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ELECTRONIC TORSIONAL VIBRATION ELIMINATION FOR SYNCHRONOUS MOTOR DRIVEN TURBOMACHINERY

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ABSTRACT

The oil and gas industry has a growing demand for electrically driven trains operated at variable speeds. Variable frequency electrical drives enable increased operational flexibility and energy efficiency. One drawback of power electronics driven systems is the generation of non-fundamental air-gap torque ripple components due to electrical harmonics. The air-gap torque ripple can interact with the mechanical system at natural torsional frequencies of the drive train. Uncontrolled excited torsional vibration can silently lead to coupling failure due to fatigue. The coincidence of electrical drive harmonics and natural torsional frequencies of the mechanical system is sometimes unavoidable, due to the large variable speed range of the compressor as for process requirements. For those types of applications, a damping system utilizing available power electronics has been developed that can be applied to new units but also as a retrofit solution in existing variable speed trains.

Electronic torsional vibration elimination (eTve) is based on an angular vibration measurement in the mechanical system and an interface to the existing inverter control of the electrical drive. An important milestone of the eTve development was achieved in 2010, in site testing this new solution to Liquid Natural Gas (LNG) production trains and demonstrating that it can completely eliminate torsional vibrations. With eTve a residual torsional vibration level was achieved that was lower than the vibration level measured while the LNG train was only gas turbine driven. This torsional performance was achieved with a standard load commutated inverter drive (LCI). LCIs are one of the most widespread electrical drive technology for gas compression trains because of excellent reliability records, and

it is the only one referenced solution for electric power larger than 45 MW.

INTRODUCTION

There is an increasing trend in the oil and gas industry towards electrical drives rated for tens of Megawatts, especially in high-power compressor trains e.g. for Liquefied Natural Gas (LNG). A present role of these drives is to act as starter and helper for the prime movers of the compressor trains. Such prime movers are typically constituted by gas turbines (GT). A typical LNG drive train configuration is shown in Figure 1. The gas turbine drives the compressors (CC) as mechanical loads. The electrical machine (EM) can operate as a GT helper. Using a power electronic converter connected to the electrical machine allows the system to operate as a variable speed drive system (VSD). The total electrical and mechanical drive power of such VSD compression trains can be in the order of 100 Megawatt. In the oil and gas industry, the reliability of production processes is extremely important for the profitability of the investment. Drive manufacturers offer a large variety of converter topologies for variable speed drives in the Megawatt power range. For compressor applications in this nominal power range, there are basically two types of inverters that can be applied: A current source inverter, known as a Load-Commutated-Inverter (LCI), and a voltage source inverter (VSI), known as pulse-width-modulated (PWM) inverter. LCIs have a significantly smaller number of components than systems based on self-commutated devices. Although VSIs can better fulfil the high performance and dynamic requirements of driven loads and can better fulfil grid codes, LCIs are still today the most widespread technology for large drives. These drives are well referenced and have been used in industrial

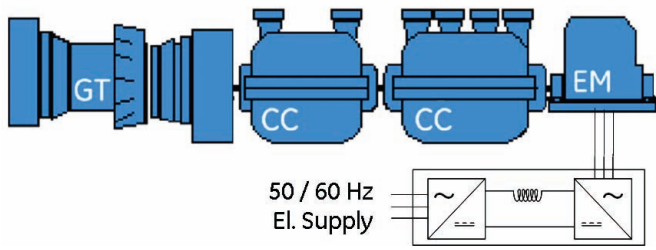


Figure 1: Typical LNG compression train configuration

applications for decades despite their drawbacks, such as high THD (total harmonic distortion) on the grid side and high torque ripple on the motor side.

The integer- and non-integer harmonics generated by LCIs cause pulsating torque harmonics on the grid side and machine side of the VSD. A coincidence of the pulsating torque harmonics with a torsional natural frequency cannot always be avoided due to the large number of potential excitations, especially in applications with a wide operating speed range. These continuously generated harmonic torque excitations could have a critical impact on the torsional behaviour of the entire train. As far as mechanical resonance excitation is concerned, the most relevant parameter of the electrical excitation source is its frequency (proximity to critical natural frequencies) and the exposure time of excitation, while magnitude has lower criticality. Therefore, a greater electrical excitation source with a frequency not close to a mechanical resonant frequency will have less impact on the shaft than a smaller excitation with a frequency at or near that mechanical resonant frequency. This explains why torsional issues are also reported in applications where the electrical drive system is based on VSIs [2], [10], [11] although VSIs are known to produce significantly less torque ripple than LCIs.

It is therefore mandatory to perform torsional analyses during the design stage of large drives, as specified in API 617. In multi-unit plants, more sophisticated analyses must be performed, which consider the fully coupled electrical and mechanical systems (Gensets, power grid and electric driven trains).

