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ELECTROMAGNETIC EFFECTS ON THE TORSIONAL NATURAL FREQUENCIES OF AN INDUCTION MOTOR DRIVEN RECIPROCATING COMPRESSOR WITH A SOFT COUPLING

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ABSTRACT

The electromagnetic field in the air gap of an electric motor is responsible for producing torque between the rotor and stator. The analysis and design of motor-driven equipment trains can be improved by including this electromagnetic (EM) effect. Torsional vibration or unsteady conditions from a reciprocating compressor are superimposed over the steady-state operation of an induction motor. In the past, these phenomena were usually neglected as typical analytic methods were unavailable for the prediction of these special effects. In some cases, not accounting for EM influence leads to substantial errors in the torsional vibration analysis (TVA).

Torsional vibration data were obtained on a motor driven reciprocating compressor system with a torsionally soft rubber coupling. 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 EM. The torsional data confirmed that this effect is significant for a motor-driven reciprocating compressor system with a torsionally soft coupling.

Comparisons of the field data are then made with theoretical predications of the TNFs with and without EM effects. A relatively simple methodology for calculating torsional stiffness and damping of EM is shown and yielded good correlation with the measured data. The accurate prediction of all dynamic characteristics of the system becomes more important when the motor is controlled by a variable frequency drive (VFD) over a large speed range so that dangerous torsional resonances can be avoided.

As a result of the torsional measurements, the minimum operating speed of this compressor system was increased to provide a sufficient separation margin (SM) from the TNF as recommended by American Petroleum Institute, API 618 for reciprocating compressors [1]. This was accomplished by reprogramming the VFD in the field. To minimize dynamic torque in the rubber coupling, it was also recommended that operating the reciprocating compressor with single-acting cylinders be avoided since such operation produced higher dynamic torque at 1× running speed and excited a torsional natural frequency of the system. In the future, the EM effect should be included in torsional analyses, especially for motor-driven reciprocating compressors with soft couplings.

BACKGROUND

Reciprocating compressors produce unsteady torque modulations, often exceeding the average transmitted torque. The torque variation in reciprocating machinery is typically much higher than that produced by rotating equipment (centrifugal compressors, fans, motors, etc.). The torque excitation produced by reciprocating machinery occurs at multiples of running speed (orders or harmonics) and can have significant amplitude, so they must be considered to avoid torsional problems.

When operating over a wide speed range (e.g., VFD motor drive), it is more likely that one or more of these torque harmonics will excite a TNF of the system. There can be several torsional resonances within the operating speed range. At resonant frequencies, the dynamic torque can be greatly amplified. Therefore, it is very important that reciprocating compressor trains be analyzed in the design stage, prior to installation. Units in critical service may also need to be field tested.

Various torsional vibration problems can occur in motor-driven reciprocating compressor systems [2]. Excessive torsional vibration can lead to failures of motor shafts, coupling components, and mechanical lubrication pumps. The shaft failures due to torsional vibration typically occur at a 45 degree angle and initiate at keyways or welds that have high stress concentration. In many cases, torsional vibration problems may not be apparent until after a failure progresses to the point of causing elevated lateral vibration.

There are several philosophies for addressing torsional vibration. First, the compressor loads can be controlled by pockets, unloaders, or deactivators while operating at constant speed. For this type of system, a disc pack coupling may be used; however, the torsional natural frequencies must be tuned between compressor orders with sufficient separation margin. The overall dynamic torque in the coupling and torsional oscillation at the oil pump must remain within acceptable limits for all load steps to avoid a possible failure.

A second approach for controlling compressor loads is to vary the operating speed. Systems with a VFD motor may require a torsionally soft coupling and/or large flywheel. These couplings can be more expensive and require periodic maintenance, but they provide torsional damping and can be used to detune the first TNF well below the minimum running speed. See Appendix on torsionally soft couplings for more information.

Stiffening effects from the motor magnetic field can affect the torsional natural frequencies (TNFs) of motor driven compressor systems, particularly when utilizing soft couplings. Knop [3] discussed how motor dynamics affect simulated results. Next, Hauptmann, Eckert, and Howes [4,5] presented an approximate method for calculating the shift in TNF and damping due to electromagnetic effects based on a few data points from available case studies. These papers described how an induction motor driving a reciprocating compressor with a torsionally soft coupling can be sensitive to torsional stiffness across the motor air gap.

Figure 1 shows a diagram of a representative compressor system discussed in this paper.

