LATERAL AND TORSIONAL VIBRATION PROBLEMS IN SYSTEMS EQUIPPED WITH VARIABLE FREQUENCY DRIVES

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ABSTRACT

Several unusual problems involving lateral vibration and torsional vibration have been seen recently in systems equipped with variable frequency drives and different types of driven equipment. The drives and motors are from different manufacturers and vary widely in size. The vibrations seem to be consistent in that there is a vibration frequency that remains constant as the shaft speed of the motor changes. Sometimes the vibrations are seen in accelerometer readings, but more consistently the vibrations are torsional. Changes in the VFD control system have helped achieve satisfactory operating conditions in systems that originally could not run without experiencing failures. Field testing and modeling have been essential to diagnose and solve the problems.

1. INTRODUCTION

Recently, Beta Machinery Analysis (BMA) has witnessed several vibration problems associated with variable frequency drives (VFDs). The cases presented in this document exhibit one, or a combination, of the following symptoms:

- A fixed frequency of vibration (either torsional or linear/lateral/transverse), independent of shaft speed, and
- Amplification of torsional vibrations, particularly on the motor side of the system.

Significant time was spent troubleshooting some of these cases, only to find out that relatively simple changes to the VFD control system, or VFD parameters, were sufficient to mitigate the problems.

Two case studies are presented in this paper. The cases include two types of machines: cooler fans and reciprocating compressors. The largest portion of this paper is related to the reciprocating compressor case.

The troubleshooting assignments included field measurements using accelerometers, strain gauges, torsional demodulation and current analysis. Several entities contributed to the diagnosis and resolution of the problems, including end users, equipment manufacturer, and consulting firms.

This paper does not address basic theory of lateral or torsional vibration, sources of excitation, finite element analysis, or data collection. The bibliography includes material related to the basic theory.

2.   CASE HISTORY 1

2.1 **System Description**
The system discussed consists of two four-throw pipeline reciprocating compressors driven by induction motors. Figure 2.1 shows a schematic configuration of one package. There is a large flywheel between the motor and the compressor. One of the compressors is normally started up using the VFD and then switched across the line, delivering the maximum capacity. The second compressor's speed is varied to match the production requirements. Any of the compressors can lead or lag. The VFD manufacturer did not supply the motors.
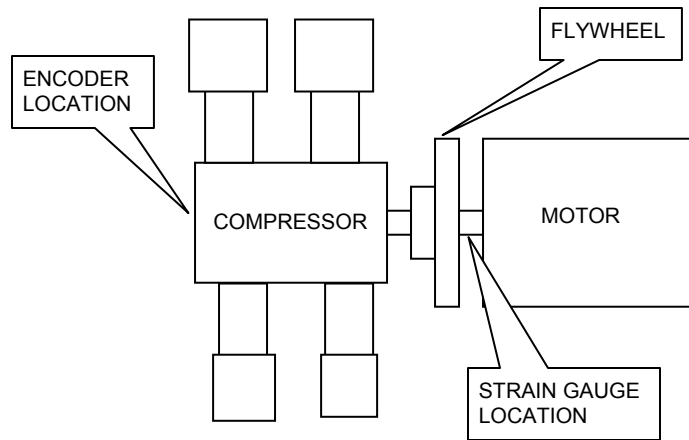

Figure 2.1. Compressor Package Layout

2.2 **Symptoms**
Failures:
Several motor shaft failures in the stub shaft area have occurred, caused by torsional vibrations and improper shaft materials.

Vibrations:
The lateral vibration at the motor drive end and at the compressor drive end were measured using accelerometers oriented parallel to the piston motion (horizontal direction). The torsional vibration at the motor shaft was measured using strain gauges and the torsional vibration at the compressor auxiliary end was measured using an encoder coupled to a signal demodulator.

The lateral vibrations were not high. The spectral analysis at the accelerometer locations showed no evidence of high amplitudes at the frequencies at which the torsional vibrations occurred. The pattern of the vibration at the two points measured matched the typical pattern of vibration on a reciprocating compressor package; that pattern consists of amplitudes at multiples of run speed, with the highest amplitudes at 1X and 2X run speed.

Torsional vibrations (TVs) were measured by analyzing the dynamic component of the torque at the motor shaft and the angular displacement at the auxiliary end of the compressor shaft. The TVs were an issue on the motor side of the flywheel. TVs were lower than predicted on the auxiliary end of the compressor. The difference in the amplitudes at these locations shows that, in order to assess the torsional behavior accurately, the measurement technique should be matched to the operating deflected shape of the train. In case of uncertainty several techniques or locations should be used, or the results of a finite element analysis should be consulted.

The original symptoms were exhibited in one of the systems. As the speed was increased and the first torsional critical speed was excited, the TVs peaked and then the amplitude of the torsional vibration stayed high at the TNF even though the motor speed was still increasing. This speed sweep process is shown in Figure 2.2. The trend shows two traces: mean torque and dynamic torque. The higher values while operating under VFD control correspond to the dynamic torque caused by torsional vibration.

The pattern of high torsional vibration was similar for loaded and unloaded conditions.

