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ELECTRIC DRIVEN CENTRIFUGAL COMPRESSOR – SUPER SYNCHRONOUS VIBRATIONS OF THE HIGH SPEED SHAFT LINE: OBSERVATIONS, ASSESSMENT AND RESOLUTION

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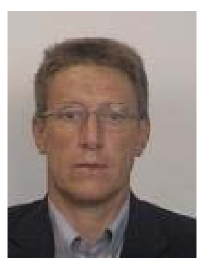

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ABSTRACT

An unacceptable vibratory behavior was encountered in 2012 during the full load string test of an electric driven centrifugal compressor. The vibrations were observed on the high speed part of the shaft line and both compressor and gearbox exhibited vibratory levels higher than 50 microns peak to peak on the second harmonic (H2) of the rotational speed. The centrifugal compressor is driven via a variable speed electrical motor through a gearbox. The main characteristics of the shaft line are the following:

- 9 MW electrical motor running from 1250 to 1875 rpm to cover the whole operating speed range of the compressor,
- Gearbox with a speed ratio of 7.035,
- Centrifugal compressor (back to back arrangement) from 72 to 167 bara (first stage) and from 165 to 307 bara (second stage),
- Low speed and high speed flexible couplings of membrane type technology.

All individual components of the train were successfully tested prior to be integrated to constitute the complete shaft line package. The aerodynamics performances of the compressor were as expected, and the gear box and the compressor did not exhibit any abnormal vibratory behavior.

During the ASME PTC10 Type 1 string test of the complete package, super-synchronous vibrations were detected on the compressor and the pinion of the gearbox while the compressor was running at around 94% speed. Extensive theoretical and experimental investigations were conducted to identify the root cause of the 2X vibration component with the full involvement of the different components suppliers:

- Potential excitation coming from the electrical drive system,
- Potential excitation coming from the gearbox,
- Aerodynamic excitation coming from the compressor,
- Torsional resonance and/or interaction with lateral behavior,
- Contact, alignment, lubrication influence, type of bearings,
- Lateral critical speed combination (train shaft line analysis),
- Type of coupling, non linearity and lack of flexibility.

The issue was solved within three months thanks to the following modifications:

- Gearbox bearing span increase and reduction of the overhang, by modifying the existing pinion design,
- New coupling with reduced moment.

The paper will describe the investigations that were conducted to understand the root cause of the super-synchronous vibrations as well as the modifications that have been successfully implemented.

INTRODUCTION: CLOV STORY

TOTAL operates Block 17 offshore Angola, holding a 40% interest, on behalf of concessionaire Sociedade Nacional de Combustiveis de Angola (SONANGOL), together with partners Statoil (23.33%), ExxonMobil (20%) and BP (16.67%). The agreement covering activities on Block 17 was signed on 15 December 1992 between SONANGOL as concessionaire for all oil and gas activities in Angola, and the original Contractor Group.

Bloc 17 is located 140 km offshore Luanda and comprises several production hubs already developed and other development areas. On stream, Girassol, Dalia and Pazflor entered in production in respectively 2001, 2006 and 2011.

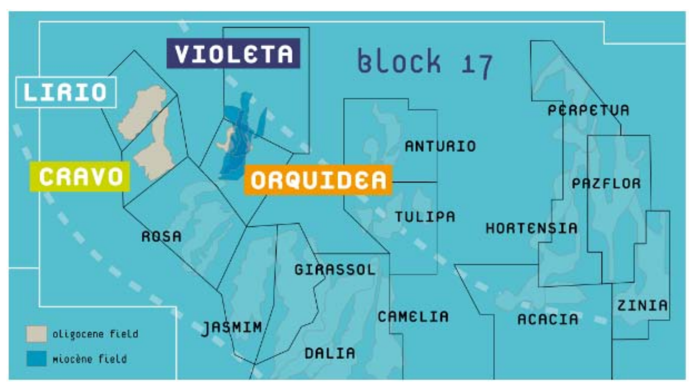


Figure 1: Block 17 development areas

In 2010, TOTAL decided to launch the development program on the CLOV fields (Cravo, Lirio, Orquidea and Violeta), thus becoming the fourth production hub on this exceptionally prolific acreage.

Cravo, Lirio, Orquide and Violeta were discovered at different times and brought together as one project. The CLOV development pole – the 4th on Block 17 – is an area of 381 km² made up of four development areas in the north western part of

Block 17 and about 35 km from the Girassol FPSO (Floating Production Storage Off-Loading facility). The overall pole is expected to produce about 505 Mb of oil over 20 years, three-quarters of it high-quality Oligocene oil.

The FPSO will be equipped to process different grades of crude oil from the Oligocene and Miocene reservoirs. The facilities are designed to improve energy efficiency and to minimize environmental impacts by recovering waste heat from turbines and collecting vent gas from storage tanks.

The FPSO is 305 meters long, 61 meters wide and 32 meters high. It is a new built hull, double-sided single bottom, spread moored with 4 legs in 1275 meters water depth. Oil process is based on wash tanks and settling down tanks for oil and water separation and oil desalting. The liquid treatment capacity is 230 kbpd, and gas compression capacity is 6.5 MSm3/d. The FPSO is an all-electric concept with variable speed drive motors, powered by (3+1) aero derivative turbogenerators rated at 28 MW each. Maximum Personnel On Board (POB) is 240.

Oil storage capacity is 1.78 Mbl in 3 rows of crude oil tanks.

CLOV is TOTAL's first deep-offshore project to use a fiber-optics data communication network as part of its subsea production control system to monitor and control production rates, water injection and distribution of chemical additives.

Figure 2: Subsea view of CLOV Project

The gas export line is the result of the decision not to re-inject the associated gas back into the reservoirs, but to transport it via the existing Angolan gas export network to the AnLNG plant in Soyo, ANGOLA.

As of February 2013, FPSO is under completion in Daewoo Shipbuilding and Marine Engineering (DSME) in Okpo shipyard, Korea and is due to sail away to Angola during summer 2013.

First oil from CLOV development is expected during the second quarter of 2014 with a plateau capacity of 160 kbpd.

FOCUS ON GAS COMPRESSION AND TESTS REQUIREMENTS

Gas is separated and compressed onboard the FPSO to supply fuel gas for the turbogenerators, to provide gas lift at riser bottom, the balance being fully exported to supply the AnLNG plant onshore.

The compression architecture is classical within TOTAL, being:

- One 100% LP1/2 compression package from 1.35 to 19.9 bara, 4890 kW electric HV (High Voltage) induction motor driven with Variable Frequency Drive (VSD), single compressor casing with 2 sections in back to back arrangement, and a speed increasing gearbox.
- Two 50% HP1 compression packages from 18.9 to 75 bara, 9610 kW each being driven by electric HV induction motor with VSD, single compressor casing, and a speed increasing gearbox.
- Two 50% HP2/3 compression packages from 72.4 to 3067 bara, 9610 kW each being driven by electric HV induction motor with VSD, single compressor casing with 2 sections in back to back arrangement, and a speed increasing gearbox.

With the exception of the compressors, both HP1 and HP2/3 packages (2 x 2) are identical including same base plates, same motors, same gear boxes, same lube oil systems and dry gas seal panels for common spare parts and standardization purpose.

CLOV compressors design is of barrel type with dry gas seals. HP1 and HP2/3 compressors speed range extends from 70% of the variable speed drive capacity up to 105%, *i.e.* between 8800 and 13193 rpm.

All packages were designed to optimize the layout and facilitate maintenance operations with intermediate access platforms. To account for pitch and roll conditions, the shaft centerline height from the top of the baseplate has been elevated to around 1800 mm (from top of the baseplate) to allow proper drainage of the lube oil from the bearings back to the reservoir.

The increase shaft height elevation was also motivated for maintenance reasons for compressor internals replacement, as internals are preassembled as a rotor and stator part. Bundles can therefore be pulled out over the lube oil auxiliary area of the package, thus diminishing scope of piping removal and thus lead time for replacement.

