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TORSIONAL VIBRATION PROTECTION SYSTEM

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ABSTRACT

The increasing use of variable frequency drives (VFDs) and island mode power generation due to remotely located sites make these Oil and Gas industry plants vulnerable to sub-synchronous torsional interactions. These interactions can occur with synchronous generators supplying island-like power systems with large VFDs, but also in synchronous motor-driven turbomachinery, especially with increased nominal power and complexity of the drive train. The increasing number of site issues evidences the lack of suitable torsional vibration measurement systems. This paper describes applications of the TVPS, a device developed to address turbomachinery torsional vibration problems. The system answers the need for ruggedness and durability for long term monitoring and can be installed on any turbocompressor or turbogenerator unit without imposing additional requirements. TVPS applications experienced by the authors will be described in this paper, including coupling of the TVPS with an active torsional mode damping control successfully tested on an LNG production train.

INTRODUCTION

The increasing demand for site torsional vibration measurements on turbogenerator and turbocompressor sets has motivated the development of smart torsional vibration measurement methods.

The common method for torsional vibration measurements consists of the installation of strain gages on the machine, typically on the coupling. This method has proven to be very problematic for field applications because it requires a major effort to comply with safety rules and extended machine downtime for installation and removal. Furthermore, strain gages are not suitable for continuous operation and have a limited life due to the harsh operating conditions (temperature and centrifugal force).

The Torsional Vibration Protection System (TVPS) can process any signal related to the machine rotational speed, such as phase reference or speed signals, to obtain the instantaneous angular vibration value of the shaft. Torsional vibration values can be calculated from the angular vibration measured using modal analysis.

With this approach, the torsional vibration in any section of the machine can be determined by simply processing the speed signal of the machine. Furthermore, since the speed signal is a fundamental parameter for turbomachinery control, it is always available with redundancy.

Installation causes a negligible impact on the operation of the machine under investigation and can be done, with some precautions, even with the machine running.

Beyond the main function (angular vibration measurement), the TVPS has additional capabilities that allow integration with the machine control system, allowing machine monitoring and protection.

NOMENCLATURE AND ABBREVIATIONS

AV – Angular Vibration
CP – Coupling Participation Factor
f_n – Torsional Natural Frequency
f_s – Sampling Frequency
k – Torsional Stiffness
N – Number of Teeth
p – Pulse Rate
t – Time
T – Alternating Torque
θ – Angle
ω – Rotational Speed

CC: Centrifugal Compressor
DAS – Data Acquisition System
DCS – Distributed Control System
FSFL – Full Speed Full Load
FSNL – Full Speed No Load
GT – Gas Turbine
GB – Gear Box
GCPP – Generator Control Panel Protection
GD – Generator Drive
GEN – Electric Generator
GG – Gas Generator
HM – Helper Motor
HSS – High Speed Shaft
LCI – Load Commutated Inverter
LNG – Liquefied Natural Gas
LSS – Low Speed Shaft
MVI – Mark VI Gas Turbine Control System
RCA – Root Causes Analysis
TMD – Torsional Mode Damper
TNF – Torsional Natural Frequency
TVPS – Torsional Vibration Protection System
VFD – Variable Frequency Drive
VIB – Vibration

TVPS DESCRIPTION

The main function of the TVPS is to convert a speed signal into a shaft angular vibration. Suitable raw signals used for angular vibration measurements with the TVPS are shown in Figure 1. The signal processing is based on tip timing, i.e., the arrival time of individual teeth or bolts.

SIGNAL	PRO	CONS
SPEED PICK-UP	Always available Bandwidth Suitable for long term monitoring	Position on the modal shape
PHASE REFERENCE	Often available Suitable for long term monitoring	Limited Bandwidth
IMPROVED PHASE REFERENCE	Bandwidth Position on the modal shape Suitable for long term monitoring	Spurios 1X
COUPLING BOLTS	Position on the modal shape Bandwidth	Temporary installation Spurios frequencies

Figure 1: Test Setup

The first step consists of sampling and interpolating the raw signal. The interpolation allows the required sampling rate to be reduced, avoiding the need for high resolution of the raw signal.
The definition of the passing time Δt between two pulses is defined by a selectable voltage threshold.
The instantaneous angular speed of the machine is given by:

$$\omega(t_i) = \frac{\Delta\theta_i}{\Delta t_i} \qquad (1)$$

The angular speed function is then integrated to obtain the angular vibration values AV(t):

$$AV(t_i) = (\omega(t_i) - \omega_m)\Delta t_i \qquad (2)$$

ω_m is the mean angular speed calculated in a convenient way.
The factors that determine the measurement quality are:

