

## NEW APPROACH TO TORSIONAL VIBRATION MONITORING

**Lorenzo Naldi**  
 Engineering Manager  
 GE Oil&Gas  
 Florence, Italy

**Mateusz Golebiowski**  
 Rotordynamics Team Leader  
 GE Oil&Gas  
 Warsaw, Poland

**Valerio Rossi**  
 Remote Monitoring and Field Testing Manager  
 GE Oil&Gas  
 Florence, Italy

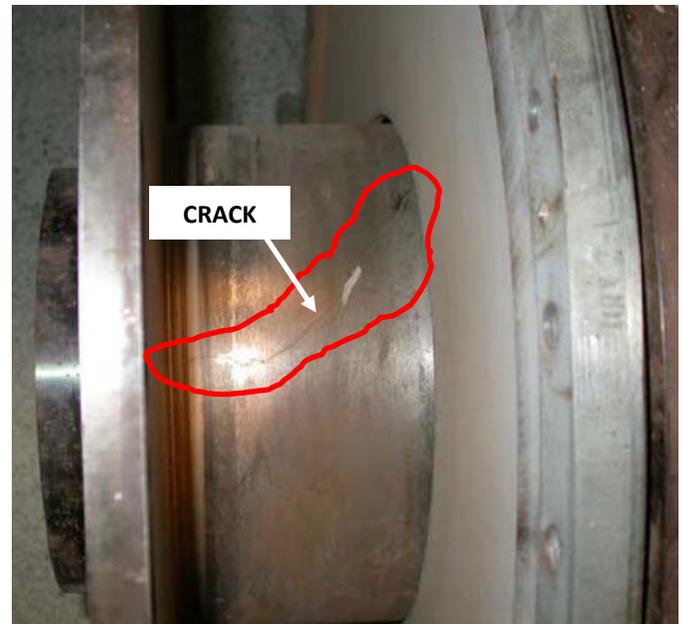
### ABSTRACT

This paper presents a summary of the authors' experience gained from implementation of model-based monitoring systems for LNG string vibration. The development of a novel train torsional vibration monitoring approach utilizing system modal analysis and lateral-torsional coupling in geared systems has been reported. A number of recent Root Cause Analysis programs related to turbocompressor string operation problems (LNG production) have faced barriers due to lack of adequate torsional vibration monitoring solutions. As the machinery problems associated with abnormal levels of train torsional vibration become more frequent, the incentives for the Oil & Gas industry to address the need for reliable and trouble-free cyclic torque evaluation gain ground. This constitutes a demand for a robust, reliable and accurate torsional vibration monitoring approach acceptable as a practical solution.

Contemporary, industry-wide accepted measurement technologies (strain gage with telemetry, optical probes, laser) fit the test needs well but their lack of ruggedness and durability as well as problematic installation impacting machine operation do not permit their consideration for long-term monitoring applications. The authors present a methodology which allows evaluation of alternating torque amplitudes in different string sections based on standard instrumentation installed on the train. The approach adopted in this paper relies heavily on the modal representation of the dynamic system being analyzed. The calculation method described utilizes the results of such analysis along with the angular vibration signal obtained from standard instrumentation installed on the train (gas turbine speed pick-up signal spectrum) and typical radial vibration data from gearbox proximity probes (when available). This means that no additional probes need to be installed to get the results. The robustness and accuracy of the proposed techniques was validated in the laboratory and benchmarked against test data from several LNG production train measurement campaigns, where the alternating torque was measured by means of standard measurement systems. The authors show the results of this verification together with prospective system applications. The symbols and abbreviations used in this paper are given in the last section.

### INTRODUCTION

Torsional vibration refers to oscillatory torsional deformations of the machine rotor sections. For large rotating machinery (Oil & Gas mechanical drive or power generation), the mechanical system will consist of several rotors that are joined by relatively flexible shafts and couplings. Each rotor in the system will oscillate following a torsional disturbance resulting in twisting in the shafts and to a lesser extent in the large diameter rotor bodies themselves. These twisting motions following severe torsional excitations may be sufficient to cause fatigue or overtorque-related damage to machine couplings and other components (Fig.1, Fig.2).



**Figure 1: Failed coupling (spacer crack). Crack angle (approx. 45deg) indicates a torsional fatigue failure.**

The torsional characteristics of the entire train are carefully studied during the design phase and investigated on a test stand, especially for LNG applications or when either a helper motor or generator is present.

The amplitudes of motor air gap torque harmonics that are generated are typically low and presence of a power electronics driven electric machine doesn't have to lead to substantial torsional excitation. Normally for LCIs the observed noninteger ripple components are lower than 1% of nominal torque (in steady state operation). The VSI type inverters (PWM) run even smoother. Progress in this area during last decade is significant (Integrated Torsional Mode Damping technology, Direct Torque control algorithm, etc.). On the other hand, the wide operating speed range, low system damping (modal damping values are usually very low and  $Q=50$  is a typical estimate used in forced response analysis) can turn this small AGT ripples into shaftline components failures. The coupled electro-mechanical study approach in train design phase is becoming more common today. It is a necessary effort to assure that no dangerous closed-loop interactions during start-up will be present (especially at low speed when high power helper motor is used to start the train). The improper setting/tuning of some control parameters inside the frequency converter (during electrical commissioning) may also lead to system stability problems (closed-loop, mechatronic interactions) which will result in continuous steady-state forcing at unacceptable levels.

A significant aspect related to the train torsional vibration phenomenon is that high levels of such vibrations can be present in the machinery for a long time. During this time, the operation of the machine can appear to be smooth and trouble-free, giving no warning before part of the rotating equipment becomes damaged.

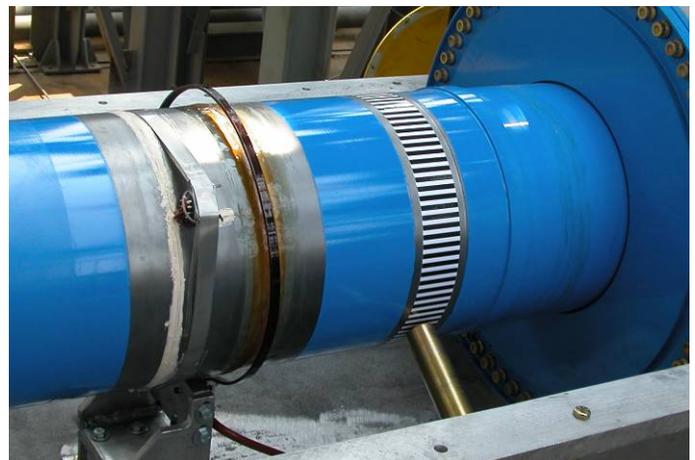


**Figure 2: Turbine/Axial Compressor coupling flange deterioration resulting from high torsional vibration (fretting marks visible)**

The concern for TV is growing among both OEMs and customers. The authors have faced a number of failures on units already in service (mechanical drive and power generation). Additionally, some recent Root Cause Analysis programs related to turbocompressor string operation problems (mainly in LNG production) have faced barriers due to the lack of

adequate torsional vibration monitoring. As an abnormal level of torsional vibration was identified as the most plausible root cause of the problem, its investigation was urgently needed. The standard measurement techniques were not considered to be applicable due to production impact, substantial test lead-time and significant costs. The most popular devices that are currently available to measure torsional vibration are strain gage (with telemetry system), rotary shaft encoder, laser vibrometer, optical sensors (Fig.3), and contactless inductive sensors. Feese and Hill (2009) provide comprehensive overview and explanation of these techniques and its limitations. These contemporary, torsional vibration evaluation technologies, which are accepted industry-wide, fit the test needs well but have a number of drawbacks. The most important aspects that must be considered and overcome, from the monitoring application viewpoint are: problematic installation impacting machine operation, lack of ruggedness and durability, cost and safety issues.

