Electromagnetic Effects On The TNFs Of A Reciprocating Compressor With A Soft Coupling

First TNF shift occurred from torsional spring to ground related to electromagnetic effect

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This paper shows how the torsional stiffening effect of the electromagnetic field can affect the torsional natural frequencies (TNFs) of a compressor system that utilizes a torsionally soft coupling. The torsional measurements documented herein show that a shift in the first TNF occurred due to the effective torsional spring to ground related to electromagnetic (EM) effect. The torsional data confirmed that this effect is significant for a motor-driven reciprocating compressor system with a torsionally soft coupling.

Case Study Introduction

Four identical compressor trains were installed at a gas plant. Initial design analyses indicated that the units should not have any torsional issues; however, the operating company requested that torsional testing be performed during commissioning. Some compressor manufacturers recommend field-testing as standard procedure to verify results of the theoretical torsional analysis when a variable speed motor drives a compressor, since it is more likely that a torsional resonance could be encountered within a relatively wide speed range.

These compressor units shared a common variable frequency drive (VFD). This VFD was used to "soft start" one motor at a time and then transfer that motor to across-the-line operation for constant speed (60 Hz electrical). Once three motors are running across the line, the VFD can actively control the speed of the fourth unit to adjust for varying plant operating conditions. A general description of the compressor package is provided in Figure 1 and Table 1.

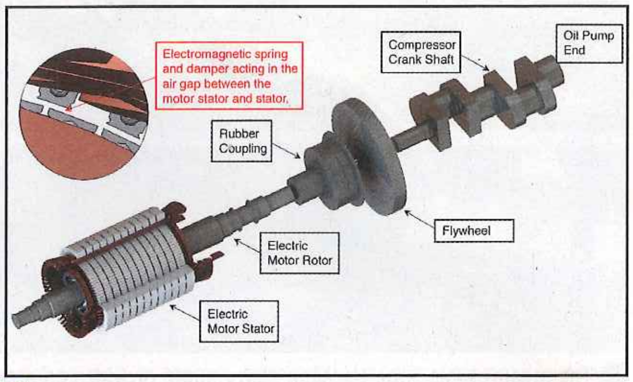

Figure 1. Diagram of VFD Motor-Driven Reciprocating Compressor System

Motor	Induction Motor, Rated 1050 hp (783 kW) Speed Controlled by Variable Frequency Drive Original Design = 750 to 1200 RPM (62% - 100%) Recommended = 780 to 1200 RPM (65% - 100%)
Coupling	Rubber-in Shear with Double Rubber Elements Torsional Stiffness = 0.49×10^6 lb.in./rad, Dynamic Magnifier = 6 Continuous Vibratory Torque = 27,700 lb.in. 0-p at 10 Hz Continuous Vibratory Torque = 13,850 lb.in. 0-p at 40 Hz
External Flywheel	Flywheel Mounted on Compressor Hub Added WR^2 = 110,000 lb.in.^{2}
Compressor	Two-Throw Reciprocating Compressor Two-Stage Residue Gas Service Cylinders Normally Double-Acting (DA)

Table 1. Description of Compressor System

Field Measurements

Measurements can be used to verify the torsional predictions. When acquired in the field, tests can be conducted

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under full load where the actual response can be determined. Measurements include dynamic torque, alternating stresses, torsional oscillation, etc. These values can then be compared to allowable limits.

Only a few points on the unit are available for measurement. Therefore, a torsional analysis is still needed to infer amplitudes at other locations that are inaccessible. By normalizing the computer model to match the measured data, conclusions and inferences can be drawn at the points of the system that are not directly measured.

Figure 2. Strain gauge telemetry system.

Strain gauges were installed on the motor shaft along with a battery-powered transmitter on the coupling hub (Figure 2). The four gauges were wired in a full Wheatstone bridge arrangement to measure only shear strain while cancelling out the effects of bending strain, axial strain, and varying temperature. The output voltage signal from the receiver was then converted to torque. The telemetry system was calibrated to measure transmitted torque (average), as well as dynamic torque with a frequency range up to 500 Hz.