Unit 2 did not exhibit the "lock-in" torsional vibration during the initial tests; however, after some changes in VFD parameters, the pattern matched that of Unit 1.
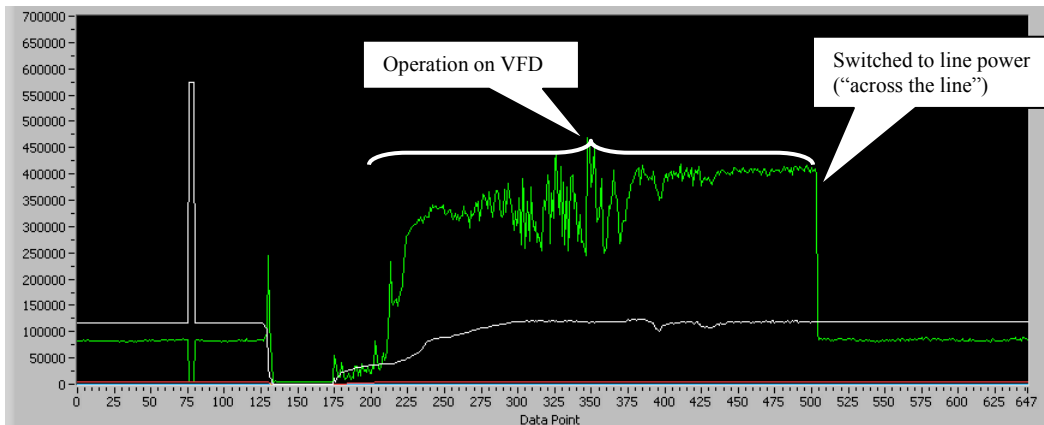

Figure 2.2.   Trend showing the peak to peak or dynamic torque fluctuation (green trace) and the Average torque (white trace). The dynamic torque drops significantly after switching from VFD operation to Line Power

Figure 2.3 shows the waterfall of the dynamic torque spectra during the speed sweep and switching process. Two orders of run speed can be seen clearly; 3X and 4X run speed. These orders increase frequency as the motor is sped up from 700 rpm to 900 rpm. The amplitude at 50 Hz increases significantly when the motor speed reaches 750 rpm and torsional resonance occurs. At higher speeds the 50 Hz peak reduces its amplitude; however, it is still higher than any other order, despite operating out of resonance.
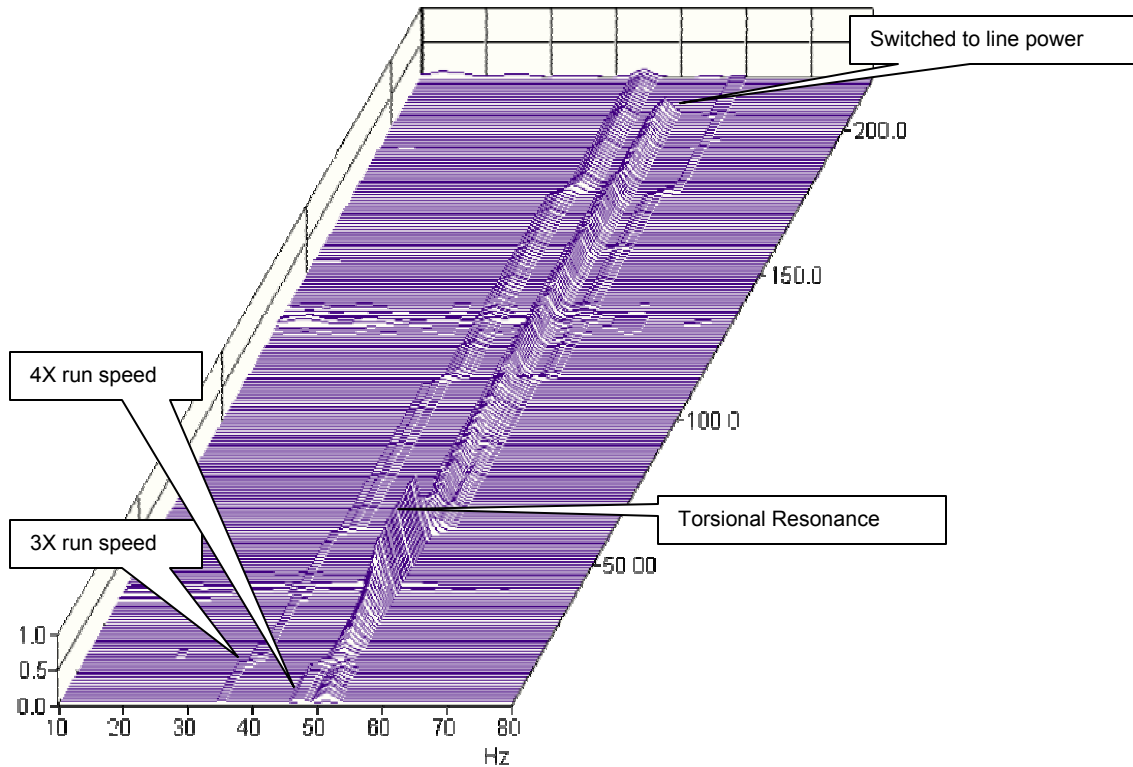
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Figure 2.3.   Torsional Vibration - motor shaft - Speed Sweep - Loaded
                    3X and 4X components vary with speed
                    Peak at 50 Hz is shows the highest amplitude
                    Switched to line power after trace 200 and 50 Hz peak disappeared

Figure 2.3 also shows that the peak at 50 Hz virtually disappears after switching to line power.