- Sampling frequency f_s
- Number of pulses per second p

The sampling frequency depends on the data acquisition hardware, while the pulse rate is given by:

$$p = \omega_m N \qquad (3)$$

A high pulse rate extends the measurement bandwidth adding information on the angular vibration behavior. However, a very high pulse rate requires a high sampling frequency to appropriately define the pulses and more processing power to analyze the sampled signal.
In practice it is necessary to compare torsional vibration values with the maximum allowable torque of the train. Usually, the maximum allowable torque is determined by the load coupling.
To calculate the torsional vibration acting on the coupling starting from angular vibration values measured in any machine section, two methods are possible. The first method, described in [1], is based on modal analysis and needs only one measurement section in a suitable position (i.e., not on the modal nodes). The alternating torque is given mathematically by:

$$T(f_n) = \left(CP(f_n)AV1(f_n)\right)k \qquad (4)$$

where k is the torsional stiffness and CP is the coupling participation factor that relates the coupling twist angle amplitude and the angle in the measurement position.
The second method requires two measurement sections in suitable positions and consists of multiplying the instantaneous

difference between the two angular vibration measurements by the torsional stiffness:

$$T(t) = \left(\left|AV1(f_n) - AV2(f_n)\right|\right)k \qquad (5)$$

In practice, the preferred method is a variant of the former in which the maximum allowable angular vibration amplitude on the measurement section is calculated:

$$AV(f_n)_{MAX} = \frac{1}{CP(f_n)} \frac{T_{MAX}}{k} \qquad (6)$$

T_{MAX} is the Max Allowable Torque. This method allows having angular vibration reference values.

The hardware used for the measurements shown in this paper had the following characteristics:

- Calculation of the angular vibration on a 40 MHz Clock FPGA;
- 50 KHz card for raw signal sampling;
- 25 KHz card for the output signal (i.e., angular vibration waveform).

A TVPS with the above characteristics was used to process four raw signals with 500 pulse/sec. simultaneously.

TVPS VALIDATION

To validate the TVPS, a dedicated angular vibration generator (speed generator) was developed. The configuration parameters of the angular vibration generator are:

- Frequency and amplitude of the angular vibration components
- Number of teeth
- Machine rotational speed

The angular vibration generator produces two output signals:

- Angular vibration time wave
- Speed signal in the form of pulses as produced by a sensor reading on a toothed wheel

Figure 2 shows a signal generated by the speed generator representing pulses collected by a sensor reading on a 60-tooth geared wheel on a machine rotating at 3600 rpm. The results shown in the following refer to these operating conditions. The "embedded" angular vibration amplitude carried by the signal is 0.1 [deg 0-pk] at 8.2 [Hz].

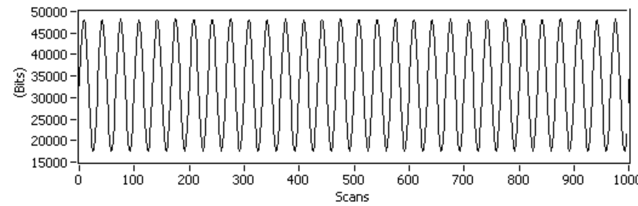

Figure 2: Speed Signal Sample.

The test setup is shown in Figure 3. The generated speed signal was processed through the TVPS and in addition, through a TK17. Outputs were sent to a DAS and were compared to the angular vibration reference signal.

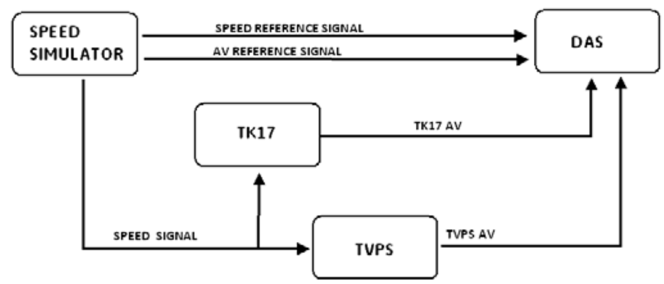

Figure 3: Test Setup for TVPS Validation.

The results for an angular vibration of 0.1 [deg 0-pk] at 8.2 [Hz] are shown in Figure 4 and Figure 5. It can be seen that there is very good agreement in terms of amplitude and frequency between the TVPS and the reference signal. A slight phase difference can be noted with the signal of the TVPS leading the reference signal. Simulation showed that the phase shift depends only on the angular vibration frequency and speed signal pulse rate but does not change with vibration amplitude.

