Shredder Torque Testing Report

Quad Plus Recycling

Scope and Objectives

- To further evaluate the torsional oscillations occurring in the shredder a data acquisition system (DAQ) was implemented to collect strain data and motor control data.
- Strain measurement equipment was installed on both inter-motor shafts.
  - On each shaft, four foil-type strain gauges were applied and configured in a full bridge to measure shear strain produced by external torques on the shaft.
  - Using the measured strain, we calculate the load present on the system and analyze the collected signals' waveforms to better understand the sequencing of the captured events in the system.
- Both inboard and outboard drives were programmed to output the following motor control data:
  - Speed
  - Current
  - DC voltage
  - AC voltage
- Several tests were performed:
  - Normal Shredding was monitored.
  - Ramp-Up and ramp-down runs with varying ramp rates.
  - Ramp-Up with an E-Stop initiated coast down.
- The objective was to witness and measure the ongoing torsional oscillation events and compile both accurately scaled drive data and strain measurement data for a complete picture of system performance.
- A modal analysis of the Motor-Motor-Driveshaft-Rotor assembly was created to determine the 1st and 2nd order natural frequencies of the system and is compared to the physical data presented in this report.

Executive Summary

- Torsional oscillations were detected throughout the operating range of the shredder at several different frequencies.
  - A 12.5Hz oscillation was detected throughout shredding operation.
    - This frequency is the 1st order natural frequency of the combined motor-motor-driveshaft-rotor system.
    - At this frequency, the system exhibits the mode 1 oscillation pattern.
    - Both gradual-growth and rapid-growth resonant events were detected, indicating an excitement through large-scale acoustical noise and impact events within the mill.
  - A 31.2Hz oscillation was detected throughout shredding operation.
    - This frequency is the 2nd order natural frequency of the combined motor-motor-driveshaft-rotor system.
    - At this frequency, the system exhibits the mode 2 oscillation pattern.
    - Present throughout the operation of the shredder, however is drown out during large scale oscillation of the 1st order.
  - During ramp-up and ramp-down, a gradual-growth resonance with frequency, f =2x(running frequency) was detected from 275-425 RPM.
    - The frequency of this resonant event is equal to two times the running frequency of the motors.
    - The cause of this oscillation unknown, however, the frequency has been shown to be f =2x(running frequency).
    - Regardless of the cause, it is another example of the under-damped system response.
  - A 10.4Hz oscillation is present in the noise floor during motor idle.
    - This oscillation is caused by an imbalanced rotor.
    - The effect of gravity on the rotor's center of mass creates an additive force for 180 degrees of the shaft's rotation (as the heavier side is lifted from bottom to top) and a subtractive force for the other 180 degrees (as the heavier side falls from top to bottom).
  - With few exceptions, torsional resonant events will only occur when the motor are being driven.
    - A loaded shaft is much more susceptible to resonance.
    - Increasing motor torque effectively increases motor mass in the mathematical model.
    - It would take a very high energy waveform to excite the unloaded system, something unlikely to come about during an unloaded, coasting state.
- The current control signals exhibited induced oscillations at two frequencies.
  - Both current waveforms exhibited the 12.5Hz oscillation during large scale torsional oscillations.
    - The measured current waveform lags the torque waveform seen on the shaft by 45-90 degrees.
    - This phase lag indicates that this current oscillation is a reaction to the 12.5Hz torque oscillation rather than the cause of it.
    - This is an expected response to the oscillations as torque on the shaft increases, the controller calls for more current; as the torque decreases, the controller calls for less current.
  - The inboard motor current also exhibited the 31.2Hz oscillation regularly.
    - This signal also lagged the torque oscillations by 45-90 degrees.
    - Only the inboard motor current exhibits this frequency because of its position in the system with respect to the mode 2 oscillation.
    - When the system oscillates in mode 2, the inboard motor experiences a condition similar to armature reaction which accounts for the diverging motor currents during extreme oscillations.
    - The maximum current fluctuation equates to 0.89% the torque caused by the 1st order oscillations.

Executive Summary Continued

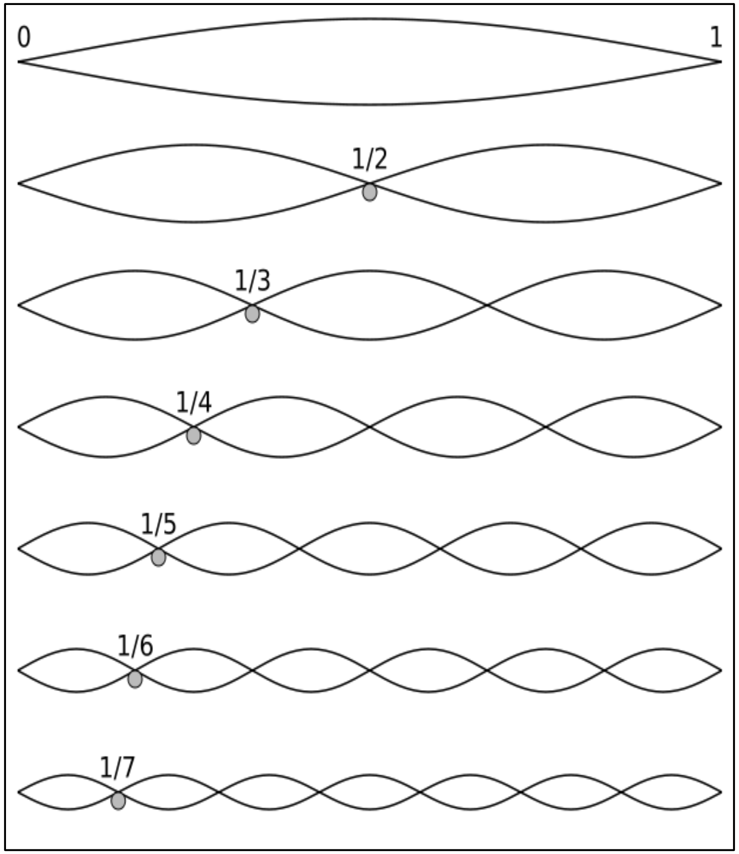
- The peak recorded magnitude of each frequency are as follows:
  - The 1st order torsional oscillations at 12.5Hz produced the greatest forces.
    - The peak torque recorded at this frequency was 100,900ft-lb.
    - The oscillations regularly unload the shaft to 0ft-lb, with instances of torque reversals caused by spring back in the shaft.
    - The largest full torque swing recorded was 115,940lb-ft, from 100,600 to -15,340ft-lb.
  - The 2nd order torsional oscillations at 31.2Hz also produced forces capable of damaging the system.
    - The peak torque recorded at this frequency was 78,720ft-lb
  - The f = 2x(running frequency) is only present during ramp-up and ramp-down.
    - Peak torque oscillations are dependent on load on the shaft, so are therefore dependant on ramp rate.
    - For a one-minute ramp, the peak torque recorded was 16,620lb-ft.
    - The oscillations regularly unload the shaft to 0ft-lb, with instances of torque reversals caused by spring back in the shaft.
  - The torsional oscillations, which exhibited magnitudes in excess of 70,000lb-ft prior to any current fluctuation, induced current oscillations of the same frequency.
    - The induced 12.5Hz current oscillation resulted in a peak variance of 3,100lb-ft in the torque output of each motor.
      - The induced current fluctuation is 2.67% the torque caused by the mechanical resonance.
    - The induced 31.2Hz current oscillation resulted in a peak variance of 1,033lb-ft in the torque output of each motor.
      - The induced current fluctuation is 0.89% the torque caused by the mechanical resonance.
      - This only occurs on the inboard motor because of mode 2 resonance causing armature reaction.
    - The current fluctuations, which begin after large-scale torsional resonance is present, are caused by the armature oscillating within the stator. This oscillation produces a condition similar to armature reaction where the armature is being pulled in and out of the neutral plane as the system vibrates.
    - The current fluctuations are not present prior to large scale torsional events, indicating the resonance is independent of motor behavior.

Executive Summary Continued

- There are three primary mechanisms occurring which excite the various frequency.
  - Impact events inside the mill include but are not limited to: "unshreddables" , "heavies", or simply a log or bale. These impact events cause rapid-growth resonance. This is also known as a step-input or an impulse.
    - Results in 1st and 2nd order natural frequency oscillations, 12.5Hz. and 31.2Hz.
    - These impacts act as step-inputs, causing an effective wind-up of energy in the system which then releases and ultimately decays back to equilibrium.
    - The large-scale torsional oscillations exhibit growth and decay curves consistent with an under-damped step-response.
    - Examples of other spring-mass-damper systems' step-response curves have been provided for comparison purposes. The data collected clearly exhibits a step-response in the system.
  - Large-scale acoustical noise produced in the shredder during operation causes gradual-growth resonance.
    - Results in 1st and 2nd order natural frequency oscillations, 12.5Hz. and 31.2Hz.
    - Gradual-growth resonance occurs when various waveforms interfere constructively, amplifying each other and ultimately exciting the natural frequency of the system.
    - The under-damped system suffers from high resonance transmissibility.
    - A mechanical system with loose components, loose bolts, worn or out of tolerance components, etc will increase resonance transmissibility exacerbating any underlying 1st and 2nd order natural frequencies.
  - Torsional oscillations were also caused by rotor imbalance. During ramp up and ramp down, motor running speed or multiples of motor running speed.
    - This results in the 10.4Hz frequency buried in the noise floor.
    - The effect of gravity on the rotor's center of mass creates an additive force for 180 degrees of the shafts rotation (as the heavier side moves from bottom to top) and a subtractive force for the other 180 degrees (as the heavier side moves from top to bottom).
  - The mechanism by which the f = 2x(running frequency) oscillation is produced during ramp-up and ramp-down is unknown, however, the relation has been proven in the data.
    - Regardless, it clearly demonstrates the under-damped response of the system.

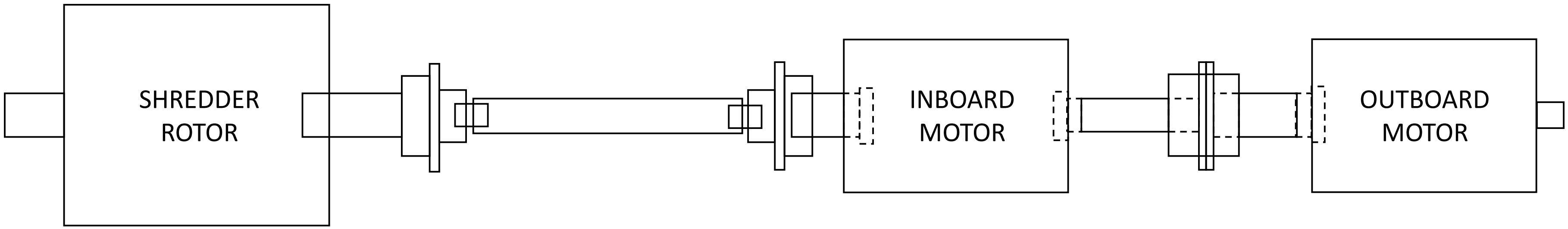
Natural Frequencies and Modes

- Every mechanical device has a natural frequency and modes of oscillation.
  - When designing mechanical systems, a full modal analysis should be performed to determine the natural frequency and the frequency of each of its overtones.
  - Any mechanical changes in the system ought to be modeled to ensure it maintains the design tolerances described in the full modal analysis of the mechanical system.
  - It is very important to know where these frequencies are in order to avoid exciting them during operation.
  - Examples of natural frequency can be seen everywhere.
    - An acoustic tuning fork is designed with a particular natural frequency in mind.
    - A child on a swing is a system that will oscillate at particular frequency. The person pushing her is an example of constructive interference.
- When exciting a natural frequency a system device will begin to resonate.
  - When an underdamped systems natural frequency is excited, resonant effects can produce forces 10-20 times greater than those produced by the system.
  - Resonance in mechanical systems occurs in characteristic patterns known as modes.
    - The simple example of a resonating string is shown on the right.
    - The oscillation mode is correlated to the natural frequency order.
    - 1st order natural frequencies resonate in mode 1, 2nd order natural frequencies resonate in mode 2, and so on.
  - Examples of resonance can be seen everywhere.
    - An acoustic tuning fork when struck with a step input resonates to produce an audible pitch.
    - Wheel-hop on an automobile doing a burnout is an example of resonance; the rubber sliding against pavement creates noise which excites the natural frequency of the suspension system.



