TORSIONAL VIBRATION PROBLEM WITH MOTOR/ID FAN SYSTEM DUE TO PWM VARIABLE FREQUENCY DRIVE

by

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ABSTRACT

Induced draft (ID) fan systems often use louvers or variable inlet guide vanes to control the flow of exhaust through the fans. An ID fan system can be made more efficient by using a variable frequency drive (VFD) to control the fan speed, which controls the air flow without adding restrictions to the flow path. To reduce energy costs, many companies are using more VFDs.

This paper discusses how a VFD was the source of high torsional vibration in a motor/ID fan system operating a sufficient margin away from the torsional natural frequencies. This type of system instability is not a classical torsional resonance and would be difficult to predict in the design stage. As a result of the high torsional vibration, several couplings were damaged and a motor shaft experienced a fatigue failure before the problem could be clearly identified and solved.

Two different fan systems at the refinery were tested and shown to exhibit similar torsional behavior, although only one of these systems actually failed. Test data showed that the dynamic torque in the couplings was excessive when the original VFD was operated at electrical frequencies above the first torsional natural frequency of the system (21 to 28.5 Hz depending on the fan and coupling arrangement). Within the normal operating speed range, there was continual reversing torque, which is considered unacceptable for centrifugal equipment.

To demonstrate that the VFD was the source of the excitation, the VFD was reconfigured as a soft starter so that it could be bypassed and the fan could then be operated at constant speed using inlet damper control. When operating across-the-line without the VFD, the dynamic torque was significantly reduced (approximately 10 percent of the motor rated torque).

After the test results were reviewed, a new VFD was developed and installed by the manufacturer to prevent the problem from reoccurring within the normal operating speed range. Final measurements are presented that show significant reduction in dynamic torque after the drive modifications were implemented.

INTRODUCTION

Improved technology has resulted in reduced torque modulation produced by a variable frequency drive (VFD) driven motor. Pulse width modulation (PWM) is generally thought to have smooth operation compared to older drive types.

There are several different types of control for PWM drives. The most basic method is Volts/Hertz, which is reported to be acceptable for applications like fans and pumps. However, if not properly tuned, these VFDs can still excite torsional natural frequencies resulting in high torsional vibration and damaged machinery.

Induced draft (ID) fans are commonly used in crude units at refineries. Older control methods utilized dampers that could be opened and closed. To increase energy efficiency, a VFD motor can be used to adjust fan speed instead of throttling flow with dampers. However, any downtime of the crude unit due to problems with the fans can quickly offset the energy savings.

In this case, the end user experienced reliability problems with the ID fan system after installing a VFD for the motor control. One unit referred to as "50 Unit" consists of a 500 hp induction motor driving an ID fan for the furnace exhaust (Figure 1). Fan speeds range up to 1200 rpm during normal operation.

Figure 1. Picture of Motor at 50 Unit.

The spacer in the coupling between the motor and ID fan failed several times, including one catastrophic failure during operation. The original coupling size was verified by the manufacturer and found to have a sufficient service factor of 3 for this application. Although the coupling was adequately sized, the end user decided to install a much larger coupling with a service factor of 15. It was thought that this would prevent further coupling failures.

Because the underlying problem was not understood, increasing the coupling size just eliminated the "fuse" and caused the next weakest link in the system to fail, the motor shaft. As discussed in this paper, it was determined through testing that the fatigue failures were caused by excessive torsional vibration due to an excitation source from the VFD.

DESCRIPTION OF EQUIPMENT

ID Fan—50 Unit

The atmospheric furnace F-50 heats the crude oil entering the refinery (between 60,000 and 108,000 barrels per day). An induced draft fan is needed for the furnace. The fan system consists of an induction motor driving an ID fan through a nonlubricated coupling as listed in Table 1. The motor speed is controlled by a VFD.

Table 1. Equipment for 50 Unit.

Induction Motor	Rated 500 HP at 1195 RPM (26,370 in-lb)
VFD	Variable Frequency Drive Pulse Width Modulation (PWM)
ID Fan	Wheel Diameter 70 inches
Original Coupling	Flexible Disc Coupling Service Factor = 3 Max. Continuous Torque = 84,100 in-lb Peak Overload Torque = 168,200 in-lb
Replacement Coupling	Flexible Disc Coupling Service Factor = 15 Max. Continuous Torque = 411,000 in-lb Peak Overload Torque = 594,000 in-lb

The coupling catalog lists allowable torque values for continuous and peak overload. The sizing procedure from the coupling manufacturer normally recommends a service factor of 1.5 for a motor driven ID fan. The service factor (SF) was 3 for the original coupling. According to the catalog, the coupling was sufficient for this service.

The allowable dynamic torque during continuous operation is not given in the coupling catalog, but can be determined from a modified Goodman diagram. The allowable alternating torque varies depending on the transmitted torque. Goodman plots are constructed with mean torque along the horizontal axis and alternating torque plotted along the vertical axis.

The allowable dynamic torque was 28,000 in-lb zero-to-peak for the original coupling. At the allowable level, the dynamic torque would be fully reversing (exceeding the transmitted torque). For smooth operation of centrifugal equipment the alternating torque should be much lower (normally less than 10 percent of the transmitted torque).

The larger coupling (SF = 15) has a much higher torque rating than the original coupling and was oversized for this application. The allowable dynamic torque is 161,000 in-lb zero-to-peak based on the modified Goodman diagram.

ID Fan—No. 3 Reformer

A second ID fan (located at the No. 3 Reformer) was also tested. This sister system has the same model motor and VFD as the 50 Unit, but has not experienced any torsional failures. However, the coupling and fan are smaller. Based on the fan curves, the maximum fan load is only 370 hp. Therefore, the coupling for the No. 3 Reformer is smaller and has a lower torque rating.

- Flexible disc coupling
- Service factor = 2
- Maximum continuous torque = 40,400 in-lb
- Peak overload torque = 80,800 in-lb

Based on the modified Goodman diagram, the allowable alternating torque would be 9000 in-lb zero-to-peak for a maximum load of 370 hp (transmitted torque = 19,500 in-lb at 1195 rpm). The allowable alternating torque would be higher for reduced fan loads. For example, with a lower mean torque of 10,000 in-lb the allowable alternating torque would be approximately 15,000 in-lb zero-to-peak.

DISCUSSION OF VFDS

PWM Variable Frequency Drive

The motor speed is controlled by an adjustable frequency alternating current (AC) drive. The main components consist of the input rectifier, direct current (DC) link, and output inverter as shown in Figure 2.

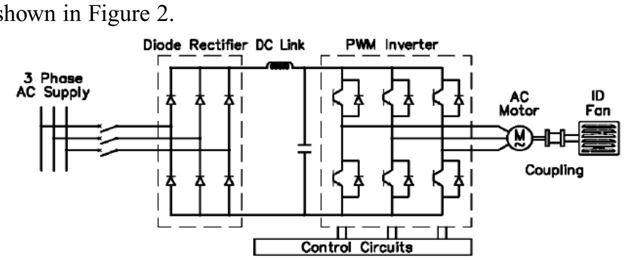

Figure 2. Components of VFD. (Courtesy of Barnes, 2003)

For a pulse width modulation drive, a diode bridge rectifier provides the intermediate DC circuit voltage. This DC voltage is then filtered by an inductor-capacitor (LC) low-pass filter. The output frequency and voltage are controlled by varying the width of the voltage pulses to the motor. This technique requires switching the inverter power devices, insulated gate bipolar transistors (IGBTs), on and off many times in order to generate the proper root mean square (rms) voltage.

Volts/Hertz

Volts per Hertz is the most basic control method and can be used for fan and pump applications. Also referred to as variable voltage variable frequency (VVVF) the proper voltage and frequency are applied to the motor to obtain the reference speed. The drive maintains a constant voltage/frequency ratio.

A block diagram is given for Volts/Hertz in Figure 3. A current limit block monitors the motor current and alters the frequency command if a predetermined value is exceeded. There is a current feedback loop shown in the block diagram between the torque current estimator and the current limit blocks. A signal from the torque current estimator is shown going to the slip estimator, which then adds the slip frequency back into the speed reference.

