COUPLING FAILURES IN VFD MOTOR / FAN SYSTEMS DUE TO TORSIONAL VIBRATION

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Abstract

Variable frequency drives (VFDs) are used to control motors over a wide operating speed range. VFDs can create torsional excitation that in some instances cause torsional fatigue failures of couplings and shafts [1,2]. This case study deals with repeated failures of disc pack couplings on three newly commissioned combustion air blowers (fans).

Field tests indicated that at certain speeds, the dynamic torque was significantly amplified by excitation of the first torsional natural frequency (TNF). The alternating torque reached 800% of full-load torque (FLT), which explained the coupling damage and could have also resulted in fatigue cracks in the motor and/or fan shafts if not corrected.

The source of the excitation was electrical energy from the VFD, and not due to turbulence or pressure pulsation from the fan. During the testing, attempts were made to reduce the electrical energy by adjusting (tuning) various VFD parameters. Although these efforts were not completely successful in solving the problem, a correlation was established with the alternating torque when VFD parameters were changed.

As a temporary solution, the VFD was reprogrammed to limit the maximum operating speed and to prevent excitation of the first TNF. The long-term solution primarily involved replacing the disc pack couplings with an alternate coupling design utilizing rubber blocks in compression. An analytical torsional vibration analysis was performed to evaluate several different coupling sizes and rubber blocks of various hardness values (durometers). It was predicted that the alternating torque amplitudes would be significantly reduced once the new couplings were installed.

Follow-up field tests confirmed that the recommended coupling reduced the alternating torque to an acceptable level over the entire operating speed range. The torsional stiffness of the rubber block coupling was lower compared to the disc pack coupling, which in turn lowered the first TNF and eliminated the coincidence with the VFD excitation within the fan operating speed range. The rubber coupling also had more damping which further reduced the overall torsional vibration levels and improved VFD stability.

INTRODUCTION

The fans are used to supply combustion air to the boilers. The fans are referred to as forced draft (FD) since they force ambient air into the system as opposed to induced draft (ID) fans, which pull hot exhaust out of a furnace or boiler. The fans are driven through disc pack type couplings by induction motors rated for 112 kW at 1,783 RPM, 575 VAC, and 140 Amps.

The fan system was originally designed for constant speed operation with damper control. However, a relatively new approach of controlling the fan speed with a VFD motor was used instead. It was thought that designing the system with a VFD would offer several benefits, such as precise control of air flow, easier integration into the plant automation system, energy savings, lower motor starting current, and elimination of possible mechanical problems with the louvers (looseness over time, sticking in position, reducing effectiveness of combustion control, lowering efficiency, vibration and noise).

This low-voltage VFD uses pulse width modulation (PWM) and operates in Volts/Hz mode over an electrical frequency range of 5 to 66 Hz. The power cable length was approximately 190 meters between the VFDs located in the Motor Control Center (MCC) and the motors located in a separate building with the boilers. For this length of cable, dv/dt filters are required on the output of the VFDs to prevent voltage spikes from occurring at the motors.

Unfortunately, a torsional analysis was not performed in the design stage prior to the installation of the VFD motor / fan units. Therefore, it was unknown if there would be a dangerous torsional resonance within the operating speed range. In addition, the VFD manufacturer was unable to provide the expected level of torque ripple produced by this model of VFD, which would be a required input for the torsional vibration analysis.

Multiple coupling failures included: cracked spacer pieces, broken bolts and disc packs as shown in Figures 1 and 2. The 45-degree angle of the crack through the coupling spacer and other broken parts are failures typically caused by high torsional vibration. These failures were not specific to just one blower unit, providing further evidence that it was a design problem.

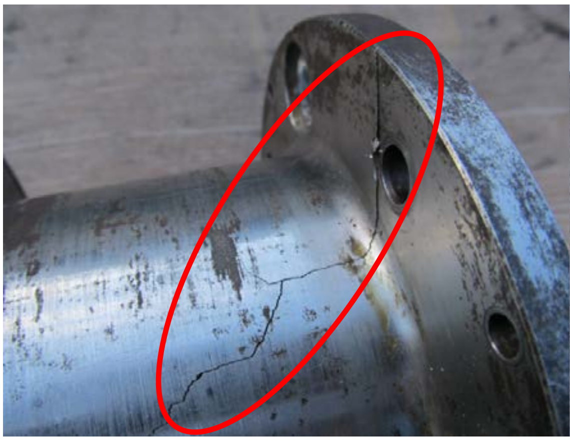

Figure 1 – Cracked Spacer Piece

Figure 2 – Damaged Disc Packs

TORSIONAL VIBRATION

Torsional vibration is sometimes referred to as “silent” because it occurs in the shaft axis of rotation that conventional vibration monitoring equipment, such as accelerometers and shaft proximity probes, will not normally detect. Coupling chatter and warping of the disc packs, also referred to as “pop canning,” are indicators of severe torsional vibration and reversing torque. In some cases, this extreme condition can actually be observed by shining a strobe light on the coupling with the guard removed. However, torsional problems are often not detected until after a failure occurs because special test equipment is required to measure torsional vibration.

VFD CONTROLS

VFDs control motor speed by varying the electrical frequency supplied to the motor. In North America, the line frequency from the electrical power grid is 60 Hz. The VFD first rectifies the supplied AC power to the DC bus. The VFD then inverts from DC back to AC power at the required electrical frequency to drive the motor at the desired speed as shown in Figure 3.

