TORSIONAL FAILURES IN HYDROGEN RECIPROCATING COMPRESSOR SYSTEM WITH STEPLESS CAPACITY CONTROL

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Abstract

Two new hydrogen reciprocating compressor systems were equipped with stepless capacity control. Initial problems after commissioning included high crosshead guide vibration, failures of the shaft-driven mechanical oil pump (MOP) and a motor cooling fan. It was later observed by personnel at the plant that elevated vibration and speed fluctuation of the system generally occurred while operating at medium compressor load. The manufacturers performed torsional vibration analyses (TVAs) in the design stage, which predicted adequate separation margins from significant compressor harmonics. Therefore, any concerns of torsional vibration were initially dismissed. Field testing with a strain gage telemetry system found that the first torsional natural frequency (TNF) was coincident with 8× running speed. This resulted in amplifying alternating torque up to 800% of full load torque (FLT) and the system being very sensitive to compressor loading due to the torsional resonance. The compressor crankshaft and motor shaft later experienced fatigue failures and had to be replaced after a relatively short time of operation. Spiral cracks occurring at a 45-degree angle to the shaft axis were consistent with failure due to high torsional vibration. As a short-term solution, the compressor was operated at a load condition that minimized the alternating torque. The TVA was then normalized to match the field data. The long-term solution involved detuning the TNFs by adding inertia to the flywheel and to the motor external cooling fan. The modified systems were retested to verify satisfactory operation over the entire compressor load range.

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INTRODUCTION

Two new synchronous motor-driven reciprocating compressors (Units A and B) were installed and commissioned at a refinery. These hydrogen compressors have four cylinders and four stages as shown in Figure 1. Suction pressure is approximately 300 psi and final discharge pressure is 2750 psi. The capacity is controlled from 20% to 100% load using stepless valve unloaders.

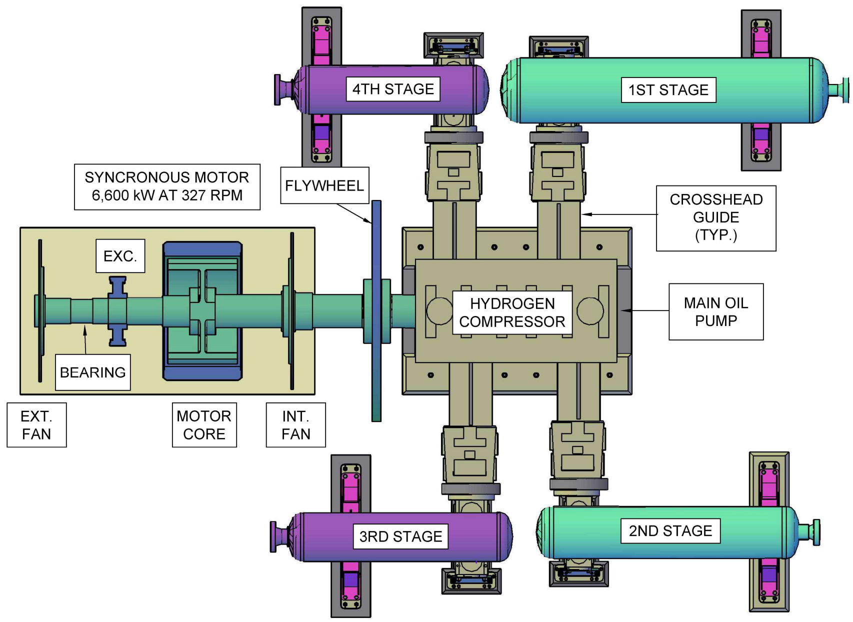

Figure 1 – System overview

The compressor is driven by a slow-speed synchronous motor that is rated for 6600 kW at 327 RPM and has a FLT of 193 kN-m. The motor shaft is directly bolted to the flywheel and compressor crankshaft with a flanged connection (no coupling). The motor has a single outboard bearing.