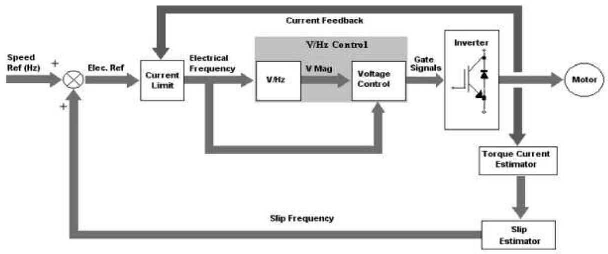

Figure 3. Block Diagram for Volts/Hertz. (Courtesy of Rockwell Automation, 2000)

A "slip compensation" block is used for improved speed control. This is a feedback loop that alters the frequency reference when the load changes to maintain the motor speed. This is necessary for induction motors since the speed will "slip" or decrease slightly with increased load.

Sensorless Vector

Sensorless vector is another speed control mode that was tried during the first field study. Sensorless vector and Volts/Hertz both operate as a frequency control drive with slip compensation keeping the actual motor speed close to the desired speed. If the slip compensation is part of the problem, then neither speed control mode would be expected to function properly. Neither of these two types of speed control (Volts/Hertz and sensorless vector) utilize a speed indicator (encoder) on the motor shaft.

FIRST FIELD TEST OF ID FAN IN 50 UNIT

The original test plan was to measure the "as-found" system with the smaller coupling and then perform another test with the larger coupling installed. However, when the system was shutdown, a large crack was found in the existing coupling spacer. In addition, several nuts were broken on the coupling bolts. Due to the extent of the coupling damage, it was decided not to run the "as-found" test.

As shown in Figure 4, the crack occurred at a 45 degree angle to the axis of rotation. The 45 degree crack is a classical indication that the failure was due to torsional vibration. The coupling was in service for less than a year. This coupling failure was the third failure within a four-year period.

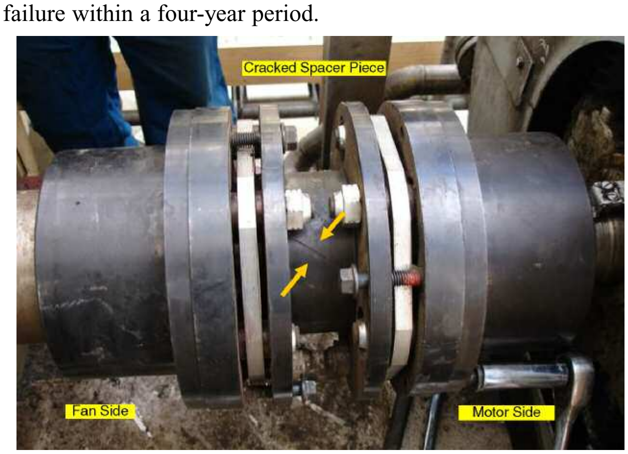

Figure 4. Cracked Coupling Spacer.

To prevent additional coupling failures, it was decided to install a much larger coupling. A comparison of the coupling sizes can be seen in Figure 5.

Measurements were taken on the unit to help diagnose the failures. Multichannel cables were run from the motor location to the VFD/switchgear building where the data acquisition computer was temporarily located. All of the signals were continuously recorded using a digital recorder. Instrumentation included:

- Strain gauges were attached to the fan shaft as shown in Figure 6. A battery powered telemetry system was mounted on the coupling hub. Gauges were oriented to measure shear strain. The output from the telemetry system was converted to static and dynamic torque.
- Flexible AC current probes were used to measure amperage. Probes were placed on electrical phases inside the VFD cabinet. Voltage signals proportional to current (1 mV/A) were recorded.
- An optical tach was used to obtain a once-per-revolution pulse from a piece of reflective tape placed on the fan shaft. The actual rotating speed of the motor/fan was determined from this tach signal.

Figure 5. Comparison of Coupling Sizes.

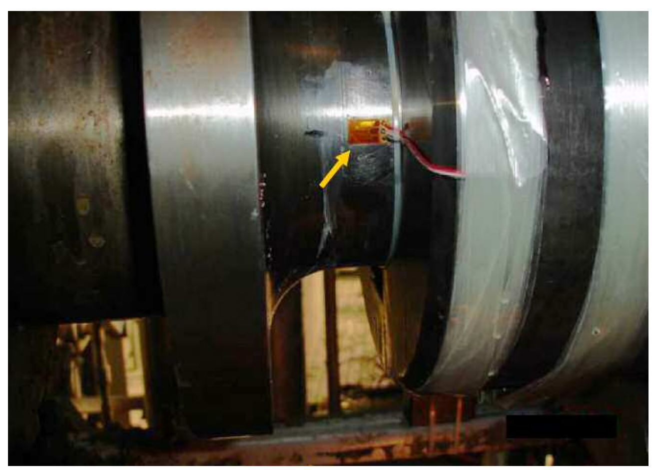

Figure 6. Strain Gauge Telemetry System.

Inlet Dampers Open

The initial tests were performed with the F-50 furnace offline (cold air) and the dampers open. Due to the increased air density, approximately twice the power was required and a maximum speed of only 910 rpm could be achieved. Therefore, the motor was overloaded and tripped during some of these tests.

The motor speed was controlled directly at the VFD panel instead of in the control room. This eliminated the possibility that the plant control system could be contributing to the problem.

Startup

Figure 7 shows a waterfall plot of the torque during startup. This plot is created by stacking multiple frequency spectra at 10 rpm speed increments. Frequency in Hertz is shown on the horizontal axis and motor speed is shown on the vertical axis. Order lines or multiples of running speed appear as diagonal lines.

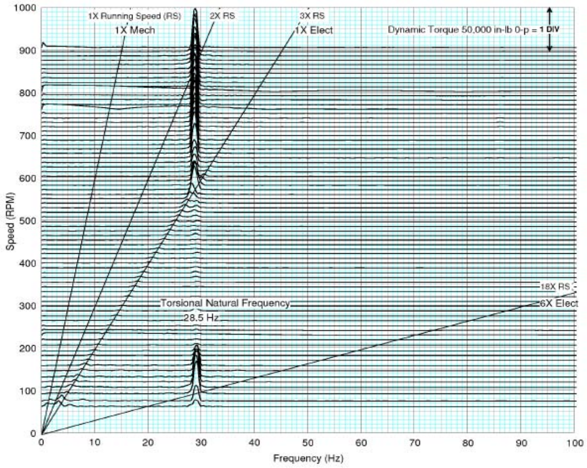

Figure 7. Waterfall Plot of Dynamic Torque During Startup.

For the six-pole motor with three pole pairs the rated speed is slightly less than 1200 rpm (3600 rpm/three pole pairs) due to slip. When running unloaded, induction motors will operate near synchronous speed. However, as load increases the motor speed slips.

The fundamental electrical frequency will be slightly greater than 3× motor running speed due to slip. VFDs typically produce excitation at several different frequencies such as 1×, 6×, 12×, etc., fundamental electrical frequency, which can excite torsional natural frequencies of the system (Wachel, et al., 1996).

The first torsional natural frequency (TNF) was measured at 28.5 Hz (1710 cpm), which is 42 percent above the rated motor speed (1195 rpm or 20 Hz). Normally, this would be considered a sufficient separation margin from the maximum running speed.

The first TNF was excited during startup by the 6× electrical frequency at 95 rpm and then by the fundamental electrical frequency at 570 rpm. When passing through these torsional resonances, the dynamic torque increases and then decreases as normally expected. Continuous operation near speeds that excite these torsional natural frequencies should be avoided by a sufficient separation margin.

Figure 8 shows a trend plot of transmitted and dynamic torque versus time during the startup, steady-state operation, and during the shutdown. The values were averaged over a one-second time window. When the motor is operating at or above 640 rpm (32 Hz fundamental electrical frequency), there is continual excitation of the first TNF at 28.5 Hz.

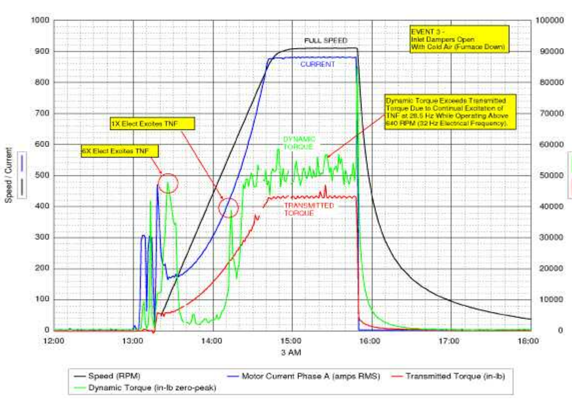

Figure 8. Trend Plot with Inlet Dampers Open.

