ASIA TURBOMACHINERY & PUMP SYMPOSIUM
SINGAPORE | 22 – 25 FEBRUARY 2016
MARINA BAY SANDS

DESIGN VALIDATION OF HIGH SPEED RATIO EPICYCLIC GEAR TECHNOLOGY IN COMPRESSION SYSTEMS

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ABSTRACT

Epicyclic gear technology is a key factor to support the compression growth strategy in electrified applications, due to the ever increasing transmission ratio required to meet the high compressor speeds. The paper collects the experience of the authors in developing a unique product through its conceptual design, as well as its mechanical and rotordynamic assessment, up to its complete validation with a full speed full load test in a complete unit arrangement.

The main target of high pressure ratio compressors is to fulfil the required compression service with one casing less than traditional technology.

The new compressor technology results in machines which run much higher in speed. To achieve a compressor ratio of ~30 in a single casing, the rotational speed is increase by roughly 40% respect to a traditional compression train, and therefore it becomes the most critical parameter.

The specific requirements of higher speeds at high powers dictate the need for the very high speed ratio gearing, which was critical to the success of the new compressor technology.
The power transmission was jointly developed by the gear and compressor manufacturers. The methods used to evaluate the gearbox configuration options are detailed in this paper, along with the gearbox specific technology design challenges faced.

A detailed overview of the selected epicyclic transmission technology is provided and how the use of the fundamental configuration was able to provide advantages over more traditional arrangements when combined with some application specific design features. Specific address will be provided to the tooth design principles applied, bearing technology and rotordynamic modelling and performance.

The paper will close with details of transmission performance testing from the gearbox manufactures test bed to the full load string test in a complete compressor unit arrangement, at different operating conditions.

INTRODUCTION

The current trend of compression technology is going towards higher and higher power density applications; this means that the future compressors shall fulfil the same service (with same or lower power) with a smaller size. This is to take the advantage of a smaller envelope, easier to fit in applications like offshore platforms. In the specific case, the technology of High Pressure Ratio Compressor (HPRC) was developed with the main target to fulfil the required compression service with one casing less than traditional technology.

The design critical parameter becomes here the rotational speed which is roughly 40% higher than a traditional compression train. At the same time a so high speed introduces mechanical and rotordynamics challenges for the compressors and the power transmission chain.

Overall the novelties content in the gear/compressor design is very high and therefore the Authors' Company decided to develop a specific prototype to validate them.

The compressor prototype was tested in the Company workshop at full speed full load and all the relevant details will be part of a dedicated paper.

Clearly the mentioned compressor technology needs a sound and robust gearbox in order to achieve the required high running speed. For the specific case a gear ratio of 12.3 at a rated power of 14.5MW was needed. The selected transmission technology for this prototype is the compound epicyclic. Of course the gearbox has been tailored on the specific application and jointly developed by the compressor and gearbox manufacturer in order to meet the performances and mitigate the technical risks during testing phase.

Actually there are no specific requirements provided by the international standards about the epicyclic gear characteristics in terms of design, manufacturing and testing. Neither API613 nor API677 show paragraphs specifically applicable to the mentioned gear technology. By the way it is technically possible to follow the API rules of the *Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services*, also for an epicyclic gearbox. It is the case of the gear presented in this paper, which can be considered full API613 compliant.

However, the application of API613 design requirements, on top to those provided by the AGMA, which shall be considered the mandatory baseline, determines a general increase of the gear rating and therefore of the gear dimensions and weight. The reason is because the API Service Factors do not consider the power distribution concept, which is intrinsically a characteristic of the epicyclic gearboxes, and which allow to accept smaller SF, without jeopardizing the operating life of the component. No technical impacts are foreseen due to the application of the API613 manufacturing and testing requirements.

Said that, when the API compliance is required, it is suggested to make exception to those requirements which can alter the optimum equilibrium between the gears required rating and components sizing.

STATE OF ART

Electrification played a fundamental role in the compression systems for Oil and Gas industry, and it will continue in the next future, providing efficient solutions to help customers address some of the world's most pressing challenges, especially in off-shore applications. Together with the driver and driven equipment, the torque transmission components, as well as the speed increaser gearboxes, have been developed during the past years to fulfill the industry needs.

Nevertheless, for what concerns the gearboxes, the Oil and Gas community is today facing the boundary limit of this developing process, at least for the most popular and used devices, i.e. the parallel offset, single stage, double or single helix gears (see Figure 1). The mentioned gearboxes are the most widely distributed in the industry, because of their simple architecture and robustness.

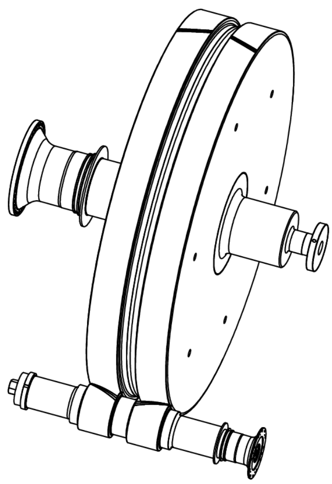


Figure 1 – Gear-set drawing for gear ratio $\tau \approx 9$

In order to define the limiting factors in gear design, it is needed to translate the external operating conditions, like transmitted power, input and output speed, into the specific gear design parameters. That done, we can summarize here below those parameters which define the technological limit in rotordynamics, fluid dynamics, kinematics, lubrication, material stress and fatigue conditions, heat treatment and manufacturing processes:

- Pitch Line velocity (PLV);
- Shaft dimension and bearing span;
- Pitting, bending and scuffing service factors;
- Bearings loads and journal velocity;
- Gear volume and L/D ratio;

The responsibility of gear manufacturer is to achieve the best compromise among these parameters, but in some cases it is not possible to maintain all of them within the comfort zone, because they have a negative impact on each other.
The references collected by the authors in the graph of Figure 2, are relevant to the parallel offset, single stage, speed increasing gearboxes for the moto-compressor applications in the Oil and Gas services.

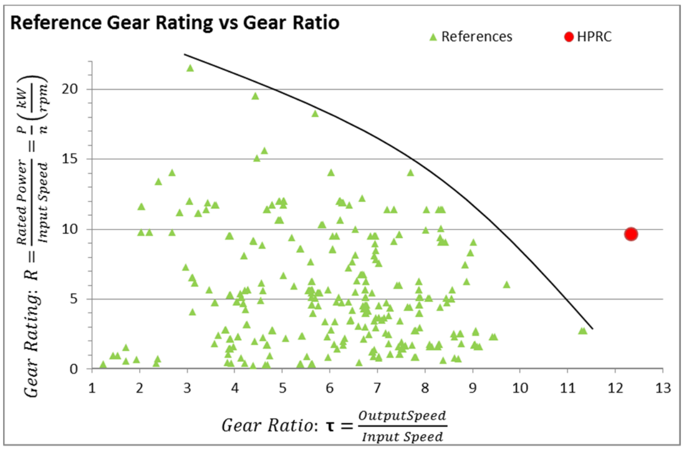


Figure 2 – Gear Rating (R) vs Gear Ratio (**τ**) – references

These references reflect the relationship between design limiting factors like the rated power (P) and therefore the gear rating (R), the input speed (n) and speed ratio (**τ**): increasing the rated power for a specific input speed, the speed ratio cannot overcome certain values, and vice versa for higher speed ratio the rated power shall be kept within certain limits.

This means that, considering all the design constrains, a parallel offset gearbox can be design to maximize the transmitting power capability or the speed ratio, as clearly shown on Figure 3. In the same figure, it is shown in red the point that represents the operating conditions (P, n and **τ**), for the HPRC speed increasing gearbox. It is clear that the state of art of the parallel offset gears cannot fulfill the next compression technology requirements, and therefore a reliable and robust solution is needed for applications with rated power and speed ratio, respectively above 10MW and 9.5÷10.

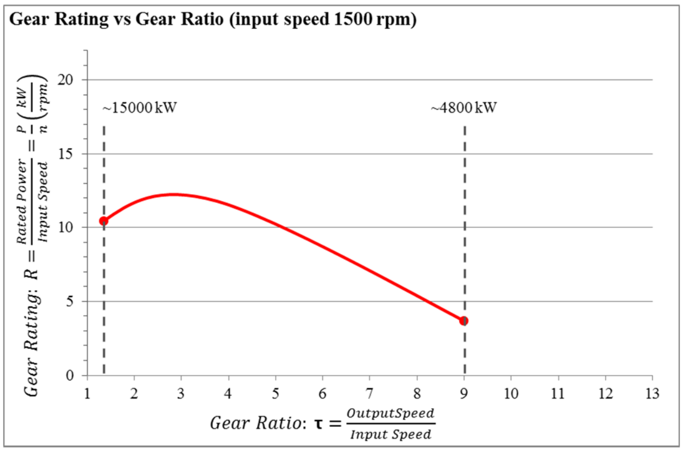


Figure 3 – Gear Rating (R) vs Gear Ratio (**τ**) – allowable limits

DESIGN CONCEPT EVALUATION

Given the step change in required performance envelope for the HPRC application in terms of coupled high speed operation with high power transmission this introduced a number of specific challenges for a traditional transmission configurations as applied in the oil and gas sector.

