COUPLING FAILURE IN ENGINE-DRIVEN PIPELINE COMPRESSOR SYSTEM

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Abstract

Engine-driven compressor units at a gas transmission pipeline station experienced multiple coupling failures. These units had a torsionally soft coupling between the engine and compressor that utilized radial leaf springs. Pressurized oil was supplied to the coupling from an engine bearing through a central bore in the engine crankshaft extension. Torsional damping occurs when oil is forced through internal clearances as the coupling springs flex.

Field measurements taken on one of the units showed that at certain operating points, the design limits of the coupling were exceeded in terms of angular oscillation and vibratory torque. Physical evidence (cracks at a 45 degree angle) and oil analysis containing copper also supported this finding. The worst condition was identified when operating near the first torsional natural frequency (TNF) with some of the compressor cylinders single-acting.

As a short-term solution, restrictions were placed on the compressor speed and load steps to avoid exciting the first TNF. The long-term recommendation involved de-tuning the first TNF below minimum running speed by adding inertia to the system with a compressor flywheel.

The purpose of the case study discussed in this paper is to raise awareness of how a torsional vibration problem can occur in reciprocating compressor systems. Several factors contributed to the coupling failure:

- The first TNF was within the operating speed range.
- Damping was over estimated for the first torsional mode.
- Not all of the compressor load steps were analyzed during the design stage.

Failure of the coupling was not anticipated based on the “ideal” operating condition. Therefore, it is important to perform a comprehensive torsional vibration analysis that encompasses all compressor operating speeds and load cases, uses conservative assumptions, and provides for sufficient separation margins from any dangerous torsional resonances.

INTRODUCTION

Torsional vibration involves the dynamic twisting of shafts while rotating. Excessive torsional vibration can lead to failures of crankshafts and couplings. These failures typically occur at a 45-degree angle to the shaft axis. Torsional problems may not be detected until after a failure because special devices are required to measure torsional vibration. In many cases, machinery experiencing high levels of torsional vibration does not vibrate laterally unless there is a gearbox, or a significant crack has already developed causing torsional vibration to cross-couple into lateral vibration.

Reciprocating machines produce torsional excitation at multiples of running speed (orders or harmonics). Four-cycle engine also produce half orders due to cylinder firing every other crank revolution. When operated over a wide speed range, it becomes more likely that one or more of these torque harmonics produced by the engine and compressor will excite a TNF.

Engine-driven reciprocating compressor trains have a potential for torsional vibration problems [1]. These systems can also be sensitive to the assembled angle between the engine and compressor crankshafts when bolted together. Depending on the relative phase angle between the two crankshafts, certain harmonics may add or cancel. At resonant frequencies, dynamic torque can be greatly amplified, possibly causing failures of crankshafts, couplings, mechanically driven oil pumps, and engine dampers.

SYSTEM DESCRIPTION

A diagram of the engine–driven compressor train is shown in Figure 1. The natural gas engine has 18 cylinders and is rated at 5,861 kW. The single-stage reciprocating compressor has a speed range of 575 to 775 RPM. The compressor cylinders are numbered 1 - 6 starting with the coupling end.

