Troubleshooting Large Turbine/Compressor Problems

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Steam turbines are often the drivers for large electric generators, pumps, and compressors in utilities and other process industries. The turbines weigh several tons and generate power from just a few kilowatts (kW) to many megawatts (MW). It's common for these machines to run for several years without repairs or bearing replacement as long as they are well monitored and maintained. However, it is essential that the rotors in these large machines retain good alignment, balance, and lubrication to avoid vibrations that can cause extensive damage. For this reason, built-in seismic and displacement vibration sensors continuously monitor the journal bearings to detect adverse vibration shaft orbit characteristics.

Many different kinds of problems can cause turbine rotor vibrations. Some common ones are: oil whirl, oil whip and rub. Their symptoms are similar, and maintenance technicians often have difficulty distinguishing one from another. Oil whip is a condition arising from a more basic problem called oil whirl, which in turn results from an uneven oil distribution (oil wedge) around the shaft in the journal bearing. Machine misalignment, improper oil viscosity, or an incorrectly designed bearing can cause this anomaly. It often generates a vibration at a frequency that is a subharmonic of the full rotor speed.

Rub is a condition where the turbine rotor contacts stationary components, including seal rings and inside diameters of the bearings. Running the machine for a short time usually clears the problem, but in some cases, the wear continues to generate larger vibration amplitudes that may become unstable and damage the machine.

Potential Solutions. Engineers and technicians typically use data acquisition systems to measure, monitor, and record time-versus-amplitude waveforms of the turbine-rotor shaft displacements within the journal bearings. The time history and spectral characteristics of the resulting plots help them determine which fault is the cause of the vibration condition.

Nelson Watson, president of Watson Engineering, has been consulting in the utilities and process industries for many years. One of his clients was faced with exactly this situation recently, and Watson was called upon to diagnose and remedy the problem.

The test system that resided at the site of the turbine compressor train was similar to those widely used by maintenance people and engineers in process industries, but in this case, some of the permanently embedded sensors used to monitor vibration and generate orbit and phase plots were defective.

IOtech Solution. A 12,000-hp steam turbine driving a centrifugal compressor had been running without a problem for several years. Two years ago, the machine developed a vibration that damaged the bearings, which were subsequently replaced with a modified version.

The bearing malfunction came from a discharge check-valve failure that forced the compressor to rotate in reverse while the system was being switched to another unit. Soon after the incident, the turbine developed another high-amplitude vibration. The machine would shut down due to the vibration before reaching operating speed each time it was energized.

Watson set up his IOtech ZonicBook as shown in Figure 1 to record signals from radial proximity sensors located at both turbine and compressor bearings. A tachometer signal was connected to the governor speed sensor and calibrated to the turbine speed using a ratio function in the ZonicBook software. Because the client's embedded sensors were not operating, it was not possible to perform the typical orbit and phase analysis normally done under these conditions. In addition, the speed sensor stopped operating when the turbine vibration became excessive near the maximum running speed.

Watson connected the ZonicBook to the eight proximity sensors, two on each of the four bearings, and four tachometer signals, one at each of the four main bearings. The total recording time was about 10 min. The instrumentation was able to record vibration signals that ranged from about 0.8 mils to 10 mils, from a slow roll to a maximum speed of 4200 rpm. As shown in Figure 2, the initial vibration amplitude during the slow roll was about 0.8 mils, then it went through a critical speed where the amplitude reached 3 mils, and finally, it backed down to less than 2 mils. As the speed increased, the amplitude reached approximately 10 mils.

Watson used the ZonicBook's software to produce a waterfall plot (Figure 3) of vibration frequencies and amplitudes that characterized the behavior of the turbine. The constant speed lines on the right side of the plot start at 1569 rpm and extend to 3824 rpm. The first critical speed appears to be about 2100 rpm, which produces 35-Hz peaks, although the manufacturer claims the critical speed should be 2850 rpm. The speed sensor stopped functioning and generated zeros when the vibration amplitude suddenly increased above 3824 rpm. The diagonal row of peaks is recorded between 1569 rpm and 3824 rpm and is the only significant vibration group observed until the running speed passes 3824 rpm. At 4200 rpm and higher, it produces peaks at 35 Hz and 70 Hz, the same frequency spikes that occur at half speed or 2100 rpm.

Figure 4 shows a spectrum when the machine is running at 4200 rpm (producing peaks at 70 Hz). The principle peak at 35 Hz is produced at half speed, 2100 rpm, although harmonics appear at multiples of the critical speed. The maximum amplitude of the turbine shaft in the horizontal direction was 6.5 mils. The amplitudes of vibration along the turbine and compressor shafts were relatively low throughout the data collection process.