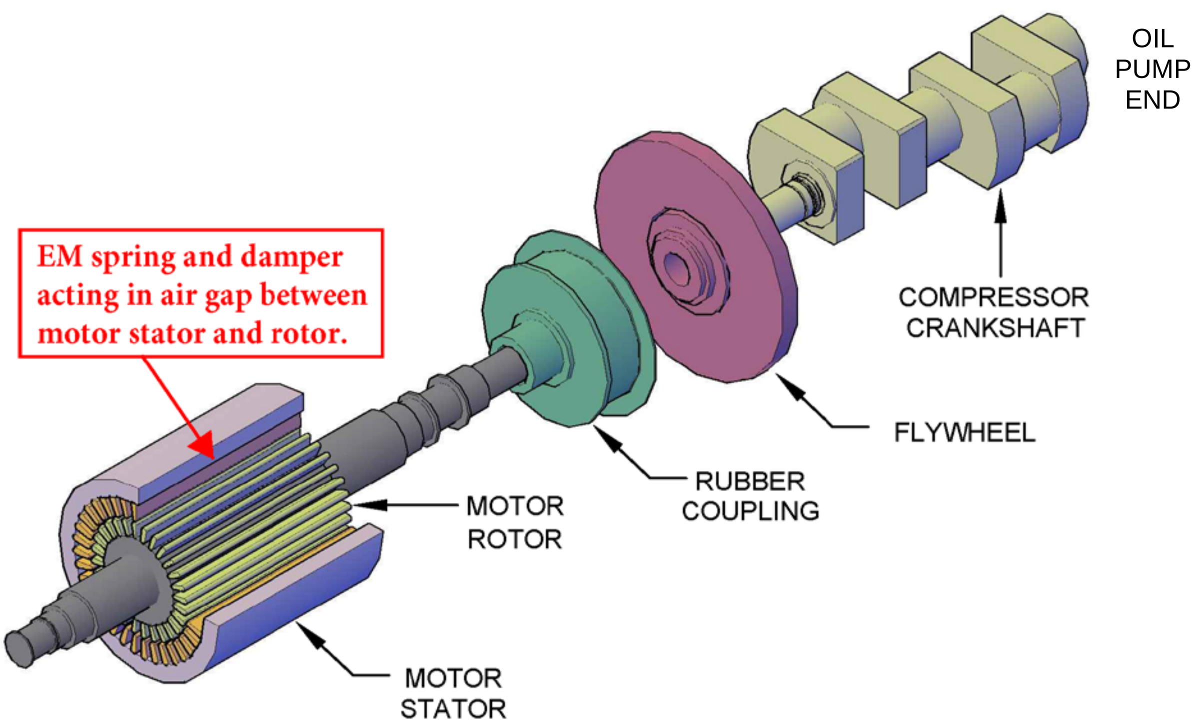


Figure 1: Diagram of VFD Motor-Driven Reciprocating Compressor System

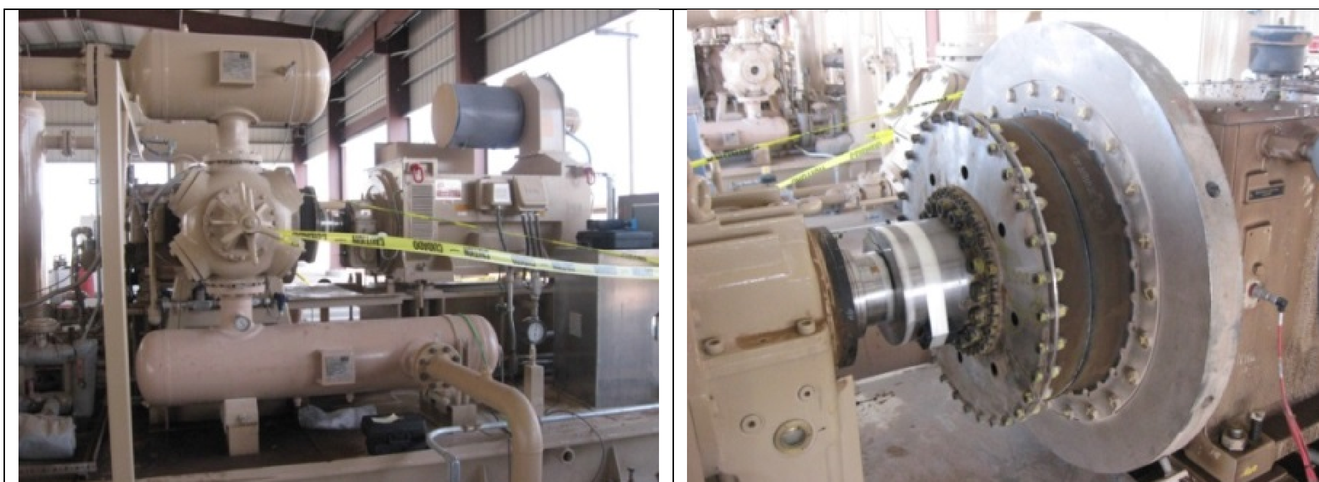
INTRODUCTION

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 to determine the acceptability of the equipment. Some compressor manufacturers recommend field testing as standard procedure to verify results of the theoretical torsional analysis when the compressor is driven by a variable speed motor, 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 Table 1. Photographs are shown in Figure 2.

Table 1: Description of 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 in-lb/rad, Dynamic Magnifier = 6 Continuous Vibratory Torque = 27,700 in-lb 0-p at 10 Hz Continuous Vibratory Torque = 13,850 in-lb 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)



a) Reciprocating Compressor and Motor

b) Rubber-in-Shear Coupling and Flywheel

Figure 2: Pictures of Compressor System

FIELD MEASUREMENTS

Measurements can be used to verify the torsional predictions. When acquired in the field, tests can be conducted under full load where the actual response can be determined, such as: 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.

Strain gages were installed on the motor shaft along with a battery-powered transmitter on the coupling hub (Figure 3). The four gages 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.

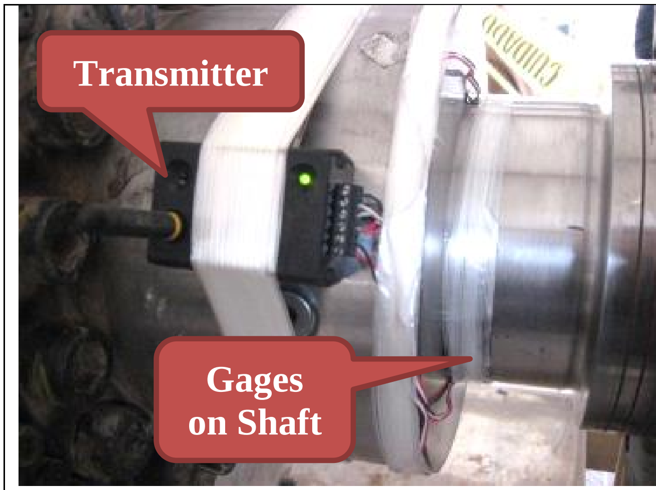


Figure 3: Strain Gage Telemetry System

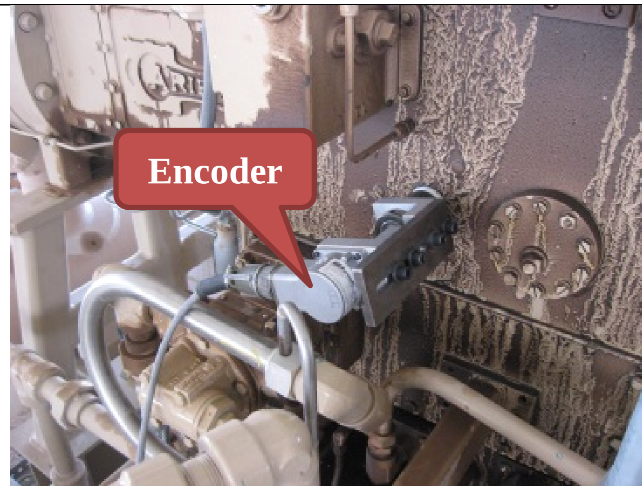


Figure 4: Shaft Encoder

An encoder was mounted on the auxiliary-end of the compressor crankshaft (Figure 4) to measure torsional oscillation. Some compressor manufacturers publish limits on the torsional velocity for the chain-driven oil pump [6]. These torsional measurements help ensure that the lubrication pump is operating within the safe zone with respect to torsional vibration. The encoder readings were acceptable and are not presented in the paper since the focus is on the electromagnetic effect which was better demonstrated in the torque measurements.

The data acquisition system consisted of 24-bit A/D converters. Proprietary software was used to control the instrumentation and digitally record the data. All of the following data plots were generated using the software.

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 5, a TNF was identified between 12 – 14 Hz.