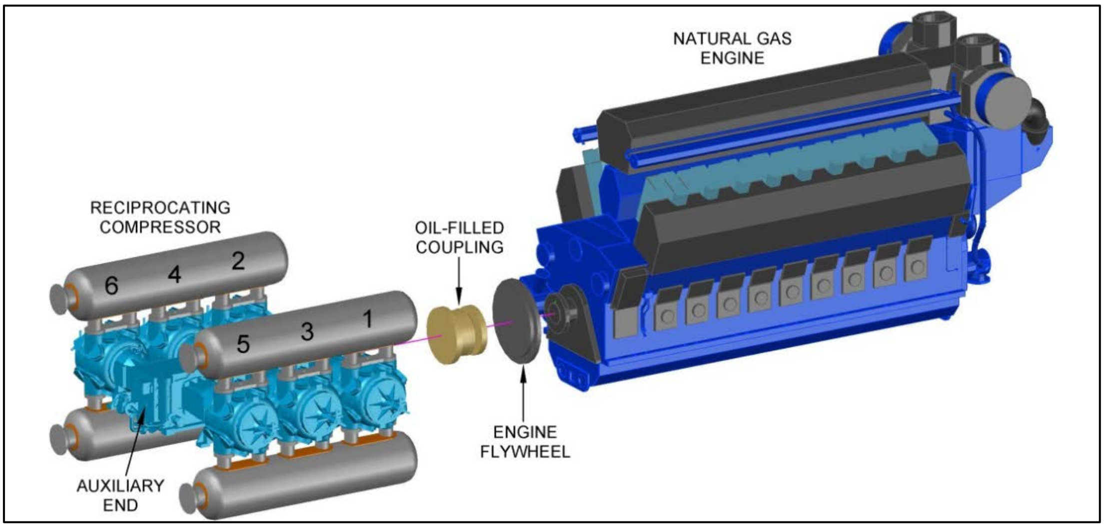

Figure 1 – Diagram of Engine-Driven Compressor Train

The coupling is oil-filled with radial leaf springs and experienced a failure as shown in Figure 2. Cracks occurred at a 45-degree angle in the center portion of the coupling where the springs fit into the axial grooves. The damage is typical of high torsional vibration. Oil samples taken prior to the failure indicated an increase in copper content possibly due to impacting of parts and wear damage inside of the coupling.

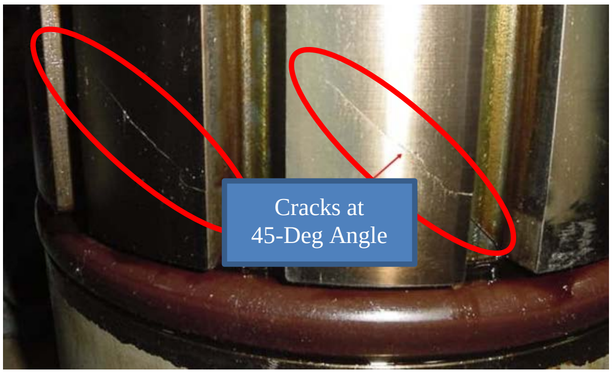

Figure 2 – Cracks in Center Portion of Coupling

The compressor has a total of 22 load steps that utilize a combination of clearance pockets and head-end (HE) unloaders to control gas flow. Table 1 lists the compressor cylinders with clearance pockets that can be opened and closed. Only cylinders 1, 3 and 5 have HE unloaders, which can be activated by holding the suction valves open. Load step 1 has the highest load with all pockets closed and all cylinders double-acting (DA). Load step 22 represents the minimum compressor load. The most dramatic changes in dynamic torque produced by the compressor occur between the load steps shown in red, as the HE of cylinders 1, 3, and 5 are unloaded.

Table 1 – Compressor Load Steps

Load Step	**Cylinder**								
	1		**2**	**3**		**4**	**5**		**6**
	Pockets	**HE**	**Pockets**	**Pockets**	**HE**	**Pockets**	**Pockets**	**HE**	**Pockets**
1 – All Cylinders DA	*Normally Closed*	*Normally Loaded*	*Normally Closed*	*Normally Closed*	*Normally Loaded*	*Normally Closed*	*Normally Closed*	*Normally Loaded*	*Normally Closed*
2						Open			
3				Open		Open			
4				Open		Open			Open
5				Open		Open	Open		Open
6			Open	Open		Open	Open		Open
7	Open		Open	Open		Open	Open		Open
8					Unloaded				
9					Unloaded	Open			
10					Unloaded	Open			Open
11					Unloaded	Open	Open		Open
12			Open		Unloaded	Open	Open		Open
13	Open		Open		Unloaded	Open	Open		Open
14					Unloaded			Unloaded	
15					Unloaded	Open		Unloaded	
16					Unloaded	Open		Unloaded	Open
17			Open		Unloaded	Open		Unloaded	Open
18	Open		Open		Unloaded	Open		Unloaded	Open
19		Unloaded			Unloaded			Unloaded	
20		Unloaded			Unloaded	Open		Unloaded	
21		Unloaded			Unloaded	Open		Unloaded	Open
22		Unloaded	Open		Unloaded	Open		Unloaded	Open

INSTRUMENTATION FOR FIELD TESTS

Field tests were performed to diagnose the cause of the coupling failure and to determine if the excitation was related to the engine and/or the compressor performance. A new coupling was installed before the field tests. The system should be in the original condition to acquire accurate data. Torsional vibration and compressor pressure-time (P-T) measurements were taken.

