Solving Pulsation Induced Vibration Problems in Centrifugal Pumps

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When a series of six main oil line pumps were expeiencing severe vibration problems, a solution was needed. These pumps were installed on two large oil platforms as part of a simultaneous drilling and production program in an offshore oil field. The pumps experienced high vibration and noise levels and were generally unreliable with numerous seal failures as well as failures of the attached small bore piping and gage lines.

The root cause of the failures was identified as an acoustic resonance of the pump internals excited by blade pass frequency. The solution was to change the number of impeller vanes, which was relatively simple compared to other options investigated.

BACKGROUND

The pumps were single stage, double volute, double suction centrifugal machines. The nominal impeller diameter was 13 inches, with an eight-inch diameter inlet nozzle and a six-inch diameter discharge nozzle. They were driven by gas turbines through a speed reducing gear. The turbines were rated for 1100 horsepower at a pump speed of 6000 rpm with a developed head of 1800 ft. The desired operating speed range was 80 to 100 percent (4800-6000 rpm).

When the pumps were first installed, low production rates resulted in low operating speeds and relatively trouble-free operation. As production rates increased, the pumps were required to operate at higher speeds and were found to produce excessive noise and vibration in the 5500-6000 rpm range. Chronic seal failures were experienced as well as fatigue failures of the attached small bore piping and instrument tubing. A chronology of the problems experienced on one of the platforms over a one-year period is shown in the table above.

TABLE 1: PUMP PROBLEMS OVER A ONE YEAR PERIOD

Date	Problem
July 15	P350 Replace O/B Seal
July 27	Gauge Failure Bourdon Tube
July 30	Gauge Failure Bourdon Tube
August 14	P350 Replaced O/B Seal
November 30	P350 Drive Coupling Failed
December 1	P340 Replaced I/B Seal
December 6	Sensing Line Failure
December 10	Sensing Line Failure
December 21	Sensing Line Failure
December 22	P350 Retube Lines
January 5	P340 Replaced I/B Seal
February 8	P350 Replace Flex Hoses Suction Transm. + S/D Switch
February 20	Suction Transm. Problem P340 Replace I/B Seal
March 11	Sensing Line Failure
April 6	P350 Replace O/B Seal
May 2	Repair Transmitters
June 8	MOL Pump Temp Recorder Failed
June 13	P350 Replaced I/B Seal
June 14	P350 Switch for O/B Seal Failure
July 12	P340 Replaced O/B Seal
July 14	P350 Suction Body Bleed Valve

As a means of reducing the pulsation levels generated by the pump, the impeller to cutwater clearance was increased by trimming the cutwater. This was suggested by the pump manufacturer and resulted in a marked reduction in the audible noise. However, the seal and piping failures persisted when the pumps were operated in the upper end of the speed range.

Various piping modifications were made to reduce the piping thermal stresses and pump nozzle loads due to the thermal growth of the suction and discharge piping. This did not resolve the failure problems, and the pumps became an industrial relations and safety issue. At this point, purchasing new pumps was given strong consideration.

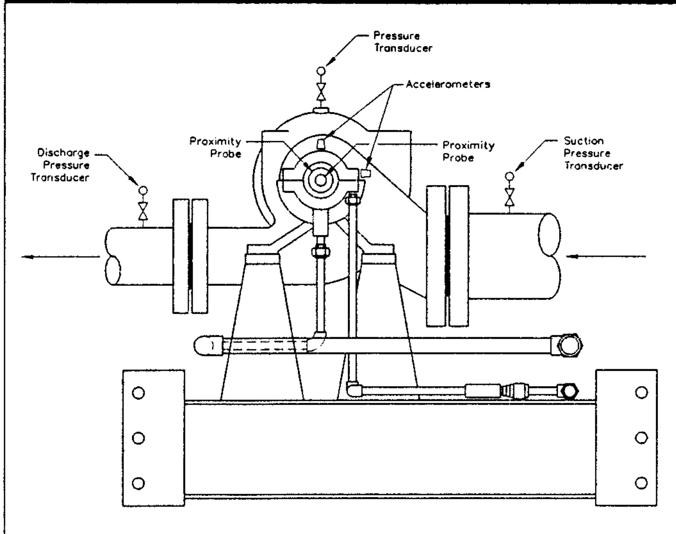
FIELD INVESTIGATION

A detailed field investigation was made to identify the root cause(s) of the failures. A sketch of the pump is shown in Figure 1. Bearing housing vibration levels were measured using accelerometers, as well as shaft-relative-to-housing vibrations levels using non-contacting proximity probes. Pressure taps were installed in the suction and discharge piping as well as the pump case. Piping vibration and sound pressure level measurements were also made. All of these data were acquired over the operating speed range of the pumps.

