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## TORSIONAL INSTABILITY OF COOLING TOWER FAN DURING INDUCTION MOTOR STARTUP

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## **ABSTRACT**

Torsional instability during starting of an induction motor has caused excessive torque oscillation and eventual failure of gearbox pinion teeth of a cooling tower fan train. A torsional instability appeared due to a steep slope of the induction motor starting torque curve, combined with a torsionally soft driveshaft and a large moment of inertia of the driven fan.

In this paper, mechanisms of torsional instability occurrence during startup of an induction motor in a cooling tower fan train are discussed, as well as details of the investigation outcomes of troubleshooting activities, including field measurements. The methods of modeling and analysis are also described. To examine the mechanism of torsional instability occurrence, in addition to transient torsional analysis, the concept of torsional stability analysis is introduced. Practical engineering guidelines to avoid this type of problem, such as a motor selection guideline, are also addressed.

## **INTRODUCTION**

Cooling towers are commonly used in many industries to extract waste heat and discharge it to the atmosphere. They are vital facilities on which manufacturing, processing, and power generation plants depend for continuous operation. Their malfunction results in reduction or complete loss of plant capacity. However, cooling tower fans are usually equipped with much less, if any, protective instrumentation compared with heavily instrumented critical equipment such as process centrifugal compressors in petrochemical plants. This is because technologies adopted in cooling tower fans are generally considered robust and well proven with simple construction and low rotational speed, and also because they are usually spared with other stand-by trains. Nevertheless, technical difficulties sometimes arise. This paper intends to shed light on one such incident.

A cooling tower in a newly constructed petrochemical plant was put into service since the early stage of the plant commissioning to provide cooling capability during the commissioning activities before handing over to the end user. During this commissioning stage, the cooling tower fans were not run continuously; instead, they operated based on cooling demand. In three months after the service commencement, however, one gearbox suddenly exhibited abnormal noise, necessitating manual stoppage of that train. Visual



inspection through a maintenance hole revealed severe damage at the helical pinion teeth. Follow-up inspection of the remaining trains identified severe damages of the helical pinion teeth in 14 out of a total of 16 trains.

Swift replacement of the damaged pinions was therefore mandatory. More importantly, the root cause of the pinion teeth damage needed to be urgently identified in order to establish effective countermeasure plans and to eliminate the possibility of damage repetition after pinion replacement.

This paper discusses the root cause investigation, its findings, and lessons learned from this troubleshooting.

## DESCRIPTION OF EQUIPMENT

An open circuit, counter flow cooling tower is equipped with a total of 16 fan trains (of which two trains are stand-by) to provide mechanical draft of cooling air. A three phase, four pole, 50 Hz induction motor with nominal output power of 295 hp (220 kW) drives each fan train. The induction motor is started direct on line without a soft starter. Nominal operating speeds are 1490 rpm for the motor, and 119 rpm for the fan. A two stage speed reduction gearbox (first stage = bevel, second stage = helical) is applied, on which the fan blades (which are made of composite material) are mounted on the hub plate installed at the output shaft. The induction motor and the gearbox are connected by a long coupling, which is termed a driveshaft in this application. The driveshaft spacer is made of carbon fiber reinforced composite material, and is approximately 15 ft (4.7 meter) in length. The induction motor and the gearbox are mounted on concrete foundations, see Figure 1.

The gearbox input shaft has a bevel pinion at one end and is connected to the driveshaft at the other end, while the intermediate shaft has a bevel gear at its upper part and a helical pinion just beneath it. The output shaft has a helical gear in the middle, and the fan blades are mounted at its top via a hub plate. The intermediate shaft is oriented perpendicular to the input shaft, and in parallel with the output shaft. Teeth damage was observed at the helical pinion, see Figure 2.

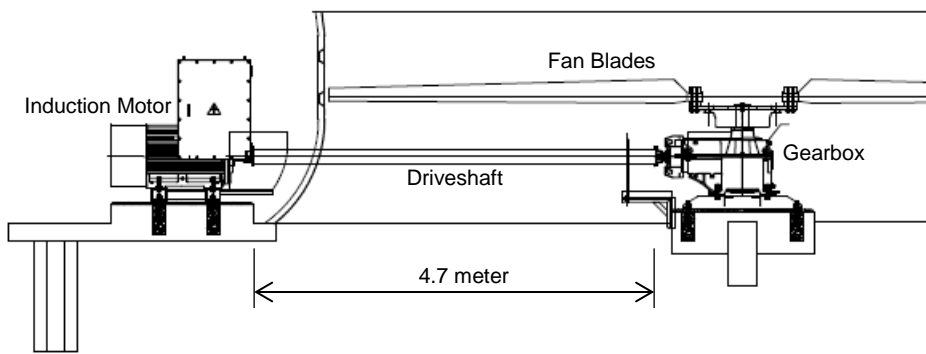


Figure 1. Outline of Cooling Tower Fan Train



Figure 2. Damaged Pinion Teeth

## INITIAL INVESTIGATION

Metal on the loaded side of the helical pinion teeth surfaces had flaked off. The damage appeared to have originated mainly from the linear cracks at the pinion teeth root. Meshing of a helical gear and its pinion could cause sliding motion leading to peak tensile stress at the root of each pinion tooth. If the pinion and gear teeth were pressed by abnormally strong loads, such damage would have resulted. It was also noticed that the severity of teeth damage was not uniform; some teeth were more severely damaged than the other teeth, and no clear pattern (such as in the case of assembly phase damage) could be found. Since the damage appearance and damaged teeth pattern were rather irregular, it was suspected that irregular contact pressure might have acted at the meshing surface. Mechanisms to cause such irregular contact pressure were, however, unclear at this initial investigation stage.

As such, possibilities of defects in design or manufacturing process of the gearboxes as well as of field installation defects were initially investigated. Possibilities of excessive teeth loading due to torsional resonance during steady state operation and of bus reclosure during momentary power interruption were also investigated.

### *Possibility of Design or Manufacturing Defects of Gearbox*

The application of a new or unproven design element was suspected; hence the gearbox manufacturer's delivery record was thoroughly rechecked. The gearbox was however not of new design; the manufacturer had delivery records of more than 300 identical units with the same design. Possibility of new or unproven design element was therefore excluded.



Teeth overloading due to poor design rating was likewise suspected. The helical gearset was designed to meet the requirements of ANSI/AGMA 2001 as well as Cooling Technology Institute Std-111 (see list of references). As per these standards, allowable power ratings in terms of surface durability (pitting resistance) and teeth strength (bending strength) were firstly calculated based on the design parameters such as gear and pinion geometry, and the calculated allowable power ratings were then divided by the motor nameplate power to derive the service factor. The service factor derived in this manner was actually 2.63 for these fan trains, thus exceeding the minimum requirement of 2.0 to accommodate design allowances. Moreover, among the manufacturer's delivery records of identical design gearboxes, service factors smaller than 2.63 were found, and they were operating without any problem. It was concluded that gearbox service factor was adequate, and that an insufficient service factor could not be the primary reason of teeth damage.

Possibilities of manufacturing defects were also investigated. Recorded metal composition of the gear and pinion material satisfied the specifications and the variation was minimal, and rechecking of the composition taken from the specimens of the damaged pinion teeth confirmed no abnormality (same composition as issued by the steel mill).

To increase surface resistance against wear, the teeth surface was carburized. Abnormally carburized structure such as over-carburized layer (e.g. spheroidal cementite) was not found in any of the investigated teeth, and the depth and hardness of the effective case hardened layer for the entire tooth section taken from the specimens of the damaged pinion teeth were all within the manufacturer's criteria.

Possibilities of gearbox shafts misalignment due to assembly mistake or bearing defect were also suspected but eventually ruled out after investigation. Teeth contact was rechecked for all the gearboxes, and no abnormality was observed, with helical pinion and gear contact pattern uniformly distributed and centered at the teeth width. Moreover, gearbox teeth accuracy was rechecked at the manufacturer's facility and confirmed to satisfy the accuracy class as per the applicable standard.

Therefore it was concluded that the damage was not primarily due to manufacturing or design defects.

#### *Possibility of Field Installation Defects*

Installation condition of the fan blades was confirmed to be adequate for all the trains (blade angle and tip clearance were within specification, and fan blades fastening torque was as specified). Measured motor voltage and current during steady state operation were also as specified in the design, indicating no abnormality in the supplied electricity. Driveshaft installation condition (alignment between the motor and the gearbox) was within both angular and axial tolerances for alignment. No improper installation condition of the gearboxes (e.g. loose anchor bolts, insufficient grouting, or cracked concrete foundation) was found.

