Practical methodologies for on-site measurements of torsional natural frequencies – application to industrial cases

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Abstract

The predictive calculation of torsional natural frequencies of a shaft line is performed in the purpose of avoiding any torsional resonance problems. However it is often delicate to select the appropriate excitation frequencies that have to be considered in the calculation, as well as to determine the modal damping factors. Moreover the bearing flexibility is rarely integrated in the boundary conditions which adds some uncertainty in the results.

Therefore it is often useful in practice to carry out experimental measurements of the torsional natural frequencies, in order to validate the calculation and to achieve the machine commissioning properly. Several methodologies can be implemented, with more or less suitable practical aspects when applied to an industrial machine: the constraints of time, cost and environment yield us to use different tricks in order to reveal the torsional modes.

We first present a torsional mode calculation using finite elements and the comparison with real measurements. We then illustrate through a few industrial case studies how we proceeded to determine the torsional frequencies: bump test machine at rest, transient analysis, response to an excitation generated by the process or by faults, and motor currant modulation analysis in stationary operating conditions.

1 Introduction

If the lateral analysis of shaft lines is now settled down it is probably not the same regarding torsional analysis, due to several possible causes: Firstly there is less standards about it even with API recommendations (American Petroleum Institute) which are yet quite complete regarding lateral vibration analysis. Second point is that the consequences of a torsional resonance are probably less obvious than for a flexural one : on line monitoring are design to detect radial and axial vibration but more rarely the torsional vibration. Lastly if the methods and means to analyse the flexural behaviour of the shaft line during the commissioning have settled down it is not the case for torsional analysis: the instrumentation for torsional vibration and stress measurement is mostly complicated in industrial plant; constraints of time, cost, and environment. It leads to use different tricks in order to reveal the torsional modes.

We first present a torsional mode calculation using finite elements and the comparison with real measurements. We then illustrate through a few industrial case studies how we proceeded to determine the torsional frequencies: bump test machine at rest, transient analysis, response to an excitation generated by the process or by faults, and motor currant modulation analysis in stationary operating conditions.

2 Example of torsional calculation of a shaft line

This study concerns a blower including a motor, coupling and a centrifugal compressor. The shaft line is supported by roller bearings. Nominal speed is from 31.5 Hz to 61.5 Hz

2.1 Analytic model

First expected torsional mode involves rotor inertia Ia and Ib supposed to be infinitely rigid and the coupling stiffness K: formulation used by coupling manufacturer is :

Fp torsion = $1/2\pi$ [K (Ia + Ib)/(Ia Ib)]1/2, with a design criteria Fptorsion > 2 Frot

For present shaft line:

- Motor shaft inertia Ia= 1.75 kgm^2
- Compressor shaft Inertia Ib= 1.66 kgm^2
- Coupling stiffness K= 0.3 MNm/rad

Calculation gives a natural frequency Fp1=94Hz

2.2 Finite element calculation

A finite element model has been designed :

- Beam elements for shaft lines.
- Torsional stiffness for the coupling from manufacturer datasheet 0.3 MNm/rad
- Lumped mass with inertia for coupling parts, impellers and motor's rotor,
- Radial stiffness of bearings : 10^8 N/m