While operating at constant speed of 910 rpm and load of 43,000 in-lb, the dynamic torque of 50,000 in-lb zero-to-peak exceeds the transmitted torque causing torque reversal in the coupling. Flexing of the disc elements could be seen with a strobe light set to a frequency of 28.5 Hz confirming that the torque was greatly oscillating. In addition, high alternating shear stresses occurred in the motor shaft, which would lead to fatigue failure of the motor shaft.

Shutdown

An increase in dynamic torque occurred just as the drive tripped. Therefore, it was decided to plot the data versus time. Figure 9 shows that the increase in dynamic torque was caused by the sudden change in transmitted or average torque as the motor was de-energized. Several other spikes appear in this plot, but are not real and were attributed to radio use near the instrumentation setup area.

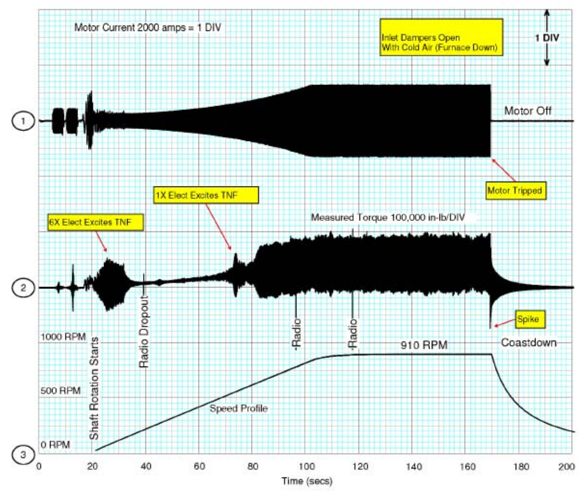

Figure 9. Time Traces.

The frequency of torque oscillation occurs at the first TNF (28.5 Hz). The torque oscillation decreases in amplitude after the motor is tripped. The amplitude decay was used to estimate the torsional damping in the system.

From the first two cycles after shutdown, the logarithmic decrement was computed to be 0.0868, which equates to an amplification factor (AF) or Q of 36. From the next two cycles the damping appeared to be less and was approximated as a Q of 42. The next two cycles have a damping value of Q = 50.

Torsional systems are lightly damped unless a rubber coupling, viscous damper, etc., is used. For fan systems with steel couplings the damping is typically Q = 30 to 50, which agrees with the measured values. At resonance the torsional excitation can be amplified by a factor of 50 times.

Adjustments to Drive Settings

It was felt that the excitation source was the VFD so possible modifications to the PWM drive parameters were discussed. A representative from the drive manufacturer assisted with subsequent testing.

The PWM drive can operate in two modes: volts per frequency (fixed boost) or sensorless vector. The motor does not have an encoder to relate actual shaft speed back to the drive. According to the VFD representative, an encoder is not normally used with fan systems. Therefore, the motor speed is estimated based on the drive electrical frequency and expected slip at certain loads. In addition, the switching frequency of the drive was set to 2000 Hz. The drive is capable of higher switching frequencies, but was not increased due to concern of inverter heating.

During subsequent tests several other drive parameters were varied one at a time. The following adjustments were made to the drive parameters to determine the effect on the torsional vibration:

- Bus regulation = enabled/disabled
- Acceleration time = 120 sec to 600 sec
- Deceleration time = 600 sec to 800 sec
- Stability gain = 0/1
- Operating mode = fixed boost or sensorless vector
- PWM comp time = 30 sec to 80 sec
- Changed controller board to update program
- Brake frequency = 0 to 17 Hz
- Skip frequency = 29 Hz with band = ±3 Hz
- Current limit lowered to 300 amps
- Flying start reverse = 0 to 60

Changes to these parameters did not prevent the first TNF from being excited when operating above 32 Hz electrical drive frequency. Dynamic torque remained excessive when operating the motor/ID fan above 640 rpm.

Inlet Dampers Closed

Running the ID fan with cold air and inlet dampers open resulted in twice the normally required power. Therefore, it was decided to perform additional testing with the inlet dampers closed so that the maximum speed of 1197 rpm could be reached.

To confirm that the strain gauge telemetry system was calibrated and functioning properly, the average or transmitted torque indicated from the telemetry system was compared to the measured motor current (amps). Neglecting motor efficiency, the ratio of transmitted torque to rated torque and the ratio of motor current to rated current were similar confirming that the measurements were correct.

As with the previous tests (inlet dampers open), the dynamic torque dramatically increased once the motor was operating above 640 rpm or (32 Hz electrical drive frequency). The dynamic torque appeared to be several times higher with the inlet dampers closed than with the dampers open. Note that the fan load was reduced with the inlet dampers closed. Therefore, the drive appeared to generate higher excitation components at lower load.

Figure 10 shows the dynamic torque increasing significantly during startup as the motor operates above 640 rpm. Note that transmitted torque and dynamic torque are plotted on the same scale shown on the right-hand side of the trend plot in Figure 10.

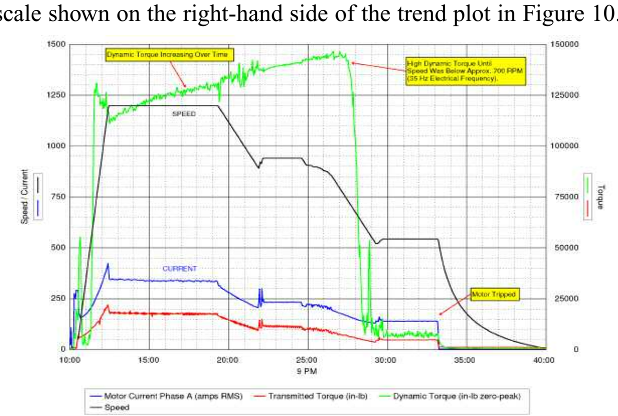

Figure 10. Trend Plot with Inlet Dampers Closed.

Figure 11 shows the torque and current signals plotted over a one-second time interval while the fan was operating at 1197 rpm. During this time, fluctuations in motor current of 20 percent were observed. This appears abnormal since the fan was operating at a constant speed and load while the data were acquired. The modulation in the 60 Hz electrical current is another indication that the VFD is the source of the torsional excitation.

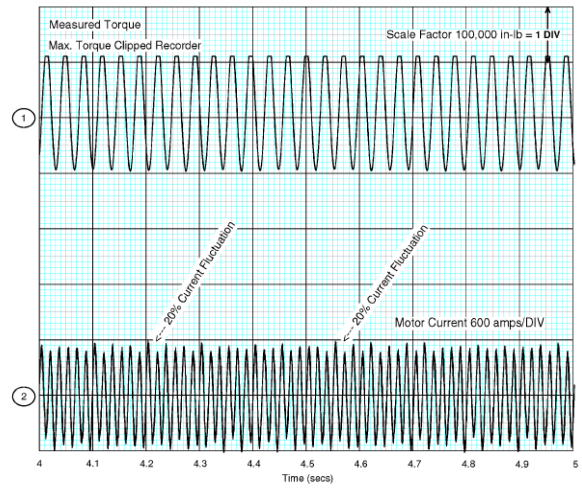

Figure 11. Time Traces with the Fan Operating at 1197 RPM.

Figure 12 shows how the dynamic torque would repeatedly jump at speeds above 640 rpm as the motor speed was increased and decreased using the variable speed drive. As demonstrated, the phenomenon was repeatable. Note that the peak in dynamic torque at 570 rpm is due to the fundamental electrical frequency exciting the first TNF and is different from the continual excitation of the first TNF when operating at or above 640 rpm.

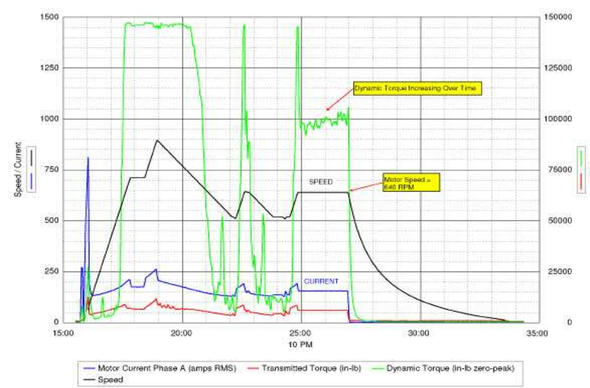

Figure 12. Trend Plot as Fan Speed Varied with Dampers Closed.