Figure 2.4 shows the waterfall of the crankshaft auxiliary end spectra during the speed sweep and switching process. Six orders of run speed can be seen clearly: 1X through 6X run speed. These orders increase frequency as the motor is sped up from 700 rpm to 900 rpm. The amplitude at 50 Hz increases significantly when the motor speed reaches 750 rpm and torsional resonance occurs, but the rise is not as much as that in the torque signal. At higher speeds the 50 Hz peak reduces its amplitude and it is still present despite operating out of resonance. After switching to line power, the 50Hz peak disappears.
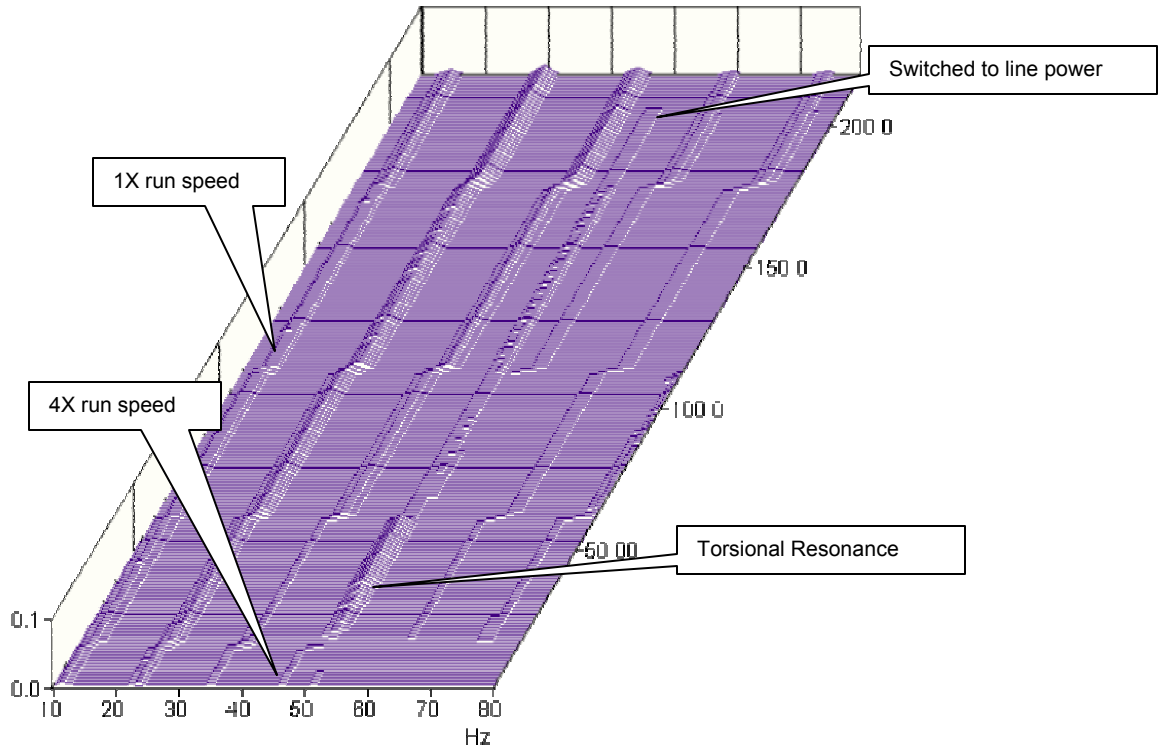
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Figure 2.4. Torsional Vibration - compressor auxiliary end - Speed Sweep - Loaded
1X through 6X components vary with speed
Peak at 50 Hz is not the highest amplitude, but is present while VFD
Switched to line power after trace 200 and 50 Hz peak disappeared