Discussion Of Measured Data

Cold condition

The torque responses were measured in the motor shaft during a cold start. From the waterfall plot shown in Figure 3, a TNF was identified between 12 and 14 Hz.

The waterfall plot was created by vertically stacking multiple frequency spectra taken at intervals of operating speed. Harmonics (multiples of running speed) appear as diagonal lines in the waterfall plot. Any torque excitation at these frequencies can excite a TNF. The "slice plot" shown at the bottom of Figure 3 was created by tracing through the 1x and 2x harmonics. Response peaks shown in the slice plot indicate the TNF. Note that some smearing of frequencies occurred because the fast ramp rate of the VFD during startup.

The torque data were re-plotted to show the time waveform during startup. By counting cycles within the small time period at resonance, a TNF was identified at approximately 13.8 Hz.

A torque spike was observed when the motor was switched from the VFD to across-the-line operation. As shown in Figure 4, during this transient event the highest torque spike was

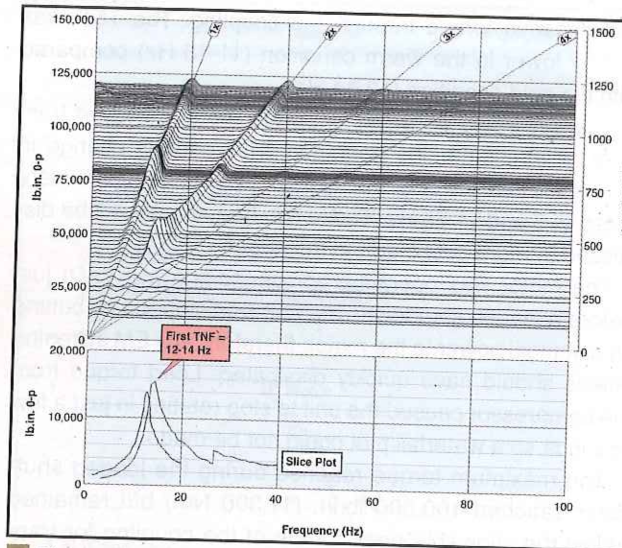

Figure 3. Waterfall plot of dynamic torque vs speed during cold start.

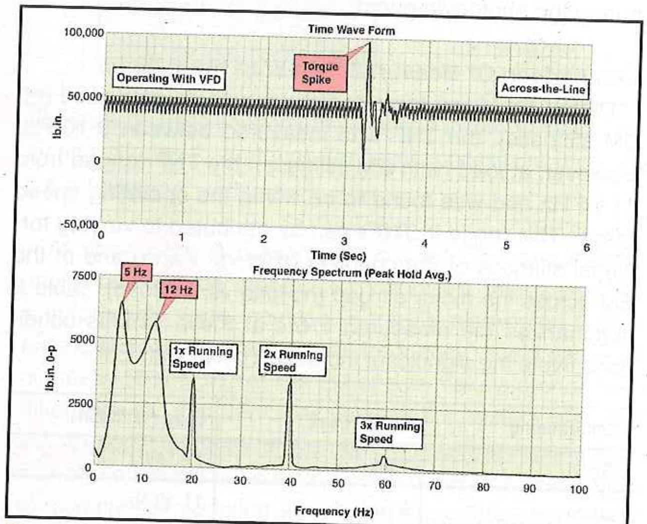

Figure 4. Torque measured during transfer from VFD to across-the-line motor operation — cold condition.

approximately 100,000 lb.in. (11,300 Nm) (twice the full load torque), which is still lower than the allowable peak torque for the rubber coupling. Using peak-hold averaging during the electrical switching event, non-synchronous response peaks were evident at 5 and 12 Hz. The 5 Hz frequency was not observed during the initial startup.

Warm condition

A second start was recorded with the unit and rubber coupling in a "warm" condition. The coupling was considered warm since the compressor unit had been operating for some time. For reference, the estimated temperature of the coupling was less than 120°F (49°C).