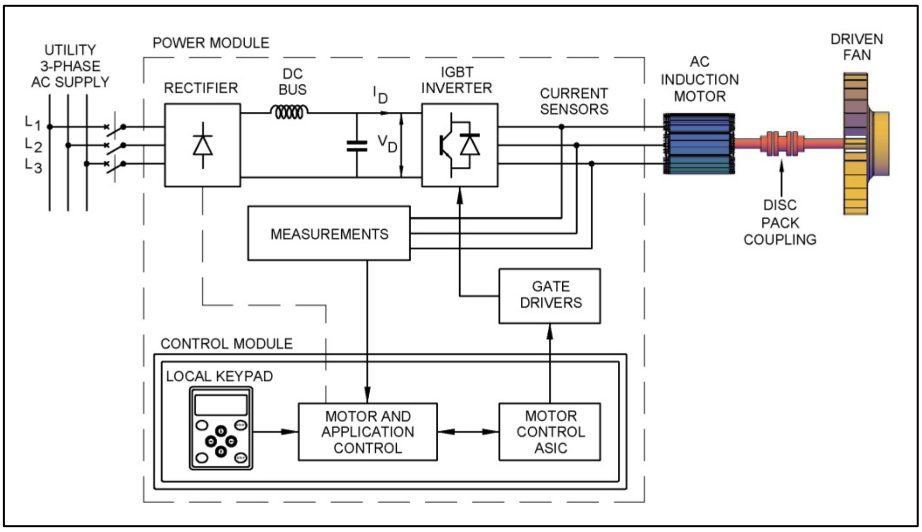

Figure 3 – Diagram of VFD Motor / Fan System

The output frequency from the drive is varied depending on the input reference speed and can even operate at frequencies above 60 Hz, but with field weakening. Because the output waveform is no longer purely sinusoidal, torque ripple can be produced. Some newer VFD technologies, such as pulse width modulation (PWM), can produce smoother waveforms, and thus reduce excitation at electrical harmonics (multiples of output electrical frequency). A multilevel DC bus with an 18-pulse inverter should produce lower amplitude harmonics than a single-level DC bus with a 6-pulse inverter. The VFD in this example uses a two-level DC bus (positive / negative). A detailed discussion of VFDs is beyond the scope of this paper. For more information on drive technology and PWM issues, see references [3,4,5].

Volts/Hz or scalar mode is the most basic VFD control method where no feedback is provided for motor speed (open loop). If the load torque changes, the induction motor will slip and the operating speed will vary slightly since the VFD output frequency remains constant. The motor current is not actively controlled by the VFD, only monitored to prevent overload. For the 4-pole motor in this case study, the fundamental (1×) electrical frequency will be approximately 2× the mechanical frequency (running speed), neglecting slip of the induction motor.

Another common control method is sensorless vector control (SVC), which is designed to provide more precise speed, or torque regulation. SVC is used for applications requiring higher dynamic performance, such as paper rolling and cranes, but is not typically required for most fan and centrifugal pump applications. In some cases, using SVC mode can cause increased excitation if the VFD is not properly configured for a high inertia system [6]. The VFD in this case study was only operated with the scalar control mode.

INSTRUMENTATION FOR FIELD TESTS

Field tests were performed to determine the cause(s) of the coupling failures. A battery-powered strain gage telemetry system was installed on the motor shaft extension (Figure 4) to measure transmitted and alternating torque. Electrical voltage and current probes were installed at the motor junction box (Figure 5) to measure the distortion in the electrical power being supplied to the motor from the VFD. All signals were continuously monitored in real-time and digitally recorded for later analysis.

During the tests, the VFD manufacturer’s representative attempted to reduce the excitation from the VFD by adjusting (tuning) many of the control parameters. When making adjustments to the VFD, it is important to have instant feedback of motor torque levels as measured by the strain telemetry system to determine whether there was an improvement, deterioration, or no effect at all. Therefore, the data acquisition computer was placed next to the local VFD panel in the MCC so the test results could be instantly accessible.

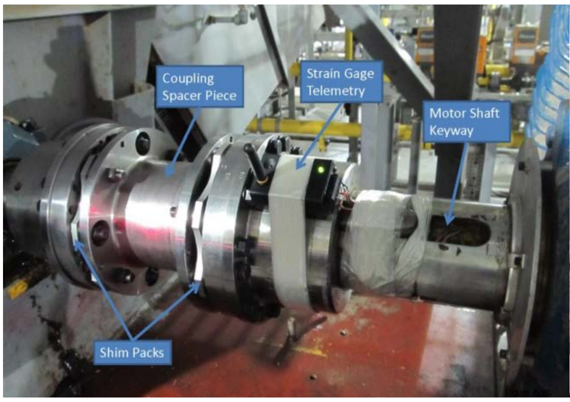

Figure 4 – Strain Gage Telemetry

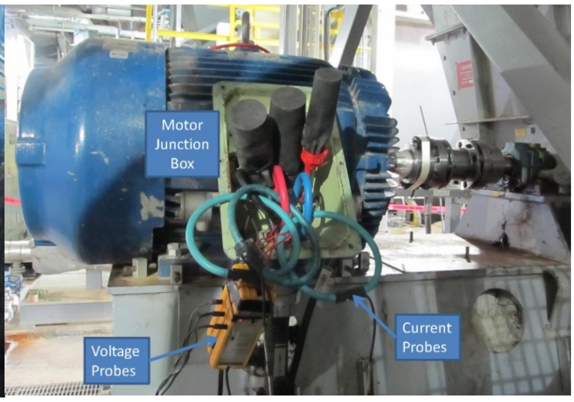

Figure 5 – Voltage and Current Probes

RESULTS OF FIELD TESTS

The fan systems were first tested with the original VFD settings (as found condition), and again while trying to optimize some of the VFD parameters. The first TNF was determined to be 57 Hz for Unit 1. As shown by the red arrow in Figure 6, the TNF was excited by energy at 1× electrical frequency when the running speed was above approximately 1,750 RPM.