The authors' objective was to develop a methodology that would match the standard testing techniques (strain gage with telemetry, optical probes, laser) in terms of alternating torque assessment capabilities, but with a different application philosophy (long-term monitoring potential).



**Figure 3: Strain gage with telemetry system installed on flexible coupling's spacer (visible also, "zebra" tape installed for non-intrusive system development purposes)**

This leads to a set of monitoring/testing solution features that are essential to meet the needs of today turbomachinery business:

- unit shut-down not needed for monitoring system installation (minimized impact on machine operation)
- additional probe deployment not required (design modification costs and time saving)
- no hazardous area concerns
- continuous measurement capability (no limitations regarding temperature, speed or measurement time)
- insight into every critical location of the shaft-line available
- reduced lead time before monitoring/test
- reduced costs

Many different attempts have been made by the authors and others involved in machinery dynamics to identify a system that would satisfy all of these conditions. There exist several techniques that facilitate the identification of torsional vibration problems, which can do the job to some extent. The primary concepts exploited here are: lateral-torsional coupling in geared systems and electro-mechanical coupling in electrical drive structures (Fig.4).

Observation of gearbox radial vibration frequency components can give an indication of some abnormality and increase in torsional vibration levels (Tanaka, et al (2009); Kita, et al (2007)). The first section of this paper gives a summary of the experience achieved by the authors in the study of the rotordynamic behavior of geared trains with the goal of defining a methodology to analyze torsional stresses adopting industrial proximity probes (installed on the gear).

Since it has been demonstrated that, at this moment, the above methods (gearbox radial vibration and electrical signal monitoring) do not allow the definition of general transfer functions that would be applicable to a wide class of geared train configurations in a full range of operating conditions, a research and development effort was initiated to introduce and validate a novel set of techniques, which are referred to as the *smart investigation methods*. This term encompasses the deployment of new instrumentation, a software platform and model based algorithms.



**Figure 4: Current probes installed on converter busbars. Rogowski coil sensing technology is commonly used to measure ac currents (inverter output, motor side).**

The accuracy of these methods was validated in the lab and benchmarked against test data from several LNG production train measurement campaigns (where the alternating torque was measured by means of standard measurement systems).

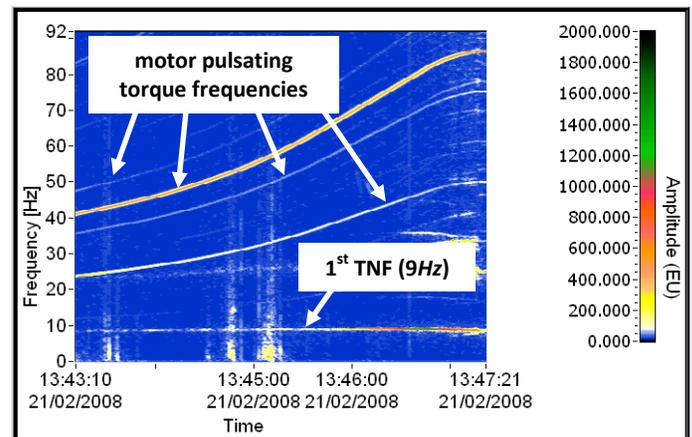
#### GEARBOX RADIAL VIBRATION MONITORING

Torsional and lateral coupled vibration in geared rotor systems has attracted much attention, both from an academic and an

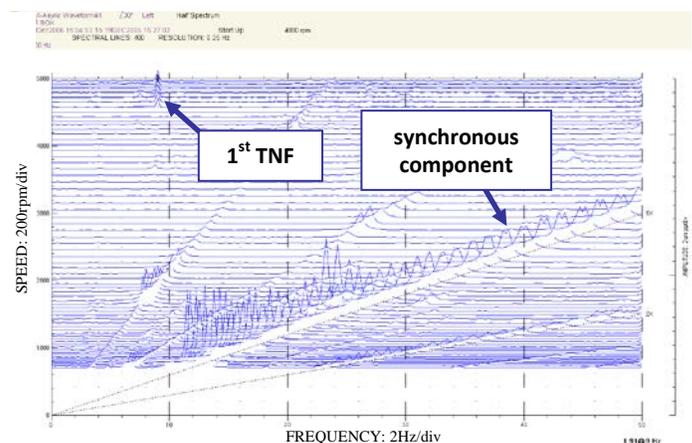
industrial point of view. To date, a great deal of analytical work has been done in order to provide a better understanding of lateral and torsional interaction and it has been widely demonstrated that torsional and lateral motion are coupled inside the gearbox due to the offset of the of the shaft centerline.

Many authors have developed mathematical formulations to predict critical frequencies, taking into account torsional-lateral coupling, which have been confirmed by good matching with field data. Wachel and Szenasi (in a work published in 1980) suggested the use of proximity probes to measure lateral vibrations in the fourth stage compressor as well as the lateral vibrations in the gearbox. The authors discovered a dominant subsynchronous vibration and noticed that the frequency of these lateral vibrations coincided with the third torsional natural frequency of the compressor train.

A similar observation from a GT-GB-GEN train is highlighted below. The acquired data show an alternating torque of 800 Nm at no load condition at the first TNF (Fig.5) with an HSS radial vibration level of 2-3  $\mu\text{m}$  at the same frequency (Fig.6). A possible correlation of roughly 260-400  $\text{Nm}/\mu\text{m}$  has been outlined.



**Figure 5: Strain gage waterfall (torsional vibration measurement)**



**Figure 6: HSS bearing cascade (no contact probe, lateral vibration)**

In order to investigate this phenomenon in detail in the context of the potential use for torsional vibration monitoring, a decision was made to conduct several test campaigns on different trains to measure lateral vibration, torsional stress and electrical parameters from the electrical machine.

The standard experimental setup was arranged to capture the full picture of the train dynamics (taking into account that the tests were done on units in service). Regarding lateral vibration, the train was fully instrumented with normal non-contact probes. Additional velocimeters and axial probes on the gas turbine were also available. The electrical signals of the electrical machine were acquired by means of dedicated probes connected inside the VFD control panel. The torsional stress measurement system was based on a strain gage torque meter technology (90° standard full bridge rosette applied in the 45° standard configuration for torque measurement). This data acquisition system was able to capture torsional stress data up to 86 kNm (maximum range) and the sensitivity of this instrument, coming from calibration, was 8.6 kNm/V.

The objective of these tests was to excite the gearset dynamics at different train speeds and VFD power levels (different transmitted torque levels). The response that was acquired allowed a trend that correlates the alternating torque at a given frequency (1<sup>st</sup> TNF, 2<sup>nd</sup> TNF) and radial vibration on the gearbox (same frequency component) to be identified. The graphs below depict such a correlation between the radial vibration amplitudes (at the gearbox bearings) and the pulsating torque at the first and second TNFs respectively.