To address this challenge from an open view point a technology capability and performance evaluation was conducted for a range of transmission configurations and technology types.

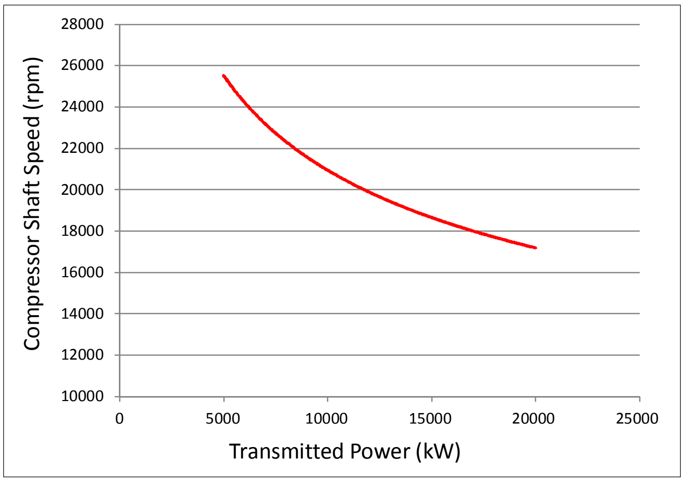


Figure 4 – HPRC Power Speed Curve

This evaluation was used to establish which transmission technology or technologies would be required to provide a solution to support the proposed HPRC family.

The base line evaluations, in terms of compressor operation, were set around the principle performance curve as presented in Figure 4. As can be seen from the power speed curve presented in relation to 'traditional' centrifugal compressor machines the ratio of absorbed power to operating speed is much higher which results in the required transmission operating in significantly higher energy state. This high energy state combined with the requirement for electric motor driven applications presents some fundamental physical challenges to the transmission operation, the kes components of these being:

- Overall speed/gearing ratio
- Pitchline velocities of gears
- Bearing loads
- Bearing journal velocities
- Rotating element thermal conditions
- Structural integrity of rotating components (dynamic stress)

The range of transmission technologies assessed were benchmarked against their ability to satisfy the required loading conditions when considering the fundamental physical parameters outlined above. The main types of technology / configuration considered were:

- Single Stage Parallel Shaft
- Multiple Stage Parallel Shaft
- Single Stage Epicyclic
- Miultiple Stage Epicyclic
- Compound Epicyclic
- Multishaft/Layshaft

Figure 5 shows the results for pitchline velocity, bearing journal velocity and power loss for parallel shaft and compound epicyclic arrangements against two design points on the extremities of the required HPRC operation curve. As can be seen the current single stage parallel shaft technology is principally incompatible with the step change introduced by the HPRC duties as a result of the high transmission ratios required. For almost all duties a two stage parallel shaft solution would be required and as Figure 5 shows the for the parallel shaft solutions both pitch line and bearing journal velocities are at or beyond the point of general operating experience for such applications. Whereas the compound epicyclic configuration overcomes some of these constraints with notably lower dynamic parameters for the defined p curve that current transmission technology would be incompatible with the step change HPRC duties.

The limiting line presented is based around API limits for a case carburized gear (the most compact solution possible within the API design criteria). The same information is also presented for the compound epicyclic which shows that for both pitchline velocities and bearing journal velocities these sit within general industry experience limitations. It is for these principle reasons that epicyclic technology provides a technical advantage above the other arrangements considered.

Specifically it is the compound epicyclic configuration which allows for both the high speed ratios and high speed-power combinations to be accommodated across the entire HPRC family whilst aligning with existing industry performance standards.

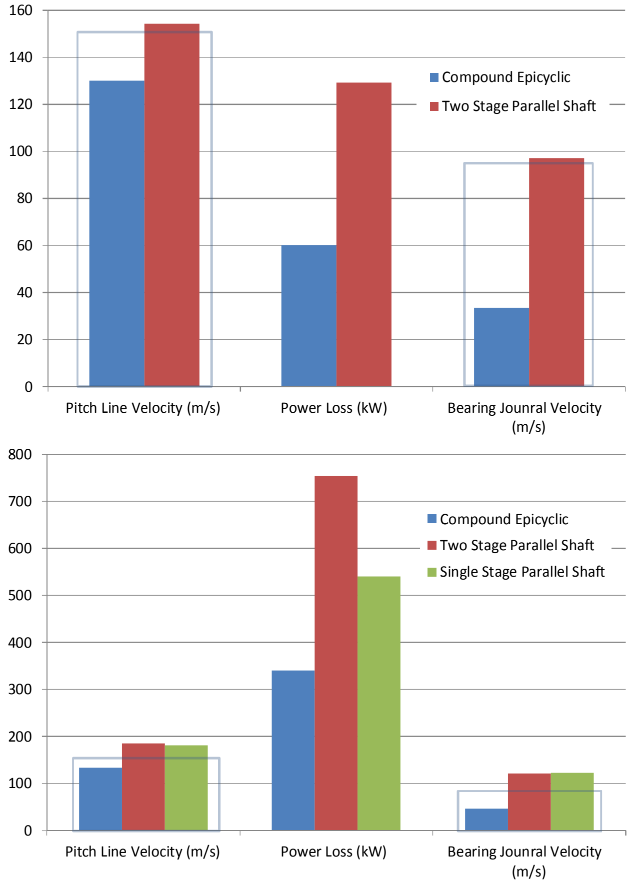


Figure 5 - Parallel Shaft vs Epicyclic Performance

Figure 6 shows the basic construction principle of the compound epicyclic. Figure 7 also shows the principles by which the epicyclic, and more specifically compound epicyclic, results in less arduous operating parameters than the equivalent single stage parallel shaft construction.

In addition to the fundamental performance parameters already addressed another key area of focus for the HPRC application is the rotordyanamics. Due to the high operating speeds coupled with the high power demands this results in elements of increased mass on the high speed rotor shaftlines than would be typical for current compressor installations.

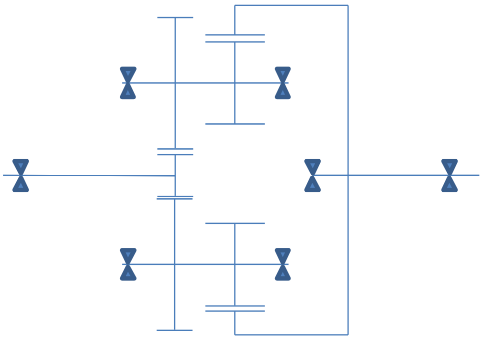
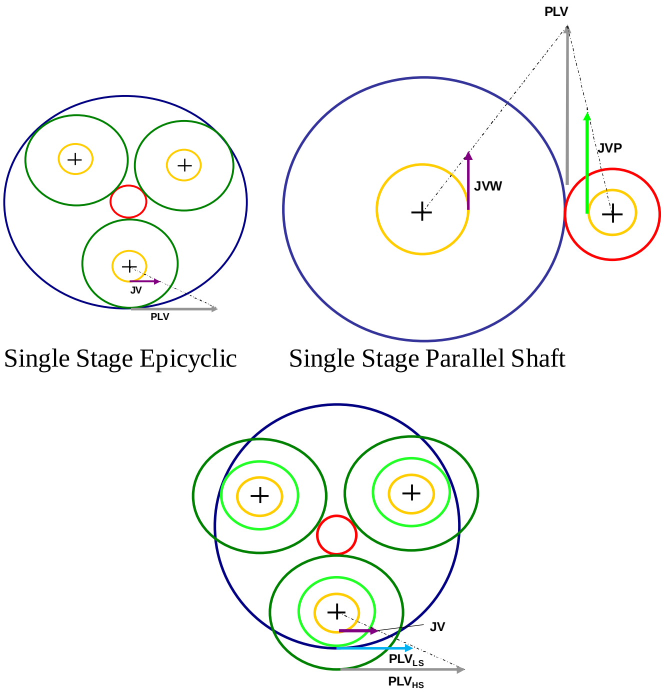


Figure 6 - Schematic of Compound Epicyclic Principle

To address this aspect in the conceptual design evaluation phase a generic rotordynamic evaluation was performed covering the breadth of the HPRC application demands to establish the feasibility of establishing a configuration that would also demonstrate rotordynamic characteristics and stability in line with the appropriate industry expectations and norms.

The preferred solution of the compound epicyclic was evaluated applying this approach and it was found that even for the relatively large half coupling masses required the analysis showed that suitable bearing-rotor arrangements could satisfy the analytical rotordynamic performance measures required. More detail on the rotordynamic evaluation is provided in the following sections of this paper.



