMENTATION, AND DIAGNOSTICS

CHARLES JACKSON
JAMES H. INGRAM

MECHANICAL PERFORMANCE

The mechanical performance of a pump would imply only the rotating mechanical masses, with no consideration given to hydraulic (process) effects. The rotating masses (impellers, sleeves, nuts, coupling, bearings, seals, and so on) can be examined as pure mechanics. A person concerned with mechanical performance should be intimately familiar with pump design, construction, and maintenance to be successful.

In discussing the mechanical performance of centrifugal pumps, two examples will be used. The first will be a horizontal, 500-hp (373-kW), single-stage (overhung impeller) American Petroleum Institute (API) process pump. The second will be a six-stage, horizontal, 1000-hp (746-kW), multistage boiler-feed pump.

Normally, the rotor dynamics will involve (a) a review of the shaft stiffness of the bearings and structure, (b) a mass model of the rotor, and (c) a critical speed analysis with mode shapes of the rotor or shaft.

SINGLE STAGE PUMP

An $8 \times 6 \times 13$ pump is operating on water at 3550 rpm with a design flow of 2500 gpm (567 m^3/h) at 600 ft (183 m) total head, 1.0 sp. gr., requiring approximately 500 hp (373 kW). The pump operated extremely rough, and the bearings and bearing housing failed. The impeller weighs 61.4 lb (27.9 kg).

If the impeller is fitted on the shaft with an eccentricity of 0.002 in (0.051 mm), a calculated centrifugal force *Fe* of 44 lb (196 N) would cause a deflection of 0.0026 in (0.066 mm) from

$$y = wl^3/3EI$$

where w = weight (force) of impeller, lb (N)
l = length of overhang, in (m)
E = modules of elasticity, lb/in^2 (N/m^2 or Pa)
I = shaft section moment of inertia, in^4 (mm^4)

This pump has a 5212 line bearing and tandem mounted (DB) 7311DB angular contact thrust bearings (40° contact angle). An extremely loose fit of the radial bearing in the bearing housing could cause the outer race to spin, which could cause a vibration equal to twice the rotation frequency. Interference fitting could lead to radial bearings' accepting thrust (for which many are not designed) from thermal expansion of the shaft or from the thrust bearing.

Frequencies Generated The following data and definitions are needed to compute the frequencies generated by defective bearings[1]:

rpm = revolutions per minute
rps = revolutions per second
FTF = fundamental train frequency, Hz
BPF_I = ball passing frequency of inner race, Hz
BPF_O = ball passing frequency of outer race, Hz
BSF = ball spin frequency, Hz
Bd = ball or roller diameter, in (mm)
Nb = number of balls or rollers
Pd = pitch diameter, in (mm)
Ø = contact angle

The formulas are

$$\text{rps} = \frac{\text{rpm}}{60}$$

$$FTF = \frac{\text{rps}}{2}\left(1 - \frac{Bd}{Pd}\cos Ø\right)$$

$$BPF_1 = \left(\frac{Nb}{2}\,\text{rps}\right)\left(1 + \frac{Bd}{Pd}\cos Ø\right)$$

$$BPF_O = \left(\frac{Nb}{2}\,\text{rps}\right)\left(1 - \frac{Bd}{Pd}\cos Ø\right)$$

$$BSF = \left(\frac{Pd}{2Bd}\,\text{rps}\right)\left[1 - \left(\frac{Bd}{Pd}\right)^2\cos^2 Ø\right]$$

The pitch diameter is the diameter measured across the bearing from ball or roller center to ball or roller center. The contact angle is measured from a line perpendicular to the shaft to the point at which the balls or rollers contact the race. The contact angle of a deep groove ball bearing is zero.

It is necessary to distinguish between the ball frequency and the impeller vane passing frequency, which is 17,750 cpm (5 vanes × 3550 rpm × 1 casing cutwater) for this example. The mode shape of the pump shaft is conical (pivotal) in the first mode of a cantilevered shaft mount. The stiffness map of the rotor looks like that shown in Figure 1.

This pump has two design faults, as can be seen from the stiffness map. The first is that an excessively large coupling is used. This heavy overhung mass at the coupling forces the first shaft resonance to be very near the pump operating speed. Normally, the shaft in this type of pump is considered to be "rigid;" that is, operating safely below the first undamped shaft resonance. In this case, the pump is affected by two negatively additive errors. The

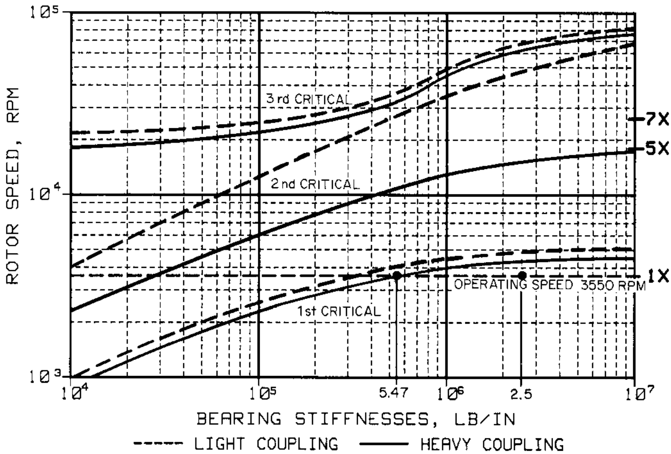


FIGURE 1 Undamped critical speed map of single-stage overhung pump, comparing normal and heavy coupling weights (lb/in × 175 = N/m)

TABLE 1 Logic of spring equivalent stiffness K_e

SPRING	ELECTRICAL RESISTOR EQUIV ANALOG	EXAMPLES OF VARIATIONS (Ke)		
m Kb Ks	Rb Rs	Kb 1 X 10^5 Ks 1 X 10^6	Kb 1 X 10^5 Ks 5 X 10^5	Kb 1 X 10^5 Ks 1 X 10^5
$1/Ke = \frac{1}{Kb} + \frac{1}{Ks}$	$1/Re = \frac{1}{Rb} + \frac{1}{Rs}$	Ke = 91k lb/in	Ke = 83k lb/in	Ke = 50k lb/in

Kb = STIFFNESS OF BEARING, lb/in (N/m) N/m = 175 X lb/in
Ks = STIFFNESS OF SUPPORT, lb/in (N/m)
Ke = EQUIVALENT STIFFNESS

NOTE: SPRINGS IN PARALLEL SIMPLY ADD, AS DOES ELECTRICAL RESISTANCE IN SERIES: Ke = Kb + Ks

heavy mass coupling effect is compounded by a weak baseplate that is not properly grouted, leaving a void under the pump supports. In rotor (shaft) supports, two spring supports in series reciprocally add, similar to electric resistors in parallel (Table 1).

The effective stiffness is

$$K_e = \frac{1}{1/K_b + 1/K_s}$$

If the bearing stiffness K_b, is 2.5 × 10^6 lb/in (4.4 × 10^6 N/m) and the support stiffness K_s, is low; that is, 7.0 × 10^5 lb/in (1.2 × 10^8 N/m), the effective stiffness is 5.47 × 10^5 lb/in (9.57 × 10^7 N/m), which moves the first mode resonance from 4300 cpm to 3550 cpm, which is the running speed of the pump.

The mode shapes of the rotor are shown in Figures 3 and 4. An animated display is used to better show the rotor gyrations in synchronous whirl. The first modes are shown with a

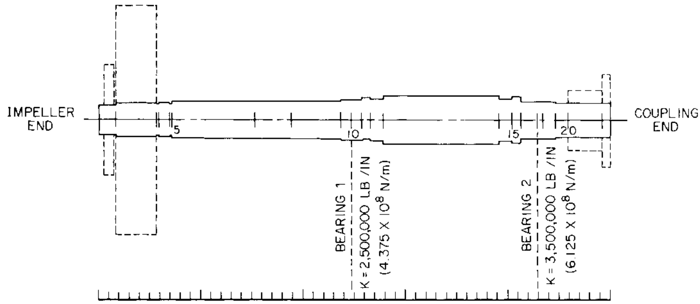


FIGURE 2 Rotor cross section of single-stage overhung pump with normal coupling weight. Rotor weight = 110.4 lb (50kg); rotor length = 30.3 in (77.0 cm); number of stations = 22; number of bearings = 2

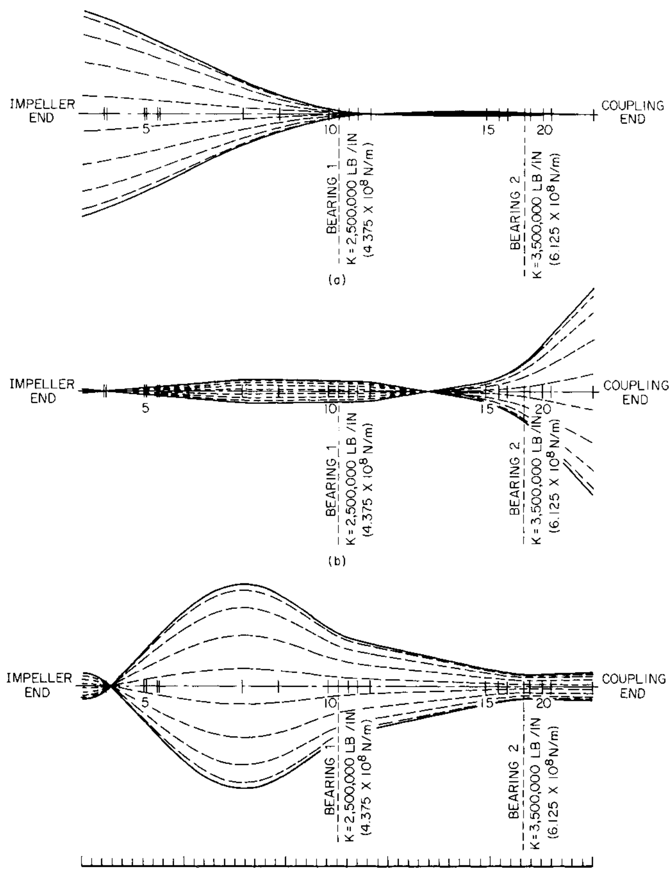
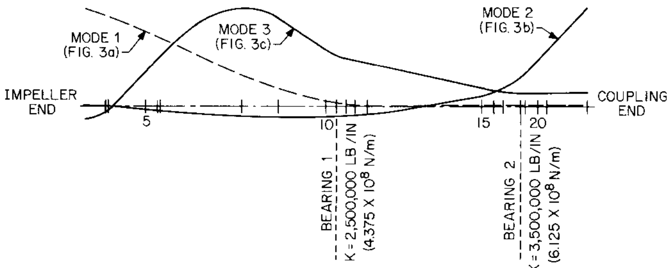


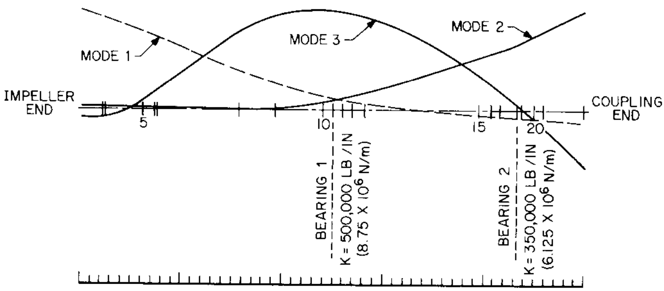
FIGURE 3 First, second, and third resonate animated mode shapes of single-stage overhung pump with light coupling of 15 lb (6.8 kg) and rigid foundation. Rotor weight = 110.4 lb (50 kg); rotor length = 30.3 in (77.0 cm); number of stations = 22; number of bearings = 2. (a) Mode 1: frequency = 4717 cpm; (b) mode 2: frequency = 53,482 cpm; (c) mode 3: frequency = 67,522 cpm