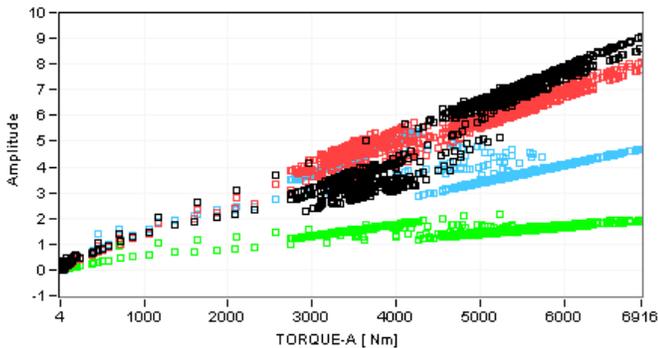


Figure 7: Torsional trend (1<sup>st</sup> TNF component)

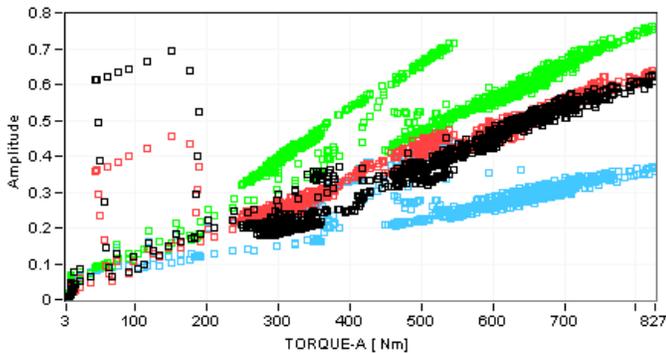


Figure 8: Torsional trend (2<sup>nd</sup> TNF component)

It was noticed that the best fitting curve is achievable when one sub-divides the measurements into 2 different groups with respect to the static load provided by the helper motor (Fig.9, Fig.10).

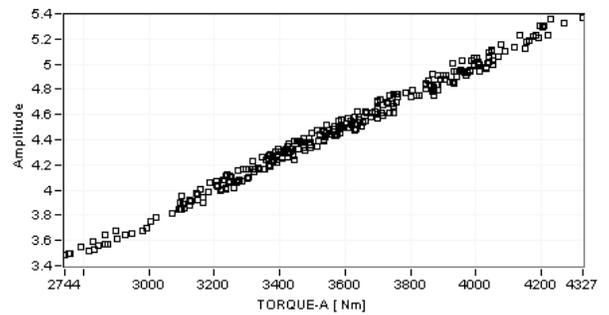
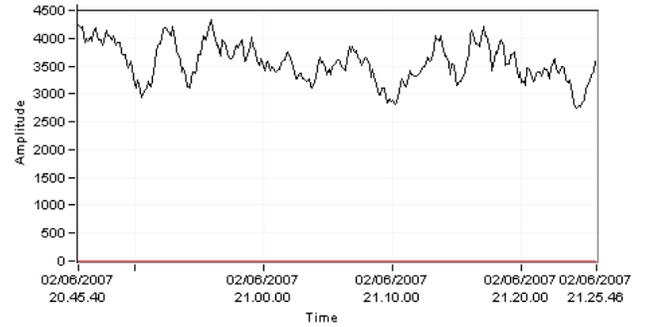


Figure 9: Torsional trend for static transmitted load < 50% (found scaling factor = 800Nm/μm)

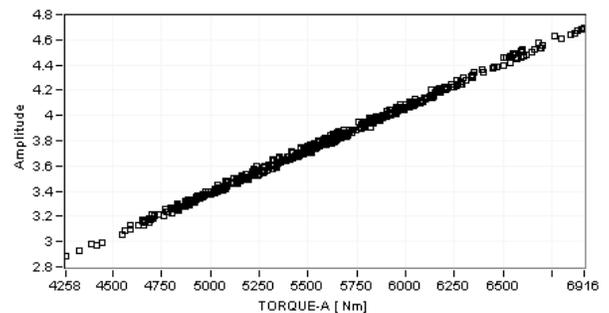
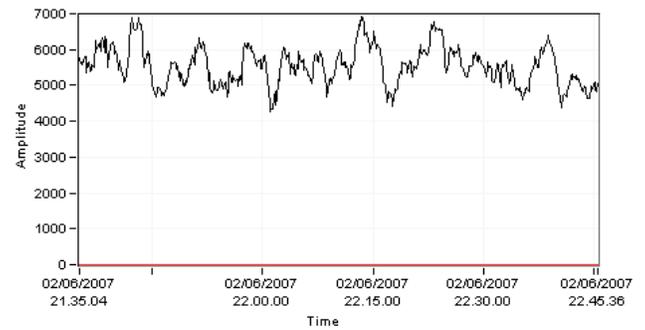


Figure 10: Torsional trend for static transmitted load > 50% (found scaling factor = 1440Nm/μm)

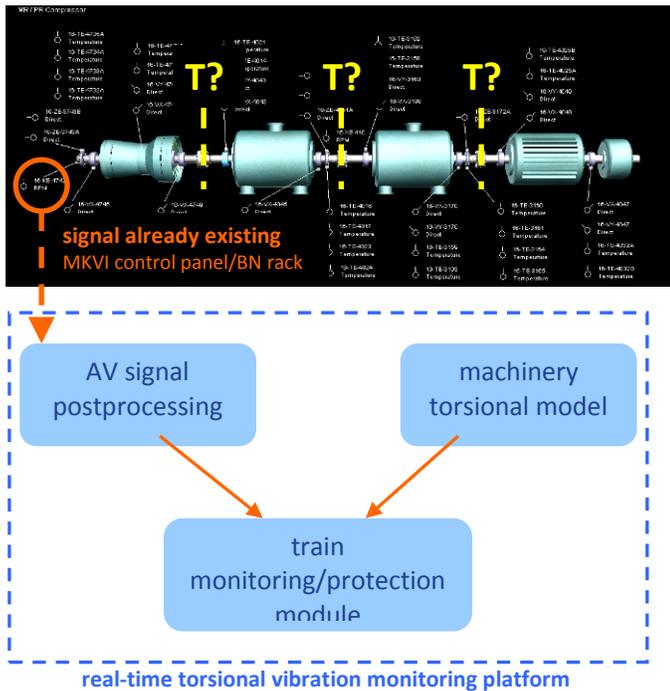
This is a result of the fact that loading the gearbox affects the train's dynamic behavior (different vibration response). These curves, which were defined for all of the tested configurations, were also compared with each other. All the linear regression functions, have the coefficients with the same order of magnitude (between 0.007 and 0.003) with  $R-Sq > 82\%$ , showing good agreement between the different system.

Despite the encouraging results, at this time it is not possible to define a unique model valid for different train configurations that allows calculating the torsional stress based on lateral vibration measurements only. The correlation must be calibrated case by case. It is possible to implement a monitoring system if this calibration can be done in the commissioning phase. Additional research effort is required to extend the database and validate the results obtained on a wider class of train configurations (trains without a GT).

**SMART INVESTIGATION METHODS (SIM) DEVELOPMENT**

A demand for a torsional vibration monitoring approach, which is sufficiently robust, reliable and accurate to be acceptable as a practical solution, triggered the development of a new system.

The *smart investigation methods* refer to the deployment of new instrumentation, software and modal calculation results. The combination of these allows the evaluation of alternating torque levels in different machine sections based on the angular vibration signal obtained from standard instrumentation installed on the train. This means that no additional probes need to be installed to obtain the results. It will also be shown that with this new approach, insight into every critical location of the shaft-line is available (Fig.11).