Oil samples taken from the gearboxes detected no abnormality such as abnormal viscosity and water content. Debris of the flaked off teeth metal were detected in the oil sample, but this was an outcome rather than the cause of the teeth damage. Oil temperature during operation was also rechecked, and it was at normal level for all the trains.

Therefore it was judged that the damage was not due to field installation defects.

#### *Possibility of Torsional Resonance during Steady State Operation*

If excessive torsional vibration occurred due to resonance, stress levels exceeding the fatigue limit of the teeth material could act when teeth were meshing. Moreover, for a geared train, torque reversal due to excessive torsional vibration could cause temporary teeth separation and recontacting, which would have resulted in shock load (Szenasi and von Nimitz 1972).

The possibility of torsional resonance was therefore checked by torsional analysis (analytical model is described in detail in a subsequent section). The calculated torsional natural frequencies were however far away from the potential excitation sources (rotation frequencies of the shafts, gearsets meshing frequencies and their harmonics, fan blade pass frequency, electrical line frequency and its harmonics, and motor slip frequency).

It was therefore concluded that teeth damage due to torsional resonance during steady state operation was unlikely.

#### *Possibility of Motor Restarting during Momentary Power Interruption*

Motor restarting during momentary power interruption could cause premature failures of cooling tower fan blades, driveshafts, and gearboxes (Heard 1997). In the event of electrical power interruption, some electrical flux could be trapped in the motor. If the bus reclosure took place shortly afterwards, it could have been in phase with the trapped electrical current, producing significant torque spikes far exceeding the equipment's service factors.

Possibility of motor restarting during momentary power interruption was therefore investigated. The electrical system was however protected against power interruption (protective circuitry did not allow immediate motor restarting in case of power



interruption), and such incident had never occurred in these trains. Teeth damage due to such mechanism was therefore concluded highly unlikely in these trains.

## FIELD MEASUREMENTS DURING MOTOR STARTUP

Teeth damage due to gearbox defects or installation mistakes was unlikely. Either a torsional resonance during steady state operation or a motor restart right after power interruption was also considered improbable.

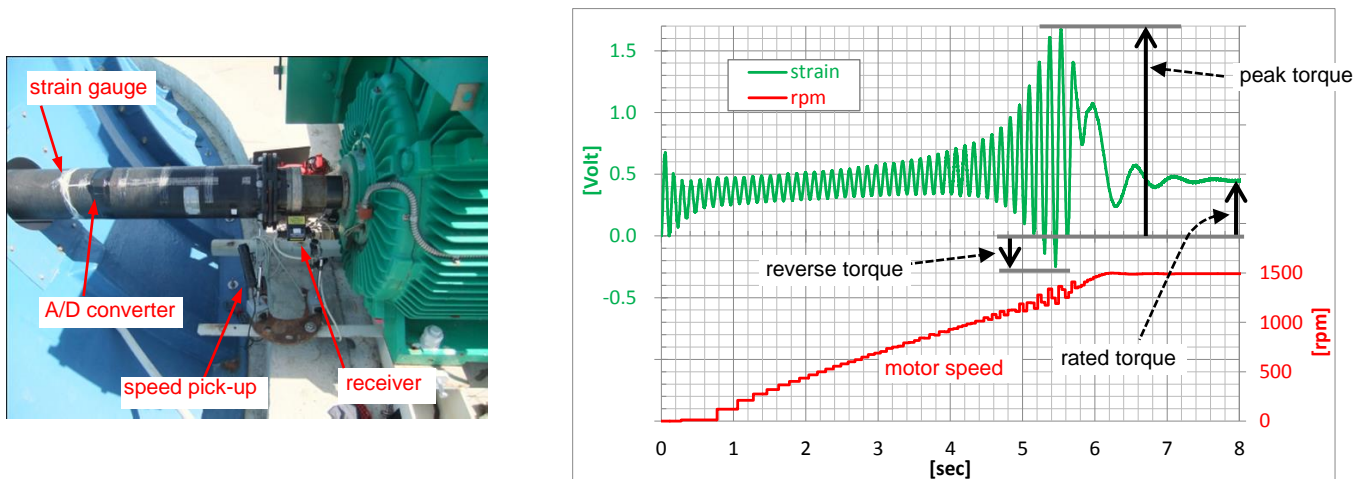
Since no abnormality was detected during steady state operation, abnormality during transient operation such as motor startup was the only remaining possibility. Moreover, rattling noise was noticed from a gearbox during the startup of a fan train. Detailed field measurements during induction motor startup were therefore judged necessary.

### *Driveshaft Strain and Motor Rotational Speed*

To confirm the transient behavior of the transmitted torque during the induction motor's startup, the driveshaft strain was measured by strain gauges. Rotational speed of the motor was also measured by a once-per-revolution marker and an optical speed pick-up sensor at the motor end of the driveshaft (Figure 3).

In Figure 3, the horizontal axis is the time from the moment of motor energizing, and the green line represents the measured strain gauge signal output while the red line represents measured shaft rotational speed. Motor rotation reaches its rated speed approximately 6.5 seconds after its starting. Torque oscillation starts to increase at around 4.0 sec, and reaches its maximum at around 5.5 sec (immediately before reaching the rated speed). At the timing of maximum torque oscillation, the motor speed also noticeably fluctuates. This correlation clearly demonstrates that the torque and the speed oscillations are caused by significant torsional vibration. Moreover, torque and motor speed oscillations appear near the calculated torsional natural frequency of the train (8.4 Hz). A key observation is that the frequency of vibration exhibits no tendency to track running speed. The frequency remains essentially constant at 8.4 Hz during the entire startup.

Because of the torque oscillation, peak torque reaches roughly four times the rated torque, far exceeding the gearbox service factor of 2.63. Although the measured peak torque factor of four is below the maximum momentary or starting load factor limit of twice the service factor recommended by the Cooling Technology Institute in CTI Std-111 (see list of references), it is still considered significant. Moreover, torque reversal also occurs during the maximum torque oscillation.



**Figure 3.** Instrumentation Set-up (left) and Measurement Result (right) of Shaft Strain and Motor Rotational Speed

### *Electrical Current*

The motor current (ampere) of one phase (out of three phases) was measured by installing a temporary ammeter probe at one of three cable terminal points of the electrical distribution panel (Figure 4).

In Figure 4, the measured output signal of motor current (pink line) is plotted together with the measured driveshaft strain (green line). Measured motor current pulsates during 5.0 ~ 6.0 sec, coinciding with the timing of maximum torque oscillation. This current



pulsation is evidently caused by torque oscillation due to torsional vibration. (Note: In Figure 4, output signal of the ammeter is plotted. Although unit of this graph is in voltage, it does represent the motor current in ampere, not motor voltage.)

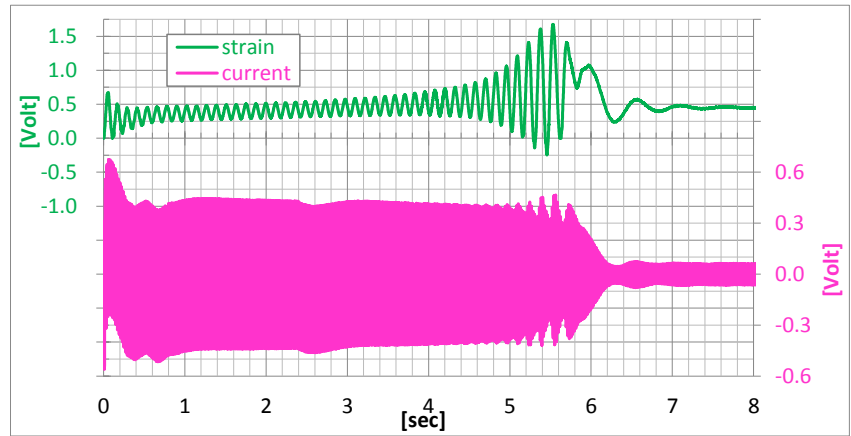
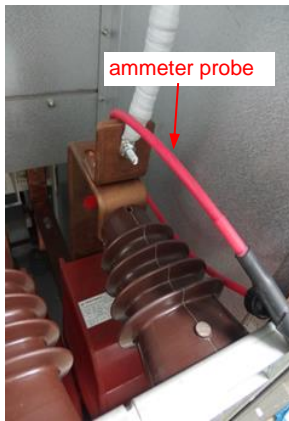


Figure 4. Ammeter for One Phase (left) and Measurement of Motor Current and Strain (right)

#### Excessive Torsional Vibration and Teeth Separation / Recontacting

The gearbox vibration was measured by an accelerometer (Figure 5).