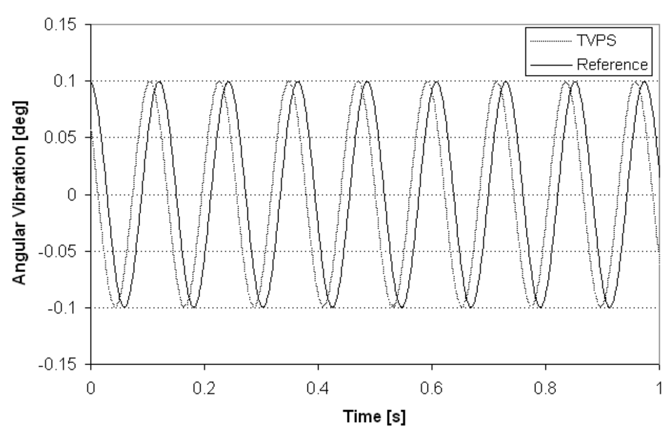

Figure 4: Reference vs TVPS signal at 8.2 Hz. Waveform.

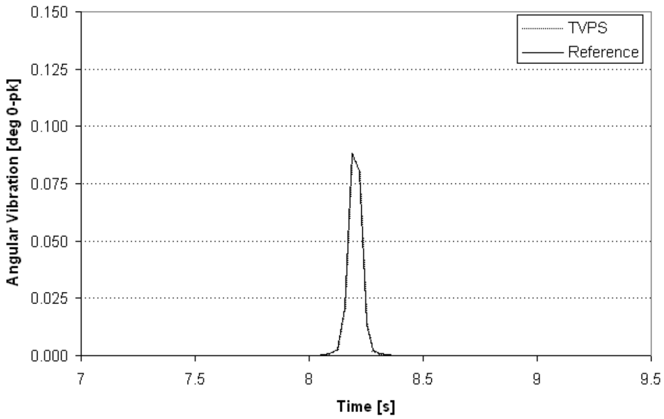

Figure 5: Reference vs TVPS signal at 8.2 Hz. Spectrum.

Figure 6 shows the system accuracy at 8.2 Hz. A speed signal containing a small angular vibration (0.001 [deg 0-pk]) at 8.2 Hz was generated. It can be observed that the measured time wave signals (reference and measured) have an offset. This offset is added by the output cards. The TVPS output signal shows some low frequency noise. Spectral analysis shows that the generated signal is captured very well by the TVPS in both amplitude and frequency.

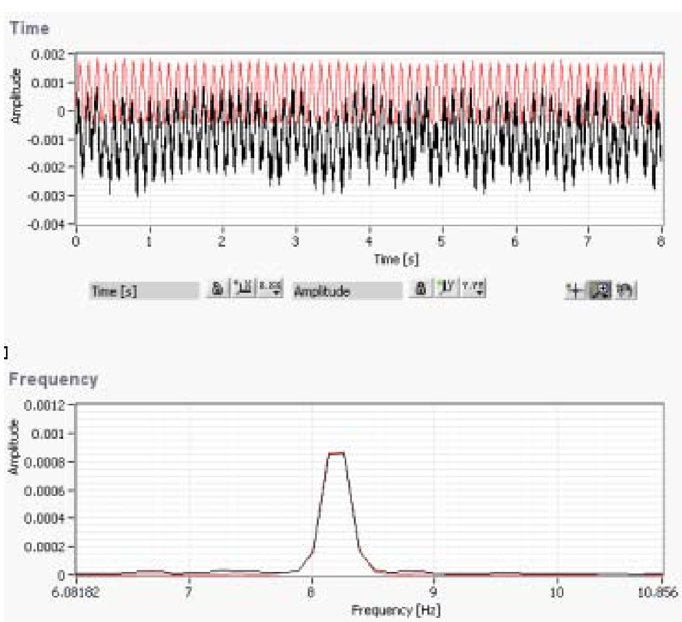

Figure 6: System Sensibility. Reference signal (red) and TVPS processed signal (black).

APPLICATION #1: RCA INVESTIGATIONS

The TVPS is used for root cause analysis investigations when torsional vibration is suspected to play a role in the problem.

In the typical test setup, the TVPS converts the speed signal into angular vibration and its output is sent to a DAS, which acquires all the other parameters important for problem resolution.