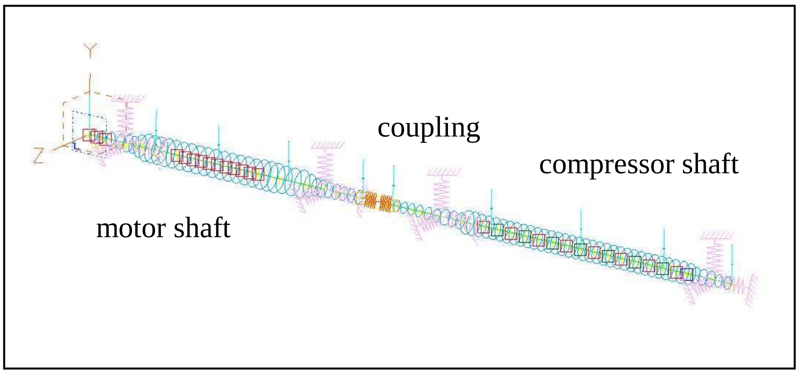
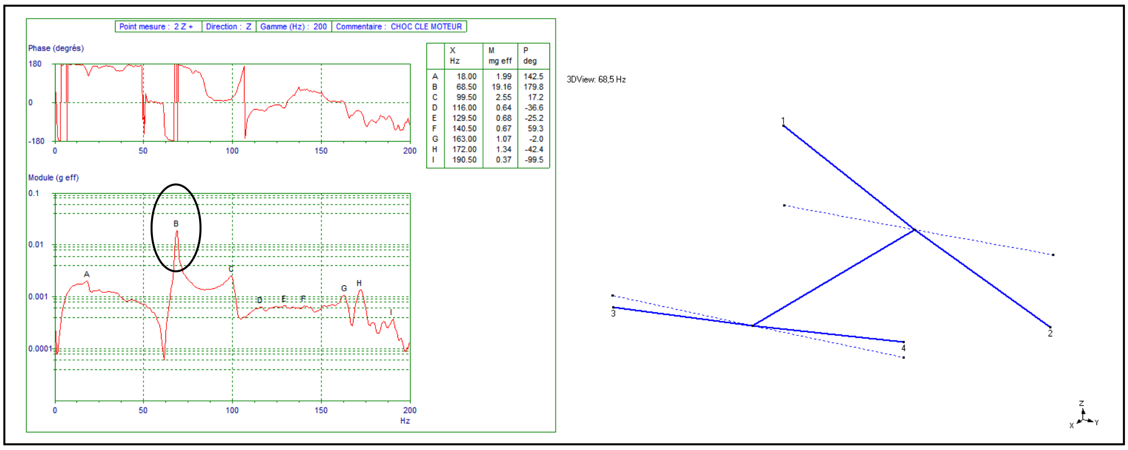


Fig 1: Model of the blower

The FEM calculation gives a first torsional mode at Fp2= 59Hz with a 180° lag of phase between the two rotors.

2.3 Comparison with experimental measurement using a shock hammer

A tangential shock has been done on the motor's shaft and the Frequency Response Function (FRF) reveals a first torsional mode at Fp3=68Hz.



	X Hz	M mg eff	P deg
A	18.00	1.99	142.5
B	68.50	19.16	179.8
C	99.50	2.55	17.2
D	116.00	0.64	-36.6
E	129.50	0.68	-25.2
F	140.50	0.67	59.3
G	163.00	1.07	-2.0
H	172.00	1.34	-42.4
I	190.50	0.37	-99.5

Fig 2: Frequency response function with a tangential shock on the shaft

There is consequently a very important gap between Fp1, Fp2 et Fp3 that we can explain as following:

- The coupling stiffness given by the manufacturer is a static stiffness surely much lower than dynamic stiffness. (We have to take a value of the stiffness three times higher to get Fp2=Fp3)
- Shafts are not infinitely rigid and there must be a part of torsional deflection coupled with the coupling mode.
- Moreover we expect to have a difference between the natural frequency at rest and in operation due to the torque and speed.

This example underlines the precautions requested regarding the input data and the interpretation of the results. It highlights the need to measure the torsional natural frequency as soon as a torsional default is suspected.

Following examples illustrate some methodologies that we use to get around the difficulties on field.

3 EXAMPLES OF NATURAL FREQUENCIES MEASUREMENTS IN INDUSTRIAL SHAFT LINES

3.1 FRF measurement with a shock hammer

Torsional excitation using a shock hammer is most of the time delicate and doesn't always reveal the torsional modes. Its main disadvantage is that it is done without torque and consequently the conditions are different from those in operation. For that reason we will apply a static torque on the shaft line when it is possible. It is particularly important in case of gearboxes to reach the contact between the teeth.

The shock is usually done in a tangential direction on a coupling sleeve and the vibratory response recorded also in tangential direction on two opposite points of the diameter. Like this and regarding phase lag we can distinguish flexural modes from torsional ones

Previous measurement on the blower clearly reveals the torsional mode. In another hand in the following case regarding a shaft line with a gearbox which for two torsional modes is known at 415 Hz and 1000Hz only the latter one is detectable. Actually the calculation indicates at 415 Hz a node on the coupling that is the only accessible place for the sensor and explains why this mode couldn't be measured.

The mode at 1000 Hz presents a maximum at the measurement location and is clearly detectable on the FRF.