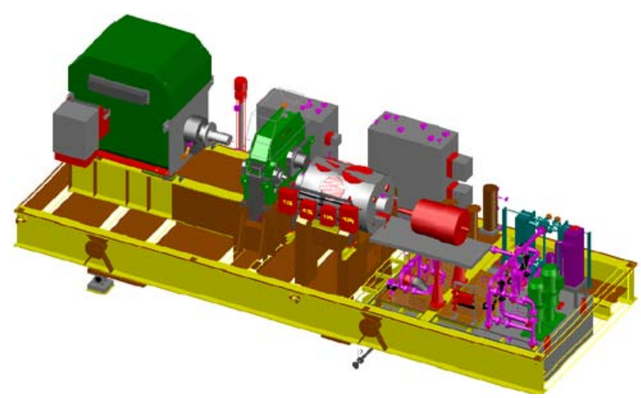


Figure 3: 3D model of the HP2/3 compressor package

All compression packages were ordered from GE O&G, Thermodyn at Le Creusot in France. As a general policy

established by the Rotating Equipment Group of TOTAL, the order included a comprehensive program of tests in addition to the standard tests specified in API 617. One of these tests is the full load string test of the gas compressor package which includes all contractual auxiliaries pertaining to each package.
The aim of a string test is to demonstrate the mechanical integrity of the whole shaft line all along the operating range, under all operating conditions (specified or not) and to check the functionality of the auxiliary systems. Activities include the following:

- Pre commissioning and Pre-test activities on the complete package,
- Individual full load tests for contract driver with back to back test (second motor being used as generator),
- ASME PTC 10 type 1 performance test at nominal conditions of the centrifugal compressor package (the first unit of each type only) using Natural Gas + CO2 mixture to comply with type 1 criteria,
- Full load / full speed test for the complete compressor package (the first unit of each type only). The aim of the full load string test is to check the mechanical integrity of the entire shaft line. Functional tests are also conducted during full load string test activity.
- Part Load or No Load String Test is also required for identical shaft lines (the first unit being full load and type 1 tested as previously mentioned).

String test also includes compressor sequences testing, such as startup, emergency and normal shutdowns, hot re-start. The mechanical behavior and integrity of the complete shaft line is also checked thanks to speed or lubricating parameter variations. The functional tests include some trip simulations, filter change over, etc. These test sequences are driven from the Unit Control Panel (UCP) directly linked to the compressor packages using serial communication link. With the software weight increase, an intensive communication test is the insurance of a reduced trouble-shooting on site.

Prior to the string test activities on the complete packages and as part as preliminary validation of each sub-component, individual tests are performed as follows:

- Electrical motor in standalone condition and back to back test,
- ASME PTC 10 type 2 performance test in similitude conditions (the first centrifugal compressor unit of each type only) in GE O&G Thermodyn test bench using Nitrogen for HP2/3 compressors,
- Mechanical running test of each compressor (including spare bundle) in no load or very low load conditions as per API617,
- Mechanical running test of all gear boxes, including the spare runner as per API613,
- Commissioning of lube oil and dry gas seal systems as per API614,
- Commissioning of UCP,
- Balancing of low and high speed flexible couplings,
- Dry gas seals in static and dynamic conditions,
- etc.

However, due to the complexity of the package and the number of sub-components, such string tests are required prior to shipment and installation on the FPSO, and of course prior to start-up at site. Failing to perform such tests may result into having non operable compressors on site, thus putting production at stake and generating unacceptable flaring for an undetermined period of time. Such risks exist and thus cannot be ignored.

All compressor packages were string-tested at GE O&G Thermodyn facilities in Le Creusot, France in the dedicated outdoors open area.

Figure 4: HP1 gas compressor package on the test facility, Le Creusot, France

OBSERVATIONS - PROBLEM STATEMENT

In March 2012, during the ASME PTC10 Type 1 test of the HP2/3 compressor train A, a high level of 2X vibrations was recorded on radial probes mainly evidenced on compressor and gearbox (pinion) drive-end (DE) side. The unit was operating at around 94% of the rated speed (~11870 rpm) in guarantee point conditions.

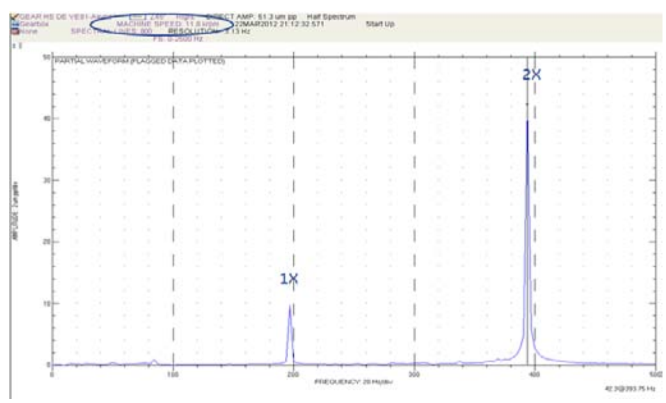


Figure 5: Vibrations spectrum

This level of 2X component was decreasing as a function of the load when moving the point from the right (stone wall) to the left (surge) limit of the operating range while the 1X component remained stable.

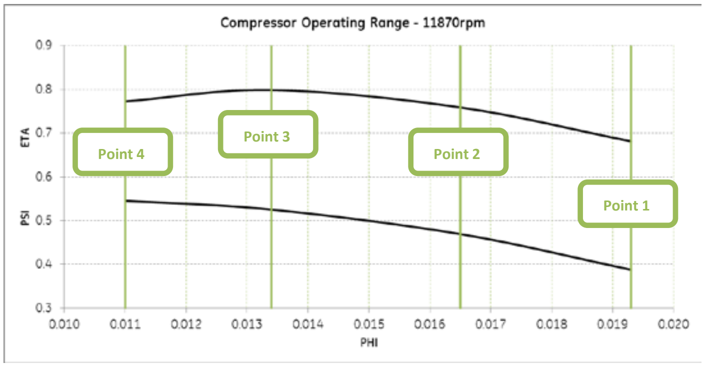


Figure 6: Compressor operating range - 11870 rpm

		Point 1	Point 2	Point 3	Point 4
Compressor	1X	10	10	10	8
	2X	28	22	16	8
Gearbox	1X	10	10	10	10
	2X	44	38	23	10

Table 1: Maximum 1X and 2X vibrations in µm peak to peak recorded on compressor and pinion DE sides over the compressor flow range

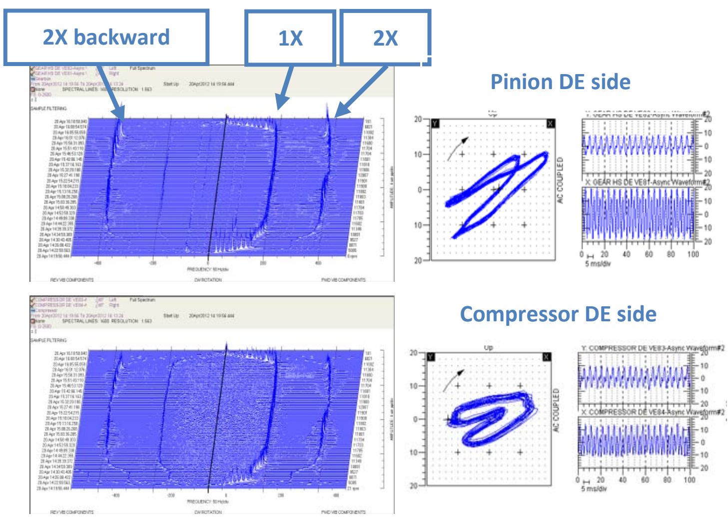


Figure 7: Waterfall and orbits recorded on respectively pinion and compressor Drive-End side during the performance test

Further to this test, the level of vibrations recorded on the two first points was not acceptable for a safe operation of the train at site. Indeed, such vibration level could compromise the operability of the train and might damage some parts during continuous operation.

Nevertheless, the mechanical running test at full pressure and full density was performed to double-check the mechanical behavior of the train at full load.
The unit ran continuously at three different speeds to cover the full compressor map:

- Maximum Continuous Speed (MCS = 13193 rpm)
- Rated Speed (100% = 12565 rpm)
- Minimum Operating Speed (MOS = 8795 rpm)

The overall and 1X vibration levels recorded at those speeds were acceptable within API 617 7th §2.6.8.8 requirements, and remarkably, no 2X vibration was observed.

However when looking at the Bode diagram of the compressor and gearbox drive-end side, a peak of 2X vibrations was clearly evidenced at a speed of 11800 rpm (2X frequency ~394 Hz), close to the guarantee point speed, with an amplification factor approaching 50.
The 1X component remained stable and acceptable over the full operating speed range.