Figure 2.5 shows a spectrum of the torque oscillation when the motor is run under VFD control at 900 rpm. This plot corresponds to a slice of the waterfall shown in Figure 2.3, just before switching to line power. Notice that the 49.87 Hz peak (referred as 50 Hz peak in other paragraphs) is much higher than any other orders of run speed. 49.87 Hz is not synchronous. A slice of the same waterfall, after switching to line power, is shown in Figure 2.6. The spectrum of the dynamic torque under line power shows no evidence of 50 Hz excitation. The change in the dynamic torque pattern is not caused by mechanical loading, as the mechanical loads do not vary when switching from VFD to line power.
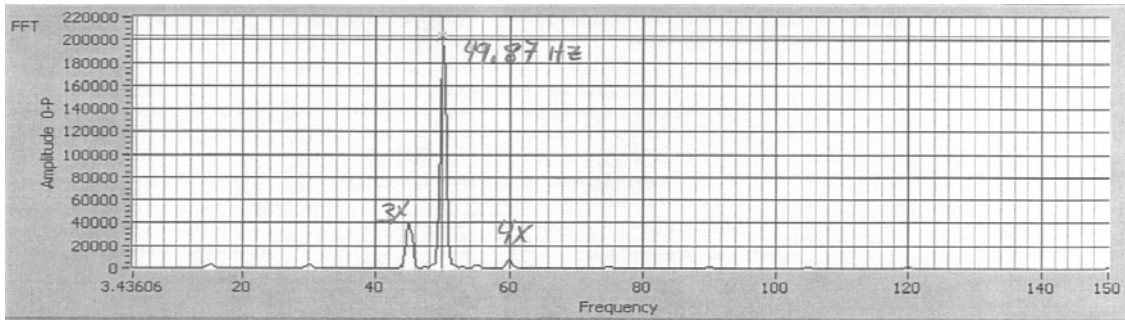
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Figure 2.5. Torque Oscillations – High amplitudes at 49.87 Hz – VFD Control 900 rpm
The peak at 49.87 Hz is not a multiple of run speed
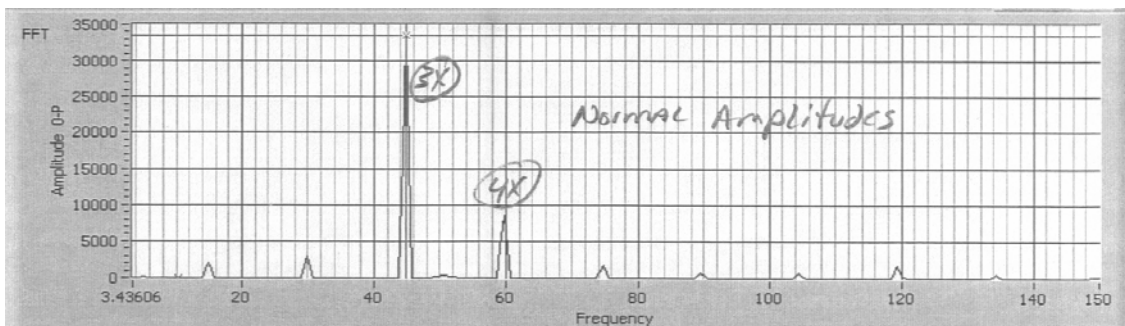See coincidence of sidebands on the current spectrum (Figure 2.7)

Figure 2.6. Torque Oscillations – Normal Amplitudes at 895 rpm – across the line
Amplitudes at multiples of run speed.
Levels are much lower than those seen under VFD Control.

Mechanical Natural Frequencies (MNFs):
Not an issue in this case.

Torsional Natural Frequencies (TNFs):
The TNF measured in the field is 50 Hz. The prediction using computer models was between 47.2 and 51.6 Hz. The measurements are in agreement with the predictions. The damping used in the predictions is consistent with the damping measured in the field.

Current Pulsations:
Figures 2.7 and 2.8 show the spectra of the current signals under two different operating modes for the as-found system: VFD control (900 rpm) and across the line (895 rpm), respectively.

The operation under VFD control shows high amplitude at 60 Hz (center frequency) and sidebands at 49.87 Hz from the center frequency. The 49.87 Hz is the same frequency at which the torque shows the highest oscillating amplitudes (see Figure 2.5). The contrast between VFD operation and across-the-line operation is the lack of sidebands in the latter.
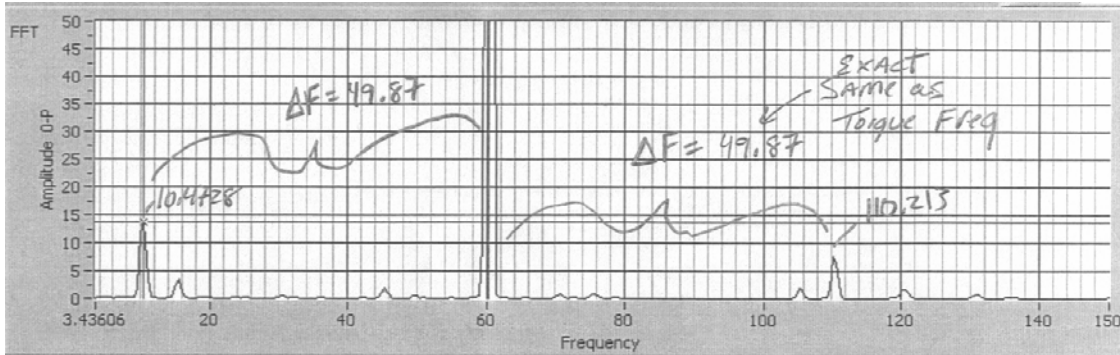
Figure 2.7.   Current Oscillations – High Amplitudes at 900 rpm – VFD Control
Sidebands at 49.87 Hz from the center frequency and small peaks at
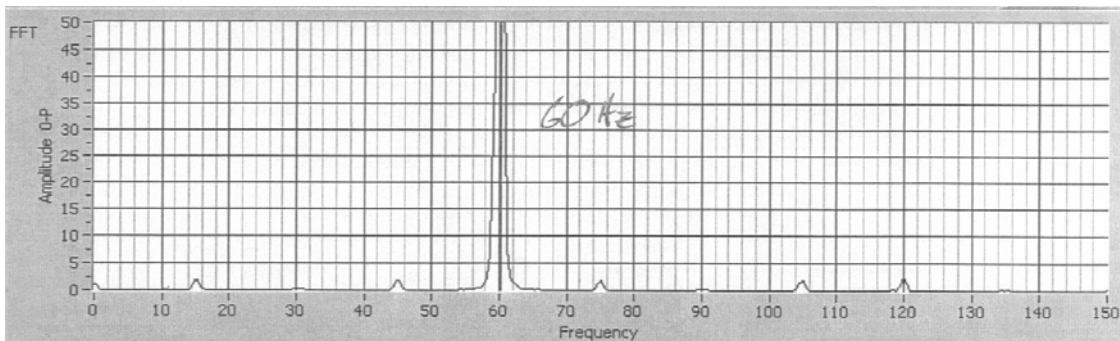multiples of run speed (15, 45, 105 and 120 Hz)