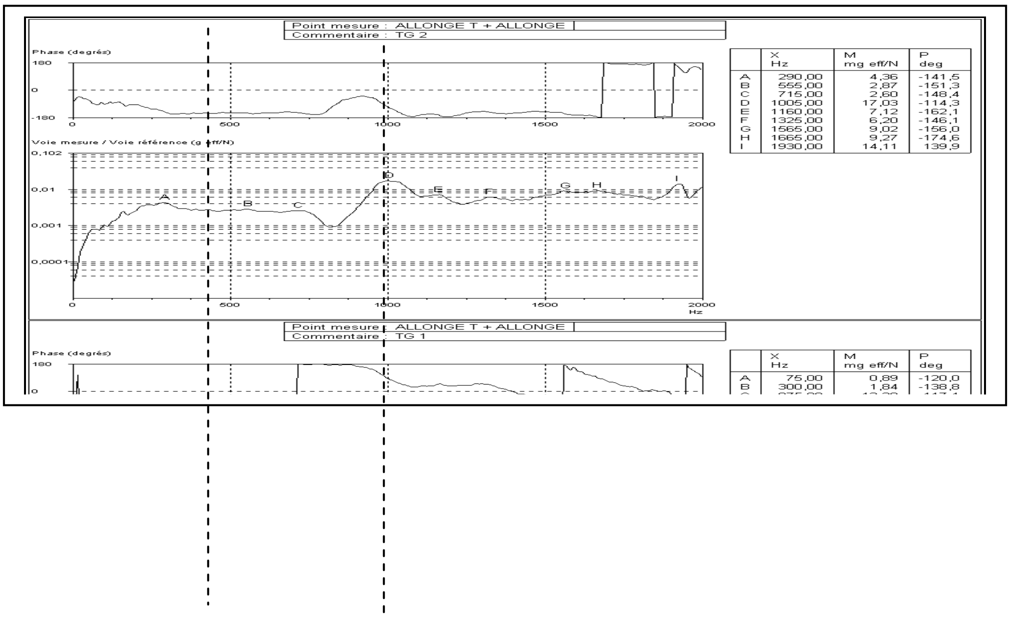


Fig 3: Frequency response function in two points of the diameter with a tangential shock on the coupling

Vibration analysis at the gear mesh frequency during machine start up indicates that the frequency of the modes are shifted at respectively 325 Hz and 805 Hz. That illustrates previous considerations.

3.2 Measurement using a torque meter .

It is for sure that the use of a torque meter is very efficient but it is also the most difficult to instrument on site.

A test bench for helicopter gearboxes is equipped with a torque meter. Since it indicates abnormal torque fluctuations that disturb the tests we are missioned to diagnose its origin.

Torque measurements reveal fluctuations art 10Hz in spite of the lack of kinematical excitation at this frequency. The amplification of torque fluctuation when the speed reaches 10 Hz confirms the presence of a torsional mode.

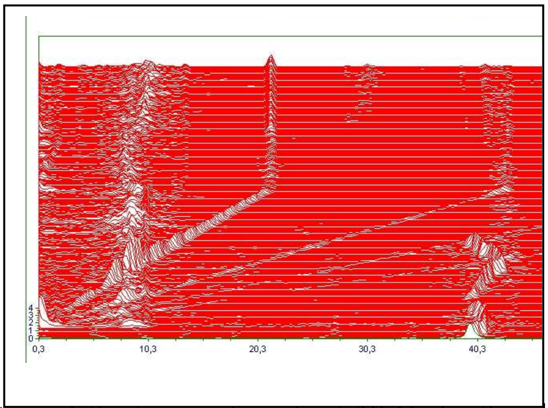


Fig 4 : Waterfall for the detection of the torsional mode at 10 Hz by the observation of basic line spectrum and transient analysis of the torque.

When the torque increases from 26 to 120 daN the natural frequency increases from 10 Hz to 13 Hz .

3.3 Electrical current analysis for torsional mode detection

In previous example we analysed the motor and generator DC current to identify the mode at 10 Hz [1]. Even if we observe a light modulation the symptoms were not obvious enough to diagnose the presence of a mode.

In another hand, without considering the symptom as a proof, the presence of wideband peaks on the baseline spectrum of the amplitude modulation function (AMF) enable most of time to reveal the torsional modes.

The example below is concerned with a ball mill made of an asynchronous motor, a gearbox speed reducer, a quill shaft and a girth gear. Spectrum analysis of the motor current shows two trains of sidebands lines: one at the quill shaft frequency (2.26Hz) and the other at the mill frequency (0.277 Hz). The lines are magnified around 40Hz and 60 Hz, i.e. at two sidebands spaced of 10 Hz apart around the carrier frequency at 50 Hz. Therefore a torsional mode at 10 Hz is suspected.