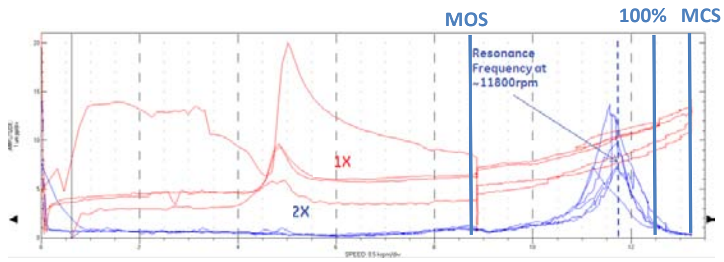


Figure 8: 1X and 2X vibrations recorded on the pinion Drive-End side during the full load test (Bode diagram)

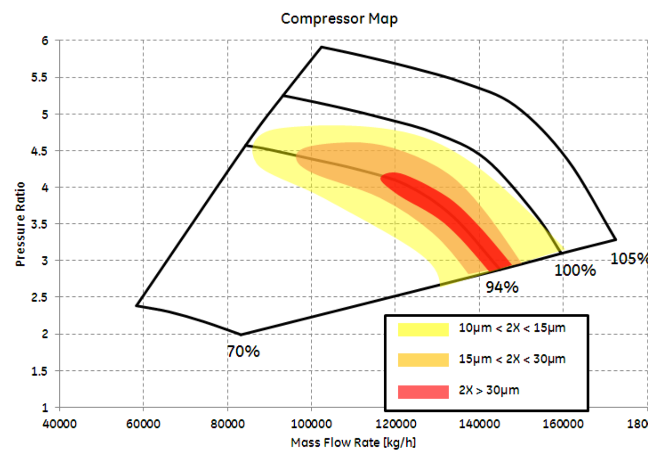


Figure 9: Compressor map and forbidden operating zone

As a consequence, the test was rejected and a test campaign was started to identify the root cause of this phenomenon.

INVESTIGATIONS AND ROOT CAUSE ANALYSIS

SHAFT-LINE INSPECTION

As a first step, a visual inspection of the whole high-speed shaft-line, including the compressor, the coupling and the pinion, was performed to check the integrity of the train.
No specific non-conformity was detected except rubbing on the pinion shaft in front of the oil retaining ring.

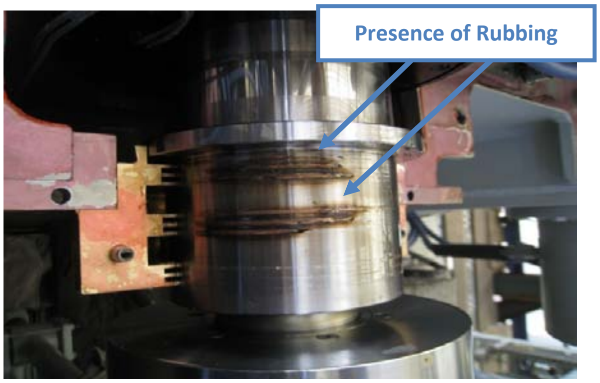


Figure 10: Pinion shaft-end in front of oil retaining ring

Such contact evidence could explain the backward 2X vibrations observed on the pinion waterfall (see figure 7).
Further to this observation, the oil retaining ring was modified from bronze to aluminum with larger clearances to avoid any rotor/stator contact.
However, after running up to the resonance frequency, the same level of forward & backward 2X vibrations was still visible on the compressor and pinion drive-end spectrum.

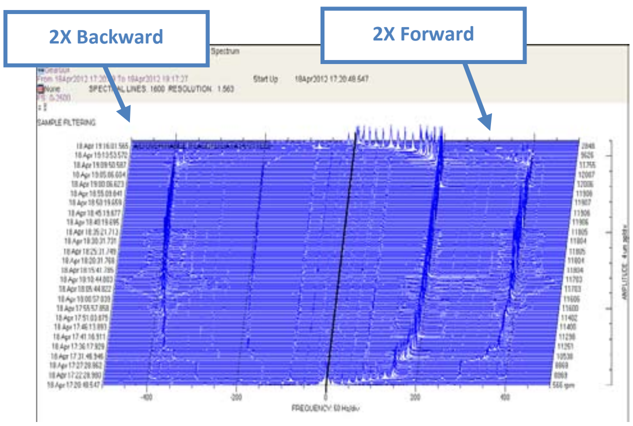


Figure 11: Waterfall recorded on pinion drive end side

Main & spare rotors were also swapped on the pinion first and then on the compressor without any impact on the super-synchronous phenomenon.

TEST CAMPAIGN

A Test campaign was carried out right after the official test to identify the source of the 2X excitation (~400 Hz) and to identify which component of the shaft-line was responding to this excitation. If at least one element could be attenuated, the phenomenon could then be minimized or even removed to ensure a safe operation at site.

Potential sources of excitation

The purpose of this first test sequence was to isolate any potential source of excitation that could affect the level of 2X vibration when operating at the resonance frequency.

- Shaft-line alignment

The 1st source that comes to mind for a 2X vibration is an excitation coming from a misalignment of the shaft-line.
First of all, the pre-stretch installed on the coupling was deeply checked taking into account the real test conditions to evaluate the thermal growth of both shaft-ends during operation (compressor and gearbox). Pre-stretch values were in line with respect to what was installed.
Then, the alignment of the high-speed shaft was checked and validated at standstill. While running at 11800 rpm, a misalignment of ~0.4mm was measured between the compressor and the gearbox high-speed shaft axis. Despite being within the coupling capability, this slight misalignment was compensated in cold conditions by -0.4mm in order to minimize the misalignment in operation when running at the resonance frequency.
After this small adjustment, no improvement was observed (~40 µm max on gearbox drive-end side and ~20 µm max on compressor drive-end side). Subsequently, the alignment and coupling pre-stretch were disregarded as the source of the phenomenon.

- Electrical excitations

The 2nd area that was thoroughly checked is the level of torque ripples and the harmonic content coming from the electrical system (frequency converter and motor). Harmonic distortions, if any, might excite torsional modes but also lateral modes of the gear high-speed shaft.
Electrical excitations from the Variable Speed Drive (VSD) were measured through the low speed torque spectrum to check the presence of frequencies close to 400 Hz.

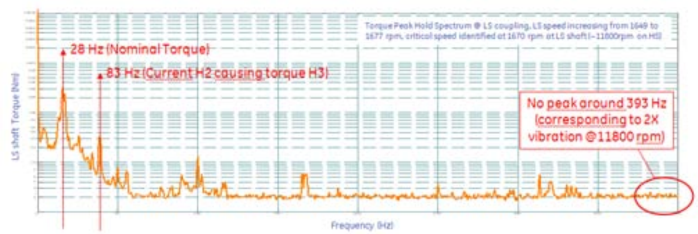


Figure 12: Torque spectrum measured on the low speed shaft

As a result of this test, the torque spectrum measured on the low-speed shaft did not show any specific frequencies at around 400 Hz.

To confirm that harmonics generated by the VSD did not have any adverse effect on the shaft-line, an emergency shutdown was performed while the unit was running at the resonance frequency. Indeed, as soon as a trip from VSD was initiated, there was no more driving torque from the motor.
The analysis of vibrations during the deceleration of the unit shows that the 2X activity still remained at the same level as before and then suddenly dropped down at 11200 rpm (roughly one second after trip) after crossing the resonance frequency.

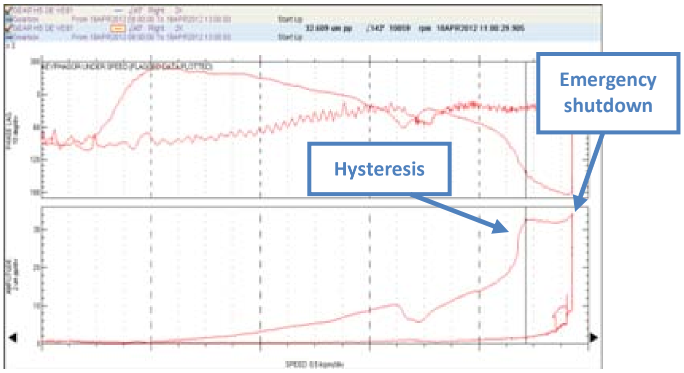


Figure 13: Vibration trend recorded on pinion DE side

As a conclusion of this test, the electrical equipment (motor and VSD) was confirmed not to be the source of the excitation. The hysteresis phenomenon seemed to be more related to a mechanical locking of the high-speed shaft-line than to an electrical phenomenon.

