Machinery MESsages

THE DYNAMICS OF A VERTICAL PUMP

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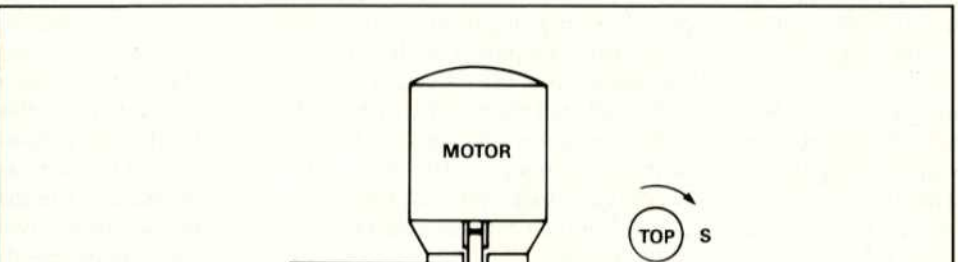

Figure 1

Legend

First Letter: Vertical location of measurement point (i.e., A, B, C, etc.)
Second Letter: Horizontal location (S for south side of pump; W for west side)
Third Letter: Type of transducer (D for displacement proximity probe; V for velocity transducer)

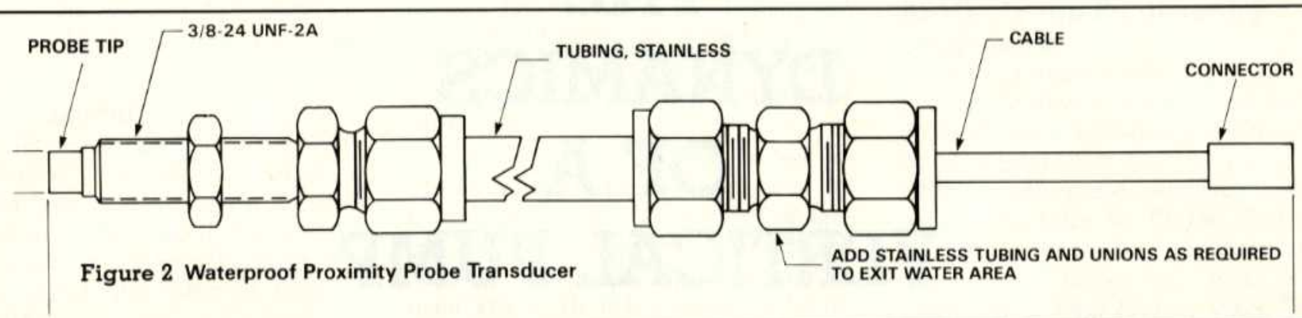

Figure 2 Waterproof Proximity Probe Transducer

Little is known about the mechanical behavior of vertical pumps, though this type of pump is typical of water pumps used throughout the power generation industry.

There are two reasons so little is known about a vertical pump's dynamics. One reason is the difficulty in modeling the rotor system of a vertical pump, all but limiting research to field testing under actual operating conditions. Yet this hasn't always been possible, which leads to the second reason: only recently has the application of submersible proximity transducers made possible the measurement of pump dynamics under field operating conditions.

Because so little is known about the dynamics of vertical pumps, a study was conducted by Bently Nevada's Mechanical Engineering Services (MES) under contract to the Bently Rotor Dynamics Research Corporation (BRDRC). The purpose of the study was to determine the proper selection and installation of vibration monitoring transducers for vertical pumps and to measure the effects of imbalance, misalignment, and hydraulics on their performance and reliability.

Machine description

A service water pump at Public Service of Indiana was chosen for the study. A number of shafts on this specific pump had failed in recent years for unknown reasons. One purpose of the study—besides the ones already discussed—was to determine the cause or causes of past shaft failures on this particular pump.

A drawing of the pump and motor is shown in **Figure 1.**

The pump's lower bearing consists of a grease-lubricated bronze bushing supported by the pump end bell. Intermediate "bearings," supported by the pump vertical column, are water lubricated and lined with scalloped rubber. This type of bearing is sometimes referred to as a "cutlass" bearing or bumper. The couplings between the shaft sections are rigid and transmit torque through two keys spaced at 180 degrees.

The pump is one of three identical pumps connected to a common header. Flow is regulated by a control valve in the discharge piping of each pump.

The rated pump performance is 30,000 gallons per minute at 140 feet. Rated brake horsepower is 1,250 hp at 720 rpm.