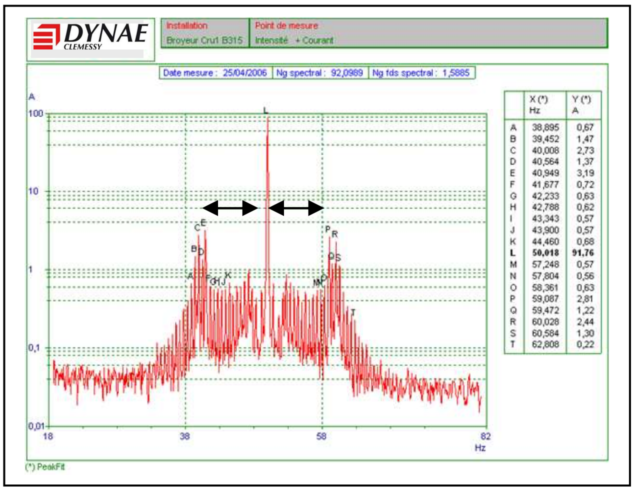


Fig 5 : Side bands amplification at +/-10Hz of the stator current that reveal a torsional mode at 10 Hz.

The AMF spectrum enables to get more details on the modulations lines and on the amplification effect accountable for a torsional mode at 10Hz. [3]

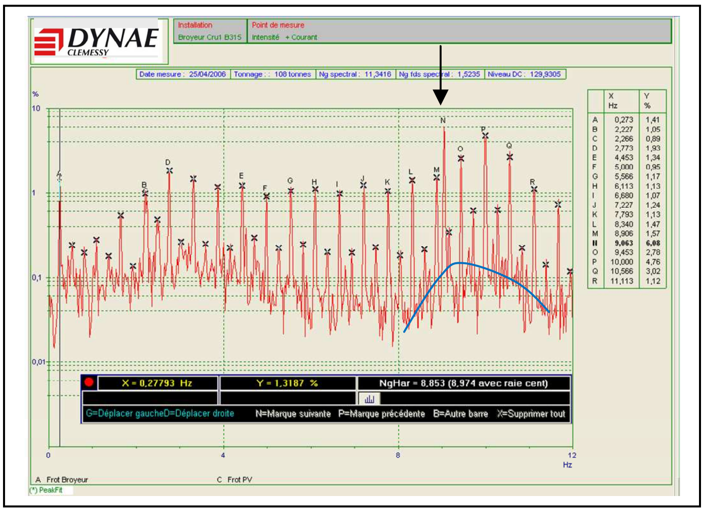


Fig 6 : FMA spectrum of motor's current at 50 Hz . Cross marks are the mill speed harmonics and the arrow indicates the fourth harmonic of the quill shaft.

However the accuracy of the position of the natural frequency remains low since we do not have a white noise excitation but a discrete one. We can only say that it is close to 10 Hz.

A methodology based on the analysis of pulse response to the shocks occurring at each junction of the girth gear which is in two parts has been developed. It enables a drastic improvement of the natural frequency resolution.

The observation of the time waveform of the AMF reveals a sudden variation of load induced by the junction of both half sectors of the ring gear occurring two times per mill revolution that yield to a torsional excitation. (Fig 7).

