Problem solving

Maki Onari, William D. Marscher, Eric Olson, and Chad Pasho, Mechanical Solutions Inc., USA, introduce a specialised testing method to resolve problems that can arise in natural gas turbines.

Aeroderivative gas turbines are used to drive LNG compressors. This article describes a problem solving process used to address a turbine problem that was introduced as part of a coupling retrofit. The specialised testing and analysis techniques are also applicable to compressors, as well as other system components (heat exchangers, fractionation towers, etc.) used in LNG facilities, particularly if some sort of unusual resonance, support system, torsional rotordynamics, or acoustic issue is the suspected root cause of the problem. An analysis process for reducing the risk of installing vibration problems is also introduced.

In this particular case, the power turbine of a 22 MW aeroderivative gas turbine driver was suffering from excessive vibration after the original rigid coupling was replaced with a flexible diaphragm coupling (Figure 1). The excessive vibration of 0.50 in./sec. rms on the power turbine (PT) occurred at
2 higher vibration vs non-idle operating conditions. This article discusses the specialised testing used to determine the root cause of the problem in order to provide specific and successful solutions.

Mechanical Solutions Inc. (MSI) performed three different specialised vibration tests as follows:

- Experimental Modal Analysis (EMA) testing to help identify potential natural frequencies and their related damping, and to determine their mode shapes.
- Continuous Monitoring (CM) of vibration and torque data, as well as strain at key structural support elements.

- Operating Deflection Shape (ODS) testing to evaluate the system and individual component relative motion and deflection, particularly at the problem operating condition (idle).

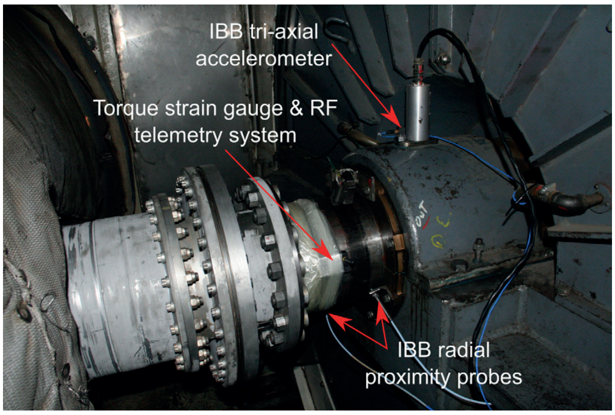

Figure 1. The power turbine (PT) of an aeroderivative gas turbine. Also shown is the recently installed new diaphragm coupling with some of the sensors used for the testing. Excessive vibration occurred after the old rigid coupling was replaced with the diaphragm coupling.

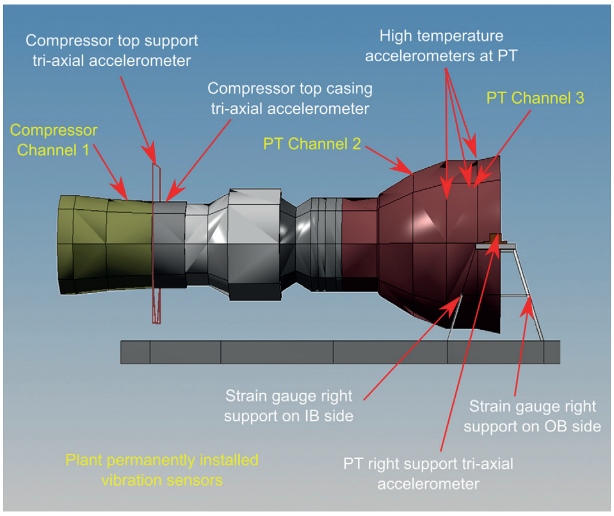

Figure 2. Gas turbine test instrumentation utilised starting at the LP compressor (left) to the PT (right).