Next, the motor was started and taken directly to the maximum speed of 1197 rpm. As shown in Figure 13, the motor operated at this maximum speed for approximately three minutes. The drive was then tripped, and the motor allowed to coastdown unpowered. Note that the dynamic torque decreased immediately when the motor was turned off although the speed remained above 640 rpm for 30 seconds. These plots indicate that the drive was the excitation source because the dynamic torque immediately decreased after the motor was tripped.

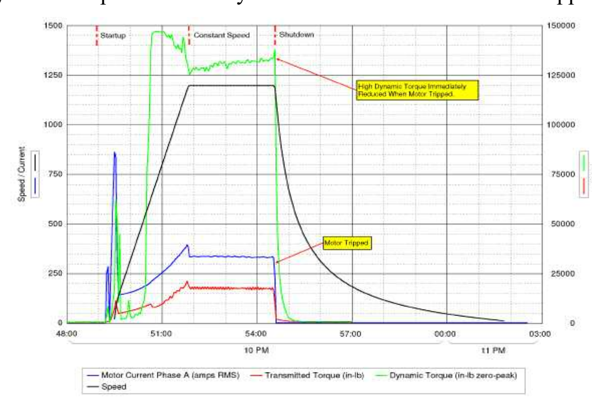

Figure 13. Trend Plot at Maximum Speed of 1197 RPM.

Figure 14 shows the measured current for motor phase A at the VFD. Note that when the dynamic torque is high (after approximately 100 seconds), side bands appear on either side of the fundamental electrical frequency. The predominant frequency for the dynamic torque occurs at the first TNF of 28.5 Hz. The side bands in the motor current are spaced ±28.5 Hz (TNF) from the fundamental electrical frequency.

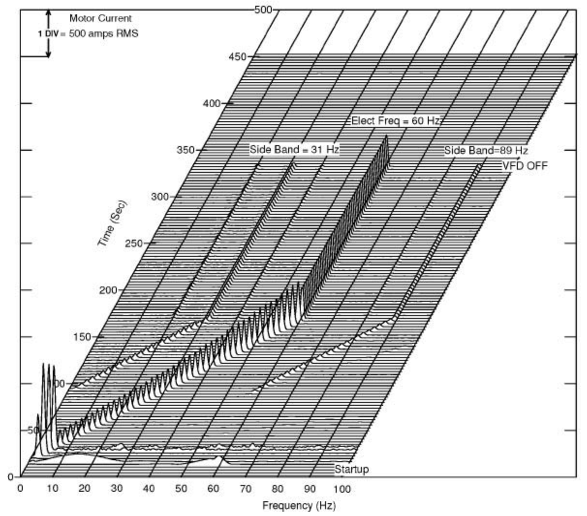

Figure 14. Time Waterfall Plot of Motor Amps.

Furnace in Operation

A final torque measurement was taken with the furnace in service. The fan speed was 680 rpm as dictated by the process conditions. Figure 15 shows the frequency spectrum of the torque signal. The dynamic torque was still occurring at the first TNF (28.5 Hz) with an amplitude of 180,000 in-lb zero-to-peak. The amount of dynamic torque is several times higher than the transmitted torque and was considered to be excessive.

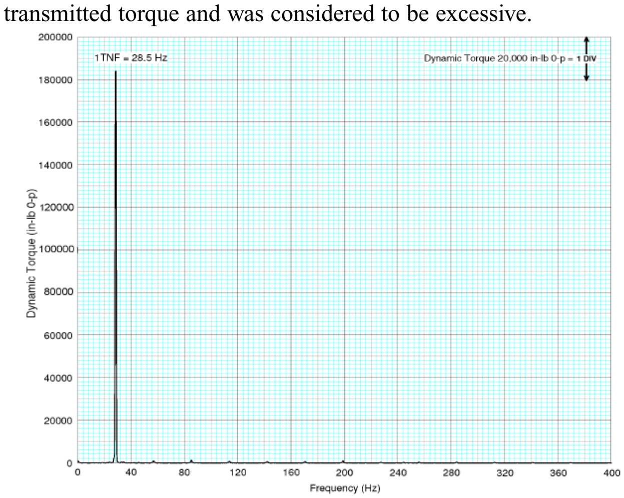

Figure 15. Frequency Spectrum of Dynamic Torque During Normal Operation with Furnace in Service.

The unintensified shear stress in the motor shaft (diameter = 3.54 inch) due to 180,000 in-lb of torque would be 20,700 psi. This alternating stress level exceeds the shear endurance limit of the shaft material. Therefore, fatigue cracks are expected to form in areas with stress risers, such as at the base of the keyway.

The flow rate through the furnace was at the minimum rate of 60,000 barrels per day. The unit is capable of 108,000 barrel per day. Since the furnace was in service, hot exhaust was passing through the ID fan. Note that the fan speed could not be varied now that the unit was back in operation.

The VFD panel was reading an average value of 240 amps rms. However, the numbers on the display were varying ±15 amps indicating significant current fluctuation. Based on the ratio of the electrical current to the motor rating (240 amps/549 amps) the load was estimated to be approximately 44 percent.

MOTOR SHAFT FAILURE

Less than a month after installing the larger coupling, it was discovered that the motor shaft failed as shown in Figure 16. The crack occurred at a 45 degree angle to the motor shaft axis indicating high torsional vibration. Note that previous failures occurred in the coupling and this was the first failure of the motor shaft.

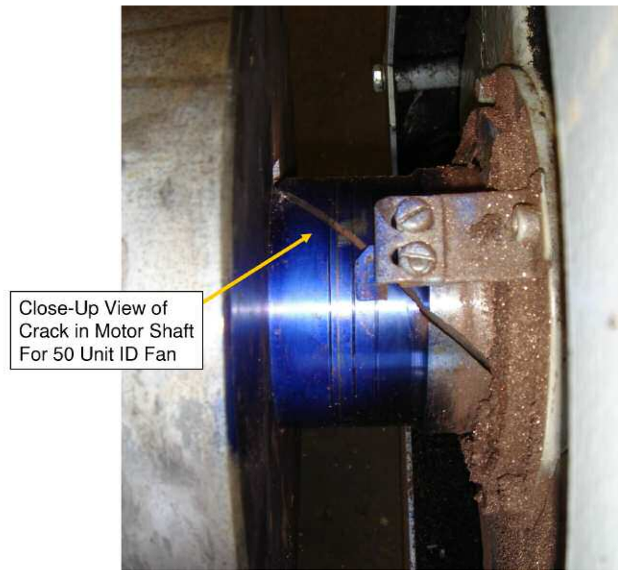

Figure 16. Crack in Motor Shaft.

Data acquired before the motor shaft cracked showed high torsional vibration at 28.5 Hz (system first TNF) while operating off resonance with the furnace in operation. The measured alternating torque was 180,000 in-lb zero-to-peak. The mean torque due to fan load was approximately 11,500 in-lb or 44 percent of the motor rated torque. Therefore, the dynamic torque was approximately 15 times higher than the transmitted torque.

This measured torque level even exceeded the allowable limit for the larger flexible disc coupling. The torsional vibration problem had not been solved by changing the coupling. Now the next weakest link in the system was the motor shaft.

Based on the initial test results, the VFD appeared to be the excitation source. To confirm this, a second field study was performed by retesting the ID fan at the 50 Unit, and also testing a similar ID fan at the No. 3 Reformer.

SECOND FIELD TEST OF ID FAN IN 50 UNIT

For the second field test, several different drive settings were evaluated in an effort to resolve the problem. When the problem persisted, the drive was reprogrammed to function only as a soft-starter so that the motor could be tested independent of the VFD. During the first field study, the system could not be tested without the drive (across the line start) because of the high breakaway torque needed to initially roll the ID fan. Bypassing the VFD and operating the motor across-the-line at constant speed helped to confirm that the VFD was the excitation source.

A new coupling that was the original size was installed along with a replacement motor after the motor shaft failure. The torsional natural frequency of the system changed from 28.5 Hz to 24 Hz because of the smaller coupling with reduced torsional stiffness. Three runs were recorded with the furnace down and are discussed as follows.

Run 1

The motor was taken up to the maximum speed for a short period of time. Figure 17 shows a waterfall plot of the dynamic torque taken during startup. The first torsional natural frequency of the system is 24 Hz or 1440 cpm with the smaller coupling, which still provides a sufficient separation margin of 20 percent above the motor speed of 1200 rpm.