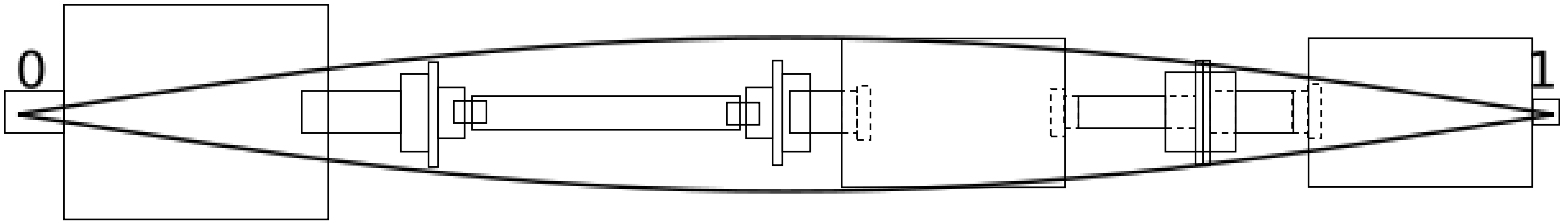
A string resonating at its fundamental frequency and its first six overtones.

Natural Frequencies and Modes



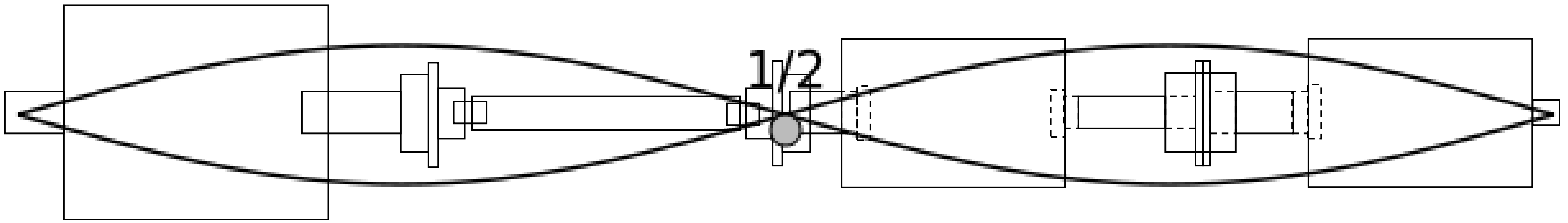
The Mode 1 Oscillation Pattern

- Mode 1 is the pattern in which the system oscillates at its 1st natural frequency.
- The torsional deflection is unidirectional across the length of the driveshaft .
- The system deflects as if one side is ground while the other is rotated.



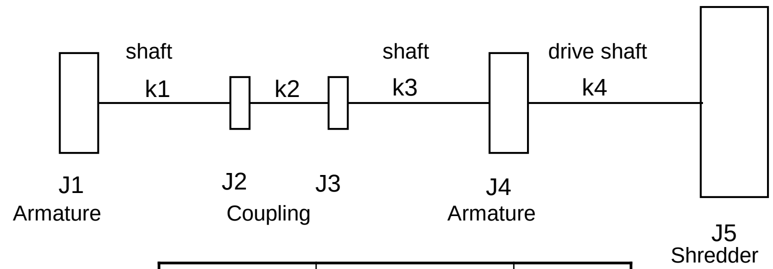
The Mode 2 Oscillation Pattern

- Mode 2 is the pattern in which the system oscillates at its 2st natural frequency.
- The torsional deflection is bi-directional across the length of the driveshaft with a portion being positive and a portion being negative.
- The system deflects as if both sides are ground while the middle is rotated.



Natural Frequencies and Modes

- A mathematical model of the motor-motor-driveshaft-rotor has been developed to verify the observed frequencies.
- The model calculates and accounts for the spring constants and polar moments of inertia of the following components:
  - Outboard armature
  - Outboard armature drive shaft
  - Inter motor coupling
  - Inboard armature extended rear shaft
  - Inboard armature
  - Inboard armature driveshaft
  - Main driveshaft
  - Rotor shaft
  - Rotor
- System natural frequencies and modes were derived using two methods.
  - Hartog Method
  - Holzer Method
- Certain estimations were made concerning the armature polar moment of inertia and spring constant.
- The calculated values for the 1st and 2nd order natural frequencies, 11.75Hz and 34.268Hz, verify that the observed 12.5Hz and 31.2Hz oscillations in the system are its true natural frequencies.
- The calculated mode 1 and mode 2 oscillation patterns have been plotted on the following slides.



J1 =	5464	lbf-in-s^2
J2 =	26	lbf-in-s^2
J3 =	20	lbf-in-s^2
J4 =	5464	lbf-in-s^2
J5 =	69372	lbf-in-s^2
k1 =	243,353,403	lb-in/rad
k2 =	1,083,000,000	lb-in/rad
k3 =	243,480,501	lb-in/rad
k4 =	57,600,320	lb-in/rad

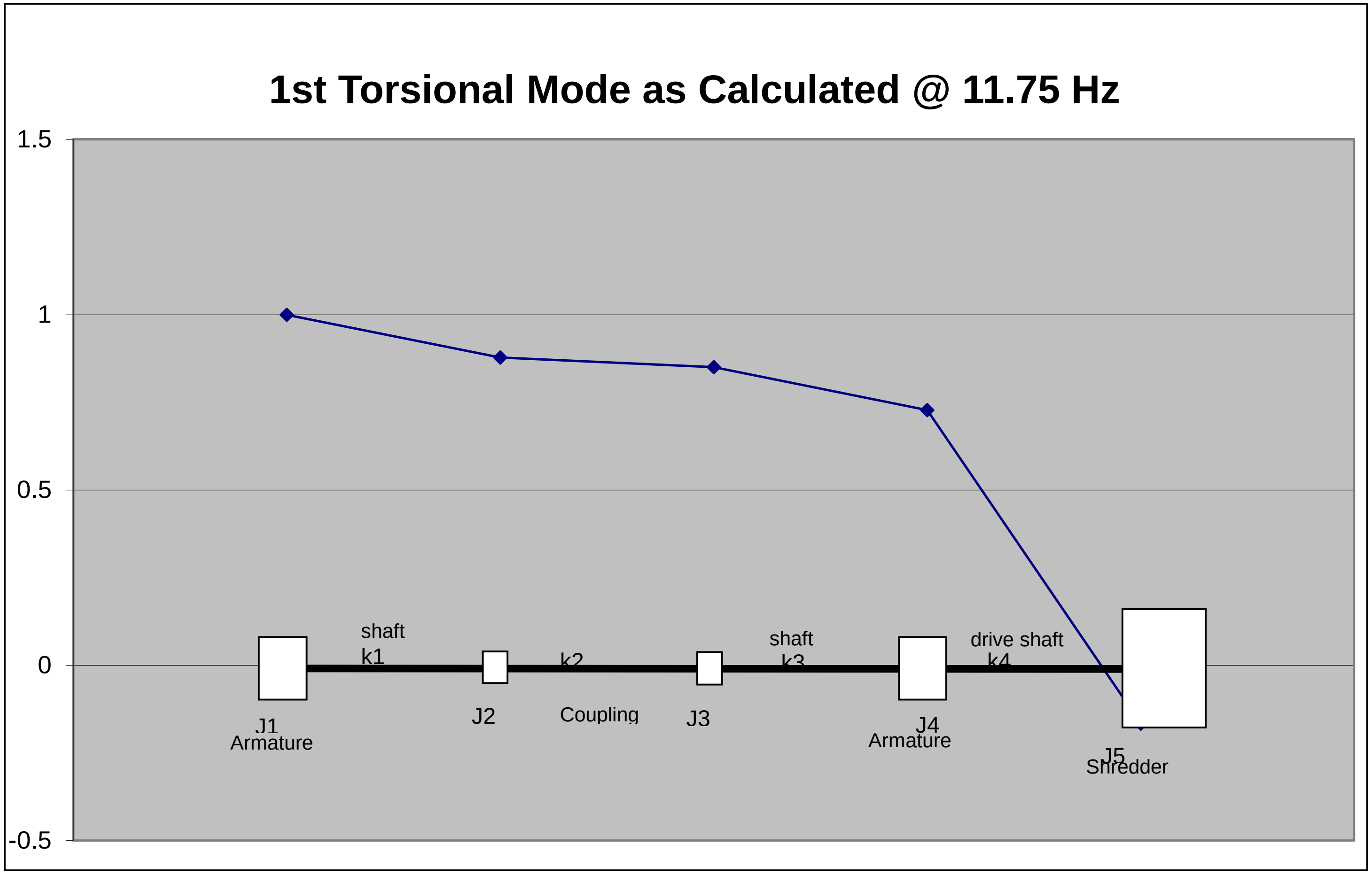
Using Hartog Method:

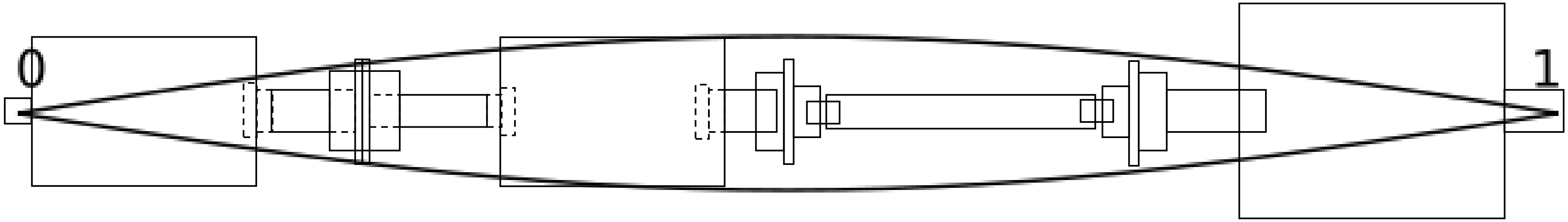
First mode ω_{n1} (Hz)	First mode ω_{n1} (cpm)	Second mode ω_{n2} (Hz)	Second mode ω_{n2} (cpm)
11.75	705.03	34.268	2053.61

Using Holzer Method:

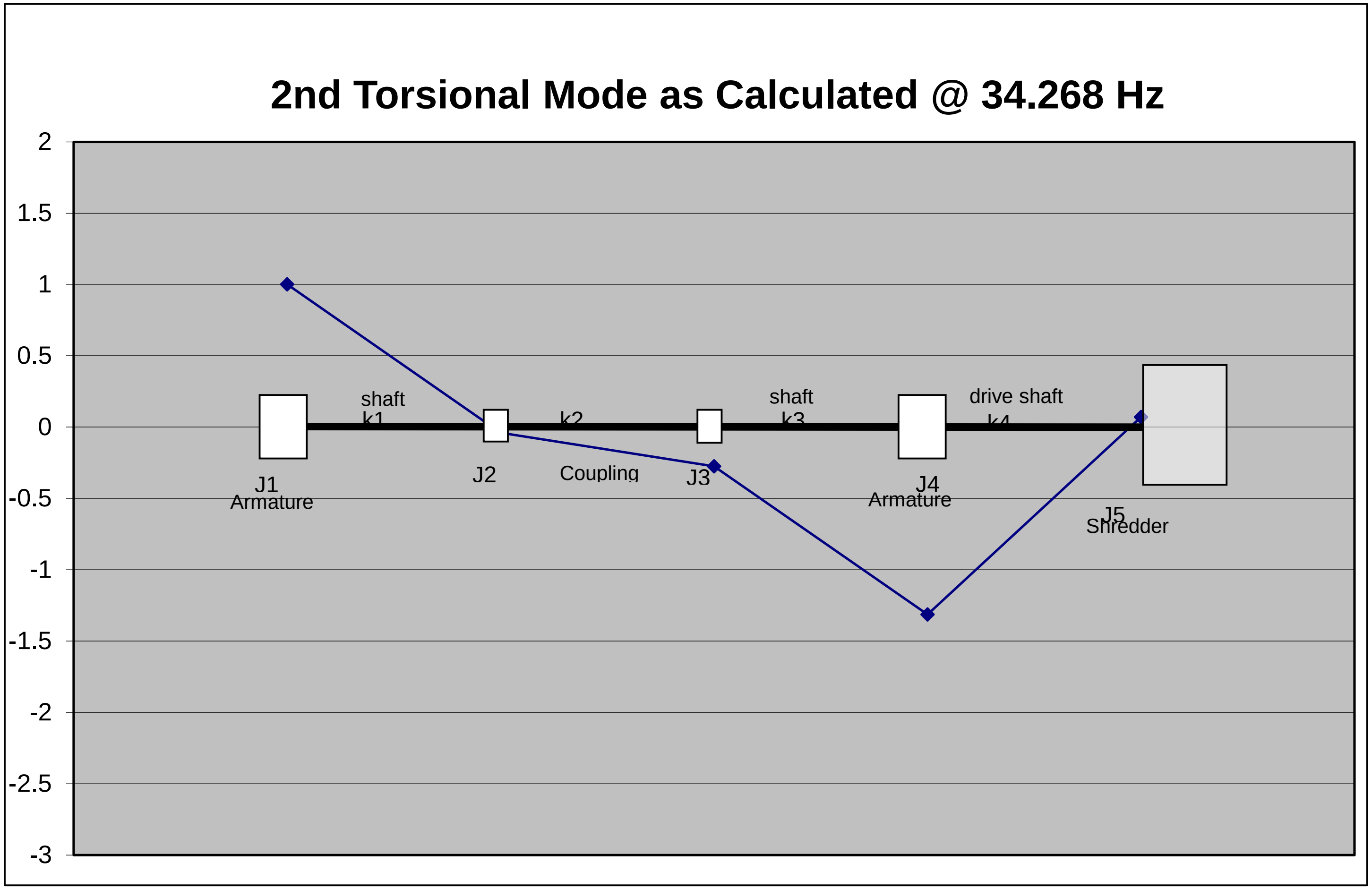
First mode ω_{n1} (Hz)	First mode ω_{n1} (cpm)	Second mode ω_{n2} (Hz)	Second mode ω_{n2} (cpm)
11.75	705.03	34.268	2053.61

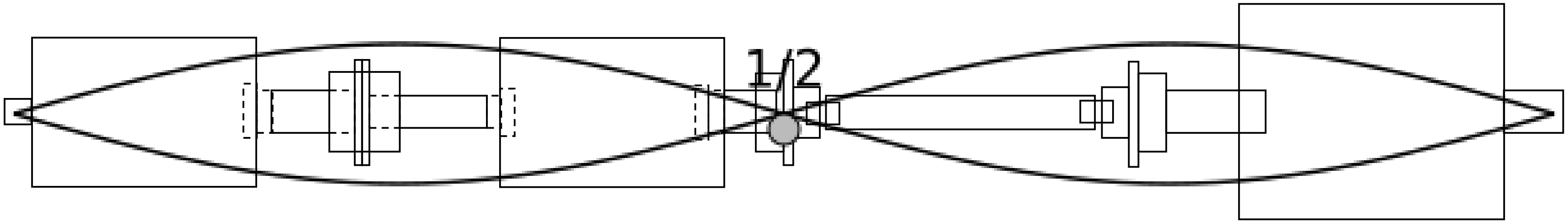
Natural Frequencies and Modes





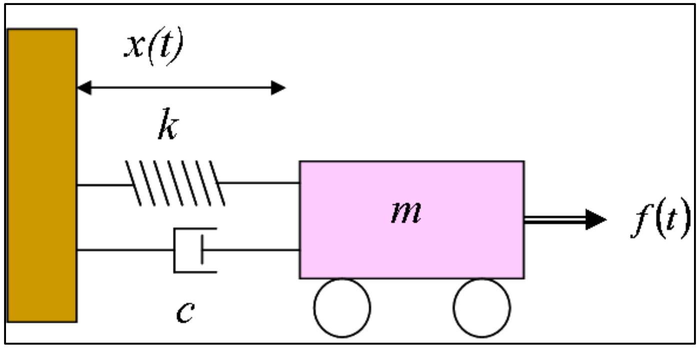
Natural Frequencies and Modes





Signal Analysis

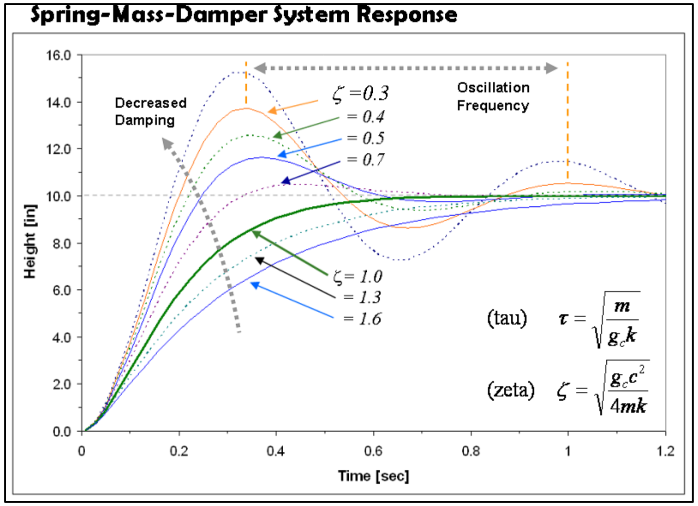
- Signal analysis is the close examination and study of sinusoidal signals and their associated waveforms to provide a clearer understanding of the events taking place in the system.
- Oscillatory signals are present everywhere in nature and are a fundamental part of mechanical design.
  - When examining mechanical systems, engineers regularly use the spring-mass-damper model to better understand the kinematics of the system.
  - A systems natural frequencies and modes are found through the mathematical differentiation of this model.
  - A diagram of the spring-mass-damper model is pictured right.
  - The spring-mass-damper model accounts for the three primary forces in a mechanical system.
    - Spring force is the 1st order force, equal to spring constant, k, multiplied by displacement, x.
    - Damping force is the 2ndorder force, equal to damping coefficient, b, multiplied by velocity, v.
    - Mass, m, multiplied by the objects acceleration, a, is the 3rd order force.
  - Examples of spring-mass-damper systems can be seen everyday.
    - An automotive suspension assembly is a perfect example of a spring mass damper system. The spring provides the 1st order force, the shock provides the 2nd order force, and the unsprung mass of the assembly provides the 3rd order force when accelerated.
- In the shredder, the driveshaft acts as the spring, frictional forces, foundation design and other internal devices provide the damping, and the mass is the total weight of the rotating assembly.
  - It is important to note that an increase in current through the motors effectively increases their mass in this model.
  - It is for this reason we only see resonance on a loaded shaft.



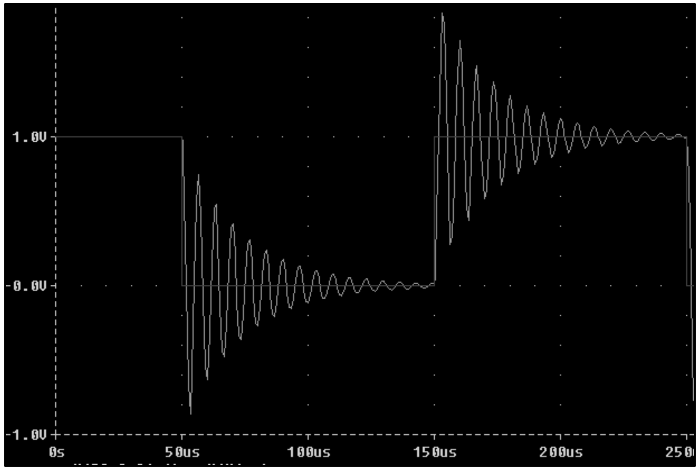
The linear interpretation of the spring mass damper model; the rotational analog is used to model the shredder system.

Signal Analysis

- With an understanding of the kinematic model, let us examine the model's response to a given input.
- Because mass and material elasticity are for the most part pre-determined when designing a system, it is difficult to design around these variables.
  - Engineers are then left with one variable to control the system.
  - Through strategic implementation of damping it is possible to reduce the transmissibility of resonance.
- Systems are either under-damped, critically damped or over-damped.
  - An under-damped system, when stuck by a step input will overshoot the step height and oscillate back and forth with gradual-decay until the system settles at the step height.
  - An over damped system, when struck by a step input will gradually approach the step height with zero oscillation.
  - A critically damped system is best and will slightly overshoot the step and settle almost immediately.
- The graphic to the right depicts a system of constant mass and spring rate responding to a step input with different values of damping
  - Low coefficients of damping yield wild oscillating responses which ultimately decay to the step height.
  - High coefficients of damping yield a slow growth response which does not overshoot the step.
- The shredder is exhibiting dramatically under-damped resonance.
  - As the mass and spring constants cannot be significantly altered, damping must be implemented to reduce the systems response.
  - Many factors contribute to the damping coefficient
    - Internal friction in the system increases damping.
    - Properly torqued hardware ensures proper mechanical connections which have low resonance transmissibility.
    - Any mechanical compliance reduces damping and increases resonance transmissibility.
- Electrical RLC circuits also exhibit this phenomenon.
  - In the graphic below, and under-damped RLC circuit responds to a step response.
  - Notice the waveform envelope.