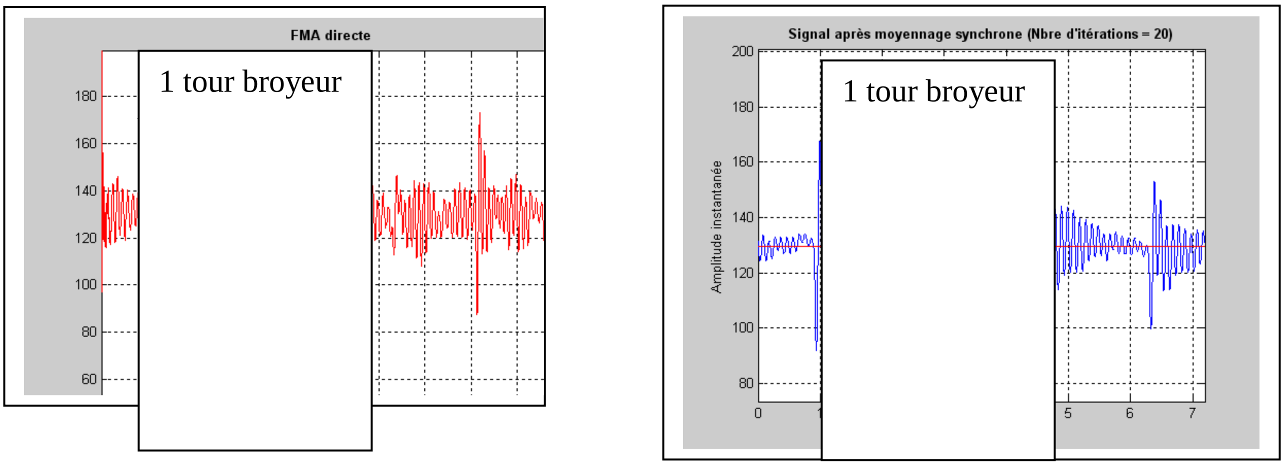


Fig 7 : raw AMF at left and AMF after angular resampling at the mill speed on right.

Angular resampling removes the components coming from the quill shaft and highlight the damped sine waves after each pulse that correspond to the torsional mode response. The measurement of the period of this sine wave gives a natural mode at 10.14 Hz with much more resolution than with the AMF spectrum.

3.4 Measurement using a Laser Rotational Vibrometer (VRL)

VRL is a double laser beam instrument that provides the instantaneous rotating speed of a shaft line.

We use it here for a raw mill similar to previous one but located on another cement plant. It presents 22 mm/s RMS on the gearbox at a ghost frequency Fx=283 Hz. Since a torsional resonance has been incriminated we used the VRL to characterize the torsional mode.

The advantages of the VRL are firstly to enable measurements without stopping the mill to set the system and secondly to enable to measure the rotational speed successively in all available sections of the shaft line we need (as long as the shaft is visible). This way it is possible to determine the mode shape. A tachometer can be used as phase reference for the different measurements.

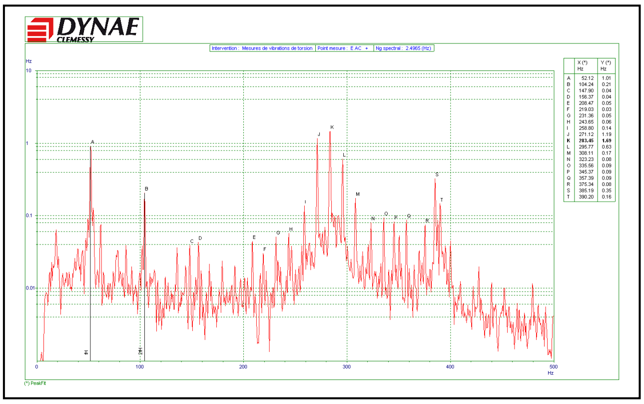


Fig 8 : Instantaneous speed spectrum 0- 500Hz located in section E.

We can see two predominant rotational vibration components on the spectrum (fig 8) : one at the girth gear frequency (line A at 52Hz) identified as a forced response, and the second one at the ghost frequency analysed as being a torsional resonance (line K at 283 Hz).

A modulation rate at 283Hz at different shaft sections gives a sketch of the mode with a maximum on the quill shaft.

A clearance in the two teeth coupling can be detected and is probably involved in the incriminated natural mode.