In Figure 5, measured accelerometer vibration (blue line) is plotted together with the measured driveshaft strain (green line). At the timing of maximum torque oscillation (5.0 ~ 6.0 sec), impulsive vibrations are observed. Torsional vibration causes massive torque oscillation, and when the torque oscillation is at its maximum, torque reversals occur. Because of the torque reversal, tooth meshing is temporarily separated, and the separated tooth travels in reverse direction within the backlash and bounces with the backside of the adjacent tooth. Afterwards, when the positive torque is recovered, it is rapidly accelerated and bounces with the mating tooth (tooth recontacting). The fact that the impulsive vibrations in accelerometer signal exactly coincide with the occurrence of each torque reversal (i.e. timing of tooth separation and recontacting) endorses this mechanism. It is judged that an impact load certainly acts due to tooth recontacting after each torque reversal.

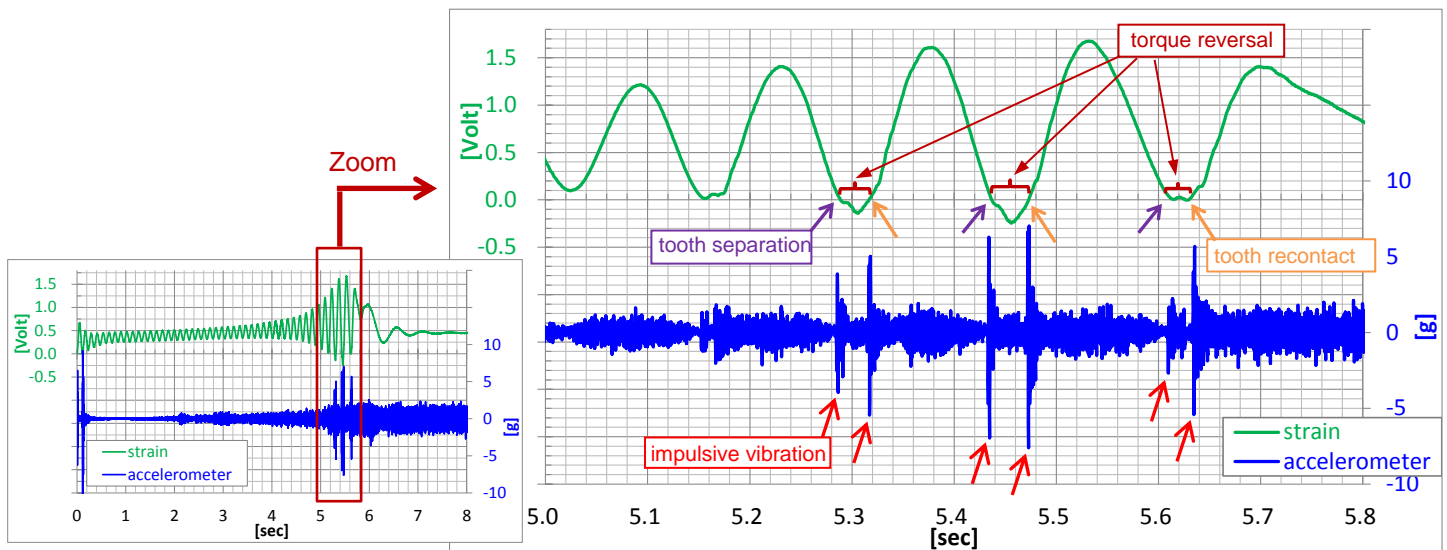


Figure 5. Measured Shaft Strain and Gearbox Vibration



### *Summary of Measurement Results*

During starting of this induction motor (i.e. during motor acceleration before reaching its rated speed), large amplitude torsional vibration occurs, generating torque and speed oscillations. The peak transmitted torque reaches four times the rated torque, far exceeding the gearbox service factor. Moreover, due to massive torque oscillation, torque reversal also occurs, causing gear teeth separation and recontacting. When the separated tooth recontacts, shock load acts at the tooth surface.

Hence, it is very likely that every time the motor is started, it would have caused certain level of tooth damage, and repeated startup would have accumulated that damage, eventually resulting in failure.

It should be noted that the pinion teeth damage occurred during the plant's commissioning period in relatively short duration (after three months since service commencement). Compared to the plant's commercial operation period, these cooling tower fans were subject to more frequent turning on and offs during this period (because their operation depended on the cooling demand, which was relatively less stable during the commissioning period). Tooth failure mechanism due mainly to startup of the motor makes sense from this aspect as well.

It is concluded that peak torque of four times the rated torque as well as shock load upon tooth recontacting would have initiated pinion tooth damage, and that the accumulated damage would have eventually resulted in tooth failure. Although it is difficult to determine which of (1) peak torque in excess of service factor, or (2) shock load due to tooth recontacting, mainly contributed to the failure, it is confidently concluded that torsional vibration during motor startup caused tooth damage.

### **TORSIONAL ANALYSIS**

Field measurements revealed occurrence of large torque oscillation during motor starting, which caused tooth damage and ultimate failure. Mechanism of large torque oscillation was however still unknown. In order to explain the torque oscillation mechanism and to establish effective countermeasure plans, detailed studies by torsional analysis were carried out.

#### *Analytical Model and Calculated First Torsional Natural Frequency*

In carrying out torsional analysis, accurate representation of the entire rotor system in the analytical model is essential (Wang et al. 2012). In order to accurately represent the entire rotor system, data from each equipment manufacturer were utilized (Figure 6).

Motor shaft dimensional data and moment of inertia of the major motor components (such as motor core and motor cooling fan) were modeled.

Torsional stiffness data and moment of inertia of the driveshaft spacer were obtained from the driveshaft manufacturer. Since the spacer is made of composite material, the torsional stiffness of the driveshaft would be nonlinear (i.e. it varies depending on the load condition) in the exact sense. However, detailed nonlinear material properties could not be obtained from the driveshaft manufacturer due to proprietary restrictions, and only the material property at the rated torque could be disclosed. A linear torsional stiffness was therefore assumed in the model. As shown in the subsequent section, this simplification did not significantly affect the overall accuracy of the analysis results.

One-third shaft penetration was assumed for the hubs of the driveshaft and of the fan flange to account for freely twisting section of the shrink fit or keyed hubs.

For each shaft of the gearbox, shaft dimensional data and moment of inertia of the major components (such as of bevel and helical gears, bearing inner races, and gearbox fan) were fully reflected into the analytical model. Tooth meshing stiffness was estimated based on the formulae presented in API RP 684. Note that, when a torque reversal occurs, tooth mesh temporarily separates and eventually recontacts. Hence, tooth meshing stiffness would actually be nonlinear (i.e. it varies depending on the meshing condition). Such nonlinearity due to tooth separation was not considered in this analysis.

Data regarding the moment of inertia of the fan blades as well as dimensional data of the fan flange and hub plate were obtained from the fan manufacturer and were fully reflected into the analytical model.

Differences of rotational speed by the gear ratio were accounted for by adjusting the inertia and stiffness in the predictions.

Note that the intermediate shaft and the output shaft of the gearbox are shown as parallel to the input shaft in the analytical model of Figure 6, while they are actually oriented perpendicular to the input shaft. Note also that geometries of bevel gear wheel and helical gear wheel are not represented in Figure 6, as gear wheels contribute to the analysis results only by inertia, which are reflected as lumped inertia in the analytical model.

The calculated first torsional natural frequency is 8.4 Hz. The mode shape appears in Figure 6 with the entire train referenced to the motor speed. In this first torsional mode, twisting motion of the driveshaft is the primary region of deformation, which makes sense because driveshaft is the torsionally softest component of the train.



The moment of inertia of the fan is by far the largest in the system. It is 1323 times larger than that of the induction motor, and when referenced to motor speed is 8.4 times larger. The large difference in inertia is reflected in the mode shape, with motion at the motor being about eight times larger than at the fan. As will be seen later, this large difference is a key factor in the instability.

Campbell diagram of the calculated torsional natural frequencies is shown in Figure 7. As shown in this figure, the calculated torsional natural frequencies have sufficient separation margins from the potential excitation sources during the steady state operation (i.e. at 1490 rpm of motor speed). Note that, although it may appear in Figure 7 that the diverging torsional vibration during motor acceleration (as observed in Figure 3) is caused by resonance of the gearbox intermediate shaft rotational speed with the first torsional natural frequency of 8.4 Hz at around 1300 rpm of motor speed, as will be shown in the subsequent sections, the diverging torsional vibration during motor acceleration is actually caused by instability, not resonance. This case demonstrates that, although a Campbell diagram is a useful tool in identifying potential resonance, it cannot predict the possibility of instability.

