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Case Study

Importance of Startup Vibration Measurements on Turbomachinery

After installing new or repaired equipment, obtaining baseline vibration measurements during startup and initial steady state operation can be invaluable in diagnosing abnormal machine behavior. Careful study of vibration characteristics can reveal a range of problems as simple as scratched probe surfaces to more serious issues of bearing oil whirl, operation near critical speeds, distortion due to piping strain, and many others.

Oil Whirl

In the case of an air compression train consisting of a steam turbine, a low pressure and a high pressure compressor and an expander, high vibration was seen after startup on the governor end bearing of the steam turbine. Examination of the steady state data at full load indicated the highest component of vibration was at a frequency of 50% of the running speed, as seen in Figure 1. This behavior is indicative of oil whirl in the bearing and if left unchecked can lead to destructive vibration levels and failure of the machine. Having identified the problem, a temporary solution of adjusting the oil supply temperature to reduce the effect of the whirl can be implemented until such time as the unit can be brought down and the bearing modified or replaced with a different design to reduce or eliminate the whirl.

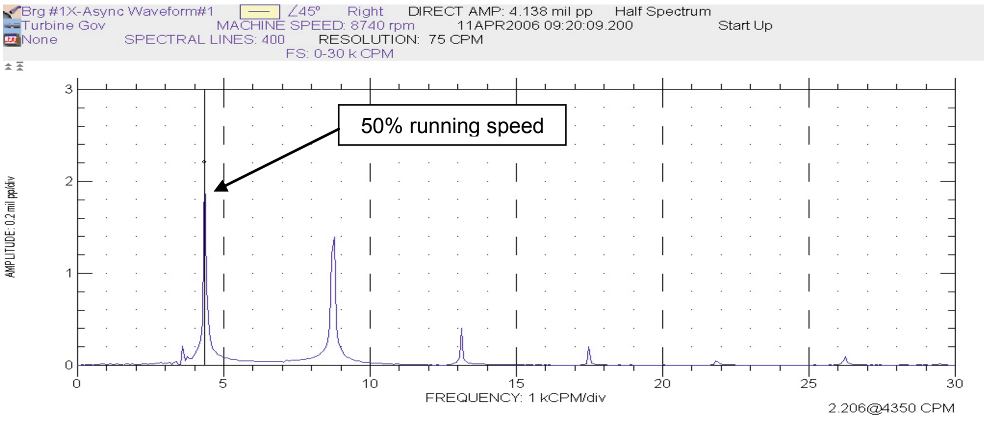

Fig. 1: Spectrum of governor end bearing on steam turbine.

Operation Near Critical Speed

Starting up a large steam turbine/generator train is a risky process in which the shaft heats and expands quicker than the large casings and generator coils. Great care is taken to dwell at certain speeds to heat soak the unit, activate seals and engage the magnetic field. During the startup of a large turbine generator train where the HP turbine had undergone significant rotor and casing rework, excessive vibration occurred while ramping through speed ranges and holding at dwell points that had previously not been an issue. Repair of the high pressure (HP) turbine had resulted in tighter bearing and sealing clearances over the previously worn condition. As seen in Figure 2, the startup vibration data indicated that the critical speed of the HP tur-

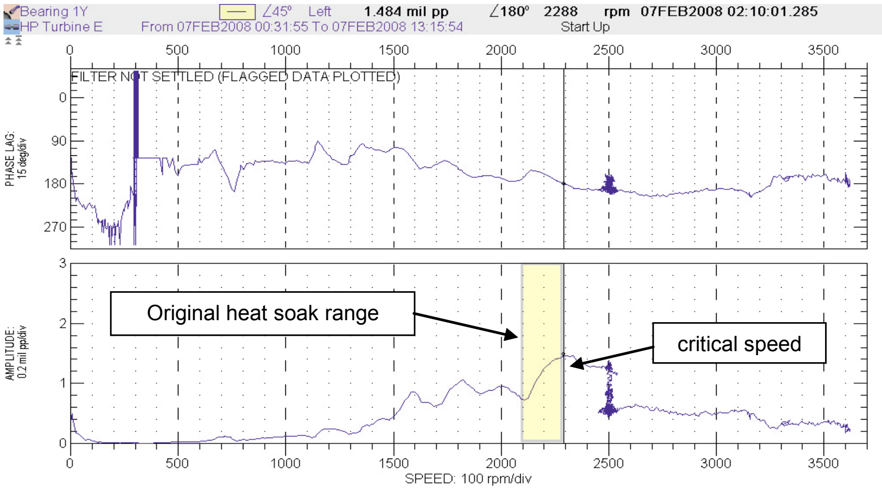

Fig. 2: Bode plot o HP turbine coupling end bearing showing critical speeds.