Although rubber-in-shear couplings are thought to have relatively constant torsional stiffness, the TNF can vary with the temperature of the rubber elements and with the mean

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and dynamic torque through the coupling. The TNF was slightly lower in the warm condition (11-13 Hz) compared with the cold condition (12-14 Hz).

A torque spike occurred when the motor switched from VFD to across-the-line operation. This sudden change in torque excited TNFs of the system. Response peaks were noted at 5 and 11.75 Hz. These two frequencies will be discussed in more detail later in the paper.

The motor was operating across-the-line (no VFD) just before a loaded shutdown. The unit was stopped by cutting off electrical power to the motor, therefore any EM stiffening effects should have quickly dissipated. Load torque from the compressor caused the unit to stop rotating in just a few seconds so a waterfall plot could not be made.

The maximum torque reached during the loaded shutdown reached 100,000 lb.in. (11,300 Nm) but remained below the allowable peak torque of the coupling for transient events, so no damage occurred to the rubber elements. The response frequencies ranged from 8-10 Hz during the shutdown event.

Comparison Of Measured Data With Predictions

During shutdown when the motor was de-energized (no EM stiffness), the TNF was measured between 9-10 Hz. However, at load (with EM stiffness), the TNF ranged from 11-14 Hz and was found to be within the operating speed range. This range in TNFs can be attributed to varying torsional stiffness of the coupling (cold vs. warm) and of the EM across the motor air gap (no-load vs. full-load). Table 2 summarizes the measured TNFs at these various conditions. Note the significant increase in TNF with EM.

Conditioning	No EM Load	Load (With EM)
Cold	10Hz	12-14 Hz
Warm	9 Hz	11-13 Hz

Table 2. Summary of measured TNFs.

In the design stage, an independent torsional vibration analysis (TVA) was performed as recommended by API. The TNF involving the coupling was calculated to be approximately 30% below the minimum running speed, which should normally be a sufficient separation margin (SM). However, the coupling and flywheel were selected using a TVA that did not include the EM stiffening effect across the motor air gap. Had the torsional stiffness due to EM been considered, a softer coupling and/or larger flywheel would likely have been specified for this compressor system to achieve the full speed range. Recall that decreasing torsional stiffness would lower the torsional natural frequency, and increasing the mass moment of inertia would also lower the torsional natural frequency.

Normalized Torsional Model

The stiffening effect associated with the EM is equivalent to a torsional spring attached between the motor core and ground. This additional stiffness explains why the measured TNF was higher compared to the calculated value that did not include the EM. The EM also creates a frequency near 5 Hz, which was observed only after the torque spike occurred transferring from VFD operation. The TNF at 5 Hz was not evident during startup because it is well below running speed (no low frequency excitation from the VFD).

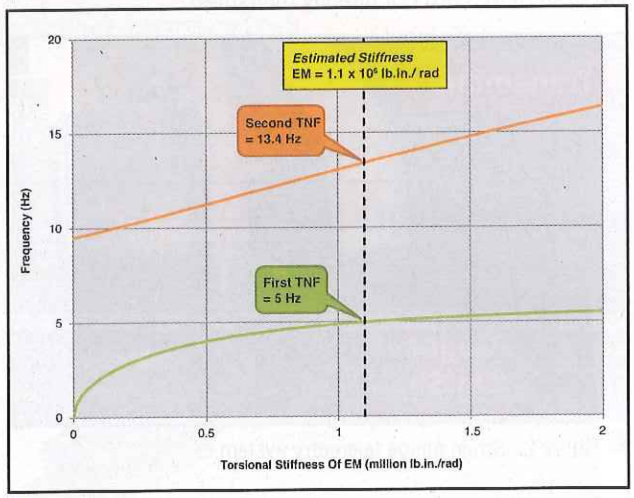


Figure 5. Parametric study of TNFs sensitivity to EM stiffness.