A variable frequency drive (VFD) is used as a soft starter for the motor. Once at synchronous speed, the motor is switched to across-the-line for constant speed operation. This avoids high in-rush current and reduces the pulsating torque normally produced by a synchronous motor during start-up.

There are various ways of controlling flow through a reciprocating compressor such as recycle, bypass, variable speed, clearance pockets, or step unloaders. The compressor loading can significantly affect the torque-effort [7]. The subject fixed-speed compressor systems utilize a stepless capacity control method, which works by delaying the closure of the suction valves [5]. Normally, the suction valves close when the pressure inside of the cylinder begins to compress and exceeds the inlet line pressure. However, the stepless capacity control can hold the suction valve in the open position for a specified portion of the stroke as shown in Figure 2. The black trace is the cylinder head end (HE) and the blue trace is the cylinder crank end (CE).

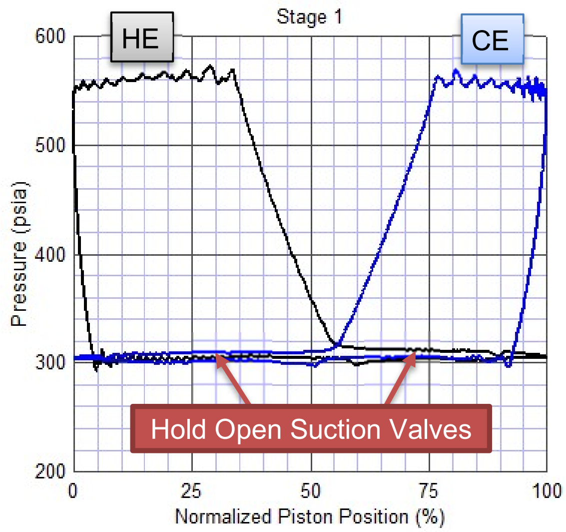

Figure 2 – Example of stepless capacity control at reduced compressor load

The varying shape of the cylinder pressure versus time (P-T) affects the torque-effort produced by the compressor at various loads. Figure 3 shows how predicted compressor harmonics change amplitude depending on the percent load. For this 4-stage compressor with stepless capacity control, higher alternating torque content is produced at 8× running speed while operating at 40% load as compared to 70% load. Note that this might not be the case for a reciprocating compressor with different cylinders utilizing another control method such as pockets, head-end unloaders, etc.

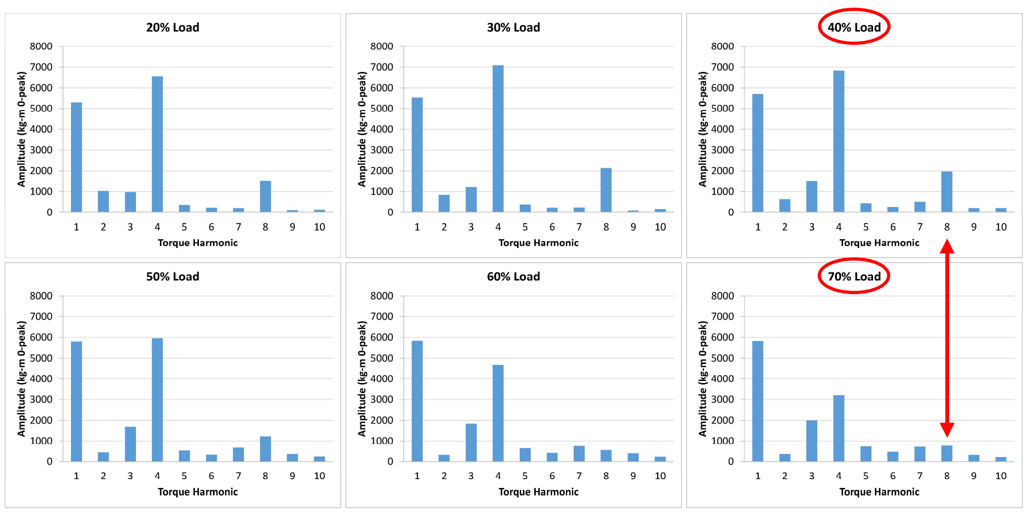

Figure 3 – Harmonic analysis of calculated torque-effort at various compressor loads

BACKGROUND INFORMATION

After commissioning, elevated vibration levels were noted on the compressor frame and crosshead guides when operating at 30% to 50% load. Significate crosshead guide vibration occurred in the frequency range of 200 to 300 Hz.