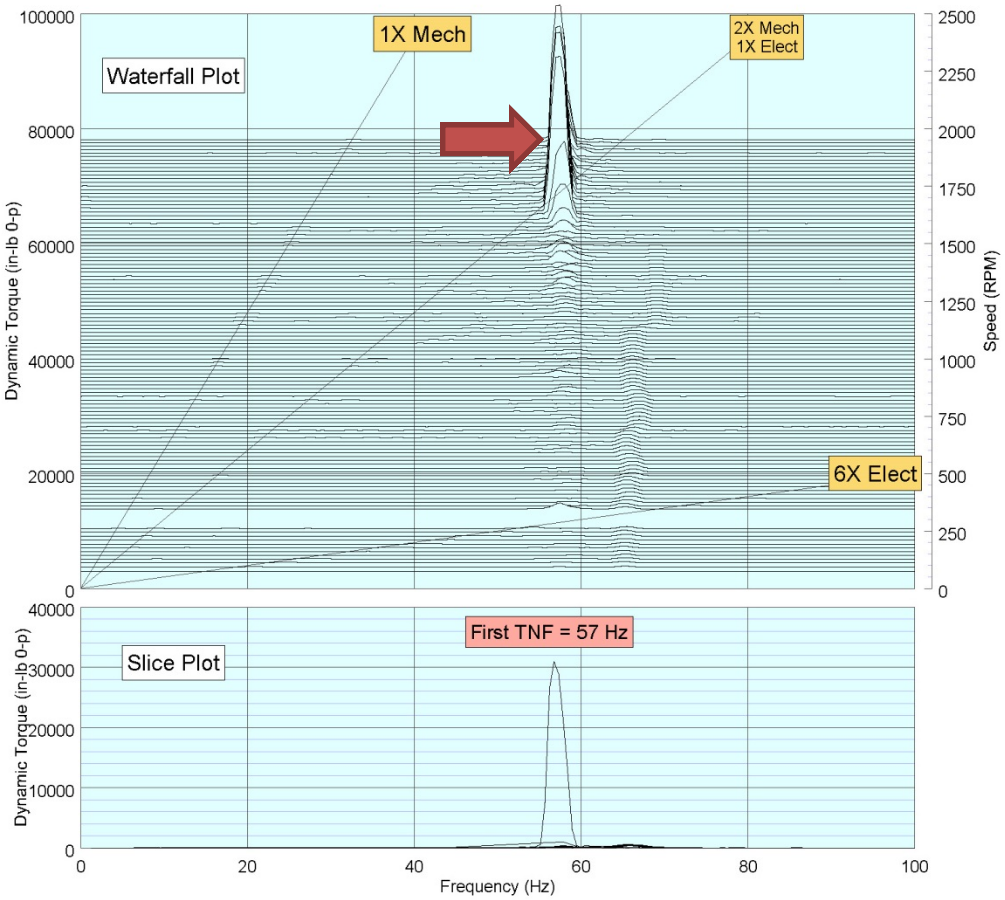

Figure 6 – Waterfall Plot of Motor Torque

As the other units were tested, the first TNF was found to vary between 55 – 57 Hz as shown in Table 1. This range in TNFs was likely due to different physical conditions of the couplings. For example, Unit 1 had a new undamaged coupling, the highest first TNF, and the highest amplification factor (AF = 180). This is considered to be a very low amount of damping, even for a torsional system. On the other hand, Unit 3 was found to have a partially damaged coupling, which resulted in a lower first TNF and AF of 35, possibly due to fretting of the cracked disc packs. Another source of variability of the TNFs could have been due to mixing of spare coupling parts in an attempt to keep the fan units in operation.

Table 1 – Measured First TNFs and Damping (Amplification Factor)

Unit	Approximate Condition	First TNF	AF
1	New disc pack coupling with limited run time	57 Hz	180
2	Newer disc pack coupling with some hours	56 Hz	80
3	Older disc pack coupling, later found damaged	55 Hz	35

The measured torsional vibration was excessive at certain operating speeds, especially when the VFD frequency was above the first TNF. As shown in the trend plot (Figure 7), the alternating torque amplitude reached 800% of FLT, which is abnormally high for centrifugal fans. Smooth fan operation would be considered 30% of FLT. Continuous operation with fully reversing torque would not only damage the coupling, but could cause fatigue cracks in the motor and/or fan shafts at the keyways, which are stress risers.