Figure 7 shows the data acquisition system used for an RCA carried out on three similar turbogenerator units.

One of the three generator units tripped frequently due to high radial vibrations detected on the non-drive end HSS GB bearing. Preliminary investigations showed that the measurement chains were working correctly and that a noticeable 11 Hz component was responsible for the increase in vibration. Since this frequency was next to the calculated first torsional natural frequency of the train, coupling between torsional and lateral vibration, as described in [2,3], was suspected.

The TVPS input signals were connected to the unit's speed pickups. Clamp current probes were installed after the generator current transformers to detect electrical phenomena and mechanical-electrical interactions. All machine vibration signals and machine control signals were connected to the DAS.

Tests were carried out running two trains at a time in different load configurations.

The measurement campaign confirmed the cause-effect relationship between torsional and radial vibrations and showed that current sags caused an increase in torsional vibration leading, in some cases, to a machine trip.

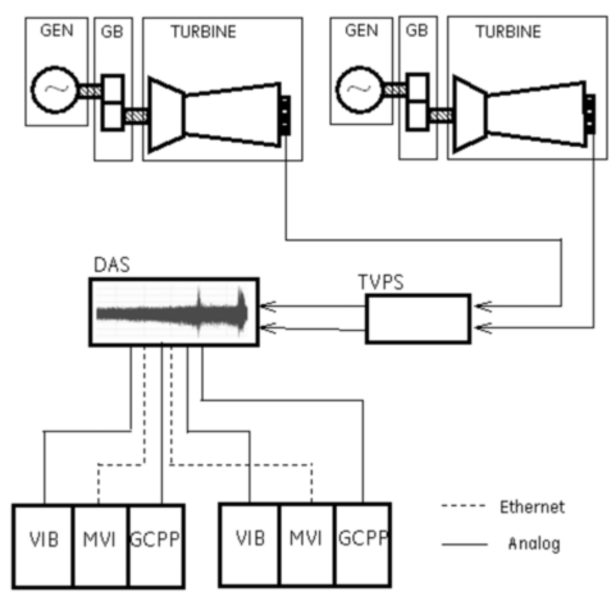

Figure 7: Data Acquisition System Layout.

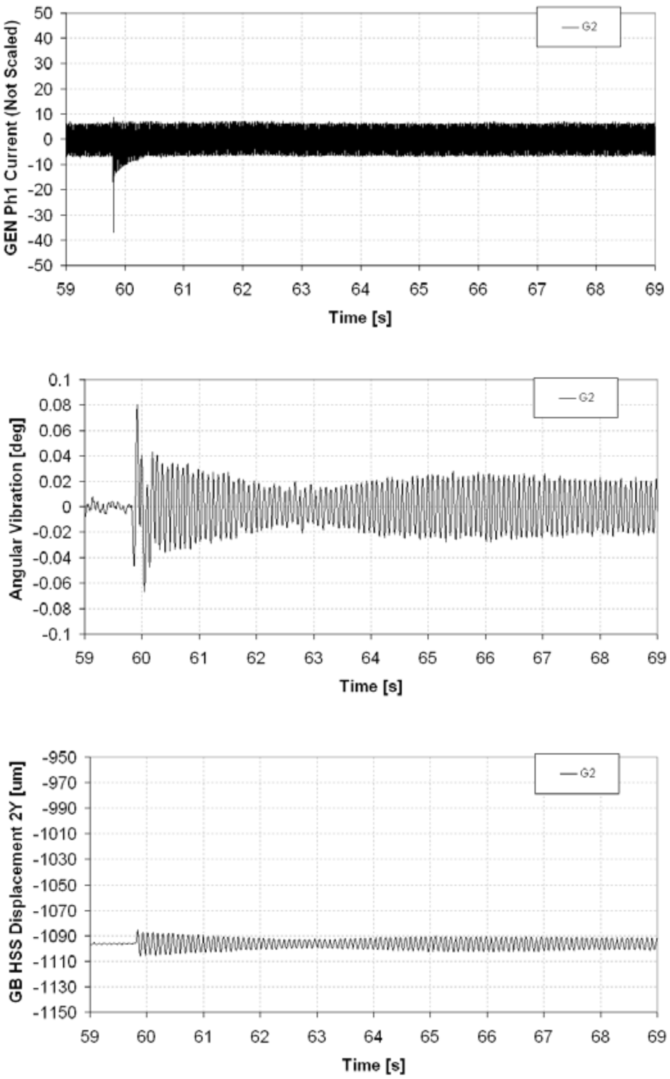

Figure 8: G2 Response to current sag.