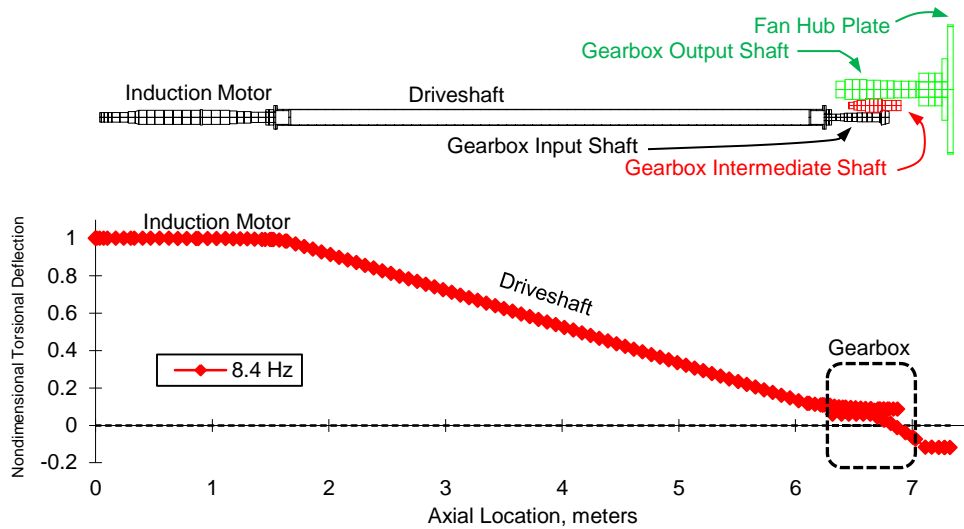


Figure 6. Torsional Analysis Model (above) and Calculated First Torsional Mode (below)

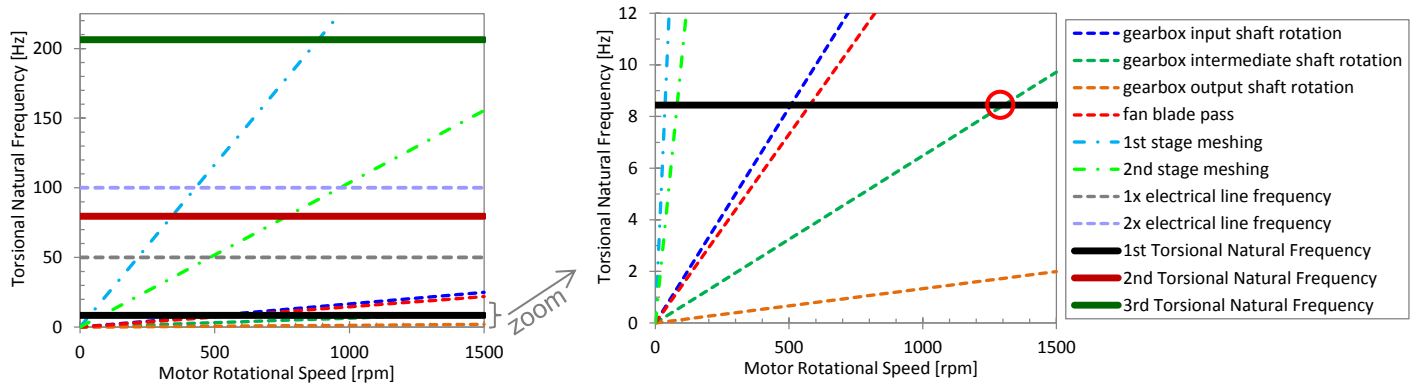


Figure 7. Campbell Diagram of Torsional Natural Frequencies

### Transient Torsional Analysis

Transient torsional analysis was carried out by applying motor output torque at the motor core and fan load torque at the fan hub plate as functions of rotational speed, and by calculating a full transient solution from standstill to steady state rotational speed. The model uses speed-torque curves. Figure 8 shows plots of torque versus speed provided by the motor and fan manufacturers. In addition to this pair of speed dependent torques, a 1.3 percent modal damping ratio for the 8.4 Hz mode was implemented as a stiffness proportional damping matrix.



The analytical results of Figure 9 show that the measured torque and speed transient simulated by the transient response analysis has remarkable fidelity compared to the measurements. It should be noted that no excitation forces were applied in the transient analysis (only motor torque and fan load torque were applied without any other forced excitations). Abrupt powering on of the motor at standstill rings the fundamental mode, which is normal. Nevertheless, the transient simulation clearly indicates torsional vibration at the rotor system's natural frequency is growing instead of decaying during acceleration. From this simulation, it became evident that the large amplitude torsional oscillations measured during startup were caused by an instability, and not resonance (i.e. self-excitation instead of forced excitation at the train's natural frequency).

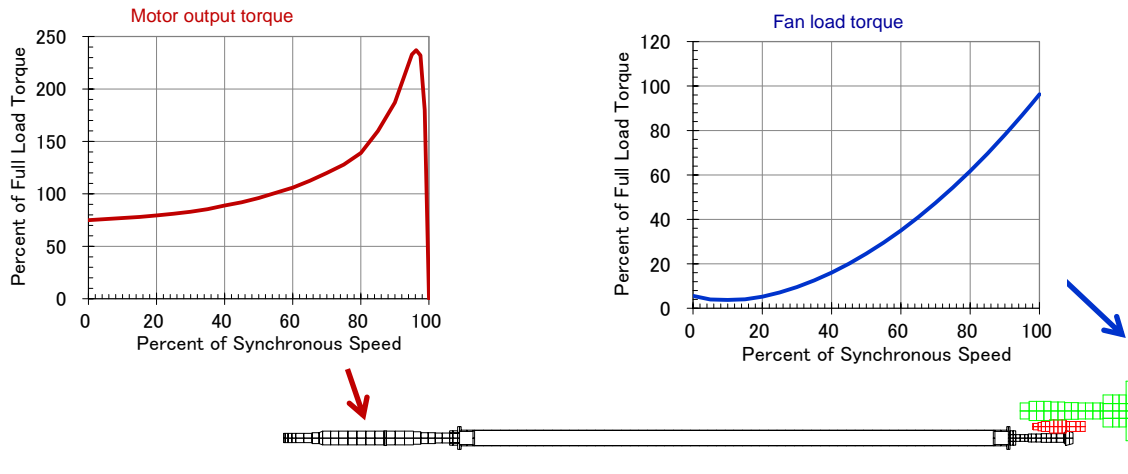


Figure 8. Transient Torsional Analysis Model

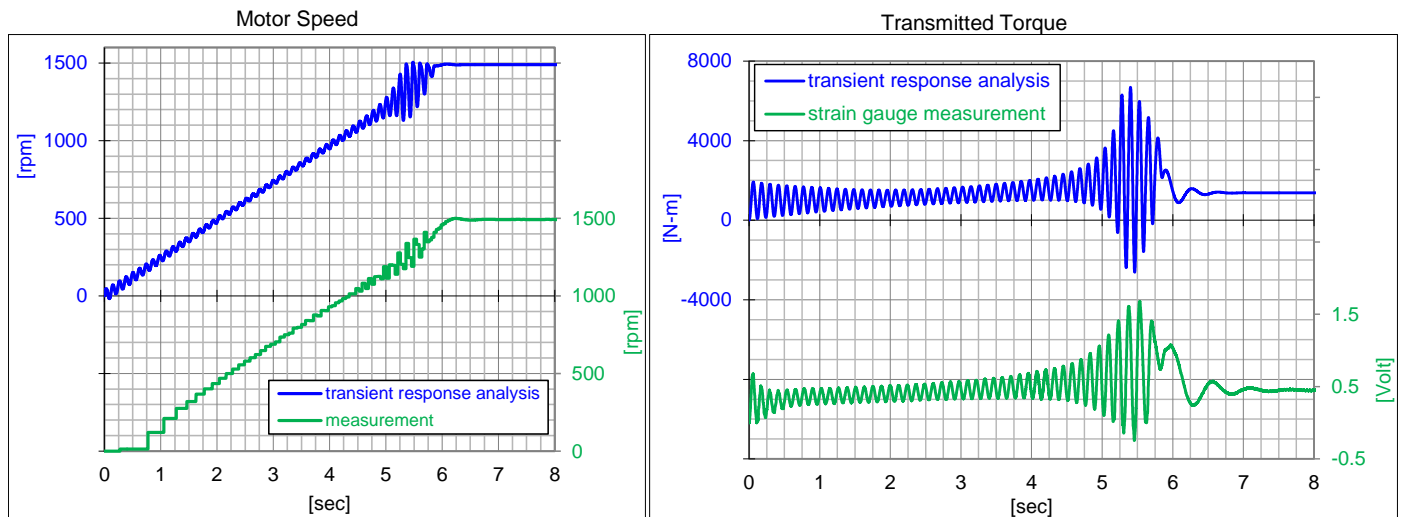


Figure 9. Comparison of Transient Response Analysis (blue) and Field Measurement (green)  
 Left: Speed; Right: Transmitted Torque

## ROOT CAUSE AND COUNTERMEASURE

A synchronous motor without soft starter generates large torque oscillations at two times the slip frequency during train startup. Because the excitation frequency decreases linearly from twice line frequency (at 0 rpm) to zero at synchronous rpm, it inevitably resonates at the train's torsional natural frequencies during acceleration. Forced vibration by such oscillations can readily attain a damaging magnitude, thereby causing failures of gearboxes, couplings or shafts. This phenomenon is widely recognized and well documented; for example, in the turbomachinery symposia, methodologies to mitigate the adversities associated with torsional





vibration during starting of a synchronous motor have been extensively discussed (Szenasi and von Nimitz 1978, Mruk 1978, Chen et al. 1983, Jackson and Leader 1983, Wachel and Szenasi 1993, Corbo and Cook 2000, Yeiser et al. 2006).