NOMENCLATURE

LCI	- Load Commutated Inverter
ITMD	- Integrated Torsional Mode Damping
LNG	- Liquefied Natural Gas
GT	- Gas Turbine
ST	- Steam Turbine
M	- Motor
G	- Generator
THD	- Total Harmonic Distortion
VSD	- Variable Speed Drive
VSI	- Voltage Source Inverter
p_m, p_g	- Pulse numbers on machine / grid side bridge
f_m, f_g	- Electrical frequency on machine / grid side

J_1, J_2	- Moments of inertia
K_{12}	- Stiffness constant
T	- Torque
θ	- Torsion angle
ω_{nat}	- Natural angular frequency
$f_{pulsating}$	- Frequency of pulsating air-gap torques
ζ	- Damping ratio
Q	- Amplification factor, $Q = 1/(2D)$
Δ	- Logarithmic decrement
CD	- Continuous Duty
FFT	- Fast Fourier Transformation
β -angle	- Motor side (inverter) firing angle
$\Delta\beta$	- Modulation of inverter firing angle
J	- Matrix of moments of inertia
D	- Damping matrix
K	- Stiffness matrix
φ	- Vector of torsion angles
u	- Vector of externally applied torques
B	- Input matrix for external torques
CTQ	- Critical to Quality
eTVe	- Electronic Torsional Vibration Elimination
TVPS	- Torsional Vibration Protection System
DOE	- Design of Experiments

LCI PULSATING TORQUE COMPONENTS

In motors driven by power electronic converters, pulsating torques of various frequencies are superposed on the air-gap torque [2]. As an example, Figure 2 shows two six-pulse thyristor bridges on the grid side (rectifier) and two on the machine side (inverter). In using secondary transformer windings that are 30 degree phase shifted, the effective pulse number of the rectifier bridge m can be increased to $m = 12$, thus reducing the voltage ripple of the rectifier stage. An increase of the pulse number to $n = 12$ is also achieved for the machine side inverter, by 30 degree phase shifted motor windings.

The dominant frequencies of the voltage ripple caused by the rectifier and inverter bridges of the 12/12-pulse LCI in Figure 2 are $f_{pulsating} = 12 * f_g$ for the rectifier bridges and $f_{pulsating} = 12 * f_m$ for the inverter bridges, f_g and f_m are the electrical frequencies on the grid and on the motor side.. The harmonic spectrum of the DC link current also contains multiples of these frequencies.

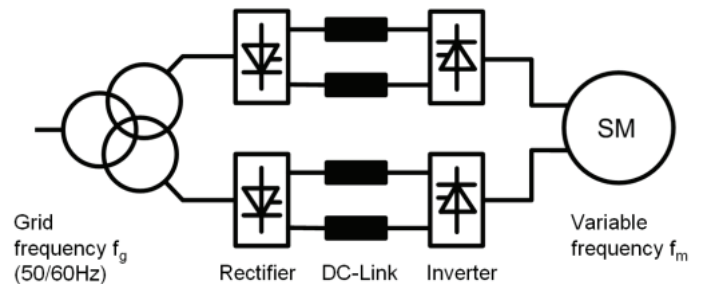


Figure 2: 12/12-Pulse LCI

These current ripple waveforms are responsible for integer- and non-integer pulsating torque components in the air-gap-torque of the electrical machine.

Assuming an electrical frequency of $f_g = 50$ Hz, the integer pulsating torque components on the grid side or at nominal speed on the motor side are at 600 Hz and higher. This is typically not an issue for the mechanical system. The non-integer pulsating torque has typically much lower amplitudes than the integer torque components. They are generated as convolution products of the grid side and machine side electrical parameters (voltages and currents). Their frequency can be calculated with Equation (1):

$$f_{\text{pulsating}} = |n_g \cdot m \cdot f_g \pm n_m \cdot n \cdot f_m| \quad n_m = 1, 2, 3 \dots \quad n_g = 1, 2, 3 \dots \quad (1)$$

where

- n, m : pulse numbers on machine / grid side bridge, $n=m=12$ in
- f_g : electr. frequency on grid side (50 or 60 Hz)
- f_m : electrical frequency of machine, $f_m = (\text{RPM}/60) \cdot p$,
where p is the motor's pole pair number, typically $p = 1$ or $p = 2$

This set of non-integer pulsating torques can be most critical in large drives; it covers a large frequency band for harmonic production close to the nominal speed. The effect can be identified in the Campbell diagram, which is a graphic representation of equation (1), with rotational speed dependent pulsating torque frequencies. The Campbell diagram enables to identify critical resonant operating speeds, in the coincidence of electrically generated frequency excitations with mechanical torsional natural frequencies that are horizontal lines in the diagram. The Campbell diagram shown in Figure 3 belongs to a train with a nominal speed of 3600 rpm and a VSDS system fed by powergrid @ 60 Hz. Potential critical operation speeds are at the crossings of electrical pulsating torque lines and natural mechanical frequencies (horizontal green lines in Figure 3), and are dependent on the amplitude of the pulsating torque components and on the modal mechanical damping of the drive train at that torsional natural frequency. For example, if the operational speed is varying within the red box, seven potential critical operating speeds can be identified in Figure 3 because of crossings of line $|6f_g - 6f_m|$, $|12f_g - 12f_m|$, $|24f_g - 24f_m|$ with the first (Mode 1) and second (Mode 2) torsional natural frequency.