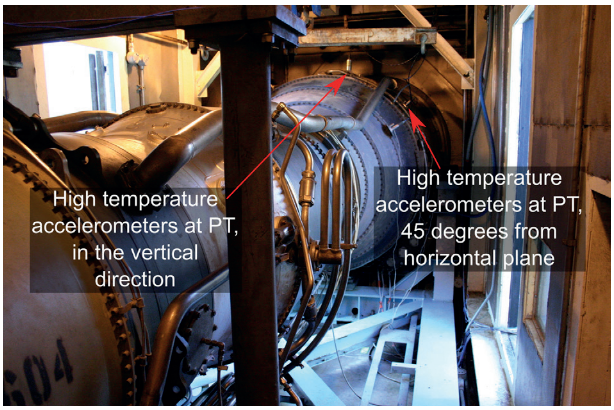

Figure 3. A view of some of the high temperature temporary uniaxial accelerometers.

Each of these techniques will be discussed individually. However, the key advantage of the testing process presented is the ability to combine the results of all three types of testing, in order to fully identify the system's dynamic characteristics. The combined test results are used to determine the problem's root cause and forms the basis for designing or recommending a solution, without resorting to trial and error problem solving. The test results (especially the ODS animations) also allow for more efficient communication with others involved in the problem solving process, including management. Figures 2 and 3 show some of the temporary instrumentation used during the testing. Data from permanently installed sensors is also leveraged.

Experimental Modal Analysis (EMA) testing

EMA testing was performed on the entire train on three different sections: engine, coupling, and the driven machine. EMA testing is performed by using a group of accelerometers to acquire response data, while an instrumented impact hammer is used to excite the engine, the coupling, and the driven machine in a given test direction (horizontal, vertical, and axial directions). The amount of force imparted to the structure at each frequency is determined by the piezoelectric crystal mounted behind the plastic head of the impulse hammer, and this 'input' force is divided into each of the acceleration responses to determine a 'frequency response function' (FRF) between the location/direction of the hammer hit and the location/direction of the accelerometer. The log amplitude of this FRF is plotted vs frequency (the log plotting allows inspection of low response, as well as high response, modes), and the peaks in this plot represent the natural frequencies of the structural system. The EMA testing results also reveal the mode shapes of each natural frequency. EMA testing can be performed with machinery operating using the Time Averaged Pulse (TAP™) technique or with the machinery not operating. In this particular case, the engine was impacted at the low speed compressor in three orthogonal directions. The coupling/shaft was impacted in the lateral/radial and the axial directions.

The EMA test determined that the newly installed coupling and shaft combination had a structural natural frequency in the lateral direction (horizontal) at 114 Hz (Figure 4). This natural frequency indicated only +0.7% separation margin from the HP rotor speed (N2), which is 113.25 Hz or 6795 rpm. Typically, the recommended separation margin between a natural frequency and the excitation source is approximately 10% based on a properly performed EMA test. The damping ratio of this natural frequency was measured to be 2.1%, which represents an amplification factor (AF) of approximately 24. This is a high AF, indicating that even a minimal amount of excitation force could lead to a resonant condition with high (and potentially damaging) vibration.

Continuous Monitoring (CM) test

Figures 2 and 3 show some of the temporarily installed accelerometers, radial proximity probes, strain gauges and a shaft torque gauge using a telemetry system. This CM test gathered vibration, strain, and torque data during idle operation for 2.5 hr and loaded operation for an additional 3.5 hr on the same day. The data indicated that the PT vibration amplitude was reduced by over a factor of 2 when the PT speed increased by only 7% from 6850 rpm to 7355 rpm. Figure 5 shows the trend plot from the strain gauges mounted on each side of the PT support 'A' frame (see Figure 2). This strain indicated that once the turbine is loaded, the strain value increased by a factor of 2 in tension, when the vibration was reduced considerably. This implies that during idle operation, the supports of the turbine need additional pre-load to reduce this high vibration. The correct pre-load needs to be applied between the pylons and the pillow-blocks, accounting for the support underneath the turbine. Also, note the variation, or 'hunting', of the right support, suggesting looseness.