bine had moved up into the heat soak speed range. Speed was also being ramped too slowly to successfully run through a lower critical speed without experiencing excessive vibration. A simple increase of the ramp rate and shift in the dwell zone speed range allowed the turbine generator to start up smoothly and eventually come online and deliver power.

Piping Strain

Maintaining proper alignment of the rotating components of each element in the machine train is critical, as well as ensuring the minimum amount of strain on the casing due to piping connections. Although many techniques exist to ensure that cold setup conditions are within specification, existing pipe strain and the effects of expansion at full load and temperature conditions can be difficult to determine. Figure 3 illustrates the shaft orbit of the #1 bearing of a gas turbine / pipeline compressor train after the installation of a regenerator. Physical inspections and cold startup vibration data indicated completely acceptable setup and performance, but as load and temperatures increased, excessive vibration occurred. As indicated by the highly elliptical, flat shape of the orbit, the shaft is heavyly preloaded in one direction. The preloaded orbit combined with the fact that this behavior was not present prior to installation of the regenerator led to a conclusion that the high vibration was due to turbine casing distortion from a complex series of piping from the compressor to the regenerator and from the regenerator to the combustion cans.

Misleading Readings

Finally, another source of high vibration readings is occasionally the installed vibration measurement equipment itself. During the startup of a different gas turbine / pipeline compressor train, excessive velocity was measured on the #1 bearing casing. Further examination revealed that the velocity transducer was mounted on an overhung bracket and the natural frequency of the bracket was very close to 2times the running speed of the turbine. Relocation of the transducer to a solid surface on the casing resulted in a large reduction in the measured vibration amplitude. Similarly, scratches on a shaft proximity probe surface can falsely indicate high machine vibration, typically showing high levels at 1time running speed and its harmonics. Such was the case on a high speed pinion compressor driven by a large bull gear. The unit was tripping on high vibration from the impeller end probes although the unit ran very quietly and no other indication of distress was present. Examination of vibration levels across the entire speed range revealed the unlikely result of nearly identical vibration regardless of speed.

Subsequent teardown of the unit revealed a series of scratches on the impeller end probe surface which caused the excessive vibration readings. Detection of faults within the measurement systems themselves can help prevent needless and expensive repair or modification of other machine components.

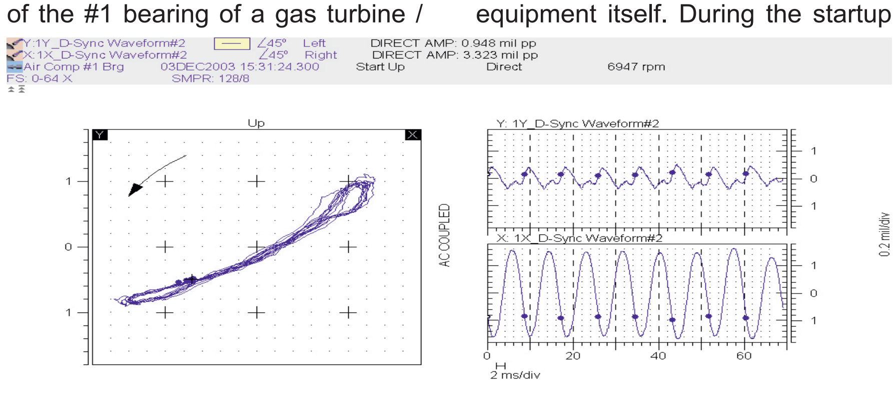

Fig. 3: Shaft orbit plot of gas turbine bearing.

Conclusion

The preceding examples illustrate a small sample of the problems that can be detected when monitoring vibration during startup. Vibration monitoring not only aids in the diagnosis and solution of machine problems, but also provides excellent baseline documentation of those much sought-after instances of uneventful, problem-free startups.

Phil Mosher
Sulzer Turbo Services Houston USA