- Load sensitivity

A test was also performed to check the phenomenon sensitivity to a load variation.
The machine was run at the resonance frequency and the load adjusted step by step.

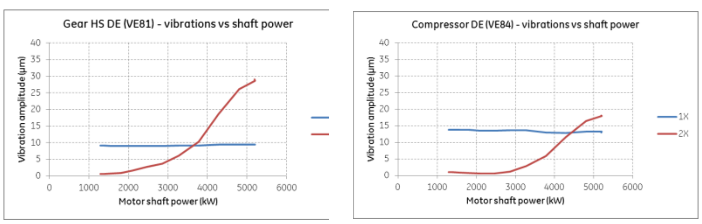


Figure 14 : Evolution of 1X and 2X components as a function of the load when operating at 11800 rpm (on resonance)

Both 2X vibrations and load seemed to be well correlated. However during this test, it was noticed that the resonance frequency might have shifted up or down mainly due to the variation of the pinion journal bearing characteristics.
In this test sequence, no speed change was performed to further quantify the trend.

Shaft-line response

The aim of this test sequence was to play either on the mass or on the stiffness/damping of the system to determine how sensitive the resonance frequency is.

- Bearing Oil Inlet Temperature Sensitivity

The purpose of this test was to measure how the temperature of the oil supplied to the gear and compressor bearings could affect the 2X vibration.
Consequently, the bearing oil inlet temperature was increased from 41°C up to 60°C to reflect the allowable range at site in continuous operation.

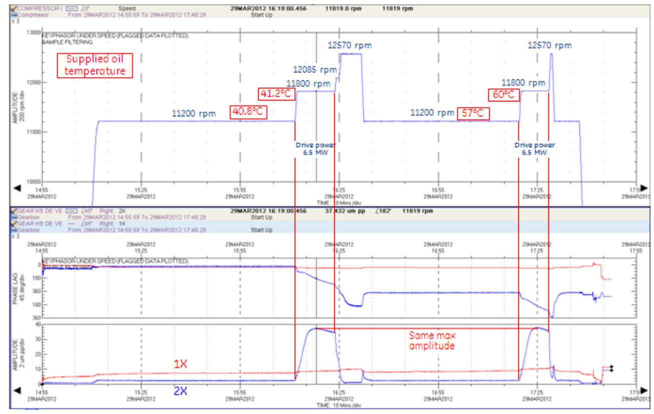


Figure 15: Vibration trend recorded on pinion DE side at different oil inlet temperature

Overall, no visible impact on the 2X vibration over the range of temperature neither on the critical speed value (~11800 rpm) nor on the vibration amplitude (~40 µm on pinion drive-end side) could be observed.

- Overhang sensitivity

Since the phenomenon was identified on the high speed shaft only, a one kg weight was added in overhung on the pinion drive-end shaft to check its impact on the resonance frequency.
This test was performed to identify whether the gear or the compressor was responding to the 2X excitation since coupling membranes were supposed to disconnect the lateral behavior of both components.
To get a relevant comparison, the test was performed in conditions similar to the original test conditions (load, oil temperature…).

Overall, the unit showed a similar behavior but the 2X peak was recorded approximately 600 rpm lower than the baseline test.

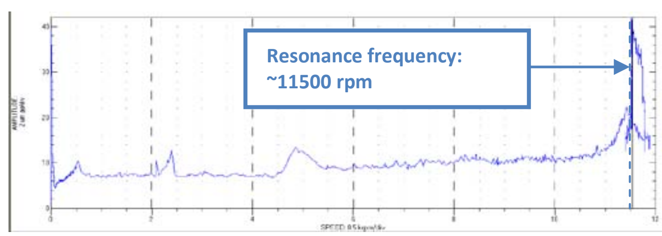


Figure 16: 2X vibrations recorded on pinion DE side during the ramp-up and the coast down of the unit

In addition, three other key-findings were noted during this sensitivity test:

While running over the resonance frequency, the compressor vibrations are fairly proportional to the vibrations recorded on the pinion but lower, whether a weight is added or not *i.e.* whatever is the resonance frequency.

Figure 17: Typical vibration trend recorded on compressor and pinion DE side while oscillating over the resonance frequency

In addition, Pinion and Compressor drive-end shaft vibrations are fairly out of phase (between 140° and 165°) auto-stimulating the 2X excitation.

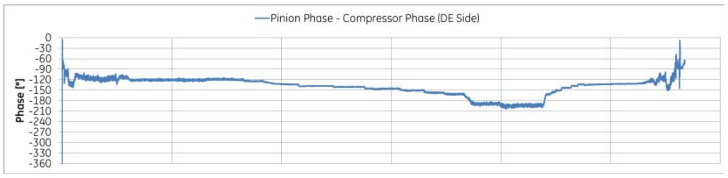


Figure 18: Pinion phase minus compressor phase while oscillating over the resonance frequency

When approaching the resonance frequency, the 2X activity starts to be visible first on the pinion and then is propagated to the compressor drive-end shaft through the high-speed coupling.

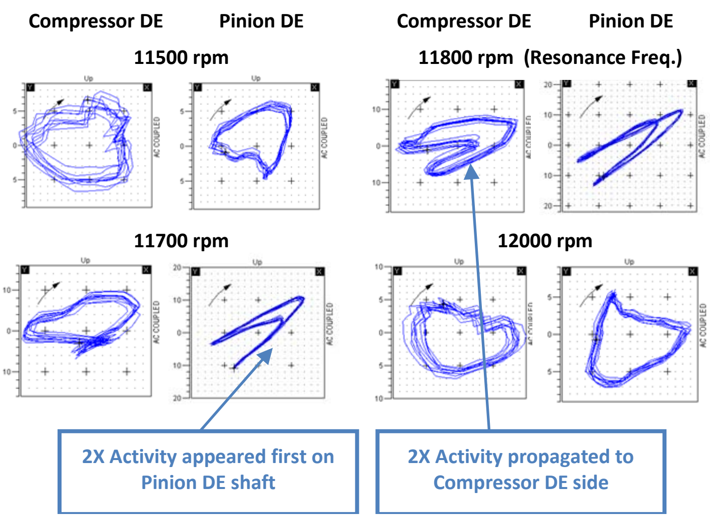


Figure 19: Compressor and pinion drive-end shaft orbits when crossing the resonance frequency

Consequently, this test demonstrated the contribution of the pinion in the response to 2X vibrations and the limited flexibility of the coupling membranes.

- Symmetrical Coupling

A final test was performed using a coupling hub with symmetrical membranes on both sides, resulting in a heavier hub on gear side but symmetrical stiffness on each side.
Initially, the coupling was composed of asymmetrical membranes since the cone diameter was larger on the compressor (100 mm on the compressor versus 80 mm on the pinion).

The purpose of this test was to check whether a symmetric coupling could eliminate the 2X excitation as a result of homogeneous stiffness and to check the reduction of the critical speed as a result of increased hub weight as observed during the previous test.
The results are presented below:

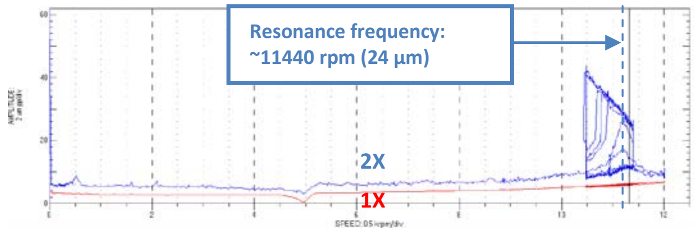


Figure 20: 1X and 2X vibrations recorded on pinion DE side during the ramp-up and the coast down of the unit

2X vibrations remained despite the use of symmetrical membranes. The critical speed moved from 11800 rpm down to 11440 rpm, *i.e.* 700 rpm in 2X as expected from the previous test due to a higher coupling weight.
Additional tests on hysteresis phenomenon showed that the longer the system is stabilized around the critical speed, the longer vibration remains when decreasing the speed (drops at a lower speed from higher amplitude). This assessment tends to confirm a progressive locking in the system when it is submitted to high amplitude vibration (*i.e.* when running at the resonance frequency).