Figure 2.8.   Current Oscillations – Normal Amplitudes at 895 rpm – across the line
No sidebands of significant amplitude
Small peaks at multiples of compressor run speed (15, 45, 75, 105 and
120Hz)

2.3 **Mechanical Changes Attempted**
The motor shaft material problems have been corrected. This has had no effect on the
vibrations, but will give better motor life.

2.4 **Electrical or Software Changes**
Several changes were made to different parameters of the VFD software by the VFD
manufacturer. We are not privy to the changes made. In some instances the torsional
vibration was increased even further.

The result of the last set of changes was that the 50 Hz excitation was significantly
reduced off-resonance. The two motors then behaved similarly under VFD control. One
system showed higher TVs than the other, but the amplitudes were improved from the as
found configuration. The lock-in pattern was eliminated and the torsional vibration
reduced as the speed moved away from resonance. There was still some remaining
torsional vibration at 50 Hz while the motor was under VFD control; however, the
amplitude was much lower than the 3X or 4X torsional components.

The torsional vibration after the changes were made to the VFD parameters showed a pattern similar to that of a traditional motor/compressor train, in which the torsional vibration increases as the speed approaches resonance and then reduces as the speed moves away from resonance. Figure 2.9 shows the waterfall of the final condition.
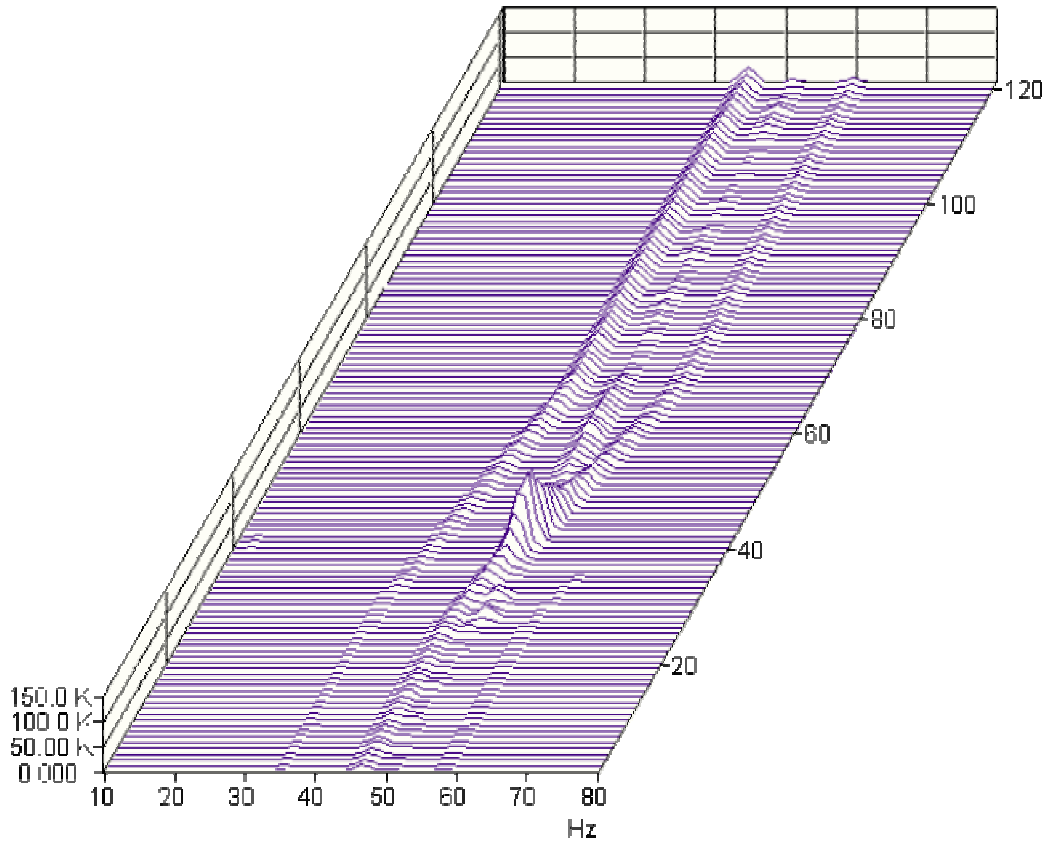