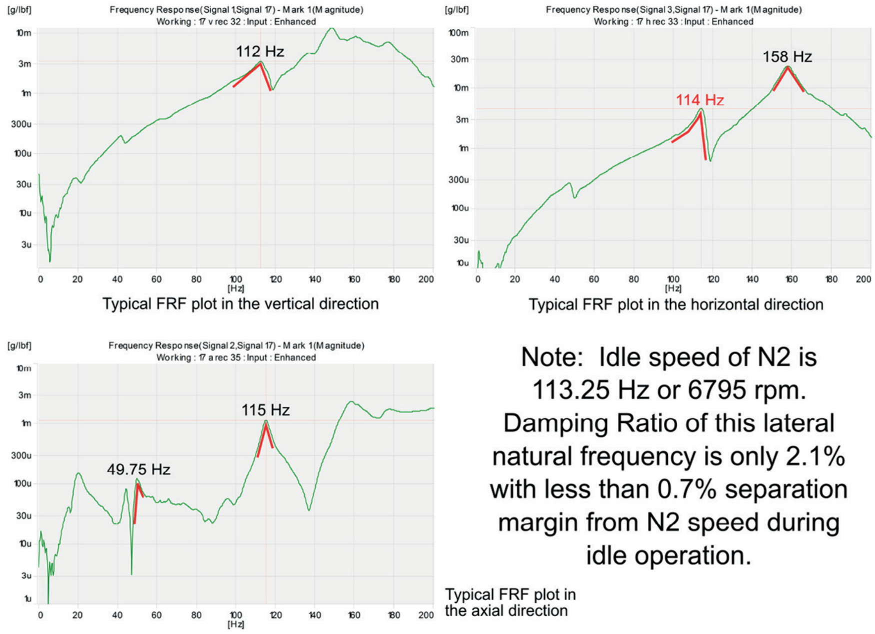

Figure 4. The Frequency Response Function (FRF) resulting from the EMA testing indicating that the coupling had a natural frequency near the HP rotor idle speed creating a potential resonant condition (high vibration).

Operating Deflection Shape (ODS) test

A thorough ODS test was performed in order to determine the most flexible location of the unit, potential soft-foot at idle load, system component relative motion, and other system dynamic characteristics at frequencies and operating conditions of interest. This testing was performed at idle operation, which was the worst condition of high vibration and was not present before the new coupling was installed. The testing process used roving triaxial accelerometers to collect vibration data at 420 measurement locations/directions on the turbine system. The ODS animation is created by specialised software that overlays the vibration measurements onto a 3D computer model of the system created using field dimensional measurements. It is an animation of real world data, not an animation of a Finite Element Analysis (FEA) model, although the test results are useful for calibrating an FEA model.

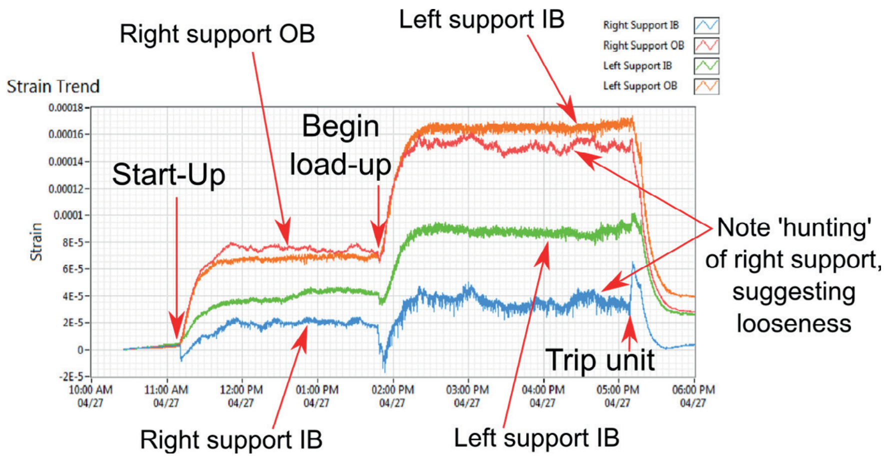

Figure 5. Strain vs time plot for the strain gauges mounted on each side of the PT support 'A' frame. A Lack of pre-load on the support was detected to be a problem contributor.