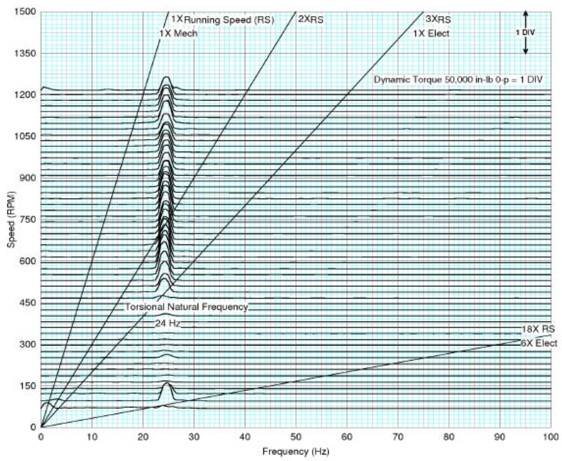

Figure 17. Waterfall Plot of Dynamic Torque During Startup.

As indicated by the peaks in Figure 17, the first TNF of 24 Hz was excited by the 6× electrical frequency at 80 rpm, and the 1× electrical frequency at 480 rpm. To maintain a minimum 10 percent separation margin from the torsional critical speed at 480 rpm, the minimum operating speed of the fan should not be less than 530 rpm.

As shown in Figure 18, the motor speed was held at 1217 rpm for less than 30 seconds and then the motor was tripped. Dynamic torque increased once the motor was operating above approximately 500 rpm. Dynamic torque reached 40,000 in-lb zero-to-peak.

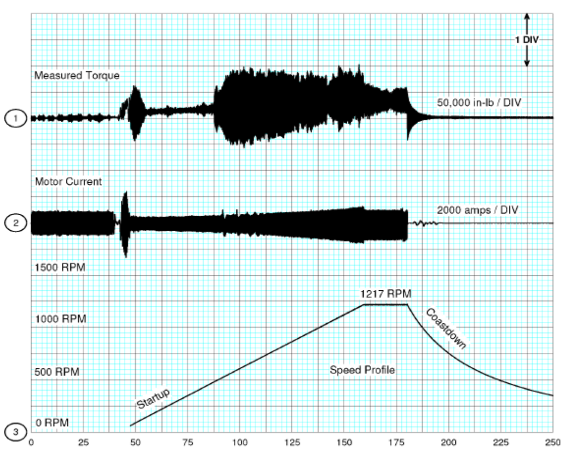

Figure 18. Time Traces and Speed Profile Taken During Startup.

Run 2

For the second test run, the carrier frequency of the PWM was increased from 2000 Hz to 3000 Hz. The inlet dampers were closed, and the motor speed increased to 1217 rpm. Changing the carrier frequency to 3000 Hz, did not solve the instability problem above 500 rpm. It was noticed that the drive excitation was higher during startup at the 1× and 2× electrical frequencies.

Run 3

For Run 3, the carrier frequency was set back to 2000 Hz. The PWM comp time was decreased from 70 seconds to 60 seconds. The startup appeared similar to Run 1.

Switch from VFD to Across-the-Line Operation

During Run 3, the motor was switched from VFD to across-the-line operation while the motor was rotating. The motor was accelerated to 1250 rpm. The drive was then turned off for 2 to 3 seconds, before the current was reapplied to the motor. Then the motor was operating across-the-line.

Figure 19 shows that during the switch the speed dropped to 1150 rpm before the current was reapplied. Because of this low speed, the current spiked to 1500 amps RMS. For a smoother transition, it would be better to have the motor speed closer to 1200 rpm before reapplying the current. To accomplish this, the maximum speed was increased to 1270 rpm so that the motor drops to the correct speed during the transition.

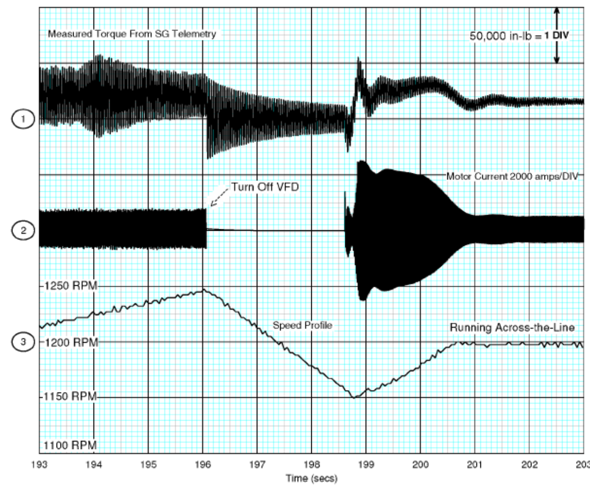

Figure 19. Switch to Across-the-Line Operation.

Reduction in Dynamic Torque

The dynamic torque was significantly reduced from 50,000 in-lb zero-to-peak to only 2,500 in-lb zero-to-peak by switching from VFD to across-the-line operation for the motor. For reference, the measured dynamic torque with the VFD in operation was approximately double the rated coupling torque of 26,000 in-lb.

The data were plotted so that time wave forms of torque and current could be viewed just before turning off the VFD. Figure 20 shows that the dynamic torque was high and that there was fluctuation in all three current signals measured at the VFD cabinet.

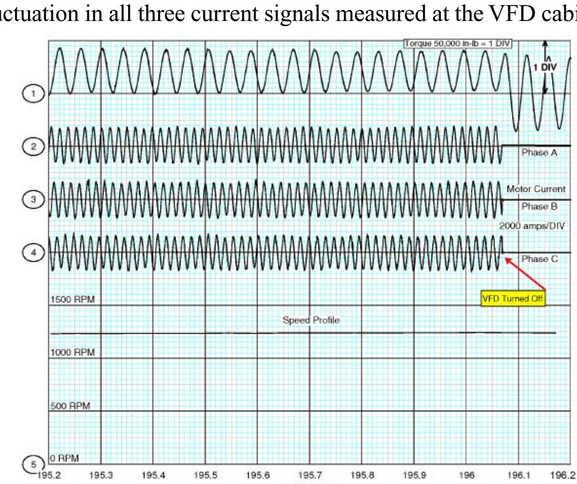

Figure 20. Time Wave Forms as VFD was Switched Off.

With the motor running across-the-line the dynamic torque was significantly reduced to approximately 10 percent of the motor rated torque. When plotted on the same scale as before, the dynamic torque is barely discernible (Figure 21).

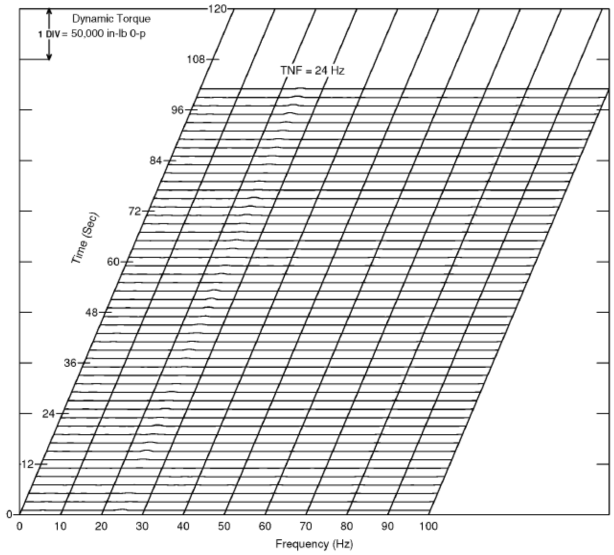

Figure 21. Time Waterfall Plot of Dynamic Torque with Motor Across the Line.

When the VFD was only used as a soft-starter, the dynamic torque was still excessive during startup so this did not represent a good long-term solution. Soft-start refers to the reduced electrical current needed to start the motor.

The short-term solution was to operate the ID fan across-the-line with inlet damper control and try to limit the number of starts. This condition was less energy efficient, but more reliable from a mechanical standpoint.

FIELD TEST OF ID FAN IN NO. 3 REFORMER

There was some discussion as to whether problems with the 50 Unit were an isolated occurrence. Therefore, a second ID fan system with the same model motor and VFD as the 50 Unit was tested. However, the fan and flexible disc coupling were smaller. Testing was performed with the inlet dampers mainly closed. The furnace was down, so air passing through the fan was at ambient temperature (60°F).

In summary, this system exhibited similar torsional behavior to the 50 Unit. The dynamic torque in the coupling was very high when the drive was operating at frequencies above the first torsional natural frequency of the system.