Figure 2.9.   TV at motor shaft - Speed Sweep – Loaded – Modified System
3X and 4X components vary with speed
Peak at 50 Hz still shows the highest amplitude when the 4X component passes through resonance, and then decreases significantly

Figures 2.10 and 2.11 show the overall torsional vibration traces for the first unit under maximum speed loaded conditions and VFD control. These traces are included to illustrate the changes. The peak to peak amplitude is displayed on the plot. The amplitude of the overall torsional vibration was reduced 54%.
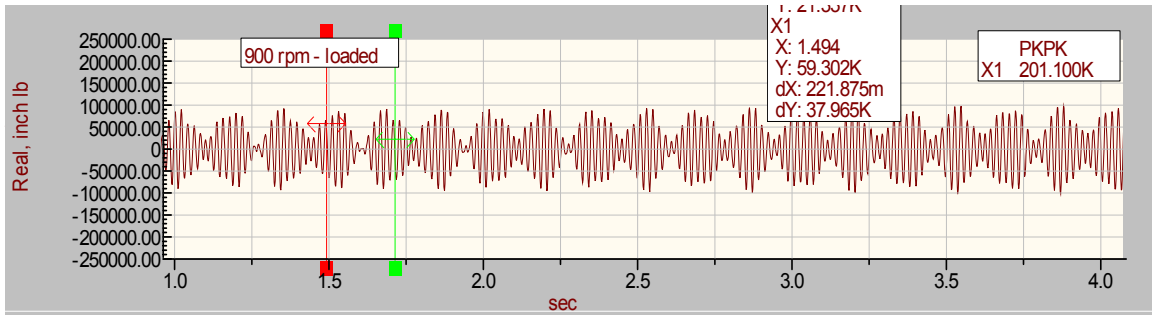
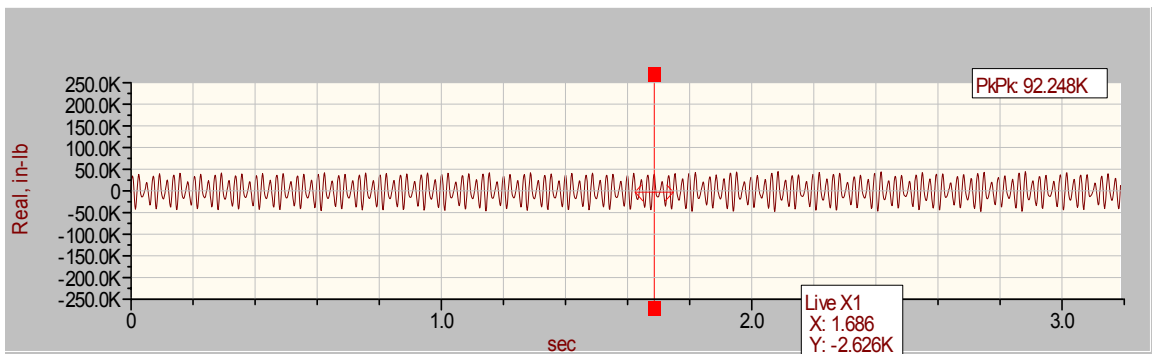**Figure 2.10.** As-Found System - Loaded – Highest Speed under VFD control

Figure 2.11. As-Left System - Loaded - Highest Speed under VFD control

2.5 Case History 1 Conclusions

We believe that the compressor torque fluctuations cause speed fluctuations in the motor, which lead to fluctuations in the variables monitored by the VFD. The VFD then changes its output to correct for the deviations from the desired operating conditions. The change in VFD output excite the torsional system, resulting in even larger torsional vibration.

The flywheel apparently isolates the compressor from the torsional vibrations (torque fluctuations) coming from the motor. It was originally installed to isolate the motor from torque fluctuations coming from the compressor.

Designers of torsional systems can only assume that Variable Frequency Drives will not amplify the torsional vibrations in a system at this time. In fact, we have seen that amplification does occur. Therefore, the VFD design should consider the torsional oscillation of the motor/compressor system to prevent or eliminate amplification.

3. CASE HISTORY 2

3.1 **System Description**

Eight motors driving cooler fans (eight blades) through cog belts. The drive manufacturer did not supply the motors. See Figures 3.1 and 3.2.

3.2 Symptoms

Vibrations:

The worst vibrations were in the east-west direction, which is the direction from the motor to the fan. The operating deflected shape (ODS) of the vibration exhibited a rotational motion about a vertical axis through the base of the motor.