Torsional oscillation of the engine side of the coupling was measured using a proximity probe that monitored the gear teeth on the engine flywheel (Figure 3). Provided the lateral vibration is low, variations in time between tooth-passing should indicate torsional vibration. These pulses were then converted to torsional vibration using the Hilbert transform [2].

Torsional oscillation of the coupling flange on the compressor side was measured using an optical sensor aimed at a laminated strip of paper with alternating black and white squares that was wrapped around the circumference of the coupling flange (Figure 4). The signal from this optical probe was also processed using the Hilbert transform to determine the torsional vibration on the compressor side of the coupling.

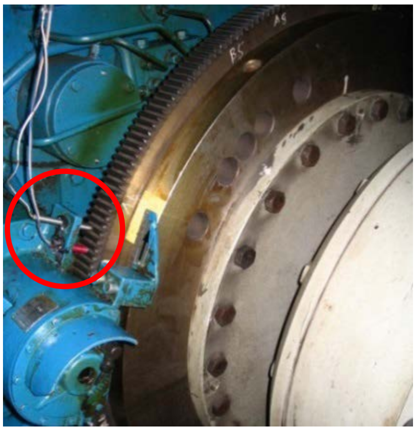

Figure 3 – Proximity Probe at the Engine Flywheel

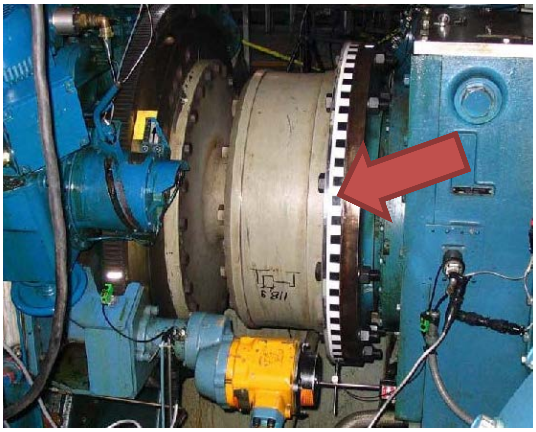

Figure 4 – Optical Sensor and Stripped Tape on the Coupling Flange

The instantaneous angular deflection across the coupling was calculated by taking the difference between the signals on the engine flywheel and the coupling flange on the compressor side. Based on the torsional stiffness of the coupling provided by the manufacturer, the dynamic torque in the coupling was then inferred. Another method would have been to use a strain gage telemetry system to measure torque; however, there was insufficient exposed shaft on either side of the coupling for installation of the strain gages.

Torsional oscillation of the auxiliary end of the compressor crankshaft was directly measured using an HBM torsiograph mounted on a threaded-adaptor as shown in Figure 5. The torsiograph uses slip rings and outputs a voltage signal proportional to the measured angular oscillation.

P-T data were measured with dynamic pressure transducers installed in all of the HE cylinders plus one transducer in the crank-end (CE) of cylinder 1 as shown in Figure 6. The shape of the P-T waveforms can change with the compressor load step, which then affects the amplitude of the torque harmonics produced by the compressor.