Analysis. The analysis begins with a closer look at bearing integrity. Because

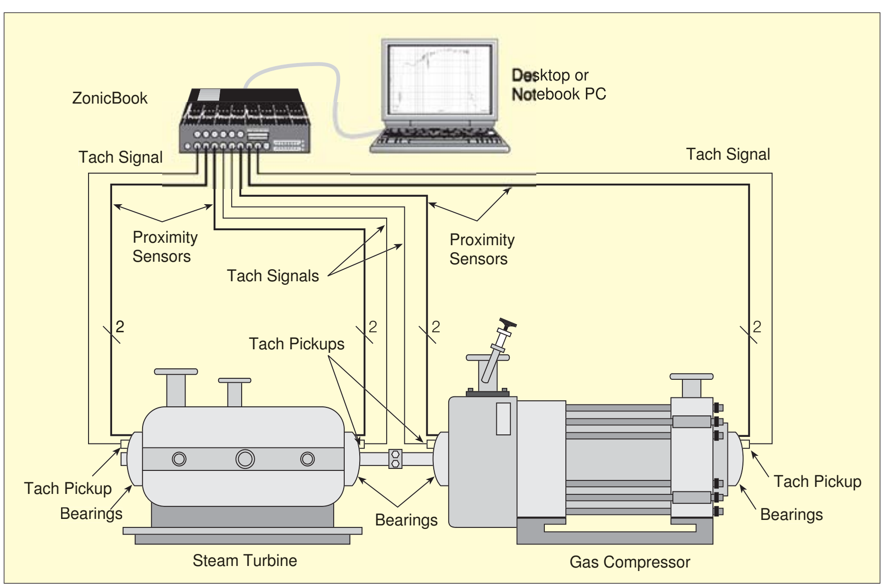

Figure 1. Steam turbine and gas compressor combinations are one of several different kinds that Watson Engineering analyzes for vibration problems. Other systems in the kilowatt to megawatt power range that develop similar vibrations include gas turbines, electric motors, compressors, pumps, and electric generators.

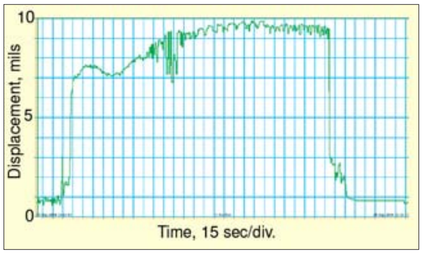

Figure 2. Turbine inboard vertical amplitude spectrum plot. The turbine inboard bearing experiences the greatest vibration amplitude of about 9.5 mils at 4200 rpm.

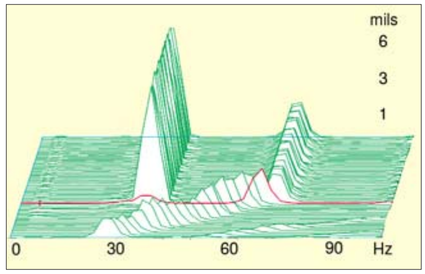

Figure 3. Waterfall plot, turbine inboard vertical, amplitude (mils) vs. speed (Hz) with constant speed lines (rpm). The plot shows vibration frequencies and amplitudes generated at turbine inboard bearings over several constant speeds. Significant vibrations of 35 Hz appear at 2100 rpm, the first critical speed, and 75 Hz beginning at about 3824, just below the 4200-rpm typical running speed.

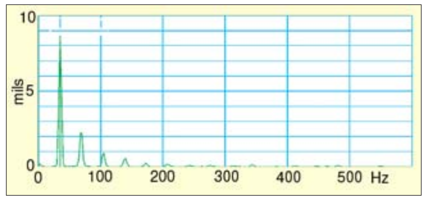

Figure 4. Turbine inboard vertical amplitude vs. frequency. This shows the spectrum when the machine is running at 4200 rpm (75 Hz). The principle peak is at 2100 rpm (35 Hz), along with several harmonics.

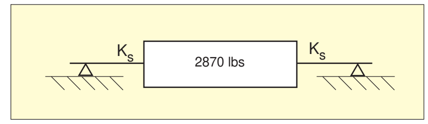

Figure 5. Flexible rotor with rigid bearings model. The diagram shows the force of the rotor shaft on rigid bearings. Using the manufacturer's specifications of 2850 rpm for critical speed, the rotor shaft stiffness can be calculated.

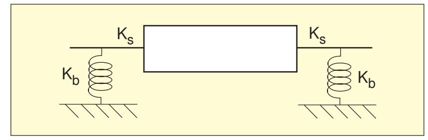

Figure 6. Actual bearing and shaft stiffness model. Actual bearings are not perfectly rigid, so their stiffness is calculated with the measured critical speed of 2100 rpm.

the bearings had been modified before the vibration problem arose, they are suspect and their stiffness needed to be determined. The first actual, critical speed appears to be about 2100 rpm, as determined by the vibration measurements during startup and coast-down. Since the embedded shaft sensor was unavailable to measure phase changes, the critical speed was verified analytically and bearing stiffness calculated in three steps.

First, the rotor shaft stiffness K_s is calculated from the manufacturer's specified critical speed of 2850 rpm. This is the rotor's characteristic only, assuming perfectly rigid bearings. Next, the total stiffness K_t which is the combined bearing and shaft stiffness, is calculated from the measured critical speed of 2100 rpm. Finally, the bearing stiffness K_b is determined from the calculated values of rotor shaft stiffness and total stiffness.