It was noticed that the dynamic torque for the No. 3 Reformer fan system peaked around 600 to 700 rpm. Fortunately, this fan normally runs between 800 and 1000 rpm. The four test runs conducted at the No. 3 Reformer are described as follows.

Run 1

Based on a waterfall plot taken during the coastdown, the first torsional natural frequency of the system was 21 Hz (1260 cpm). The separation margin from the first TNF was only 5 percent above the motor speed of 1200 rpm. Normally, a separation margin of at least 10 percent is required by American Petroleum Institute (API) specifications.

In most cases, a torsionally stiffer coupling can increase the first TNF and provide an acceptable separation margin from the maximum motor speed. The final coupling selection should be verified with a torsional analysis.

The trend plot for Run 1 is shown in Figure 22. The ramp rate during startup was 10 rpm/sec or 0.5 Hz electrical frequency per second. There were two instances during startup where the dynamic torque in the motor shaft reached at least 50,000 in-lb zero-to-peak. The fan system was operated at 1198 rpm for almost two minutes, before tripping the motor. The speed profile during coastdown is also shown in Figure 23.

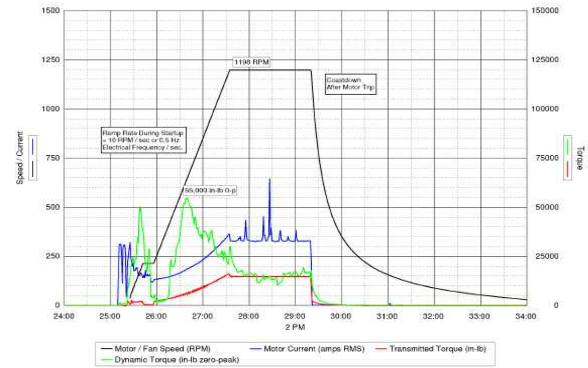

Figure 22. Trend Plot from Test Run 1.

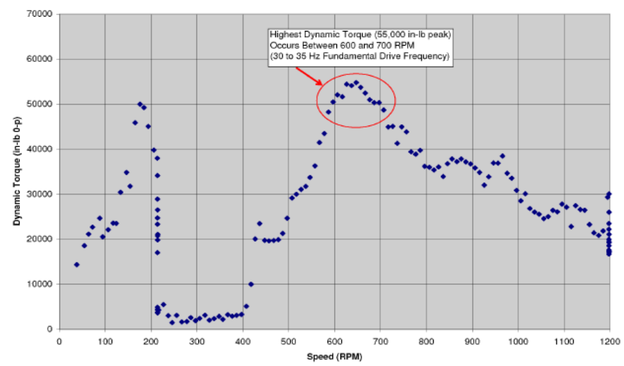

Figure 23. Plot of Dynamic Torque Versus Speed.

The trend data were replotted in Figure 23 to show dynamic torque versus motor speed. The highest dynamic torque was 55,000 in-lb zero-to-peak and occurred between 600 and 700 rpm. This corresponds to an electrical frequency range of 30 to 35 Hz. What is interesting about Figure 23 is that the amplitude of dynamic torque was reduced at higher speeds.

During Run 1, the dynamic torque was 100 percent of the transmitted or more. To have continual reversing torque during steady-state operation of a rotating equipment train is considered highly unusual and unacceptable especially for centrifugal fans. The dynamic torque levels should be reduced for long-term reliability. The data indicated that operating at speeds from 600 to 700 rpm would likely fail the coupling.

Figure 24 shows the time wave form of the torque and current signals along with the speed profile during part of the startup. From 34 to 49 seconds, the motor speed remained a constant 214 rpm and it was assumed that the fan load would be constant as well. However, there was considerable torque variation during this time period and the current signal appeared erratic as noted on Figure 24.

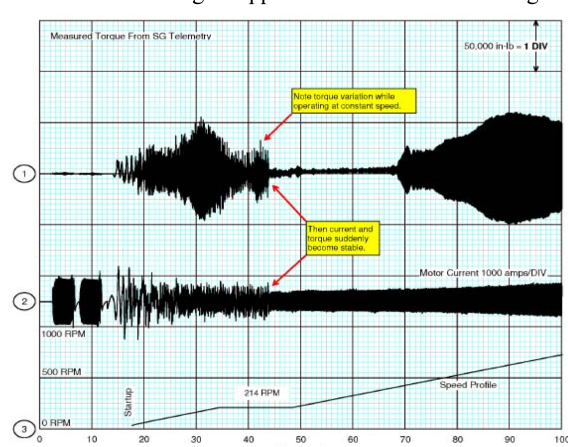

Figure 24. Time Traces from Test Run 1.

At 44 seconds, the dynamic torque suddenly reduced in amplitude and was minimal although the operating conditions had not changed. At this same instant in time, the current signal suddenly became stable, which was a very interesting event.

It turns out that 214 rpm was close to where the 2× electrical frequency from the VFD would excite the first TNF of the motor/fan system. The drive appeared to be reacting to the torsional resonance, but then after a period of time was able to achieve stable operation as shown in Figure 25.

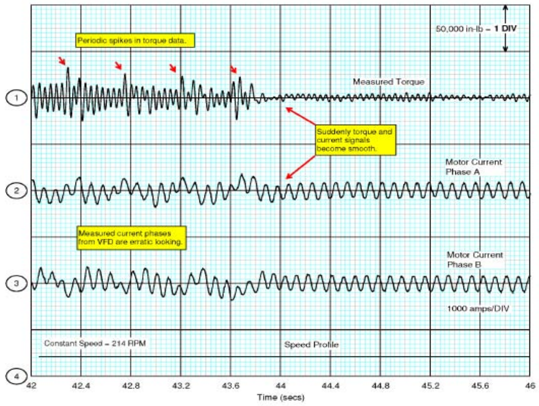

Figure 25. Measured Torque and Motor Current Versus Time.

The motor was increased to maximum operating speed of 1198 rpm. Figure 26 shows the torque before and after the motor was tripped. Considerable torque modulation can be seen during constant speed operation before the unpowered coastdown.

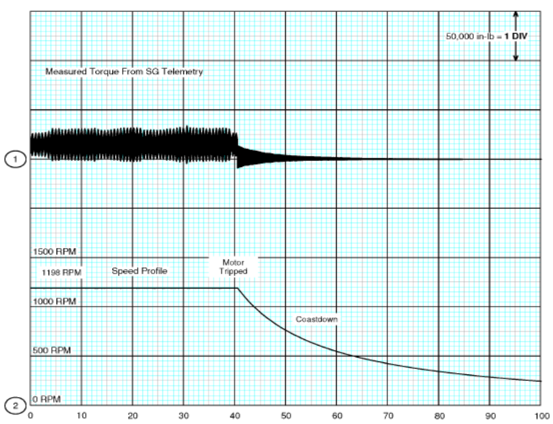

Figure 26. Trip Event and Coastdown at End of Test Run 1.

Run 2

A second run was performed to check for repeatability. The 1×, 2×, and 6× electrical harmonics excited the first TNF during startup. Even when the fundamental electrical frequency of the VFD was above 21 Hz, the first TNF continued to be excited (ring). This was the same torsional behavior as observed in the 50 Unit.

The fan was then increased to the maximum speed of 1198 rpm. As in Run 1, there was high dynamic torque of 50,000 in-lb zero-to-peak near 200 rpm and 55,000 in-lb zero-to-peak when operating between 600 and 700 rpm. Therefore, the test results were repeatable.

Run 3

After startup, the motor speed was held at several speeds to determine the dynamic torque without any acceleration effects. Table 2 summarizes the measured levels. The results presented in Table 2 are slightly different than previous data. The highest dynamic torque during steady-state operation occurred at 721 rpm, which was outside the range of 600 to 700 rpm mentioned previously.

Table 2. Dynamic Torque in ID Fan System for No. 3 Reformer.

Operating Speed	Dynamic Torque
618 RPM	30,000 in-lb 0-p
721 RPM	50,000 in-lb 0-p
817 RPM	38,000 in-lb 0-p
1010 RPM	25,000 in-lb 0-p

Figure 27 shows that the dynamic torque jumped when operating at 721 rpm and was discontinuous compared with the surrounding speeds of 618 rpm and 817 rpm. As shown, the transmitted torque remained at 10,500 in-lb or below for all of the speeds tested. The dynamic torque amplitudes were more than double the transmitted torque, which is considered to be excessive.