No. UNITS	CRITICAL SPEED rpm	(Hz)	WMODE, lb	ITMODE, lb/in^2	KMODE, lb/in	DIM. STRAIN ENERGY USHAFT	UBEARING
1	4717	(79)	57.0	1.84E-01	3.60E+04	91	9
2	53482	(891)	6.0	3.59E-01	4.90E+05	36	64
3	67522	(1125)	9.8	6.42E-01	1.27E+06	42	58



(a)

No. UNITS	CRITICAL SPEED rpm	(Hz)	WMODE, lb	ITMODE, lb/in^2	KMODE, lb/in	DIM. STRAIN ENERGY USHAFT	UBEARING
1	4085	(68)	58.0	1.81E-01	2.75E+04	59	41
2	21251	(354)	13.8	3.50E-01	1.76E+05	10	90
3	40036	(667)	17.1	1.34E-00	7.78E+05	37	63



(b)

FIGURE 4 Synchronous critical speed summary with first three mode shapes of a single-stage overhung pump with light coupling of 15 lb (6.8 kg) and rigid or flexible foundation. Rotor weight = 110.4 lb (50 kg); rotor length = 30.3 in (77.0 cm); number of stations = 22; number of bearings = 2. (a) Rigid support, modes 1, 2, and 3 respectively: 4717, 58,482, and 67,522 rpm; (b) flexible support, modes 1, 2, and 3 respectively: 4085, 21,251, and 40,036 rpm. (lb $\times$ 0.454 = kg; lb/in $\times$ 175 = N/m)

lighter, normal, and correct coupling (15 lb; 6.8 kg). Figure 2 shows the mathematical model of this pump with the impeller at station 2, the radial bearing at station 11 at 2.5 $\times$ 10^6 lb/in (4.4 $\times$ 10^8 N/m) stiffness, the outboard thrust/radial bearing at station 18 at 3.5 $\times$ 10^6 lb/in (6.1 $\times$ 10^8 N/m) and the coupling at station 21.

Figure 3a, obtained from a finite element computer analysis of the mathematical model in Figure 1, shows the first mode with a rotor weight of 110.4 lb (50 kg), a rotor length of 30.3 in (77.0 cm), and a first mode undamped resonance (critical) of 4717 cpm. This is a pivotal mode, with 100% of the normalized motion at the impeller end. Any motion at the antifriction bearings is greatly restrained.

Figure 3b shows the second resonance mode, at 53,482 cpm, which is not to be encountered. The coupling motion is now the greatest motion.

Figure 3c shows the third mode, at 67,522 cpm.

Figure 4a shows an overlay of all three modes with a summary of the criticals, the modal mass, and the relative strain energy (91) in the shaft at station 10 (impeller side of radial bearing). A lesser strain energy is at the radial bearing (station 11).

Figure 4b summarizes what happens to critical speed modes if either a more *flexible* bearing or a soft structure is provided intentionally or unintentionally. Also note that the criticals are lowered significantly and the strain energy is transferred more from the shaft into the bearings; that is, strain values under the *U-shaft* column are less than under *U-bearing* column. The first critical is 4085 cpm at a pump speed of 3550 rpm (+15%). A 15% margin of separation may be close enough to excite (cause a rise in vibration) the rotor if the resonance response envelope is too wide. However, this is unlikely on antifriction bearings (spiky/narrow response), but possible on sleeve bearings (low/broad response).

Figure 5a is a summary which shows the response of a rigid support and an *excessively heavy* (62 lb = 28 kg) coupling, which is as heavy as the impeller. Note that the first mode is again only slightly above the operating speed; that is, 4279 cpm compared with 3550 (+21%). The bearing stiffness is assumed to be the controlling stiffness. Many assume that the structure or base stiffness is one order above the bearing stiffness ($K_s = 10K_b$). This assumption that the bearing stiffness is the controlling stiffness variable is often a very poor assumption. The larger the pump size, the more this is true. That is why an 8 × 6 × 13 pump was used as an example.

Further, the second mode, at 15,865 cpm, is in an area where the blade passing frequency (5 × 3550 = 17,750 cpm) can easily excite this mode, given little variation in support stiffness. Figure 5*b* is a summary sheet that best illustrates the problem:

- The baseplate was improperly installed and grouted.
- The elastomeric coupling designed for low-duty, low-speed, and torsional damping was *too* heavy; that is, too much overhung weight.

Note that the first critical is in sympathy with the pump operating speed, which becomes intolerable with the operating time *limited* to one to two days, due to bearing failures.

The stiffness on antifriction bearings was determined from a program written by M. E. Leader of Monsanto, using values projected by an article written by F. F. Garguilo, DuPont.[2] The correction consisted of converting the 62-lb (28-kg) coupling to a 15-lb (6.8-kg) series dry flex disk-type coupling and stiffening the support by flushing the baseplate cavity with a degreasing fluid and pressure injection of epoxy to fill the baseplate voids.

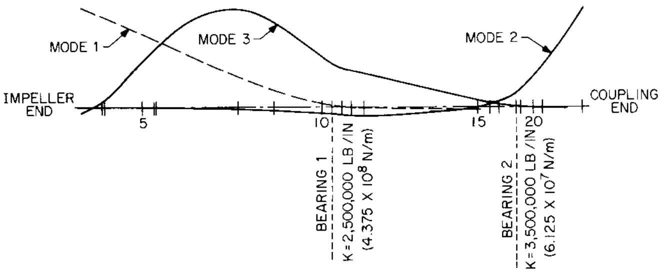
It should be remembered that the blade passing frequencies will normally be the strongest exciting force. On this pump, the frequency is five times running speed (five vanes times each cutwater). Because there are two cutwaters, there can also be a frequency at 10 times running speed. The 5× frequency is shown on Figure 1. Also, this 5× frequency excitation could excite the second mode because the second mode critical could fall anywhere between the solid and dashed lines, depending on baseplate stiffness.

The instruments used in diagnosing this problem were force-effective seismic sensors (velocity or piezoelectric accelerometers). They are preferred for pumps, particularly those with antifriction bearings.

MULTISTAGE PUMP EXAMPLE

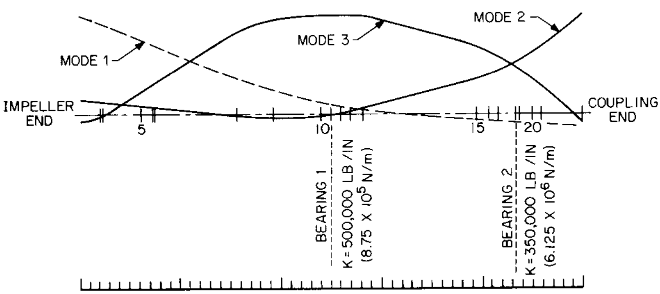
To show the mechanical rotor variations, a six-stage boiler-feed pump with a design capacity of 1250 gpm (284 m^3/h), 2200 ft (670 m) total head, and driven by a 1000-hp (746-kW), two-pole motor has been selected. This pump utilizes interstage bushings as support bearings to the rotor. The contribution of these bushings as bearings will probably be less than might be assumed.

No. UNITS	CRITICAL SPEED rpm	(Hz)	WMODE, lb	ITMODE, lb/in^2	KMODE, lb/in	DIM. STRAIN ENERGY USHAFT	UBEARING
1	4279	(71)	65.2	1.80E-01	3.39E+04	91	9
2	15865	(264)	53.3	3.28E-01	3.81E+05	73	27
3	64970	(1083)	8.9	6.54E-01	1.07E+06	51	49



(a)

No. UNITS	CRITICAL SPEED rpm	(Hz)	WMODE, lb	ITMODE, lb/in^2	KMODE, lb/in	DIM. STRAIN ENERGY USHAFT	UBEARING
1	3580	(60)	66.6	1.79E-01	2.43E+04	61	39
2	9339	(156)	59.6	3.47E-01	1.48E+05	35	65
3	35365	(589)	22.0	7.01E-01	7.82E+05	25	75



(b)

FIGURE 5 Synchronous critical speed summary with first three mode shapes of a single-stage overhung pump with heavy coupling of 62 lb (28 kg) and rigid foundation. Rotor weight = 157.4 lb (71.4 kg); rotor length = 30.3 in (77.0 cm); number of stations = 22; number of bearings = 2. (a) Rigid support, modes 1, 2, and 3 respectively: 4279, 15,865, and 64,970 cpm; (b) flexible support, modes 1, 2, and 3 respectively; 3580, 9339, and 35,565 cpm. (lb × 0.454 = kg; lb/in × 175 = N/m)

Pressure or seal leakage control bushings contribute rotor support if they are long. The hot feedwater has very low viscosity and little damping. The bearing stiffness will be relative to the eccentricity ratio of the shaft in the bushings. An eccentricity ratio of unity (maximum) implies that the shaft is rubbing directly on its bushing.

The impeller weight is increased by the water trapped in each impeller. Many pump manufacturers improperly list pump undamped critical speeds from *dry* pump data or calculations. The bushings, labyrinths, and wear rings all contribute to the actual critical speed. Also, bearing housing resonances are more common than expected.

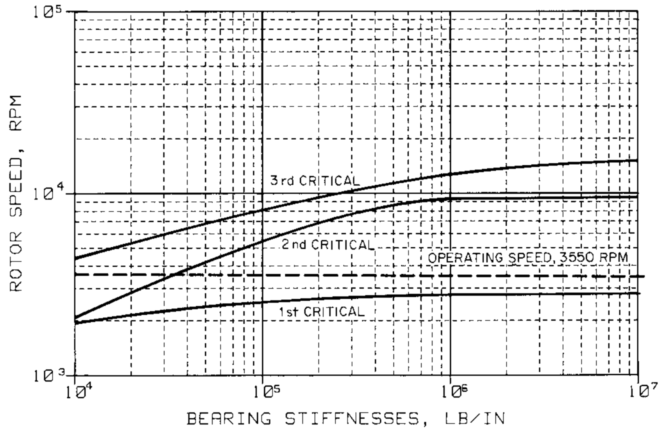


FIGURE 6 Undamped critical speed map of multistage boiler-feed pump with plain journal bearings (lb/in × 175 = N/m)

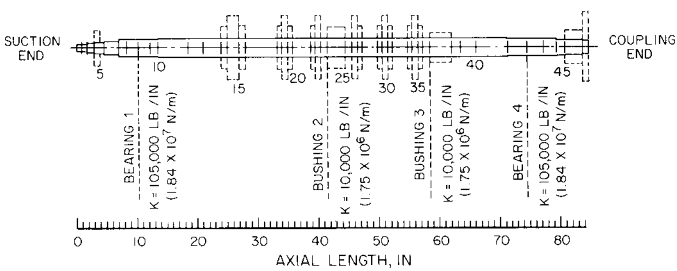


FIGURE 7 Rotor cross-section of multistage high-pressure boiler-feed pump with plain journal bearings. Rotor weight = 377.7 lb (171.3 kg), rotor length = 84.6 in (215 cm); number of stations = 47, number of bearings or bushings = 4 (in × 2.54 = cm)

Figures 6 to 9 illustrate it in the same fashion as the previous example the design audit of a steam-turbine-driven, $4 \times 8 \times 10\frac{1}{2}$, six-stage, boiler-feed pump using hydrodynamic radial and thrust bearings. No problems were experienced with this pump.