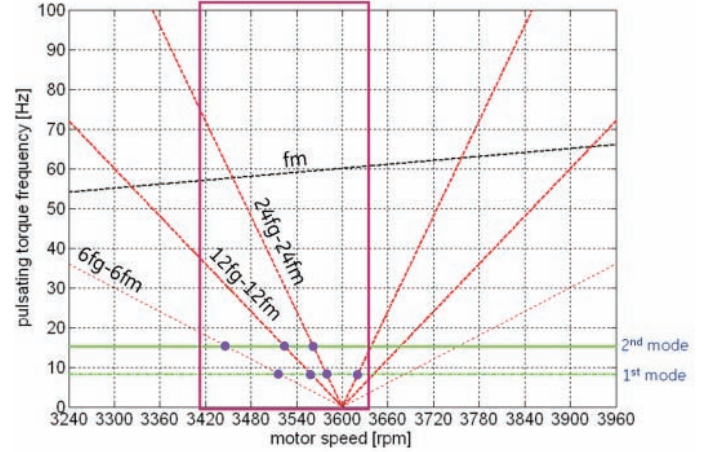


Figure 3: Campbell Diagram of a Compression Train Driven by a 12-Pulse LCI

It is important to know that the amplitudes of non-integer pulsating torques are typically low in the nominal speed range of a compression train, e. g., less than one percent of the nominal torque, as also reported by Simond, et al. (2005) [5]. This low level of electrical excitation at a critical frequency of the mechanical system has also been found by air gap torque measurements during the tests performed with a 30 MW compression train.[8]

Having confirmed that electrical (air-gap) torque components with amplitudes lower than one percent of the nominal air-gap torque can cause relevant torsional excitations, the dynamic behavior of the coupled electrical and mechanical system needs to be investigated in more detail. For this purpose, reduced order models for relevant drive train designs are derived in the next section, and recommendations are given for modal damping parameters of models representing synchronous motor driven turbomachines.

MECHANICAL EFFECTS OF PULSATING TORQUES

The first step in a torsional analysis is to generate a lumped parameter model of the mechanical system, as described in more detail by Ong (1999) [4]. The differential equation belonging to the reduced order model of two rotating masses (Figure 4) is described by Equation (2) and obtained by applying the Newton's second law to a two-mass rotating system with respective inertias J_1 and J_2 and a torsional stiffness K_{12} . θ_1 and θ_2 are the torsion angles. Internal Damping (D_{12}) is neglected. With this approach the eigenvalues of an undamped system can be calculated. This approach is also applied to large drive trains if negligible modal damping is an appropriate assumption [8].

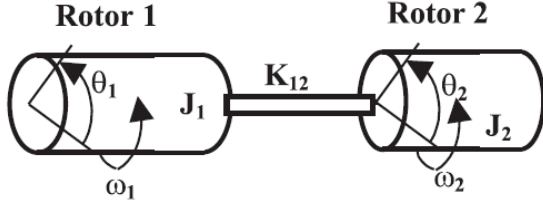


Figure 4: Reduced Order Model of Two Rotating Masses

Assuming that an external torque of magnitude T_a is applied, and the same pulsation as the natural frequency (ω_{nat}) of the mechanical shaft, the torsion angles will oscillate according to Equation (3).

$$\begin{aligned} J_1 \frac{d^2 \Theta_1}{dt^2} &= K_{12}(\Theta_2 - \Theta_1) \\ J_2 \frac{d^2 \Theta_2}{dt^2} &= K_{12}(\Theta_1 - \Theta_2) + T_a \cos(\omega_{nat} t) \end{aligned} \quad (2)$$

with $\omega_{nat}^2 = K_{12} \frac{J_1 + J_2}{J_1 J_2}$

with

$$\begin{aligned} \Theta_1(t) &= \lambda_1 (t \sin(\omega_{nat} t) + \cos(\omega_{nat} t)) \\ \Theta_2(t) &= \lambda_1 \left(\frac{J_1}{J_2} t \sin(\omega_{nat} t) - \cos(\omega_{nat} t) \right) \end{aligned} \quad (3)$$

with $\lambda_1 = \frac{T_a}{2(J_1 + J_2)} \omega_{nat}$

Equation 3 shows that if the externally applied torque has the same frequency as the mechanical resonance frequency of the mechanical system, the torsion angles will increase linearly over time if the mechanical system has no torsional damping. A generalized multi-mass system can be described in form of a, differential equation system given by:

$$\mathbf{J}\ddot{\boldsymbol{\theta}} + \mathbf{D}\dot{\boldsymbol{\theta}} + \mathbf{K}\boldsymbol{\theta} = \mathbf{T}_{ext} \quad (4)$$

The characteristic equation of the mechanical system, neglecting damping and assuming harmonic motion, that is $\ddot{\boldsymbol{\theta}} = -\lambda \boldsymbol{\theta}$, the natural (unforced, $\mathbf{T}_{ext} = 0$) system equation becomes:

$$|\mathbf{J}^{-1}\mathbf{K} - \lambda \mathbf{I}| = 0, \quad (5)$$

where $\mathbf{I}$ is the identity matrix. The shaft natural frequencies $\omega_{nat,j}$ can be calculated as follows, taking into account the roots λ_j (eigenvalues) of the characteristic equations:

$$\omega_{nat,j} = \sqrt{\lambda_j}, \quad j = 1 \dots n-1$$

For an n -mass system, there are $n-1$ eigenfrequencies.