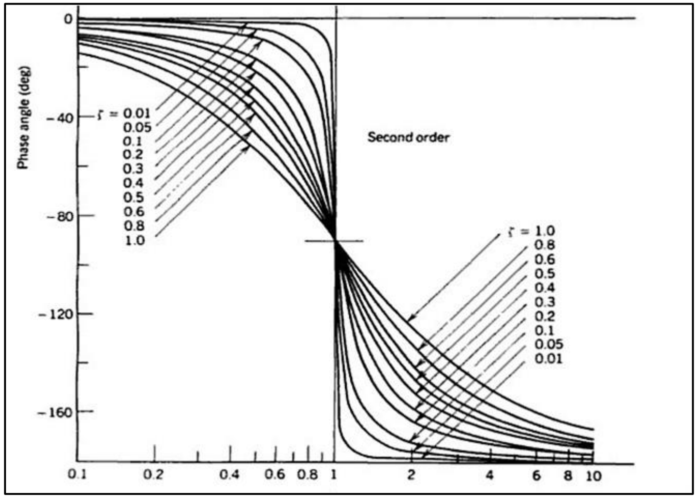
The step-response of a spring-mass-damper system with varying coefficients of damping; the shredder exhibits under-damped step-responses through out its operation.

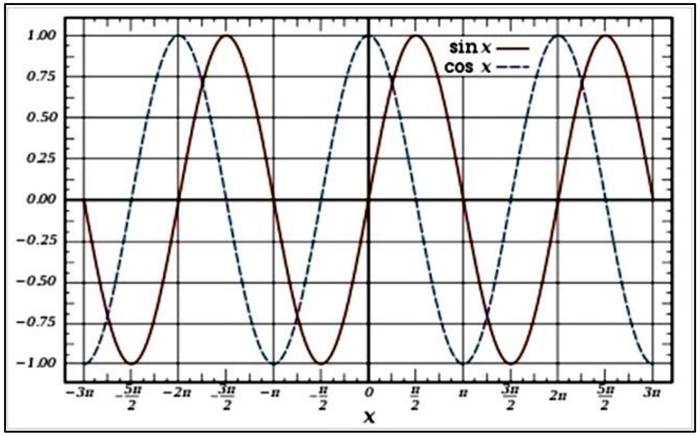


The step-response of an under-damped RLC circuit; the waveform envelope is similar to those seen in the shredder data

Signal Analysis

- With an understanding of a spring-mass-damper system's resonant response, we examine the waveforms of the signals produced during these events to identify the driving and driven waveforms.
- Examining the phase relation of two or more waveforms allows us to construct a picture of the sequencing of their events.
  - As a resonating system approaches its 1st order natural frequency resonance, the driven signal will approach a 90 degree phase lag with respect to the driving signal.
  - As seen in the graphic above, at resonant frequency, a value of 1 on the X-axis, the phase angle shift of a driven signal is 90 degrees.
  - When the frequency ratio is well below or above a 1:1 relation, the phase angle approaches 0 and 180 degrees respectively.
  - The amount of damping present effects the rate of this phase shift.
  - This explains the current waveform's 90 degree phase lag with respect to the torsional oscillations.
- The graphic below depicts two sinusoidal waveforms; one driver and one driven.
  - The blue cos(x) trace is the driver
  - The red sin(x) trace is the driven
  - Notice the 90 degree phase lag.





Conclusions and Recommendations

- Thermal Management and updated software algorithms
  - The thermal management algorithm is a calculation based on motor current. This is used as another layer of motor protection. The only action this takes is to limit effective current limit when certain criteria are met. The concept of dynamically limiting current has been in use since the system was commissioned. This has no impact on the tuning of the speed and current PI algorithms inside the DC drive and can not cause current divergence between the inboard and outboard motors as noted in this report.
  - Speed dependant current limitation was also added. This uses the same ramp technique as the other motor protection schemes used since the system was commissioned. Again, this has no impact on the tuning of the speed and current PI algorithms inside the DC drive and can not cause current divergence between the inboard and outboard motors as noted in this report.
- Drive Instability Causes
  - Drive instability would be caused from feedback errors to the speed control or current control PI algorithms, poor tuning in either PI algorithm, or defective drive components. This instability would be present consistently within the system and often times manifest itself as wild current swings (i.e. +/- 125% current swing) many times per second. These issues are easily identified as their magnitudes are often noticeable by operators and operations staff. This sort of instability is not present in this system.

Conclusions and Recommendations

- Torsional oscillations detected at multiple frequencies are excited by acoustical noise and impact events inside the shredder.
  - The oscillations are of system damaging magnitudes with dynamic loading well above the endurance limit of system components.
  - Large-scale torsional oscillations induce a current oscillation in the motors as seen by the lagging current waveform.
  - The 2nd order resonance in the system is causing a condition similar to armature reaction in the inboard motor, resulting in a 31.4Hz current fluctuation on that motor only.
- Other torsional oscillations were detected.
  - There is a f =2x(running frequency) present during ramp-up and ramp-down.
    - The mechanism by which this oscillation is produced is unknown, however, the event is another example of the systems under-damped response.
  - The 10.4Hz present during motor idling is caused by an imbalanced rotor.
    - The effect of gravity on the rotor's center of mass creates an additive force for 180 degrees of the shafts rotation (as the heavier side moves from bottom to top) and a subtractive force for the other 180 degrees (as the heavier side moves from top to bottom).
    - It is recommended to better balance the mill rotor to reduce these resonant events.
  - These oscillations are not currently of damaging proportions.
- The motors and drives are not the source of the various resonant events taking place in the system.
  - The torsional oscillations are present at magnitudes in excess of 70,000lb-ft before any current fluctuation is seen on the motors.
  - Analysis of the waveforms show that the DC Drive's current output lags the torsional oscillation events by 90 degrees, a fundamental characteristic of a driven signal.
  - The fact that the current fluctuations are not present until large scale oscillations occur and the 90 degree phase lag they exhibit are proof that the DC Drives are not driving the resonance. The current fluctuations are being driven by the torsional oscillations which is clearly demonstrated in the attached data.
  - The torsional oscillations occurring in the system are not produced by current oscillations.
- We have examined the drive control data, strain gauge data, and have shown the problem to be a mechanical resonance which leads the current fluctuations.
  - The three variables which affect a systems resonance are, mass, spring constants, and damping coefficients.
  - As the mass of the mill and material properties of the construction materials cannot be easily altered, damping must be strategically implemented throughout the system to reduce the resonance.
- It is the responsibility of the site's overseeing mechanical systems engineer to determine the sources of the resonance and dampen them accordingly.
  - Many mechanisms exist which combat resonance in rotating machinery.
  - These mechanisms effectively increase the damping coefficient of the system, reducing resonant growth rates and increasing resonant decay rates.
  - It is advised, that an expert in mechanical resonance specify these mechanisms.
- Shredder Feed Control can also have a significant impact on the magnitudes or the forces present in the mechanical system.
  - If mill is near or at full current great care should be taken to avoid allowing any more material in the mill. This addition of acoustical noise may further excite oscillations already present or ringing in the mechanical system