Figure 5 – HBM Torsiograph on Auxiliary End of Compressor

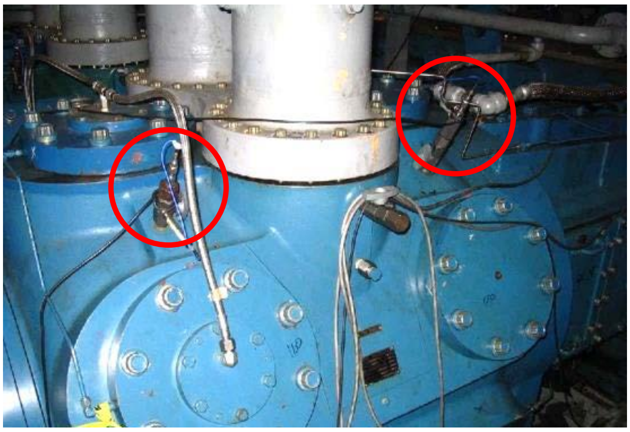

Figure 6 – Pressure Transducers in Compressor Cylinders

RESULTS OF FIELD TESTS

During the tests, all of the signals were simultaneously monitored and digitally recorded over the full operating speed range and at as many load steps as possible; however, based on the pipeline conditions only load steps 10 through 22 were available. The engine speed was slowly reduced from 775 to 575 RPM, and then increased back to 775 RPM. The compressor load steps were allowed to change automatically from 22 to 10. As shown in Figure 7, the torsional oscillation of the engine flywheel reached 1.75 degrees peak-to-peak when operating the compressor at lower speeds and load steps. A sudden jump of the 1× torsional amplitude was noted between load steps 19 and 18, which corresponded to loading the HE of compressor cylinder 1.

The dynamic differential angular displacement across the coupling is shown in Figure 8. The maximum amplitude occurred while operating near 675 - 680 RPM with compressor load steps 12 and 13. Cylinder 3 was single-acting which increased the dynamic torque at 1× running speed. As shown at 3:20 PM, the maximum differential across the coupling was 7 degrees peak-to-peak, or 61 milli-radians zero-peak.

Figure 8 shows that sudden jumps in torsional vibration occurred when switching between compressor load steps 18-19 and between steps 13-14. These steps correspond to loading the HE of cylinders 1 and 5, respectively. The transmitted torque results in approximately 80 milli-rad of constant angular displacement in the coupling. Combining the maximum dynamic angle of 61 milli-rad with the angle caused by the transmitted torque, sums to 141 milli-rad across the coupling. This exceeded the manufacturer's limit of 125 milli-rad, and indicated possible contact of parts inside of the coupling.