The eigenvectors derived from equation (5) represent the mode shapes of the mechanical system. Figure 5 includes the mode shape representing the first natural frequency of the given drive train configuration. The mode shape is an important tool to understand e.g. the basic behavior of the drive train at that critical frequency, which part of the drive train oscillates against which part (mode 1: GT against rest of the drive train), but also the sensitivity of the mechanical drive train to be excited from the electric machine or the optimal sensing location for that mode.

Modal damping must be applied to the model, in order to limit the response at resonance to finite values. The modal damping ratio ζ is expressed as a percent of the value for a critically damped system. A critically damped system

converges to zero faster than any other without oscillation ($\zeta = 100$ percent).

A more detailed description of damping properties and Q -factor determination for large drive trains can be found in Sihler et al. (2009) [8]. Table 1 summarizes the amplification factors (Q -factors) reported for synchronous machines driving turbomachinery and for power generation applications, together with the rated power of the synchronous machines. As can be seen from Table 1, generic values for the torsional modal damping must be applied with extreme care, especially in new applications with synchronous motor-driven turbomachinery and high power ratings. Instead, conservative values derived from similar designs should be used until measurement results become available.

Table 1: Summary of Amplification Factors Q Reported in Literature for High Power Trains driven by Synchronous Machines

Application	Typical Nominal Power	Relevant Torsional Nat.Freq	Q (conservative)
Geared turbomachinery	≤ 50 MW	≤ 200 Hz	25-33
Ungeared turbomachinery	≤ 50 MW	≤ 200 Hz	40
Flywheel generator	150 MW	20-30 Hz	200
Power generation (utility steam turbine)	≤ 1000 MW	8-150 Hz	400 (full load) 4000 (light load)

Torsional Vibration Sensing Based on Tip-Timing

The sensing method applied for eTve is the so called tip-timing method (Golebiowski et al (2010)) [6], which is based on targeting a n -teeth geared wheel. A clear advantages of using this sensing technology is that the sensing is based on well referenced and well proven instrumentation required for the control of the gas turbine. The basic functional operation is the following: Raw signals acquired from the probes are compared with voltage thresholds (fixed by the user) to identify tooth tip passing time. Interpolation on the raw signal rising edge is possible to increase the tip timing accuracy. Instantaneous rotational speed $d\theta/dt$ is calculated as tooth-to-tooth angle ($d\theta = n/360$ [°]) divided by the measured tooth-to-tooth time (dt [s]). Speed versus time distribution is then integrated to obtain angular vibration versus time waveform.

The required real-time performance for tip-timing sensing and calculating angular vibration (online) depends on:

- Number of tooth tips per rotation,
- Operational rotational speed,
- Relevant mode shapes distribution along the shaft at the desired natural frequencies together with the sensing plane location and

The embedded damping controller is disabled until $t > 3s$. The alternating torque component oscillating at the first torsional natural frequency is growing within the first 0.5s at critical speed operation. After enabling the embedded controller ($t > 3s$), the dynamic torque component is decreasing because of additional damping power P_{damp} . The remaining dynamic torque component depends on the closed loop gain of the damping control. The optimum phase relation for effective damping is $\phi = -\frac{\pi}{2}$, based on the theory of mechanical oscillations. But damping can theoretically be achieved with $0 < \phi < -\pi$, which makes this method robust against small phase variations.

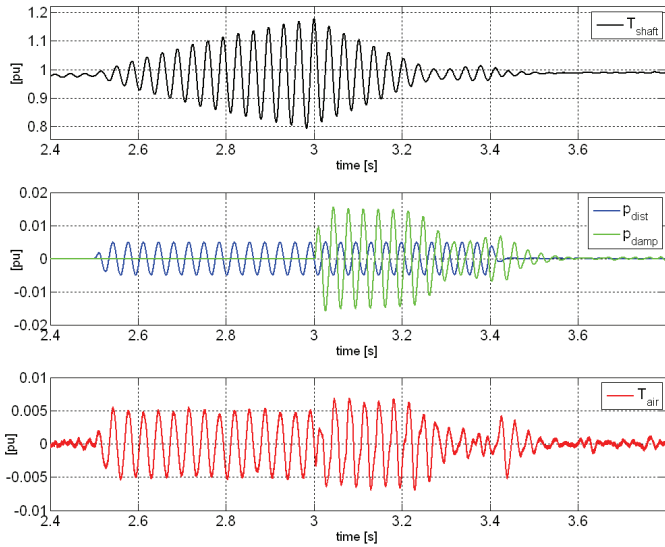


Figure 9a: Electromechanical Simulation: Simulated Effect of Torsional Mode Damping applied to an LCI Driven Compression Train.