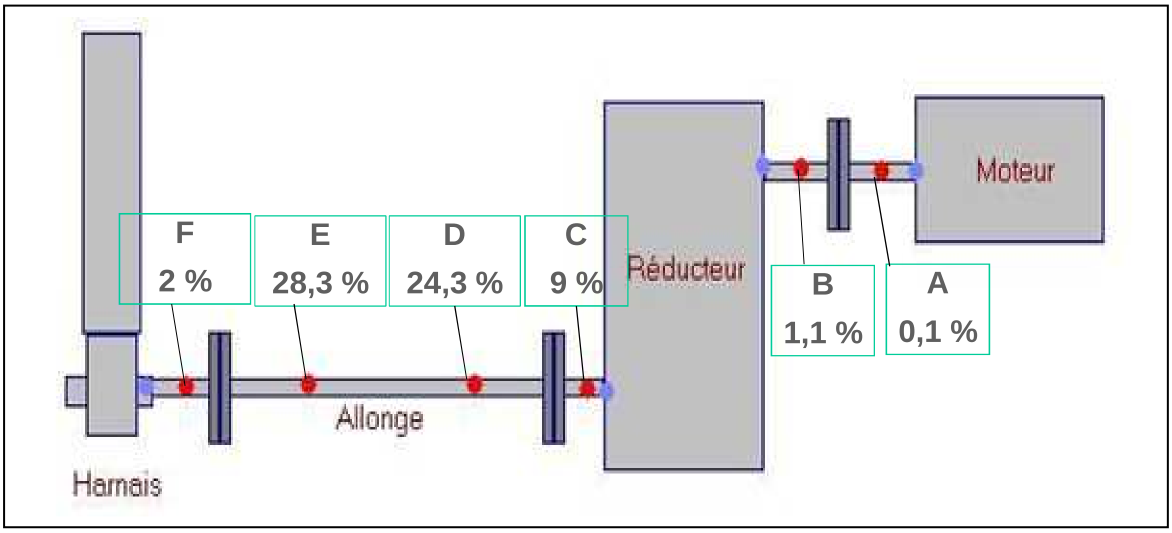


Fig 9: Speed fluctuation at 283Hz giving a sketch of the torsional mode.

3.5 Transient measurements

Previous measurement methods can also be used during run up or shut down [3] . Run up seems preferable since the torque is closer to the nominal torque than during the shut down.

Two indicators can be analyzed :

- the profile of kinematic frequencies and mostly those that involve a torsional excitation such as gear mesh frequencies.
- The response to torque pulses that often occur during start up : pulse at the first revolution of the shaft line, shock due to the clearance in coupling or gear teeth , pulse when the motor is short circuited and so on.

Here again torque and instantaneous speed (encoder or VRL) measurements give accurate results. In another hand the modulations of the motor or generator current , easier to implement, usually enables to measure the torsional frequencies. . Fig 10 displays torque and current signals during the start up of another cement mill: the left curve is at the very beginning of the start up and the current doesn't reveal the natural mode against the torque . however the right curve at the end of the start up illustrates a good coherence between torque and current fluctuations when the torsional resonance is reached just before the nominal speed.

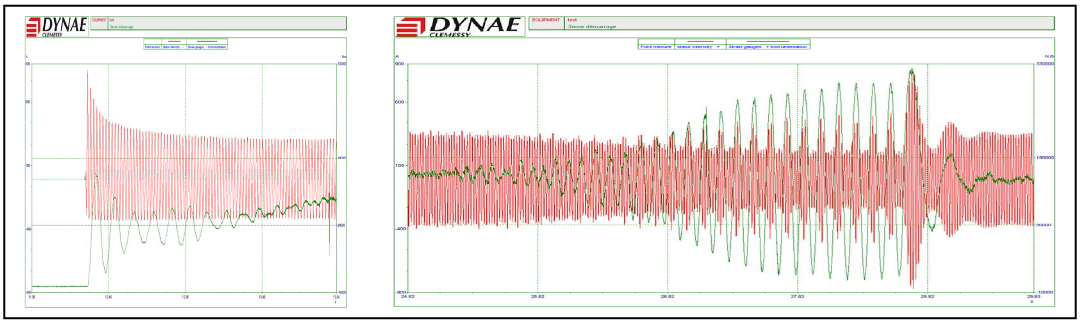


Fig 10: torque (green) and current (red) signal during the start up :the left curve is at the very beginning of the start up and the right curve at the end .

4 CONCLUSION

These experiences on industrial equipments illustrate several issues to characterize the natural torsional modes of a shaft line.

Firstly the multiplicity of the physical parameters that can give information: torque, speed, modulation of the motor current , FRF… Torque measurement is the most accurate and efficient way to measure the torsional natural frequencies and at the same time the most difficult to instrument. We can note however that the wireless FM system are limited to moderate speed because of centrifugal forces. Instantaneous speed is also really efficient and the VRL easier to implement than optical or magnetic encoders. At last current modulation doesn't always give the results but when it is useful, I mean when there is really a torsional trouble on a shaft line, the current modulation works. The advantage is the non-intrusive and low cost measurement aspect without the need to stop the shaft line.

FRF measurements with a shock hammer are delicate and we must remind the frequency gap of natural frequency between stopped and running conditions.

Our recommendation is to use several methods at the same time, depending on the problematic and what it is possible to do on site, to confront the results. The comparison between results of calculation can yield to important gaps coming from the hypothesis on boundary conditions or the difference between static and dynamic stiffness of the coupling. That is why acceptance test at new equipment commissioning seems pertinent.

References

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