A more complete listing of data from analysis is shown with an added breakdown of the rotor model and the output of the first, second, and third resonant modes. The first critical (rotor resonance) is now a cylindrical mode and not a conical mode, as previously seen for the overhung impeller of the single-stage process pump.

ALIGNMENT OF PUMPS AND DRIVERS

Outside of serious unbalance of pump components, there is no single contributor of poor mechanical performance more significant than poor alignment. Incorrect alignment between a pump and its driver can cause

- Extreme heat in couplings
- Extreme wear in gear couplings and fatigue in dry element couplings

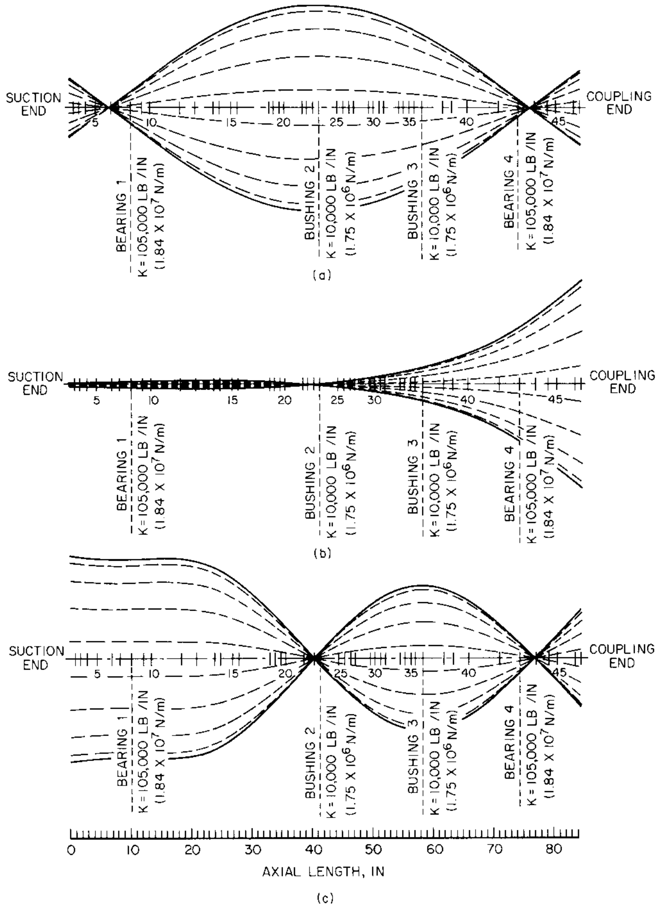


FIGURE 8 First, second, and third resonate animated mode shapes of multistage high-pressure boiler-feed pump with plain journal bearings. Rotor weight = 377.7 lb (171.3 kg); rotor length = 84.6 in (215 cm); number of stations = 47; number of bearings or bushings = 4. (a) Mode 1: frequency = 2614 cpm; (b) mode 2: frequency = 5223 cpm; (c) modes: frequency = 8134 rpm. (in × 2.54 = cm)

- Cracked shafts and totally failed shafts, with failure due to reverse bending fatigue transverse to the shaft axis initiating at the change of section between the large end of the coupling hub taper and the shaft
- Preload on bearings (evident by an elliptical and flattened orbit resembling a deflated beach ball); pure asymmetry of vertical and horizontal vibration can be misleading because the bearing spring constants could vary greatly in the k_{yy} (vertical) and the k_{xx} (horizontal) axis.
- Bearing failures plus thrust transmission through the coupling, which can be totally locked (axial vibration checks across the coupling; that is, at each adjacent machine, will generally confirm this condition)

Significant changes in the cold nonrunning alignment of a pump and driver can take place if the temperature rise in each machine is different and if the piping imposes forces on the pump.

No. UNITS	CRITICAL SPEED rpm	(Hz)	WMODE, lb	ITMODE, lb/in^2	KMODE, lb/in	DIM. STRAIN ENERGY USHAFT	UBEARING
1	2614	(44)	200.6	8.23E-01	3.89E+04	51	49
2	5223	(87)	51.2	2.40E-01	3.96E+04	17	83
3	8134	(134)	127.8	1.38E-00	2.40E+05	46	54

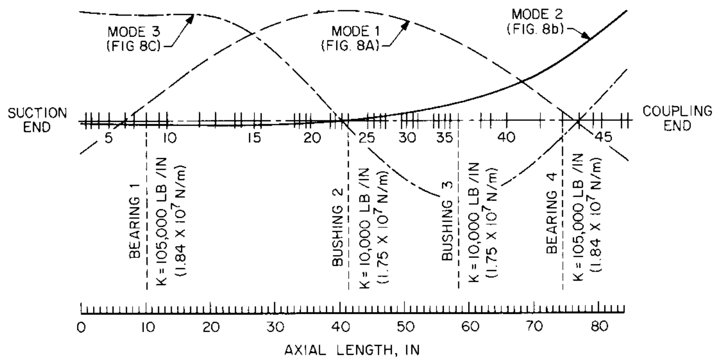


FIGURE 9 First three critical speed mode shapes of multistage high-speed boiler-feed pump superimposed (in × 2.54 = cm)

Therefore, alignment under actual operating conditions must be predicted or, if unknown, confirmed by instrumentation. In either case, an allowance must be made in the initial cold alignment to compensate for changes in alignment from cold idle to hot running.

There are several techniques for measuring cold and hot alignment. The cold alignment is generally measured by either face and rim (Figure 10) or reverse dial indicator (Figure 11) methods.

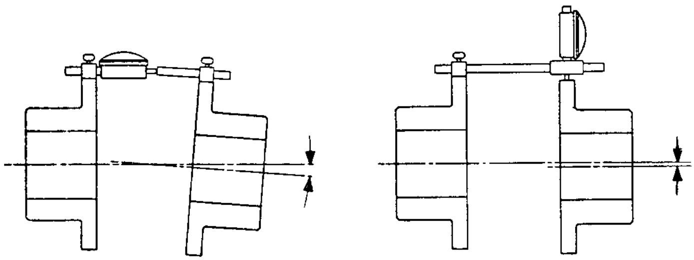
The face and rim method has a sensitivity advantage when the diameter of a coupling exceeds the indicator span of reverse indicator bracket tooling. This is rare, as the pump will generally have a spacer coupling and the reach of the reverse indicators can be increased by clamping onto the shaft behind each coupling half. The face and rim method would also have an advantage if either the driver or the gear could not be rotated, as it seems unlikely that the pump could not be rotated. In order to compensate for the measuring surface's not being circular or smooth, both shafts should be rotated together when using this method.

Disadvantages of Face and Rim Method

1. Diameters of the rim must be true (circular) and smooth and the face reading surface must be flat and smooth, unless both shafts are rotated together.
2. The driver and pump cannot float axially while a reading is being taken or an error will be introduced into the face (angular) reading. (A fixed axial stop will assist in reducing errors.)

Reverse Dial Procedure for Measuring Alignment (Hot or Cold) Several procedures have been suggested by various people to estimate or actually measure alignment while a pump is running at operating temperature. Some techniques are

1. Shutdown after temperatures have stabilized for "hot check" by dial indicators
2. Optical measurements cold to hot (A. J. Campbell, Compressor Engineering Corp., Houston)
3. Dodd bars (DynAlign) technique (B. Dodd, Chevron[3])



Check for Angular Misalignment Dial indicator measures maximum longitudinal variation in hub spacing through 360° rotation.

1. Attach dial indicator to hub, as with a hose clamp; rotate 360° to locate point of minimum reading on dial; and then rotate body or face of indicator so the zero reading lines up with pointer.
2. Rotate both half couplings together 360°. Watch indicator for misalignment reading.
3. Driver and driven units will be lined up when dial indicator reading comes within maximum allowable variation for that coupling style. Refer to specific installation instruction sheet for the coupling being installed. Note: If both shafts cannot be rotated together, connect dial indicator to the shaft that is rotated.

Check for Parallel Misalignment Dial indicator measures displacement of one shaft center line from the other.

4. Reset pointer to zero and repeat operations 1 and 2 when either driven unit or driver is moved during aligning trials.
5. Check for parallel misalignment as shown. Move or shim units so parallel misalignment is brought within the maximum allowable variations for the coupling style.
6. Rotate couplings several revolutions to make sure no "end-wise creep" in connected shafts is measured.
7. Tighten all lockouts or capscrews.
8. Recheck and tighten all locknuts or capscrews after several hours of operation.

FIGURE 10 Face and rim dial indicator method (Courtesy Rexnord)

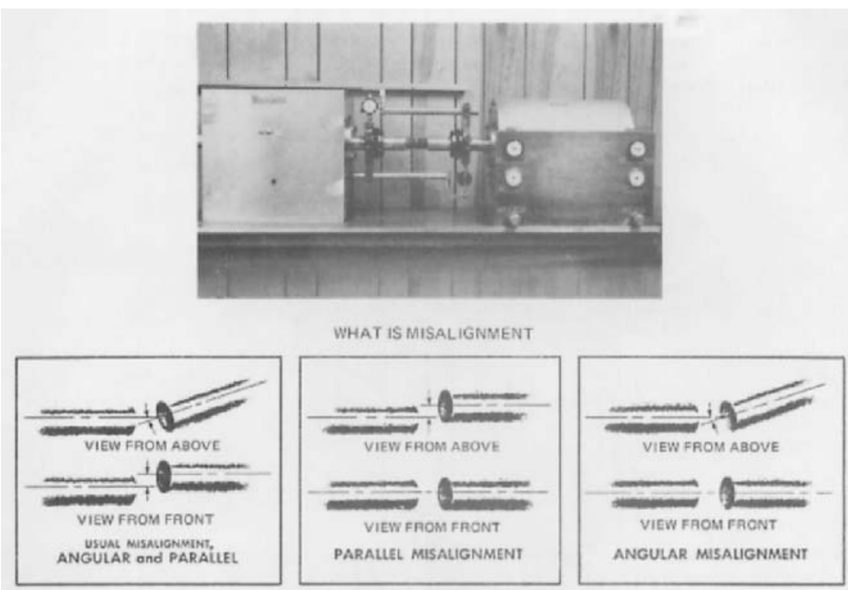


FIGURE 11 Model used for training machinist in the use of reverse dial indicator technique for alignment of machine shafts. Misalignment can be measured as parallel or angular offset.

4. Acculign bench mark gauges (I. Essinger, Shell Oil)
5. Water-cooled probe stands (C. Jackson, Monsanto)
6. Instrumented coupling (for example, Indikon)

Refer to Reference 4 for more specific information on the above techniques.

After measurements have been made and actual misalignment is determined for the hot running condition of the pump, the next step is to calculate the alignment correction required between machines to bring them into alignment during operation. Trial-and-error methods should be discouraged. Plotting actual shaft positions to scale allows one to graphically measure the required cold alignment corrections. Other methods—for example, those that can best be carried out with the aid of a programmable calculator—quickly and accurately calculate vertical, horizontal. inboard or outboard changes in machine positions to accomplish the correct cold alignment. See Reference 11 for a calculator program for both reverse dial and face and rim methods.

Offered here are a graphical procedure and an example for obtaining the desired alignment between a pump and driver for a case where the thermal growth has been estimated and a calculated cold alignment is desired to achieve the final alignment during operation.