Single Stage Epicyclic Single Stage Parallel Shaft

Compound Epicyclic

Figure 7 - Relative Gear Pitchline and Bearing Velocities

All considered, the epicyclic gear technology looks the most appropriate answer to the HPRC demand, being it able to share the transmitted load through a multiple meshes, and therefore able to reduce the internal forces. Moreover, the epicyclic gears can be configured in three basic gearing arrangements (see Figure 8):

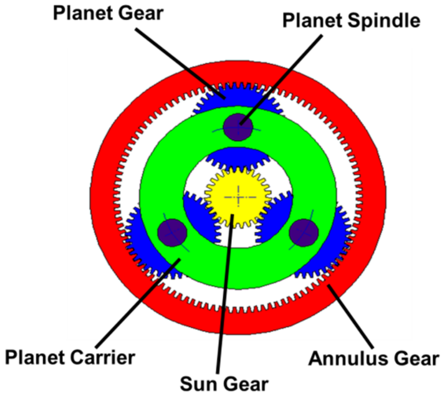


Figure 8 – Epicyclic gear schematic

1. **Planetary**
   - Sun gear and carrier rotate;
   - Annulus gear fixed;
   - Highest ratio range;
2. **Star**
   - Sun and Annulus gears rotate;
   - Planet carrier fixed;
   - High ratio range;
3. **Solar**
   - Annulus gear and carrier rotate;
   - Sun gear fixed;
   - Low ratio range;

TECHNOLOGICAL ASSESSMENT

As part of the development program a prototype HPRC drive train was commissioned for which the power transmission employing the compound epicyclic construction was incorporated. The prototype unit headline specification is given below:

- Required Output Power : 14500 kW
- Nominal Input Speed : 1500 rpm
- Nominal Output Speed : 18500 rpm
- Max Continuous Output Speed : 19425 rpm
- Over speed : 30%
- Gear Rating : API 613

The mechanical design and assessment of the power transmission was conducted applying a combination of manufacturer custom high speed epicyclic design principles with industry specific qualifications. As the predominate application for the HPRC product will be for the oil and gas industry the application to API design principles was a key consideration in the design and qualification of the gearbox. Unlike the equivalent parallel shaft technology, at the time of construction, no specific API specification was in place for the rating and qualification of epicyclic; however it was a core principle throughout the design to embody the key elements of the API rating, construction and testing principles.

A wide range of specific design considerations were required in the development of the transmission not all are presented here but some of the key elements are summarized in this section.

Gearing

The toothing design of the power transmission was sized and rated in accordance with API 613 loading levels and assessment criteria, meeting all typical API criteria attaining the service factor requirement of 1.6 for the intended application. The tooth configuration consists of a high helix angle double helical high speed stage and high load capacity spur low speed stage, with all running teeth being case hardened by nitriding. In addition to the principle API rating all toothing was analyzed in detail for stress distributions and deflection characterization using advanced 3D tooth modelling techniques.

All tooth profiles had applied microgeometry to optimize the gear mesh performance from both a durability and dynamic excitation perspective. A range of operational conditions were considered during the toothing analysis covering the entire HPRC operational envelop, considering also the effects of misalignments due to production and operational variations. In the Figure 9 it is shown the LH and RH helix tooth contact pattern for HPRC prototype gearbox.

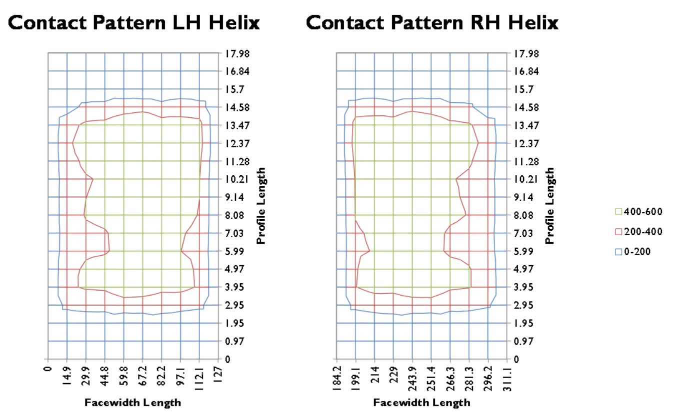


Figure 9 - HPRC Prototype Gearbox

Bearings

A key principle or the epicyclic, and moreover the compound epicyclic, is the removal of the need for load carrying high speed bearings. This eliminates then issues around having high journal velocities coupled with high loads, with the only gear load supporting bearings being located on the lower speed compound shaft rotor. As such , whilst still a critical feature, the bearings are operated well within existing design and operational experience without the need for special treatments or configurations. In the prototype design the loaded compound shaft bearings were of the fixed geometry type incorporating modified profile bore geometry applying the taper dam principle. This configuration enables both good operational performance in terms of temperatures, oil flow and efficiency to be obtained whilst also providing good rotordyanmic characteristics.

For the arrangement developed for HPRC there is also a single high speed bearing provided on the sunwheel shaft line. The bearing which is separated axially form the gear mesh is used to ensure sound rotordyanmic performance can be achieved with the use of flexible membrane type couplings, whereby the overhung coupling mass is supported by the single bearing on the gearbox side. A key feature of this bearing is that no radial forces are imparted by the gearbox, only the small amounts of mass of the high speed shaft and coupling impart any force. This results in a bearing that can be small in diameter and that operates a very low load with the only sizing criteria relating the shaft sizing due to the transmitted torque. This low load (typically <100 kPa bearing pressure) but high speed operation mandates that care is taken to ensure stable rotordyanmic behavior in achieved.

The bearing configurations used to achieve this can be either specially profiled fixed geometry bearings or variable geometry tilting pad type bearings. In the HPRC case a 5 shoe tilting pad type bearing with directed lubrication was utilized with modified geometry to accommodate for the very low load operation whilst maintaining low predicted pad operating temperatures. The final bearing design operated at a nominal bearing load/pressure of 68 kPa with a journal velocity of 87 m/s. Throughout the bearing selection and analysis the rotordyanmic behavior was a critical component driving the design. Whilst a challenging duty the configuration and bearing arrangement employed was one that Allen Gear had employed on many units prior making the analysis and selection process significantly lower risk.

Structural Dynamics

When operating high torque capacity rotors with the associated housings at such high speeds a key factor introduced beyond current applications is the criticality of the system dynamics, and this relates not only to the rotor dynamics considerations but also the support structures and housings. As such it was a critical aspect to appropriately capture the dynamic characteristics of the bearing supports and gear housings. From a general structural perspective the challenge introduced is a function of increased structural mass, due to the large housings involved for the higher power/torque delivered, coupled with the condition that due to the high operating speeds the excitation frequencies (specifically 1st and 2nd order shaft rotations) are notably increased.

These results in a condition where the frequencies of housing key natural modes are reduced and an increase in the excitation frequencies is introduced, therefore closing the typical separation margin inherently realized for current installations.

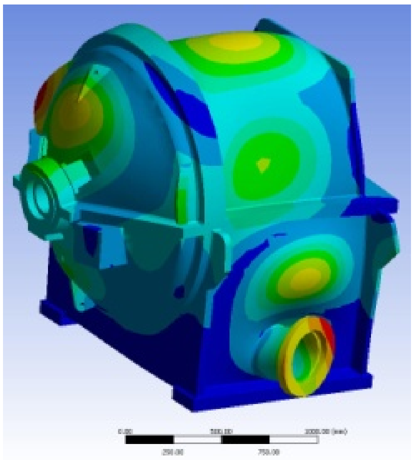


Figure 10 - Analytical Structural Dynamic Analysis Model

This was addressed by detailed structural evaluations employing high fidelity finite element based modelling techniques which were coupled to an iterative design procedure (see Figure 10). The dynamic characterization response work was not limited to analytical activity alone but also practical modal analysis using seismic force-response measurement techniques which were used to validate and develop the modeling methods. Using this analysis process it was possible to develop casing structures displaying dynamic response characteristics that were compatible with the full breath of excitation frequencies generated across the operating range, it was also possible by following this process to achieve this result without an increase in the collective mass of the gearbox structures.

Summarizing the mechanical design, whilst the HPRC application provided a duty which in terms of combined power and speed produced a new reference point as shown in Figure 2, it was possible, by employing the compound epicyclic configuration, to maintain all key performance parameters relating to the gearing, bearings and rotors within existing experiences and references. Providing for a technical solution, that contains lower technical risk than other traditional oil and gas power transmission configurations.

LATERAL ROTOR-DYNAMIC DESIGN ASSESSMENT

With all high speed rotating equipment it is imperative that an accurate and comprehensive rotodynmaic analysis is completed early in the design phase. The lateral analysis allows the designer to predict the shaft critical speeds, associated log decrement values and shaft unbalance response characteristics.

There are two primary types of lateral analysis, a damped stability analysis and unbalance response analysis. The damped stability analysis calculates, for a given load and speed the damped natural frequencies and their associated log decrement values. This allows the generation of a Campbell Diagram and the verification of any critical speeds within the operating range of the gearbox. The unbalance response analysis calculates the elliptical whirl response for a give unbalance excitation which can be utilized to investigate the sensitivity of the shaft in terms of possible unbalance forces.