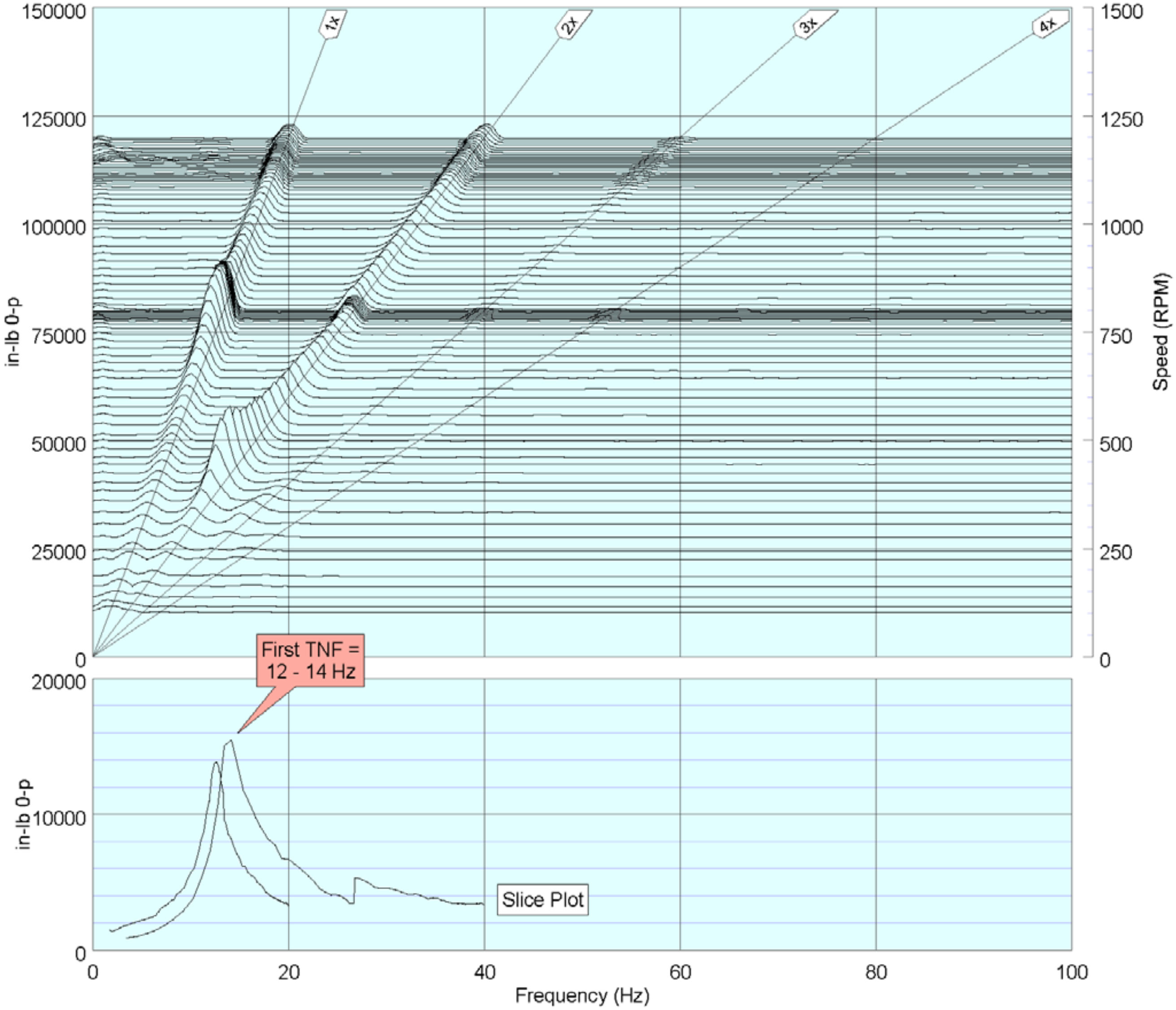


Figure 5: Waterfall Plot of Dynamic Torque vs Speed During Cold Start

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 5 was created by tracing through the 1× and 2× 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 in Figure 6 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.

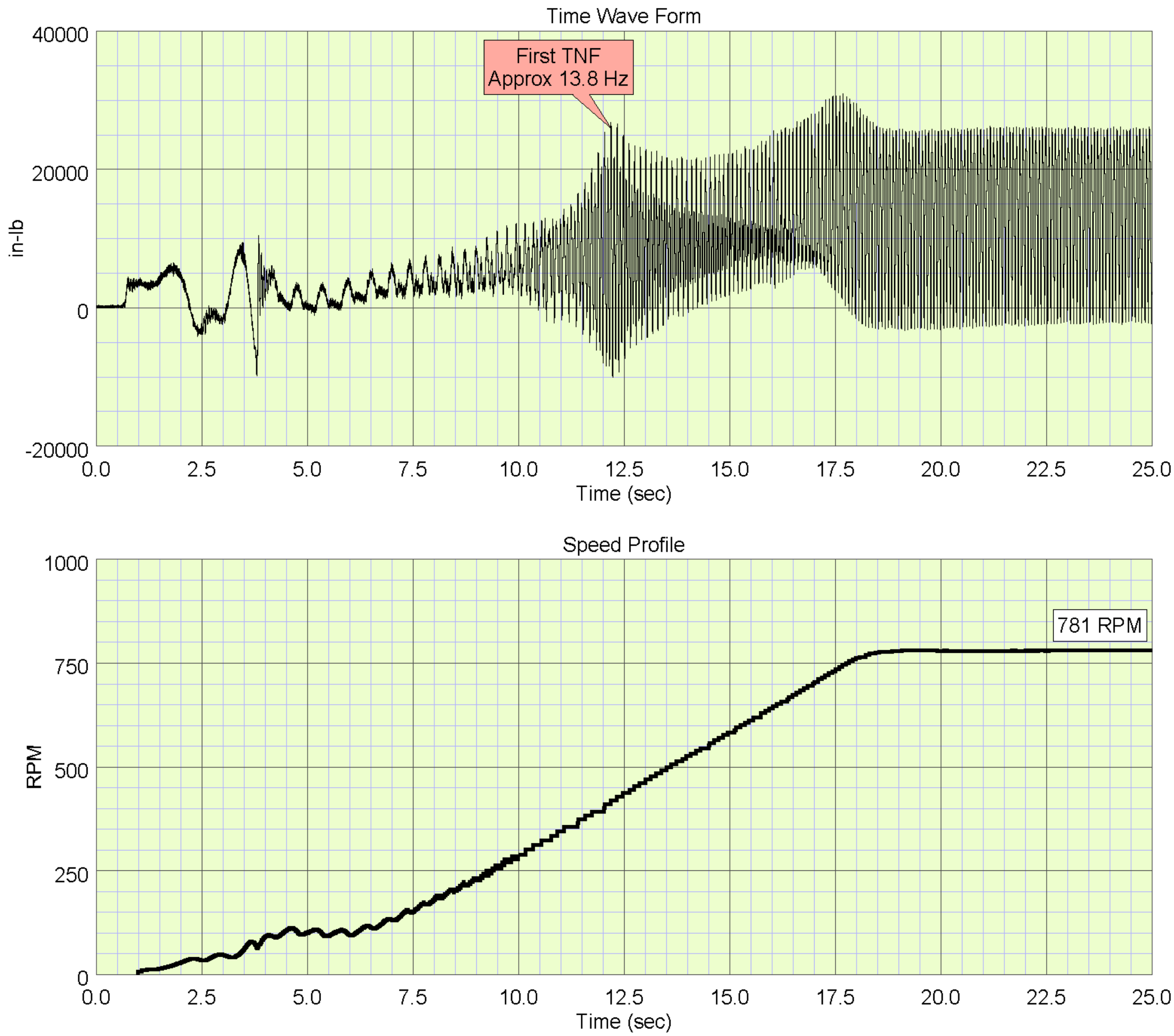


Figure 6: Time Waveform of Torque During Cold Start
(Torque shown in top trace and speed shown in bottom trace.)

A torque spike was observed when the motor was switched from the VFD to across-the-line operation. As shown in Figure 7, during this transient event the highest torque spike was approximately 100,000 in-lb (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 Hz and 12 Hz. Note that the 5 Hz frequency was not observed during the initial startup (Figure 5).