Inspection of the compressors revealed that the orifice plates located between the compressor cylinders and pulsation bottles, as specified from the acoustical analysis, were not installed. These single-hole orifice plates were subsequently installed, and the vibration levels were reportedly slightly reduced. However, vibration levels were still considered high when operating at 40% to 50% load when compared with the EFRC Guidelines [6].

To address the crosshead vibration, a third-party consulting firm recommended using multi-hole orifice plates [3], which increased pressure drop in the cylinder nozzles, but did not solve the vibration issues.

A series of MOP failures occurred on both compressors. These failures included broken fastening bolts and oil pump shafts. According to the compressor manufacturer, these oil pump shaft failures appeared to be due to reverse bending, and not shear stresses. Furthermore, no damage was found on the oil pump gears.

The compressor manufacturer made recommendations to improve the alignment of the lube oil pump, strengthen the shaft, and reduce vibration of the attached lube oil piping with additional bracing. However, the vibration around the MOP was not improved. It was also reported that the compressors were experiencing higher than normal speed fluctuation of up to 6% during operation at reduced compressor load.

TESTS OF ORIGINAL SYSTEM

Field tests were conducted to assist in identifying the cause(s) of the failures. Initial testing performed on Unit A included impact data on the external motor cooling fan, which was used to create mode shapes of the fan impeller [14]. Data were gathered as the compressor load was varied from 20% to 70% via the stepless valve control system. Simultaneous acquisition of vibration, pulsation, cylinder pressure, motor current/voltage, torsional oscillation and torque were obtained. Vibration levels were measured on the compressor frame, crosshead guides, motor tail bearing, and pulsation bottles. Pulsation levels were measured in the suction and discharge piping of all four stages.

Transmitted and alternating torque data were measured on the motor shaft next to the flywheel as shown in Figure 4. Shear style strain gages were installed on both sides of the motor shaft and wired as a full Wheatstone bridge to cancel any bending, axial or temperature effects [9]. It is important to use all four strain gages because the weight of the motor core is supported between the flywheel and a single outboard bearing causing some bending in the shaft.

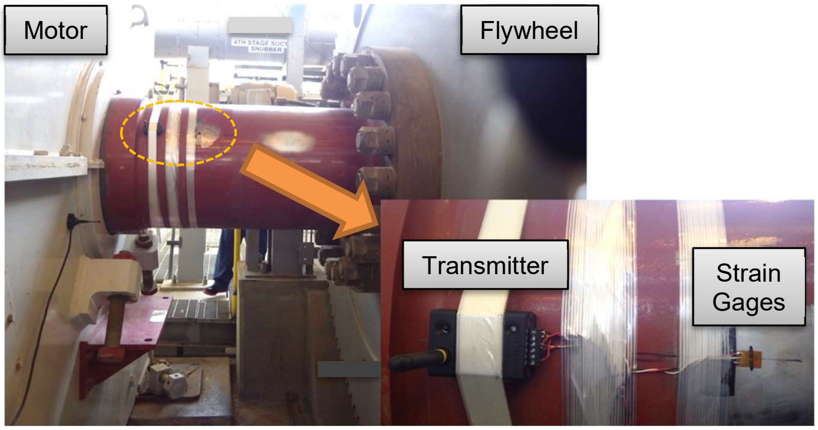

Figure 4 – Strain gage telemetry system mounted on motor shaft to measure torque

As shown in Figure 5, the first TNF was measured at 44.5 Hz, which is within 2% of 8× running speed. Consequently, the alternating torque levels were very sensitive to compressor load as shown in Figure 6. Alternating torque levels reached approximately ±800% of FLT with 55% compressor load, which is excessive. Alternating torque at the other compressor harmonics were much lower in comparison to 8× running speed. For reference, the GMRC torsional document provides a guideline for designing motors shafts up to ±200% alternating torque when driving a four-throw compressor [10].