The example is illustrated in Figure 12. Consider a steam-turbine-driven boiler-feed pump that has the following heat rise predictions by the manufacturers: steam end 0.002 in (0.051 mm), exhaust end 0.012 in (0.305 mm), pump inboard support 0.006 in

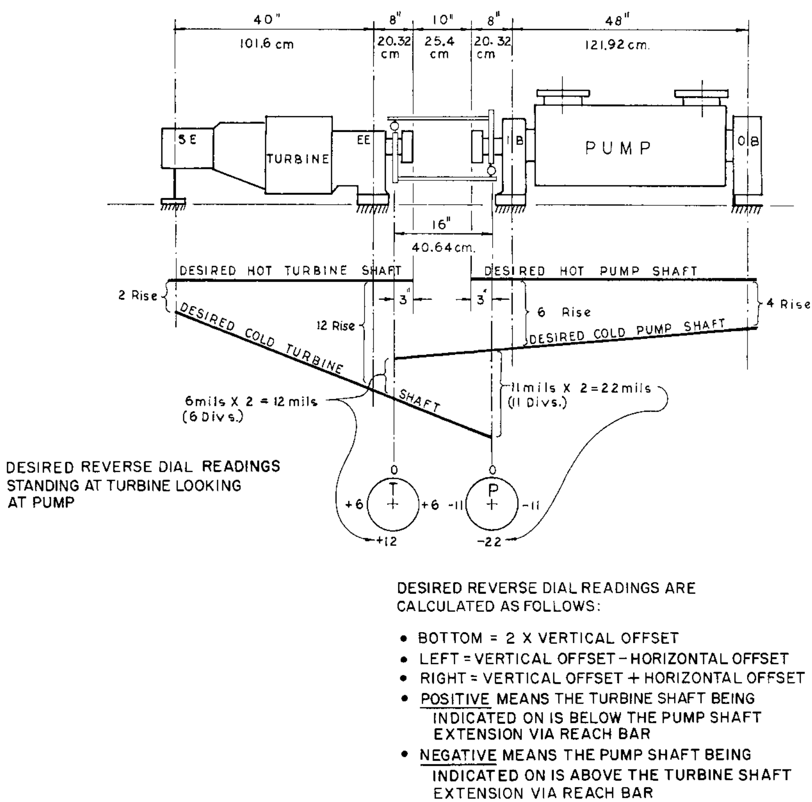


FIGURE 12 Alignment example of a turbine-driven boiler-feed pump with heat-rise data from the manufacturers plotted to calculate the desired cold alignment

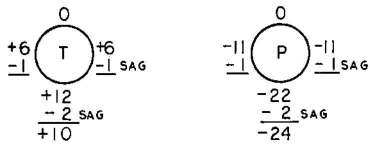


FIGURE 13 Desired dial indicator readings corrected for indicator bar sag

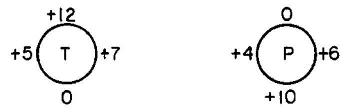


FIGURE 14 Actual alignment readings

FIGURE 15 Dial readings corrected to position zero at top

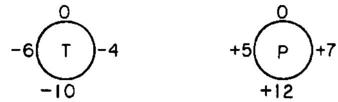


FIGURE 16 Actual alignment readings corrected for indicator bar sag

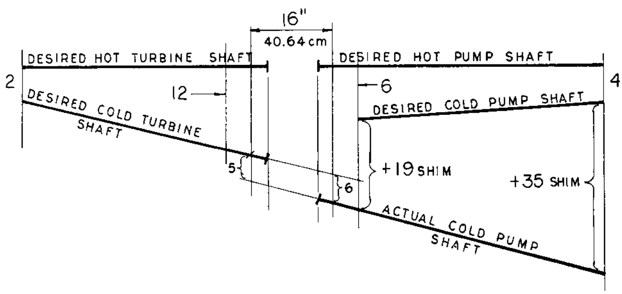


FIGURE 17 Plot of absolute vertical shaft positions. It can be seen from the graph that a 0.019-in (0.483-mm) shim is required to raise the inboard end of the pump and a 0.035-in (0.889-mm) shim is required to raise the outboard end of the pump to compensate properly for thermal growth

(0.152 mm), and outboard bearing 0.004 in (0.102 mm). The horizontal length is laid on a graph with 1 div. = 1 in (25.4 mm). The vertical movement plots are laid out with 1 div. = 0.001 in (1 mil; 0.0254 mm).

Based on 0.001-in (0.0254-mm) sag (see Figure 19), the field readings needed to meet the above absolute requirements are shown in Figure 13.

The actual cold alignment is checked, and the reverse dial readings are as recorded in Figure 14. There are 0.125-in (3.175-mm) shims under all support feet. It is decided to move the pump rather than the turbine, and therefore the correct cold position of the pump must be calculated.

To correct the turbine dial readings to position zero at the top, simply add −12 to all four turbine readings (Figure 15). To correct for sag (see Figure 19 for explanation), *subtract* 1 from the left and right readings and 2 from the bottom reading (Figure 16). Finally, plot the absolute shaft positions on graph paper, leaving the turbine "in place," so to speak, thereby determining two points across the 16-in (40.64-cm) indicator span to define *where* the pump shaft lies with *respect to* the turbine (Figure 17).

To plot the horizontal corrections, reduce the final horizontal readings *only* to zero on the least numerical reading, by adding 4 to the turbine readings and −5 to the pump reading, as shown in Figure 18.

Dial indicator readings can be in metric units and the scale in centimeters rather than inches. A scale of between 500:1 and 1000:1 is suggested. A 1000:1 scale is in use here (horizontal scale equals 1000 times vertical scale).

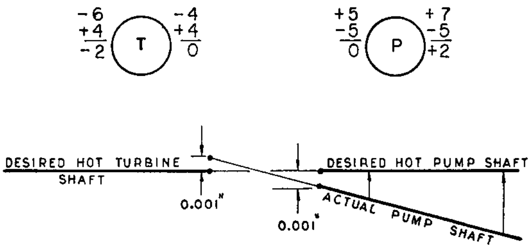
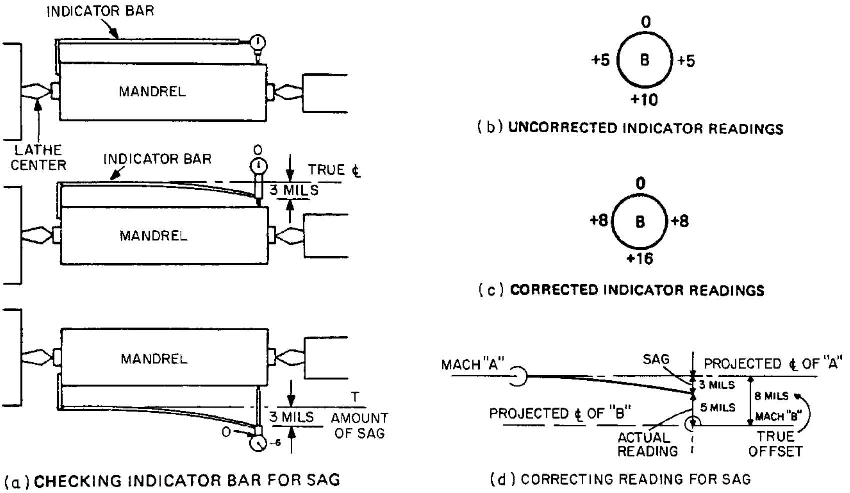


FIGURE 18 Required horizontal corrections

As shown in Figure 12, the driver and pump are purposely misaligned so actual operating temperatures will put the two shafts within acceptable limits. The acceptable limits for pump final alignment are 0.001 in/in (0.025 mm/mm) of coupling flex plane separation. If a spacer coupling is 5 in (13 cm) between flex planes on the flexible coupling, the shafts must be within 0.005 in (0.127 mm) vertical or horizontal offset. Pure angular misalignment in one plane is not desired, as it reduces the tolerance by 2:1 for gear couplings (dry coupling would have severe fatigue in one flex plane). The above limits should be reduced by one-half.

INSTALLATION SUGGESTIONS AND USE OF DIAL INDICATORS

1. Nonferrous shim packs should be installed under all feet of the pump and driver, particularly when installing a new pump. The amount should be 0.125 to 0.250 in (3.175 to 6.35 mm) in no more than three pieces to start; for example, one 0.125-in (3.175 mm) and two 0.0625-in (1.59-mm) full shims of stainless steel.
2. Motors have four feet generally, and any "soft foot" should be compensated first. A soft foot is one that is shorter than the other two or three feet, a condition that puts a twist or strain in the equipment. Simply place a dial indicator stem vertically against the motor foot and release the hold-down bolts sequentially around the unit, recording and retightening at each step. If a 0.002-in (0.050-mm) spring-up occurs on three feet, for example, and 0.006 in (0.152 mm) occurs on the fourth foot, add 0.004 in (0.10 mm) of shim to the fourth foot, eliminating the soft foot.
3. Provide low-sag tooling to reach over the coupling (coupling left in place) for reverse indicator alignment. A 0.001- to 0.0015-in (0.025- to 0.038-mm) sag is easy to accomplish on indicator reach bars.
4. Let the indicator indicate on its own bracket or bracket pin, thus preventing any poor surface condition of shaft or coupling from contributing to poor measurements.
5. Support the dial indicator weight on the motor or pump shaft so it does not contribute to "reach bar" sag.
6. Do not overlook the fact that many times one can clamp to the shaft behind each coupling hub and obtain more span and therefore better accuracy.
7. Record all data looking the *same* way down the unit; that is, top east, bottom, west *or* top, north bottom, south *or* top, right bottom, left. It is suggested that the driver-pump always be viewed from the driver end.
8. Turn the shafts in the direction they normally turn and approach the 90° points in a precise manner (do not back up and introduce backlash errors). Turning in the normal direction is good training because, on gear units, it reduces helix angle lift errors.



Indicator bar sag can be determined by firmly affixing it to a sag-free shaft mandrel, usually 4-in (10-cm) diameter or larger, dependent on length. The mandrel may be supported between lathe centers, mounted on knife edges, or held and rotated by hand. With the indicator bar positioned on top of the mandrel, the sag of the bar will be down toward the mandrel. Set indicator face to read zero at this position. By zeroing the indicator, you have errored the indicator by the amount of the sag. Rotate the mandrel 180° (indicator at bottom position). The indicator bar will sag away from the mandrel; hence the indicator reading will be twice the actual bar sag and will read negative as shown in (a).

$$\left(\frac{\text{TIR}}{2} = \text{Sag}\right)$$

Once the indicator bar sag is determined, it should be permanently stamped on the bar. This true sag must be accounted for when determining sweep readings.

To correct your sweep readings for sag, subtract twice the amount of true sag from the bottom reading (B) and correct the side readings (R & L) by subtracting the amount of the sag.

Sweep reading for an 8 mil vertical offset before making correction for a 3 mil indicator bar sag would be as shown in (b)

As shown in (c) and (d), correct sweep readings for indicator bar sag (amount of sag was 3 mils):

$$\begin{aligned} B &= (+10) - (-6) \text{ or } +16 \\ R &= (\ \ +5) - (-3) \text{ or } \ \ +8 \\ L &= (\ \ +5) - (-3) \text{ or } \ \ +8 \end{aligned}$$

FIGURE 19A through D Illustrative procedure to determine the amount of sag in an indicator bar (bracket) (Courtesy Reference 3)

9. If the motor can be turned down from the end opposite the end from which the measurements are taken, do so. Regardless, always release the strap wrench or spanner bar before recording each $\frac{1}{4}$-point reading.
10. Obtain center zero dial indicators *or* revolution counter indicators *or* carefully note all indicator movements with a mirror to assure, for example, that 0.090 in was not really −0.010 in. The algebraic sum of horizontal and vertical readings should be near equal.

INSTRUMENTS FOR VIBRATION ANALYSIS

One fact about end-suction and between-bearing pumps is that external visual evidence of mechanical problems is very limited. Only three gauges for mechanical trouble exist: temperature, vibration, and sound.