Outcome

Overall, during this test campaign it was confirmed that the source of 2X excitation was not electrical but rather mechanical. It engaged the whole high-speed shaft-line and the phenomenon looked to be correlated to the load and to be sensitive to the overhung momentum of the pinion. Finally, the orbit shapes close to the resonance frequency showed that the pinion seemed to respond first and then drove the compressor drive-end shaft through a coupling that was not as flexible as expected.

ROTORDYNAMIC MODELLING

Compressor & Gearbox Critical Speeds

First of all, the lateral behavior of the compressor and the pinion was calculated separately since both units were supposed to be disconnected through the flexible coupling membranes.
This analysis reflected the “as-tested” bearing micro-geometry (clearance, preload…) and oil inlet conditions.
As a result, the 2nd bending mode of the compressor crossed the 2nd harmonic of the operating speed at 367 Hz.
The corresponding amplification factor was high (~60) since vibration nodes were located close to the bearing axis.

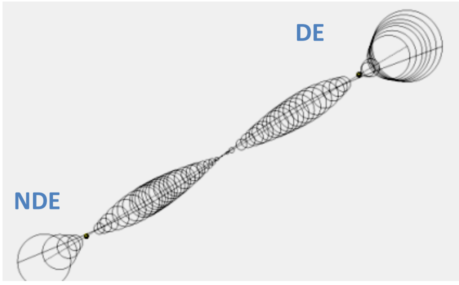


Figure 21: Compressor 2nd bending mode shape (367 Hz)

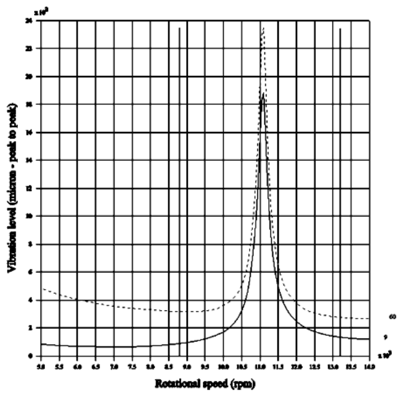


Figure 22: Response of the compressor alone to 2X excitation at probes location (DE and NDE)

On the other side, the 1st pinion natural frequency (overhang mode) crossed the 2nd harmonic of the rotating speed close to 355 Hz. The calculated amplification factor was around 6.

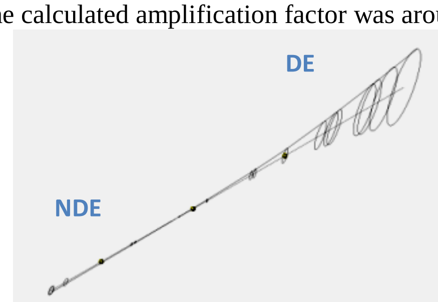


Figure 23 : Pinion overhang mode shape (355 Hz)

High-Speed Shaft-Line Rotordynamic Assessment

- Critical Speeds Calculation

The full high-speed train was then schematized into mass and stiffness in order to check the impact of the coupling in the lateral behavior of the shaft-line. As for single models, bearings were described as matrix composed of direct and cross-coupled stiffness and damping terms.
An angular stiffness of 100N.m/deg was also taken into consideration (as indicated on Manufacturer's coupling drawing) to simulate the theoretical flexibility of the coupling membranes.

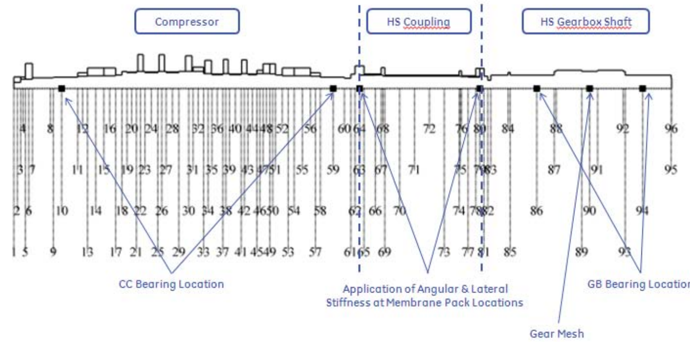


Figure 24: High-speed shaft-line modeling

The calculated Campbell diagram is presented below as well as the mode shapes of interest:

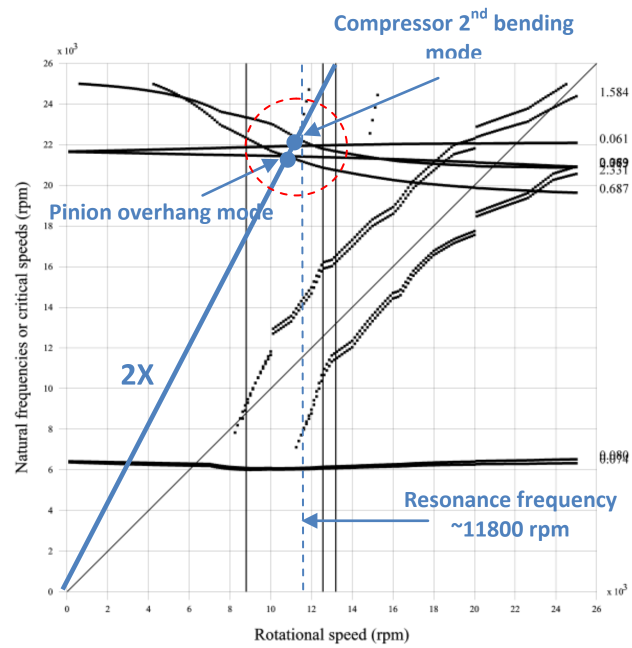


Figure 25: High-speed shaft line Campbell diagram

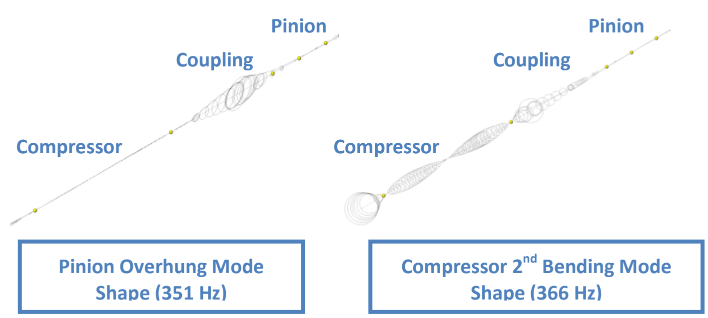


Figure 26: High-speed shaft-line mode shapes around 400 Hz

Frequencies calculated on the full shaft-line model were in line with those calculated from the single model.
It was noted that the amplification factor measured during the test looked to be fairly in line with the amplification factor calculated on the Compressor 2nd bending mode.

- Overhung Momentum Sensitivity

The overhung momentum of the pinion was increased in the model to check whether the critical speeds were as sensitive as what had been observed during the test campaign.
This analysis was only focused on the compressor 2nd bending mode and the pinion overhung mode close to the measured resonance frequency.

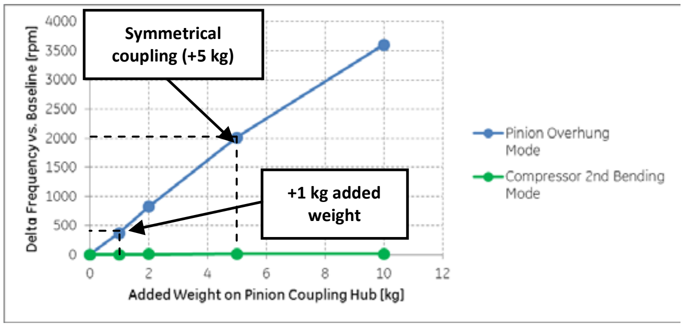


Figure 27: Sensitivity of compressor 2nd bending and pinion overhang modes to added weight on pinion coupling hub

As expected, considering a full flexible coupling, the variation of overhung weight at pinion shaft end only impacted the pinion natural frequency and not the compressor 2nd bending natural frequency. However, compared to the test results, the model was a bit overestimating the impact (~400 rpm per added kg) when the added weight was getting significant.
The prediction remained almost valid for the trial test with +1 kg but not for the test using a coupling with symmetrical membranes (~+5 kg).
This analysis was not reflecting the behavior that was observed during the tests.