The vibrations were so large they were visible. They were so extreme that the system could not have been run. In fact, a Unit F motor bearing may have been damaged in the relatively limited testing that was done in the attempts to troubleshoot the vibration problem.

Figure 3.1.   Motor, belt and driven sprocket on unmodified system

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Figure 3.2.   Fan and driven sprocket viewed from below

Table 3.1, below, summarizes the key vibration amplitudes and frequencies. The dominant frequency was observed after the speed increased to that frequency. After the speed increased further, the dominant frequency remained and only increased in amplitude as the motor speed increased.

Table 3.1. Summary of Vibration Data from Testing Modifications		
System Test Description	**Dominant Fixed Frequency (Hz)**	**Frequencies and Ratios**
Unit F, as found	20.8	Rated motor speed = 30 Hz Speed range = 5 to 30 Hz Belt ratio = 4.86:1
Unit F with Brace 1	22.3	7% increase in dominant frequency, but vibrations still very high
Unit F with Brace 1 and 4 of 8 fan blades removed	24.6	24.6/22.3 = 1.1 (calculated TNF ratio with 4 and 8 blades = 1.1)
Unit D, as found	21.56	Vibrations: 5.3 ips pk
Unit D with Brace 2 (tube)	20.5	Vibrations reduced to 1.6 ips pk
Unit F without VFD	n/a	Vibrations reduced to 0.32 ips pk

Mechanical Natural Frequencies:

The relevant MNFs are shown in Table 3.2 below. Figure 3.3 below shows a typical mechanical natural frequency measurement.

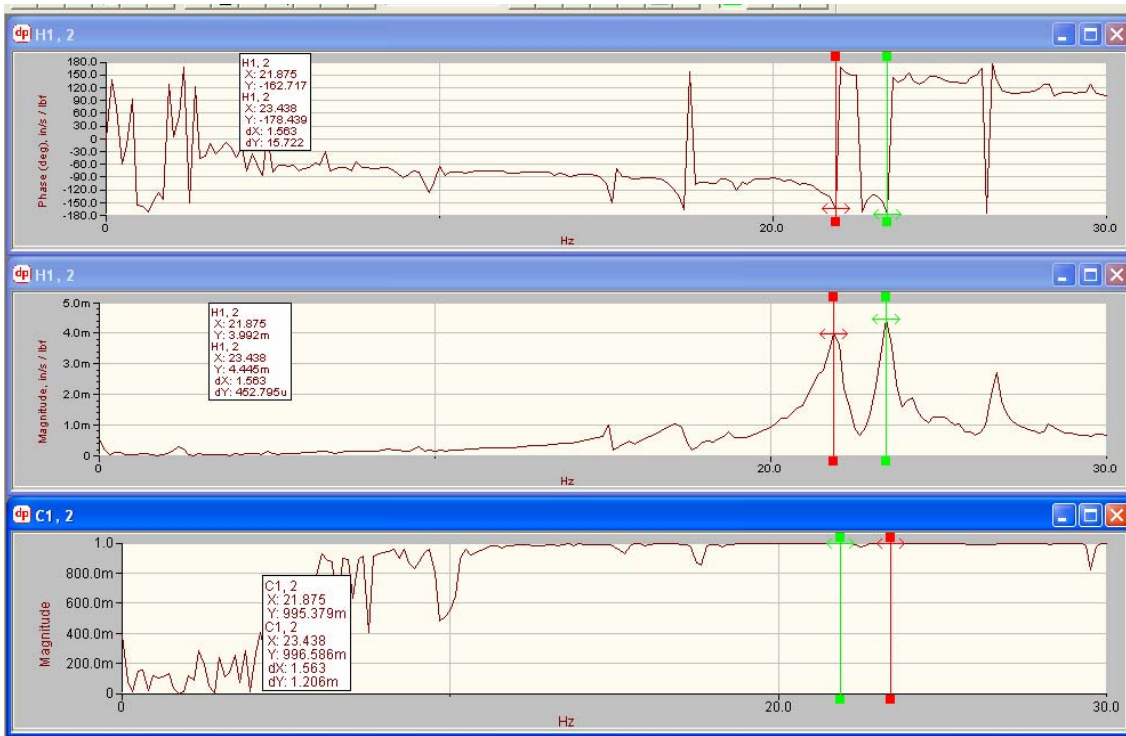| Table 3.2: Mechanical Natural Frequencies of the Motor		
Unit	Modification	Description of MNF
D	As found	21.9, rotational mode about a vertical axis through the motor base, and 23.4 Hz
D	Brace 1	Not much change
D	Brace 2	27.6 Hz, rotational mode about a vertical axis, and 23.4 Hz unchanged

Figure 3.3. Typical MNF, measured in the east-west direction on a motor, as found

Torsional Natural Frequencies:
Not measured, but the calculated values shown in Table 3.3 were consistent with the linear vibration frequencies observed.

Stiffnesses Included in the Calculation	Inertias Included	TNF Hz
Table 3.3: Calculated Torsional Natural Frequencies (done after leaving the site)		
Fan shaft and belt	8 blades and motor	31.5
Fan shaft and belt	4 blades and motor	34.8
Fan shaft, belt and motor stub shaft	8 blades and motor	23.8
Belt	8 blades and motor	144

Note: It is interesting to see that the cog belt is stiffer than the shaft.