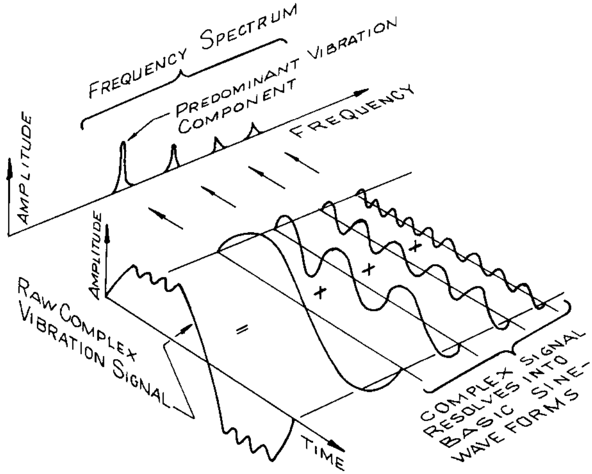


FIGURE 20 Complex vibration signal resolves into sine wave spectrum

It is normal for a machine to vibrate at some level; such vibrations are caused by manufacturing defects, design limits of the pump, casting irregularities, less than optimum application, and a maintenance/installation problem. When the velocity vibration level starts to increase 0.1 in/s (2.5 mm/s) zero to peak (0-P) above the "as new installed level," the vibration should be analyzed to determine the possible sources of the mechanical and/or hydraulic problem. Several mechanical and/or hydraulic problems may be producing, for instance, the 1× running speed frequency vibration. The key in using vibration to define the mechanical and/or hydraulic problems is to determine the *frequency* at which the vibration occurs. Vibration *amplitude* is also an important factor because it indicates the severity of the vibration. Field vibration data are normally a complex vibration waveform. By using a tunable analyzer, the complex vibration signal, as shown in Figure 20, can be filtered or tuned into its basic frequency components; that is, all complex signals are summations of the harmonics and subharmonics 1×, 0.5×, 6×, 30×, and so on. By comparing these filtered components of the complexed vibration signal with an analysis chart and some common-sense experience, probable causes of the vibration can be listed.

The first step toward resolving the vibration problem is to convert the mechanical movement to an equivalent electrical signal so it can be filtered and measured.

Often used analyzer systems are

1. A turbine ac-battery-powered analyzer with strobe light, providing amplitude, frequency, and phase. A plotter accessory can also be attached for copies of the data.
2. A small battery-powered, internally driven, tunable analyzer with a built-in plotter using an accelerometer or velocity sensor.
3. A spectrum analyzer, ac powered, that receives the signal directly from a vibration transducer or the recorded signal from a battery-powered four-channel FM/AM cassette tape recorder.

For startup or where the problem is tougher, one can add

1. Eight-channel FM tape recorder
2. Four-channel oscilloscope with blanking and time display
3. Tracking filter displaying revolutions per minute, amplitude, and phase, capable of tracking runup/rundown data

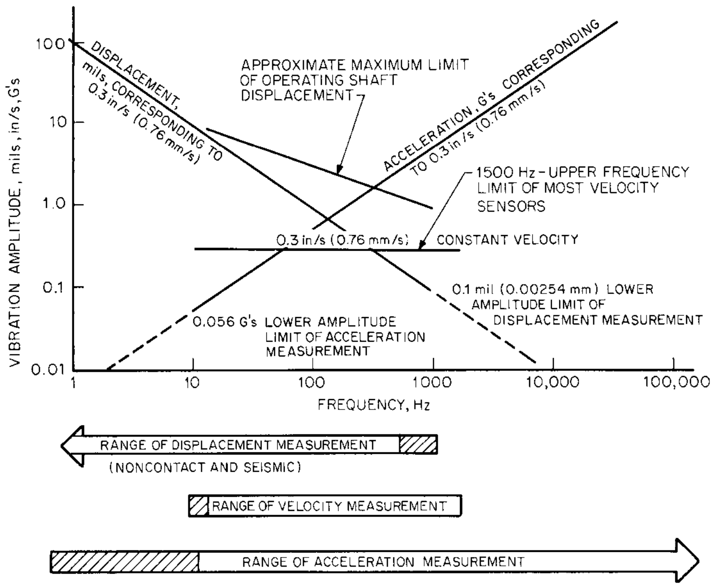


FIGURE 21 Limitations on machinery vibration analysis systems and transducers (mils × 0.0254 = mm; in/s × 25.4 = mm/s) (Reference 10)

Provisions should be made for the use of all types of sensors, as there are advantages in each. As more complex problems continue to appear, tunable analyzers with a sensor are not just a requirement but a necessity in any maintenance reliability program. The choice of a displacement sensor (eddy current probe), velocity or seismic sensor, or an accelerometer depends on the frequency range to be analyzed and the type of pumping equipment. There is no one vibration sensor for all jobs.

Of the three types of vibration measurements, acceleration and displacement are dependent on frequency and velocity is independent of frequency. Most engineers and technicians select a measurement that is independent of frequency for a datum to judge the general health of new and used pumps. With the exception of low-speed pumps and motors, 1750 rpm or less, unfiltered velocity and filtered velocity are used for most basic data. Figure 21 shows the frequency relationships (log) versus output (log) of three different measurement sensors with reference to a constant velocity of 0.3 in/s (7.6 mm/s). The figure gives an overview of present sensor limits and shows that each sensor is like a window through which portions of the frequency spectrum may be observed. The figure also shows that the accelerometer is the choice sensor at high frequency because it measures the square of the frequency. The advantage of displacement at low frequencies is due to its high output; the disadvantage of displacement at high frequencies is that the output signal will disappear into the background noise of most measuring systems.

One should not confuse the measurement parameters (displacement, velocity, and acceleration) with the sensors (eddy current probes for displacement, velocity sensors, and accelerometers). The basic relationship of these measurement parameters with commonly used units are shown on a simple sine wave in Figure 22.

Although the velocity sensor is not necessarily the *best* all-around type of sensor, it does have the advantage of high self-generating output (up to 1000 ft [300 m] of cable), can be mounted in any position, and is influenced only slightly (less than 5%) by transverse sensitivity (side forces). The disadvantages are that the output signal below 600 cpm is significantly nonlinear but can be corrected, the accuracy is limited at ±8% to 1000 Hz, and

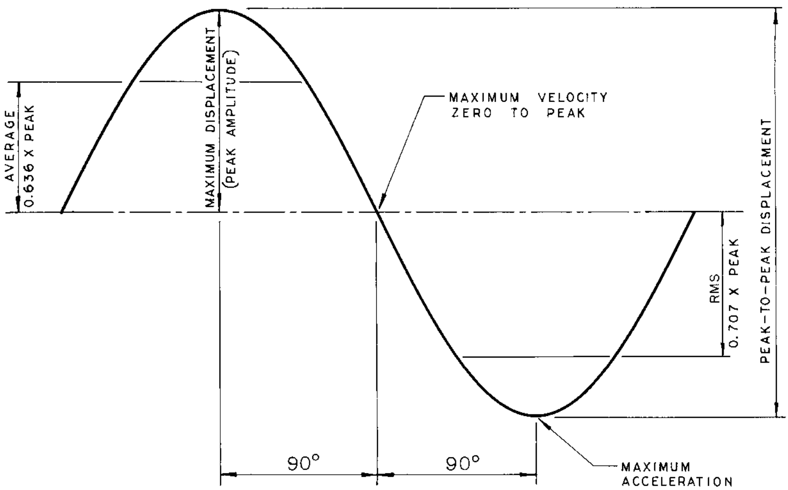


FIGURE 22 Basic relationship of measured parameters with a simple sinusoidal vibration

the sensor will most likely have problems in one to two years when mounted in field applications where vibration is high, especially vane passing frequencies.

The piezoelectric accelerometer is a very light and compact sensor that measures vibration using a mass mounted on a piezoelectric crystal. Its output is low and requires a charge amplifier in the lead even with very short leads. The accelerometer is small and can be mounted virtually anywhere; it has a 1 to 3% influence factor from transverse side forces. A good rule of thumb on the usable frequency range is one-fifth to one-third of the resonant frequency. The disadvantages are that the sensor is sensitive to mounting torque, although stud mount is the best method to mount accelerometers. A lot of data are produced, of which some may be the data from an excited accelerometer resonance or cable noise. An impedance matching device can be built into the accelerometer for use at temperatures below 250°F (120°C), and cable noise can be greatly reduced with the voltage and charge sensitivity greatly improved; for example, 100 mV/g and 50 to 100 pC/g (where g = number of accelerations of gravity). For higher temperatures, the accelerometer will need a separate charge amplifier and may need heat insulation, such as MICA wafers (refer to API 678, dated 1981).

Most so-called ultrasonic analyzers use the accelerometer as a structural microphone. Many have chosen a carrier frequency in the megahertz range to improve the signal-to-noise ratio and make characteristic high-frequency patterns. Most of these systems are still in the development stage.

Techniques for Taking Data The second most important part of a vibration analysis program is the type of data taken and the techniques used to take the data. The purpose of taking vibration data on a pump is to either perform an analysis because someone noticed a noise or increased vibration level, or as a part of a periodic preventative maintenance program. It has been proven from experience that the velocity measurement is the best method for determining acceptable levels of centrifugal pump vibration. This is not to say that displacement and acceleration are not measures of vibration severity; they are, but it is necessary to know the frequency of the vibration. Displacement is preferred by a few for frequencies less than 6000 cpm. Accelerometers matched with analyzers can be purchased with signal integration that will give reliable readings in velocity in the 3000- to 60,000-cpm range. For readings above 60,000 cpm, the sensor would generally be an accelerometer reading in g's, peak or integrated to read velocity zero to peak or 0-P.

Vibration amplitude is an important parameter because it indicates the approximate severity of the dynamic stress levels in the pump. Experience has shown that the shaft bearings and seal will probably fail in a pump with a velocity reading of 0.5 in/s (13 mm/s) 0-P. Also, catastrophic failures will probably occur when a pump is at 1 in/s (2.5 mm/s) 0-P. Pumps with velocity readings of 0.05 to 0.15 in/s (1.3 to 3.8 mm/s) 0-P, will perform well mechanically. Vibration readings are taken in the horizontal and vertical planes on the bearing housing of horizontal-shaft pumps.

To have a worthwhile maintenance reliability program with pumps, vibration readings must be recorded regularly (that is, monthly). This can range from a trend plotting of unfiltered vibration to a full vibration analysis using a real-time analyzer to generate the frequency spectrum. A standard method used by many companies consists of taping pump vibrations with a battery-powered cassette recorder using a velocity sensor. Readings can then be processed through a real-time analyzer and recorded on an *XYY′* plotter. The best application of this method is during startup and repair evaluation.

As an alternate method, a spectrum analyzer/plotter that produces a spectrum on a 4 in × 6 in (l0 mm × 15 mm) card with a frequency plot versus amplitude can be used. This procedure has in some installations detected and corrected 95% of the mechanical problems before failure. Experience has shown that had unfiltered displacement readings been taken, only 60 to 70% of the mechanical problems would have been observed. During these recordings, emphasis should be placed on the change in vibration levels, which is a better indication of a mechanical problem than absolute vibration.

One of the best pieces of data available for the pump's equipment file is a vibration record taken during the manufacturer's test or during water batching or commissioning. It is advisable to request a witness performance test on key or critical pumps. The purpose of this test is to assure mechanical reliability *along with* performance.