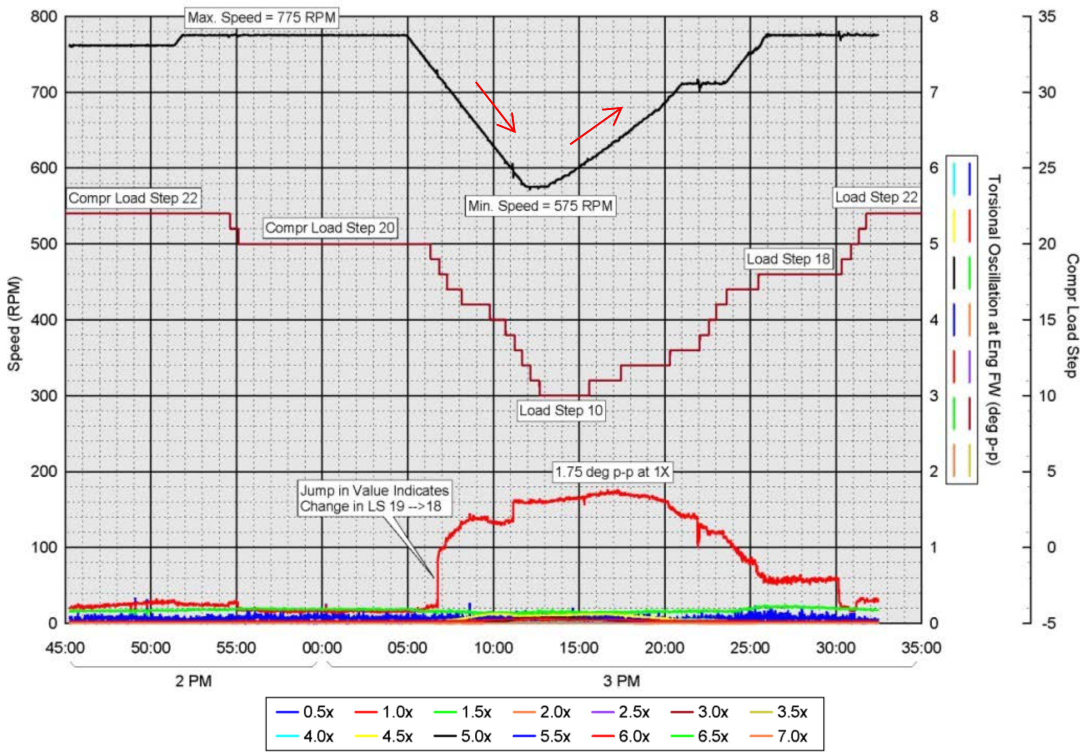

Figure 7 – Trend Plot of Speed, Compressor Load Steps, and Torsional Oscillation

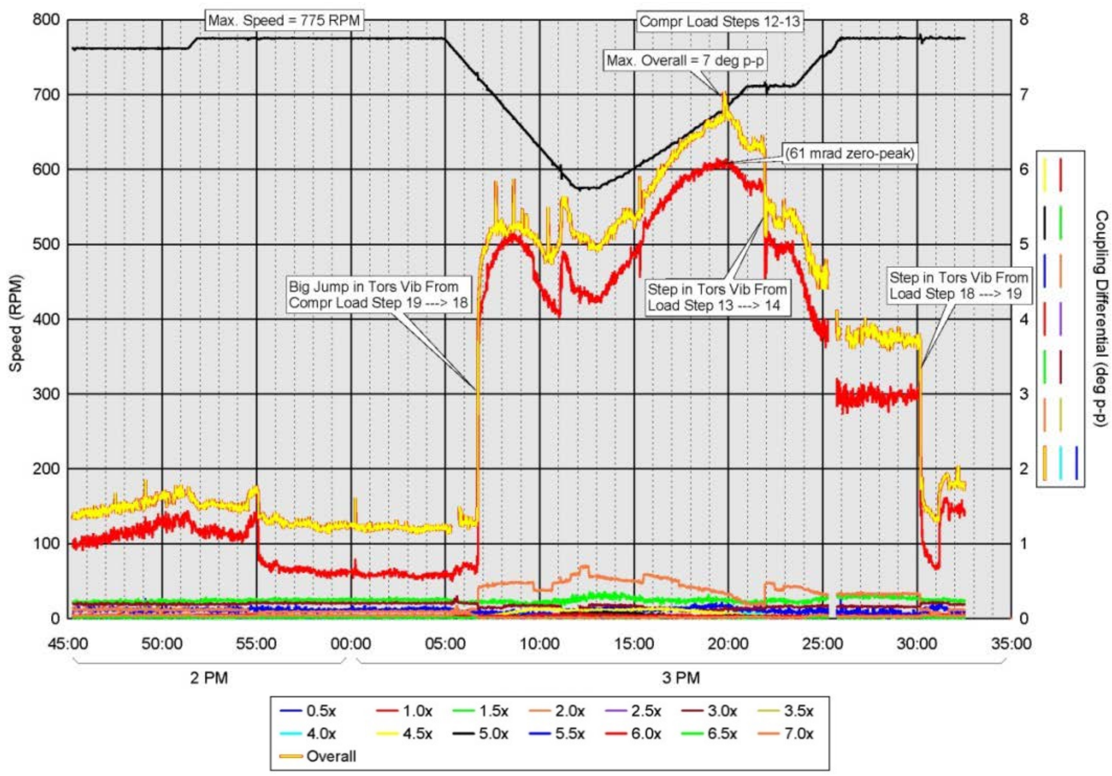

Figure 8 – Trend Plot Showing Differential Angular Displacement across Coupling

Figure 9 shows a waterfall plot of torsional vibration at the coupling flange on the compressor side as the unit speed was decreasing. The sudden jump in amplitude at 1× running speed was due to changing compressor load steps and excitation of the first TNF. The slice plot for the 1× harmonic shows the first TNF at 11.3 Hz or 678 CPM, which was within the normal operating speed range. A curve fit was used to estimate the amplification factor (AF) of 6, indicating that the actual system had less damping than was used in the original torsional analysis.