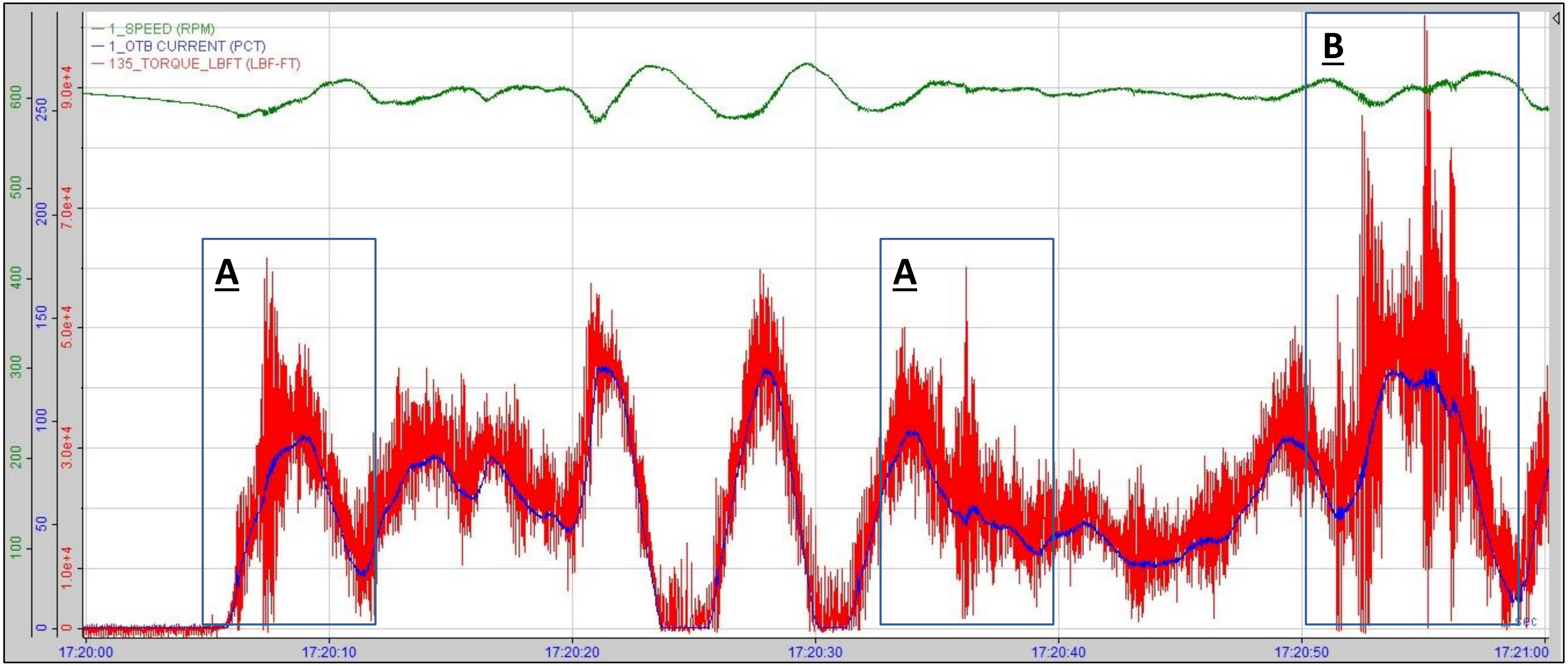
Data Analysis

Quad Plus
Variable Speed Drives and Systems for Industry

Channels Logged on DAQ

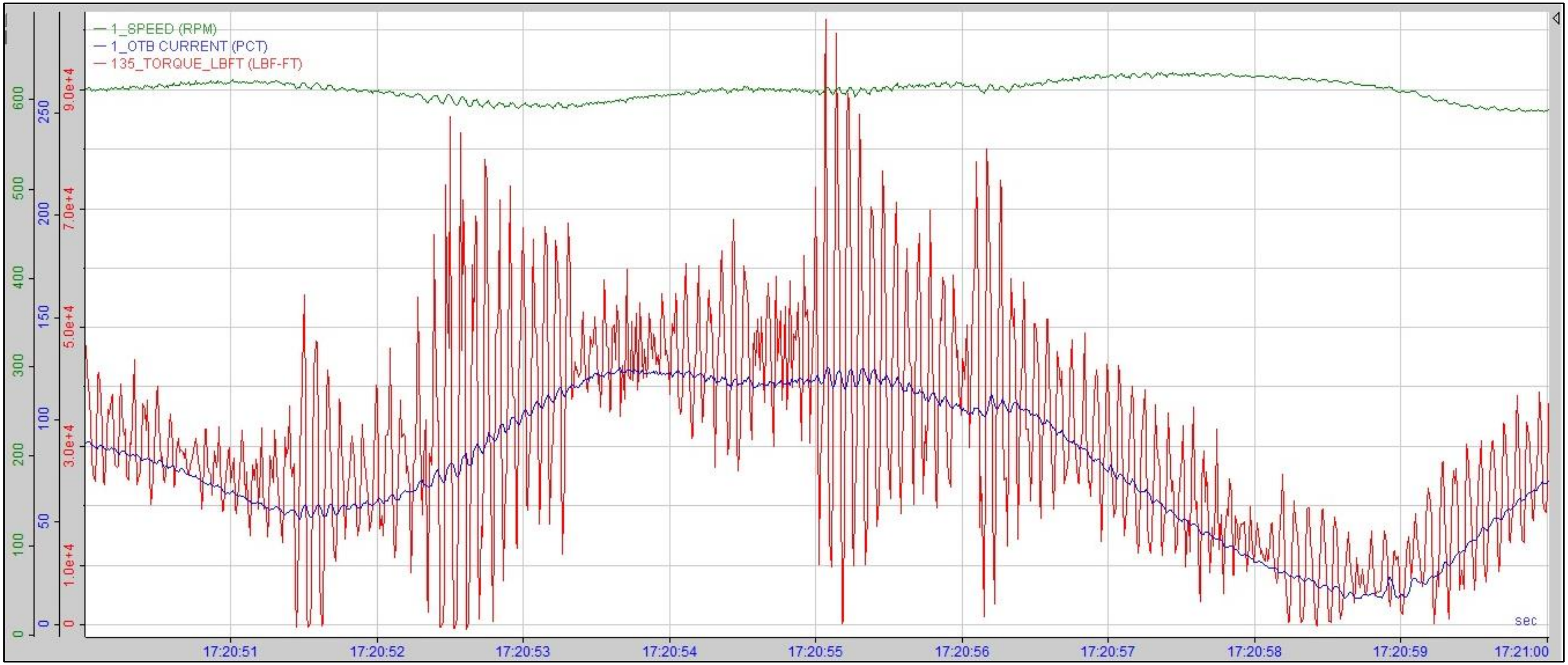
Channel	Signal	Source
0:0	Speed	1.02
0:1	Outboard Motor Current	1.06
0:1	Outboard Armature Voltage	1.13
0:3	Outboard Main AC Voltage	1.11
0:4	Inboard Motor Current	1.06
0:5	Inboard Armature Voltage	1.13
0:6	Inboard Main AC Voltage	1.11
1:0	8.25" Shaft Strain	SG-Link1
1:1	6.875" Shaft Strain	SG-Link2

One Minute of Shredding Operation



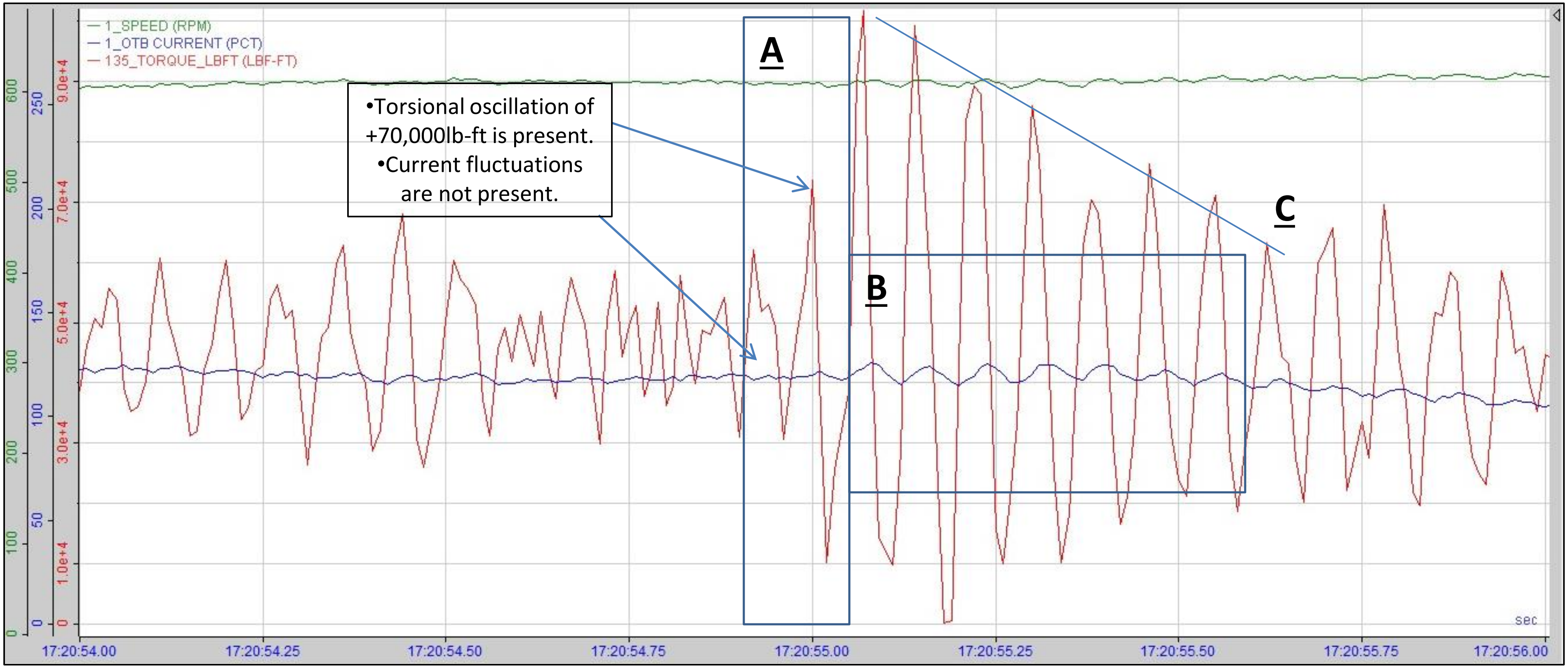
- This plot shows speed, outboard motor current, and the torque on the driveshaft over one minute of shredding operation.
  - Although the current and torque signals have different units, percent and ft-lb respectively, they are relatively scaled.
  - This allows us to see the magnitude relation between torque produced by the motors and torque produced by resonance.
- A: Notable torsional resonance events occur throughout shredding operation.
  - They oscillate at the 1st and 2nd order natural frequency of 12.5Hz and 31.4Hz respectively.
  - These events are of concerning magnitudes and occur with alarming frequency with an average event peak of approximately 80,000 lb-ft.
  - The frequencies are excited by impact forces on a heavily loaded shaft and constructive interference of acoustical noise.
- B: Extreme torsional oscillations were seen periodically during shredding.
  - They oscillate at the 1st and 2nd order natural frequency of 12.5Hz and 31.4Hz respectively.
  - The trace exhibits full-scale torque swings of over 105,000 lb-ft in magnitude.
- This event is plotted and discussed on the next three slides using different scales to better examine the waveforms.

10-Second Shredding



- This plot is a 10-second window of the previous plot and focuses our attention on the extreme resonance occurring.
- At this scale, the plot depicts the envelope in which the oscillations occur.
  - The growth and decay of the resonance is easily identified.
  - The waveform envelope is congruent with the spring-mass-damper- model previously discussed.
- The scaling of the current and torque signals remains relative in this plot for magnitude comparison.

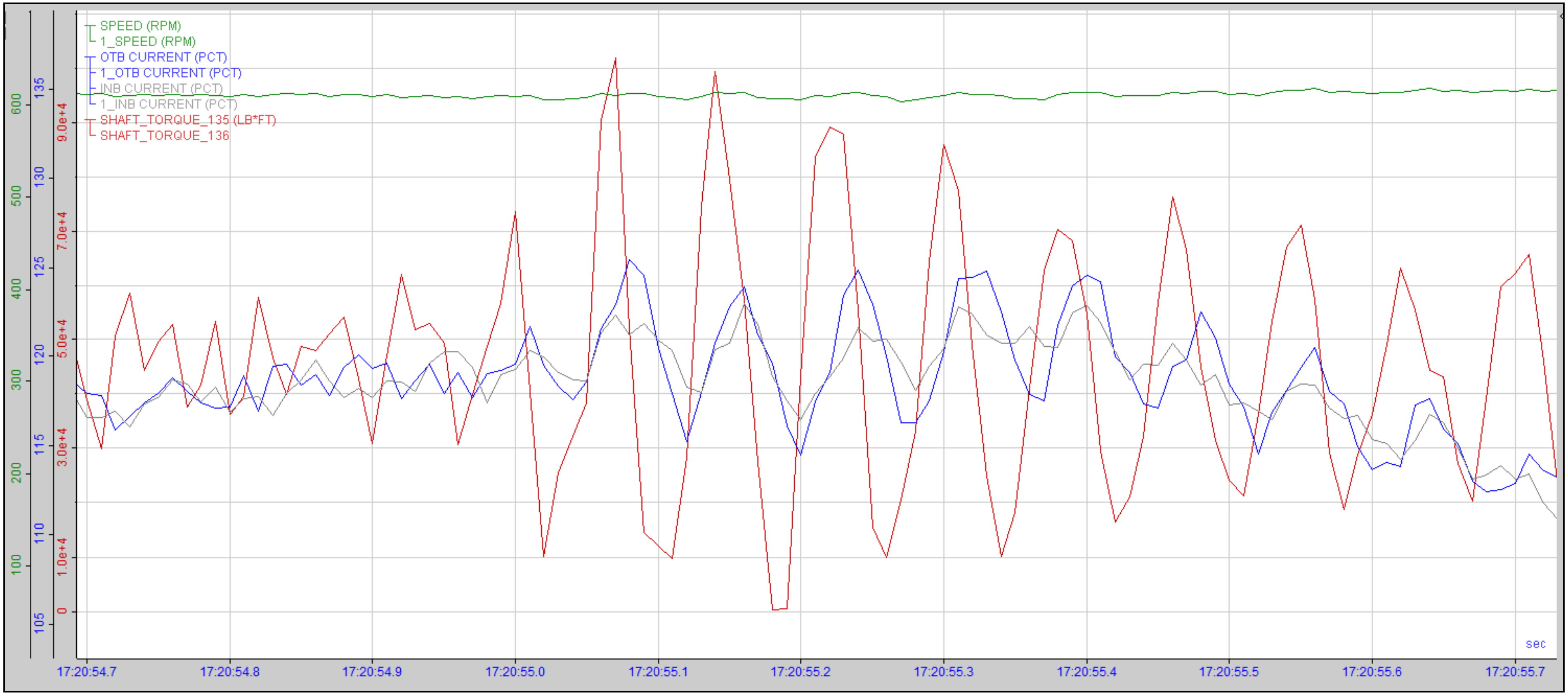
2-Second Shredding



- This plot shows the extreme torsional resonance shown with a 2-second time range.
- The scaling of the current and torque signals remains relative in this plot for magnitude comparison.
- The plot exhibits two notable occurrences.
  - A: Prior to any current fluctuation, the 12.5Hz torsional oscillation exhibits growth and a magnitude in excess of 70,000lb-ft.
  - B: The current waveform lags the torque waveform by 90 degrees, a fundamental characteristic of driven signals.
    - This phase separation shows the drives behaving in reactance to the torsional oscillation.
    - When compared to the spring-mass-damper example previously discussed, we see that the torsional oscillation is the driving waveform while the current is the driven waveform.
    - The torsional oscillations in mode 1 and 2 are causing load changes on the shaft. Just like every other load change on the shaft, the drives react to deliver the proper torque.
    - The next slide offers a view of the two wave forms with current scaling increased to better examine their phase relation.
  - C: Decay begins during maximum current oscillations.
    - This indicates a step-response in which the current fluctuations have nothing to do with.