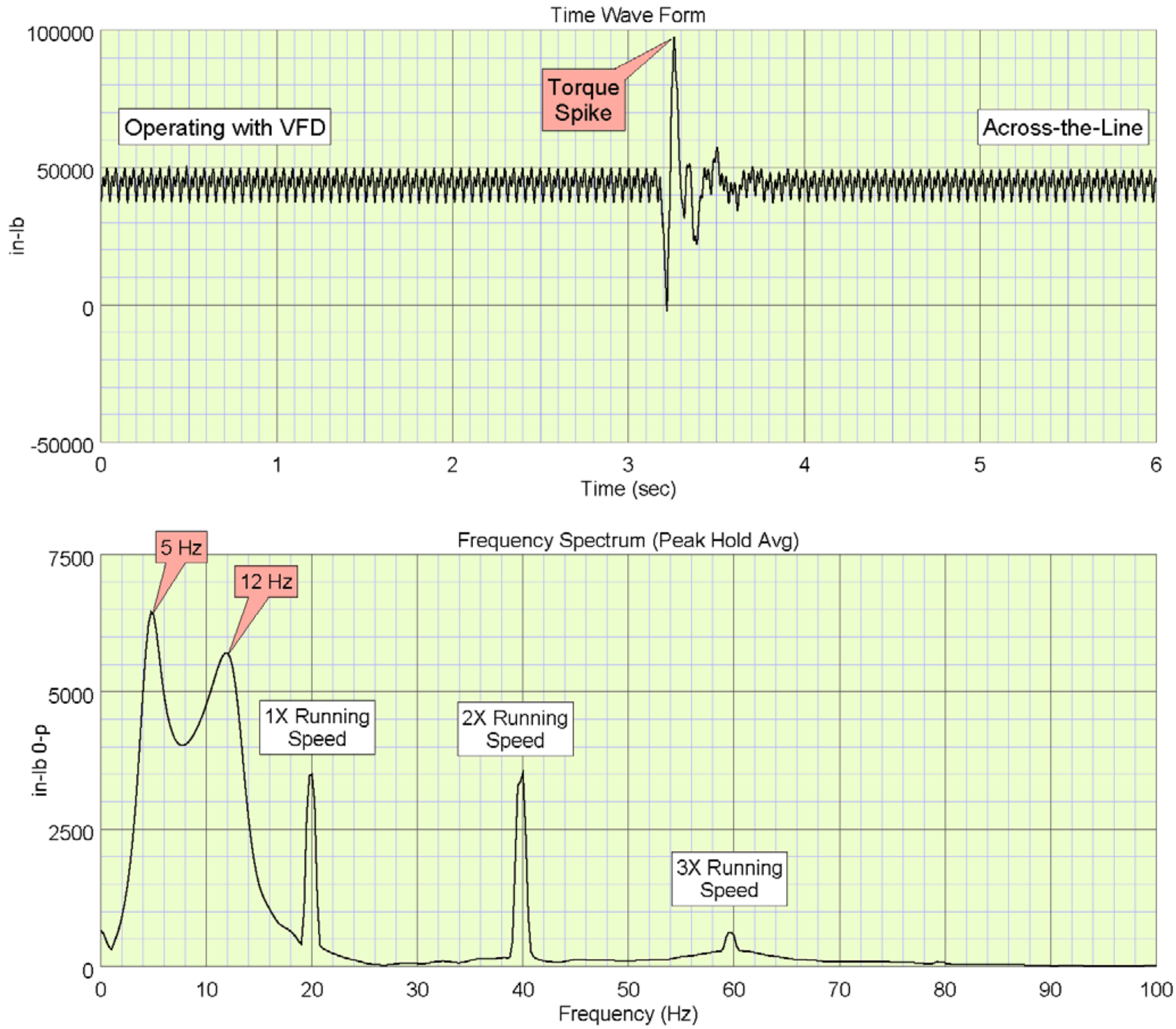


Figure 7: Torque Measured During Transfer from VFD to Across-the-Line Motor Operation – Cold Condition
(Torque vs time in top trace and frequency spectrum in bottom trace.)

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.

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 and dynamic torque through the coupling. As shown in Figure 8, the TNF was slightly lower in the warm condition (11 – 13 Hz) compared with the cold condition (12 – 14 Hz).

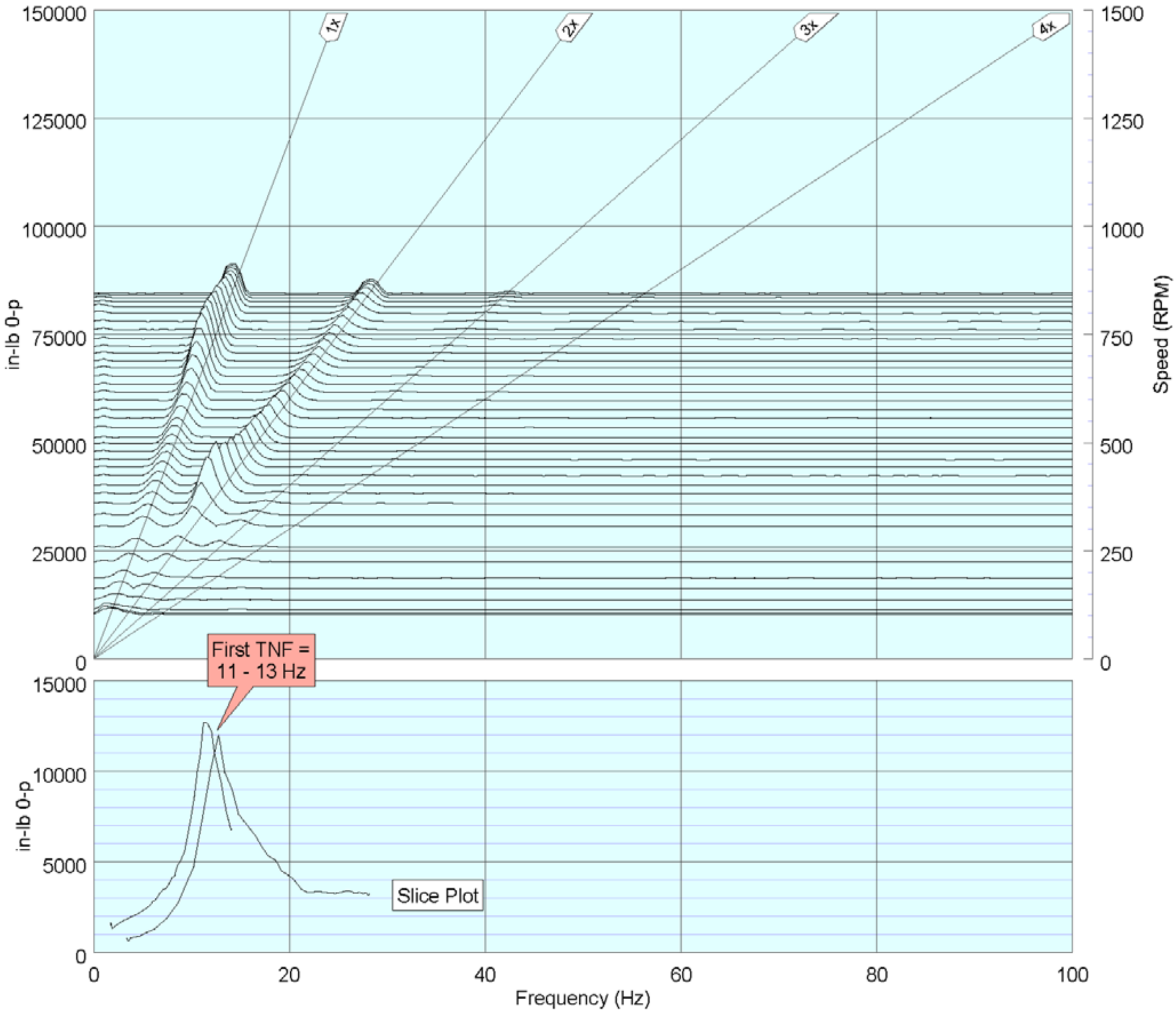


Figure 8: Waterfall Plot of Dynamic Torque vs Speed During Warm Start

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 Hz and 11.75 Hz as shown in Figure 9. These two frequencies will be discussed in more detail later in the paper.