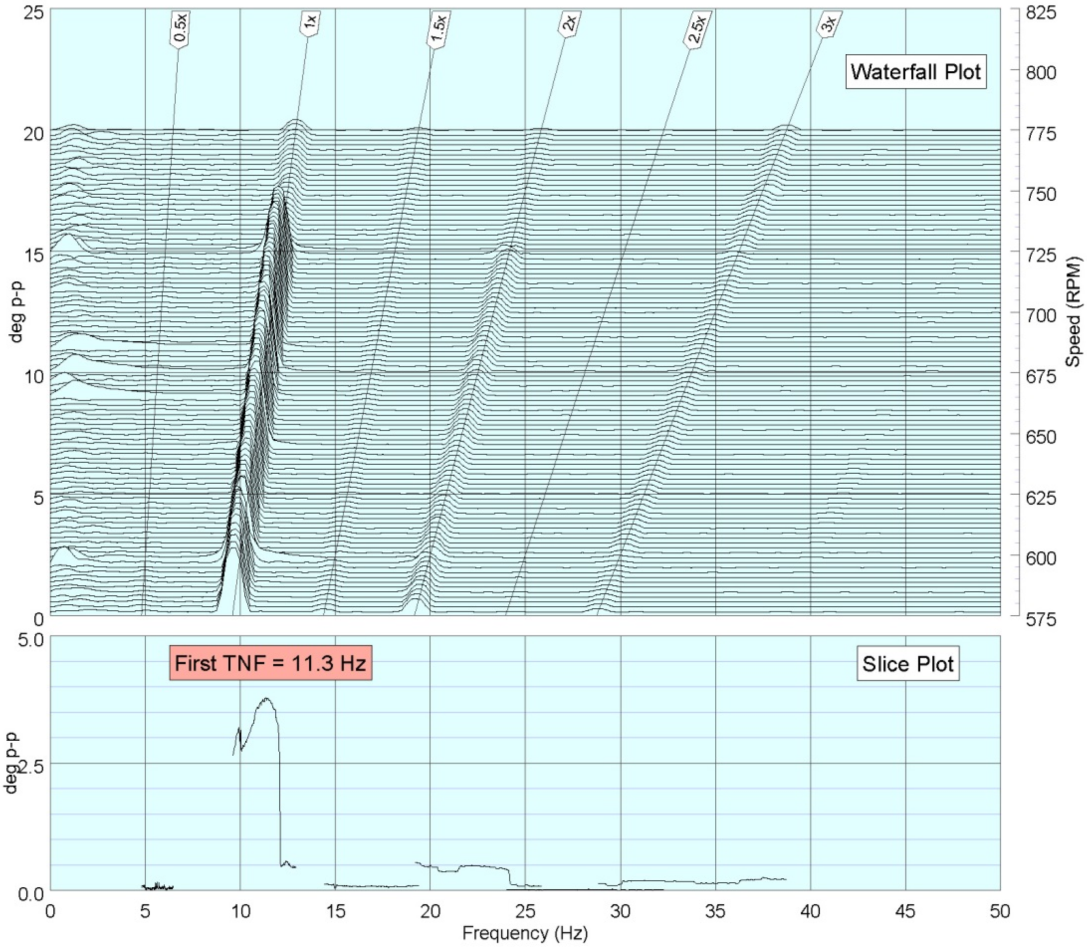

Figure 9 – Torsional Vibration of Coupling Flange on the Compressor Side

In subsequent tests, when the torsional oscillation was lower amplitude, the first TNF was found to be 10.7 Hz versus 11.3 Hz. It is thought that when the system was experiencing high torsional vibration, the springs were contacting the stops inside the coupling, thus increasing the torsional stiffness and the first TNF.