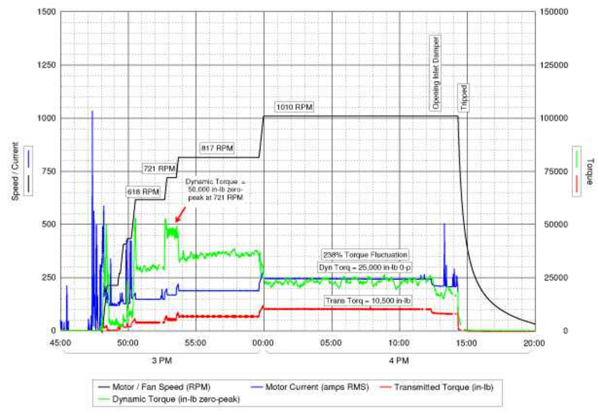

Figure 27. Trend Plot.

Run 4

The purpose of this final test (Run 4) was to measure the dynamic torque at the motor speed of 1133 rpm. This test point was chosen because previous data showed a reduction in dynamic torque when operating at 1133 rpm. Also, this is the maximum operating speed in order to maintain a 10 percent separation margin from the first TNF at 21 Hz.

The motor speed was increased directly to 95 percent speed (1133 rpm). While operating at 1133 rpm, the dynamic torque remained at 20,000 in-lb zero-to-peak or below (Figure 28). Based on the test data with cold air, this would appear to be the preferred operating speed to minimize the dynamic torque until the VFD could be corrected. Additional data were not acquired with the furnace online and hot exhaust flowing through the ID fan.

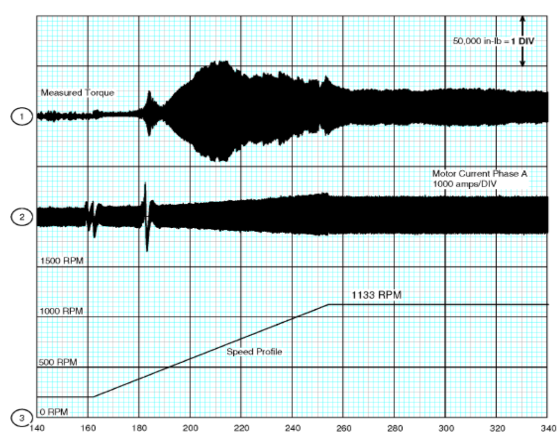

Figure 28. Time Traces from Test Run 4.

In summary, these four test runs showed that the motor/fan system in the No. 3 Reformer exhibited similar torsional behavior to the motor/fan system in the 50 Unit. The VFDs were the same model so the results were not unexpected. The only reason the motor shaft and coupling in the No. 3 Reformer had not failed was because damaging torsional vibration was not produced at the normal operating speed. However, the amount of dynamic torque was still unusually high and operating the motor/fan at slightly different speeds would have likely failed the system. Therefore, it was recommended that the VFDs be replaced in both the 50 Unit and the No. 3 Reformer.

SIMPLE TORSIONAL MODEL

The VFD manufacturer needed to include the torsional model of the mechanical system in their electrical simulations. Therefore, a simple torsional model for the motor/ID fan system was provided. Only the first torsional mode was of concern so a simple lumped model was created with two inertias (one representing the motor and the other for the fan). An equivalent torsional spring was calculated to model the motor shaft, coupling, and fan shaft.

The first torsional natural frequency of the motor/fan system at the 50 Unit was 24 Hz (1440 cpm) with the smaller size coupling. With the larger coupling, the first torsional natural frequency was increased to 28.5 Hz (1710 cpm). The primary difference between the two cases is the torsional stiffness of the couplings. The difference in inertia between the two coupling sizes was very minor in comparison to the motor and ID fan inertia values.

Equation (1) is used to calculate the torsional natural frequency of the idealized two inertia system. The calculated frequency ω_n will be in rad/sec. To convert from rad/sec to Hz, the frequency must be divided by 2π (150.8 rad/sec = 24 Hz).

$$\omega_n = \sqrt{K_t\left(\frac{J_m + J_f}{J_m J_f}\right)} \tag{1}$$

The mass-elastic values in Table 3 can be used to calculate the first TNF of the mechanical system.

Table 3. Mass-Elastic Values.

J_m	Motor Inertia	438 in-lb-sec^2
J_f	Fan Inertia	2538 in-lb-sec^2
K_t	Eqv. Tors. Stiffness w/ Small Coupling	8.5 million in-lb/rad
K_t	Eqv. Tors. Stiffness w/ Large Coupling	12.0 million in-lb/rad

Torsional systems without viscous dampers or rubber couplings are lightly damped. The first author's company determined a Q of approximately 36 to 50 for the first torsional mode. To be conservative, the minimum damping of Q = 50, which is equivalent to a damping ratio of 0.01 or 1 percent was used for the torsional model.

For example, a damping value of 1127 in-lb-sec should be used in the model between the motor and fan inertias for the system with the small coupling to achieve the proper Q of 50 (damping ratio of 0.01). For the larger coupling, the equivalent damping value is 1340 in-lb-sec.

A simplified torsional analysis was performed using the two inertia model to obtain the first torsional mode shape. The ID fan has a large inertia and is located near a node (point of minimal torsional oscillation) for the first torsional mode. The motor is located at the antinode, which is the most sensitive location (point of highest torsional oscillation). Therefore, any torque variation at the motor could easily excite this mode.

DRIVE SIMULATIONS AND TESTING

All available data were supplied to the drive manufacturer. They performed simulations and additional testing at the factory. Discussion of the electrical model is beyond the scope of this paper. The VFD manufacturer has presented a paper, which discusses the electrical issues in greater detail (Kerkman, et al., 2008).

The following is a brief summary of the electrical simulations and testing. Four possible sources of disturbance were considered.

- Induction motor instabilities
- VFD induced dynamic torque
- Inverter dead time
- PWM (discrete modulation, bus voltage feedback, and carrier comparison)

Instability of the induction motor was ruled out for frequencies above 7 Hz. Next the VFD manufacturer examined fundamental components of the drive:

- Modulator design
- Sampling process
- Feedback filtering
- DC bus component design
- Cabling

The drive manufacturer used analysis and simulation software to evaluate the existing electrical system combined with the simple torsional model. Results confirmed that once beyond 30 Hz operating frequency the first TNF would be continually excited because of frequency smearing (Figure 29). The bus voltage is sensed and filtered. Any distortion will be amplified by the triangle comparison pulse generator.

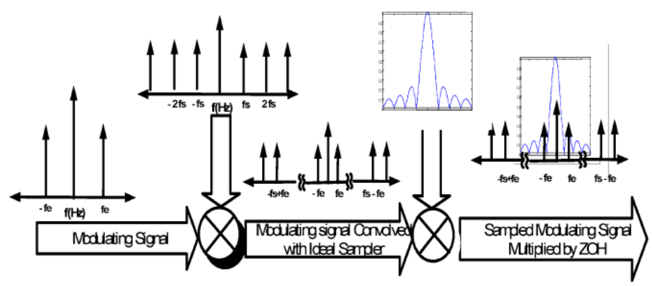

Figure 29. PWM Model Showing Spectrum Smearing. (Courtesy of Kerkman, et al., 2008)

The higher load stresses the DC link components, saturating the DC choke, causing distortion. This distortion can be a source of torsional excitation. The source was determined to be the combination of installation/component/feedback/modulator. High torsional vibration was induced by excitation components generated at the inverter output.

By increasing to "twice per carrier" updates of the PWM registers, the analysis by the drive manufacturer showed a significant improvement. Table 4 shows major differences in the drive control and hardware. Based on the analysis results, cabling modifications were not recommended.

Table 4. Drive Comparison.

Item	Original Drive	New Drive
Duty cycle update	1/carrier cycle	2/carrier cycle
Dead time comp	1/carrier cycle	2/carrier cycle
Bus voltage feedback filtering	Non-linear filter	Moving average
DC link choke	saturating	Non-saturating

FINAL FIELD TEST

Based on the results of the simulations, the VFD manufacturer designed a new drive. Several test runs were performed with the new drive installed at the refinery. During these tests, the furnace at the 50 Unit was not in operation. Therefore, to simulate full load the inlet dampers were gradually opened while operating the ID fan at full speed and with cold air. The full load torque of 26,000 in-lb was reached when the dampers were 10 percent open.

The first torsional natural frequency of the system was 24 Hz and was excited by multiple electrical harmonics during startup as shown in Figure 30. The dynamic torque exceeded the transmitted torque during startup as shown in Figure 31.