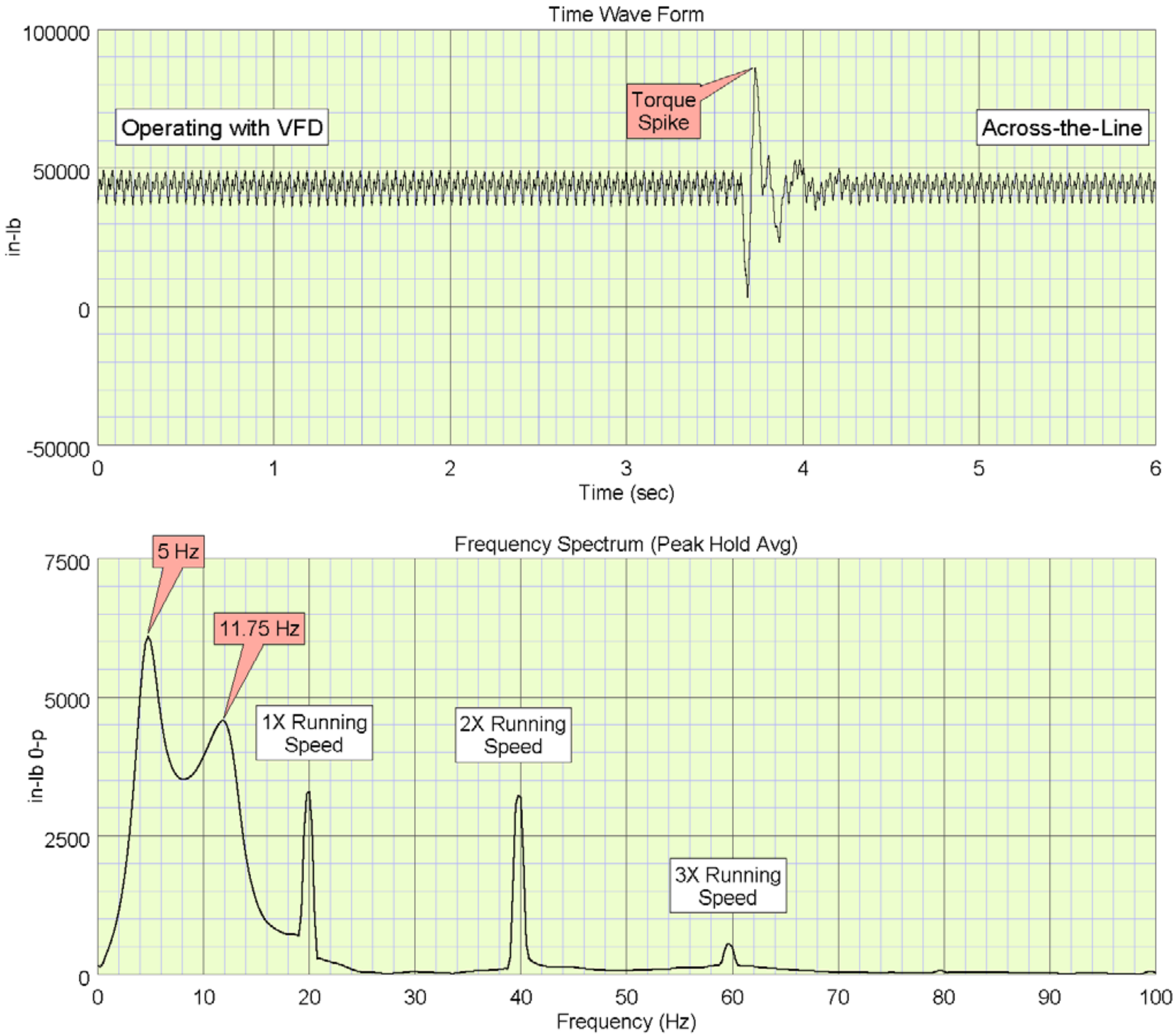


Figure 9: Torque Measured During Transfer from VFD to Across-the-Line Motor Operation – Warm Condition
(Torque vs time in top trace and frequency spectrum in bottom trace.)

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.

Figure 10 shows the time wave form of the measured torque during the loaded shutdown. Note that the maximum torque reached 100,000 in-lb, 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 as shown in the bottom half of Figure 10.

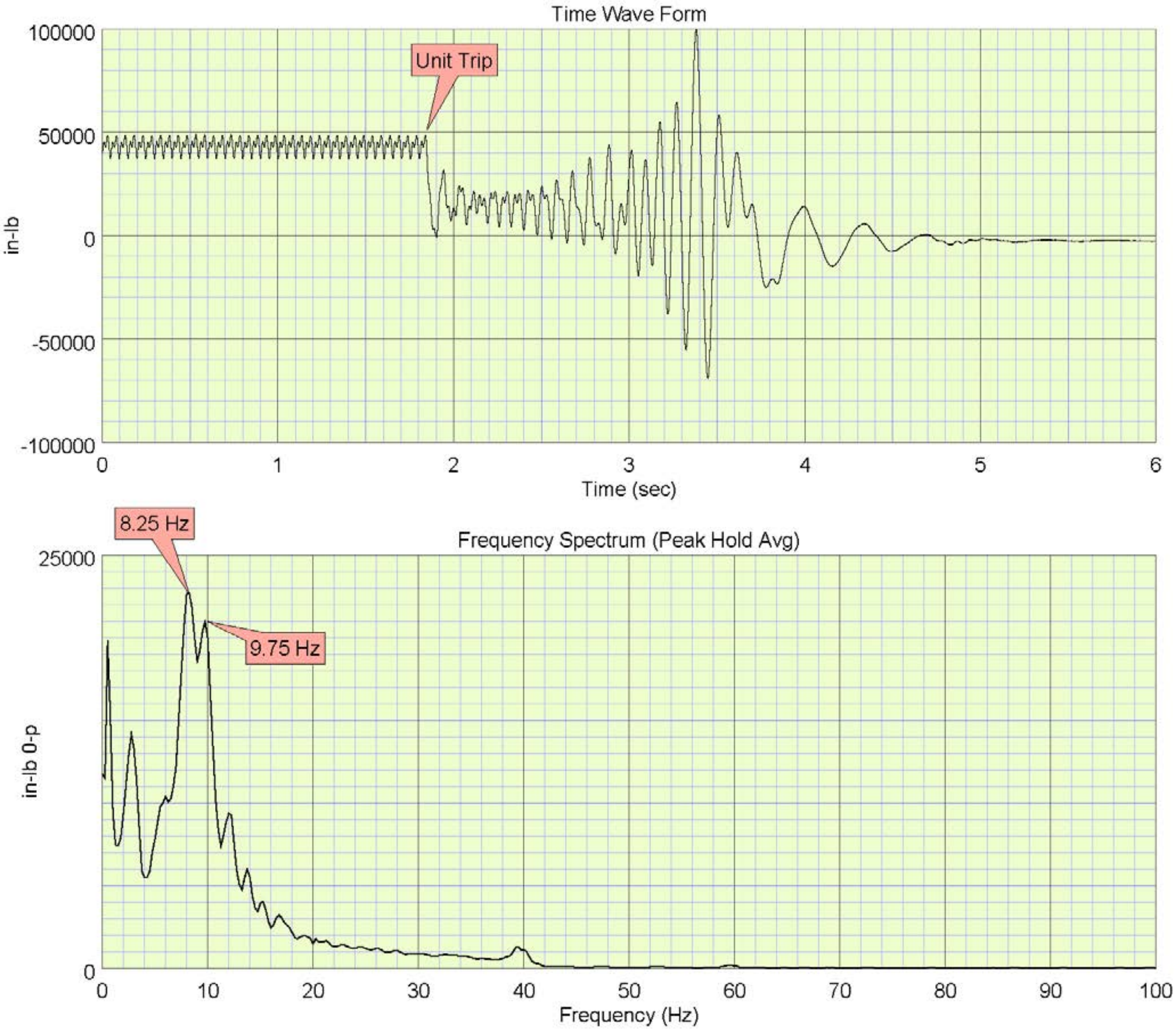


Figure 10: Measured Torque During Loaded Shutdown – Warm Condition
(Torque vs time in top trace and frequency spectrum in bottom trace.)

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.

Table 2: Summary of Measured TNFs

Condition	No Load	Load (with EM)
Cold	10 Hz	12 – 14 Hz
Warm	9 Hz	11 – 13 Hz

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 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).

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 11, the EM torsional stiffness at the motor air gap would need to be approximately 1.1×10^6 in-lb/rad to correlate with the first two TNFs that were measured in the field.