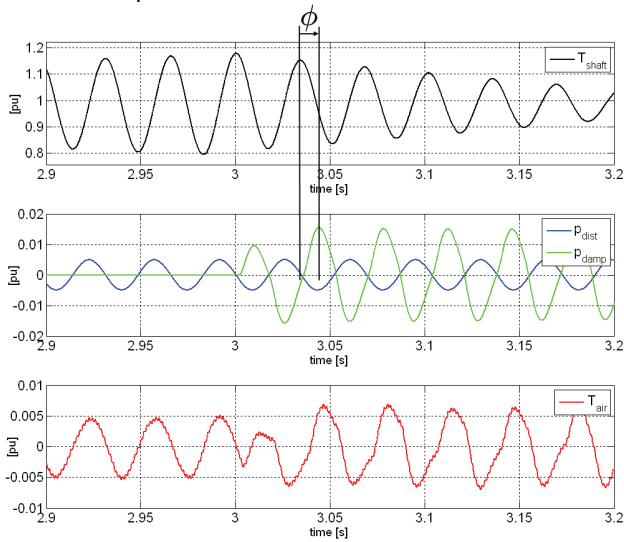


Figure 9b: Zoom of Figure 9a: Damping torque acting with $\phi = -\frac{\pi}{2}$ phase at ω_{nat} between shaft torque T_{shaft} and effective damping torque component P_{damp}

In the simulation a β -angle modulation of 1.5° was applied which corresponds to an active power modulation of less than 0.5 percent of the nominal power of the converter. The damping power with a frequency ω_{nat} is partially taken from the DC-link inductor, and partially from the grid, dependent on the control settings of the DC link current controller. The effect of the active power modulation on the machine side power factor is negligible, considering that the $\Delta\beta$ signal oscillates between 0 and 1.5° . This results in an average value of 0.75° that is subtracted from a static β -angle of typically 150° for the short time of damping with full damping power. The amount of damping power applied can safely be limited to the value that corresponds to the maximum β -angle modulation, e.g. to 2 or 3 degrees.

Thus, the TMD control unit shown in Figure 6/7 can be designed as an external, retrofit extension for a given LCI control system. The TVPS angular vibration measurement is typically based on already existing instrumentation. Therefore, the application of eTVe does not require changes to the mechanical setup or drive system design, only an additional interface to the machine side (inverter) β -control and the additional control blocks shown in Figure 7.

TORSIONAL INTERACTION ANALYSIS

LNG plants typically have two, almost identical drive trains (Figure 10). Interactions between these two drive trains cannot always be excluded, they depend on the grid strength and potential decoupling of the two VFD's, but also on VFD control settings.

A detailed simulation model including the mechanical system, electric machine, VFD and control has been used to investigate potential electro-mechanical interactions and to verify eTVe for different operational and system conditions. The variables used in the DOE are:

- Total control loop dead time due to initial uncertainties in sensing and VFD interface implementation up to 300ms
- Total control dead time variation during operation - $10ms \leq t \leq 10ms$
- TVPS dynamic accuracy 1% to 10% of nominal torque
- TVPS SNR 20 to 1

The simulation results have shown that eTVe operation improves the damping properties of the drive train even under the worst case assumptions of the DOE performed at least by a factor of two.

The variable with highest influence on the effective damping performance is the dynamic accuracy of the measurement system, which turned out to be much more accurate than the conservative assumptions made in the DOE.

TESTING OF ETVE DURING LNG PLANT COMMISSIONING

The eTve system was tested during commissioning of a LNG plant, the application of such solution has been as a backup solution in case of development of unexpected high torque response, to improve the damping performance of both drive trains simultaneously. A basic system block diagram of the two trains and the eTve control hardware applied during site commissioning is shown in Figure 10.

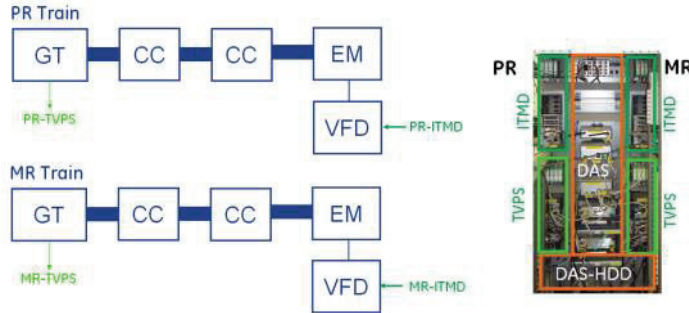


Figure 10: LNG System including PR and MR Train. TVPS and TMD control cabinet, together with additional Data Acquisition System (DAS), shown on the right side.

The additional control hardware needed for eTve is comparable small. Most of the space inside the cabinet was used by the signal monitoring and data acquisition system. This equipment is typically only needed during plant commissioning and can be removed afterwards.

MEASUREMENT RESULTS

Figures 10-14 show characteristic measurement results recorded during the LNG plant commissioning. Figure 11 is a similar case as simulated in Figure 9. Here, a β -angle modulation of max. 1° was applied (green curve). The resulting β -angle of the machine side (inverter) bridge is shown in the bottom. It oscillates between about 150° and 151° , because of the superimposed $\Delta\beta$ signal. This results in a unidirectional modulation towards lower firing angles. The damping power generated by means of this modulation is sufficient to totally suppress the angular vibration (red curve) that is caused by air-gap torque harmonics at critical speed (around 3512 rpm – see Figure 3).