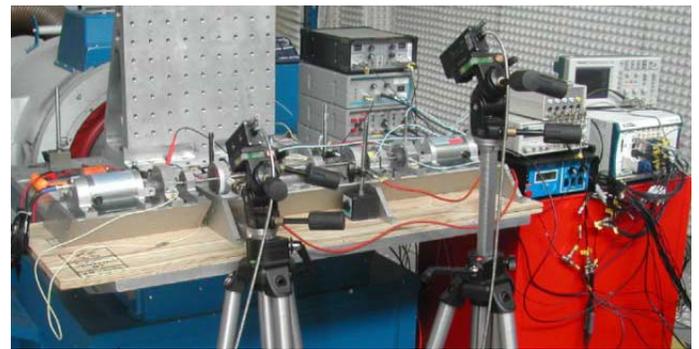


**Figure 11: Proposed monitoring approach scheme**

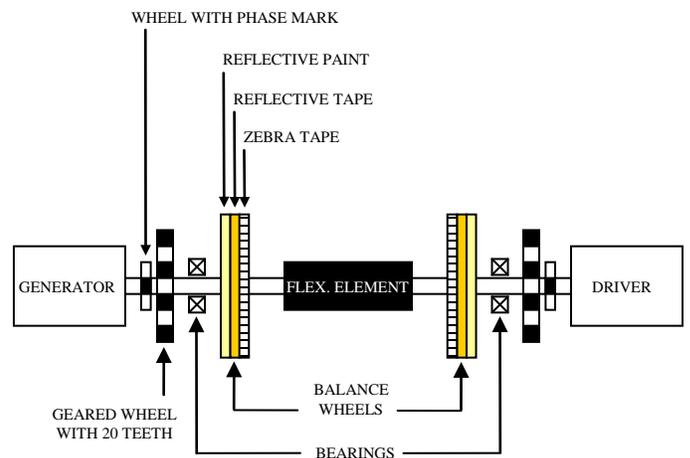
Flexible element deformations, as well as the absolute value of the angular vibration, are suitable quantities to be measured for torsional vibration level assessment. As the prospective system had to be absolutely non-intrusive (avoiding installation of instruments on the rotating parts or mounting/dismounting of coupling guards, covers or any other component), only two kinds of signals could be considered. These are signals that are already available in train control/monitoring loops and signals from non-contact sensors.

The technique that fit these requirements best is to capture the shaft angular velocity oscillations about the average rotational speed of the rotor.

Extensive tests were carried out to verify the state-of-the-art solutions available on the market and to understand the limitations of typical signal processing algorithms that can provide angular vibration spectra based on speed pick-up or keyphasor signal. For the purpose of testing of this new instrumentation and preliminary comparative analysis a dedicated software and torsional vibration simulator were developed (Fig.12, Fig.13).



**Figure 12: Torsional vibration simulator**



**Figure 13: Torsional vibration simulator layout**

The first TNF of the simulator is 9.45 Hz, which is within the range of typical first TNF values of the compression trains under investigation. The torsional excitation is introduced by means of a transistor system and a function generator.

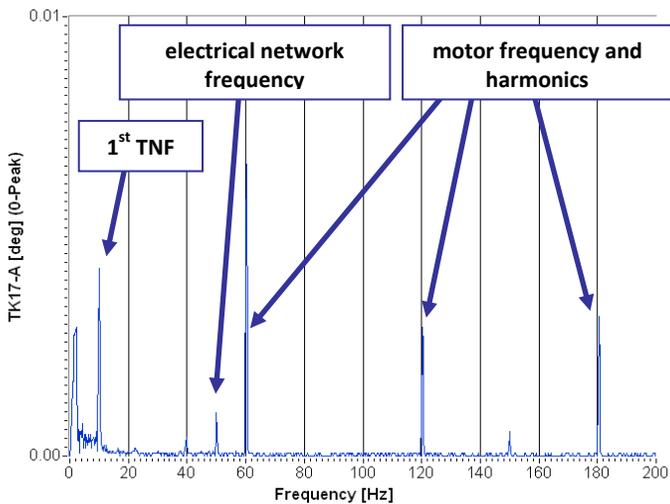
Three different sensor types were installed in the rig:

- standard eddy current probes (targeting the geared wheel, bolts, and phase mark)
- optical probes (targeting the "zebra" tape)
- laser Doppler probe (targeting stainless steel, reflective tape, and paint)

The eddy current probe appeared to be the best candidate for on-site measurement because of its durability and very high rejection of electrical noise. The output signal from the eddy current probe can be processed in different ways. The results presented in this paper were obtained with two different methods.

#### Signal conditioner output postprocessing

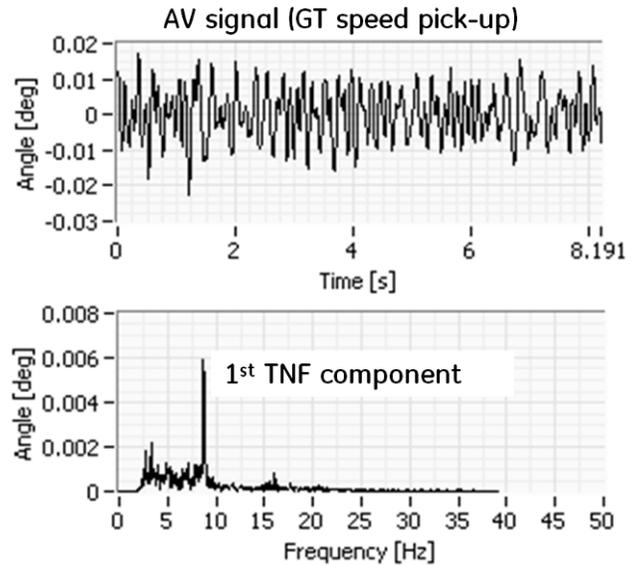
One choice is to deploy analog signal conditioners dedicated for torsional vibration measurement (for example Bently Nevada TK-17). TK-17 provides the AV signal already scaled in degrees, which can then be directly connected to a spectrum analyzer. Figure 14 is an example of BN TK-17 output signal frequency spectrum.



**Figure 14: Angular vibration signal (frequency domain) – BN TK-17 output**

#### Transducer signal postprocessing

In the second method, referred to as the "software method", one directly acquires and analyzes the raw transducer signal. The in-house developed real-time software identifies the tooth tip passing time (based on voltage thresholds fixed by the user). The instantaneous rotational speed is calculated as the tooth-to-tooth angle divided by the tooth-to-tooth time. The speed signal is then integrated to the angular vibration waveform, which is filtered and re-sampled to obtain vibration values uniformly distributed in time. Re-sampling is performed with respect to the requested spectral bandwidth.



**Figure 15: Angular vibration signal based on 60x encoder (time and frequency domain) – "software method"**

Both methods were examined in laboratory tests. It was demonstrated that the "software method" gives practically the same results as the analog signal conditioner when targeting a geared wheel with 20 teeth. The advantage of the "software method" is a wider applicability region (use of keyphasor signal) as there are no limitations regarding the minimum number of events per shaft revolution (20 for BN TK-17). The problem however, arises when the tooth passing frequency decreases below a certain level. The angular vibration amplitudes estimated by the software are attenuated. The magnitude of this effect is related to the vibration frequency of interest, shaft speed and number of pulses per revolution (tooth passing frequency). The computer simulation developed provides correction factors that must be applied to obtain the correct AV amplitudes.

The frequency spectrum of the shaft torsional oscillations will usually show most response in the lower order torsional modes (Fig.15), with some components at the harmonics of the electrical drive or network frequencies (Fig.14).

Since the laboratory investigations validated the AV measurement accuracy, the next steps were taken to use this data most efficiently to evaluate the alternating torque levels in different machine sections.

The following part of this section will address the calculation technique which incorporates angular vibration information into the existing modal model of the turbomachinery mechanical system.