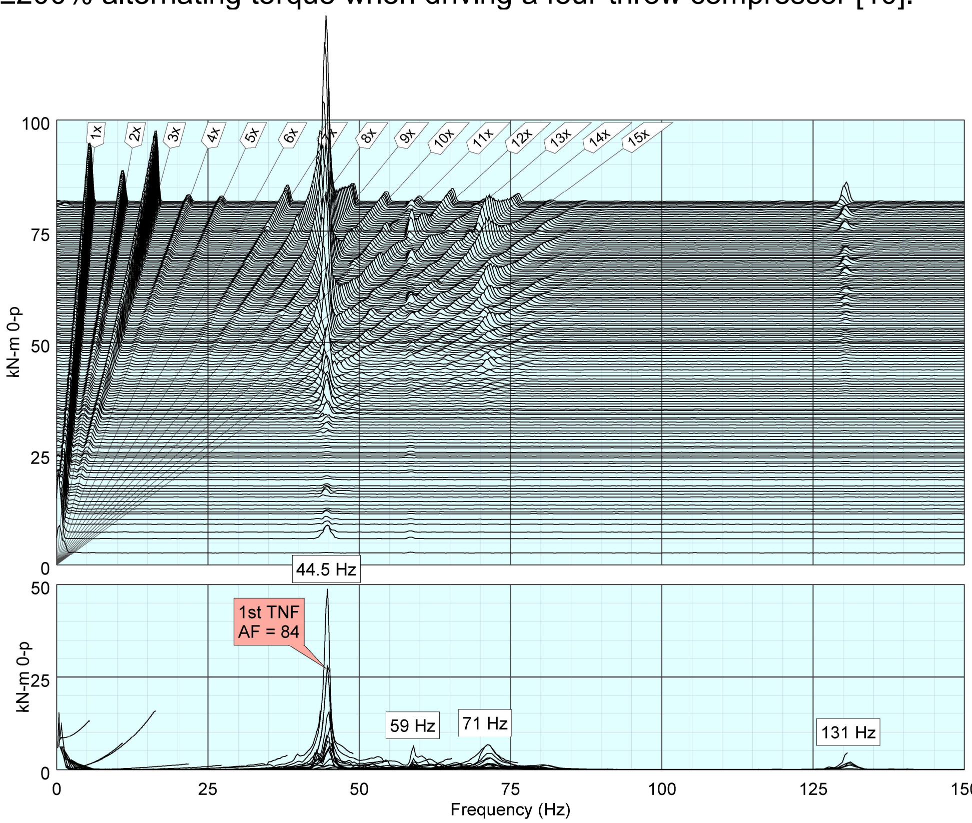

Figure 5 – Torque waterfall plot

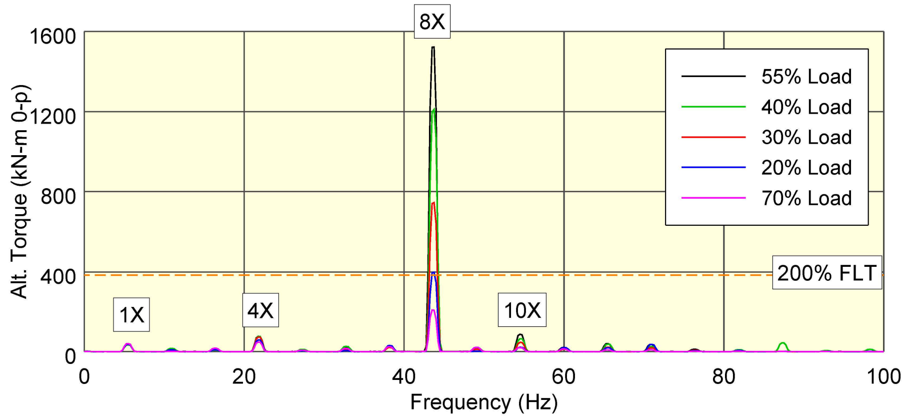

Figure 6 – Alternating torque versus compressor load

Torsional oscillation of the motor shaft measured near the external cooling fan showed elevated levels at 8× running speed and correlated with the alternating torque data. Lateral vibration readings on the compressor frame, motor bearing, and pulsation bottles also correlated with alternating torque.