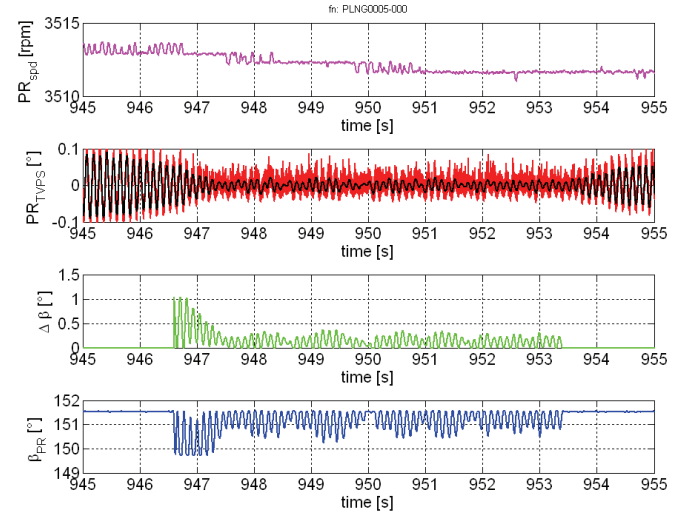


Figure 11: Suppression of a torsional vibration measured at the PR train (eTve activated during time interval $946.5 \text{ s} < t < 953.5 \text{ s}$)

The same effect of eTve could be observed at any speed in the nominal operating range where torsional vibrations occurred. Figure 12 shows another example where the PR train was operated close to a critical speed and eTve was switched on for six seconds, switched off for ten seconds, and then switched on again, to demonstrate the strong torsional damping effect of eTve despite modulating the beta-angle by less than 2° in this example.

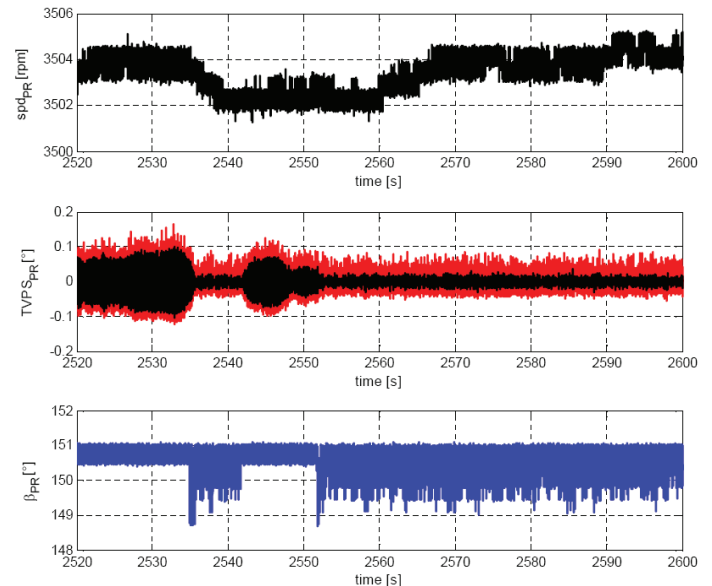


Figure 12: Operation at around critical speed

The benefit of using eTve control becomes even more obvious when monitoring the angular vibration level of the compression train in a longer time frame and comparing the rotational performance of the VSD driven train (with and without eTve)

with the performance when the drive train is only mechanically driven, by means of a FR7 gas turbine. This measurement result is shown in Figure 13. Initially, the compression train is only gas turbine driven. With additional drive power from the VSD, starting shortly before 4pm, the torsional vibration level increases (well within maximum allowed value). But it falls below the vibration level achieved with mechanical drive only during time windows when eTVe is activated. Figure 13 shows that eTVe can reduce the torsional vibration level caused by interharmonic excitation through the VSD (Load Commutated Inverter) below the values achieved when the drive train is only gas turbine driven. This is an important result because it shows that eTVe enables to achieve smooth rotational performance of compression trains driven by standard VSDs, such as LCIs, which produce electrical waveforms with a comparably larger harmonic content. These results have been confirmed by continuous monitoring during continuous eTVe testing all over the commissioning of the LNG plant in all operating points. Also shown in Figure 13 is the mechanical limit for angular vibrations. The vibration level during time periods when eTVe control is not activated is well within the mechanical vibration limit. Therefore eTVe was only tested during the commissioning period but not continuously applied to improve the damping performance.

Another measurement result, showing single and simultaneous application of eTVe while both compression trains are in operation (MR and PR train), is shown in Figure 14.

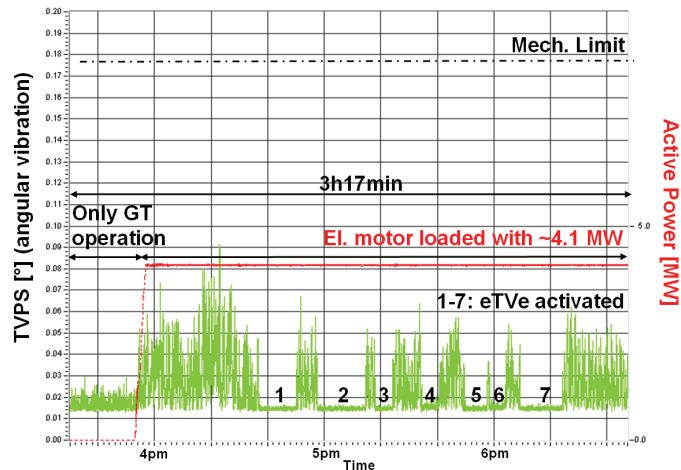


Figure 13: Residual torsional vibration level when PR Train is only GT driven and VSD driven (with and w/o eTVe)

Again, the time periods when the measured angular vibrations don't exceed 0.02°, is when eTVe was switched on. Without eTVe those angular vibration reached up to 0.1°, depending on the operational speed (a value below train components mechanical limits). This type of control has a very positive effect on the torsional behavior of the compression train and negligible impact on the electrical system, as will be explained

in more detail by the measurement examples below. Figure 15 shows a zoomed view of activating eTVe while both compression trains are operated at a critical speed.