Based on the calculated mode shape (or steady state forced response analysis result for electrical drive excitation) one can calculate the coupling participation, defined as follows

$$CP(\omega) = \frac{\alpha}{\delta} \quad (1)$$

where:

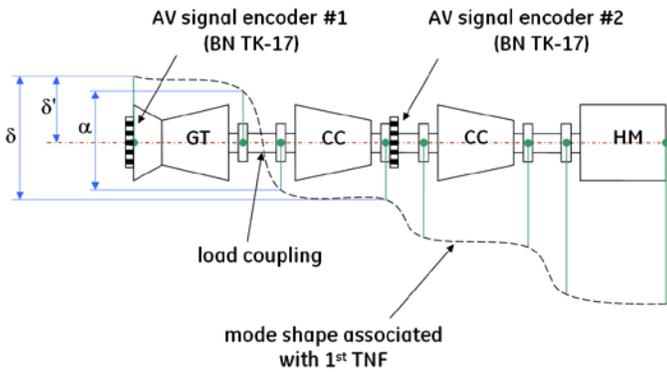
$CP(\omega)$  coupling participation  
 $\alpha$  twist angle of a coupling  
 $\delta$  twist angle (2 different sections of a machine)

This value describes the twist angle amplitude of the coupling related to the twist angle between two sections of the machine where AV probes are available.

An alternative formulation, relevant to the situation when the AV measurement is available only in one location, reads

$$CP'(\omega) = \frac{\alpha}{\delta'} \quad (2)$$

where  $CP'(\omega)$  is a coupling participation (for single plane measurement) and  $\delta'$  is AV amplitude at frequency  $\omega$ .



**Figure 16: Typical turbocompressor string with 1st TNF mode shape indication and load coupling twist value**

Having the coupling torsional stiffness  $k$  and the angular vibration signal with the frequency of interest filtered one can calculate the section torque amplitude (pk value).

$$T(\omega) = (CP(\omega) | AV1_f \pm AV2_f |)k \quad (3)$$

where:

$T(\omega)$  dynamic torque amplitude  
 $AV1_f$  Angular Vibration (in section #1 at freq.  $f$ )  
 $k$  coupling torsional stiffness

Similarly, when the AV measurement is available only in one location, the section torque amplitude can be calculated as

$$T'(\omega) = (CP'(\omega)AV1_f)k \quad (4)$$

For example, a typical compression train torsional mode shape (deformation predominantly in couplings at first natural frequency) the ratio between load coupling twist angle and GT shaft AV amplitudes (on suction side) is  $CP'(\omega_{nat})=2.6-3.2$ . For a value from this range a dangerous level of alternating torque of 50 kNm in GT coupling section is corresponding to 0.17 deg

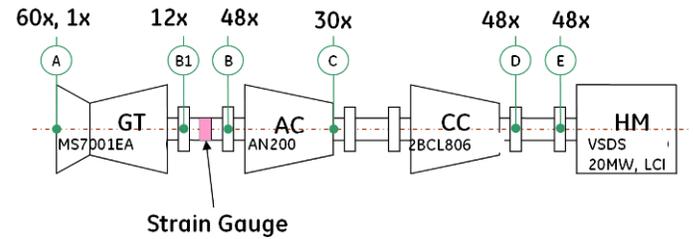
of vibration amplitude in speed pickup plane, which can be set as an alarm limit.

## SMART INVESTIGATION METHODS (SIM) FIELD IMPLEMENTATION

The accuracy and reliability of the *smart investigation methods* was benchmarked against test data from several LNG production train measurement campaigns. The reference alternating torque measurements were based on standard measurement systems. The results of this verification are outlined in the sections below.

### Field experience #1 (Monitoring)

The case described here is an LNG mixed refrigerant compression unit test campaign, which took place in May of 2009. The overall test setup was organized to occur during the planned shutdown of the train, however, some signals used in the test are typically present in train control system loops (except for SG and non-contact probes on the coupling bolts) and were already available. The reference alternating torque signal source was a strain gage installed on the load coupling's spacer. Due to speed limitations of the strain gage and telemetry system, validation was possible only up to the cranking phase of the train start-up. Before fired start-up, the train had to be stopped and the strain gage system removed.



**Figure 17: LNG compression train (field experience #1)**

The angular vibration signal sources accessible were:

- A speed pick-up probe (60-tooth wheel) and keyphasor on turbine suction side
- B1 single eddy current probe on 12 spacer bolts
- B single eddy current probe on 48 spacer bolts (Fig.18)
- C geared nut on the axial compressor NDE side (30 teeth)
- D, E eddy current probes on 48 spacer bolts



Figure 18: Proximity probes on coupling flange bolts

The torsional mode shapes predictions are presented in Figure 19.

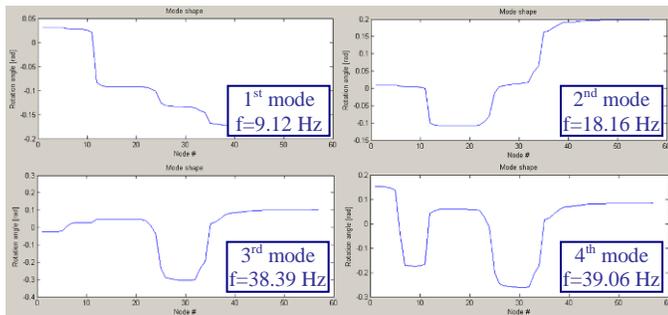


Figure 19: Train modal analysis results.

Figure 20 shows the captured load coupling section torque (strain gage measurement) related to a start-up when the train was accelerated to crank speed only by the helper motor. This type of plot format (waterfall) may be very useful in torsional vibration analysis and was extensively employed by the authors. The horizontal axis represents time, the vertical, frequency, and the colors denote the amplitude values. The train speed trend plot is complementary and positioned below the waterfall plot.

The observed frequency content (good matching of the first 3 TNFs with predictions) confirms the lumped parameter model, which was developed.

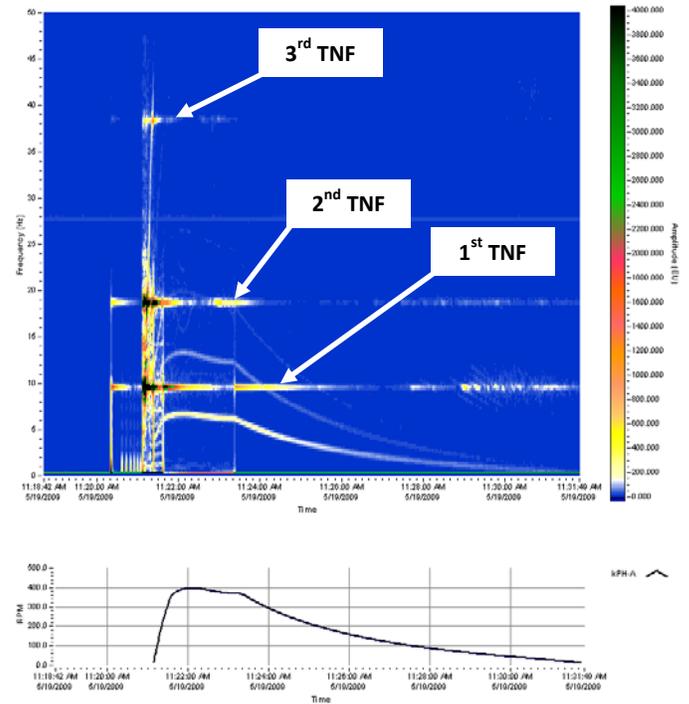


Figure 20: Strain gage signal waterfall (alternating torque in Nm pk)

The results of the alternating torque evaluation (TQ-Tip Timing) together with the strain gage readings (TQ-SG, used as a reference) are summarized below. For each point shown in the below plots (Fig. 21-24) the abscissa corresponds to the reference torque amplitude value (SG measurement) while the y coordinate equals the torque estimation based on AV signal spectrum. The location of these points with respect to the dotted line quantifies the estimation error. The presented findings basically fall into 2 categories: double plane and single plane evaluation.