The manufacturer should be asked about the availability and type of vibration analysis equipment and sensors. Regardless of the instruments used, the vibration data sheet for the tested pump should have a sketch of where all vibration points were taken. The manufacturer should also supply a complete mechanical description of the number of impeller vanes, number of casing volute cutwaters or diffuser vanes, type of coupling, length of coupling spacer, and so on.

There are several different methods for taking periodic vibration data on pumps:

1. Using a handheld battery-powered velocity probe/readout, a machinist or operator logs unfiltered readings taken at one or two points on the bearing housing. When the reading reaches 0.3 to 0.5 in/s (8 to 13 mm/s) 0-P, the pump is pulled for maintenance. Readings are usually taken every two weeks.
2. Vibration points in the vertical, horizontal, and axial directions are recorded on a tabulated chart in unfiltered and filtered velocity at the various peak amplitudes, using a battery-powered tunable analyzer with a velocity sensor.
3. Vibrations in the vertical, horizontal, and axial direction are taken at each bearing, using a velocity sensor. The signal is recorded on a tape recorder, preferably a battery-powered FM/AM cassette. These data are then processed through a real-time analyzer. A spectrum hard copy is made on an *XYY′* plotter of velocity versus frequency.
4. Key vibration points are fed directly from a velocity sensor or an accelerometer/charge amplifier through a long extension cable to a safe area, where a real-time analyzer processes the signal into a velocity versus frequency spectrum or a *g*'s (acceleration) versus frequency spectrum. Hard copies for records are made on a *XYY′* plotter. This method requires two technicians with radios.

The most accurate are methods 3 and 4. The most costly to run in workerhours per point is method 2. The least accurate is, of course, method 1, but it is a popular screening technique.

Use of Vibration Sensors The use of a handheld velocity sensor with an aluminum extension rod or a light-duty vise grip with the probe mounted on the top of the grip has produced some high and misleading vibration readings because of extension resonances. For instance, the vise grip should not be used because of a 5000-cpm resonance. A 9-in (23 cm) long by $\frac{3}{8}$-in (0.95-cm) diameter extension to the velocity pickup should not be used above 16,000 cpm. The approximate axial natural frequency in cycles per minute for a rod extension from the probe, in tension and compression, can be expressed as in USCS units $$f_n = 188\sqrt{\frac{AE}{WL}}$$

in SI units $$f_n = 946\sqrt{\frac{AE}{WL}}$$

where W = pickup weight (force), lb (N)
L = length of rod, in (m)
A = cross-sectional area of rod, in^2 (m^2)
E = modulus of elasticity of rod,° lb/in^2 (kPa)

Use of attachments above these listed frequencies will produce a higher amplitude.

The best and simplest method of holding a velocity probe to a pump is a two-bar magnetic holder on the end of a velocity probe. Proper cleaning and some paint removal are generally necessary for good attachment. Periodic wiping of the magnetic bar to remove iron filings is also necessary. After mounting the probe, give it a light twist and a rocking motion; if it twists easily or rocks, change locations or reclean the surface. This location should be marked and future readings taken on the same spot; otherwise, the trend plots will vary.

If there is a concern about an extension or magnetic holder resonance, test this by holding the sensor, without the extension or magnetic holder, directly to a reasonably flat spot and noting any differences. Holding the sensor on a flat spot is generally safe up to 60,000 cpm.

When measuring vibration on an electric motor, there is always the possibility of false readings at 60 and/or 120 Hz due to electrical induction by the motor. This can be checked by two methods:

1. Hold the sensor by its cord and move it toward the motor, noting any increase in amplitude.
2. Using a two-channel oscilloscope, trigger the filtered signal against line voltage. In-phase signals mean the vibration is electrically induced.

For field use, one usually does not have to contend with temperatures above 250°F (120°C) direct to the sensor; thus, accelerometers with built-in impedance matching devices can be used and 100 mV/*g* voltage sensitivities can be obtained and transmitted 300 ft (90 m) if necessary. If the frequency range is low, and it is for pumps, a charge sensitivity in the order of 100 pC/*g* can also be obtained.

Techniques for Taking Preliminary Vibration Readings Some key points to remember before you start your analysis of the vibration problem:

1. Do not reach a decision on what the problem is before you record and analyze the data. By deciding too quickly what the problem is, you will most likely neglect other important factors.
2. Before you take the data, take time to review maintenance logs, talk with the area mechanic and operator, and make notes on the following:
   - **a.** Are there any unusual sounds (cavitation, bearings, and so on)?
   - **b.** Is there any movement in the discharge pressure gage?
   - **c.** What is the direction of rotation?
   - **d.** Are the flush and cooling lines lined up properly?
   - **e.** Is there any movement in the coupling shim pack?
   - **f.** Are there any foundation cracks?
   - **g.** Are pipe supports functioning properly?
   - **h.** Has a suction screen been installed?

°For aluminumm, $E = 10.3 \times 10^6$ lb/in^2 (71×10^6 kPa). For steel, $E = 30 \times 10^6$ lb/in^2 (207×10^6 kPa)

i. What is the magnitude of the liquid velocity in the suction line?

j. Is the automatic oiler level adjusted correctly?

k. Where is the pump being operated?

l. What are the flow, suction, and discharge pressure?

m. Is the pump's minimum flow bypass system in service?

n. Has the process changed?

o. What is the suction valve stem orientation?

p. Have there been any color changes in the paint?

q. Are there any loose parts, including the coupling guard?

r. Determine the color and feel of the oil (if possible).

s. What is the bearing housing temperature?

t. How is the coupling guard attached (attachment to the bearing housing is poor practice)?

You will be surprised how much this information aids in an analysis. Example: you record a high 1× in the radial plane, low values of 2× and 5×, and several high-frequency components at about 0.15 in/s (4 mm/s) 0-P. The 1× could be a bent shaft, loose coupling, plugged impeller, bad coupling unbalance, upper and lower case halves misaligned, and a whole list of running frequencies symptoms. During your review of maintenance logs, you noted that the impeller had been replaced because there had been distillation column tray part damage. The next questions you should ask are, "Did maintenance reinstall the suction screen?" (if not, another tray part may have lodged in the impeller) and, "Was the impeller rebalanced after it was trimmed from maximum diameter?"

3. Do not try to interpret partial vibration readings for someone looking over your shoulder before you have even taken all the readings. Sit down in a quiet place with your notes on installation and maintenance, a symptoms list, and a severity chart and then make the analysis. Analysis is not a simple task, but with some experience you will build confidence and it will become second nature.

VIBRATION DIAGNOSTICS

Analysis Symptoms The vibration severity chart and vibration identification chart are guides, but experience, a set of procedures, and study of the literature will make diagnostics easier. To be effective, one must be thoroughly familiar with the machinery's internal construction, installation, and basic control system. Both mechanical and hydraulic mechanisms can produce symptoms of vibration.

The vibration analysis symptoms, or vibration severity criteria, chart has taken on many forms since the Rathbone chart of 1939. Perhaps the most widely used symptoms chart in the turbo-machinery field today is the original paper published by Sohre.[5] A condensation and revision of the original paper is shown in Table 2. Although this chart includes some symptoms that will never appear in pumps, it is one of the better references for vibration analysis. The way the table gives percentages of cases showing the symptoms for the causes listed is unique. As one learns to use the chart and modifies it with experience, a good diagnostic tool will be developed.

A good guide for unfiltered bearing cap velocity limits on field installed pumps is given in Table 3. A guide for shop testing new and rebuilt pumps is given in Reference 6.

Comments on Table 2 In the following comments, the numbers correspond to "Cause of vibration" in Table 2.

1. Long, high-speed rotors often require field balancing at full speed to make adjustments for rotor deflection and final support conditions. Corrections can be made at balancing rings or at coupling bolts.

TABLE 2 Vibration analysis symptoms

Cause of vibration	Predominant frequencies												
	0–40%	40–50%	50–100%	1× running frequency	2× running frequency	Higher multiples	½ running frequency	¼ running frequency	Lower multiples	Odd frequency	Very high frequency	Vert.	Hor.
1. Initial unbalance	..	..	..	90	5	5	..	..	..	..	..	40	50
2. Permanent bow or lost rotor parts (vanes)	..	..	..	90	5	5	..	..	..	..	..		
3. Temporary rotor bow	..	..	..	90	5	5	..	..	..	..	..		
4. Casing distortion — Temporary	10			80	5	5	..	..	..	..	..		
4. Casing distortion — Permanent	10			80	5	5	..	..	..	..	..		
5. Foundation distortion	..	20	..	50	20	..	..	..	..	10	..		
6. Seal rub	10	10	10	20	10	10	..	..	10	10	10	30	40
7. Rotor rub, axial	20			30	10	10	..	..	10	10	10	30	40
8. Misalignment	..	..	..	40	50	10	..	..	..	..	..	20	30
9. Piping forces	..	..	..	40	50	10	..	..	..	..	..	20	30
10. Journal & bearing eccentricity	..	..	..	80	20	..	..	..	..	..	..	40	50
11. Bearing damage	20			40	20	..	..	..	..	..	20	30	40
12. Bearing & support excited vibration (oil whirls, etc.)	10	70	..	..	..	..	10	10	..	..	..	40	50
13. Unequal bearing stiffness horizontal-vertical	..	..	..	..	80	20	..	..	..	..	..	40	50
14. Thrust bearing damage	90					..	..	..	..	..	10	20	30
Insufficient tightness in assembly of:	Predominant frequency will show at lowest critical or resonant frequency							..	..	..	..		
15. Rotor (shrink fits)	40	40	10	..	..	..	..	..	..	10	..	40	50
16. Bearing liner	90		..	..	..	..	..	..	..	10	..	40	50
17. Bearing cases	90		..	..	..	..	..	..	..	10	..	40	50
18. Casing & support	50		..	..	..	..	..	..	..	50	..	40	50
19. Gear inaccuracy	..	..	..	..	..	20	..	..	..	20	60	30	50
20. Coupling inaccuracy or damage	10	20	10	20	30	10	..	..	..	..	..	30	40

Numbers indicate percent of cases showing previous symptoms, for causes listed in vertical column at left.

Source: *The Practical Vibration Primer* by Charles Jackson. Copyright (c) 1979 by Gulf Publishing Company, Houston, Texas.

TABLE 2 Continued.

Direction and location of predominant amplitude							Amplitude response to speed variation during vibration-test runs											
							Coming up						Slowing down					
Axial	Shaft	Bearings	Casing	Foundation	Piping	Coupling	Stays same	Increases	Decreases	Peaks	Comes suddenly	Drops out suddenly	Stays same	Increases	Decreases	Comes suddenly	Drops out suddenly	Cause
10	90	10	..	..	..	..	..	100	..	Peaks at critical	..	..	..	..	100	..	..	1
			..	..	..	..	..	100	..		..	..	..	..	..	..	..	2
			..	..	..	..	30	60	5		5	..	30	5	50	5	10	3
			..	..	..	..	30	50	5		5	10	30	5	50	5	10	4
			..	..	..	..	40	60	..		..	..	40	..	60	..	..	
	40	30	10	10	10	..	20	80	..		..	..	20	..	80	..	..	5
30	80	10	10	..	..	..	10	70	..		10	10	10	..	70	10	10	6
30	70	10	20	..	..	..	10	40	10		20	20	10	..	50	20	20	7
50	80	10	10	..	..	..	20	30	10		20	20	20	..	40	20	20	8
50	80	10	10	..	..	..	20	40	..		20	20	20	..	40	20	20	9
10	90	10	..	..	..	..	40	50	10		..	..	40	10	50	..	..	10
30	70	20	10	..	..	..	10	50	10		20	10	10	10	50	10	20	11
10	50	20	20	20	..	..	..	10	..	..	90	..	..	..	10	..	90	12
10	40	30	30	..	..	..	..	40	..	50	10	..	..	..	40	..	10	13
50	60	20	20	..	..	..	20	50	10	..	10	10	20	10	50	10	10	14
10	60	20	20	..	..	..	..	..	..	..	90	10	..	..	..	10	90	15
10	80	10	10	..	..	..	..	..	..	..	90	10	..	..	..	10	90	16
10	70	20	10	..	..	..	..	..	..	..	90	10	..	..	..	10	90	17
10	50	20	30	..	..	..	..	..	..	..	90	10	..	..	..	10	90	18
20	80	10	10	..	..	..	20	20	20	20	10	10	20	20	20	10	10	19
30	70	20	..	..	..	10	10	20	..	20	Loose sleeve, friction or dirt 40 in teeth 10		10	..	20	10	40	20

TABLE 2 Continued.