On the other hand, torsional vibration during starting of an induction motor is relatively lesser known. Nevertheless, this phenomenon has been documented in several articles (Concordia 1952, Godwin 1976, Rivin 1980, Hyde and Brinner 1986, Shadley et al. 1992, Ran et al. 1996). While torsional vibration during startup by a synchronous motor is mainly due to resonance (forced vibration at the train’s natural frequency), one of the major concerns of torsional vibration during startup by an induction motor is instability (self-excited vibration due to negative damping). In this section, mechanisms of torsional instability occurrence during starting of an induction motor in a cooling tower fan train are discussed, as well as methods of mitigation.

### Speed-Torque Curve

During acceleration of an induction motor, the driving torque developed by the motor varies as a function of motor speed. Typical speed - torque characteristics curve of a generic induction motor is shown in Figure 10. As shown in this figure, the locked rotor torque, also sometimes referred to as starting torque, is the torque delivered by a motor at 0 rpm, while pull-up torque and breakdown torque are the minimum and maximum torques that can be developed, and full load torque is the torque required to produce the rated power at full speed (NEMA 2014).

By definition, the motor startup torque curve has a positive slope in the speed range between pull-up torque and breakdown torque. In this speed range, when a small torque fluctuation occurs, motor output torque and rotational speed tend to change in the following manner:

- Increase of output torque due to fluctuation → increase of rotational speed due to more motor torque → increase of output torque due to rotational speed increase (note the positive slope of torque curve) → more and more torque and speed
- Decrease of output torque due to fluctuation → decrease of rotational speed due to less motor torque → decrease of output torque due to rotational speed decrease → less and less torque and speed

In this manner, small perturbations in torque and rotational speed become amplified by the positive slope of motor’s output torque curve. In other words, positive slope of motor torque curve can be regarded as negative damping. Conversely, negative slope of motor torque curve results in positive damping (induction motor’s speed-torque curve usually has very steep negative slope when nearing rated speed. Torsionally stabilizing effect by the slip of induction motor when at or near rated speed is attributable to this damping mechanism). On the other hand, positive slope of a load torque (i.e. the fan) is equivalent to positive damping. It can be shown that the effective damping of load torque is equal to the slope of that load torque curve (Vance 1987). For example, effective damping of the example load torque curve of Figure 11 is 1328 in-lbf-s/rad (150 N-m-s/rad) at the angular rotational speed of 8.5 rad/s (81 rpm).

As such, before reaching breakdown torque, negative effective damping is generated by an induction motor’s positive slope of its speed-torque curve, and the degree of negative damping is larger with a steeper slope.

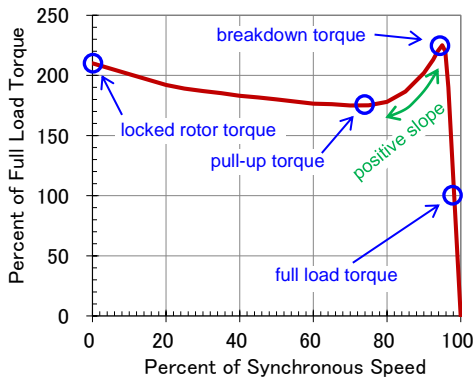


Figure 10. Typical Speed-Torque Curve (Motor)

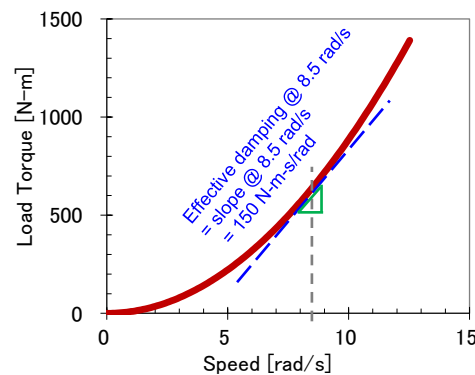


Figure 11. Effective Damping of Load Torque (Fan)

(Note: Figures 10 & 11 are typical torque curves, and they are not the curves of the equipment described in this paper.)

### Mechanism of Torsional Instability

Reexamining the torque curve of this induction motor (Figure 8), it has a small locked rotor torque (75 percent of full load torque at 0 rpm), no corresponding pull-up torque (because the locked rotor torque is the minimum torque), and a considerably greater breakdown torque than the locked rotor torque (240 percent of full load torque, which is more than three times greater than the locked



rotor torque), which all lead to a continuously positive and steep slope of the motor torque until reaching breakdown torque during its acceleration. The slope of this motor torque curve is especially steep over the speed range of 70 ~ 95 percent of synchronous speed. In the measurement results of Figure 3, during 5 ~ 6 sec after motor starting (which corresponds to the speed range with steep positive slope of the torque curve), torque and speed oscillation is at the maximum. This oscillation is caused by an instability mechanism which is essentially attributable to the steep positive slope of the motor torque curve.

In addition, as discussed in the previous section, the large moment of inertia of the fan results in its modal amplitude being only about one-eighth that of the motor. So the negative damping of the motor is at a point of large deflection, and the positive damping of the fan is at a point of small deflection. The ratio of deflections is about eight, which means the fan damping would need to be much larger than the motor damping for the two damping effects to offset each other in case no other damping mechanism exists.

It should however be noted that none of the discussed characteristics are by themselves abnormal;

- (1) Because load torque of turbomachinery (such as centrifugal blowers and pumps) usually increases proportionally to square of the rotational speed increase (square torque characteristics, see Figure 11 as an example), a big locked rotor torque is unnecessary to drive the train. Consequently, thousands of industrial induction motors with similar torque characteristics as that of Figure 8 are applied as prime movers of turbomachinery without any difficulties.
- (2) Composite driveshafts have been successfully applied in numerous cooling towers without problems. Although the driveshaft is torsionally the most flexible element of this fan train, it still has sufficient strength and endurance for this application, and no associated problems are expected due to torsional flexure (except for the issues discussed in this paper).
- (3) Large fan inertia is typical in cooling tower service because large diameter fans are needed to provide substantial amount of mechanical draft of cooling air.

It is the combination of (1) steep positive slope of induction motor, (2) torsionally flexible driveshaft spacer, and (3) large moment of inertia of fan blades, which has produced the torsional instability problem in this train.

#### *Application of Larger Diameter Driveshaft*

The root cause of tooth damage was identified as large torsional vibration during the motor startup and which causes excessive tooth loading beyond its ratings. Because a large amplitude torsional vibration was essentially due to steep slope of the motor's torque curve, the most straightforward mitigation measures would be:

- Replace the induction motor with the one having relatively flat speed-torque curve
- Add soft starter in the electrical room to modify the startup profile, and thereby flatten the speed-torque curve
- Replace the gearbox with the one having higher service factor

However, any of these mitigation measures would have necessitated significant modifications to the already completed plant arrangement. Moreover, required delivery time for the additional or replacement components was unacceptably long, which would have adversely affected the plant's operation schedule. Additionally, procurement and construction cost for any of these modification plans was significant.

Alternative mitigation measures were therefore sought. Since the primary reason of the torsional instability was due to the steep slope of the induction motor torque curve in combination with torsionally soft driveshaft and large load inertia, instead of modifying the motor torque curve, application of either (a) a torsionally stiffer driveshaft spacer, or (b) a reduced moment of inertia fan, could possibly mitigate the instability. Significantly reducing the moment of inertia of the fan blades was not practical because they were already made of lightweight composite material. On the other hand, stiffening the driveshaft spacer could be done relatively easily and quickly by replacing it with a larger diameter element.