Quad Plus
Variable Speed Drives and Systems for Industry

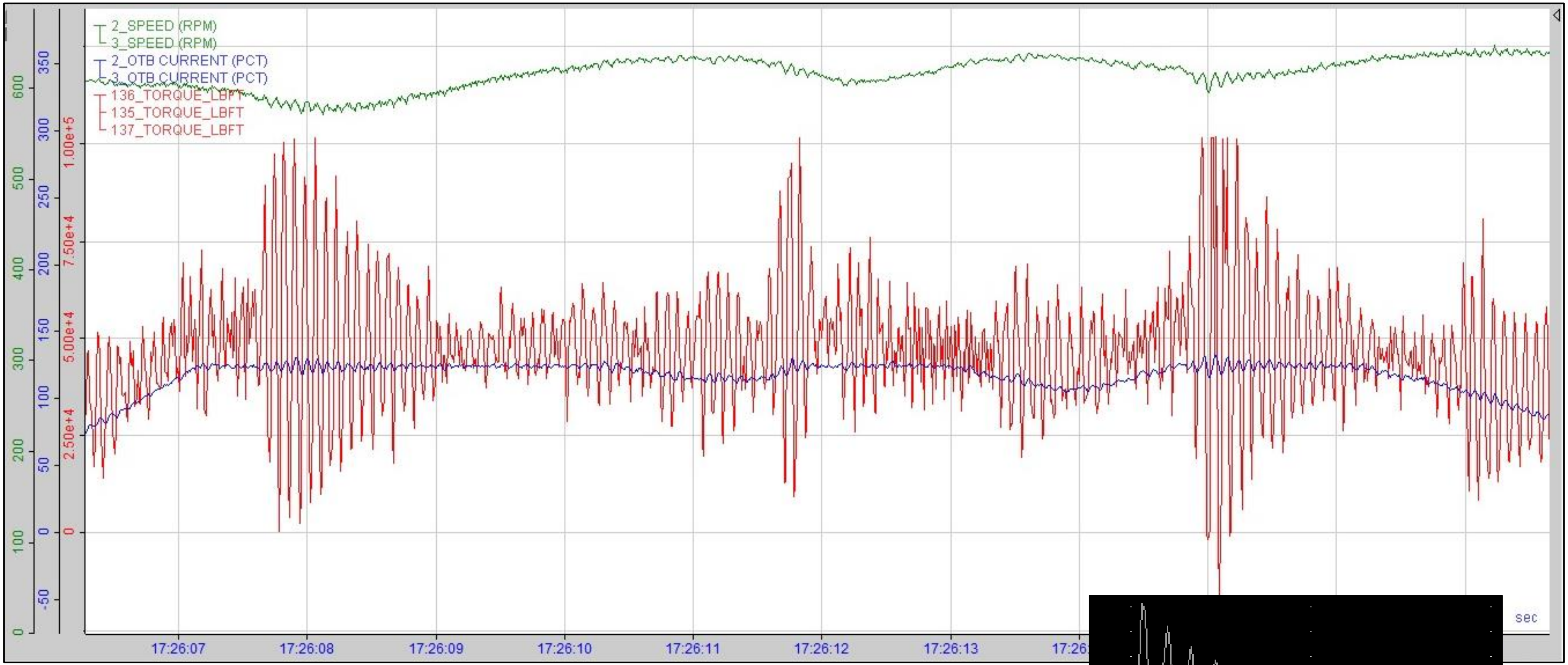
1-Second Shredding



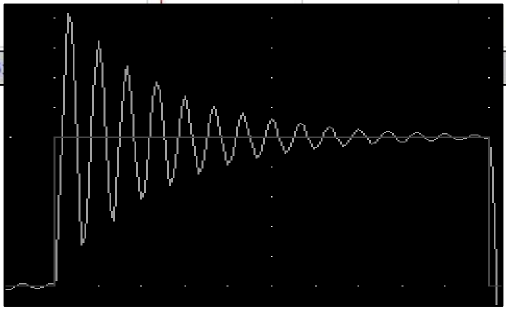
- This plot shows the extreme torsional resonance shown with a 1-second time range.
- The scaling of the current signals has been dramatized with respect to the torque signal to offer better examination of the two waveforms’ phase relation.
  - With this scaling, it is clear that the current oscillations are lagging the torsional oscillations by 90 degrees.
- The inboard motor current trace has been added to the plot.
  - The 2nd order natural frequency of 31.4Hz causes a current fluctuation on the inboard motor only.
  - Mode 2 resonance causes the inboard motor to oscillate within the stator, causing armature reactance.
  - As the armature oscillates within the stator, it is effecting the EMF of the motor which translates into voltage fluctuations on the armature.
  - These voltage fluctuations are seen by the inboard drive and the controllers act accordingly.

Quad Plus
Variable Speed Drives and Systems for Industry

Shredder Step-Response

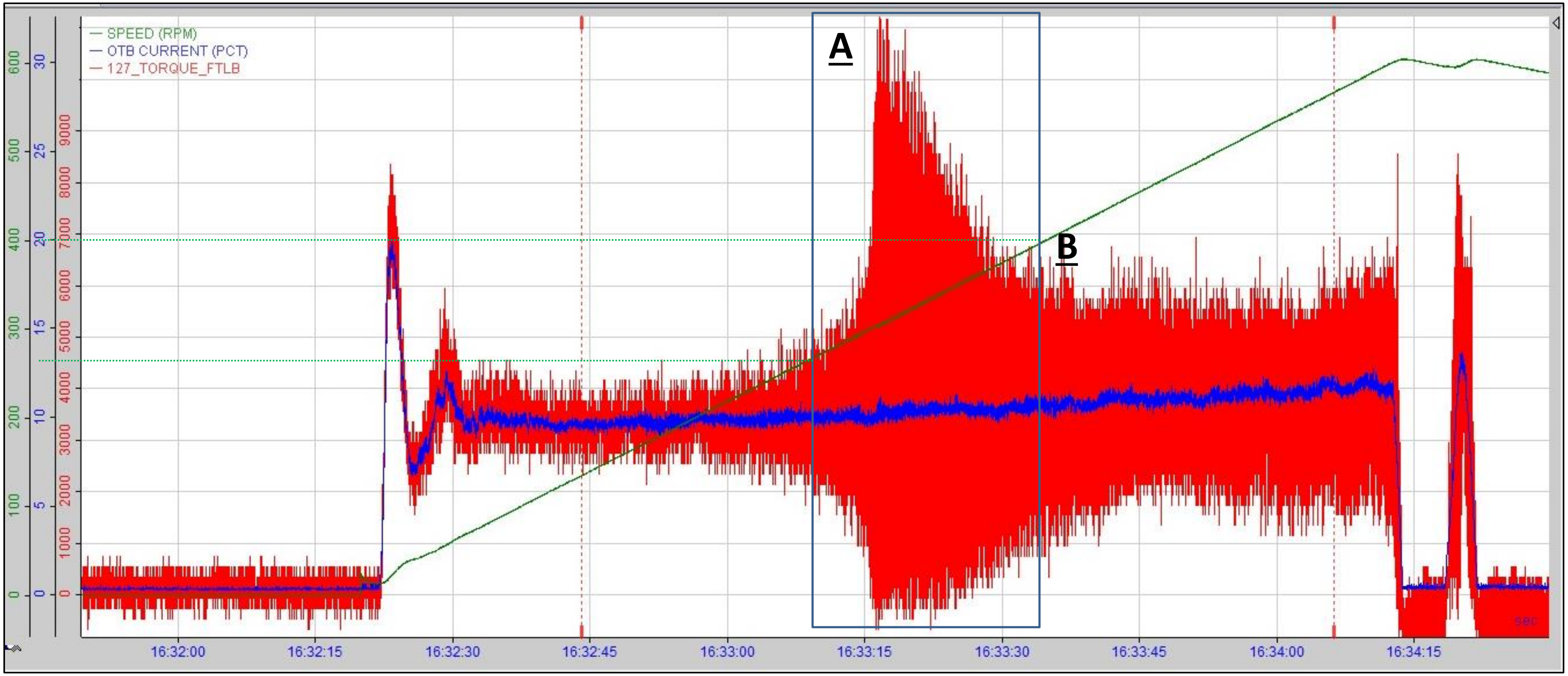


- This plot shows just over 10 seconds of shredding with the current and torque signals scaled respectively for magnitude comparison.
- This window occurs 5-6 minutes after the previous plot and three more extreme torsional resonance events can be seen.
- Each event shows full-scale 12.5Hz oscillations of over 100,000lb-ft.
- Examining the waveform envelope, we see this is an under-damped step-response, reflective of every other under-damped spring-mass-damper model and RLC circuit step-response.
  - When introducing an “unshreddable” or a “heavy” to the rotor, the impact of these objects act as step-inputs on the motor-motor-driveshaft-rotor assembly.
  - The induced current fluctuations are also visible and can be seen to begin after large-scale oscillations are present. Further, the 90 degree phase shift, a characteristic of a driven signal, is present.



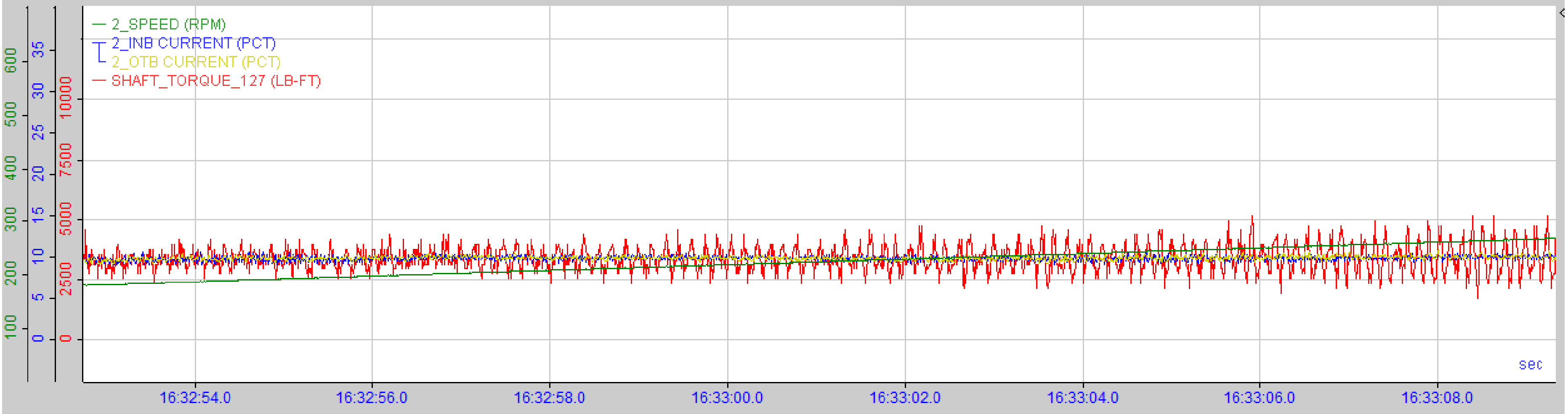
Recalling the step-response plot previously discussed, please note the similarity between resonant events and the step-response.