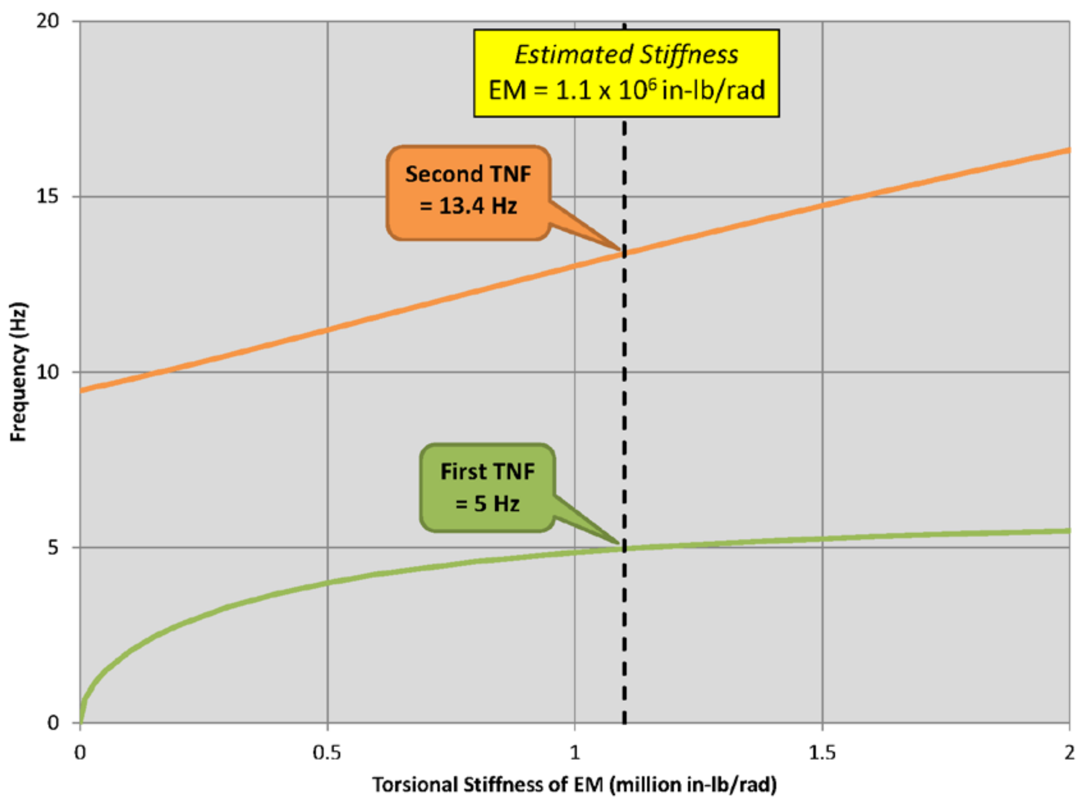


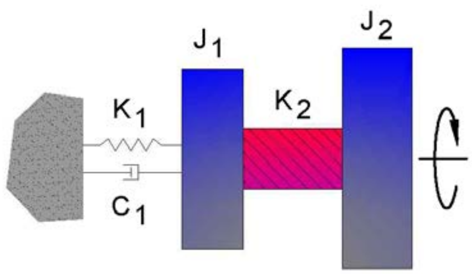
Figure 11: Parametric Study of TNFs Sensitivity to EM Stiffness

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 0.9×10^6 in-lb/rad (98.7 kNm/rad). Holopainen [7] reported a torsional stiffness of 1.4×10^6 in-lb/rad (160 kNm/rad) for an induction motor rated for 1250 HP (932 kW). In comparison, the EM torsional stiffness of 1.1×10^6 in-lb/rad is approximately 2.2× 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.

Table 3: Equivalent Mass-Elastic Values



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

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

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

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 12 and 13. 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.

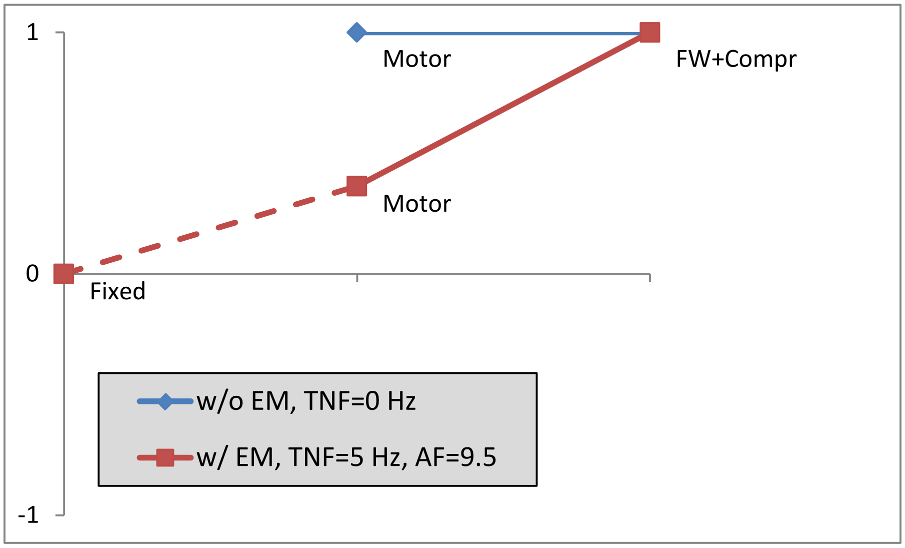


Figure 12: Torsional Modeshape Involving Rigid Body Motion

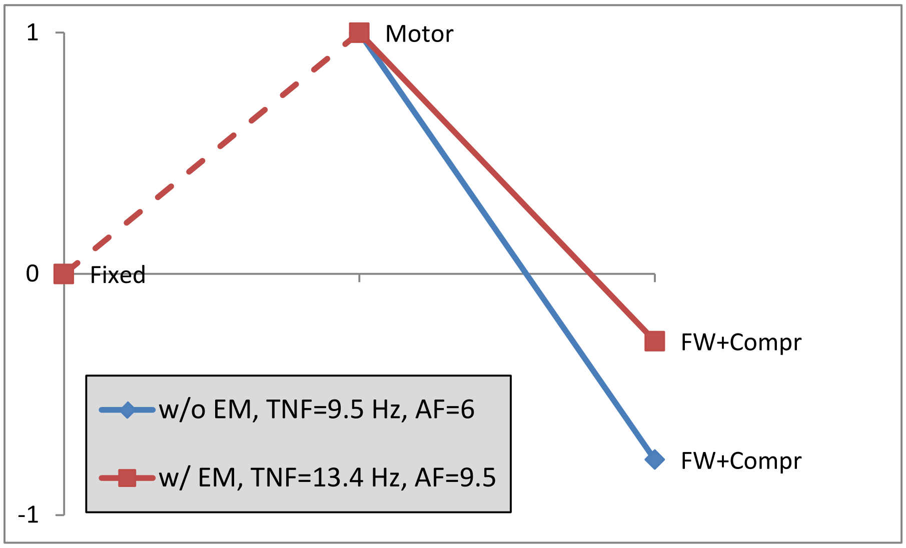


Figure 13: Torsional Modeshape Involving the Coupling

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%.

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” can 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 14 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 14, 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 13, 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.