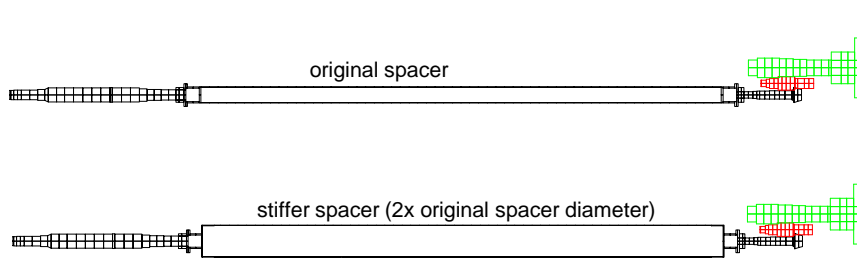
While stiffening the driveshaft spacer was essential, it was also important to keep the other components in the train unaffected because any modification to them would have required a long delivery time. It was therefore decided to increase the driveshaft spacer diameter by two times, which was the largest diameter spacer compatible with the installed hubs. In this manner, modification could be done without changing anything else in the train (Figure 12).

Prior to actually incorporating the stiffer spacer, its effectiveness was predicted by transient torsional analysis. Figure 13 shows the transient analysis result with the stiffer spacer, and the unstable oscillations have been suppressed (Figure 13, left).

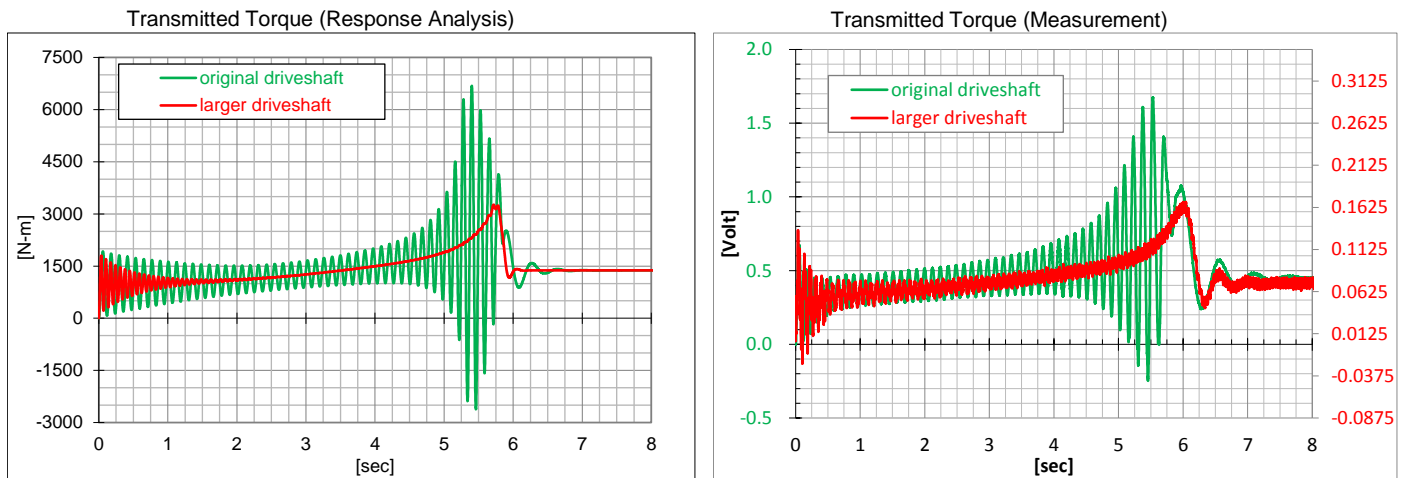
Figure 13, right, shows field measurements with the stiffer driveshaft, and confirms that the torsional instability has been suppressed. Peak transmitted torque was reduced to just 2.4 times the rated torque (at around 6 sec) with the new driveshaft, as opposed to four times in case of the original driveshaft. Transmitted torque of 2.4 times rated torque was due to the breakdown torque of the induction motor, which was inevitable unless soft starter were used.

Field measurements of electrical current and gearbox vibration also confirmed that motor current pulsation and gearbox impulsive vibration right before reaching the rated speed successfully disappeared with the larger diameter driveshaft (Figure 14 & Figure 15).

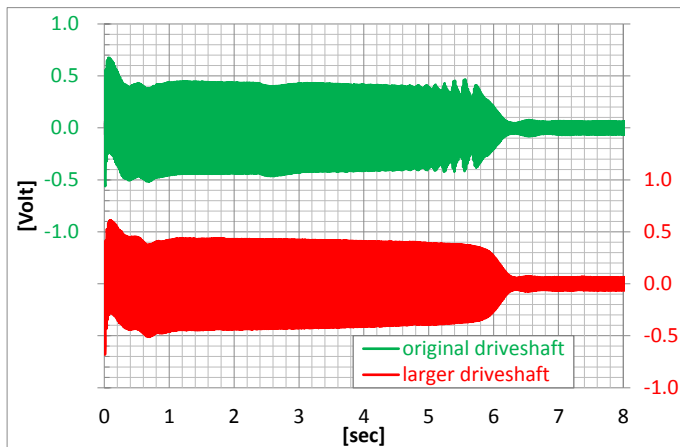
It was therefore concluded that application of larger diameter driveshaft was entirely effective in suppressing the torsional instability in this train. After implementation of this countermeasure, no other problems have been observed.



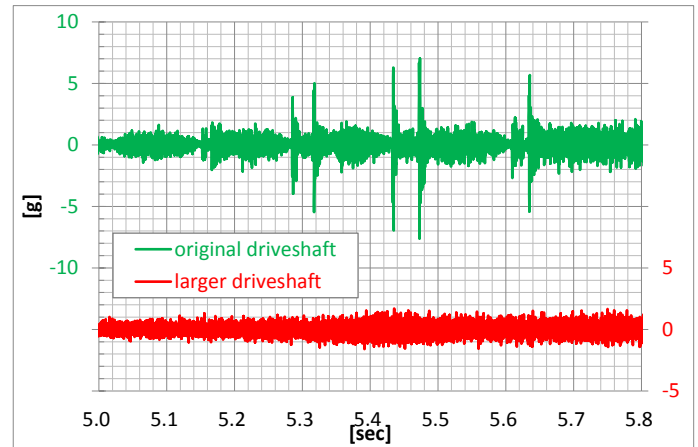
**Figure 12.** Comparison of Torsional Analysis Model (above: original driveshaft, below: larger driveshaft)



**Figure 13.** Comparison of Original and Larger Driveshafts (left: torsional analysis, right: field measurement)



**Figure 14.** Measured Motor Current



**Figure 15.** Measured Gearbox Vibration

### *Torsional Stability Analysis*

Centrifugal compressors for high pressure applications can suffer from high levels of lateral subsynchronous vibration due to an instability. Such instability is essentially attributable to destabilizing aerodynamic forces at labyrinth seals and impellers in excess of stabilizing effects of oil films at bearings, thus making the overall damping of the rotor system negative. To assess the potential for a lateral instability, it is common practice to perform a stability analysis for centrifugal compressors. In the lateral stability analysis,



speed dependent damping coefficients are defined at the bearings (which promote stability), and labyrinth seals and impellers (which promote instability). Then the log decrement for the first lateral natural frequency is calculated by a damped eigenvalue analysis at each rotational speed. Log decrement ( $\delta$ ) and damping ratio ( $\zeta \approx \delta / 2\pi$ ) are the measure of system stability. A rotor system is stable when the damping ratio is above zero; otherwise it is unstable. Larger damping ratios mean there is more margin against system instability. On the contrary, smaller damping ratio implies stronger tendencies toward instability. Hence, by calculating the damping ratio, the stability of a rotor system can be quantified. Because a higher rotational speed often results in increased likelihood of an instability, calculated damping ratios are usually plotted against rotational speed such that system's tendency toward instability can be visualized.

Torsional stability of a rotor system can be evaluated in essentially the same manner as lateral stability. As shown in Figure 11, effective damping of a load torque is equivalent to the slope of the load torque curve, and the effective damping of a motor torque is equivalent to the slope of the motor torque curve multiplied by -1 (because it is opposite of load torque). Hence, effective damping due to fan and motor torque curves can be derived by calculating the slope of torque curves at incremental values of rotational speed (Figure 16 & Figure 17). By applying these speed dependent damping coefficients at the locations of motor core and fan hub plate, the damping ratio for the first torsional natural frequency can be calculated at each rotational speed by damped torsional eigenvalue analysis.