- Coupling Membrane Stiffness Sensitivity

Since the coupling was suspected not to be flexible enough, another analysis was performed to check how the frequencies and the response to overhung would be sensitive to an increase of angular stiffness in coupling membranes.
For this calculation, both membranes were assumed to have the same mechanical properties and to provide no additional damping to the system.
A membrane was modeled considering 2 separate nodes at the same location linked by an infinite stiffness in the transversal direction to avoid any lateral motion between the 2 nodes.

The complete stiffness matrix considered in the coupling membranes is presented below:

	X1	Z1	θx1	θz1	X2	Z2	θx2	θz2
Fx1	Kxx	0	0	0	-Kxx	0	0	0
Fz1	0	Kzz	0	0	0	-Kzz	0	0
Fθx1	0	0	Jθx	0	0	0	Jθx	0
Fθz1	0	0	0	Jθz	0	0	0	Jθz
Fx2	-Kxx	0	0	0	Kxx	0	0	0
Fz2	0	-Kzz	0	0	0	Kzz	0	0
Fθx2	0	0	-Jθx	0	0	0	-Jθx	0
Fθz2	0	0	0	-Jθz	0	0	0	-Jθz

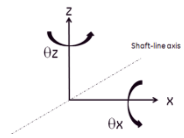


Figure 28 : Stiffness matrix definition in coupling membranes including angular stiffness values

Where:
F is the force [N].
x, z, θx and **θz** are respectively the lateral and angular displacement with respect to x and z axis [m].
K is the lateral stiffness [N/m].
Jθ is the angular stiffness [N.m/deg].

As a result, the Compressor 2nd bending mode remained at the same frequency whatever the angular stiffness. However, it was sensitive to a variation of overhang on pinion coupling hub. On the other side, the pinion sensitivity was slightly reduced and might have explained part of the discrepancy observed between test and calculation.

Compressor 2nd bending mode — Pinion overhang mode

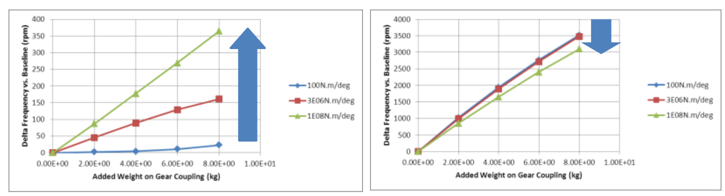


Figure 29: Compressor and pinion mode sensitivity to overhang by increasing the membranes angular stiffness

A similar analysis was performed by increasing the shear cross-coupled stiffness in the coupling membrane:

	X1	Z1	θx1	θz1	X2	Z2	θx2	θz2
Fx1	Kxx	-Kxz	0	0	-Kxx	0	0	0
Fz1	Kzx	Kzz	0	0	0	-Kzz	0	0
Fθx1	0	0	0	0	0	0	0	0
Fθz1	0	0	0	0	0	0	0	0
Fx2	-Kxx	0	0	0	Kxx	0	0	0
Fz2	0	-Kzz	0	0	0	Kzz	0	0
Fθx2	0	0	0	0	0	0	0	0
Fθz2	0	0	0	0	0	0	0	0

Figure 30: Stiffness matrix definition in coupling membranes including shear cross-coupled stiffness values

The same trend was observed regarding the sensitivity to the overhung:

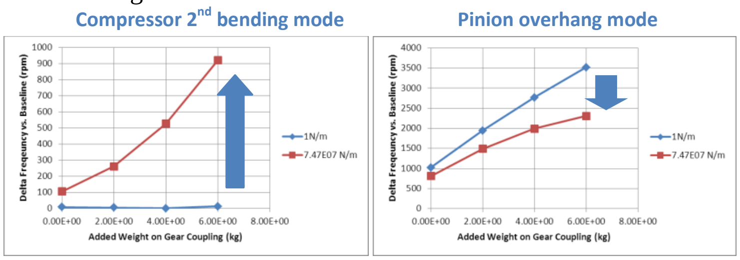


Figure 31: Compressor and pinion mode sensitivity to overhang by increasing the membranes shear cross-coupled stiffness

The damping of the system was also dropping quickly when the cross-coupled stiffness was in the range of the lateral stiffness of each component:

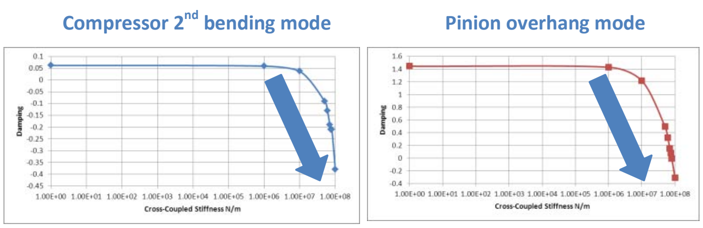


Figure 32: Compressor and pinion modes damping as a function of the membranes shear cross-coupled stiffness

This trend could explain a part of the measured amplification factor when the coupling membranes were constrained out of phase while running at the resonance frequency.

Finally a last calculation was performed playing on the angular cross-coupled stiffness with a marginal impact on the system.

Despite the complexity to simulate precisely what was observed at the test bed, all those results tended to confirm there was something happening in the coupling membranes.

Outcome

The test campaign as well as the calculation results confirmed that the phenomenon is a shaft-line resonance responding to a 2X excitation. This phenomenon is generated by the coincidence of 2 critical speeds (compressor and pinion) crossing the 2^{nd} harmonic of the rotating speed and connected by coupling membranes not as flexible as expected at such high level of frequency.
It was demonstrated during the test campaign that the source of the 2X vibrations was rather mechanical but had not been clearly identified. The only key-parameter was to play on the overhung momentum at the pinion coupling hub to move out the resonance frequency. The design was upgraded accordingly.

DESIGN UPGRADE

The shortest way to move this undesirable frequency out of the operating speed range was to push this resonance frequency beyond the maximum continuous speed. For a safe operation at site, it was commonly agreed between TOTAL & GE to get this mode beyond the trip speed.
As a consequence, the overhung momentum had to be reduced to get the reverse behavior as the one observed during the test while adding weight on the coupling.
This operation was done into 2 steps.

STEP 1: PINION OVERHUNG LENGTH REDUCTION

Purpose of the modification

As a first step, the lateral stiffness of the pinion shaft-end (from drive-end bearing axis to the half coupling center of gravity) was increased to move up the resonance. This modification was done directly on the existing shaft to speed-up the execution.
2 operations were carried out:
-50 mm length reduction on the drive end
-12 mm bearing span increase playing on the position of the drive-end journal bearing.
Overall, the shaft overhang length was reduced by 62 mm.

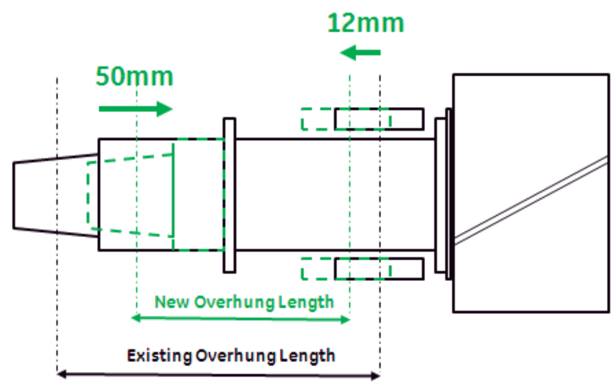


Figure 33: Pinion drive-end shaft upgrade

Consequently, the coupling DBSE (Distance Between Shaft-Ends) had to be increased by 50 mm to fit with the new pinion design without any major impact on the coupling weight (+300 g). Only the spacer was replaced keeping symmetrical membranes as a trial test (heavier hub on the pinion).
With respect to the original design, such modification enabled increasing the stiffness over mass ratio (K/M) by 60%, just considering the lateral stiffness of the shaft-end (from bearing axis to the coupling hub center of gravity) and the mass of the half coupling as the main contributor in the location of the resonance frequency.