Cause of vibration	0–40%	40–50%	50–100%	1 × running frequency	2 × running frequency	Higher multiples	½ running frequency	¼ running frequency	Lower multiples	Odd frequency	Very high frequency	Vert.	Hor.
	Predominant frequencies												
21. Rotor & bearing system critical	..	..	..	100	..	..	..	..	..	..	..	40	50
22. Coupling critical	..	..	..	100	Also make sure tooth fit is *tight!*							20	40
23. Overhang critical	..	..	..	100	..	..	..	..	..	..	..	40	50
Structural resonance of: 24. Casing	..	10	..	70	10	..	10	..	..	..	..	40	50
25. Supports	..	10	..	70	10	..	10	..	..	..	..	40	50
26. Foundation	..	20	..	60	10	..	10	..	..	..	..	30	40
27. Pressure pulsations	Most troublesome if combined with resonance									100	..	30	40
28. Electrically excited vibration	..	..	..	..	..	↓	..	..	..	..	..	30	40
29. Vibration transmission	..	..	..	..	..		..	..	..	90	..	30	40
30. Valve vibration	..	..	..	..	..	↓	..	..	..	..	100	30	40
Problem	The section below is meant to identify basic mechanisms												
31. Subharmonic resonance	..	Rare—Look for aerodynamic origin (seals)					100			..	..	30	30
32. Harmonic resonance	..	..	..	..	100		..	..	..	..	..	40	40
33. Friction induced whirl	80	10	10	..	..	..	..	..	..	..	..	40	50
34. Critical speed	..	..	..	100	..	..	..	..	..	..	..	40	50
35. Resonant vibration	..	..	..	100	..	..	..	..	..	..	..	40	40
36. Oil whirl	..	100	Watch for aerodynamic rotor-lift (partial admission, etc.)									40	50
37. Resonant whirl	..	100	..	..	..	..	..	..	..	..	..	40	50
38. Dry whirl	..	..	..	..	..	..	..	..	..	..	100	30	40
39. Clearance induced vibrations	10	80	10	..	..	..	..	..	..	..	..	40	50
40. Torsional resonance	..	..	..	40	20	20	..	..	..	20	..	Torsional	
41. Transient torsional	..	..	..	50	..	..	..	..	..	50	..	..	↓

TABLE 2 Continued.

Direction and location of predominant amplitude							Amplitude response to speed variation during vibration-test runs											
							Coming up						Slowing down					
Axial	Shaft	Bearings	Casing	Foundation	Piping	Coupling	Stays same	Increases	Decreases	Peaks	Comes suddenly	Drops out suddenly	Stays same	Increases	Decreases	Comes suddenly	Drops out suddenly	Cause
10	70	30	..	..	..	..	..	20	..	80	..	..	..	..	20	..	..	21
40	10	10	..	..	..	80	..	20	..	80	..	If loose	..	..	20	50 If loose	..	22
10	70	10	..	..	..	20	..	30	..	70	..	..	..	..	30	..	..	23
10	..	40	40	10	10	..	..	20	..	80	..	..	..	..	20	..	..	24
10	..	20	50	20	10	..	..	20	..	80	..	..	..	..	20	..	..	25
30	..	10	40	40	10	..	..	20	..	80	..	..	..	..	20	..	..	26
30	Can excite whirls or resonance		30	30	40	..	90		10%—Depending on origin of disturbance				90	10 →				27
30	↓	↓	40	40	20	..	90	..	..	↓	..	..	90	↓	..	..	..	28
30	↓	↓	40	40	20	..	90	..	..	↓	..	..	90	↓	..	..	..	29
30	..	..	80	10	10	..	80	..	..	..	10	10	80	..	..	10	10	30
40	20 If bearing is excited	80	20	20	20	..	..	20	..	20	30	30	..	..	20	30	30	31
20	20	10	10	30	30	..	20	20	..	60	..	..	20	..	20	..	..	32
10	80	20	..	..	..	..	..	..	..	..	90	10	..	..	..	10	90	33
10	60	40	..	..	..	..	..	20	..	80	..	..	..	..	20	..	..	34
20	20	10	20	30	20	..	..	20	..	80	..	..	..	..	20	..	..	35
10	80	20	..	..	..	..	..	..	..	..	100	..	..	..	..	..	100	36
10	20	20	20	20	20	..	..	..	..	..	80	20	..	..	..	20	80	37
30	40	20	20	10	..	10	..	..	..	..	80	20	80	..	..	..	20	38
10	70	10	10	..	..	10	..	..	..	..	80	20	20	..	..	20	60	39
..	100	Lateral amplitude 40	40	..	..	10	..	20	..	30	30	20	20	..	..	20	30	40
..	Torsion 100	40	40	..	..	10	..	..	..	50	30	20	..	..	..	30	20	41

TABLE 3 Bearing cap data-velocity unfiltered

Smooth	Acceptable	Marginal	Planned shutdown for repairs	Immediate shutdown
0.1 in/s (p) and less	0.1–0.2 in/s (p)	0.2–0.3 in/s (p)	0.3–0.5 in/s (p)	0.5 in/s (p)

Note: For gearing, add 0.1 in/s to all values.
p = peak
mm/s = 25.4 × in/s

2. Bent rotors can sometimes be straightened by the "hot-spot" procedure, but this should be regarded as a temporary solution because bow will come back in time. Several rotor failures have resulted from this practice. If blades or disks have failed, check for corrosion fatigue, stress corrosion, resonance, off-design operation.
3. Straighten bow slowly, running on turning gear or at low speed. If rubbing occurs, trip unit immediately and keep the rotor turning 90° using a shaft wrench every 5 minutes until the rub clears; resume slow run. This may take 12 to 24 h.
4. Often requires complete rework or new case, but sometimes a mild distortion corrects itself with time (requires periodic internal and external realignment). Usually caused by excessive piping forces or thermal shock.
5. Usually caused by poor mat under the foundation or thermal stress (hot spots) or unequal shrinkage. May require extensive and costly repairs.
6. Slight rub may clear, but trip the unit immediately if a high-speed rub gets worse. Turn by hand until clear.
7. Unless thrust bearing has failed, this is caused by rapid changes of load and temperature. Machine should be opened and inspected.
8. Usually caused by excessive pipe strain or inadequate mounting and foundation, but is sometimes caused by local heat from pipes or the sun's heating the base and foundation.
9. Most trouble is caused by poor pipe supports (should use spring hangers), improperly used expansion joints, and poor pipe line up at casing connections. Foundation setting can also cause severe strain.
10. Bearings may become distorted from heat. Make a hot check, if possible, observing contact.
11. Watch for brown discoloration, which often precedes recurring failures. This indicates very high local oil film temperatures. Check rotor for vibration. Check bearing design and hot clearances. Check condition of oil, especially viscosity.
12. Check clearances and roundness of journal, as well as contact and tight bearing fit in the case. Watch out for vibration transmission from other sources and check the frequency. May require antiwhirl bearings or tilting-shoe bearings. Check especially for resonances at whirl frequency (or multiples) in foundation and piping.
13. This can excite resonances and criticals and combinations thereof at two times running frequency. Usually difficult to field balance because, when horizontal vibration improves, vertical vibration gets worse and vice versa. It may be necessary to increase horizontal bearing support stiffness (or mass) if the problem is severe.
14. Usually the result of slugging the machine with fluid, solids built up on rotor, or off-design operation (especially surging).
15. The frequency at rotor support critical is characteristic. Disks and sleeves may have lost their interference fit by rapid temperature changes. Parts usually are not loose at standstill.

16. It is often confused with oil whirl because the characteristics are essentially the same. Before suspecting any whirl, make sure everything in the bearing assembly is absolutely tight with an interference fit.

17. This should always be checked.

18. It usually involves shading pedestals and casing feet. Check for friction, proper clearance, and piping strains.

19. To obtain frequencies, tape a microphone to the gear case and record noise on magnetic tape.

20. Loose coupling sleeves are notorious troublemakers, especially in conjunction with long, heavy spacers. Check tooth fit by placing indicators on top, then lifting by hand or a jack and noting looseness (should not be more than 1–2 mils [0.025 to 0.05 mm] at standstill, at most). Use hollow coupling spacers. Make sure coupling hubs have at least 1 mil/in (1 mm/m) interference fit on shaft. Loose hubs have caused many shaft failures and serious vibration problems.

21. Try field balancing; more viscous oil (colder); larger, longer bearings with minimum clearance and tight fit; stiffen bearing supports and other structures between bearing and ground. This is basically a design problem. It may require additional stabilizing bearings or a solid coupling. It is difficult to correct in the field. With high-speed machines, adding mass at the bearing case helps considerably.

22. These are criticals of the spacer-teeth-overhang subsystem. Often encountered with long spacers. Make sure of tight-fitting teeth with a slight interference at standstill and make the spacer as light and stiff as possible (tubular). Consider using a solid or membrane coupling if the problem is severe. Check coupling balance.

23. Overhang criticals can be exceedingly troublesome. Long overhangs shift the nodal point of the rotor deflection line (free-free mode) toward the bearing, robbing the bearing of its damping capability. This can make critical speeds so rough it is impossible to pass through these speeds. Shorten the overhang or put in an outboard bearing for stabilization.

24. Casing resonance is also called case drumming. It can be very persistent but is sometimes harmless. The danger is that parts may come loose and fall into the machine. Also, rotor/casing interaction may be involved. Diaphragm drumming is serious because it can cause catastrophic failure of the diaphragm.

25. Local drumming is usually harmless, but major resonances, resulting in vibration of the entire case as a unit, are potentially dangerous because of possible rubs and component failures, as well as possible excitation of other vibrations.

26. Similar problems exist as in 24 and 25 with the added complications of settling, cracking, warping, and misalignment. This cause may also produce piping troubles and possible case warpage. Foundation resonance is serious and greatly reduces unit reliability.

27. Pressure pulsations can excite other vibrations with possible serious consequences. Eliminate such vibrations using restraints, flexible pipe supports, sway braces, shock absorbers, and so on, plus isolation of the foundation from piping, building, basement, and operating floor.

28. It occurs mostly at two times line frequency (7200 cpm), coming from motor and generator fields. Turn the fields off to verify the source. It is usually harmless, but if the foundation or other components (rotor critical or torsional) are resonant, the vibrations may be severe. There is a risk of catastrophic failure if there is a short circuit or other upsets.

29. This can excite serious vibrations or cause bearing failures. Isolate the piping and foundation and use shock absorbers and sway braces.

30. Valve vibration is rare but sometimes very violent. Such vibrations are aerodynamically excited. Change the valve shape to reduce turbulence and increase rigidity in the valve gear. Make sure the valve cannot spin.