Figure 18 is a plot of the torsional damping ratio. With the original driveshaft, the damping ratio  $\zeta$  becomes negative (i.e. unstable) above 600 rpm motor speed, and peaks at 1420 rpm with a damping ratio of -0.17. In contrast, with the stiffer driveshaft the rotor system is stable up to 1150 rpm, and its peak negative value is down to -0.07. From this torsional stability analysis, it is clearly shown that the larger diameter driveshaft indeed contributes to stabilize the rotor system against torsional vibration. Greater stability by the stiffer driveshaft is essentially attributable to the increased elastic twist throughout the drive train instead of being concentrated in the driveshaft (i.e. other parts of the rotor system also participate to dissipate the energy in case of the stiffer driveshaft via the 1.3% proportional damping matrix described earlier). Dissipation by fan load torque was found to be negligible. See Figure 19 for the comparison of the mode shapes.

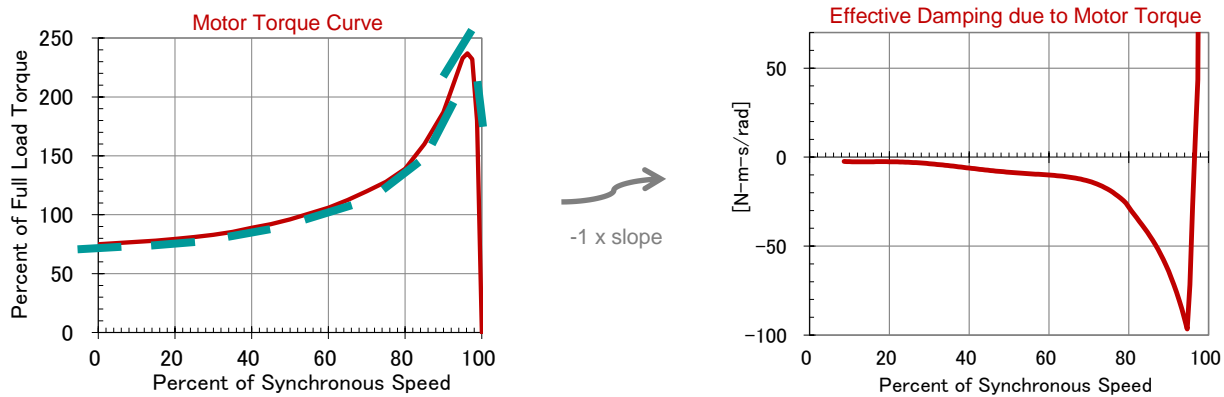


Figure 16. Derivation of Effective Damping from Slope of Motor Torque Curve

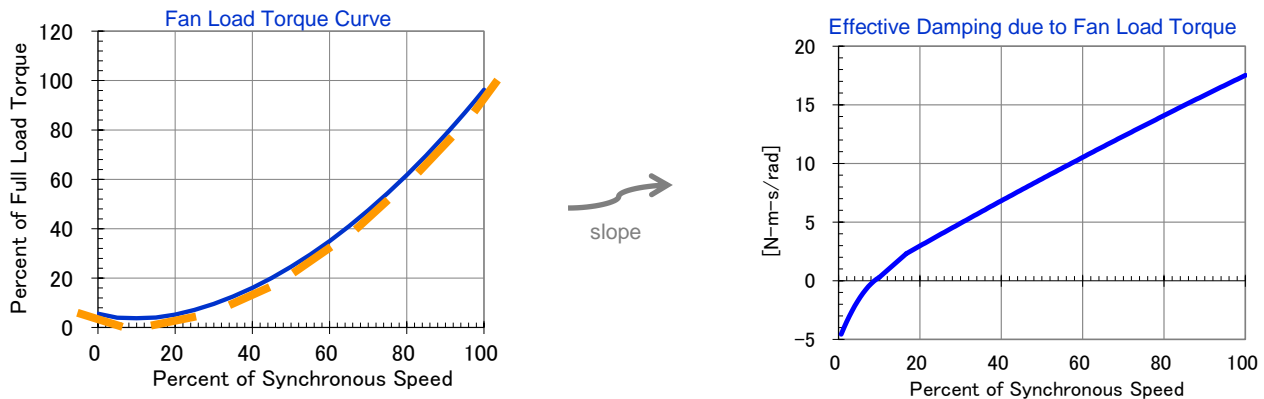
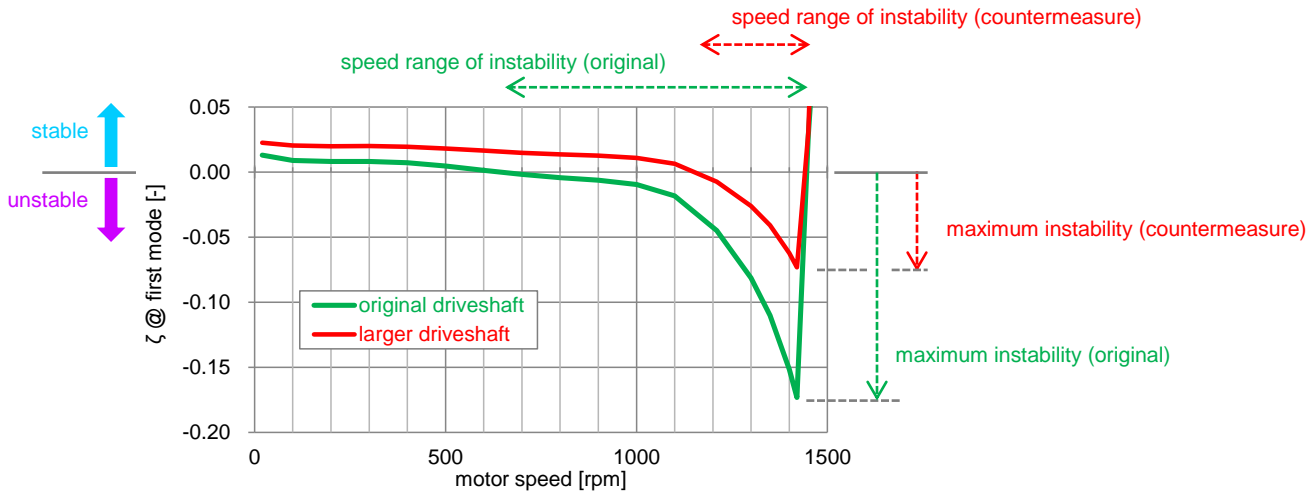
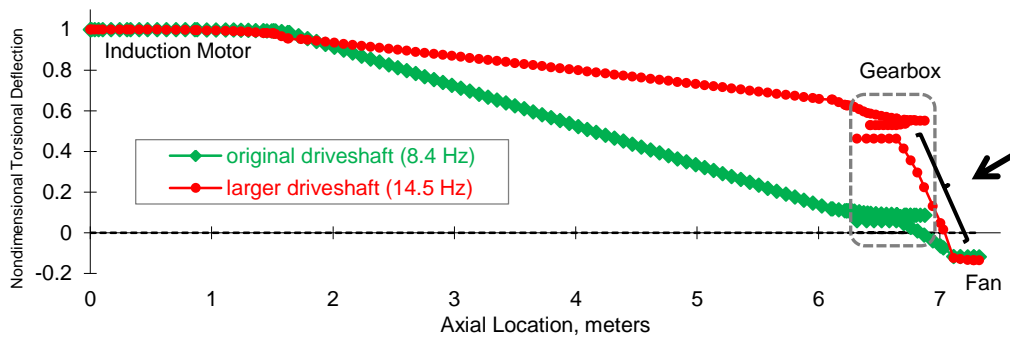


Figure 17. Derivation of Effective Damping from Slope of Fan Torque Curve



**Figure 18.** Calculated Torsional Damping Ratio  $\zeta$  (green: original, red: countermeasure)



**Figure 19.** Comparison of Mode Shapes (green: original, red: countermeasure)

## LESSONS LEARNED

In the process of troubleshooting activities, many lessons have been learned. In this section, among many such lessons, those which are most relevant to engineering of cooling tower fans are discussed.

### Motor Selection Guideline

The root cause of gearbox teeth damage is attributable to a torsional instability during motor startup, which is essentially due to steep slope of the induction motor torque curve. Hence, in order to prevent instability occurrence, unless a soft starter is equipped, induction motors for cooling tower fan trains should be carefully selected to avoid their steep slope of torque curve. In other words, motors with relatively flat torque curve are preferable. To ensure motor selection with a flat torque curve, it is important to keep the locked rotor torque above a certain level.