In addition to the operating vibration, noise, and pulsation data, impact tests of the bearing housing structure and various piping components were conducted with the pumps shut down to identify mechanical natural frequencies.

The bearing housing vibration spectra were plotted versus pump speed in a speed rastor (cascade) format as shown in Figure 2. Two resonant frequencies were excited by blade pass frequency (5x running speed) as shown. The noise levels near the pump showed similar spectral content. Initially, it was not known if these frequencies were mechanical natural frequencies or acoustic natural frequencies. The discharge pulsation data indicated that these were acoustic resonances. Impact tests with the pump

FIGURE 1: PUMP TEST POINT LOCATIONS



shut down verified that mechanical natural frequencies of the pump components (i.e., bearing housings, shaft, etc.) were not involved.

The discharge pulsation data from the same speed run was also processed using a tracking filter to plot amplitude and phase of the 5x running speed vibration versus frequency (Bode plot). This data, shown in Figure 3, clearly identified two response peaks which represented the acoustic natural frequencies of the system. Note that the peak amplitudes for this pump were 90-140 psi (p-p), which was considered excessive in the 550 psig discharge piping. Similar data from other pumps showed pulsation levels in excess of 185 psi (p-p).

The acoustic natural frequencies were similar in all six pumps tested. Two modes were found within the operating speed range. Each pump seemed to have a predominant response peak near 450-480 Hz range that appeared to be independent of the attached piping. Additional resonances in the 250-410 Hz range appeared to be influenced by the attached piping, which varied slightly from pump to pump. The fact that the frequencies were different on some pumps indicated that the attached piping configuration had some influence on these modes, or that some difference in the pump internal dimension might exist.

For this reason, a test was made with an orifice installed in the discharge flange of one pump. This discharge was on a 6" line, therefore an orifice of 3" was chosen since a diameter ratio of 2:1 usually results in a noticeable change. A comparison of the discharge pulsation with and without the orifice is shown in Figure 4. The two spectra plotted in Figure 4 are "peak-hold" spectra. This means that the maximum amplitude at each frequency is stored during the chosen sample time. For this plot, the analyzer was allowed to sample the pulsation data as the speed was varied from 4000-5850 rpm. The data was then repeated with the orifice installed. During the speed range of 4000-5850 rpm (67 to 98% speed), the BPF ranges from 333-488 Hz as noted on Figure 4. The effect of the orifice was to lower the first mode from approximately 380 Hz to 340 Hz and attenuate the amplitude of the 450 Hz mode from 140 psi (p-p) to 50 psi (p-p).

This was a significant improvement in the blade pass frequency pulsation and related vibration. Unfortunately, the restriction in the discharge line caused excessive pressure drop which could not be tolerated for long-term operation. Therefore, it was recommended that the orifice be removed.

In order to solve the acoustic resonance problem, either the acoustic natural frequency or the excitation frequency had to be changed. Because of the difficulty in identifying the complex acoustic mode shapes, the most practical solution appeared to be changing the blade passing frequency of the pump by changing the number of impeller vanes.

FIGURE 2: PUMP BEARING HOUSING VIBRATION 4500-5950 RPM

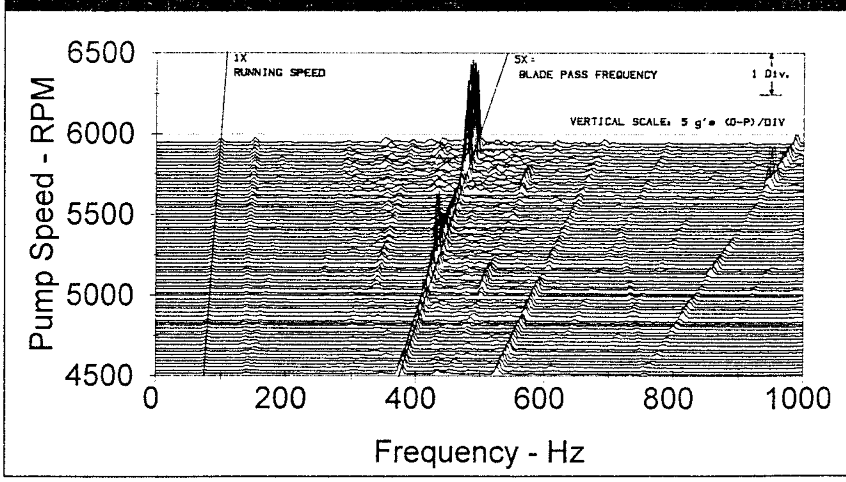


TABLE 2: BEARING HOUSING VIBRATION

Location		Bearing Housing Vibration (ips, 0-p) 5-vane	7-vane
Inboard	Horizontal	0.83	0.12
	Vertical	0.35	0.04
Outboard	Horizontal	0.12	0.02
	Vertical	0.04	0.04