The source of excitation was electrical energy from the VFD, not turbulence from the fan. Another indication of electrical excitation is when side-bands appear in the frequency spectrum analysis of the motor current. As shown in Figure 8, the spacing of the side-bands is equal to +/- the first TNF. To verify that the VFD was the excitation source, the motor was de-energized at full speed, allowing the fan to coast down unpowered. As soon as the power to the motor was turned off, the torsional vibration at the electrical frequency immediately dissipated.

Some reductions in the electrical excitations were obtained by adjusting the VFD switching frequency. This model of VFD uses insulated gate bipolar transistor (IGBT) inverters, which are capable of switching up to 6 kHz; however, the motor manufacturer specified a limit of 3 kHz. Based on the test results, the optimum switching frequencies were different for each unit, and ranged from 1.0 kHz to 1.6 kHz.

The motor torque was stable when the torque producing current was higher and the flux was lower. This may be related to the magnetizing current of the induction motor. A simple test was performed to change the motor load by turning on and off the air pre-heater in the fan inlet duct. Without changing speed of the motor or any of the VFD parameters, the increased load from cold inlet air (higher density) versus warm inlet air (lower density) noticeably improved the stability of the VFD.

Historical trend data from the plant computer showed that Units 1 and 2 were operating within the worst portion of the speed range for dynamic torque. However, Unit 3 was primarily operating below the first TNF. This explains why Unit 3 was able to operate for a longer time before experiencing a coupling failure compared with Units 1 and 2, which experienced multiple coupling failures within shorter operating times.

As a temporary solution, the units were limited to 72% operating speed (VFD frequency of 49 Hz). Since the plant was still in start-up mode, the limitation on the fan speeds did not negatively impact the production. For the long-term, a mechanical solution was pursued since tuning of the VFDs while at the plant did not reduce the torsional vibration to acceptable levels.