The rotor shaft stiffness is determined from the critical speed equation by solving for K_s. A model of this configuration is shown in Figure 5.

Given the critical speed equation:

$$N_c = (60/2\pi)(gK_s/W)^{1/2} \tag{1}$$

where:
N_c = specified critical speed, 2850 rpm
W = effective rotor weight = 2/3 of total rotor weight, 2870 lb
g = gravity constant, 386 in/sec^2
K_s = shaft stiffness, lb/in
Solve for K_s:

$$K_s = (W/g)(2\pi N_c/60)^2 \tag{2}$$

$$K_s = (2870/386)\left[(2\pi)(2850)/60\right]^2$$

$$K_s = 661 \times 10^3 \text{ lb/in}$$

Secondly, the model is now modified to determine the bearing stiffness, shown in Figure 6. The total stiffness K_t which is the combined bearing and shaft stiffness, is calculated from the measured critical speed of 2100 rpm:

$$K_t = (W/g)(2\pi N_{cm}/60)^2 \tag{3}$$

$$K_t = (2870/386)\left[(2\pi)(2100)/60\right]^2$$

$$K_t = 359 \times 10^3 \text{ lb/in}$$

The relationship among the variables is expressed as:

$$1/K_t = 1/2K_b + 1/K_s \tag{4}$$

where:
K_t = total series spring rate, lb/in
$2K_b$ = bearing spring rate, lb/in
K_s = shaft spring rate, lb/in
Solve for K_t:

$$K_t = 2K_sK_b/(K_s + 2K_b) \tag{5}$$

Thirdly, the bearing spring rate for the two bearings $2K_b$ now is calculated from:

$$2K_b = (K_tK_s)/(K_s - K_t) \tag{6}$$

$$2K_b = \left(359 \times 662 \times 10^6\right)/\left[(662 - 359) \times 10^3\right]$$

$$2K_b = 784 \times 10^3 \text{ lb/in}$$

The calculated bearing stiffness and measured critical speed of 2100 rpm appear to be reasonable values for this machine based on the above procedure.

Oil Whip Theory. The oil whirl condition usually precedes the oil whip condition. Spectral and orbit analyses can be used to identify either situation. When this occurs, usually a sub-synchronous frequency can be measured in a range less than half of rotor speed. Oil whirl produces a distinctive orbit pattern, but because the client's phase-sensitive transducers were not working, the orbit could not be displayed. It would have shown metal-to-metal contact within a bearing. Because the critical speed is close to half of the running speed, it is difficult to determine whether the problem was due to oil whip or rub.

The machine behavior is characteristic of oil whip, since the vibration frequency is at the first critical speed of 2100 rpm for a machine that reaches about 3900 rpm. However, most literature discussing oil whip indicates an increasing vibration frequency that starts from zero until the machine reaches the critical speed. Then the vibration frequency stays constant even though the machine continues to increase in speed. In this case, however, the vibration frequency stays at the running speed until the rotor reaches 3900 rpm. Then the machine suddenly vibrates again at 35 Hz and 70 Hz, and the amplitude increases from 2.0 mils to 7.5 mils. The fact that the machine suddenly jumps in amplitude when the speed is well above the critical speed may indicate another phenomenon taking effect.

Rub Theory. The behavior of the machine also has the characteristics of a rub based on the spectrum plot shown in Figure 4. The first peak appears to be at 2100 rpm, with harmonics at 4200 rpm, 6300 rpm, and so forth. There was evidence of a packing rub when the machine was inspected internally. It is possible that the rub was a secondary fault, since the amplitude reached almost 10 mils before shutdown.

The fact that the critical speed is extremely close to half the running speed makes it difficult to distinguish between oil whip and rub as the primary fault. The most likely fault is oil whip, with a secondary fault of an internal rub when the vibration amplitude increased. The turbine inboard bearing stability parameters appear to have shifted due to changing forces caused by compressor bearing damage.

The bearings were inspected, and one packing-bearing rub was evident in both the compressor and turbine. The babbitt in one compressor bearing was smeared, which reduced the clearance on one side of the bearing. There was an indication of a rub on one of the packing rings in the turbine. The coupling was found to be in good condition.

The most likely condition is oil whip, which caused the rub when the vibration amplitude increased dramatically. The turbine inboard bearing instability appears to have been caused by compressor bearing damage. The instability developed after the bearing failure and was apparently mitigated after bearing repair. The two problems were corrected and the machine ran smoothly.

Conclusion. Nelson Watson uses an IOtech ZonicBook to troubleshoot large turbine and compressor problems that originate from a variety of sources, including unbalanced rotors and bearing problems due to oil whip, and rub. In a typical test, he connects eight proximity sensors to the bearings and records the vibration frequencies and amplitudes. The EZTomas software helps him analyze the data and pinpoint the source of the problem.

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