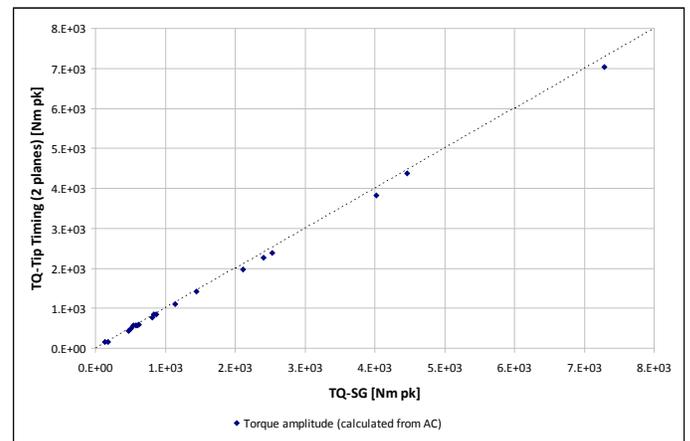
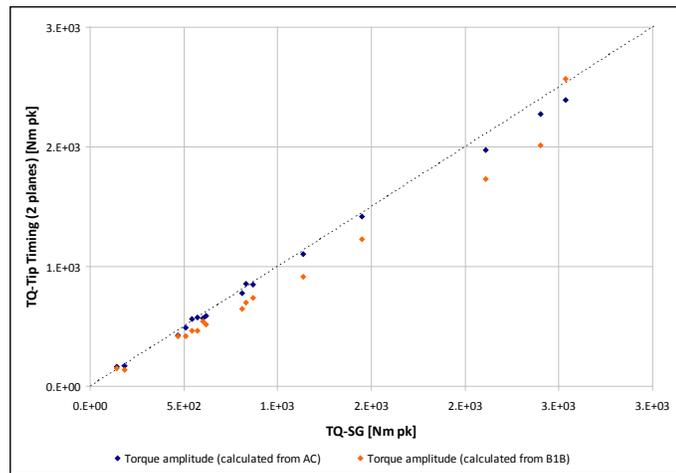


Figure 21: Calculated alternating torque amplitudes (1st TNF, double plane {AC})

The double plane calculation refers to the situation where one can measure a twist angle between two sections of the machine (2 AV probes are available). The second case is relevant to the situation where the AV measurement is available only in one

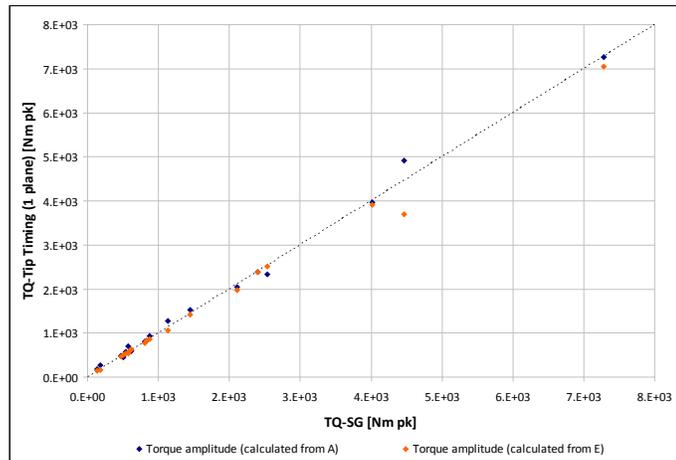
location (alternative formulation of the coupling participation  $CP'(\omega)$  should then be used).

The single plane approach is suitable for most turbomachinery applications (with encoder-based speed measurement, keyphasor signal, etc.).



**Figure 22: Calculated alternating torque amplitudes (1st TNF, double plane {AC}, {B1B})**

The double plane calculation at the 1st TNF resulted in underestimated torque amplitude values (mean 97.5%,  $\sigma=5.6\%$  utilizing planes A and C).

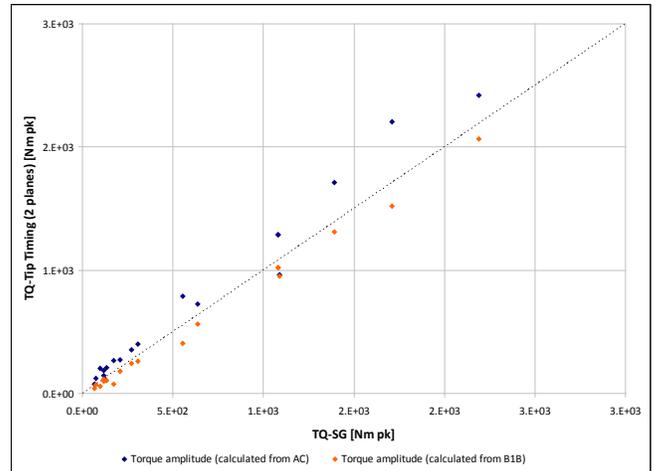


**Figure 23: Calculated alternating torque amplitudes (1st TNF, single plane A, E)**

The cyclic torque estimate, obtained using the AV measurement in only one plane, is also close to the SG values (mean 105.6%,  $\sigma=14.1\%$  for plane A and mean 96.5%,  $\sigma=4.1\%$  for plane E).

The lower accuracy of the 2<sup>nd</sup> TNF component prediction (mean 131.7%,  $\sigma=27.1\%$  utilizing planes A and C and mean 83.0%,  $\sigma=13.9\%$  for planes B1 and B) possibly has several sources. The lower quality of the AV readings in several locations at the frequency of interest is most likely the root cause. As the frequency of interest increases, it takes more

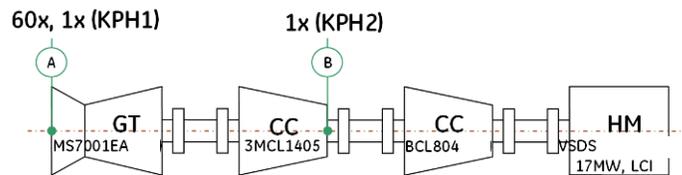
effort to properly post-process the tip-timing signal, mainly due to the attenuation effect.



**Figure 24: Calculated alternating torque amplitudes (2nd TNF, double plane {AC}, {B1B})**

**Field experience #2 (Protection)**

Another opportunity for the authors to validate the proposed technique was the on-site, torsional test of an LNG propane compression unit, which was carried out in March of 2009. Figure 25 shows the train arrangement together with the associated installed instrumentation outline.



**Figure 25: LNG compression train (field experience #2)**

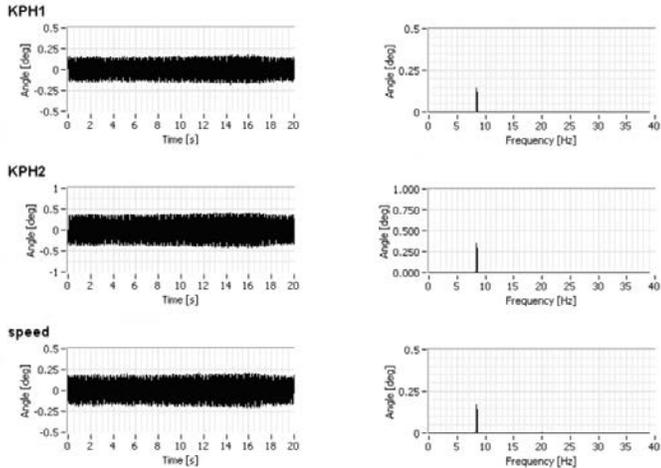
The angular vibration sources available in this case were:

- A speed pick-up probe (60-tooth wheel) and keyphasor on turbine suction side (KPH1)
- B spare keyphasor on the centrifugal compressor NDE side (KPH2)

The test circumstances were different from the previous example. Due to a load coupling (GT/CC) failure, a thorough investigation was started (within a dedicated RCA program). Since the abnormal level of torsional vibration was considered to be a potential cause of the coupling damage, torsional testing was scheduled followed by the installation of a dedicated protection system (Fig.28). As agreed with the customer, the torsional vibration level had to be estimated without any impact on plant operations. This naturally ruled out a train shutdown and the installation of standard measurement systems. The scope of the test operating conditions covered full speed train operation (3600rpm) with helper motor load steps (1MW

increments).