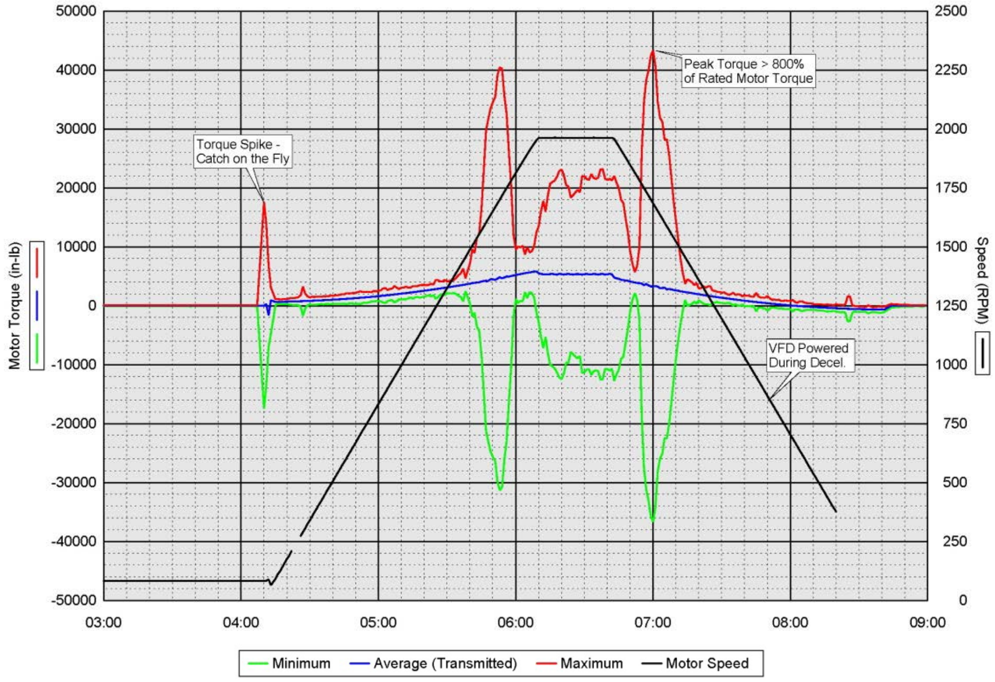

Figure 7 – Trend Plot of Motor Torque

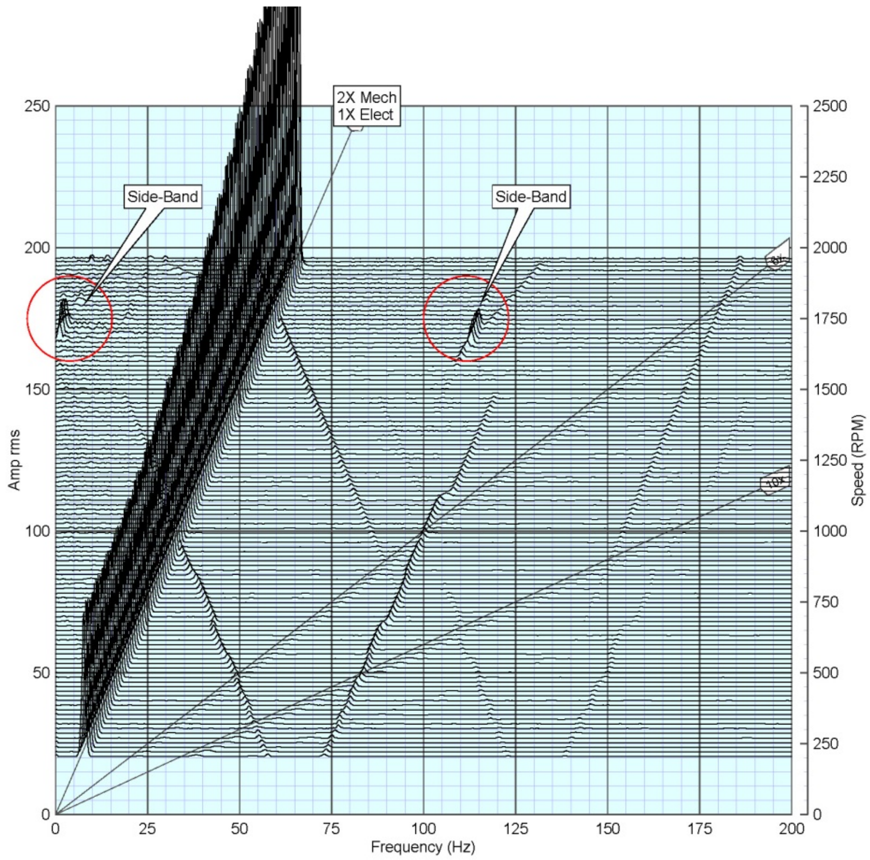

Figure 8 – Waterfall Plot of Motor Current

TORSIONAL VIBRATION ANALYSIS

A mass-elastic model was created to represent the fan system “as found” with the original disc pack coupling. Based on the information and drawings provided for the motor, coupling, and fan, the system first TNF was computed to be 40.5 Hz. However, this computed value did not agree with the measured TNF of 57 Hz. The source of the erroneous stiffness and/or inertia information was unknown. The computed TNF would normally be expected to match more closely [7]. The mass moment of inertia (WR^2) of the motor was adjusted to force the computer model to agree with the measured TNF. Modifications were then evaluated using this normalized mass-elastic model.

The end user initially requested that a larger diameter fan shaft (increased torsional stiffness) be analyzed to determine if the first TNF could be raised above the maximum operating speed. However, the predicted first TNF was only increased to 61 Hz, which would still be excited by the electrical energy from the VFD in the upper portion of the speed range. Therefore, increasing the diameter of the fan shaft was not a viable solution.

Next, alternate couplings were investigated to determine if a coupling modification would allow the fan system to safely operate throughout the entire speed range. Resilient style rubber block couplings were considered because of additional torsional damping to the mechanical system. A larger than normal service factor of 5 was specified to ensure that the rubber blocks would not be damaged by the high torsional excitation levels from the VFD.

The rubber blocks are available in various durometers (shore hardness). A torsional analysis was performed to select the durometer block with optimum torsional stiffness and damping properties. A coupling with rubber blocks in compression has a progressive torsional stiffness that increases with transmitted torque. Based on the fan affinity laws, the load torque will vary with speed squared which changes the torsional stiffness of the coupling and the resulting TNFs. Therefore, the first TNF is not a constant frequency, but rather a range of values which vary with the transmitted torque.

Figure 9 shows an interference or Campbell diagram, which is a plot of the computed first TNFs versus speed for three types of rubber blocks (durometers 60, 70, and 80), and excitation frequencies (1× mechanical and 1× electrical). The 60 durometer blocks are the softest and have the lowest damping (highest amplification factors). The 80 durometer blocks are the hardest and have the highest damping (lowest amplification factors).

As shown in Figure 9, the first TNF increases with the running speed as the transmitted torque is increased. The coupling with the 60 durometer blocks is unacceptable because the first TNF is coincident with the 1× electrical frequency near a running speed of 1,700 RPM (see green arrow in Figure 9), which is well within the operating speed range. The coupling with the 70 durometer blocks is also unacceptable because the first TNF is coincident with the 1× electrical excitation near the maximum running speed.