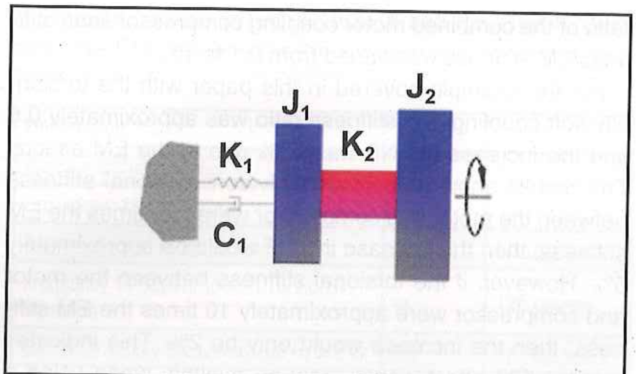
Once the torsional computer model was normalized to match the no-load case, a parametric study was performed to evaluate the sensitivity of the TNFs to EM stiffness. As shown in Figure 5, the EM torsional stiffness at the motor air gap would need to be approximately 1.1 x 10^6 lb.in./rad (120.6 kNm/rad) to correlate with the first two TNFs that were measured in the field.

The estimated EM stiffness across the motor air gap appears to be reasonable because "Drive B" in the Hauptmann, et al., paper had a similarly rated motor with a reported stiffness of 0.9 x 10^6 lb.in/rad (98.7 kNm/rad). Holopainen [7] reported a torsional stiffness of 1.4 x 10^6 lb.in./rad (160 kNm/rad) for an induction motor rated for 1250 HP (932 kW). In comparison, the EM torsional stiffness of 1.1 x 10^6 lb.in./rad (120.6 kNm/rad) is approximately 2.2 x the equivalent torsional stiffness of the rubber coupling elements (i.e., 1.1 / 0.49 = 2.2).

Simplified Torsional Model

Although a detailed mass-elastic model is normally required for an accurate torsional analysis, a simplified two-inertia model is created here for illustration purposes. Equivalent mass-elastic values are listed in Table 3. In addition to torsional stiffness, EM can also provide damping.

The dynamic magnifier given by the manufacturer for the rubber element in the coupling is an indication of damping. However, if the damping is neglected, then the following equation can be used. For two unequal inertias,



Electromagnetic (EM) Estimated From Measurements	$K_1 = 1.1 \times 10^6$ lb.in./rad (120.6 kNm/rad) $C_1 = 780$ lb.in.-s/rad (88.1 Nm-s/rad)
Motor Rotor	$J_1 = 242$ lb.in.-s^2 (27.3 Nm-s^2)
Rubber Coupling	$K_2 = 0.49 \times 10^6$ lb.in/rad (53.7 kNm/rad) Dynamic Magnifier = 6
Flywheel (FW) + Compressor	$J_2 = 314$ lb.in.-s^2 (35.5 Nm-s^2)

Table 3. Equivalent mass-elastic values.

and two unequal springs, the undamped torsional natural frequencies can be calculated using the following equation from Blevins [8]:

$$f_i = \frac{1}{2^{\frac{3}{2}}\pi}\left\{\frac{k_1}{J_1}+\frac{k_2}{J_1}+\frac{k_2}{J_2}\mp\left[\left(\frac{k_1}{J_1}+\frac{k_2}{J_1}+\frac{k_2}{J_2}\right)^2-\frac{4\,k_1\,k_2}{J_1\,J_2}\right]^{\frac{1}{2}}\right\}^{\frac{1}{2}}$$

Equation 1

Without the EM effects, the results from the simple model are $f_1 = 0$ Hz and $f_2 = 9.5$ Hz. With EM effect, the results from the simple model are $f_1 = 5$ Hz and $f_2 = 13.4$ Hz, which reasonably match the measured results.

The first two torsional modeshapes are plotted in Figures 6 and 7. Comparisons are made with and without the EM effects. Torsional modeshapes are a little difficult to represent on paper. Torsional oscillation is normalized to maximum of +1 or -1 (unitless). A point of maximum oscillation is called an anti-node. Significant twisting through a shaft or coupling is indicated by a line crossing zero, and this point is referred to as a node (where no oscillation occurs).

For example, a zero mode would be predicted without EM because the torsional system is not attached to ground (free to rotate). Adding the EM stiffness between the motor core and ground, as indicted by the dashed line, increases the TNF from 0 to 5 Hz.