Transducer selection

Since very little information is available on the rotor dynamic characteristics of vertical pumps, both proximity and velocity transducers were installed at each bearing and at the motor coupling. Transducer mounting locations are shown in **Figure 1.** Transducers are identified using the three-letter code shown in the legend with **Figure 1.**

Transducers at locations C and D are below river level, dictating the use of a waterproof transducer design. In addition, the transducer had to be designed to withstand the abrasive action of sand in the flowing water. The probe design used is shown in **Figure 2.**

Static testing

The pump's rotor system was tested experimentally for both lateral and torsional resonances. Testing was performed by suspending the instrumented pump shaft, motor rotor, and pump impeller vertically and perturbing the assembly with an instrumented force hammer. The pump speed of 720 rpm dictates that the system will run above the second and below the third lateral balance resonance.

The first torsional resonance of the rotor occurred in the 690 to 765 cycles per minute range. Since this range includes the running speed, torsional analysis was warranted. Strain gauges with telemetry were employed to measure shear strain. Values were extremely low—well below the fatigue limit—indicating a torsionally well-damped system.

Field testing

The pump impeller was modified to allow the addition of balance weights to the impeller blades without pulling the pump. This was accomplished by using a scuba diver to swim into the pump intake and add or remove weights. The pump impeller was then shop-balanced and installed in the pump case. The pump was assembled with the proximity and velocity transducers mentioned earlier.

Extreme care was taken to align the pump and motor. The axial position of the pump was set to the manufacturer's specification, establishing the clearance between the leading edge of the impeller and casing.

Imbalance

The balance condition of the impeller was changed while holding the other variables constant. The amount of imbalance added was 80 oz.-in., resulting in a centrifugal force equal to 84 pounds. For a ▶

rotor weighing 1,000 pounds, this equals 8.4 percent of the rotor weight.

Figure 3 shows the orbit and time base plots of shaft motion with and without the imbalance weight. These measurements were taken at the lower bearing (location D). Note that the plots clearly show the effects of unbalance. Similar plots at the upper bearing (location B) do not show a significant change.

Misalignment response testing

The alignment between the motor shaft and pump shaft was changed by adding a 0.010-inch shim under one side of the motor. This results in an angular misalignment of approximately two degrees. Radial alignment changes are impossible with this type of coupling due to a babbitted fit between the coupling halves.

Orbit and time base plots of shaft motion at location B are shown in **Figure 4**. The elliptical shape of the orbit indicates a preload caused by the misalignment. Shaft misalignment is not apparent in the spectrum plots of the same data **(Figure 5)**. Since the misalignment manifests itself as 1X vibration, no strong indication of traditional 2X vibration is indicated on the spectrum plot. The preload was not as apparent at the bottom bearing, as expected.

Hydraulic response testing

The pump was run with two different clearances between the leading edge of the impeller and the casing. Tests were conducted with the design clearance of 0.043 inch and an excessive clearance of 0.200 inch. The discharge pressure decreased by 20 psig with the increased impeller clearance, which is an indication of the amount of recirculation and efficiency losses.

With design clearance, spectrum plots of shaft vibration data from the lower bearing show strong subsynchronous response at 372 cycles per minute **(Figure 6)**. This frequency is the first bending natural frequency (critical) of the pump shaft. When the clearance was increased, the subsynchronous vibration became less stable and occurred over a broader frequency range.