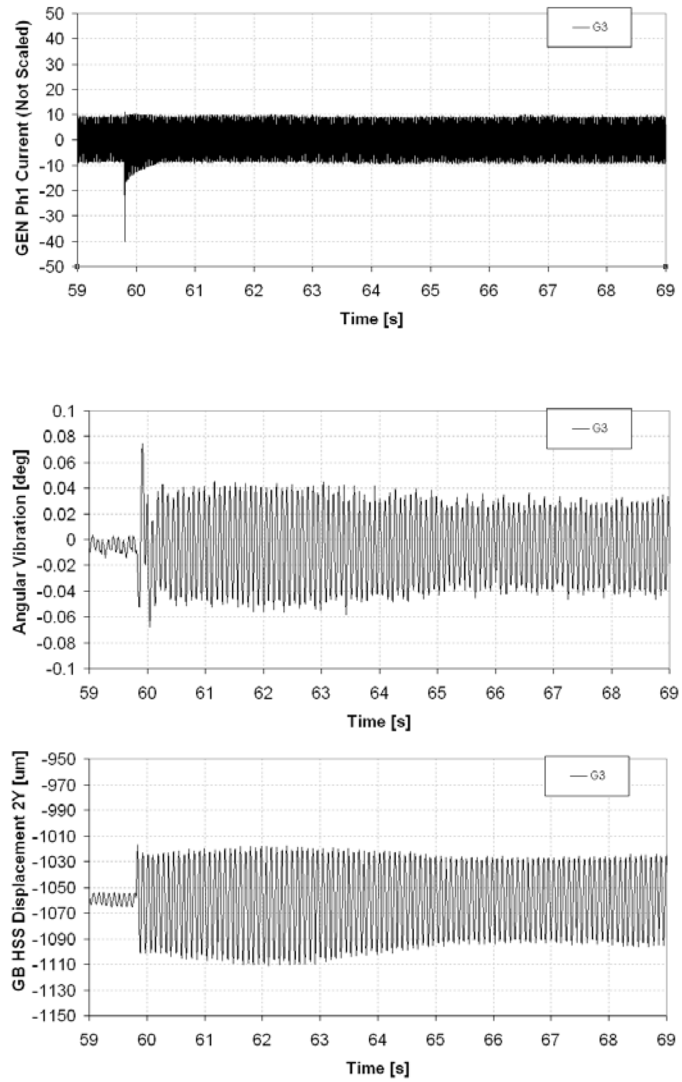

Figure 9: G3 Response to current sag

Figure 8 and Figure 9 show the response of the trains in terms of angular vibration and lateral vibration on the gearbox due to the current sag. The two figures refer to the same instant in time. It can be seen that while the angular vibrations are comparable, the gearbox lateral vibration on train G3 was much higher than that on G2 (90 [um pp] and 20 [um pp] respectively). The angular vibration values were within the limits (0.4 [deg 0-pk]).

Because the gearbox lateral response on G3 was much different from that of the other units, it was decided to inspect the bearing. The inspection revealed pieces of rubber inside the bearing that caused the incorrect development of the oil fluid film and an increased sensitivity of the bearing to excitations. Cleaning of the bearing solved the problem.

APPLICATION #2: MACHINE MONITORING AND PROTECTION

Another application for which the TVPS is employed is torsional vibration monitoring and protection. In this case the TVPS is installed for permanent service and operates without supervision.

In this case, the preferred input signal of the TVPS is the speed pickup signal due to its reliability. If the speed pickup is not suitable because of its position on the modal shape (e.g., a node) a measurement plane in a different position is needed. Typically, the second choice is the phase reference signal because the installation of additional sensors is not necessary. In some cases, the phase reference signal can be improved with limited effort by adding additional marks to obtain a geared wheel. In order to avoid loosing the phase reference, the additional marks are machined to a smaller depth. Figure 10 shows a compressor lock ring with 24 additional marks. The benefit of the additional marks is the extension of the angular vibration measurement bandwidth and a decrease in the lowest unit speed at which the measurement is reliable.