For the torsional mode involving significant twisting through the coupling, the TNF is 9.5 Hz without EM and increases to 13.4 Hz with EM. This represents an increase in frequency of approximately 40%.

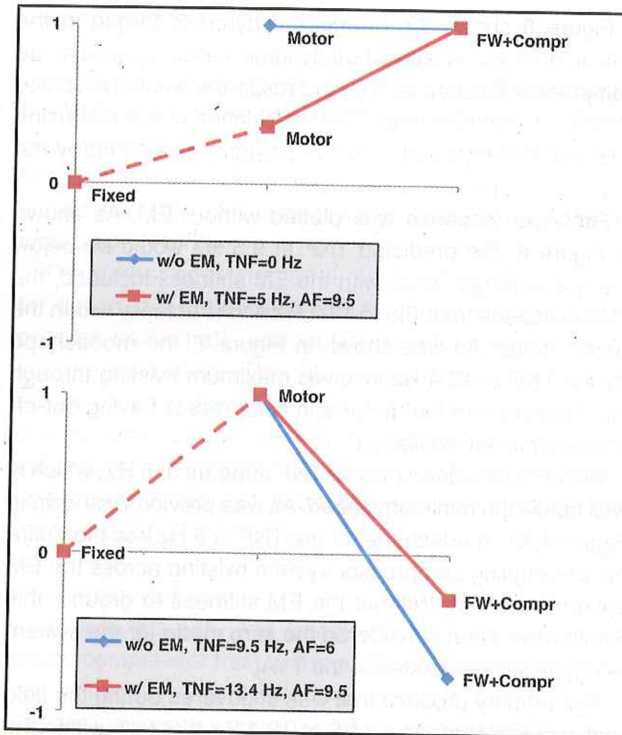


Figure 6. (top) Torsional modeshape involving rigid body motion. Figure 7. (bottom) Torsional modeshape involving the coupling.

As shown, the increase in TNFs due to EM is significant and should not be ignored. Although this may seem like a surprisingly large increase, Knop [3] also noted that in his experience the "real natural frequency" could easily be higher by 50% or more. The EM effects make a pronounced difference for this particular compressor system because the torsional stiffness of the rubber coupling is less than half of the EM stiffness. Later in this paper, systems with a torsionally stiff coupling are shown to be less sensitive to the EM effect.

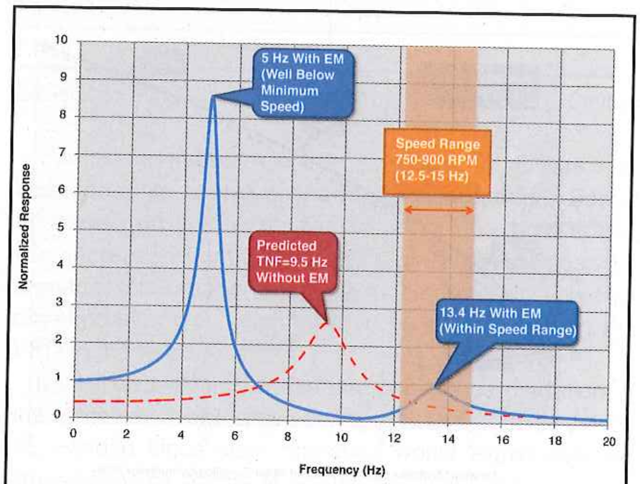


Figure 8. Comparison of calculated normalized dynamic torque in coupling — with and without em.

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Figure 8 shows the normalized dynamic torque in the coupling based on a constant dynamic torque applied at the compressor location so that the response would be scaled to unity at zero frequency. This calculation was made using a proprietary torsional vibration program developed by the author's company.

First, the response was plotted without EM. As shown in Figure 8, the predicted TNF at 9.5 Hz would be below the speed range. Then with the EM stiffness included, the TNF increases from 9.5 to 13.4 Hz and is actually within the speed range. As was shown in Figure 7, the modeshape for the TNF at 13.4 Hz involves maximum twisting through the coupling with the motor and compressor having out-of-phase torsional oscillation.