31. The vibration is exactly one-half, one-quarter, one-eighth of the exciting frequency. It can be excited only in nonlinear systems; therefore, look for such things as looseness and aerodynamic or hydrodynamic excitations. It may involve rotor "shuttling." If so, check the seal system, thrust clearances, couplings, and rotor-stator clearance effects.

32. The vibrations are at two, three, and four times exciting frequency. The treatment is the same as for direct resonance: change the frequency and add damping.

33. If the cause is intermittent, look into temperature variations. Usually the rotor must be rebuilt, but first try to increase stator damping, add larger bearings (tilting-shoe), increase stator mass and stiffness, and improve the foundation. This problem is usually caused by maloperation, such as quick temperature changes and fluid slugging. Use membrane-type coupling.

34. This is basically a design problem, but is often aggravated by poor balancing and a poor foundation. Try to field-balance the rotor at operating speed, lower oil temperature, and use larger and tighter bearings.

35. Add mass or change stiffness to shift the resonant frequency. Add damping. Reduce excitation and improve system isolation. Reducing mass or stiffness can leave the amplitude the same even if resonant frequency shifts because of stronger amplification. Check "mobility."

36. Stiffen the foundation or bearing structure. Add mass at the bearing, increase critical speed, or use tilting-shoe bearings (which is the best solution). First, check for loose fit of bearings in bearing case.

37. Same comments as 36 with additional resonance of rotor, stator, foundation, piping, or external excitation; find the resonant members and the sources of excitation. Tilting-shoe bearings are the best. Check for loose bearings.

38. Sometimes you can bear the squeal of a bearing or seal, but frequency is usually ultrasonic—very destructive. Check for rotor vanes hitting the stator, especially if clearances are smaller than the oil film thickness plus rotor deflection while passing through the critical speed.

39. Usually accompanied by rocking motions and beating within clearances. It is serious especially in the bearing assembly. Frequencies are often below running frequency. Make sure everything is absolutely tight, with some interference. Line-in-line fits are usually not sufficient to positively prevent this type of problem.

40. This problem is very destructive and difficult to find. The symptoms are gear noise, wear on the hack side of teeth, strong electrical noise or vibration, loose coupling bolts, and fretting corrosion under the coupling bolts. There is wear on both sides of coupling teeth and possibly torsional-fatigue cracks in keyway ends. The best solution is to install properly tuned torsional vibration dampers.

41. It is similar to 40, but encountered only during startup and shutdown because of very strong torsional pulsations. It occurs in reciprocating machinery and synchronous motors. Check for torsional cracks.

Impeller Unbalance Impeller unbalance appears as a 1× running speed frequency vibration approximately 90% of the time and may be mechanical or hydraulic in origin. Impeller mechanical unbalance is a frequent cause of mechanical seal and bearing failures. Many mechanics will never think of checking impeller balance until heavily pitted areas appear. Because of the nonhomogeneous nature of most castings, corrosion is usually more aggressive in one area of the impeller. The degree of etching or surface pitting is a judgmental indicator of balance change. Impeller balancing should be part of the shop repair procedure for impellers over 10 in (25 cm) at 3600 rpm. It is good practice, when balancing an impeller, to keep the impeller bore to balance mandrel fit no greater than 0.001 in (0.0254 mm) loose. Installing the impeller with the keyway up on the balance mandrel and pump shaft will help eliminate some of the unbalance due to shaft centerline shift.

EXAMPLE A 38-lb (17.2 kg), 15 $\frac{1}{2}$-in (39.4-cm) diameter impeller operating at 3600 rpm is balanced on a machine good to 25×10^{-6} in (635×10^{-6} mm) using an expanding mandrel. The impeller is then installed on its shaft, which has a loose fit of 0.0035 in (0.0889 mm). The forces created by this shift in the center of mass is calculated as follows:

in USCS units

$$\begin{aligned}\text{Unbalance} &= \text{eccentricity of impeller (in)} \times \text{impeller wt. force (oz)}\\ &= \frac{0.0035}{2} \times 38 \times 16 = 1.064 \text{ oz} \cdot \text{in}\\ \text{Unbalance force} &= 1.77 \left(\frac{\text{rpm}}{1000}\right)^2 \times \text{unbalance (oz} \cdot \text{in)}\\ &= 1.77 \left(\frac{3600}{1000}\right)^2 \times 1.064 = 24.4 \text{ lb}\end{aligned}$$

in SI units

$$\begin{aligned}\text{Unbalance} &= \text{eccentricity of impeller (mm)} \times \text{impeller wt, mass (g)}\\ &= \frac{0.0889}{2} \times 17.2 \times 1000 = 765 \text{ g} \cdot \text{mm}\\ \text{Unbalance force} &= 0.01094 \left(\frac{\text{rpm}}{1000}\right)^2 \times \text{unbalance (g} \cdot \text{mm)}\\ &= 0.01094 \left(\frac{3600}{1000}\right)^2 \times 765 = 108.5 \text{ N}\end{aligned}$$

The example also points out that the unbalance force generated by loose fit impellers with keyways mounted in one plane could be quite high. This force could be minimized by staggering keyways or randomly orienting the impellers on the balance mandrel. Shifting of shrink-fitted, well-balanced impellers on multistage and high-speed double-suction pumps after a period of operation can result in unbalance. The shifting of the impeller is due to the relaxation of residual stresses that built up as the impeller cooled and contracted around the shaft. Shaft vibration and flexing tend to relieve the residual stress and cause the impeller to cock or bow the shaft from the original balance centerline.

Standards should be referred to for balancing pumps and their drivers. When balancing, consideration must be given to the need for balancing at rated speed in order to properly evaluate the importance of shaft deflection due to modal components of unbalance. See References 12 and 13.

Hydraulic Unbalance Uneven flow distribution entering the impeller can cause a 1× running speed frequency type of vibration. The unbalance occurs because the flow is not equal in all vane passages. An example of this is a double-suction impeller with a short, straight run to the pump and an elbow in the horizontal plane. Flow from the elbow does not have time to straighten and therefore enters both sides of the impeller unequally. A similar condition results if suction is taken from a tee off the main header. Unequal and unsteady flow into the impeller may cause axial thrust and high axial vibrations. Thus, it is good design practice to install elbows vertically in double-suction pumps.

In double-suction pumps, the nonsymmetrical positioning of the impeller or the offset of the upper case half of the lower case half will cause a 1× unbalance due to nonsymmetrical flow.

Recirculation forces and pulsation recirculation within a pump (which can occur when flow is less than design) may manifest themselves in the form of a noise and/or vibration with random frequencies, along with pressure pulsation that may be seen on a pressure gage. Recirculation may also appear in the piping system as vibration and noise. Increased *NPSHA* has helped in a few cases, especially if the recirculation is mainly on the suction side of the impeller. After a pump has a recirculation problem in a given system and the flow cannot be increased using a bypass, little can be done to the pump itself unless the system characteristics allow an impeller change.

Antifriction Bearings Vibrations generated by ball bearings cover a wide range of high frequencies that are not necessarily a multiple of the shaft running speed. The frequency readings obtained during analysis are somewhat unsteady because of the resolution of the filters in a hand-tuned analyzer. The amplitude reading may also be somewhat unsteady.

Experience has shown that hand-tuned field analyzers tend to show the last stage of the bearing failure. Monitoring of stress waves or shock pulses (impact energy) on the pump bearing housing will show failure trends that will generally precede an increase in the detectable level of mechanical vibration. This method of failure detection is called acoustic high-frequency monitoring or incipient-failure detection (1FD). A comparison between conventional methods and the acoustic high-frequency method is shown in Figure 23.

Accurate analysis of pump bearings and other machinery can also be made using a velocity sensor good to 1500 Hz or an accelerometer. Data should be recorded and processed through a real-time analyzer with at least 256-line resolution capability and a band selectable analysis option. Analysis of antifriction bearings using a real-time analyzer and equations for calculating frequencies generated by defective bearings can be found in Reference 7.

Accurate and extended analysis of pump bearing vibration is generally not needed. A majority of bearing problems are currently identified by an acoustic noise during operation. What is needed by maintenance personnel is a quick and reliable method to monitor bearings and determine when a bearing is failing and the rate of bearing deterioration. At present, there are some expensive instruments that can be purchased, but none of them meet maintenance requirements.

Currently, the best method for reducing bearings analysis is prevention of the failure. Most antifriction pump bearings fail for one of several reasons: (1) water gets into the oil, (2) automatic oiler is not adjusted properly (this cause is most often overlooked and will continue to produce a short life to failure cycle), (3) product gets into the oil, (4) acidic vapors condense and break down the oil, (5) mounting techniques or fits are improper, (6) new hearing is defective. The solution to high-humidity problems and problems with acidic units is the use of an oil mist lubrication system. If this cannot be economically installed, an aggressive preventative maintenance program on a monthly to every-other-month basis is required.

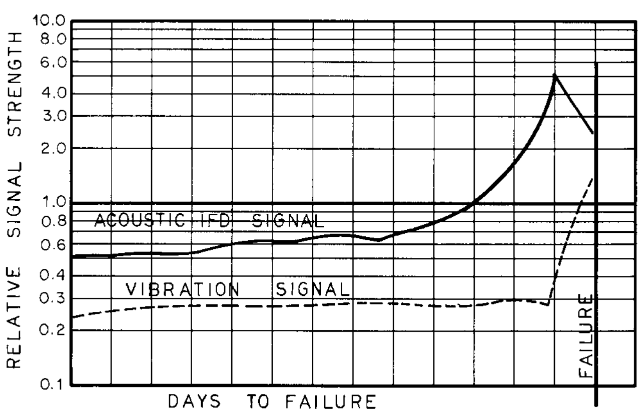


FIGURE 23 Relative signal strength versus days to failure for acoustic IFD and conventional vibration monitoring methods

Baseplates With the change from low rotating speeds and cast iron baseplates to the less rigid fabricated steel baseplates and higher rotating speeds, higher operating temperatures, and larger impellers have greatly increased the probability of baseplate vibrations, distortion, and a decreased stiffness for rotor dynamics. Reference 6 has increased coverage for piping loads and the option of a heavy-duty baseplate over the proposed standard baseplate. The reference specifies a standard baseplate and pedestal support that are twice as rigid as specified in the fifth edition of the reference standards.

Baseplate vibration problems can be resolved at the design stage or during construction. Engineering specifications should call for leveling screws, grout filling holes [4 in (10 cm) minimum, 6 in (15 cm) preferred for each bulkhead section], venting holes [$4\frac{1}{2}$ in (11 cm) holes to each bulkhead section], and corrosion protection. A check of the outline dimension approval drawing should also be made for proper grout hole placement (bulkhead and cross bracing must be shown on the drawing). Construction specifications should call for proper baseplate preparation before grouting.[8] API pumps should have epoxy grout bonding the baseplate to the concrete foundation.

After proper cure, the baseplate should be tapped for voids, especially between and under the pump centerline supports.

> EXAMPLE A high vertical vibration occurred on the coupling end bearing at vane passing frequency on a multistage volute pump. The vertical vibration was 2.5 times the horizontal; thus, a check of the bearing pedestal and baseplate was in order. The hammer test on the pan of the baseplate under the bearing pedestal showed complete lack of grout. Vertical vibration on the pan was 1.8 times the horizontal. Regrouting eliminated the pan vibration and reduced the vertical vibration to one-half of the horizontal. Lack of proper stiffness from the baseplate can and will lower some pump critical speeds into the operating range.

When a complete analysis is being done on a problem pump, take several pedestal readings (top, middle, and bottom on the side and end) and several readings on the pan of the baseplate. The middle, bottom, and pan readings should show good attenuation.

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