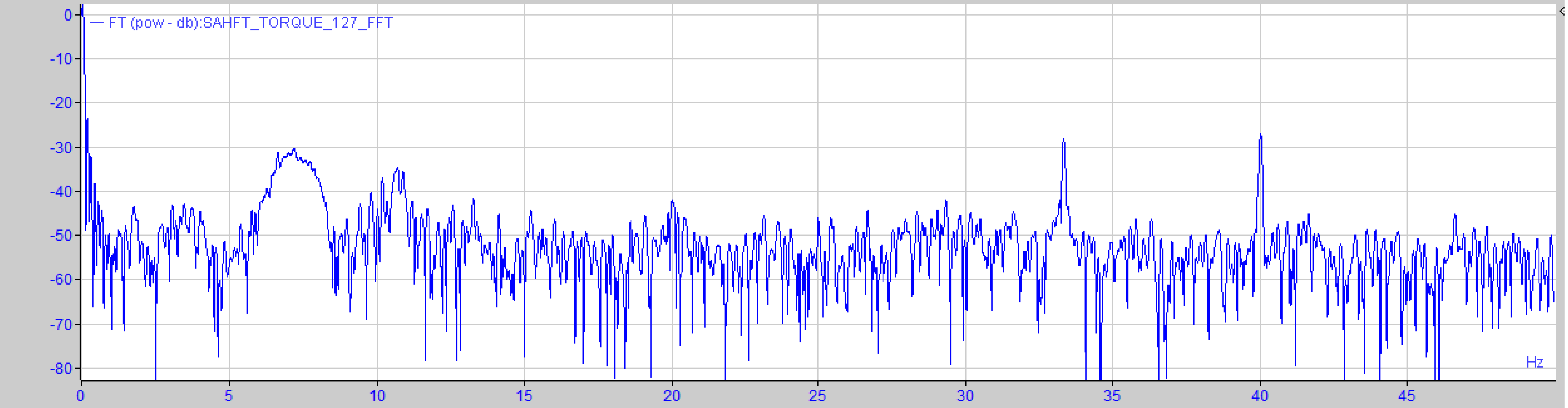
Two Minute Acceleration



- The plot shows a two-minute ramp-up.
- The peak torque due to resonance during this event was 11,109 lbf-ft
- The torsional oscillations exhibited a frequency shift with motor speed; *f* = 2x(Running Frequency)
- From 275-425RPM, the *f* = 2x(Running Frequency) is closest to the 1st order natural frequency of the system explaining the gradual-growth resonance.
- The next three slides will examine this frequency shift.

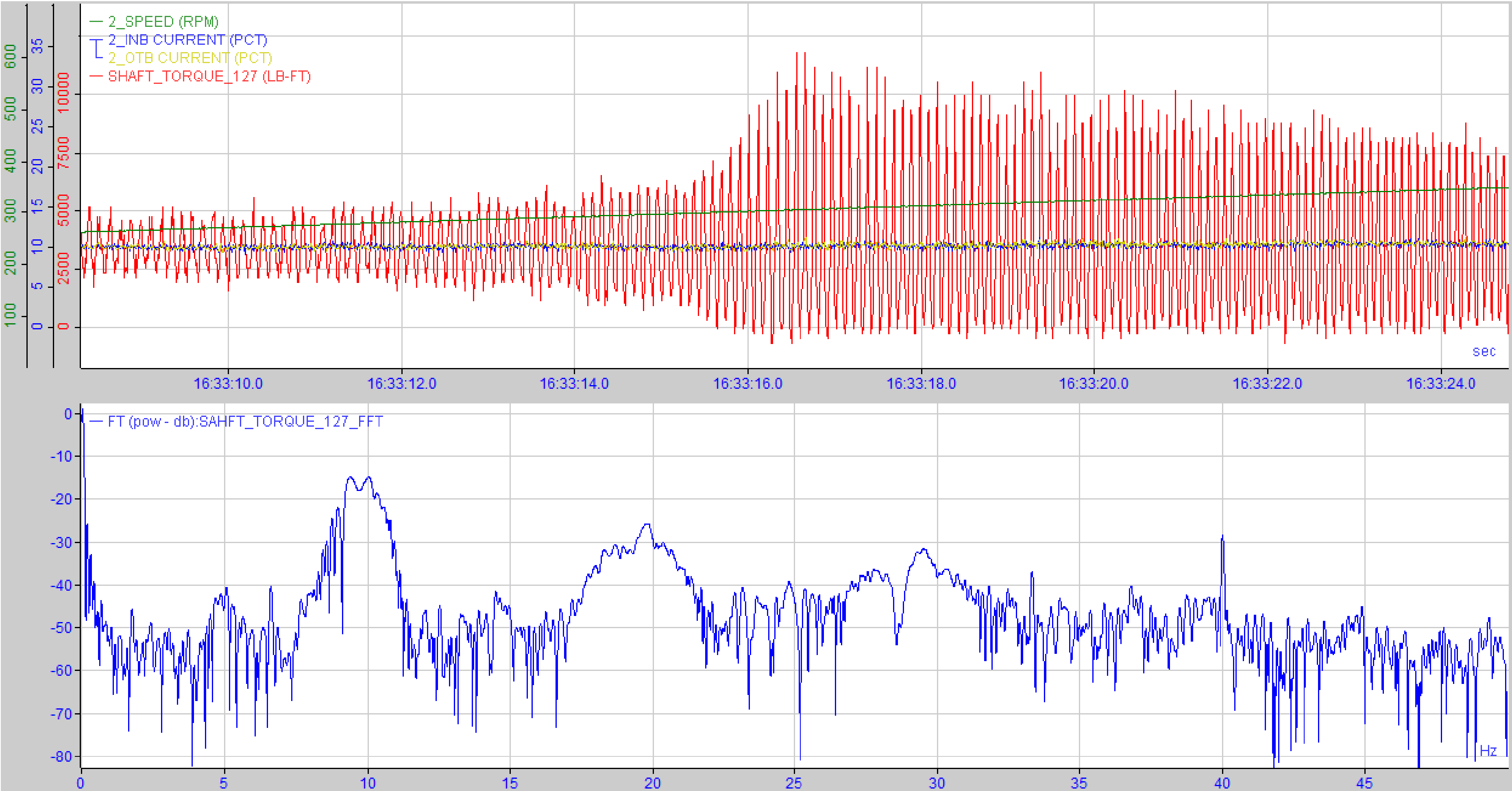
Ramp-up Resonance Frequency Shift





- The two plots above show a window of the start up resonance event centered at 200RPM
- Examining the FFT analysis plot, we see a predominant resonance frequency centered about 6.8Hz or 408CPM which is nearly twice the running speed.

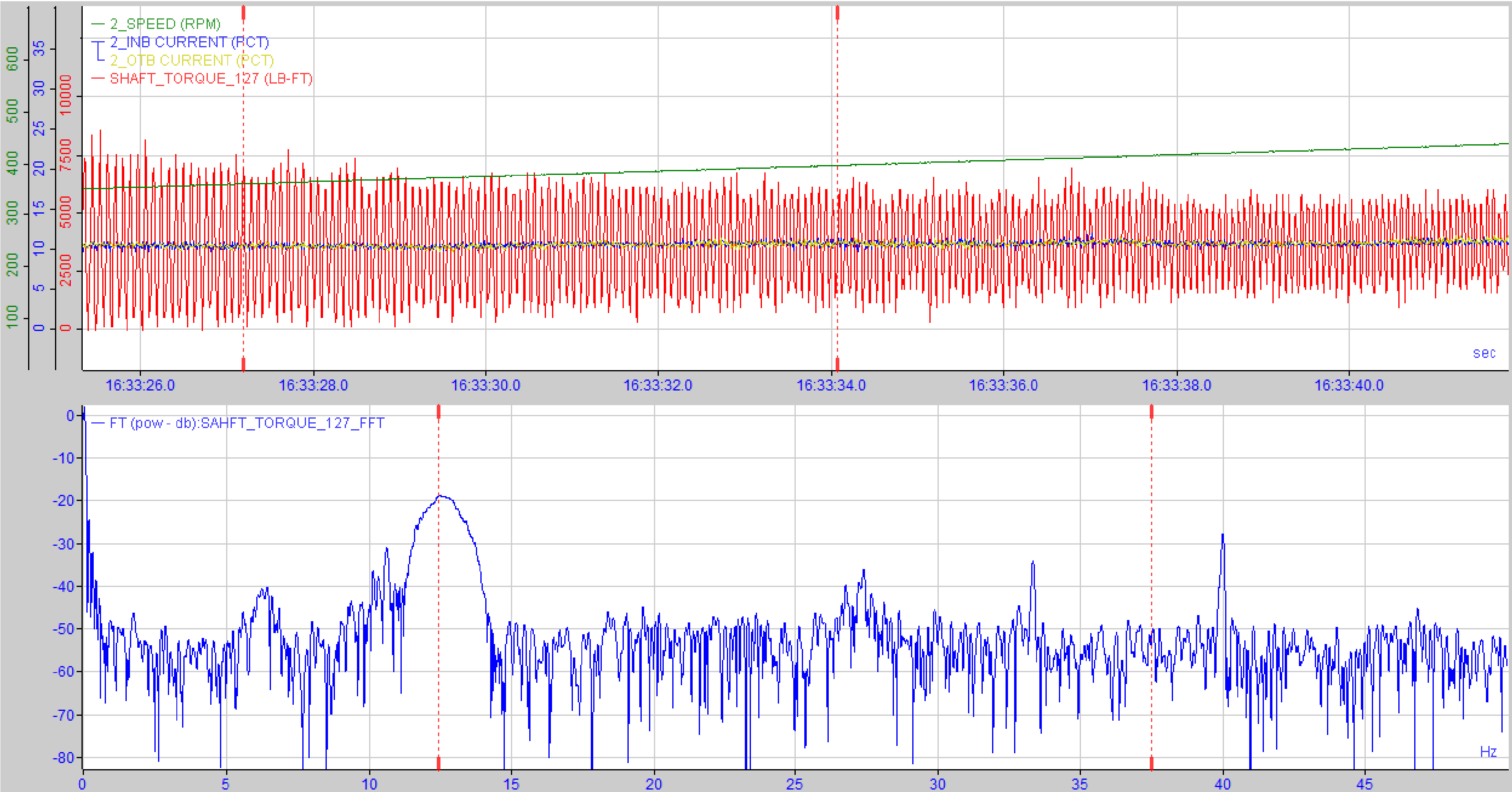
Ramp-up Resonance Frequency Shift



- The two plots above show a window of the start up resonance event centered at 300RPM
- Examining the FFT analysis plot, we see a predominant resonance frequency centered about 9.9Hz or 594CPM which is nearly twice the running speed.