Since the power factor (PF) of the motor was being automatically controlled, but the compressor load torque varies during a shaft revolution, the electrical system was checked. The current pulsation was determined to be less than 30%, which meets the NEMA guideline of 66% [13]. No electrical issues were found with the motor, and the total inertia of the compressor system was sufficient to prevent excessive current pulsation.

The frequency analysis of the motor current is shown in Figure 7. The highest sidebands at ±1× operating speed (red circles) are typical of the rigid-body torsional mode. However, there were also significant sidebands at ±8× operating speed (blue circles) due to the first TNF, which could be detected.

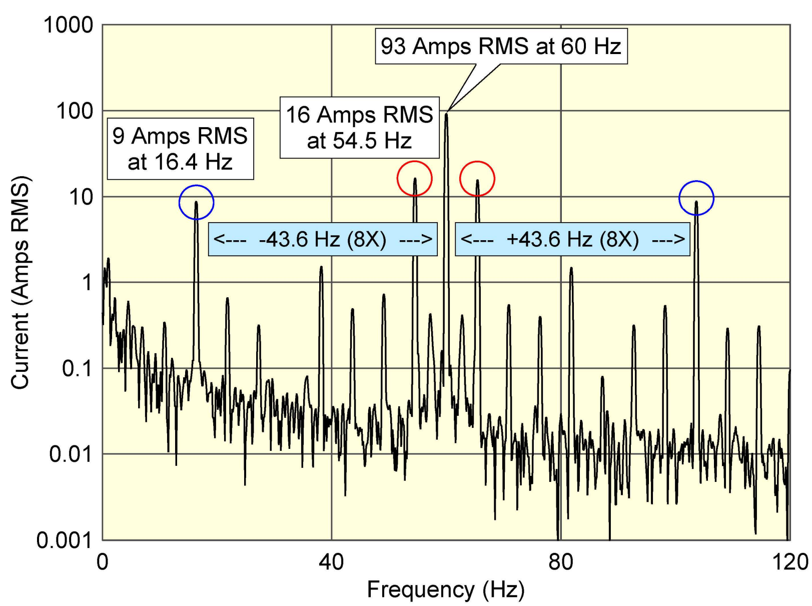

Figure 7 – Frequency analysis of motor current [8]

Data were acquired with the existing single-hole orifice plates and with the multi-hole orifice plates installed for comparison. Pulsation amplitudes were measured in the piping on the process side of the filter bottles and found to be below the American Petroleum Institute Standard, API 618 [2]. Single-hole versus multi-hole orifices did not significantly affect the vibration levels. The acoustic system was considered acceptable and not the source of the vibration. One of the authors had prior experience where an acoustic nozzle resonance increased the torsional vibration level in another motor-driven compressor system [9]. However, this was not the case for the subject units.

Before Unit B could be tested, it tripped on high compressor bearing temperature. Inspection revealed a spiral crack in the compressor crankshaft likely originating from a lubrication oil hole on the drive-end of the shaft. Lube oil holes are common stress risers in compressor and engine crankshafts.

The test data from Unit A and the crankshaft failure in Unit B indicated that amplification of alternating torque due to the excitation of the first TNF by the 8× compressor harmonic was the cause of high compressor crosshead vibration, failures of the MOP and failure of the motor external cooling fan.

ADDITIONAL TORSIONAL VIBRATION ANALYSIS

In the original TVA, the first TNF was predicted by the manufacturers to be between 8× and 9× running speed. Unfortunately, the first TNF was measured to be 44.5 Hz, which is too close to 8× compressor speed, and does not meet the recommended minimum API separation margin (SM). There were some discrepancies in the mass-elastic model for the motor-compressor system [15].