For epicyclic gearing, manufacturers have developed a specific analysis procedure that breaks the gear transmission down into a number of discrete shaft lines. Dependent on the configuration, the high speed shaft is modelled with a high speed bearing support and an equivalent gear mesh stiffness. The compound and low speed lines are modeled as simply supported shafts in two hydrodynamic bearings. Figure 11 shows the three discrete shaft lines. The fixed geometry hydrodynamic bearings are modelled with 8 stiffness and damping coefficients derived through internal bearing performance software while the high speed tilting pad bearing characteristics are provide by the bearing supplier.

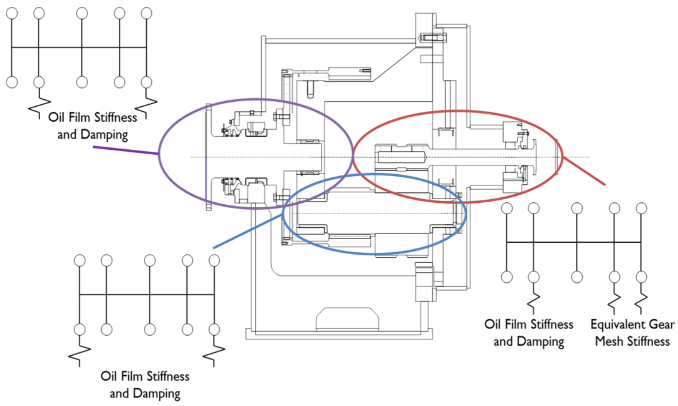


Figure 11 - Epicyclic Gear Rotordynamic Schematic

The analysis algorithm used is based on a transfer matrix solution with unsymmetrical matrices for the bearing supports. The software utilizes a lumped mass representation for the inertia properties of the rotating assembly, with the rotor divided into a number of sections with half the mass of each section represented by an infinitely thin disc positioned at either end of the section. The procedure includes a stability analysis and verification against log decrement stability criteria (log dec >0 preferred, loaded conditions > 0.3, No -ve values permitted).

Due to the specific nature of the HPRC project the primary focus was put on the performance and characterization of the gearbox high speed shaft. The high power density and API requirements imposed challenging shaft vibration limits which highlighted the importance of the rotordynamic analysis procedure in ensuring the gearbox met the required specification and performed satisfactorily. This report therefore focuses on the high speed shaft analysis. Figure 12 shows the gear high speed shaft and the sections modelled.

To encompass all possible operating conditions, the minimum and maximum bearing clearances were considered along with extreme lubrication temperature and pressure conditions. For simplicity only the minimum clearance data has been presented in this report.

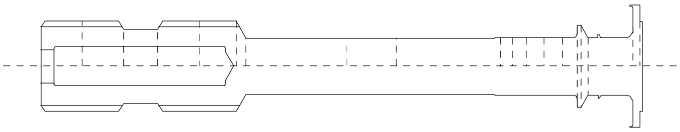


Figure 12 – HPRC High Speed Shaft Gear

The lateral analysis predicted the first 4 shaft natural frequencies. These are shown below in the Campbell Diagram, Figure 13.

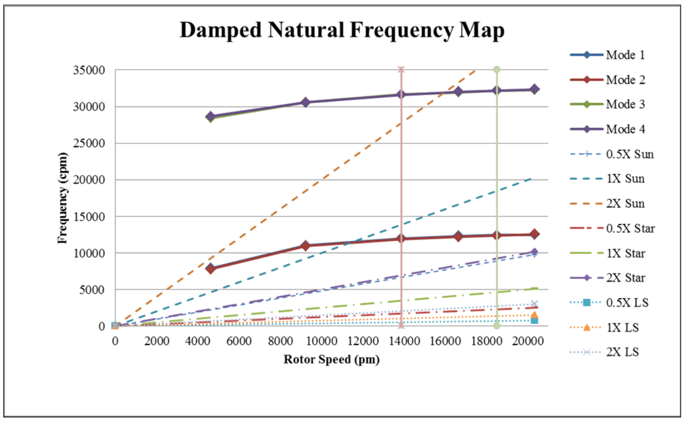


Figure 13 – Campbell Diagram

The first natural frequency was found to be a bearing and shaft overhung mode, see the typical mode shapes in Figure 14.

While this mode does intersect the high speed shaft rotational frequency, this occurs below the minimum operating speed of the gearbox and was found to have high log decrement values thus is theoretically well damped.

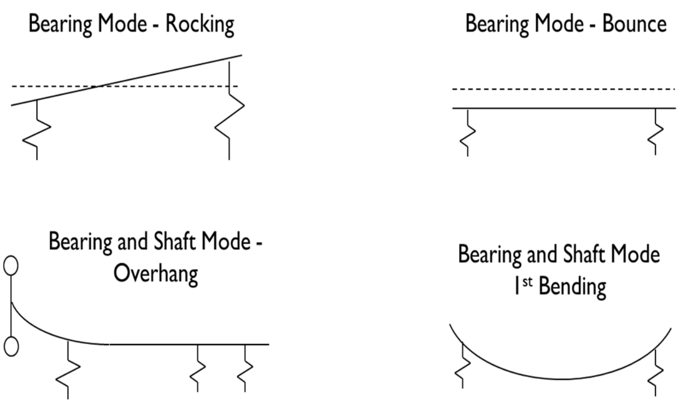


Figure 14 – Principle Rotordynamic Mode Shapes

Intuitively this was the most likely to be the first mode of the shaft when considering the shaft geometry, bearing support locations and overhung mass. The first shaft bending mode was shown to be well above the maximum operating speed of the gearbox, > 30,000rpm. Again this mode was found to be highly damped with log decrement values > 0.5.

A damped unbalance response analysis was completed and the results are shown in Figure 15 below. The response analysis was conducted in-line with the requirements of API 613 that is “the magnitude of unbalance shall be 4 times the value of U as calculated”. Where U is defined as,

$$U = 6350 \times \frac{W}{N}$$

U = input unbalance, g-mm
N = operating speed, rpm
W = journal static load/overhung mass, kg

The analysis showed a pk-pk vibration at full speed of nominally 20 microns with the maximum shaft amplitude occurring at the coupling flange location. The peak response was found to be at approximately 15,000rpm, ~80% full speed. The Half-Power Bandwidth principle was used to determine the amplification factor associated with the response analysis. An amplification factor of ~1.5 was calculated highlighting the low maximum vibration amplitude at resonance.

The lateral rotordynamic analysis did not highlight any primary critical speeds within the operating range of the gearbox. The response amplification factor was shown to be small, indicating a well damped system with minimal vibration amplitude at resonance. The stability analysis also shows a well damped shaft line with log decrement values > 0.5. The results indicate that the rotodynamic performance of the gearbox will be satisfactory and will perform, well across its whole operating range.

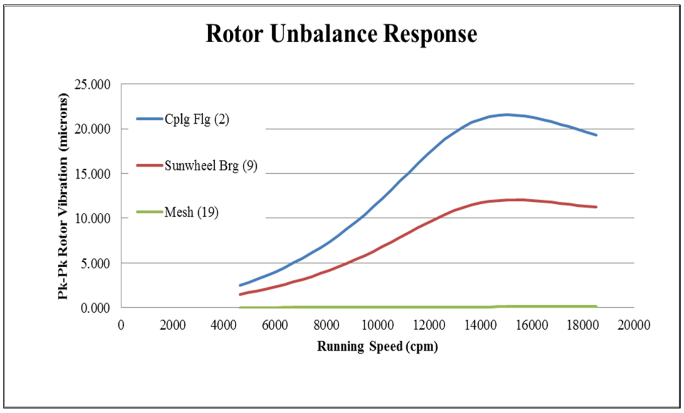


Figure 15 - Damped unbalance response

TORSIONAL ROTOR-DYNAMIC ASSESSMENT

The load pulsations and more in general the electromechanical interactions within the moto-compressor trains or between different turbomachinery trains (especially in island-mode electrical installation), are critical operating conditions for the geared systems, resulting in alternating stresses which can be significant enough in amplitude to affect the fatigue life of the gear components (when run in torsional resonance condition).

Several epicyclic gears have typically floating members that adjust their running positions in response to changing load vectors, providing the risk to jeopardize the oil film stability between the mating teeth (potential failure mode: micropitting). In order to minimize this risk the specific gear architecture is such that each shaft is equipped with appropriate journal bearings, except the High Speed Shaft, which is supported by a single bearing at the flexible coupling end and by the sunwheel gear mesh at the other end, split in three axial-symmetric components.