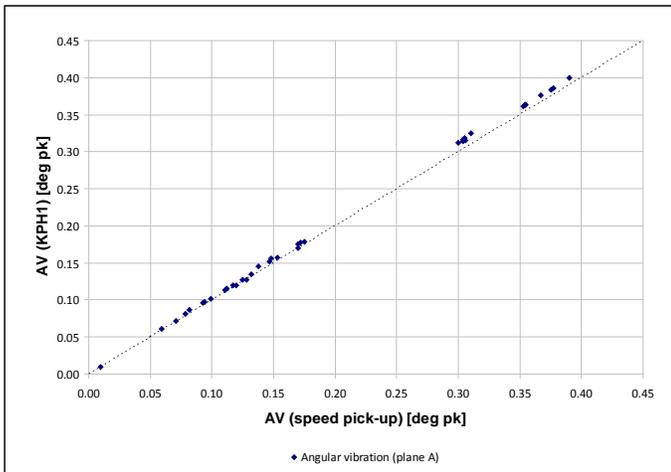
The use of a keyphasor as an AV signal source requires significant post-processing effort, which was undertaken by the signal processing routines embedded in the in-house developed software. Due to limitations (the required minimum number of events per revolution) of the analog torsional vibration signal conditioners (Bently Nevada TK-17), they could not be used in this situation.



**Figure 26: Obtained AV signals in time and frequency domain (12MW helper motor load)**

Validation of the angular vibration amplitude values (for the 1st TNF), which were extracted from the acquired phase reference signals, was critical to correctly estimating the torque. Good matching of this data with the speed sensor signal (based on a 60-tooth wheel), taken as a reference, is presented in Figure 27.

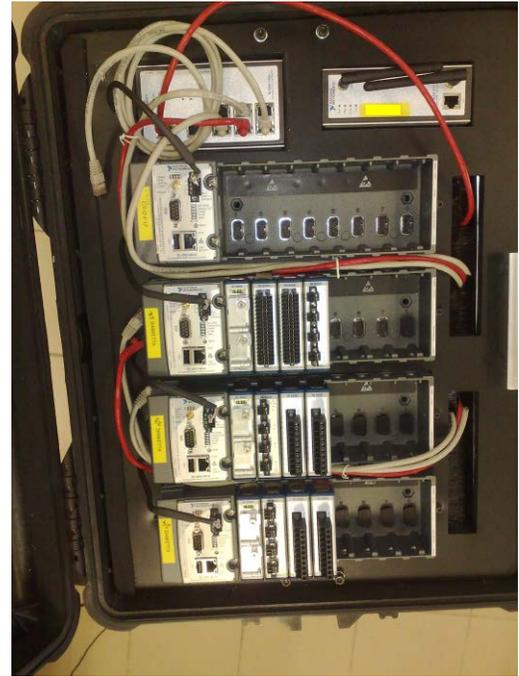
In this particular case, analyses were limited only to the 1st mode of torsional vibration, as the observed response of higher modes can be considered negligible.



**Figure 27: Validation of AV amplitudes extracted from KPH1 signals (1st TNF component)**

Figure 28 shows the standalone protection system that was installed on site. The interface contains 2 TVPS units (for 2

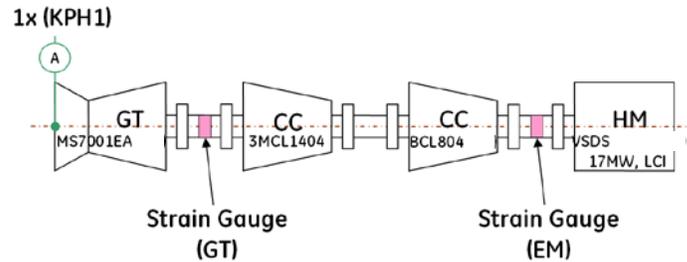
compression strings) based on CRIO technology. CRIO's collect the analog signals and send it to the master unit through EGD (ethernet cable). The master unit collects EGD data from TVPS's and from MVI and produces trends files stored on an USB disk.



**Figure 28: Standalone protection system's interface**

**Field experience #3 (Monitoring)**

Another example of the deployment of *smart investigation methods* comes from a string test of an LNG propane compression unit (Fig.29), which was performed in February of 2008.



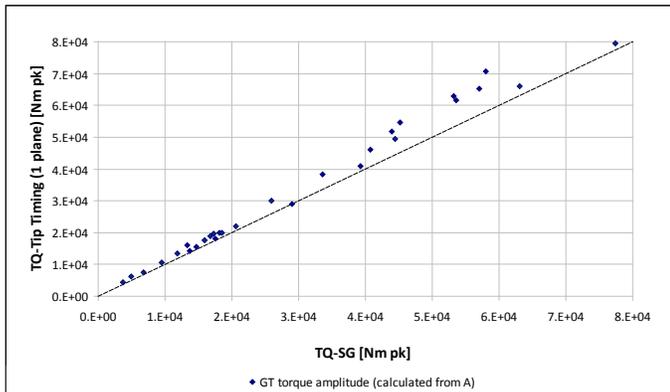
**Figure 29: LNG compression train (field experience #3)**

The only source of angular vibration signal used for estimating the alternating torque was the keyphasor on the turbine suction side (KPH1).

The goal was to benchmark the method, comparing the estimated torque values in the GT/CC and CC/HM coupling sections (referred to as GT and EM couplings respectively) with reference strain gage system measurements. The scope of the test operating conditions encompasses a broad range of train speed and helper motor load values (the full spectrum of

machinery operation conditions), which is suitable for thorough validation of the proposed calculation method. Before starting the tests the SG systems (four element, full bridge) were shunt calibrated with a resistance to simulate a strain of 201  $\mu\text{m}/\text{m}$  (micro-strain). Each of the four elements of the strain gage was shunt calibrated to demonstrate correct function.

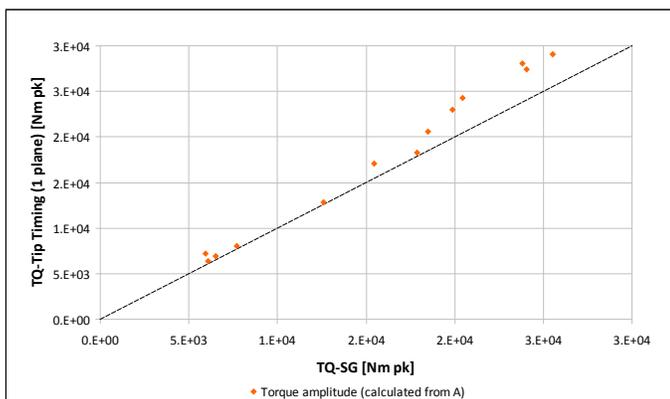
The results of the validation of the torque estimates are presented below.