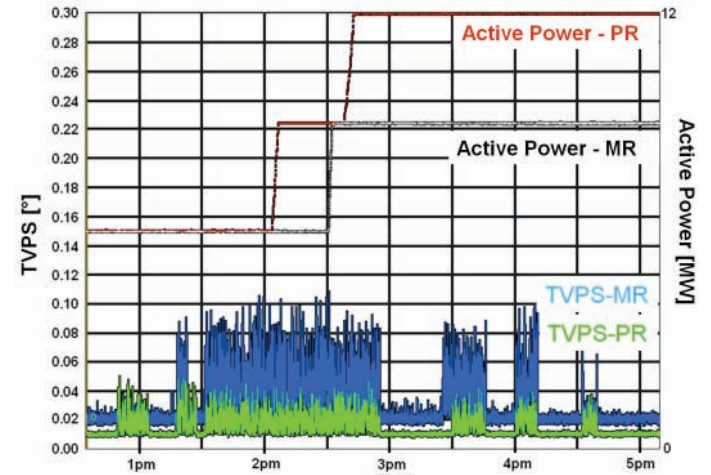


Figure 14: Parallel operation of both compressor trains with different loading levels on the electric machines, with and without eTVe (independent from each other).

Both torsional damping control loops are completely independent from each other, as shown in Figure 10, and have the same block diagram shown in Figure 7. The eTVe control for the MR train is activated at $t = 125.1s$, the one for the PR train at $t = 127.6s$ (Figure 15). In both cases, the maximum beta angle modulation was limited to 1°.

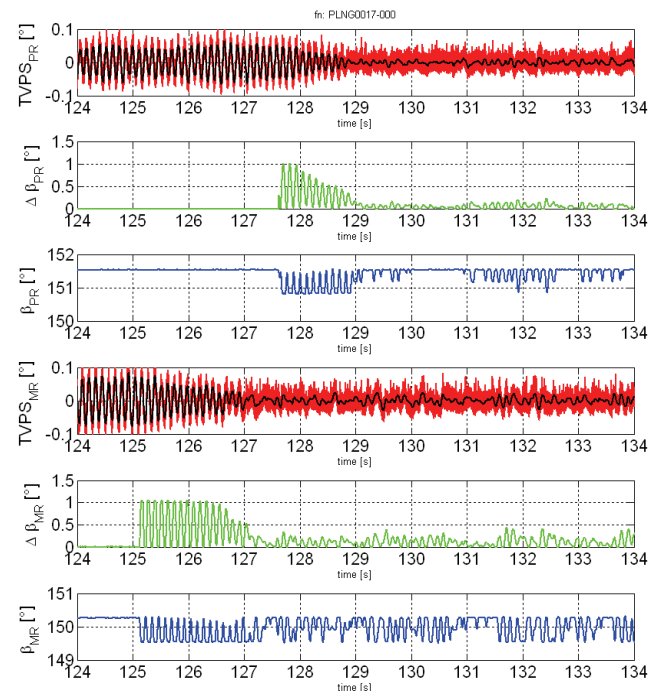


Figure 15: Successive enabling of eTVe for both drive trains, showing suppression of relevant torsional TNF (red curves)

Figure 15 shows that a one degree beta angle modulation around 151.5° (PR) respectively 150.3° (MR) was sufficient to suppress torsional vibrations in this application. Modulating the beta angle by a few degrees in a unipolar way, as shown here, has negligible effect for the electrical drive system. The one degree beta angle modulation is not even visible in the measured DC link current of the LCI, as can be seen in Figure 16.

Although the electronic damping system effectiveness has been proven beyond the authors' expectations, its application was not necessary in this specific application, because the compression train torsional oscillations without eTve turned out to be well within the mechanical limits. Nonetheless eTve has demonstrated to be a reliable solution to decrease risk of unfavorable electromechanical torsional interactions for challenging future applications, e.g. larger VSDs, longer shaft lines, multi shafts/multi geared applications, etc.

POWER SYSTEM RESPONSE

Besides increasing the torsional stability of the compression trains, eTve control also had a stabilizing effect on the grid frequency of the local power system which was an island power system powered by gas turbines. Both large VSDs exchange interharmonics through the local power system. But the nuisance effects of this interaction was lower while eTve control was activated, which can be explained by the improved damping performance of both compression trains.