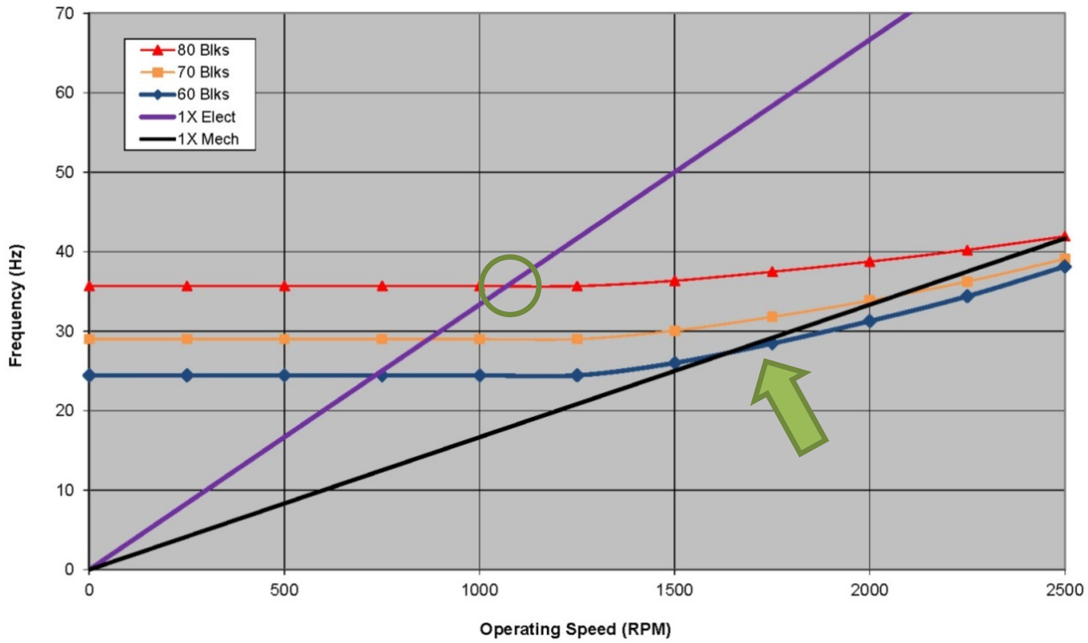

Figure 9 – Interference Diagram Showing First TNF with Various Rubber Blocks

The green circle in Figure 9 shows the torsional resonance where the 1× electrical frequency is predicted to intersect the first TNF. The 80 durometer blocks are the best choice because the first TNF is not coincident with the electrical frequency within the operating range, and the 80 durometer blocks have more damping which should further reduce the alternating torques.

Steady-state forced response calculations were performed using a proprietary computer program developed by the author (Figure 10). Although the VFD manufacturer did not provide the expected torque ripple, based on previous measurements the electrical excitation was determined to be 5% to 7% of FLT. The worst case was applied to the motor in the torsional model.

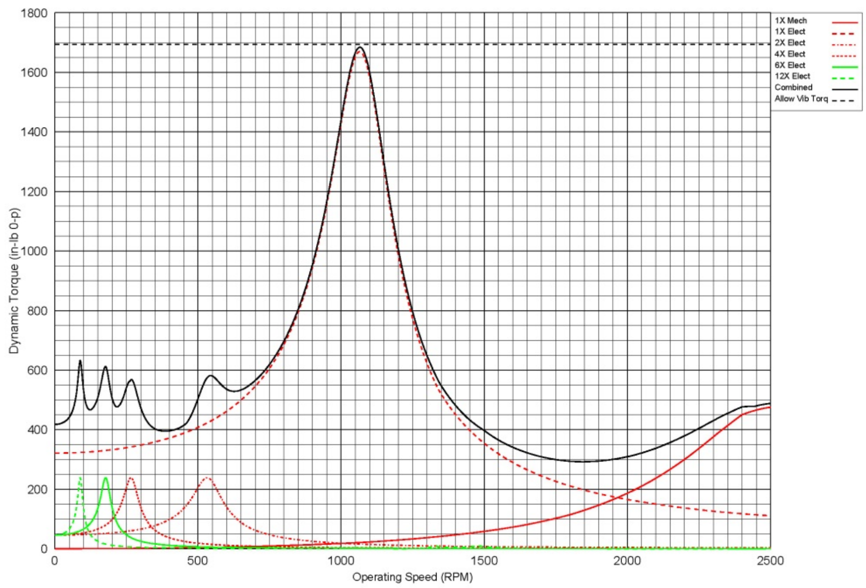

Figure 10 – Predicted Alternating Torque in Coupling with 80 Durometer Blocks

Results were compared with the allowable limits for vibratory torque and heat load provided by the coupling manufacturer. It is important to de-rate these coupling limits if the predominant torsional vibration frequency differs from the coupling manufacturer's catalog vibration frequency (typically 10 Hz).

Note that rubber is a natural material and can have up to 25% uncertainty associated with the reported torsional stiffness values in the manufacturer's catalog. Temperature and age can also affect the torsional stiffness of the rubber blocks. To allow for these uncertainties, it is preferred to have a separation margin of at least 10% to 15% between TNFs and excitation frequencies. However, for a system with a wide operating speed range, this may not always be practical. Therefore, any torsional resonances within or near the operating speed range should be fully evaluated.

The torsional analysis results showed that additional damping from the rubber blocks should limit the alternating torque to an acceptable level in the coupling, as well as significantly reducing the alternating shear stresses in the motor and fan shafts. Therefore, the replacement couplings were ordered with 80 durometer rubber blocks and custom spacer pieces to avoid having to relocate the axial position of the motors relative to the fans.

FOLLOW-UP FIELD TESTS

Follow-up field tests were performed on all three units after the new couplings were installed. The coupling with cover plate removed is shown in Figure 11.

Figure 11 – Coupling with Wedge Style Rubber Blocks

The VFDs were tested with the modified settings and with the factory default parameters. All three units performed similarly during the tests. The first TNF of all three units was measured to be 33 – 34 Hz, which agreed with the computed TNF and verified the normalized mass-elastic model. The damping values were similar on the three units (AF = 4 to 7) and agreed with the catalog value for the 80 durometer rubber blocks. Note that these AFs are significantly lower than the value measured with the original undamaged disc pack coupling (AF = 180).

The first TNF and damping values were determined for Unit 1 by analyzing the torque versus time waveform that was measured immediately after de-energizing the VFD. As shown in Figure 12, the first TNF was excited when the driving torque was suddenly removed. The first TNF was determined from the inverse of the time period between cycles and taking a peak hold FFT of the torque signal. To estimate the damping of the coupling, the logarithmic decrement (δ) was calculated from the decreasing torque amplitudes, and converted to an amplification factor [8], where AF ≈ π / δ.