**Figure 30: Calculated GT coupling alternating torque amplitudes (1st TNF, single plane A)**

Calculation of the alternating torque amplitude (1st TNF) in the GT coupling provided overestimated values (mean 112.5%,  $\sigma=7.5\%$ ). The maximum dynamic torque value recorded during this test is 77.4 kNm. The torsional excitation required to obtain this amplitude of response was calculated as 1.3 kNm which is approximately 2.9% of a nominal torque provided by the HM in FSFL condition. For LCI driven equipment the torque ripple components are typically lower than 1% of nominal torque (in steady state operation).

Correspondingly, the EM coupling section cyclic torque estimates are higher than the strain gage measurements taken for a reference (mean 111.7%,  $\sigma=6.8\%$ ).



**Figure 31: Calculated EM coupling alternating torque amplitudes (1st TNF, single plane A)**

This provided proof that the deployment of a train model can give equally accurate torque estimates for the sections located relatively far from the AV signal source (EM coupling section).

## CONCLUSIONS

The summary of the authors' experience gained from the development and implementation of model-based monitoring/protection systems for LNG string torsional vibration was presented.

It was shown that the combined use of a non-intrusive approach and model results (referred to as the *smart investigation methods*) allows reliably evaluation of the alternating torque levels in different machine sections based on the angular vibration signal obtained from standard instrumentation installed on the train. The number of necessary signal sources was reduced to one, which is very important from a practical standpoint, as it makes the approach suitable for most existing turbomachinery (already equipped with an encoder-based speed measurement system).

Several long term monitoring and protection systems employing the described approach are already installed on a number of critical units in LNG plants, providing valuable insight into the train torsional vibration.

Future work will be done to extend the experience, incorporating a wider class of machinery into the study.

## NOMENCLATURE

AC	Axial Compressor
AGT	Air Gap Torque
AV	Angular Vibration (signal)
CC	Centrifugal Compressor
EGD	Ethernet Global Data
EM	Electric Motor
GT	Gas Turbine
HM	Helper Motor
HSS	High Speed Shaft
KPH	KeyPhasor
LCI	Load Commutated Inverter
LNG	Liquefied Natural Gas
PWM	Pulse Width Modulation
RCA	Root Cause Analysis
SG	Strain Gage
TNF	Torsional Natural Frequency
TQ	Torque
TV	Torsional Vibration
TVPS	Torsional Vibration Protection System
VFD	Variable Frequency Drive
VSI	Voltage Source Inverter
$\omega$	rotational speed

## REFERENCES

- Wachel J.C., Szenasi F.R., 1980, "Field Verification of Lateral-Torsional Coupling Effects on Rotor Instabilities in Centrifugal Compressors", NASA Conference Publication No. 2147
- Okabe A., Kudo T., Yoda H., Sakurai S., Matsushita O.,

- Shiohata K., 2009, "Rotor-Blade Coupled Vibration Analysis by Measuring Modal Parameters of Actual Rotor", Proceedings of ASME Turbo Expo 2009, Orlando
- Lebold M., McClintic K., Campbell R., Byington C., Maynard K., 2000, "Review of Vibration Analysis Methods for Gearbox Diagnostics and Prognostics", Proc. 54<sup>th</sup> Meeting of the Society for MFPT, pp. 623-625
- Yang, M.-T., Griffin, J.H., 1998, "Theory and Methodology of Optimally Measuring Vibratory Strains in Closely Spaced Modes," Journal of Engineering for Gas Turbines and Power, Vol. 120, October 1998, pp. 801-807.
- Duncan N. Walker, 2004, *Torsional Vibration of Turbomachinery*, The McGraw-Hill Companies, Inc.
- Rossi V., Naldi L., Depau V., 2009, "Torsional Vibration in Rotordynamic Systems: Smart Investigation Methods" Proceedings of ASME Turbo Expo 2009, Orlando
- Naldi L., Biondi R., Rossi V., 2008, "Torsional Vibration in Rotordynamic Systems, Identified by Monitoring Gearbox Behavior" Proceedings of ASME Turbo Expo 2008, Berlin
- Watson M., Sheldon J., Amin S., Lee H., Byington C., Begin M., 2007, "A Comprehensive High Frequency Vibration Monitoring System for Incipient Fault Detection and Isolation of Gears, Bearings and Shafts/Couplings in Turbine Engines and Accessories" Proceedings of ASME Turbo Expo 2007, Montreal
- Chancey V.C., Flowers G.T., Howard C.L., 2001, "A Harmonic Wavelets Approach for Extracting Transient Patterns From Measured Rotor Vibration Data" Proceedings of ASME Turbo Expo 2001, New Orleans
- An Sung Lee, Jin Woong Ha, Dong-Hoon Choi, 2003, "Coupled Lateral and Torsional Vibration Characteristics of a Speed Increasing Geared Rotor-Bearing System", Journal of Sound and Vibration 263, pp. 725-742
- Wang P., Davies P., Starkey J.M., Routson R.L., 1992, "A Torsional Vibration Measurement System", IEEE Transactions on Instrumentation and Measurement, Vol. 41, No. 6, December 1992
- Den Hartog J.P., 1985, *Mechanical Vibrations*, Dover Publications, Inc., New York
- Nestorides E.J., 1958, *A Handbook on Torsional Vibration*, British Internal Combustion Engine Research Association (BICERA), pp. 84-88
- Swan P., Betts R.G., 2001, "Improved Vibration Monitoring on Aero-derivative Gas Generators with Internal Transducers" Proceedings of ASME Turbo Expo 2001, New Orleans
- Lim, T.W., 1993, "Actuator/Sensor Placement for Modal Parameter Identification of Flexible Structures" Modal Analysis: the International Journal of Analytical and Experimental Modal Analysis, Vol. 8, No. 1, pp. 1-13.
- Bently D.E., Muszynska A., Goldman P., 1991, "Torsional/Lateral Vibration Cross-Coupled Response Due To Shaft Anisotropy: a New Tool in Shaft Crack Detection," BRDRC Rep. No. 5, Bently Rotor Dynamics Research Corporation, Minden, NV
- Salhi B., Berthillier M., Lardies J., Voinis P., Bodel C., 2007, "A Subspace Method for Modal Identification of Bladed Assemblies Using Blade Tip-Timing Data" Proceedings of ASME Turbo Expo 2007, Montreal
- Ginsberg J.H., Wagner B.B., 2005, "On the Use of Modal Parameters as Metrics for Condition-Based Monitoring of Rotating Machinery" Proceedings of ASME Turbo Expo 2005, Reno
- Fraunhofer ITWM, 2007, "A Contactless Sensor for Online Monitoring of Torsional Oscillations"
- Golebiowski M., Naldi L., Rossi V., Ponticelli S., 2010, "Train Torsional Vibrations: Monitoring System Based On Model Result" Proceedings of ASME Turbo Expo 2010, Glasgow
- Feese T., Hill C., 2009, "Prevention of Torsional Vibration Problems in Reciprocating Machinery" 38<sup>th</sup> Turbomachinery Symposium, Houston
- Tanaka K., Adachi A., Takahashi N., Fukushima Y., 2009, "Torsional-Lateral Coupled Vibration of Centrifugal Compressor System at Interharmonic Frequencies Related to Control Loop Frequencies in Voltage Source PWM Inverter" Case Study #14, 38<sup>th</sup> Turbomachinery Symposium, Houston
- Kita M., Hataya T., Tokimasa Y., 2007, "Study of a Rotordynamic Analysis Method That Considers Torsional and Lateral Coupled Vibrations in Compressor Trains with a Gearbox" 36<sup>th</sup> Turbomachinery Symposium, Houston

## ACKNOWLEDGEMENTS

With sincere thanks and appreciation to the entire team that supported the authors, and especially to Valerio Depau, Stefano Ponticelli, Adolfo Agresta, Adam Borowa and Andrea Papini.