The gearing arrangement is shown in Figure 16, it consists in:

- Two gear ratio stages to enable high ratio in a compact gear width and diameter;
- Double helical gearing on high speed stage of sunwheel and compound star pinion;
- High Contact Ratio Spur gearing on low speed mesh between compound star pinion and annulus ring gear;
- Fixed carrier housing supports star pinions in journal bearings, offering a wide bearing span with reduced sensitivity to misalignment;
- High Speed shaft supported by a single bearing;
- Low Speed shaft mounted in hydrodynamic taper dam sleeve bearings, it provides the connection to the annulus ring gear through a single gear tooth coupling;

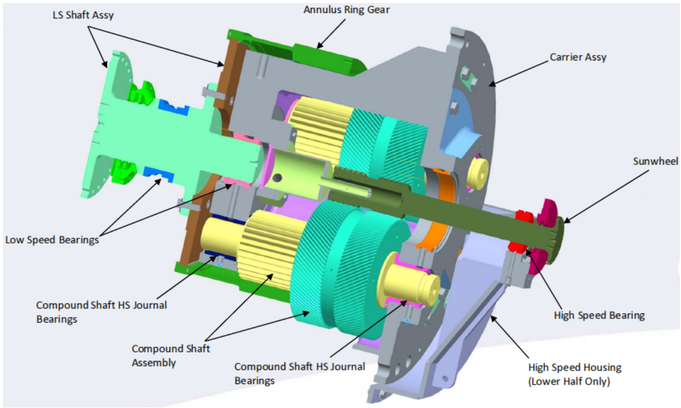


Figure 16 – Gearing Arrangement.

The train torsional analysis was performed in order to assess its safe behaviour from a torsional dynamics viewpoint. The model was including all the shaft line stiffness / inertia contributions as shown in Figure 17.

The following items are worth of special mention:

- Torquemeter
- Epicyclic gearbox modelization

The torquemeter was able to measure either static or dynamic torque so it was used to measure the power absorbed by the compression train and to check the torsional dynamics.

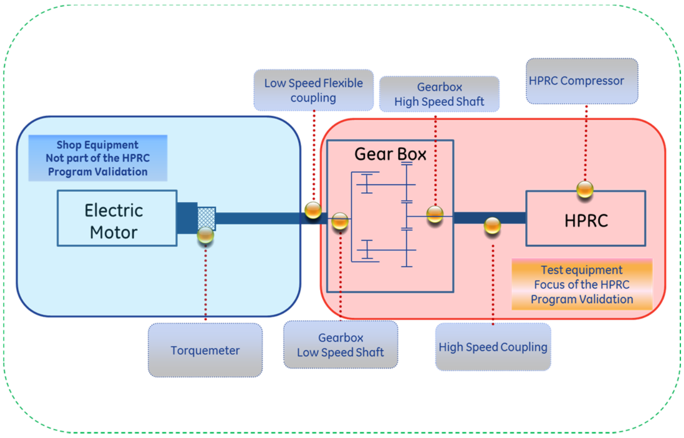


Figure 17 – Shaft-line schematic.

The epicyclic gearbox was modeled as shown in Figure 18 according to the Authors' experience and best practices. Specifically there are 4 lumped stiffness terms coming from the following components:

- High Speed Shaft (K1)
- Compound Shaft (K2, an equivalent shaft made by 3 single shafts in parallel)
- Annulus Gear (K3)
- Low Speed Shaft (K4)

The inertia terms are lumped according to the same criterion.

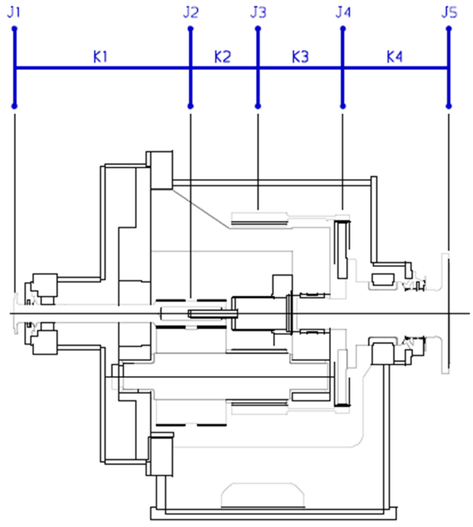


Figure 18 – Gearbox torsional modelization.

The torsional analysis was performed through a set of tools developed by the University of Virginia through a research consortium (Brockett L.) and (Orsey R.S.).

Figure 19 shows the predicted Campbell diagram where torsional natural frequencies are plotted versus rotational speed together with the excitation lines.

The couplings selection was done with the goal to avoid any resonance of 1st or 2nd torsional mode in the operating speed range (±10% margin). If a resonance with a higher order mode is found, a specific stress and fatigue analysis is performed in order to verify that there is no concern for the shaft infinite life.

It is worth to note that the first torsional natural frequency was predicted to be at 16Hz, and that the torquemeter dynamic data allowed measuring the torsional frequency response of the train, therefore it is possible to have an experimental validation of the modal analysis.

Additionally the torsional analysis included a forced response analysis to the following excitations:

- Electric motor short circuit (2-phase, 3-phase)
- Electric motor Air Gap Ripple torque

The first analysis was performed as usual when an electric motor is present in order to check the safety factor in case of electric malfunction. The second analysis was done since the electric motor was here driven by a VSD and this component might inject some torsional excitation in the train due to the specific control pattern.

Both verifications were fully satisfactory according to the Authors' relevant Design Practice.

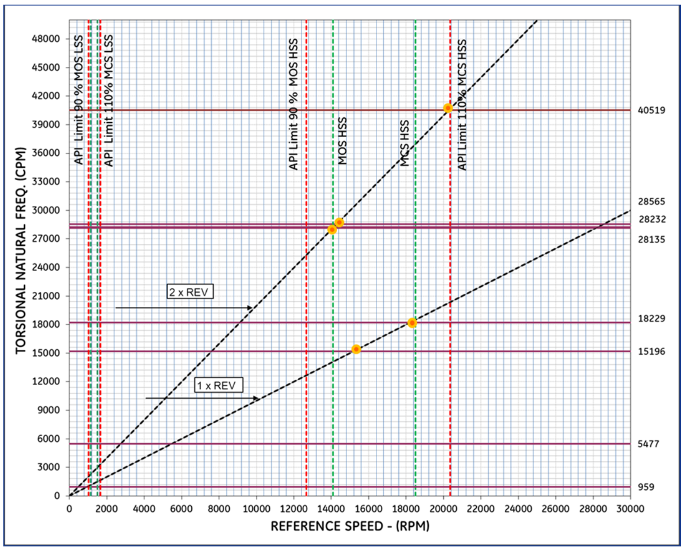


Figure 19 – HPRC train Campbell diagram.

FULL SPEED NO LOAD TEST RESULTS

To validate the gear design a Factory Acceptance Test (FAT) was completed. A full speed, part load spin test was conducted at the gear manufacturer workshop. The test setup is shown in Figure 20.

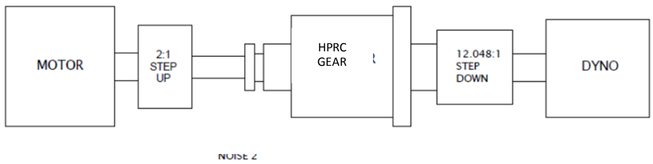


Figure 20 - Factory Test arrangement

The gear was driven by an electric motor through a 2:1 step up gearbox. Load of approximately 650kW was applied to the unit through a step down gearbox and dynamometer. A summary of the gearbox test procedure is outlined below,

- Controlled ramp-up to full speed. (25% increments)
- Application of ~650kW at the dynamometer
- Run continuously for 4 hours
- Run at low inlet pressure and high inlet temperature
- Run at overspeed for a minimum of 15 minutes
- Rundown

The equipment was incorporating a comprehensive instrumentation system to provide condition monitoring parameters and methods for failure detection. The wide-ranging instrumentation setup also allowed the author's to better characterize the prototype gearbox performance. The key instrumentation probes are summarized in Table 1 below.

Component	Instrumentation
High Speed Shaft	Radial Proximity Probes
High Speed Bearing	Journal Temperature Probes
Compound Shaft	Radial and Axial Proximity Probes
Compound Shaft Bearings	Journal and Thrust Temperature Prb.
Low Speed Shaft	Radial Proximity Probes
Low Speed Shaft Bearings	Journal Temperature Probes
Annulus Gear	Radial Proximity Probes
Gearcase	Accelerometers
Gearcase	Temperature Probes
Oil Inlet	Pressure, Flow and Temperature Prb.
Oil Outlet	Temperature Probes

Table 1

The gearbox component temperature trend is shown in Figure 21 and it clearly highlights the thermal stability of the system. The performance of the hydrodynamic journal bearings and high speed shaft tilting pad bearing was shown to be within the operating limits and performing satisfactorily. During oil inlet trip conditions and overspeed the bearings also performed within acceptable limits.