FIGURE 3: DISCHARGE PULSATION TRACKING 5X PUMP SPEED

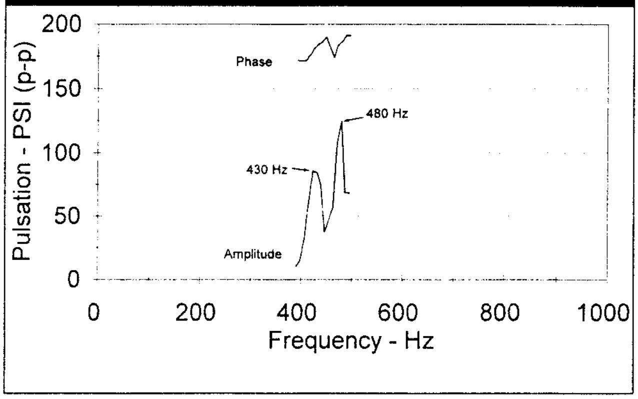
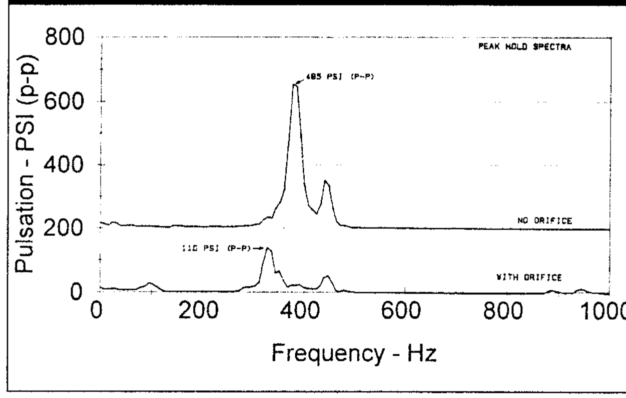


FIGURE 4: EFFECT OF ORIFICE PLATE ON DISCHARGE PULSATION



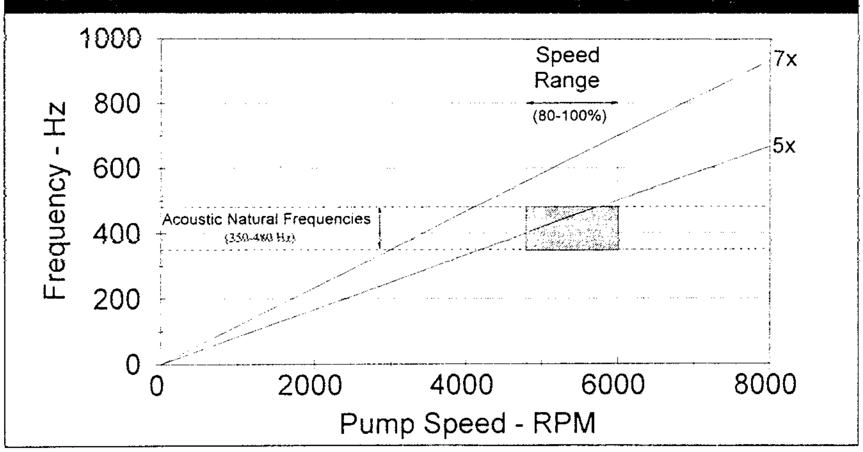
SOLUTION

An interference diagram was constructed as shown in Figure 5 to evaluate the acoustic resonances. This shows the relationship between the excitation frequencies and the acoustic natural frequencies. The range of acoustic natural frequencies measured was 350 Hz to 480 Hz and is represented by the horizontal lines in the figure. The desired operating speed from the pumps was 4800 to 6000 rpm and is represented by the vertical lines as noted. The intersections of the horizontal and vertical lines develop the cross-hatched rectangle, which is the area that should be avoided by any system excitation frequency. The blade pass frequency (5x running speed) excitation is represented by the upward sloping diagonal line in Figure 5. It intersects the cross-hatched area. The blade pass frequency excitation line for a seven-vane impeller (7x running speed) is also shown in Figure 5. This line does not intersect the cross-hatched area, which indicates that the acoustic natural frequencies would not be excited within the desired operating speed range, if a seven-vane impeller were used.

> ***"The discharge pulsation data...clearly identified two acoustic natural frequencies of the system."***

The pump manufacturer was able to supply seven-vane impellers for the pumps, which resulted in significant reductions in overall vibration and noise levels. The detailed pulsation and vibration tests were not repeated after the seven-vane impellers were installed, since the improvement was evident. However some overall bearing housing vibration levels were available before and after the modifications (Table 2). These pumps have operated reliably for seven years as of this writing with only routine maintenance required. ■

FIGURE 5: INTERFERENCE DIAGRAM MAIN OIL LINE PUMPS



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