	Existing Pinion + Asymmetrical Membranes (Baseline)	Existing Pinion + Symmetrical Membranes	Upgraded Pinion + Symmetrical Membranes
Pinion Overhung Length (L) [mm]	277.9	279.1	217.1
Half Coupling Weight (M) [kg]	16	20.8	21.1
Ratio $1/(ML^3)$ $[kg]^{-1}[m]^{-3}$	2.91	2.21	4.63
Discrepancy versus Baseline	-	-24%	59%

Indeed, the pinion shaft-end could be approximated as an embedded beam from the bearing axis to the coupling hub center of gravity where the lateral stiffness $\mathbf{K = 3EI/L^3}$, **E** is the Young Modulus, **I** the polar inertia and **L** the length of the beam.

$$\Rightarrow \quad (K/M)_{upgrade}/(K/M)_{baseline} = (ML^3)_{baseline}/(ML^3)_{upgrade}$$

Table 2: Pinion upgrade impact on the mechanical properties of the shaft-line

Validation Test

To get a comparative approach, the train was tested in the same conditions as during the official test.
As expected, the resonance frequency moved up at a higher frequency close to 12400 rpm on the 2X component with less amplitude than the initial test.

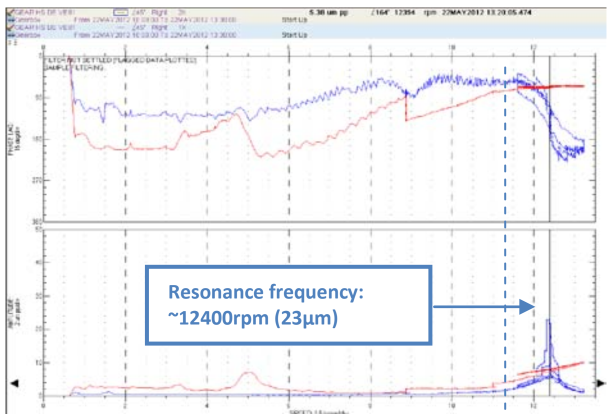


Figure 34: Bode diagram 2X vibrations recorded on the pinion drive-end side

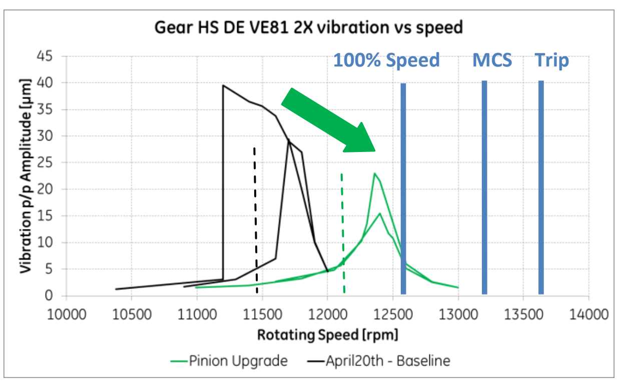


Figure 35: 2X vibrations recorded during the test (green) versus initial test result (in black)

This test confirmed the high contribution of the pinion shaft-end on the super-synchronous phenomenon.
It was also noted that the hysteresis phenomenon observed during the previous tests while coasting down of the unit was no more visible.

Model calibration and prediction

Further to these tests, the full shaft-line rotordynamic model was calibrated to reproduce the sensitivity of the phenomenon to the overhang and to predict the steps forward.
Both angular stiffness and shear cross-coupled stiffness were artificially increased to simulate shaft-line frequencies on the high-speed train.
The aim of this calibration was not to get an absolute value of the frequencies but more to catch the sensitivity of the resonance frequency as a function of the pinion overhang.
The best tradeoff to simulate the frequency variation between each run with respect to test measurements was to consider 1E08N/m and 1E08N.m/deg. respectively as shear cross-coupled and angular stiffness in coupling membranes.
Such high value of angular stiffness could reflect the fact that membranes are not as flexible at high frequency and may be responsible of the locking phenomenon (hysteresis). As a consequence, both Compressor and Pinion shafts are fully connected.

That could be also the main reason why the sensitivity to the overhang is reduced with respect to what was calculated on a single model considering 100% flexible coupling membranes.

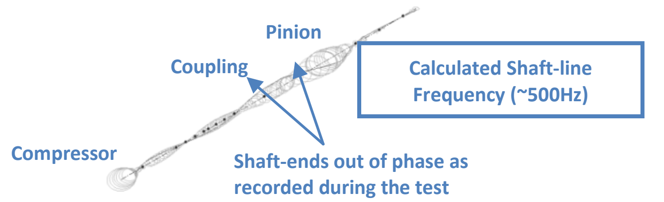


Figure 36: High speed shaft line mode shape around 500 Hz

Difference of 2X Resonance Frequency [rpm]	Test	Calculation considering 100% flexible coupling membranes	Calculation including angular stiffness & shear cross-coupled stiffness
Test 1 - Baseline	~-700	-1384	-886
Test 2 - Test 1	~2000	5500	2476

Figure 37 : Comparison test and calculation while increasing angular stiffness

Baseline: Existing Pinion + Asymmetrical Coupling
Test 1: Existing Pinion + Symmetrical Coupling
Test 2: Upgraded Pinion + Symmetrical Coupling

Nevertheless, although this model was able to capture fairly well the sensitivity to the overhang, it has to be used cautiously since the calculated shaft-line frequency and damping values are still far from what was observed.
Indeed, the accuracy of such high frequency value could be affected by the non linearity of the coupling membrane pack, that was not included in the modeling. In addition, looking at the mode shape, the damping of the system is very sensitive to the shaft-ends deflection.

STEP 2: COUPLING OVERHANG REDUCTION

Purpose of the modification

In addition to the pinion shaft-end upgrade, a second modification was carried out on the coupling itself to minimize as much as possible the overhang momentum.

The purpose of this modification was first to reduce the half coupling weight by reducing the membranes external diameter (back to a non-symmetrical coupling) and then to shift the center of gravity of the overhang weight by considering a coplanar (low moment type) coupling as presented on figure 38.

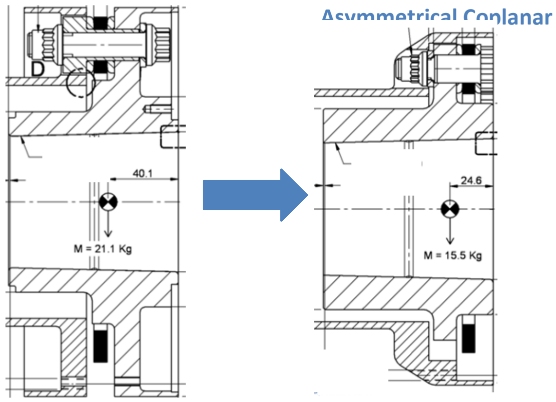


Figure 38: Coupling upgrade

	Actual	Assymetrical + Coplanar
Distance from bearing axis to coupling [mm]	178	178
Coupling Center of Gravity [mm]	40.1	24.6
Half Coupling Weight [kg]	21.1	15.5
Overhung Momentum [kg.mm]	4602	3140

	Existing Pinion + Asymmetrical Membranes (Baseline)	Upgraded Pinion + Symmetrical Membranes (STEP 1)	Upgraded Pinion + Asymmetrical Steel Coplanar Coupling (STEP 2)
Pinion Overhung Length (L) [mm]	277.9	217.1	201.6
Half Coupling Weight (M) [kg]	16	21.1	15.5
Ratio 1/(ML³) [kg]⁻¹[m]⁻³	2.91	4.63	7.87
Discrepancy versus Baseline	-	59%	170%

Table 2: Coupling upgrade impact on the mechanical properties of the shaft-line

Result is approximately 30% overhang momentum reduction with respect to the previous test (Step 1) and ~2.7 time K/M increase compared to the original desing.

Considering the last calibrated model, such coupling arrangement would move the resonance up to 13300 rpm.

Final Test

The new train configuration was finally tested considering the previous test conditions as a matter of comparison. While increasing the speed, the phenomenon appeared close to 13250 rpm, right at the Maximum Continuous Speed at very low amplitude that would be acceptable for a safe operation at site.