Figure 13 is a waterfall plot of the measured alternating torque versus speed with the new coupling. As shown, the first TNF was no longer excited. In addition, the levels at 1× running speed and 1× electrical frequency (≈ 2× running speed) were significantly reduced due to the increased damping from the rubber blocks in the new coupling. For clarity, the vertical Y-scale in Figure 13 was changed by a factor of 20 (full scale = 5,000 lb-in 0-p) compared to the vertical Y-scale in Figure 6 (full scale = 100,000 lb-in 0-p).

VFD operation remained stable with and without using the pre-heaters; therefore, the speed restrictions were removed from the units. The VFDs were auto-tuned and left with the uniform settings (VFD switching frequency of 3.0 kHz). The plant reports that these fan units have been in operation for approximately four years without any more failures.

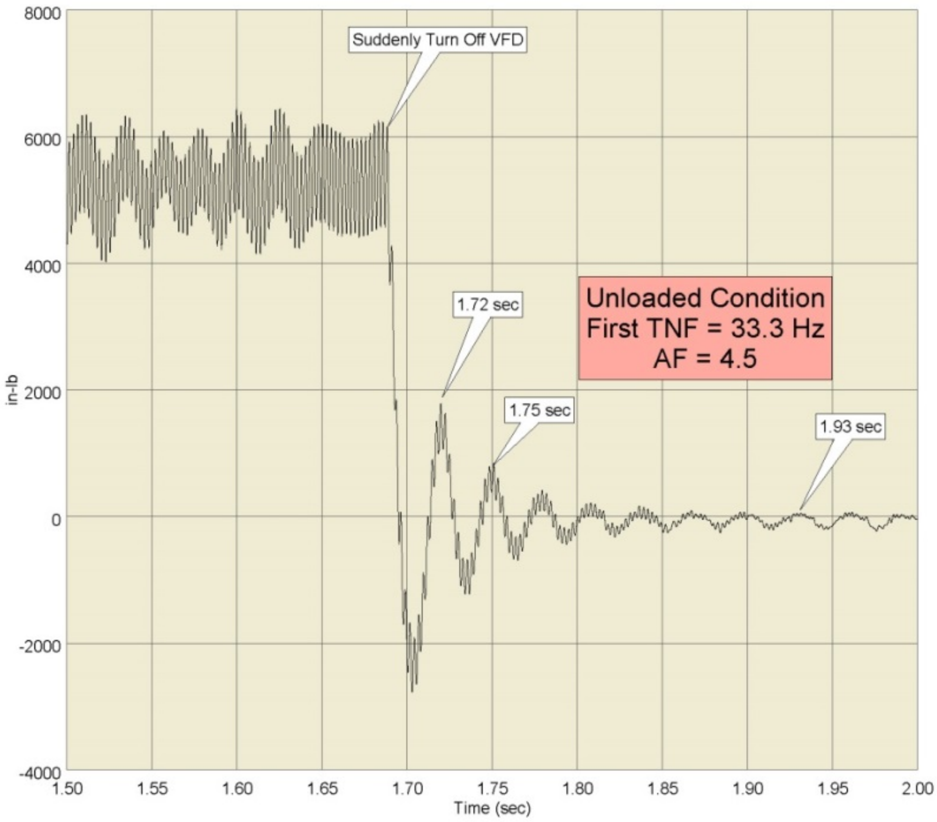

Figure 12 – Time Waveform of Torque with New Rubber Coupling

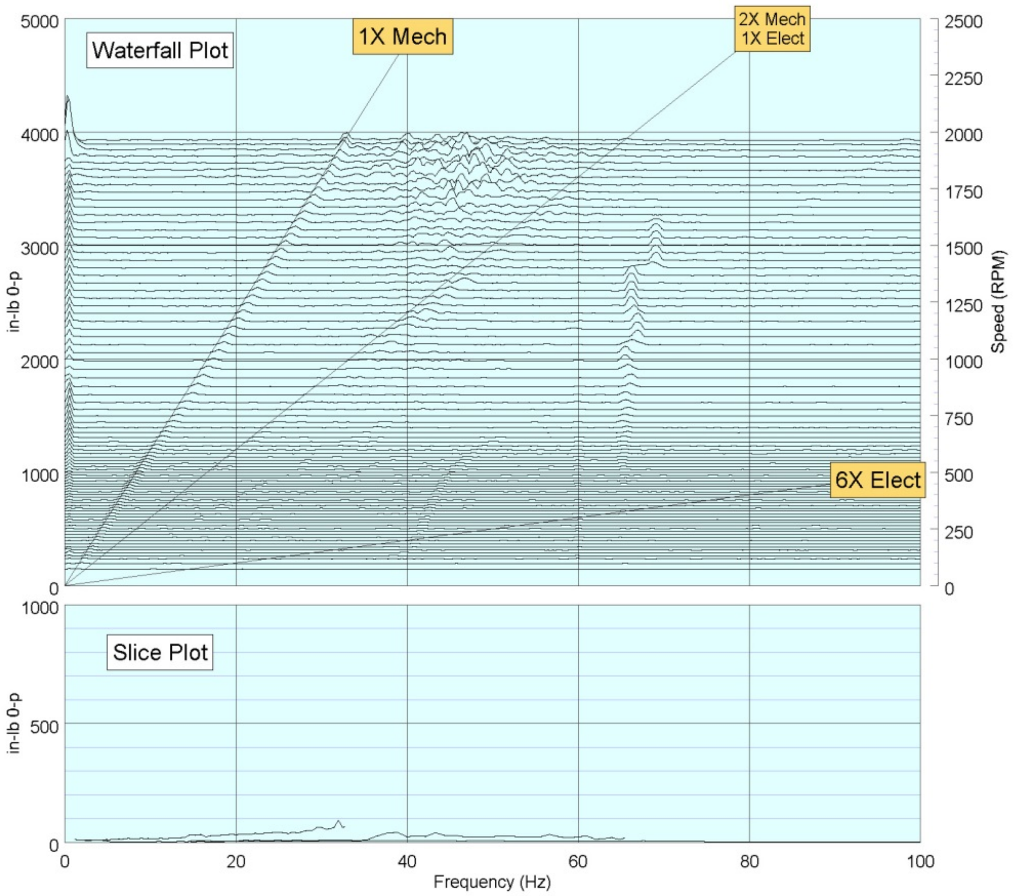

Figure 13 –Waterfall Plot of Torque with New Rubber Coupling

CONCLUSIONS AND RECOMMENDATIONS

Sufficient separation margins from dangerous torsional resonances are required to prevent failures. However, for this system with the original disc pack couplings, the first TNF continued to be excited, even when operating well above the torsional resonance. This behavior and the side-bands observed in the motor current are indications of unstable VFD control, and/or issues with the PWM inverter. Contributing factors to VFD instability may also include extremely low mechanical damping of the disc pack coupling, and long cable length between the VFD cabinet and the motor, which results in increased capacitance and susceptibility to electrical noise. Other references have reported electrical noise to be a source of increased VFD excitation [9].

This system had a large fan to motor inertia ratio, exceeding 30 to 1. For the first torsional mode, the motor is near the anti-node (point of maximum oscillation) and acts as a torsional pendulum. Whereas, the fan is at the node (very low oscillation) and acts as a large flywheel that resists sudden speed changes. For a torsionally stiff and lightly damped mechanical system such as this, the motor will be very sensitive to electrical excitation or sudden speed adjustments from the VFD. However, the fan impeller is relatively insensitive to torsional excitations due to the “flywheel effect.”

Improved reliability of these fan systems was achieved by installing couplings with rubber blocks, which added damping and detuned the first TNF. This type of coupling is commonly used in large fan systems at power plants. When specifying a coupling with rubber elements in VFD service, consider oversizing the coupling (use greater service factor) to ensure that the rubber blocks do not prematurely crack or overheat. Periodic inspection and replacement of rubber blocks are recommended per the coupling manufacturer. In addition, the end user has specified that any new VFD motor / fan systems be designed with 18-pulse VFDs for smoother output.