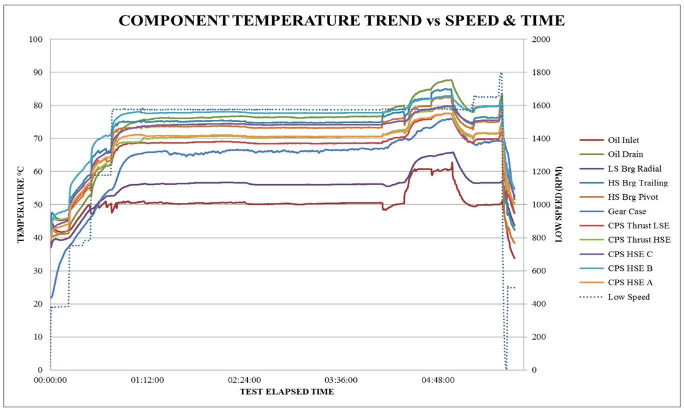


Figure 21 – Component Temperature Trend

Due to the significant volume of data and information collected during the extensive testing this section focuses on the high speed shaft characteristics. Figure 21 shows the shaft proximity probe trend data during the test. The contract high speed shaft probes are located inboard of the high speed shaft bearing. The vibration levels for both X and Y probes were within the 43μm test limit and the variation between the two probes can be associated to the elliptical orbit of the shaft.

During the steady state running several spectra were recorded so that a more detailed investigation into the dominating frequencies could be completed. Figures 23 & 24 show FFT spectra plots respectively of the high speed shaft probes and the casing accelerometers. The high speed probe data clearly demonstrates that the dominating frequency is the high speed shaft synchronous frequency (~324Hz).

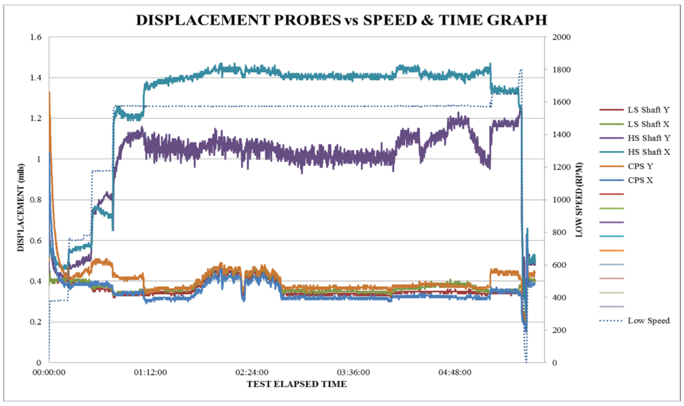


Figure 22 – Component Vibration Trend

The accelerometer frequency domain plot (10Hz-10kHz) also shows high speed 1X synchronous frequency as the dominant component. Although this plot shows a relatively broad bandwidth, encompassing tooth meshing frequencies due to the very light load they are not apparent.

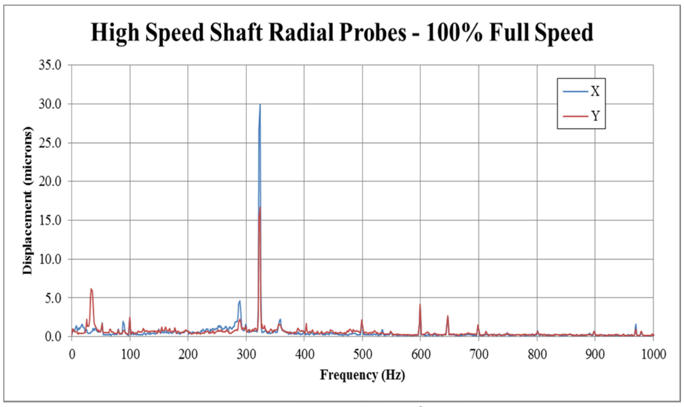


Figure 23 – High Speed Shaft Probes FFT

The gearbox rundown was captured and the resulting bode plot for the high speed shaft probes can be seen in Figure 25.

Figure 25 shows three interesting features. Firstly, the fundamental structural resonance of the gearbox can be seen by the phase shift and increased vibration levels below 5,000rpm. This mode was predicted through a detailed FEA of the gearcase. Secondly there is a gradual phase shift between 10,000rpm and 15,000rpm. This aligns with the increased amplitude of vibration and the traversing of the 1st shaft critical speed, overhung mode. This mode is clearly well damped with a low amplification factor. Thirdly, overall the bode plot shows the predicted increase in shaft vibration as a function of speed.

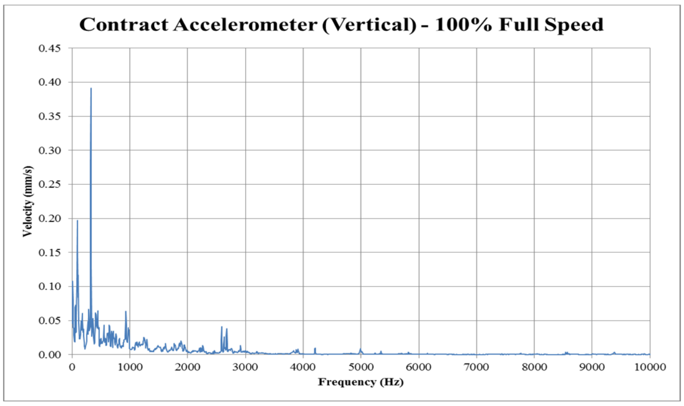


Figure 24 – Contract Accelerometer FFT

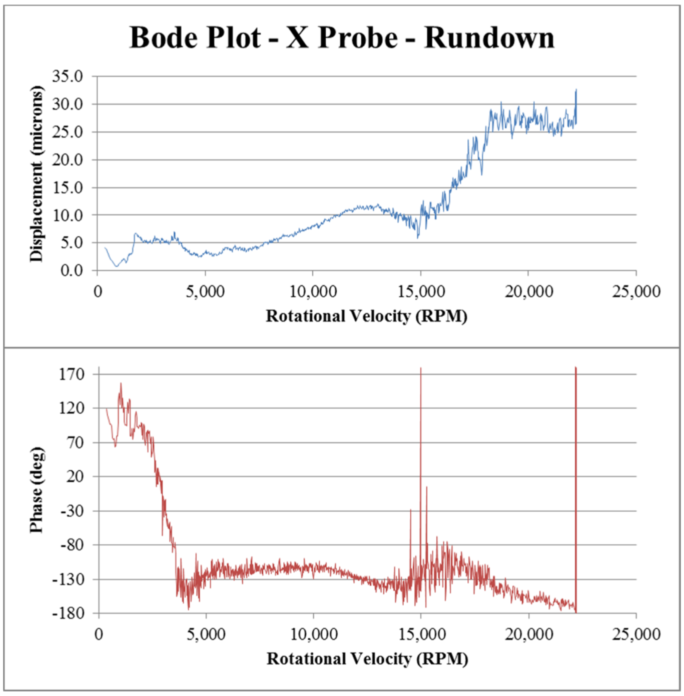


Figure 25 – Rundown Bode Plot

The FAT results proved the performance and overall concept of the gearbox and provided good correlation with the rotordynamic and structural analysis.

FULL SPEED FULL LOAD TEST RESULTS

The mechanical running test, full speed and partial load, at the gear manufacturer's workshop, is an important validation step, by the way, being the gearbox a load dependent piece of the rotating equipment, its dynamic behavior shall be verified at the full speed and full load, where it is possible to replicate the real operating conditions in terms of journal bearings load, the tooth contact pattern and tooth stresses.

Purpose of the string test is therefore to demonstrate the mechanical integrity and efficiency of the integrated shaft-line and the aerodynamic stability of the compressors, in the specified operating conditions, as well as to validate the rotordynamic performances of all the rotating equipment, especially the speed increasing gearbox (test bed in Figure 26).

Figure 26 – String Test Bed

The material presented in this paper is based on the outcomes of a complete unit string test carried out at compressor full Speed, full load, full pressure, and exploring its performance curves. Figure 27 shows the specified test procedure in terms of gear output speed and compressor operating points (i.e. transmitted power).

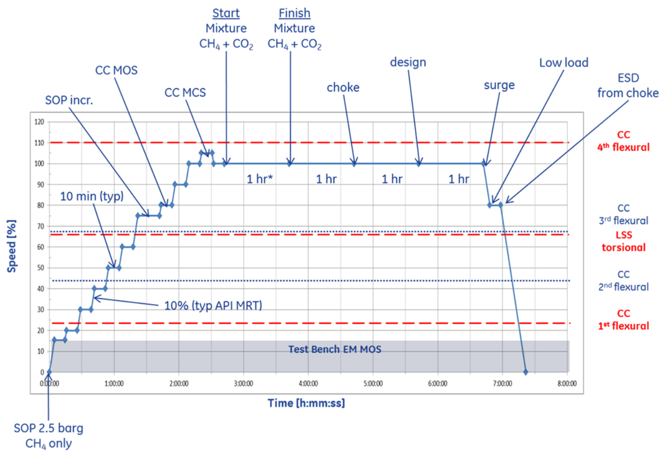


Figure 27 – String Test Procedure

The Figure 28 shows the speed and load trends recorded during the string test, as carried out; at the nominal speed (100%) the transmitted power changes with the compressor process inlet/outlet setting pressure.