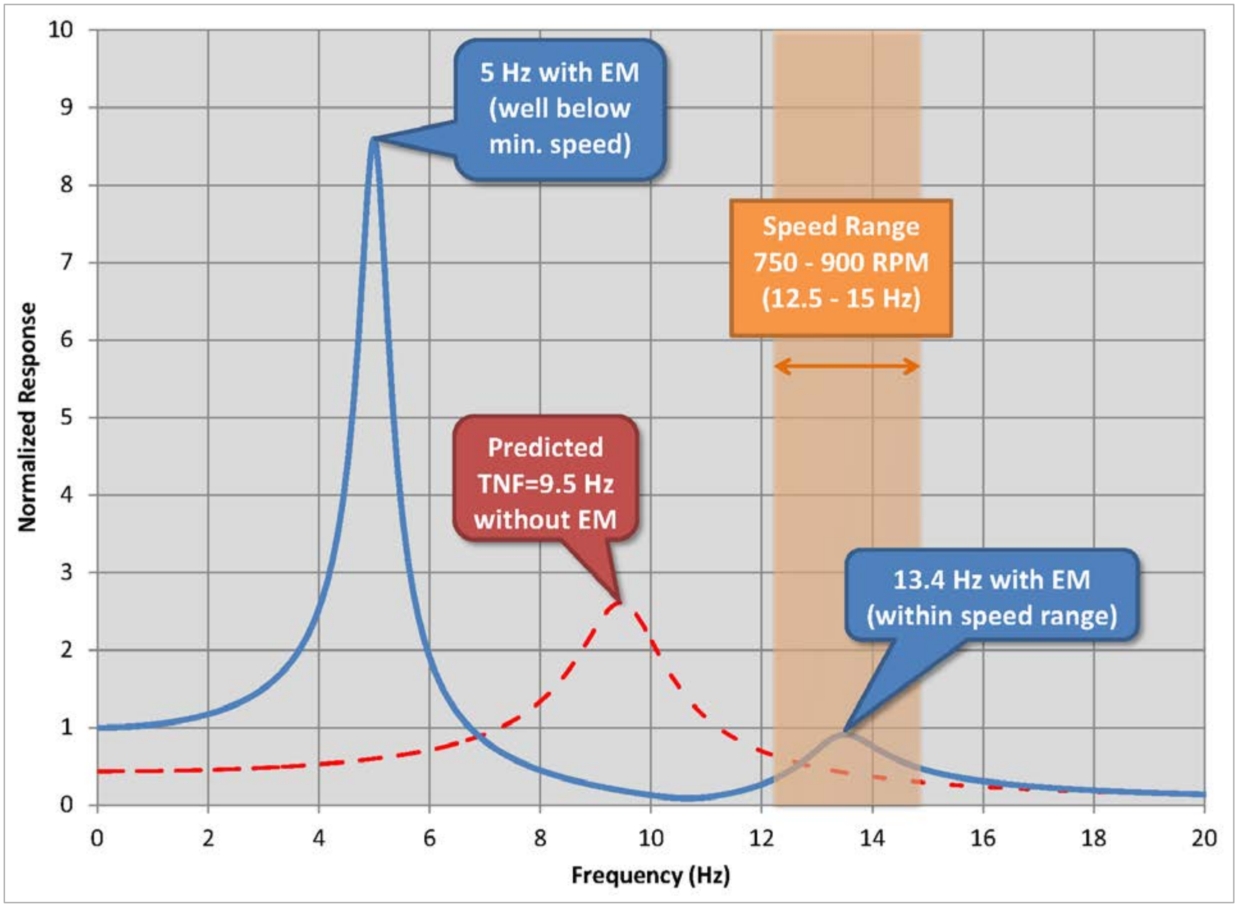


Figure 14: Comparison of Calculated Normalized Dynamic Torque in Coupling – With and Without EM

With EM included another TNF appears at 5 Hz, which is well below the minimum speed. As was previously shown in Figure 12, 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 was used instead of the torsionally soft rubber coupling. This was accomplished by performing a parametric analysis as shown in Figure 15 where the torsional stiffness ratio of the combined motor-coupling-compressor shaft stiffness / EM stiffness was varied from 0.1 to 12.

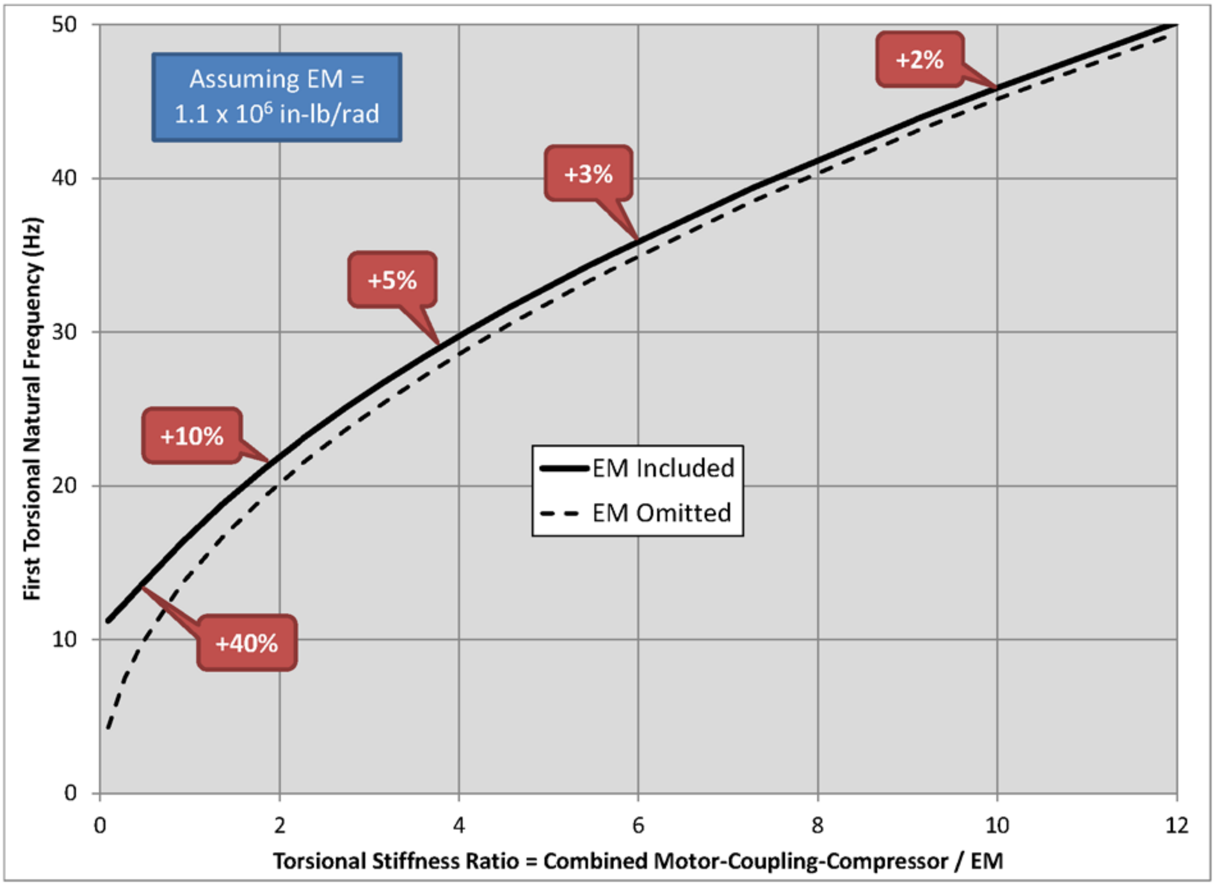


Figure 15: Parametric Study of TNF vs Stiffness Ratio

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 is four times the EM stiffness, then the increase in TNF would be approximately 5%. However, if the torsional stiffness between the motor and compressor is approximately ten times the EM stiffness, then 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. A more detailed discussion of torsionally soft couplings is contained in the Appendix.

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.

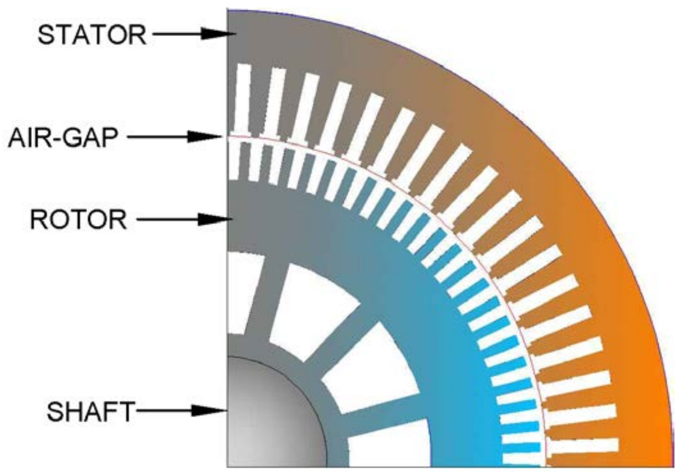


Figure 16: Diagram Showing Motor Air Gap

There are some types of machines which require more detailed analysis of the electromagnetic interaction when applied in 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 electromagnetic interaction and the effects upon the lateral dynamics of the rotor were first presented by Rosenberg [9]. The author presents a formulation to consider the UMP in the critical speed calculation. There are numerous publications concerning the influence of this interaction in electric machinery, various methodologies and models dedicated to dynamic and stability analysis of rotors.

In relation to torsional effects, a new method for torsional study is presented by Tenhunen et al [10]. These authors propose a methodology to generate a frequency response function (FRF) directly from an impulse response of the rotor. The stiffness and damping parameters are then evaluated considering the displacement caused by the impulsive force.