Additional torsional calculations were performed to evaluate possible modifications to “de-tune” the first TNF away from significant compressor harmonics. The first step was to normalize the mass-elastic model to match the measured TNFs. A finite element analysis (FEA) was performed on the compressor crankshaft to calculate torsional stiffness. The results were compared to stiffness equations for crankshafts found in Ker Wilson [11].

The first torsional mode shape involves twisting through the bolted flange connection with the motor and compressor oscillating out-of-phase. The amplification factor (AF) for the first TNF was very high indicating low damping. The AF of 84 was determined from coastdown data using the half-power point method [12] and included into the computer model. Based on experience with similar units, high AF (low torsional damping) is typical for a system with large diameter shafts that are directly bolted together with a flanged connection [9].

Excessive alternating torque and angular oscillation were measured at 8× running speed due to the excitation of the first TNF. The magnitude of the alternating torque depended on the compressor load controlled by the stepless capacity system. The measured P-T data were indexed using a multi-tooth wheel mounted on the oil pump end of the compressor and then converted to torque-effort for various load conditions. The forced excitation calculations were checked with the measured test results to verify the model was accurate.

Based on the new TVA, it was recommended to increase the flywheel inertia to tune the first TNF between 7× and 8× running speed. An increase in the inertia of the motor external cooling fan was also recommended to provide a separation margin (SM) for the second TNF.

API 546 [1] and API 618 [2] specify that a TVA should be performed, and that TNFs of the complete compressor system shall not be within 10% of any operating shaft speed or within 5% of any multiple of operating speed in the rotating system up to and including the tenth multiple. For motor-driven compressors, TNFs should ideally be separated from the first and second multiples of electrical power frequency by 10% and 5%, respectively.

TESTS OF MODIFIED SYSTEM

As a result of the initial testing and subsequent TVA, bolt-on inertia rings were manufactured and installed on the flywheel and external fan of Unit A. Prior to testing, the compressor crankshaft in Unit A was inspected, which revealed spiral cracks similar to those previously found in Unit B. Damage was also observed on the bearing cover as shown in Figure 8.

Figure 8 – Compressor crankshaft cracks and bearing damage

The test results of the modified system indicated that the first TNF shifted to 40.25 Hz or 7.4× running speed. Alternating torque levels in the motor shaft were found to be greatly reduced to a maximum of approximately ±120% FLT. Vibration levels were also significantly reduced to acceptable levels. Pulsation levels were unaffected by the modifications and still considered acceptable.

Trending of the torque data indicated that the first TNF appeared to shift down in frequency during a one-hour load test to approximately 39.25 Hz. The cause of this shift of the first TNF was believed to be caused by the cracks in the compressor crankshaft. As these cracks progress in size, the first TNF would likely be closer to 7× running speed, thereby amplifying, and increasing the alternating torque due to resonance.

A new compressor crankshaft was delivered and installed on Unit B. Inertia rings were also installed on the flywheel and motor fan. Testing was requested on Unit B to confirm that it behaved similarly to Unit A. The results showed that during start-up the first TNF was 39.5 Hz. However, the first TNF decreased during operation to 38.3 Hz, which was lower than predicted. The trend plot showed some variation in the minimum and maximum alternating torque values. This indicated the possibility of additional looseness or broken parts in Unit B. For example, one of the authors had previous experience diagnosing a loose pole on a synchronous motor driving a reciprocating compressor when the first TNF changed with speed during a coastdown [9].

After a thorough inspection, cracks were found in the motor shaft originating at the base of the keyway on the drive-end as shown in Figure 9. The key was also found to be slightly cocked, which can increase stresses in the keyway [4]. These cracks were believed to be the result of the high alternating torque experienced in the motor shaft prior to the installation of the inertia rings. The motor for Unit A was later inspected and found to have similar cracks. The motor manufacturer began fabricating new shafts with the added inertia integrated into the external cooling fan design. New one-piece flywheels were also fabricated with the specified increased inertia.