Quad Plus
Variable Speed Drives and Systems for Industry

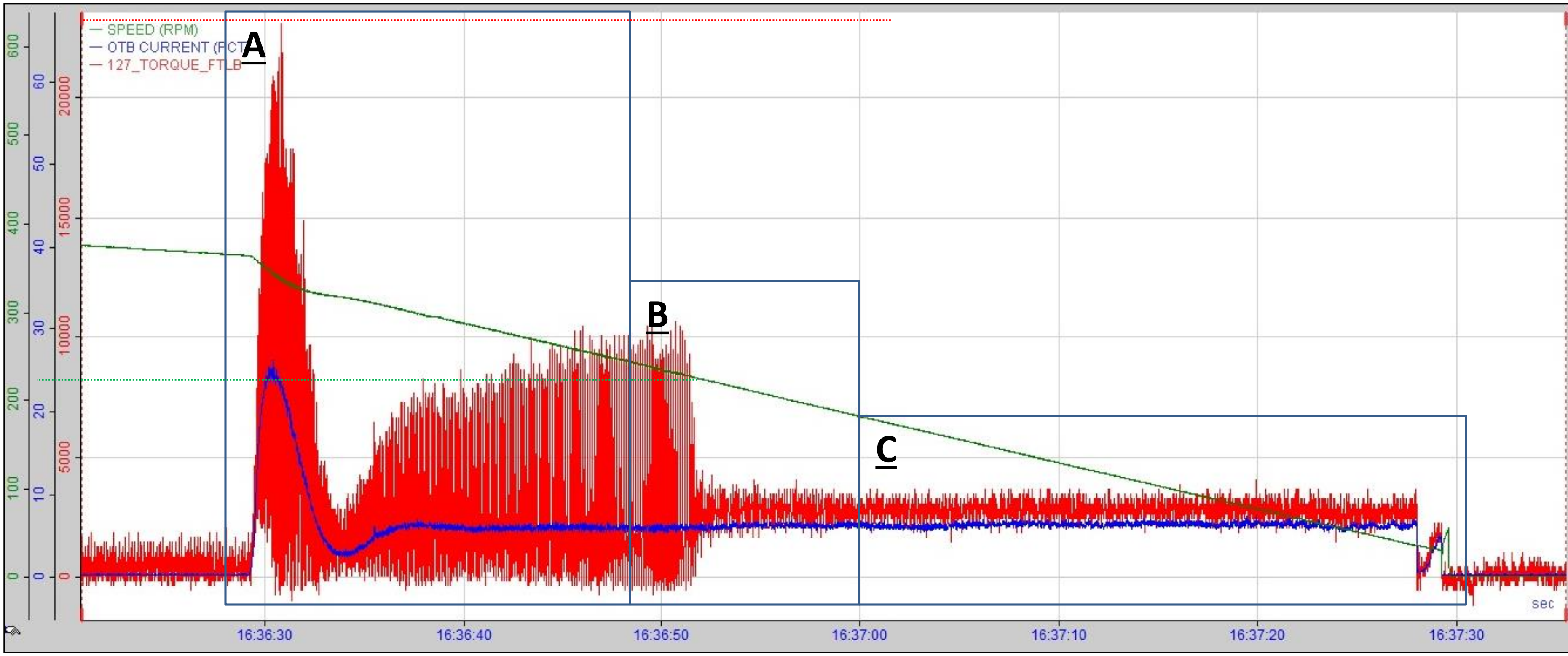
Ramp-up Resonance Frequency Shift



- The two plots above show a window of the start up resonance event centered at 400RPM
- Examining the FFT analysis plot, we see a predominant resonance frequency centered about 12.5Hz or 750CPM.
- This is not exactly *f* = 2x(Running Frequency) because at this running speed, the *f* = 2x(Running Frequency) is close enough to the 1st order natural frequency of the system to excite mode 1 oscillations.
- Mode 1 oscillations continue until the ramp-up is complete.

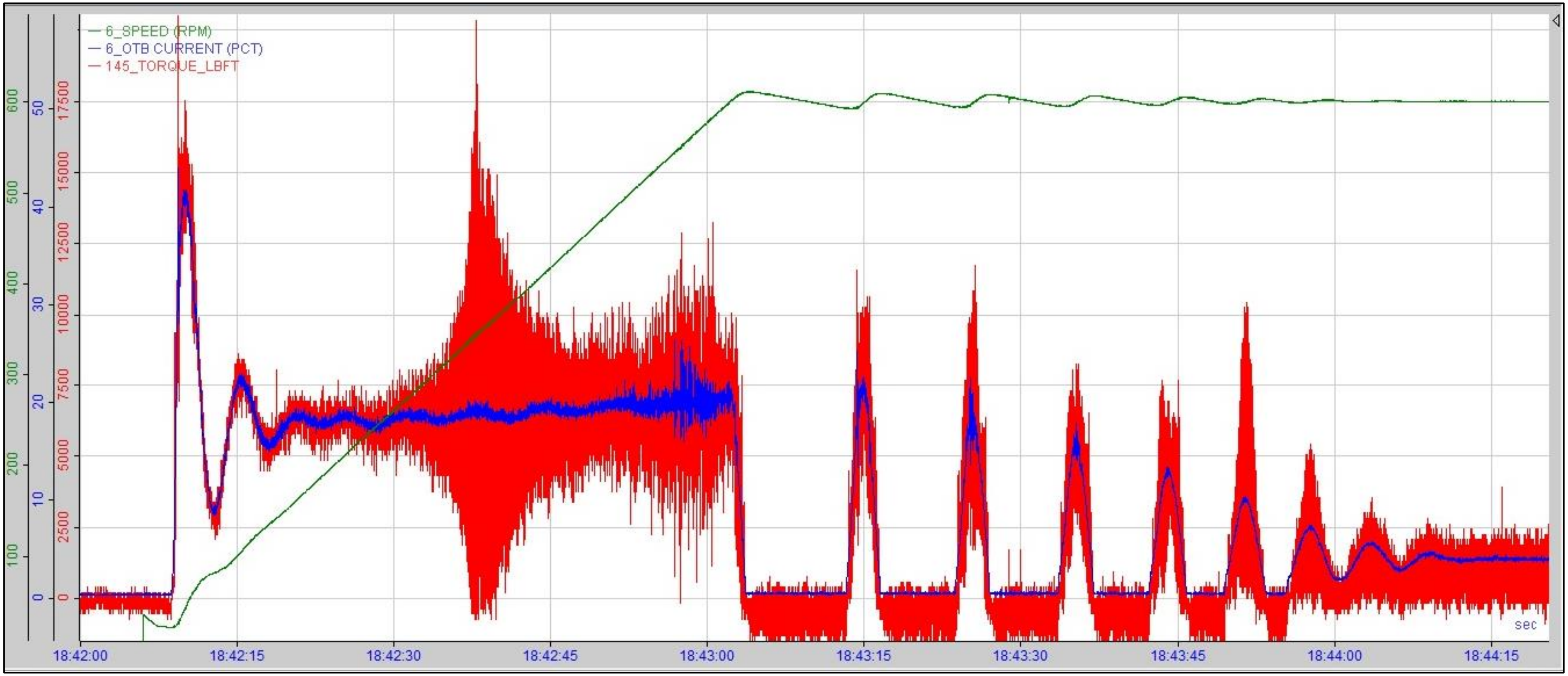
Quad Plus
Variable Speed Drives and Systems for Industry

One Minute Deceleration



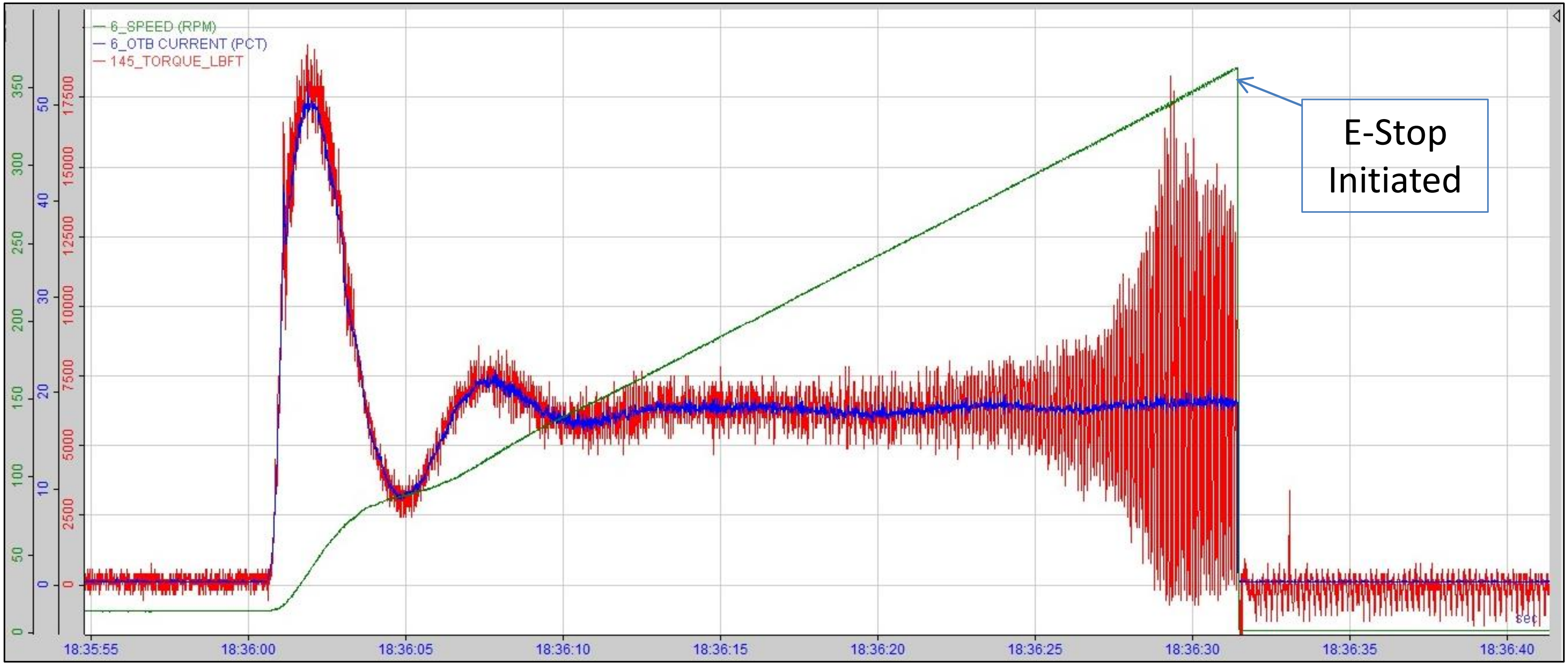
- The plot shows a 1-minute ramp-down.
- The peak torque due to resonance during this event were greater than the 2-minute ramp up due to the increased torque required to accelerate the rotor to the same speed over a shorter period of time.
- The event exhibited the same torsional oscillations with *f* = 2x(Running Frequency).

Effects of Adding Filter Time to Speed Control



- It was recommended by the drives expert on site during the test to add filter time to the speed controller.
- The effect of adding 200ms of filter time to the speed controller on ramp-up can be seen in the plot above.
  - The added filter time was ineffective.
  - The additional filter time results in poor speed regulation and ineffective current control.
- The *f* = 2x(Running Frequency) relation existed during all ramp-up and ramp-downs.

Acceleration with E-Stop Initiation



- During a 1-minute ramp-up, an E-Stop was initiated during the *f* = 2x(Running Frequency) torsional resonance.
- As load was removed from shaft, resonance exhibited rapid-decay as expected.
  - The torsional oscillation continued for 3 more cycles after e-stop was initiated before decaying to rest.
  - It is very difficult to excite an unloaded shaft. An unloaded shaft will not exhibit torsional resonance unless excited by an extremely high energy waveform, the type that are not present during coasting.
- The next slide examines the rapid-decay of the *f* = 2x(Running Frequency) torsional oscillation.

Quad Plus
Variable Speed Drives and Systems for Industry

Acceleration with E-Stop Initiation

- The plot shows the 1-second window during which the E-Stop was initiated.
- The rapid-decay resonance continues for 3 more cycles after e-stop is initiated, even producing a negative torque before settling.
- The speed and current values drop to zero because the drive from which the data was collected were powered off.