Figure 6 is an example of one still-frame ODS animation of the entire unit at the idle speed. The animation shows the relative motion of the turbine system components in exaggerated fashion. From the animations, it is evident that there is an excessive orbiting or corkscrewing motion of the PT, mostly in the vertical direction. In addition, soft-foot was detected between the PT pillow-blocks and the 'A' frame support structure. The lateral rocking motion of the left support could be attributed to the severe vertical motion of the PT casing. The ODS results supported the need to increase the pre-load of each support of the PT (lateral and bottom supports), in order to reduce the excessive motion of the unit at idle operation. As for the natural frequency of the

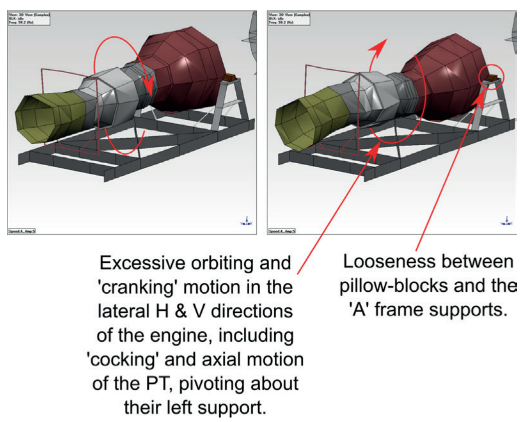

Figure 6. Still-frame of one ODS animation of the entire turbine unit at the synchronous speed. The animation shows the relative motion of the turbine system components in an exaggerated fashion.

coupling at 114 Hz, it was recommended to increase the PT speed by approximately 7% at the idle condition in order to reduce vibration on the right support during idle operation. The ODS animations at full turbine speed indicate much less motion (and vibration), since under load, the machine steady movement under torque tightens this joint, which resulted in lack of support pre-load and resulting looseness.

Conclusions

Based on the modal testing performed on the newly installed coupling, it was clear that the main issue was that a strong lateral natural frequency was at 114 Hz – close enough to the idle speed of the HPT to cause resonant response at 113.25 Hz. As the turbine powered through this speed to a 7% higher speed, vibration dropped dramatically.

Based on the ODS testing, during idle operation MSI detected looseness in the support assembly, which allowed a large orbiting motion of the PT casing mostly in the vertical direction, causing possible damage to the bearings. When combined with the support leg strain gauge strain vs turbine load results, this visibly evident support looseness in the ODS animations indicated a mechanically unloaded condition of the centre support under the turbine at idle. In turn, this mechanically unloaded the right-side lateral supports, consistent with the ODS (Figure 6).

Based on CM of the strain gauge data on the shaft coupling measuring the drive torque, the vibration of the turbine was proven to not be associated with a torsional resonance problem, which could have been a time-consuming and awkward issue to troubleshoot using other methods. The first and second torsional modes could be documented accurately at 21 Hz and 285 Hz, respectively.

Recommendations implemented to solve the problem

The following specific recommendations were implemented to successfully reduce the vibration of the PT during idle operation without creating further problems:

- The coupling resonance issue was resolved by changing the HP rotor speed to increase the separation margin with the lateral mode of the coupling from 0.7% to approximately 10%. Another option was to replace the new coupling with a different new coupling with a slightly different mass.
- The PT casing vibration issue was resolved by increasing preloading of the centre support by adding shims underneath the main support plate. Similarly, the lateral supports were also pre-loaded by a larger amount for safety.

These problems would have been difficult and expensive to efficiently characterise and solve without the use of specialised testing and analysis techniques similar to those described in this article. Experienced turbine maintenance personnel understand the difficulty in trial and error solutions involving a complex situation, such as the one described in this case.

The risk of introducing a resonance problem during a retrofit or for new installations can be reduced so that the type of troubleshooting described above is not necessary. Machinery manufacturers and independent analysis and testing companies are sources for risk-reduction dynamic analysis services prior to 'cutting metal' using modern FEA and rotordynamic analysis techniques. Further details on these types of risk-reduction analysis techniques are deserving of a separate article. **LNG**