With EM included, another TNF appears at 5 Hz, which is well below the minimum speed. As was previously shown in Figure 8, the modeshape for the TNF at 5 Hz has the entire motor-coupling-compressor system twisting across the EM spring to ground. Without the EM stiffness to ground, this would have been considered the zero mode for the system and typically not plotted in the TVA.

The primary problem that was uncovered during the field test was that there is a TNF at 13.4 Hz that falls within the operating speed range and therefore does not have the API recommended separation margin. Although the TNF at 5 Hz appears to have a higher peak response, it is well below the minimum operating speed and not a concern. In fact, the TNF at 5 Hz may not have even been noticed except for the switching from VFD to across-the-line that momentarily produced a small torque spike.

Discussion Of Predicted Vs. Measured Results

An evaluation was made to determine how the motor EM stiffness would affect the system if a stiffer disc type coupling were used instead of the torsionally soft rubber coupling. This was accomplished by performing a parametric analysis as shown in Figure 9 where the torsional stiffness

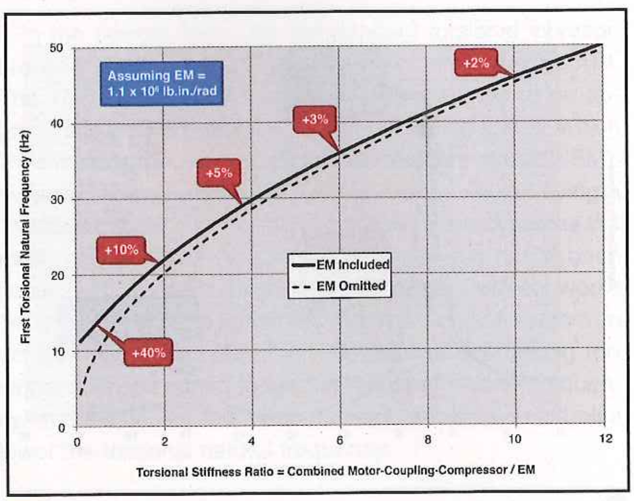

Figure 9. Comparison of calculated normalized dynamic torque in coupling — with and without EM.

ratio of the combined motor coupling compressor shaft stiffness/EM stiffness was varied from 0.1 to 12.

For the example covered in this paper with the torsionally soft coupling, the stiffness ratio was approximately 0.5 and the increase in TNF was 40% due to the EM effects. The results show that if the equivalent torsional stiffness between the motor and compressor were four times the EM stiffness, then the increase in TNF would be approximately 5%. However, if the torsional stiffness between the motor and compressor were approximately 10 times the EM stiffness, the increase would only be 2%. This indicates that the EM effect would likely be minimal when using a disc-type coupling.

Another discrepancy between the predicted and measured results can be due to the torsional stiffness variation of the rubber coupling. Tolerances in rubber, which is a natural product, can mean that the dynamic torsional stiffness will deviate by ±15% from the values provided in the catalog. The dynamic torsional stiffness can also be influenced by the temperature of the rubber elements, steady torque transmitted through the coupling, and by the frequency and amplitude of the vibratory torque. For example, some coupling manufacturers determine the torsional stiffness on a test rig at a frequency of 10 Hz and then recommend a correction factor for other frequencies.

Electromagnetic Model

Up to this point, the EM effect has been shown to exist, and the EM stiffness value was inferred from the torsional model and field test data. The estimated torsional stiffness was also shown to fall within a range of stiffness values previously published in other papers. The following section discusses how the EM can be calculated in the design stage based on supplied motor information and simplified equations.

The electromagnetic field in the air gap of an electric motor induces forces in the radial and tangential direction. The steady-state torque and power output of an induction motor are the result of electromagnetic fields, which act across the air gap between stator and rotor. The motor rotor has tangential forces applied to it, which are responsible for the torque generation. No resultant radial load is expected when the motor rotor is centered within the stator.