In order to limit the number of graphs and simplify the outcomes report, the authors are showing only the "X" radial vibration records of the input shaft (LSS), of one compound shaft (CS), and of the output shaft (HSS).

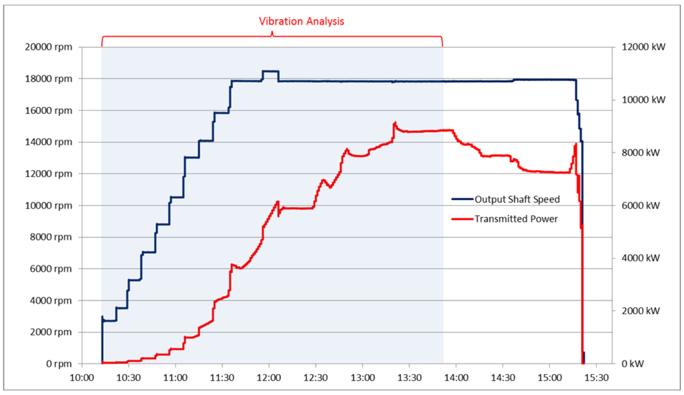


Figure 28 – Measured speed and load trends

In figures 29 and 30 are shown respectively the bearings temperature sensors and the radial vibration probes setup.

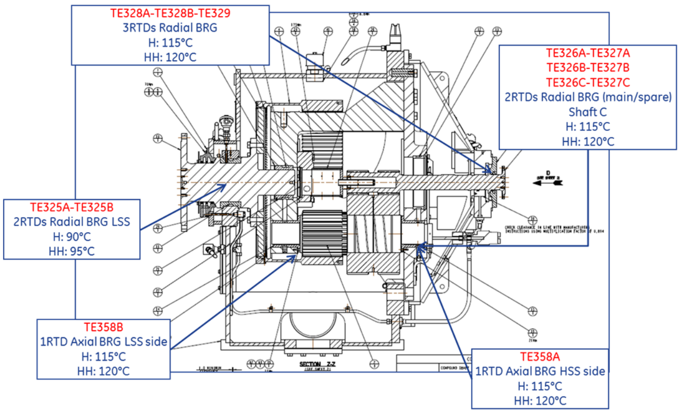


Figure 29 – Bearings temperature RTD's setup

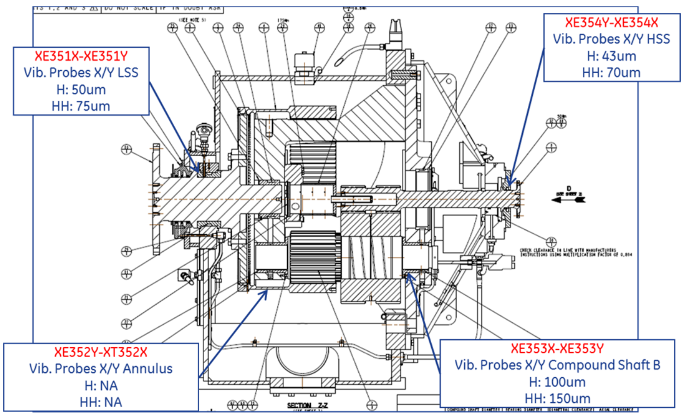


Figure 30 – Radial vibration probes setup

Figure 31 collects the gear journal and trust bearings operating temperature vs speed and load; the bearings operating temperature looks stable at constant speed and marginally affected by the transmitted load, as typically happen in an epicyclic gear, where the meshing forces are split in an axial symmetric arrangement (this does not happen in the parallel offset gearboxes).

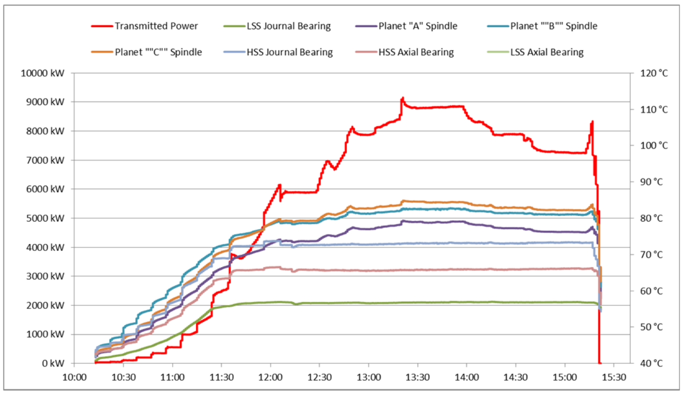


Figure 31 – Trends of bearings temperature vs load

Figures from 32 to 38 show the radial vibration behaviour of the most significant gear shafts (LSS – CS – HSS), in terms of direct vibration trends, and vibration spectra. The gear exhibits stable shaft vibrations amplitude, well within the comfortable zone, with frequency content consistent to the expectations (see the gear characteristic frequencies in Table 2).

From the frequency analysis it is possible to observe on the HSS a significant contribution of the LSS 1X Rev., and a light contribution of the 2X and 4X Rev. which are clearly present on the LSS spectrum.

	MOS (minimum operative speed)	MCS (maximum continuous speed)
Characteristic Frequencies	**Hz**	**Hz**
Meshing frequency (High Speed)	5550	8325
Meshing frequency (Low Speed)	1699	2549
High Speed Shaft 1 X Rev.	206	308
Compound planet shaft 1 X Rev.	57	85
Low Speed Shaft 1 X Rev.	17	25

Table 2 – Gear characteristic frequencies at MOS/MCS

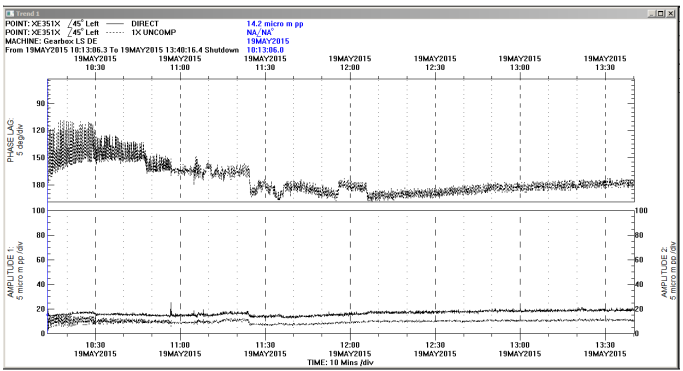


Figure 32 – LSS vibration trend and phase

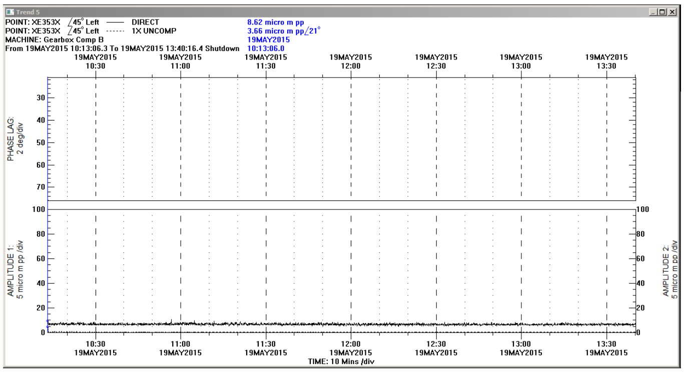


Figure 33 – Planet shaft vibration trend (not key-phasor)

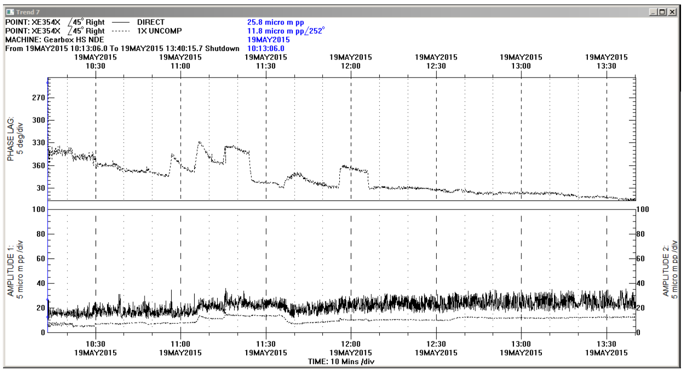


Figure 34 – HSS vibration trend and phase

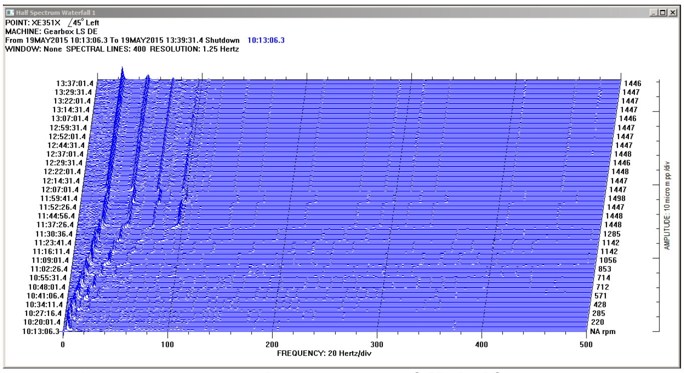


Figure 35 – LSS vibration Waterfall (half spectrum)