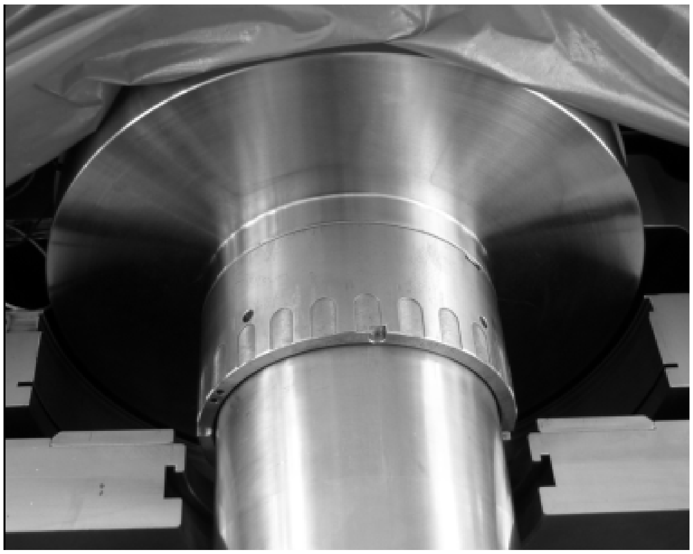

Figure 10: Lock ring with additional marks for improved phase reference signal.

A monitoring application was implemented in an island power system composed of four generators mainly supplying VFDs driving pumps. The turbogenerator units tripped unexpectedly due to high vibrations of the gearbox HSS.

Vibration measurements showed that the increased vibration level was due to an 11 Hz sub-synchronous component that was related to the plant load. It was clear that the problem was due to Sub-synchronous Torsional Interactions.

Adjustment of the VFD control was suggested and proved to be beneficial. Due to the fact that it was not feasible to test all site operating conditions and that the site was in an expansion phase, each turbogenerator was provided with a TVPS for torsional vibration monitoring.

The TVPSs were interfaced with the machine control systems in order to:

- Display on the HMI amplitudes at pre-selected frequency ranges
- Implement alarms on the filtered frequencies

The pre-selected frequencies were set around 5 Hz and 11 Hz (first natural frequency of the train).

An alarm was implemented on the machine control system at 0.1 [deg 0-pk]. When the alarm condition is reached, the operators have to decrease the plant loads following a "last in first out" sequence until the angular vibration values are back to normal. Because the TVPSs are monitoring torsional vibrations, it was possible to increase the alarm and trip limits of lateral vibration of the gearboxes.

In addition to angular vibration measurements, the TVPS has the capability of collecting the following data:

- All protection and control parameter signals (through Ethernet)
- Lateral Vibration (through additional analog input modules)

Data are stored at a 1 second sampling rate on local memory. Reaching the angular vibration alarm level triggers the collection of lateral and angular vibration waterfalls.

The TVPS is hence the "black box" of the machine and can be used to analyze the root causes of future problems or failures, even those not strictly related to torsional vibrations.

Figure 11 shows angular vibration trends stored by the TVPS over a period of two weeks after the adjustment of the VFD control. It can be seen that the angular vibration values are within the limits (0.1 [deg 0-pk]). The second filtered frequency corresponds to the first torsional natural frequency of the train (11 Hz). The overall value can be useful for detecting increases in any other frequencies not specifically monitored.

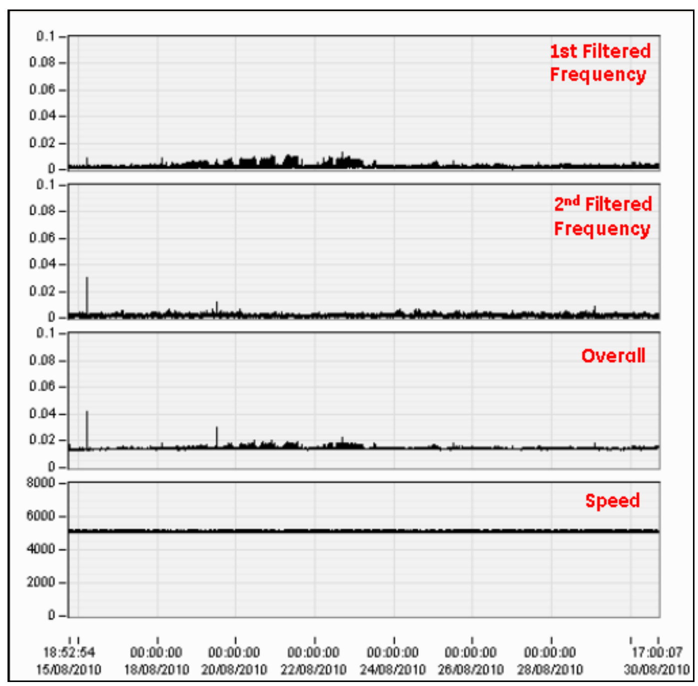

Figure 11: TVPS trends over a period of two weeks.