Additional research is recommended to better understand why certain VFD parameters significantly affect the electrical excitation of the motor. These could include: PWM switching frequency, linear versus squared V/Hz ratio, and effect of magnetizing current. For this case study, lower VFD switching frequencies of 1.0 – 1.6 kHz initially worked better than the higher switching frequency of 3.0 kHz. Using a squared V/Hz ratio also seemed to work better for this system than the linear V/Hz ratio. According to the VFD manual, using the squared V/Hz option will cause the motor to operate under magnetized (reduced motor flux), which will produce less electromechanical noise. Finally, when the inlet air was cooler, it was demonstrated that the VFD was more stable presumably due to the higher active torque producing current relative to the reactive magnetizing current. After installing the rubber couplings, the VFD motor / fan systems were much less sensitive to these VFD parameters and various operating conditions.

SUMMARY

The problem with repeated failures of the disc pack couplings was solved using a combination of field measurements and analytical techniques:

- The coupling failures were identified as typically caused by high torsional vibration (crack at 45-degree angle through coupling spacer).
- Initial field tests were conducted to obtain torsional vibration and electrical data. Excessive alternating torque was measured at certain operating conditions and found to be caused by the VFD. While at the plant, attempts were made to optimize (tune) various VFD parameters.
- A computer model was created to compute the TNFs. The computer model was adjusted to agree with the measured data.
- Possible modifications to the fan system were evaluated using the normalized computer model. The proper coupling size and durometer blocks were selected based on the worst case excitation levels.
- After the new rubber block couplings were installed, follow-up field tests confirmed that the torsional vibration problem was solved.

NOMENCLATURE

AC	alternating current
AF	amplification factor, non-dimensional
CPM	cycles per minute, unit for vibration frequency
DC	direct current
dv/dt	filter device for limiting voltage spikes
FD	forced draft fan
FFT	Fast Fourier Transform
FLT	full-load torque
Hz	Hertz is unit for vibration frequency in cycles per second
ID	induced draft fan
IGBT	insulated gate bipolar transistor used in VFD inverter
kW	one kilowatt is equal to one thousand watts, unit for power
MCC	Motor Control Center, building where VFDs are located
p-p	peak-to-peak, double amplitude
PWM	pulse width modulation
RPM	revolutions per minute, unit for speed
SVC	sensorless vector control method for VFDs
TNF	torsional natural frequency
V/Hz	voltage per frequency in Hz, VFD scalar mode
VAC	voltage for AC motor
VFD	variable frequency drive
WR^2	mass moment of inertia
0-p	zero-peak, amplitude
δ	logarithmic decrement

ACKNOWLEDGEMENTS

Special thanks to Jay Dalcourt and Donald Smith for their very helpful reviews of this paper, and to Mark Broom for preparing Figure 3.

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