More recently Knop [3] suggests analytical expressions for stiffness and damping evaluation through linearization of the electromagnetic field equations. Hauptmann et al [4,5] then applied some analytical manipulation and presented equations for calculating the electromagnetic stiffness ($\mathrm{K_{em}}$) and damping ($\mathrm{C_{em}}$).

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

Table 4: Data from Electric Motor Manufacturer

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 × 2π)	Ω_s	377 rad/sec
Slip at rated load	s_r	0.0092

The first step is to calculate the electrical time constant:

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

$$T_L = 0.0544 \text{ sec}$$

Then the EM stiffness will be:

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

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

And finally the EM damping is:

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

$$C_{em} \approx 1{,}000 \text{ in-lb-s/rad}$$

As shown, the calculated EM stiffness of 0.9×10^6 in-lb/rad compares favorably with the EM stiffness of 1.1×10^6 in-lb/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.

SUMMARY AND CONCLUSIONS

- The electromagnetic interaction can be important and simple analytical methods shown in this paper should be used to estimate the EM stiffness. When included in the TVA, this will help increase accuracy of the numerical prediction.
- For this VFD motor – compressor system, the TNF associated with the coupling was measured to be 40% higher than predicted because the EM effect was not included in the initial torsional vibration analysis. Had the EM stiffness been included in the TVA, a softer coupling and/or larger flywheel would likely have been specified for this system.
- Another indication of the EM stiffness effect was the measured frequency near 5 Hz, which appeared during a transient event while the motor was fully energized. Data presented in this paper shows two peaks when the EM spring is present during loaded condition.
- Using the measured data, an equivalent spring to ground across the motor air gap was estimated to be 1.1×10^6 in-lb/rad. This agreed with the range of values (0.9 to 1.4×10^6 in-lb/rad) found in publications with similarly sized motors.
- The EM stiffness value calculated from available equations was 0.9×10^6 in-lb/rad, and is considered to be a good approximation for use in a TVA.
- 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 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 below for error estimates when EM is not considered.

Table 5: Summary of Estimated Error if the EM Effect is Omitted

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%

- 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 1× running speed, and could prematurely wear out the rubber coupling elements.

NOMENCLATURE

A/D = Analog to digital conversation
AF = Amplification factor (Indication of damping. High AF means low damping.)
API = American Petroleum Institute
C_{em} = Torsional damping due to EM
EM = Electromagnetic
FRF = Frequency response function
FW = Flywheel
Hz = Hertz (Frequency expressed as cycles per second.)
K_{em} = Torsional stiffness due to EM
N1 = Number of stator poles
P-T = Pressure - time (P-T card shows pressure versus time in a compressor or engine cylinder.)
PWM = Pulse width modulation
SM = Separation margin (Percent difference between a critical speed and the operating speed.)
s_r = Slip at rated load
T_B = Breakdown motor torque [lb-in]
T_L = Electrical time constant [sec]
T_R = Rated full load motor torque [lb-in]
TNF = Torsional natural frequency [Hz]
TVA = Torsional vibration analysis (Predictions using a theoretical mathematical model entered into a computer program.)
UMP = Unbalanced magnetic pull
VFD = Variable frequency drive
WR^2 = Polar mass moment of inertia [lb-in^2]
ω = Frequency of the superimposed torsional vibration [rad/s]
Ω_s = Electrical supply frequency [rad/s]

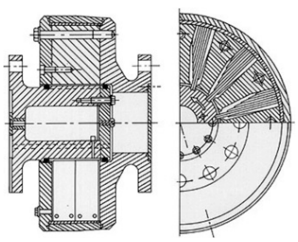
APPENDIX – TORSIONALLY SOFT COUPLINGS

A coupling is considered "soft" when it has a torsional stiffness that is much less than the shafts of the machinery. In contrast, a rigid connection would be direct flange-to-flange (no coupling). A disc coupling, commonly used for motor and engine applications, would fall somewhere between a soft coupling and a rigid connection. Soft couplings can be beneficial for controlling torsional vibration in the following instances:

- **Detune a torsional natural frequency**– A variable speed reciprocating compressor may have several damaging torsional resonances within the speed range due to the compressor excitation harmonics intersecting a torsional natural frequency. With the use of a soft coupling, a torsional natural frequency can be tuned below the minimum operating speed. In many instances, this is the only way to achieve the full desired speed range. Otherwise, the speed range has to be limited, or certain speed bands have to be avoided.
- **Add damping to the system** – Some soft couplings can attenuate high torsional amplitudes that are the result of a resonant condition. For couplings with rubber elements, this is accomplished through hysteretic damping. Hysteretic damping (internal friction) dissipates heat when the shaft or coupling material is twisted due to torsional vibration. The leaf spring type of coupling provides viscous damping since it is oil-filled. The dynamic magnifier for most resilient couplings, which is a measure of the damping properties, typically ranges from 4 to 10.

Several types of soft couplings with constant torsional stiffness are listed in Table A.1. The table lists some advantages and disadvantages of each type of coupling. Rubber-in-compression could also be considered to be a soft coupling, but is not discussed in this table because of progressive torsional stiffness that changes with applied torque.

Table A.1: Torsionally Soft Couplings

Example	Description	Advantage	Disadvantage
	Rubber-in-Shear Uses rubber-in-shear elements with a constant torsional stiffness.	Low torsional stiffness and high damping.	Limited life of rubber elements due to generated heat. High service factor required for reciprocating applications.
	Leaf-Spring Radially arranged steel leaf springs with constant torsional stiffness. Usually filled with oil for damping.	Low torsional stiffness and high damping.	Normally requires pressurized oil supply through a hollow engine or compressor crankshaft.
	Helical-Spring Uses compressed steel springs with constant torsional stiffness	Wide range of torque and stiffness values. Damper option available.	Helical-spring couplings can be massive and expensive. Special tools are required for spring replacement.

Selecting couplings with rubber elements is usually considered a “last resort” because of potential trade-offs with increased cost, longer delivery time, and periodic inspection and maintenance. For example, rubber elements will degrade over time due to heat, ozone, and other factors. Most coupling manufacturers state that the life of the rubber elements can be 3 to 5 years. However, the actual life may be significantly less if the coupling is subjected to excessive torque or harsh environment.

Although these couplings may have increased damping, excessive torques can still occur when operating near a torsional resonance, during a start-up, or loaded shutdown. Running at or near a resonant condition for just a few minutes could elevate the internal temperature beyond the melting point of the rubber elements and damage the coupling. Also, the TNFs of a system could vary for cold starts versus hot restarts. Cases have occurred where assumed damping was much higher than actually realized in the field, resulting in a non-conservative design and premature coupling failures.

When selecting a rubber-in-shear coupling, a generous service factor should be considered to allow for possible torque overload during start-up, shutdown, or any other unexpected conditions. Coupling manufacturers may require that the nominal torque rating of the selected coupling size be at least 1.5 to 2 times the transmitted torque. This would correspond to a service factor of 4.5 to 6 since the catalog rating is typically 3 times the transmitted torque.

The manufacturer should verify the final coupling selection. The vibratory torque and heat dissipation (for damped couplings) must be calculated and should also be reviewed by the coupling manufacturer for acceptability. Other factors such as end-float and allowable misalignment should also be addressed with the coupling manufacturer.

In instances where a very soft coupling is required to achieve an acceptable system, a multi-row coupling (two or three rows of soft elements in series) can sometimes be used; however, this is not normally recommended since a torsional resonance of the coupling pieces can result. Instead, a standard soft coupling in conjunction with a flywheel is preferred. For example, significant inertia can be added with a flywheel at the coupling hub or with internal flywheels sometimes called “donuts” or “detuners” mounted on the crankshaft spreader inside the compressor frame. Rotating counterweights can also provide additional inertia similar to flywheels.

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