Current Pulsations:
We wanted to measure the current pulsations, but the problem was solved before we could install a current transformer. Frequencies in the motor currents should be a useful tool in troubleshooting this type of problem.

3.3 **Mechanical Changes Attempted**
- **Brace 1**
  This brace was installed before BMA arrived to the site. It did not make much difference to the vibrations. The dominant frequency increased about 7%, as shown in Table 3.1.

Figure 3.4.   Braces in the north-south direction.

- **Brace 2**
  This brace was designed to increase the stiffness in the rotational direction about a vertical axis. It was quite effective and the vibrations were reduced dramatically, but not enough to be acceptable. The dominant, or fixed, frequency of vibration actually went down by about 7% with this brace installed.

Figure 3.5.   Looking west at the rotationally stiff brace on motor D

- **Remove 4 Blades**
  One of the tests done before BMA arrived to the site was the removal of half of the fan blades. The only obvious effect of this change was to raise the torsional natural frequency of the system, as discussed earlier. The dominant frequency of the vibration increased in the same ratio as the calculated TNFs changed. The vibrations were still very high.

- **Remove Belt**
  A motor was run alone. The vibrations were very low.

- **Remove VFD**
  Unit F motor was run without the VFD. This took some time to arrange as a starter had to be located and installed. The vibrations were acceptable. This was the same result as reported by the manufacturer of the cooler when a motor-fan was tested in the factory without a VFD.

3.4   **Electrical or Software Changes:**
The pulse frequency was changed from 1.5 to 3.5 kHz, based on advice over the telephone from the drive manufacturer's representative. Subtle changes in the vibrations were noted, but the changes were of no practical significance.

The "load reactor" was removed from the electrical system, but no changes resulted.

The load reactor was returned and the "line filter" was removed. The effect of this change was dramatic. The vibrations on the motor dropped to an acceptable level. Unfortunately, the line filter is required to protect the motor from overheating due to the harmonics in the power system.

The drive manufacturer had a software change available to correct the problem. This was installed and after some adjustments of parameters, the vibration problem went away.

3.5 **Case History 2 Conclusions**

The presence of a torsional natural frequency in the system, in an apparently critical frequency range, caused motor speed fluctuations that sent current fluctuations from the motor to the VFD. The VFD and the electrical system combined to amplify the current fluctuations.

The mechanical natural frequency of the motor further amplified the vibrational response of the system.

Although the mechanical system responses had a significant effect on the frequency and amplitude of the vibrations, the fundamental cause of the high vibrations was a feedback of energy from the VFD.

4. STUDY CONCLUSIONS

- Two different VFD manufacturers in these cases were reluctant to admit that mechanical vibration problems can be caused by the electrical side of the system. This experience suggests that most VFD manufacturers will behave the same way.
- A system approach to the design of a drive-motor-driven machine may be helpful. The system should include an understanding of the torsional response of the mechanical system.
- A thorough start-up lateral and torsional vibration, and current pulsation check for all systems with Variable Frequency Drives should be done. The measurement techniques should be matched to the operating deflected shape of the system. If the operating deflected shape of the system is unknown, several techniques can be used simultaneously, or finite element analysis can be conducted.
- There is a coupling mechanism between electrical and mechanical systems through the torsional vibrations imposed on the motor by the system. Speed fluctuations of the motor cause fluctuations in the signals monitored by the VFD control. The VFD output changes as a function of the monitored signals and the changes reflect in increased torsional oscillation.
- Some drive manufacturers suggest that the drive and the motor should be supplied by the same manufacturer, so that responsibility for the matching of the motor to the drive is clear. We conclude that, as a minimum, when different manufacturers are involved, one of the manufacturers should be given the responsibility to ensure that the system design is correct.

5. ACKNOWLEDGEMENTS

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7. BIOGRAPHIES

Brian Howes

Brian graduated from the University of Calgary with a Master of Science in Solid Mechanics. His thesis was entitled *Acoustical Pulsations in Reciprocating Compressor Systems*. Brian has worked with Beta Machinery Analysis since 1972. In his present position as Chief Engineer for Beta, he has performed troubleshooting services all over the world. Brian has many technical papers to his credit. The range of machinery problems they cover includes all manner of reciprocating and rotating machinery and piping systems, balancing and alignment of machines, finite element analysis, modeling of pressure pulsation, torsional vibration testing and modeling, flow induced pulsation troubleshooting and design, pulp and paper equipment, such as pulp refiners, etc. Brian has also worked on hundreds of reciprocating compressor installations.

Luis De La Roche

Luis has a Bachelors Degree in Mechanical Engineering from the National University of Colombia. During his 7 years with Beta Machinery Analysis, Luis has conducted design studies, finite element modeling and troubleshooting services in a variety of machines. His experience includes reciprocating, screw and centrifugal compressors, piping systems, motors, pumps, fans and engines.