On the other hand, possibilities of gearbox damage due to excessive locked rotor torque can also be problematic (Drago 2007). If the locked rotor torque far exceeds the full load torque, transmitted torque during motor's initial acceleration (i.e. at 0 rpm) may exceed the maximum sustainable load of the gearbox tooth, resulting in tooth surface damage. It would therefore be preferable to keep the locked rotor torque below certain level as well in order to surely prevent the damage occurrence by such mechanism.

To examine the influence of motor torque curve onto the transmitted torque, a case study analysis is carried out by transient analysis with various motor torque curve profiles. In this case study, everything including the rotor geometry, inertias and fan load torque are kept identical to the original configuration except for the motor output torque curve.

Figure 20 is the result of this case study. When the locked rotor torque is increased to 128 percent of full load torque (Motor (b)) from 75 percent in the Base Case, the instability occurrence becomes milder, because of milder slope of the torque curve. When it is increased to 206 percent (Motor (c)), the instability is almost completely suppressed. On the other hand, because of a bigger locked



rotor torque, initial torque right after starting (near 0 sec) becomes bigger (around three times the rated torque). And, when it is 270 percent (Motor (d)), instability completely disappears, but the initial transmitted torque near 0 sec becomes significantly bigger.

Note that, as the locked rotor torque increases, the time required to reach the rated speed reduces because of the quicker acceleration due to more output torque of the motor (e.g. it takes three seconds by Motor (d) to reach the rated speed as opposed to 6.5 sec by Motor (a)). Consequently, a larger number of cycles is experienced during motor acceleration with Base Case (Motor (a)).

It is therefore proposed that, for the cooling tower fan application, aiming to select a motor with locked rotor torque around 200 percent of the full load torque might be recommendable from the standpoint of limiting the peak torque during motor acceleration.

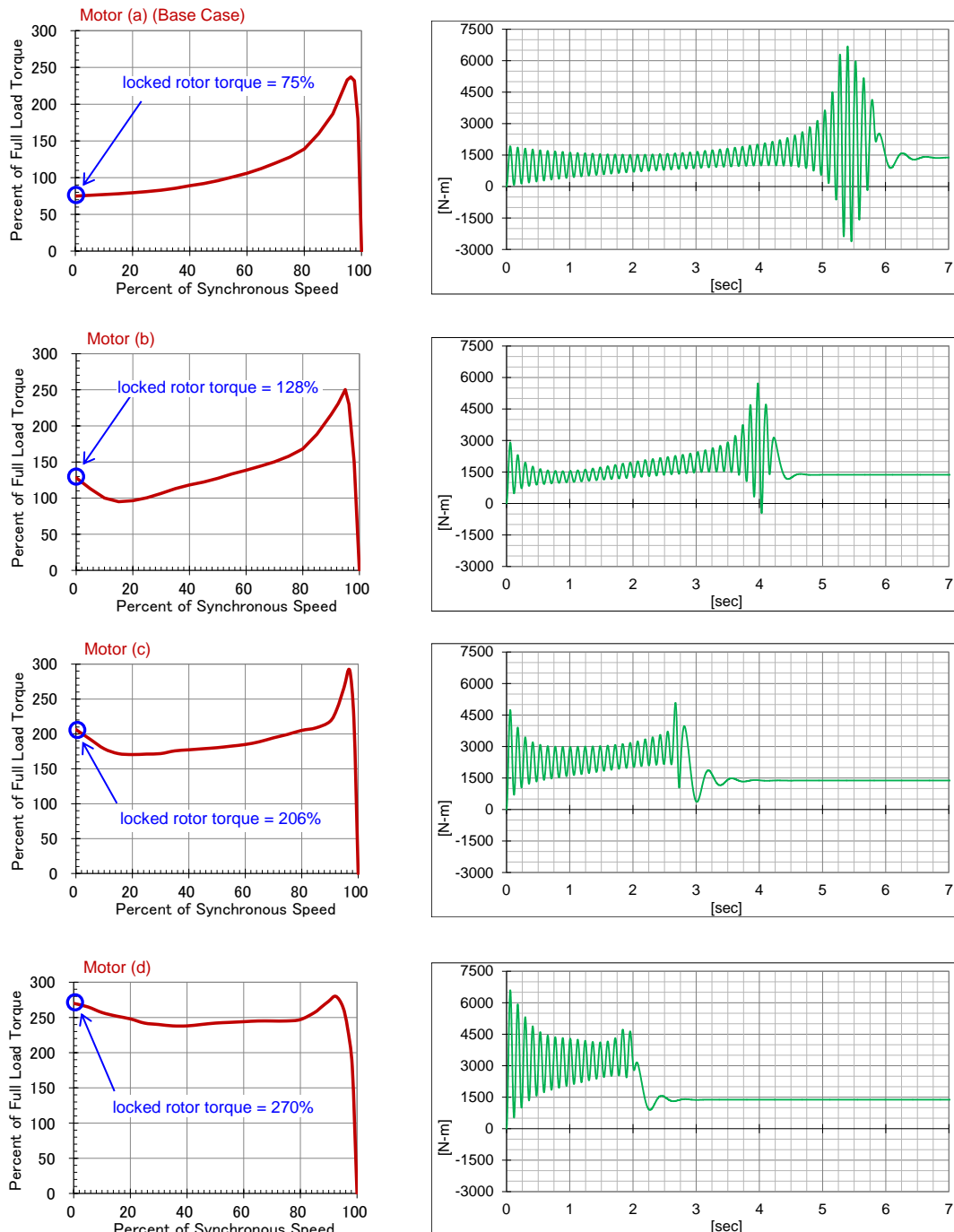


Figure 20. Various Motor Torque Curves (left) and Calculated Transmitted Torque Predictions (right)



### *Gearbox Service Factor*

Gearboxes of cooling tower fans are usually designed to satisfy the requirements of CTI Std-111. As per this standard, the minimum service factor of 2.0 is required. However, during acceleration of the induction motor, a momentary load far beyond this factor can occur as discussed in this paper.

Moreover, as per CTI Std-111, the gearbox service factor is referenced to the motor nameplate rating, not to the required load. Because service factor is referenced in this way (which helps to ensure design allowances), depending on the power margin of the selected motor against the required load, the gearbox service factor can be different even when everything else is identical in the train except for the motor nameplate power rating. For example, in the following two cases, gearbox service factors are different by factor of 1.14 (= 335 hp / 295 hp) even if identical gears are applied for exactly the same service:

- Fan required power is 261 hp (195 kW), Motor nameplate power rating = 335 hp (250 kW)
- Fan required power is 261 hp (195 kW), Motor nameplate power rating = 295 hp (220 kW)

As such, simply ensuring the gearbox selection with bigger service factor than that found in the reference list may be insufficient. In comparing the gearbox service factor to that in the reference list, careful attention should be paid to (1) the possibility of difference in the actual momentary load factor, and (2) the possibility of difference in power margins of the selected motors.

### *Usefulness of Torsional Analysis*

Transient torsional analysis proved to be a powerful and useful tool to examine the transient behavior of the train torsional flexure during startup. Torsional stability analysis is also shown to be a powerful tool to examine the likelihood of instability occurrence in the train.

It may be a good practice to carry out torsional analysis for a cooling tower fan train under the following situations:

- Gearbox service factor is smaller than or close to the lower limit of the reference gearboxes.
- Induction motor with very steep slope torque curve is applied without soft starter.
- Cooling tower fan train is suffering from anomalous behavior.

## **CONCLUSIONS**

Gearbox pinion damage in 14 of 16 cooling tower fan trains was shown to have been caused by a torsional instability of the train during startup. Field measurement by strain gauges exhibited excessive torque oscillation as much as four times the rated torque. Moreover, impulsive vibrations at the gearbox confirmed occurrence of shock load at the moment of pinion teeth recontacting of the separated teeth during torque reversal.

The torsional instability was due to the combination of a steep slope of the induction motor's speed-torque curve, a torsionally soft driveshaft spacer, and a fan inertia much greater than the motor inertia. Transient response torsional analysis successfully replicated the measured phenomena with remarkable fidelity. In addition, torsional stability analysis also predicted the presence of the instability, and its mitigation via a stiffer driveshaft. As a countermeasure to overcome the instability, a stiffer driveshaft was installed in each field unit, and confirmed by measurements to have resolved the problem.

Additionally, as a guideline, motor selection with a locked rotor torque of around 200 percent of the full load torque is proposed in order to limit the peak transmitted torque during startup. Also discussed is the importance of considering the effect of the motor nameplate power rating margin on the gearbox service factor.

## **NOMENCLATURE**

- $\delta$  = Logarithmic Decrement (-)  
 $\zeta$  = Damping Ratio ( $\approx \delta / 2\pi$ ) (-)

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