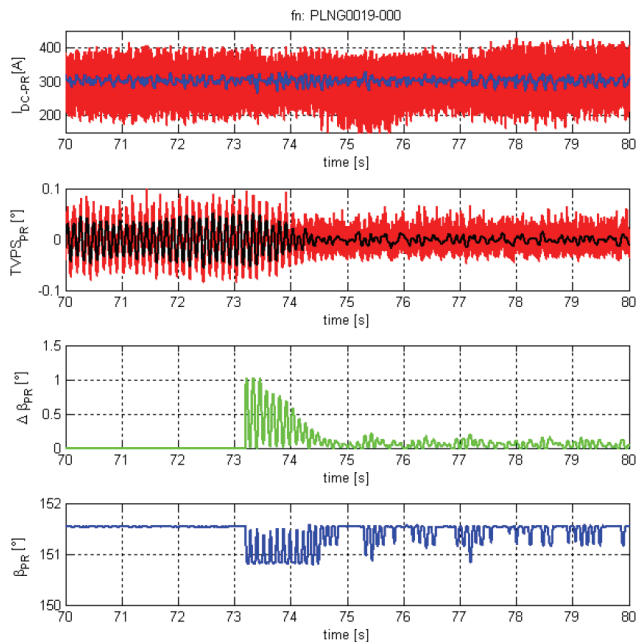


Figure 16: Effect of eTve control on DC link current (uppermost curve)

CONCLUSIONS

Variable speed electric drives enable increased operational flexibility and energy efficiency. Issues with variable speed driven systems may arise from the generation of air-gap torque ripple components due to electrical harmonics and the fact that very large synchronous motor-driven turbomachinery can have inherently low mechanical damping values.

An integrated electronic Torsional Vibration elimination (eTve) controller can decrease the sensitivity of large electrical machinery to harmonic torque ripple components and other disturbances. The damping control provided by eTve is typically based on input from well-proven and well-referenced standard instrumentation of the compression train, so that no additional specific sensors need to be installed. Torsional vibration sensing based on tip timing, as being used for monitoring of complex compression trains, has been successfully tested to deliver the right input signals for eTve control. It enables to completely eliminate torsional vibrations, below a level that is achieved when the compression train is only gas turbine driven. The torsional mode damping provided by eTve involves only low power, e. g. far below one percent of the nominal power of the electrical system. Thus eTve can be offered as a retrofit for existing drive system designs, e. g. in cases where torsional issues have already occurred or during string tests with particular complex shaft lines. Since it does not require changes to the existing drive or control system, it can be provided as an additional control system that can be switched on and off during operation.

The major advantage of eTve is that it enables torsional vibration-free performance of large compression trains with robust and reliable VFD topologies (thyristor and IGCT based). With eTve, smooth rotational performance can be achieved at minimum complexity and cost of the electrical drive system.

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REFERENCES

- [1] API Standard 617, 2002, "Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services", Seventh Edition, American Petroleum Institute, Washington, D. C.
- [2] IEEE Task Force on Harmonics Modeling and Simulation "Interharmonics: Theory and Modeling", IEEE

- Transactions on Power Delivery Vol.22, NO.4 October 2007
- [3] R.-D. Klug, A. Mertens, “ Reliability of Megawatt Drive Concepts”, 1CIT2003, Maribor, Slovenia, pp.636-641
 - [4] C. M. Ong, “Dynamic Simulation of Electric Machinery using Matlab/ Simulink”, Prentice Hall, Purdue, 1999
 - [5] J.-J. Simond, A. Sapin, M. Tu Xuan, R. Wetter, P. Burmeister, “12-Pulse LCI Synchronous Drive for a 20 MW Compressor - Modeling, Simulation and Measurements”, IEEE Transactions On Industry Applications, 2005, pp. 2302-2308
 - [6] M. Golebiowski, L. Naldi, V. Rossi, S. Ponticelli, “Train Torsional Vibrations: Monitoring System Based on Model Result”, Proceedings of ASME Turbo Expo 2010: Power for Land, Sea and Air, GT2010, June 14-18, 2010, Glasgow, UK
 - [7] V. Rossi, V. Depau, S. Ponticelli, “Torsional Vibration Protection System”, Proceedings of ASME Turbo Expo 2011 GT2011, 2011, Vancouver, Canada
 - [8] C. Sihler, S. Schramm, J. Song-Manguelle, P. Rotondo, S. D. Puglia, and E. Larsen, “Torsional Mode Damping For Electrically Driven Gas Compression Trains In Extended Variable Speed Operation,” No. 38, (Turbomachinery Laboratory, Texas A&M University, College Station, Texas), pp. 81–92, Turbomachinery Symposium, 2009.
 - [9] S. Schramm, C. Sihler, J. Song-Manguelle, and P. Rotondo, “Damping Torsional Interaction Effects of Large Drives,” vol. 25, pp. 1090–1098, IEEE Transactions on Power Electronics, 2010
 - [10] K. Tanaka, A. Adachi, N. Takahashi, Y. Fukushima “Torsional-Lateral Coupled Vibration of Centrifugal Compressor System at Interharmonic Frequencies Related to Control Loop Frequencies in Voltage Source Inverter” No. 38, (Turbomachinery Laboratory, Texas A&M University, College Station, Texas), Case Study # 14, Turbomachinery Symposium, 2009.
 - [11] J.A. Kocur, J.P. Corcoran “VFD Induced Coupling Failure” No. 37, (Turbomachinery Laboratory, Texas A&M University, College Station, Texas), Case Study # 09, Turbomachinery Symposium, 2008.