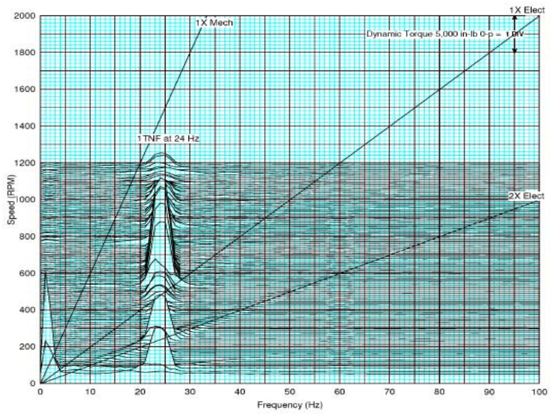

Figure 30. Waterfall Plot of Dynamic Torque with New VFD.

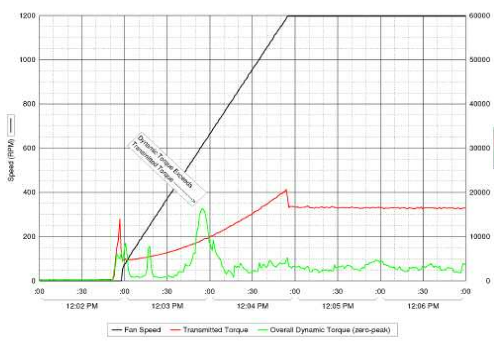

Figure 31. Trend Plot Taken During Startup and Operation with New VFD.

At motor speeds of 560 rpm and 600 rpm (28 Hz and 30 Hz) the dynamic torque greatly exceeded the transmitted torque, which indicated unstable drive operation. In order to have a sufficient separation margin from the operating speeds that create high dynamic torque, it was recommended that the continuous operating speed range be limited to 700 rpm to 1200 rpm (35 Hz to 60 Hz electrical frequencies). Figure 32 shows another test run where the motor was held at various speeds so that the dynamic torque could be evaluated under constant operating conditions.

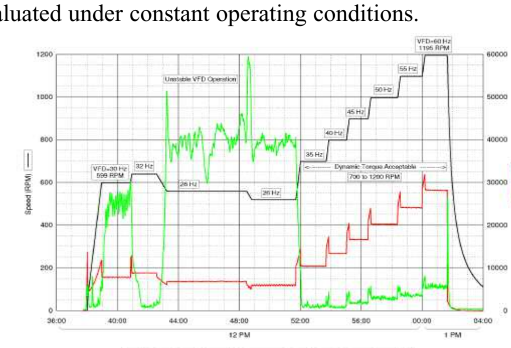

Figure 32. Trend Plot from Multiple Speed Test with New VFD.

From 700 to 1200 rpm, the dynamic torque was approximately 12 to 18 percent of the transmitted torque and was considered acceptable. Recall that the previous VFD model produced dynamic torque as high as 180,000 in-lb zero-to-peak or 682 percent of full-load torque. The new VFD had much smoother operating in this speed range compared to the older model drive. Figures 33 and 34 show the torsional response at various operating speeds (electrical drive frequencies).

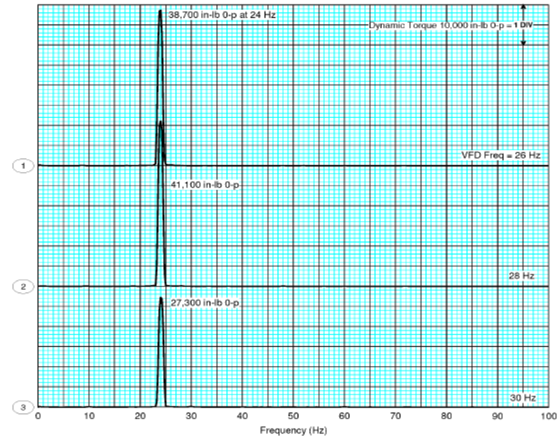

Figure 33. Frequency Spectra of Dynamic Torque with New VFD.

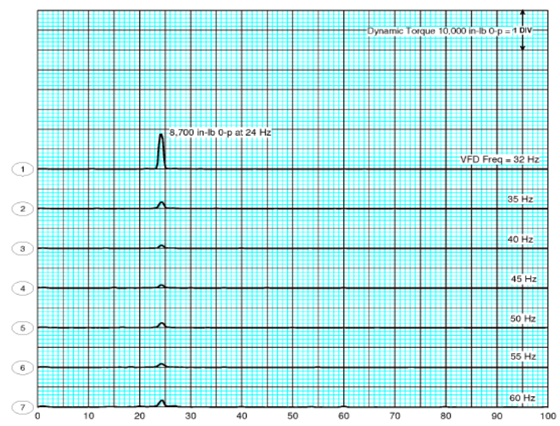

Figure 34. Frequency Spectra of Dynamic Torque with New VFD (Continued).

Testing was concluded after one hour because catalyst was scheduled to be added to the unit. With additional testing, it may have been possible to tune additional parameters to improve the drive performance at the lower speeds.

In summary, the new VFD was much smoother than the older version while operating within the normal speed range required for the unit. However, high dynamic torque still occurred at lower operating speeds. Therefore, the minimum speed of the motor/ID fan system had to be restricted in order to avoid another torsional vibration problem.

CONCLUSIONS

To increase energy savings, a VFD motor can be used to adjust fan speed instead of throttling air flow with dampers. However, any downtime of the fan system and the crude unit can quickly overshadow the amount of potential energy savings from using the VFD. Therefore, it is important that the system be safe and reliable.

The end user experienced reliability problems with the ID fan system after installing a VFD for the motor speed control. Coupling failures were not initially attributed to the torsional vibration problem. Measurements later showed that excessive dynamic torque was the root cause of the multiple coupling failures.

Through testing, it was determined that the VFD was the source of the excitation. Even when operating above the first TNF with a sufficient separation margin, the first TNF was still being continually excited. This type of phenomenon was not a classical torsional resonance and would be difficult to predict in the design stage.

Due to the light damping in the torsional mechanical system, other VFDs could have a similar problem. In fact, the sister unit exhibited similar behavior, but had not failed because it normally operated at a speed that did not produce damaging torsional vibration. However, the amount of dynamic torque in the No. 3 Reformer was still unusually high and operating the motor/fan at slightly lower speeds would have likely failed the system.

Changing to a coupling with rubber blocks would have added damping to the torsional system. However, there was concern that if the drive still exhibited unstable behavior, the rubber blocks could be damaged and have reduced life. Therefore, it was decided to pursue only modifications to the VFD. Additional modeling and testing by the drive manufacturer resulting in a new VFD design.

The new replacement VFD still caused high dynamic torque in the system while operating at speeds of 520 rpm to 600 rpm (26 Hz to 30 Hz electrical frequencies). However, the new VFD proved to be acceptable from 700 rpm to 1200 rpm (35 Hz to 60 Hz electrical frequency), which covered the normal speed range for the motor/ID fan system at the 50 Unit.

RECOMMENDATIONS

To avoid torsional problems, it is recommended that a steady-state torsional analysis be performed. Such an analysis should include:

- Calculating torsional natural frequencies and mode shapes.
- Plotting a Campbell or interference diagram.
- Providing forced response calculations for comparison to allowable torque and stress limits.
- The VFD manufacturer should provide torque harmonics produced by the drive over the entire operating range.

For critical applications, field testing may be needed to ensure successful operation. If any changes are later made to equipment (motor, coupling, fan, or VFD) the analysis should be updated.

SUMMARY

As shown in this paper, the excessive torsional vibration and resulting coupling and motor shaft failures were caused by an interaction between the PWM inverter and the first torsional mode which was lightly damped. This required mechanical and electrical engineers to work together to solve the system problem. Care must be exercised before applying a VFD to a motor/ID fan system. If not, downtime of the crude unit at a refinery due to problems with the motor/ID fan system could offset the potential energy savings of using a VFD instead of damper control.

NOMENCLATURE

cpm	= Cycles per minute
K_t	= Torsional stiffness, million in-lb/rad
Hz	= Hertz, cycles per second
J_m	= Inertia of motor, in-lb-sec^2
J_f	= Inertia of fan, in-lb-sec^2
N	= Speed, rpm
p-p	= Peak-to-peak
0-p	= Zero-to-peak amplitude
rpm	= Revolutions per minute
TNF	= Torsional natural frequency
ω_n	= Natural frequency, rad/sec

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