APPLICATION #3: TORSIONAL VIBRATION SUPPRESSION

A further application of the TVPS is coupling to the torsional mode damper (described in detail in [4]) designed to damp torsional vibrations in trains driven by electric motors. This application was validated on a turbocompressor train driven by an 85 MW gas turbine and a 17 MW LCI Helper Motor, and is described in detail in [5].

The torsional mode damper analyzes the angular vibration signal and gives a modulation signal that is added to the converter control logic to damp the pulsating components of the torque.

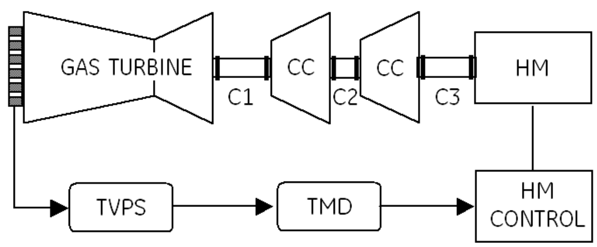

Figure 12: TVPS used for torsional vibration suppression.

The TMD requirements for the TVPS processed angular vibration signal were:

- Maximum delay between effective shaft torque and processed signal: 25 ms
- Minimum required dynamic accuracy: 5% of full scale

Tests were carried out with the speed simulator to evaluate the time delay between the reference signal (shaft torque) and the TVPS angular vibration signal. Figure 13 and Figure 14 show the results for the two angular vibration frequencies equal to the first and second torsional natural frequencies of the train under test. The highlighted area represents the unit operating speed range, which is also the range where the TMD was allowed to be operated. The time delay variation within the operating range is lower than 0.5 ms.

During the site tests, the angular vibration was measured in four positions along the shaft line:

- Two speed pickups (Section A)
- Improved phase reference signal (Section C)
- Non-Contact Probe on flange bolts (Sections D and E)

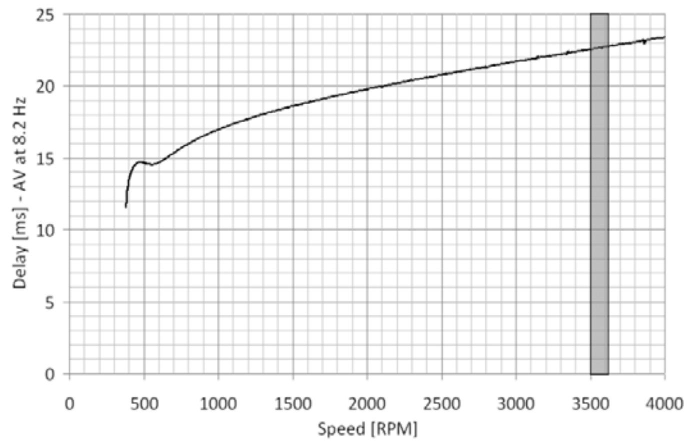

Figure 13: Time delay between reference and measured angular vibration at 8,2 Hz.

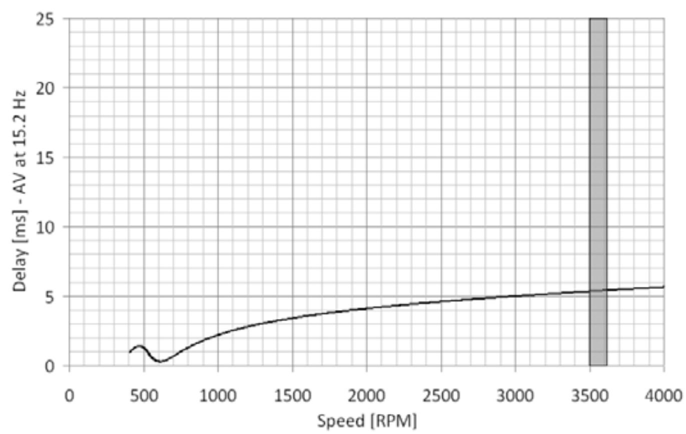

Figure 14: Time delay between reference and measured angular vibration at 15,2 Hz.

Only the output of one speed pickup was fed into the TMD (Section A in Figure 15). The reasons for using the speed pickup signal are as follows:

- Speed pickups are always available on turbocompressor trains;
- The sensors are reliable and used for machine control;
- At least three sensors are installed on each geared wheel, allowing redundancy logic to be implemented.