REVIEW OF TORSIONAL ANALYSIS

A torsional analysis was performed in the design stage as is good practice. The report indicated that the system would be satisfactory, although the first TNF was predicted within the operating speed range. This was because the first torsional mode was assumed to be well-damped due to the oil-filled coupling. During the field tests it was determined that the torsional damping for the first torsional mode was actually less than had been assumed in the analysis. The measured dynamic torque in the coupling was nearly double the calculated level at resonance.

Upon further review, it was found that only a few of the 22 compressor load cases had been evaluated in the torsional analysis. If all of the compressor load steps were considered, the results would have shown that the allowable limit for the coupling was exceeded for some of the compressor load conditions, particularly with single-acting cylinders which tend to produce higher dynamic torque at 1× running speed.

The original torsional analysis matched the measured TNFs fairly close, but needed to have the damping factors adjusted and all compressor load cases included. It was recommended that the first TNF be lowered out of the operating speed range by adding inertia to the compressor.

CONCLUSIONS

- Test results showed that the design limits of the coupling were exceeded in terms of angular oscillation and vibratory torque for some compressor operating conditions. Physical evidence such as cracks at a 45-degree angle, observation of high torsional vibration with a strobe light, and oil analysis with increasing copper content also supported this finding.
- The worst conditions occurred while operating near the first TNF. This torsional resonance was then strongly excited by the compressor load steps with single-acting cylinders. Harmonics from the engine were not found to be a contributor to the coupling failures.

RECOMMENDATIONS

- The short-term solution involved placing restrictions on the compressor operating speed and load steps to avoid exciting the first TNF and damaging the coupling.
- For the long-term solution, additional compressor inertia (using an external flywheel or internal rings attached to the spreader) was recommended to de-tune the first TNF below minimum running speed.

GENERAL GUIDELINES FOR PREVENTING TORSIONAL VIBRATION PROBLEMS

1. Torsional problems can often be avoided by performing a torsional vibration analysis with conservative assumptions in the design stage. A complete analysis should be performed and usually includes the following:
   a. Torsional natural frequencies, vibration mode shapes, interference diagram, shaft shear stress, coupling vibratory torque, torsional oscillation (particularly at the auxiliary end of a compressor or damper end of engine), heat dissipation (for systems with dampers or rubber couplings), and realistic damping effects.
   b. All load cases should be considered over the full operating speed range. It may also be necessary to analyze different phase angles between the engine and compressor crankshafts.
   c. Results should be compared to allowable separation margins, endurance limits of shaft material, allowable coupling torque, oscillation, or heat build-up.
   d. If a problem is predicted with a system as initially designed, possible modifications include: selecting an alternate coupling size or type, adding inertia (external or internal flywheels), or changing the engine damper, or adding a second damper.
2. Testing may be specified to verify the predicted TNFs and responses.
3. Avoid continuous operation near potentially dangerous torsional resonances by maintaining acceptable separation margins per API [3].
4. Torque harmonics that are *theoretically* low can still cause failures due to abnormal operating conditions such as: engine bank-to-bank imbalance, engine misfire, failed compressor valves, or acoustic resonances in the process piping. Therefore, periodic engine and compressor performance monitoring is recommended for improved reliability.

NOMENCLATURE

AF	amplification factor, non-dimensional
API	American Petroleum Institute
CE	crank end of compressor cylinder
CPM	cycles per minute, unit for vibration frequency
DA	double-acting compressor cylinder
HBM	Hottinger Baldwin Measurements, manufacturer of torsiograph
HE	head end of compressor cylinder
Hz	Hertz is unit for vibration frequency in cycles per second
kW	one kilowatt is equal to a thousand watts, unit for power
milli-rad	one thousandth of a radian, unit for angle
peak-to-peak	double amplitude
P-T	pressure in compressor cylinder plotted verses time
RPM	revolutions per minute, unit for speed
SA	single-acting compressor cylinder
TNF	torsional natural frequency
zero-peak	amplitude

ACKNOWLEDGEMENTS

The author appreciates the opportunity to present at this conference. Special thanks to Mark Broom for preparing Figure 1, and to Donald Smith for his helpful reviews of this paper.

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