Vibration Study and Root Cause Analysis

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An 1,800 MW power station in the Midwest experienced constant trouble with four of its four-stage boiler feed pumps, which were driven by steam turbines and operated at variable speed to meet the required plant load.

According to the plant, these pumps had always exhibited high vibration. In 2003, the plant approached an aftermarket service provider to resolve this problem. Unfortunately, the pumps were not fitted with vibration monitoring equipment, so vibration trends were not readily available. Since the problem existed on all four pumps, a systematic analysis was recommended to determine the root cause. The plant and repair shop developed a plan to a) undertake a field study and conduct a vibration trend analysis, b) identify possible causes by performing hydraulic and structural analysis, c) analyze suitable modifications and d) rebuild the pump accordingly.

Field Observations

1. The vibration level varied by machine, from 0.3 ips to a maximum of 0.7 ips.
2. Maximum vibration levels were at one times running speed on the inboard bearing housing and two times running speed on the outboard bearing housing.
3. The vibration level was ~0.15 ips at impeller vane pass frequency at five times running speed (the impeller had five vanes).
4. In the past, the suction impellers were repaired or replaced every two years due to impeller vane erosion damage.

Analyses and Results

Engineers believed that the root cause for vibration problems could be attributed to the combined effect of structural stability and hydraulic phenomena. For the purpose of analysis both of these effects were studied separately.

Results of Structural Analysis

The recommendation was that the plant employ a third party consulting firm specializing in vibration analysis to conduct a field and analytical study to determine the full operating deflection shape (ODS) of the pumps. Impact testing was also performed to determine the natural frequencies and mode shapes of each unit. The ODS study revealed the following:

1. Modal impact testing showed a lightly damped natural frequency of approximately 88 Hz on three pumps and 98 Hz on the fourth pump. As the typical running speed was between 4,400 rpm (73.3 Hz) and 5,400 rpm (90 Hz), the pumps were operating at or near critical speeds.
2. Analysis showed that the one times running speed vibration was a result of the entire pump twisting on its pedestals, and not a result of localized motion of the bearing housings (see Figure 1). The pedestals were deforming their cross-sectional shape to produce this twisting motion. The inboard bearing housing moved in-phase with the barrel at this frequency. Applying a fix to the coupling guard or inboard bearing housing could slightly reduce vibration levels, but did not address the vibration's root cause.
3. The outboard bearing housing vibrated in the vertical direction at two times running speed due to a nearby vertical natural frequency of the outboard bearing housing. In some cases, the amplitude of this vibration exceeded that of the one times vibration on the inboard bearing housing.
4. The outboard bearing housing vibrated in the horizontal direction at vane pass frequency (five times running speed = 375 Hz to 450 Hz). There was a natural frequency at approximately 380 Hz that produced at least some significant vibration over this entire speed range.

Figure 1. Data acquired during testing shows the pump twisting on its pedestals.

Recommendations based on Structural Analysis

1. Structurally stiffen the pump pedestals in the direction of deformation as identified in the modal and ODS analysis. This could be done either by filling the hollow pedestals with concrete or by

welding stiffening ribs in the appropriate locations. The idea was to prevent the rectangular cross-sections of the pedestal (as viewed from the top) from deforming into parallelograms.

2. Stiffen the outboard bearing housing support (drip pocket) so as to shift its natural frequency higher and reduce vibration in the vertical direction. The drip pocket that supports the bearing housing could be modified to provide support at the top of the bearing housing as well as the existing support at the bottom of the bearing housing.
3. Stiffening the outboard bearing housing in the horizontal direction to counteract vibration in the horizontal direction should not be attempted. Stiffening in the horizontal direction would increase the natural frequency, which is excited by vane pass frequency, and would risk increasing the amplitude of the vane pass frequency vibration. Another approach to reducing the vane pass frequency would be to install impellers with a different number of vanes.

Results of Hydraulic Analysis

Before the analysis, the dismantled pump components were inspected. Figure 2 shows the damage in the inboard bearing. Figure 3 shows the erosion damage of impeller vanes. Figure 4 shows a diagram indicating sealing gaps on impellers.

Figure 2. The inboard bearing was found to be nearly destroyed due to excessive vibration.

1. The impeller and balancing drum sealing gaps during operation were increased due to vibration. The effect of increased sealing gap meant increasing leakage flow, thereby reducing pump efficiency by an estimated 3.2 percent and reducing damping effect of sealing gaps and decreasing natural frequency of the rotor, which may lead to reducing the margin of critical speed.
2. The impeller had five vanes, and the volute casing had two tongues. Theoretical evaluation showed that this vane number combination will result in unbalanced blade forces, which excite lateral rotor vibrations. Increasing the number of impeller vanes to six, however, would result in high fluctuations of pressure, torque and axial loads leading to high degree of vibration. However, a seven-vane impeller would be optimum.
3. A close look at the eroded suction impeller indicated the signature of suction cavitation erosion. Since this damage was present in every pump, insufficient NSPHA/NPSHR ratios were suspected. This varied from 1.0 to 2.6 depending on operating speed and flow. The lowest ratio occurs at the highest capacity near the BEP. (For high energy pumps, according to Hydraulic Institute, this ratio should be at least 2.0 to minimize erosion damage due to cavitation).

Figure 3. Results of cavitation on the 1st stage impeller.

Figure 4. Though the radial alignment was good, excessive clearances which existed needed to be reduced. Large clearances result in low efficiency and high operating costs.

The low NPSH ratio results in an impeller life of 2.7 years and explains why the suction impellers have been repaired or replaced every two years. According to the plant, little could be done on NPSHA, so changing suction impeller design was the recommended option. Design parameters were agreed upon and the goal was set to increase the operating life of suction impellers to 40,000 hours, or 4.5 years. By varying the suction inlet and vane inlet geometries, several suction impeller designs were studied and one final design was chosen that would increase operating life to four years.

Recommendations Based on Hydraulic Analysis

1. Correct the sealing gaps to the acceptable limit.
2. Replace five-vane impellers with a new seven-vane design.
3. Replace existing suction impeller with a new one designed for better suction behavior and aimed to increase its operational life.

Conclusion

The plant accepted the design recommendations, and the pump was completely rebuilt to stringent acceptance criteria. Installed in 2005, the redesigned pump continues to be in operation today without any vibration problems. It was a challenging job, but addressing the problem at its root and exploring means to correct the problem was the key to success.

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