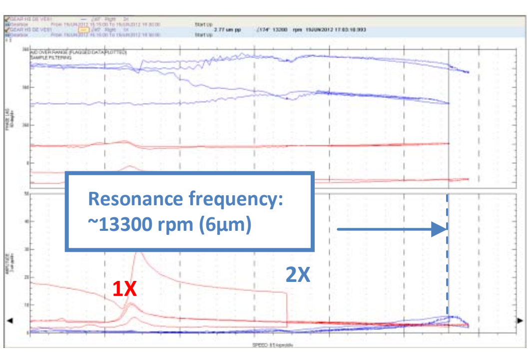


Figure 39: Bode diagram – 2X vibrations recorded on the pinion drive-end side

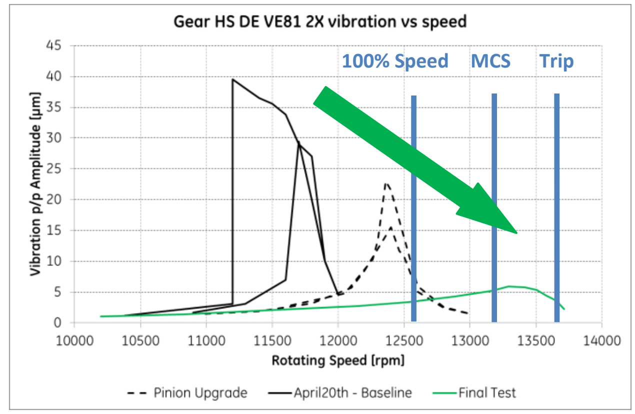
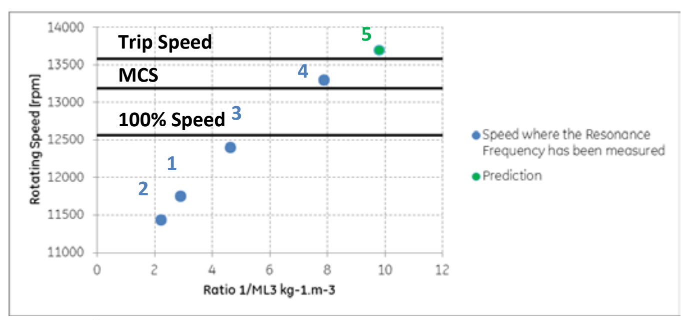


Figure 40: 2X vibrations recorded during the test (in green) versus initial test results (in black)

As a result of this test, the unit was shipped to site as it was. The final coupling spacer was upgraded in titanium to move the phenomenon further out, beyond the trip-speed (~13700 rpm as per the calibrated model).

	Existing Pinion + Asymmetrical Membranes (Baseline)	Upgraded Pinion + Symmetrical Membranes (STEP 1)	Upgraded Pinion + Asymmetrical Steel Coplanar Coupling (STEP 2)	Upgraded Pinion + Asymmetrical Titanium Coplanar Coupling (FINAL STEP)
Pinion Overhung Length (L) [mm]	277.9	217.1	201.6	204.8
Half Coupling Weight (M) [kg]	16	21.1	15.5	11.9
Ratio 1/(ML³) [kg]⁻¹[m]⁻³	2.91	4.63	7.87	9.78
Discrepancy versus Baseline	-	59%	170%	236%

Table 3: Final coupling upgrade impact on the mechanical properties of the shaft-line



1 – Official Test (Baseline)
2 – Symmetrical coupling (heavier hub on pinion side)
3 – Symmetrical coupling + Pinion shaft-end upgrade
4 – Steel asymmetrical coplanar coupling + Pinion shaft-end upgrade (Final Test)
5 – Titanium asymmetrical coplanar coupling + Pinion shaft-end upgrade (Prediction for Site Operation)

Figure 41: Resonance frequency evolution as a function of the K/M ratio

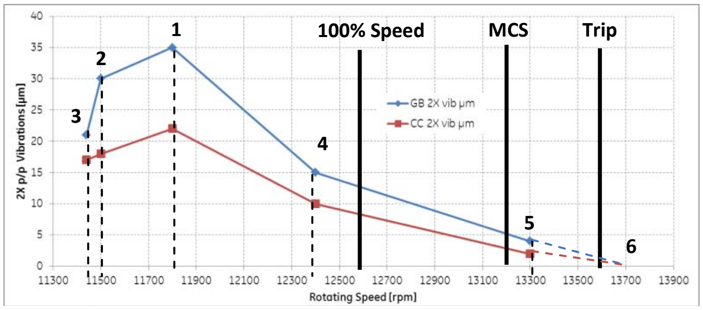
Lessons Learned

The super-synchronous phenomenon was solved by disconnecting the pinion first critical speed from the compressor 4th (2nd bending) critical speed.

Both, pinion and coupling design were upgraded to get 2X vibrations within allowable levels.

The increase of the pinion shaft-end lateral stiffness combined with the reduction of the coupling overhang momentum enabled to increase the ratio K/M by a factor of ~3.5 to move up the 1st natural frequency above the trip speed.

Indeed, the pinion acted as an amplifier of the phenomenon through a coupling that was not able to disconnect the vibrations at such high level of frequency.
The higher was the separation, the less was the amplitude.



1 – Official Test (Baseline)
2 – Test with +1kg added on the pinion coupling hub
3 – Symmetrical coupling (heavier hub on pinion side)
4 – Symmetrical coupling + Pinion shaft-end upgrade
5 – Steel asymmetrical coplanar coupling + Pinion shaft-end upgrade (Final Test)
6 – Titanium asymmetrical coplanar coupling + Pinion shaft-end upgrade (Prediction only)

Figure 42: Evolution of the phenomenon as a function of the overhung on the pinion coupling hub

This experience also revealed the loss of flexibility of the membrane type couplings at high frequency as well as the presence of internal destabilizing forces related to friction forces inside the membrane stack.

Other alternative solutions would have consisted in using sleeve bearings instead of tilting pads on the pinion to increase the global support stiffness, or using a coupling with diaphragms instead of membranes to get known coupling characteristics whatever the frequency and deflection.

As a lesson learned of this test, any coincidence of natural frequencies within the operating speed range of 2 components connected together must be avoided.

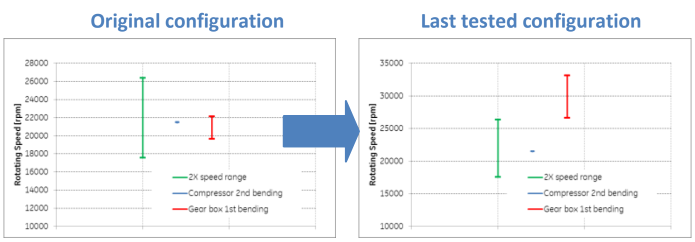


Figure 43: Separation margin between Compressor 2nd bending mode and Pinion overhung mode

A comfortable separation margin shall be considered during the design phase over the full range of bearing stiffness to prevent any issue. On that specific case, the pinion overhang mode and the compressor 2^{nd} bending mode were separated by around 5000 rpm on the 2^{nd} harmonic of the operating speed.

Conclusion

The comprehensive testing sequence of the CLOV HP2/3A compressor allowed the observation of a super-synchronous mode at the guarantee point speed.

The subsequent course of investigations revealed that the flexibility of the coupling membranes were not as well known as engineering team thought it was. Reduced flexibility involved increased cross coupling stiffness terms leading eventually to the phenomenon that has been described in this article.

The lesson learnt from this issue is an additional set of checking on top of the usual verification such as the study of coincidental modes between shaft line components. From now on, coinciding modes between driving and driven equipment shall lead to an increased attention to the coupling type selection.

This issue that appeared furtively during the compressor HP2/3A ASME PTC 10 type 1 test at 94% speed could have had dramatic consequence on the compressor availability and thus on the production and flaring, if it not had been observed and addressed correctly during the factory tests.

In our case, the issue could be investigated and solution was found in three months. It is impossible to estimate how long it would have taken if compressors had been simply shipped and immediately installed on board FPSO. It is not difficult to hypothesize that startup operation could have been delayed by much more than these three months of investigations and tarnish CLOV project story.

This event clearly demonstrates that comprehensive testing helps to reduce equipment unavailability risks.

Testing scope should always be elaborated in a way to reproduce site conditions through testing according to ASME PTC10 type 1 test. Compressor packages should also be tested all over their operating range to exhibit, when existing, all potential vibration phenomena.

Finally, this example must be kept in mind, when discussions on Project schedule tend to consider equipment testing period as a potential time contingency to deliver project quicker. Indeed, it clearly demonstrates that full equipment testing at the factory is mandatory to ensure successful compressors operability on site and eventually the success of the project.

References:

API Standard 617: Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Services.

ASME PTC 10: Performance Test Code on Compressors and Exhausters.