There are some types of machines that require more detailed analysis of the electromagnetic interaction when applied in a motor-driven system. Motors with sleeve bearings present eccentricity due to the bearing requirements for operation, and, according to the level of deviation, could be relevant to the resultant load. High-speed motors or generators, operating near a rotor lateral critical speed, have the eccentricity increased due to bending and whirling of the rotor. The total resultant force in the air gap is called Unbalanced Magnetic Pull (UMP) and different methodologies can be used to evaluate the influence of the electromagnetic parameters in the dynamics of the rotor.

The information listed in Table 4 can be used to compute the EM stiffness for the motor used in the example for this paper.

Description	Variable	Value
Number Of Stator Poles	N1	6
Breakdown Motor Torque	T_B	146,232 lb.in.
Rated Full Load Motor Torque	T_R	55.147 lb.in.
Frequency Of Superimposed Torsional Vibration	ω	126 rad/sec
Electrical Supply Frequency (60 Hz x 2π)	Ω_s	377 rad/sec
Slip At Rated Load	S_r	0.0092

Table 4. Data from Electric Motor Manufacturer.

The first step is to calculate the electrical time constant:

$$T_L=\left(\frac{1}{\Omega_s}\right)\cdot\left(\frac{1}{2s_r}\right)\cdot\left(\frac{T_R}{T_B}\right)$$

Equation 2

T_L=0.0544 sec.

Then the EM stiffness will be:

$$K_{em}=N1\cdot T_B\frac{(\omega\cdot T_L)^2}{(1+(\omega\cdot T_L)^2)}$$

$$K_{em}\approx 0.9\times 10^6 \text{ in-lb/rad}$$

Equations 3 and 4

And finally the EM damping is:

$$C_{em}=K_{em}\cdot\frac{T_L}{(\omega\cdot T_L)^2}$$

Equation 5

$C_{em}\approx$1000 in-lb-s/rad

As shown, the calculated EM stiffness of 0.9 x 10^6 lb.in./rad (98.7 kNm/rad) compares favorably with the EM stiffness of 1.1 x 10^6 lb.in./rad (120.6 kNm/rad) that was based upon the measured field data. This indicates that these equations can be used to estimate the effective EM stiffness.

Using the calculated EM stiffness with the simple two-inertia model previously developed, the first two TNFs were re-calculated to be 4.7 Hz and f_2 = 12.6 Hz. These calculated TNFs closely compared with the measured frequencies.

Conclusions

Torsional excitation from the VFD motor was insignificant compared to the reciprocating compressor. This is because the VFD is a modern type pulse width modulation (PWM) with relatively low torque ripple. However, authors have seen other cases where excessive torsional vibration occurred when the VFD control was improperly tuned for the application [11,12].

The rubber coupling had a much lower allowable limit for dynamic torque than the steel motor shaft and compressor crankshaft, and would therefore be the "weak link" in the system. The rubber coupling would typically act as a fuse in the system and likely fail before the other components. Cracks were not evident in the rubber elements, which indicated that the newly-commissioned compressors had not been operating for long periods with excessive torsional vibration.

Recommendations

To ensure a safe and reliable design of a VFD motor-driven compressor train, a complete simulation of the entire unit is necessary and field tests are also recommended. Separation margins should meet API or the unit should be shown to have acceptable torsional response at all load conditions and operating speeds.

The EM effect should be included in future TVAs, especially when the coupling is torsionally soft (stiffness ratio less than four). See Table 5 for error estimates when EM is not considered.

Torsional Stiffness Ratio = Combined Motor-Coupling-Compressor Shaft / EM	Increase Of TNF With EM Compared To Predicted First TNF Without EM
0.5	+40%
1	+20%
2	+10%
4	+5%
6	+3%
10	+2%

Table 5. Summary of Estimated Error If The EM Effect Is Omitted

For the compressor unit in the example, it was recommended to limit the operating speed range. Based on measured field data, the minimum operating speed was increased from 750 to 780 rpm to prevent possible damage to the rubber elements in the coupling. This recommendation was accomplished by reprogramming the VFD in the field.

It was also recommended that continuous operation of the compressor with single-acting cylinders or failed valves be avoided since such operation would significantly increase dynamic torque at 1x running speed, and could prematurely wear out the rubber coupling elements. CT2

See references at: ct2.co/references