It is worthwhile to note that the measurement section was on the opposite side of the section from where the feedback action occurred (the helper motor).

The other measurement sections were held as backups and used for reference during the tests.

An alarm signal was routed to the machine control system in case the angular vibration exceeded 0.4 [deg 0-pk].

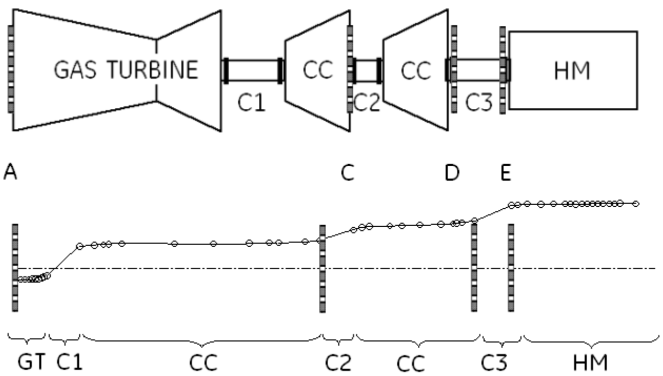

Figure 15: Turbocompressor train with associated 1st TNF mode shape and measurement sections.

Figure 16 shows results obtained during the site test. It can be seen that when the TMD is enabled, the angular vibration decreases to very small values.

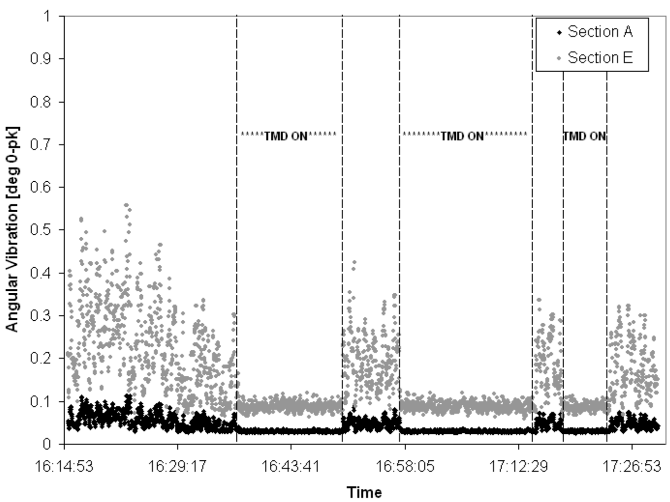

Figure 16: Angular Vibration trends measured by TVPS on sections A and E (Overall Value).

CONCLUSIONS

An increasing number of site vibration measurements are being requested in order to prevent and/or solve issues related to torsional vibrations. In some cases, permanent torsional vibration monitoring and protection of the machine has been needed to solve the problem.

To overcome the practical difficulties of measuring torsional vibration on a live plant, especially on a continuous basis, a device called a TVPS, based on tip timing, has been developed.

Through the TVPS and the modal analysis of the train it is possible to easily measure torsional vibrations. In fact, in most cases it is possible to measure angular vibration on one plane using signals that are already used by the machine control system, such as speed pickup or the phase reference signals. The reliability of the sensor allows torsional vibration monitoring and protection of the machine.

The reliability of the measuring system and the output signal characteristics match the requirement for the torsional mode damper (TMD).

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REFERENCES

[1] Golebiowski, M., Naldi, L., Rossi, V., Ponticelli, S., 2010, "Train Torsional Vibrations: Monitoring System Based on Model Result", ASME Turbo Expo 2010, Glasgow, UK, June 14-18.

[2] Iida, H., Tamura, A., Yamada, Y., 1978 "Coupled Torsional-Lateral Vibration of Shaft in a Geared Rotor System I", Bulletin of JSME, Vol23, No186, December 1980.

[3] Iida, H., Tamura, A., Oonishi, M., 1984 "Coupled Torsional-Lateral Vibration of Shaft in a Geared Rotor System II", Bulletin of JSME, Vol28, No245, December 1985.

[4] Sihler, C., Schramm, S., Song-Manguelle, J., Rotondo, P., Del Puglia, S., Larsen, E., 2009, "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.

[5] Sihler, C., Schramm, S., Rossi, V., Lenzi, A., Depau, V., 2011 "Electronic Torsional Vibration Elimination for Synchronous Motor Driven Turbomachinery", ASME Turbo Expo 2011, Vancouver, Canada, June 6-10.