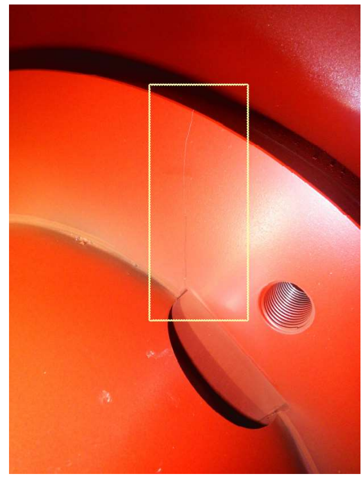
Figure 9 – Crack in motor

FINAL VERIFICATION TESTS OF REPAIRED SYSTEMS

A new motor shaft and flywheel were installed on Unit B. Test data showed that the first TNF was 40.4 Hz (7.4× running speed), which was the optimal separation margin. With the system now in “like new” condition, the measured TNFs agreed well with the TVA.

As with the previous tests, alternating torque reached a maximum value near 40% to 50% compressor load. However, now the maximum value was only ±120% of FLT as shown in Table 1. This was a significant reduction in the alternating torque levels and considered to be safe by the manufacturers.

Table 1. Alternating torque in motor shaft as percentage of FLT

Compressor Load	**Alternating Torque (zero-peak)**	
	Original System	**Mod. Flywheel**
20%	270%	80%
30%	455%	95%
40%	650%	110%
50% - 55%	Over 800%	120%
70%	150%	85%

Speed fluctuations measured at the motor external cooling fan were acceptable. In addition, compressor frame, crosshead guide, oil pump, and tail bearing vibration levels were low. It was therefore recommended that an identical motor shaft and flywheel be installed on Unit A. The modified compressor units have reportedly been operating successfully for four years.

CONCLUSIONS AND RECOMMENDATIONS

For long-term reliability and safe operation of a reciprocating compressor with stepless capacity control, the following should be carefully considered:

- The separation margin between higher compressor harmonics is very narrow so a small error in calculations could result in a torsional resonance. While the manufacturers performed TVAs in the design stage, there was an unexpected torsional resonance near 8× running speed. The alternating torque was greatly amplified due to the low torsional damping (AF=84). This was the cause for failures of the compressor crankshafts and motor shafts.
- The tendency of excitation torque for the stepless capacity control system is different than step control. This compressor system was very sensitive to load, and the "worst case" did not occur at the highest load. For example, the alternating torque levels in the motor shaft increased during compressor load changes from 20% to 55%, but then decreased at loads above 65%. Alternating torque at 70% load was relatively smooth and allowed for short-term operation until a solution was developed. Therefore, a comprehensive TVA should include all load conditions and follow the API recommendations.
- Torsional measurements are strongly recommended on critically important equipment during the shop test and/or commissioning at the plant. Strain gage telemetry systems can help verify system integrity. It is important to compare TNFs during startup and coastdown for repeatability. Variation in the first TNF between cold and hot conditions and variation in alternating torque during tests with constant compressor load could indicate cracks in the compressor crankshaft, motor hub, looseness, etc.
- To develop a long-term solution, the torsional model was normalized to match the measured data and used to analyze possible solutions. The size of the flywheel was increased, and inertia was added to the external cooling fan of the motor. The final measurements of the modified and fully repaired system confirmed:
    - Acceptable API separation margins from the first and second TNFs
    - Reduced sensitivity to compressor load
    - Decreased torsional vibration / speed fluctuation
    - Alternating torque below the GMRC guideline of ±200% FLT
    - Acceptable vibration levels on motor and compressor components
- It is recommended that the compressor performance be periodically checked to verify that the capacity control system is operating properly and there are no failed valves that could negatively affect the alternating torque.
- Initially, vibration problems and MOP failures were not recognized as being related to torsional issues. For example, high-frequency vibration on the compressor crosshead was mistakenly attributed to an acoustic resonance in the cylinder nozzle instead of excessive torsional vibration. Additional field measurements helped determine the proper course of action.

ACKNOWLEDGEMENTS

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