As observed earlier with the balance response testing, the transducers located ▶

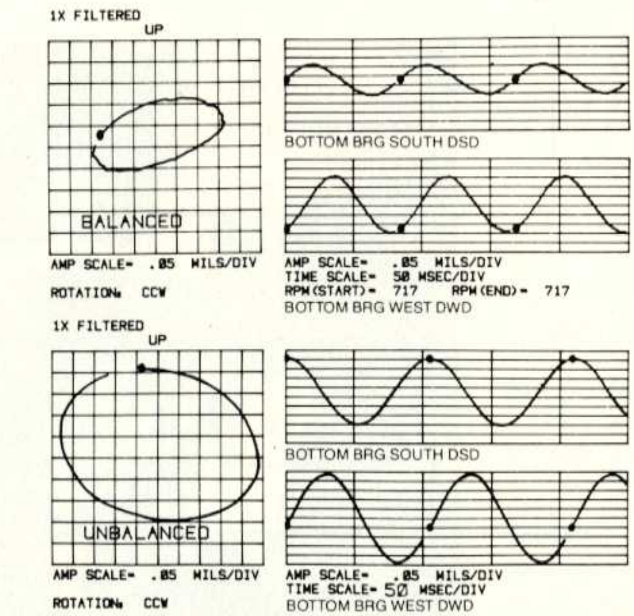

Figure 3

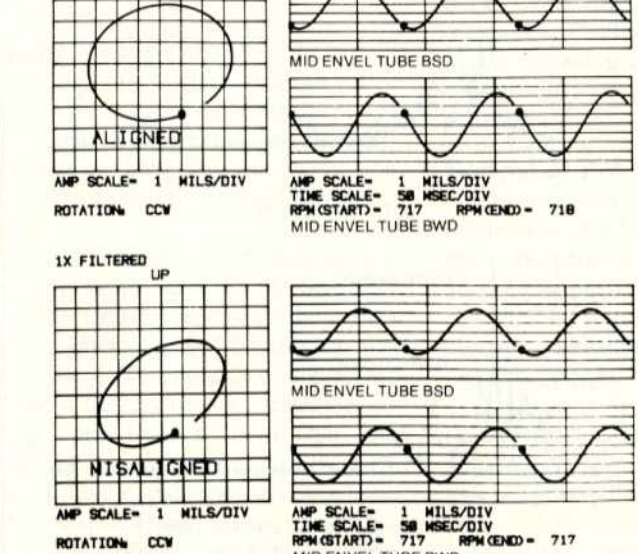

Figure 4

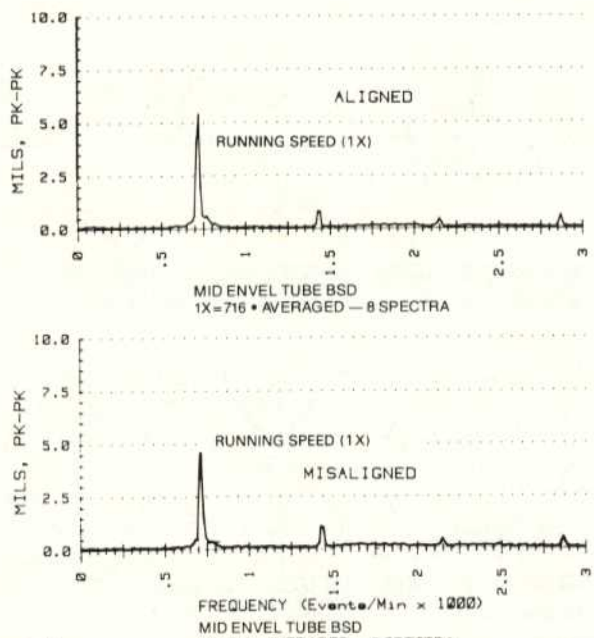

Figure 5

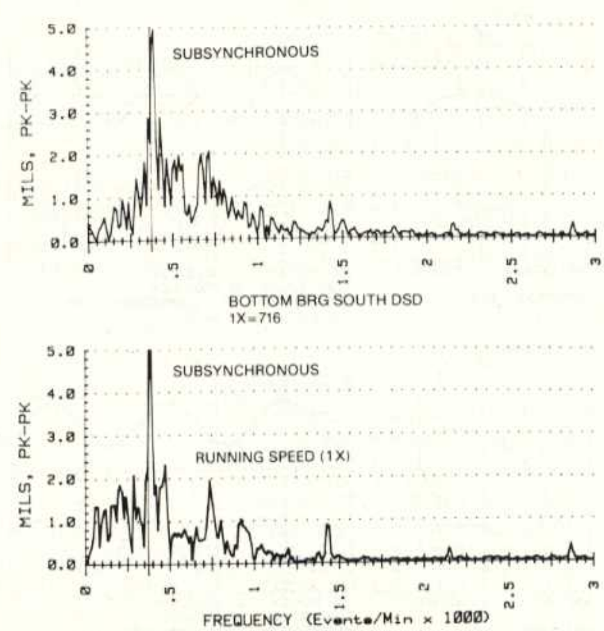

Figure 6

near the motor were unable to monitor the motion at the impeller. The subsynchronous activity on the impeller was not measurable at the coupling.

The hydraulic response testing determined that the re-excitation of the first balance resonance by the subsynchronous vibration was the cause of the recent shaft failures. These failures were identified by metallurgical analysis as bending fatigue failures.

Even so, several unanswered questions still remained:

1. Why is the vibration higher with design clearance than with excessive clearance?
2. Why did the pumps operate without shaft failures until recently?
3. Why is the discharge pressure only 62 psig when rated pressure at river level is 85 psig?

The answer to these questions was discovered by researching the maintenance records. It was found that the start of shaft failures was preceded by a change in impeller supplier. Recent impellers were not purchased from the original equipment manufacturer (OEM), but from another vendor. The OEM has since indicated that subtle changes in the blade attack angle could cause the re-excitation of the first bending natural frequency (critical) due to vortex shedding.

Conclusion

Our study of the dynamics of a vertical pump has led to two major conclusions.

The first conclusion is that adequate machinery monitoring of long vertical pumps cannot be accomplished without measuring the motion of the shaft near the impeller. This pump was analyzed on many occasions using transducers installed at location A. These transducers failed to detect the large subsynchronous vibration occurring at the first balance resonant frequency of the shaft with the pump at running speed.

The second conclusion is that hydraulic effects appear to be several orders of magnitude higher than imbalance and alignment effects. All diagnostic effort should include the correlation of hydraulic data. ■