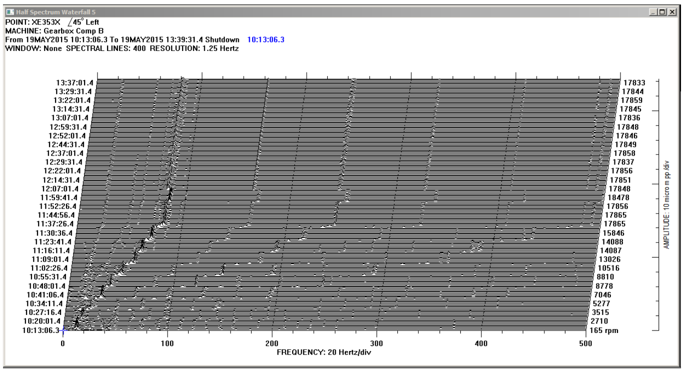


Figure 36 – Planet shaft vibration Waterfall (half spectrum)

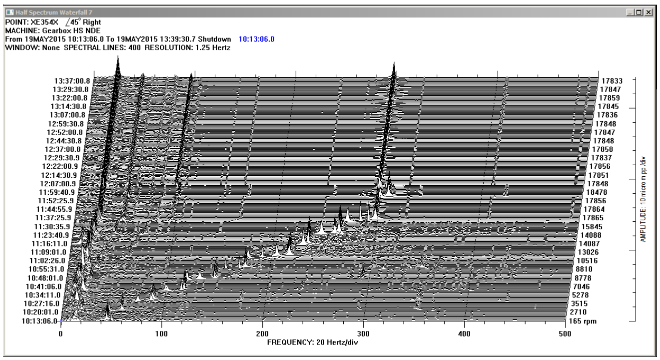


Figure 37 – HSS vibration Waterfall (half spectrum)

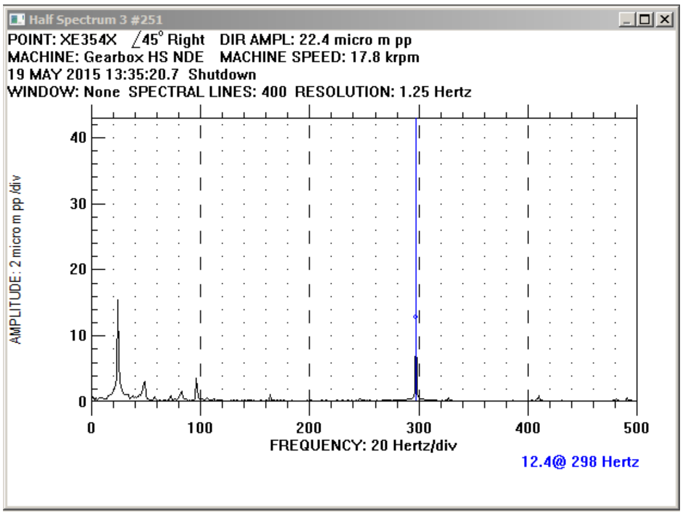


Figure 38 – HSS vibration spectrum @ FSFL

Figures 39, 40 and 41, respectively shows the LSS, CS and HSS vibrations vs speed during the coast-down (Bode plots). Also in this case there is nothing to highlight as abnormal behaviour (no sign of lateral critical speeds, as calculated).

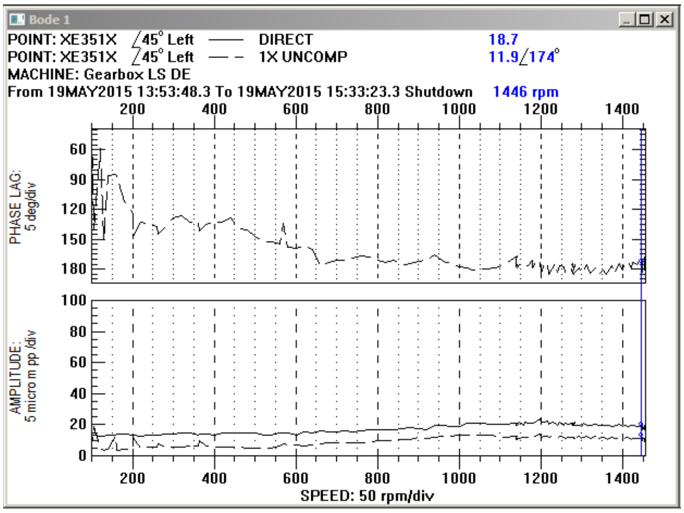


Figure 39 – LSS vibration Bode at shut-down

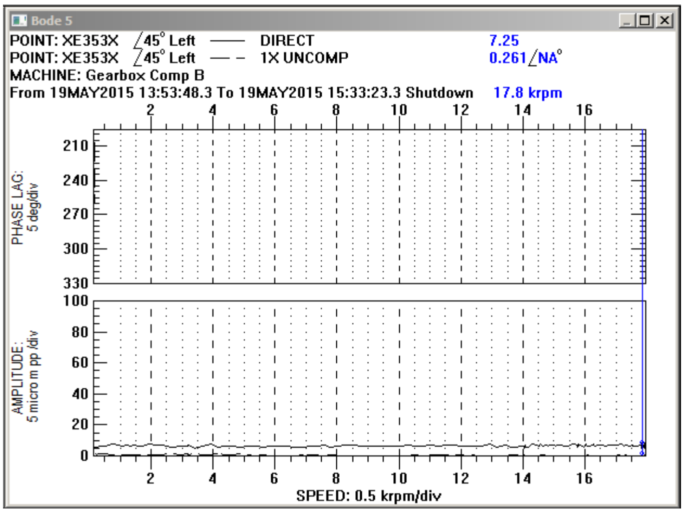


Figure 40 – Planet shaft vibration Bode at shut-down

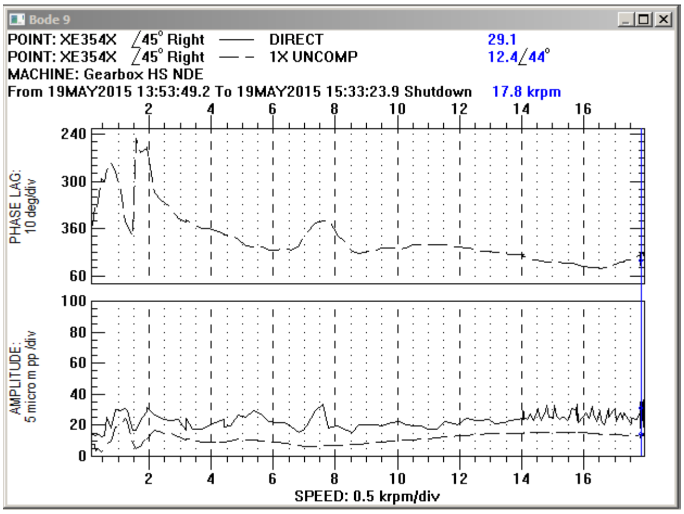


Figure 41 – HSS vibration Bode at shut-down

As anticipated in the torsional analysis section, the train is equipped with a torquemeter on the low speed shaft. The torquemeter is able to monitor the dynamic torques and to get information about the torsional natural frequencies. The figure 42 shows a waterfall plot of the dynamic torque measured during the full load test. Here it is depicted the all-day recording including the start-up, full load test at 100% speed and final shutdown. It is clear that the major resonance region is around 18Hz which is the measured first torsional natural frequency.

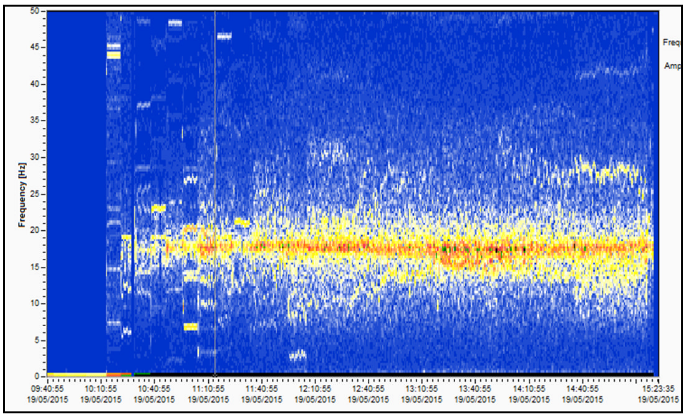


Figure 42 – Torquemeter waterfall plot.

This means that the discrepancy between measurement and predictions is less than 10% which can be considered acceptable considering the complexity of the shaft line. The torque response shown here is about 3% of the coupling rated torque so it is not a matter of concern for the component integrity as well as for the whole shaft line.

The gear casing vibrations were monitored as well as shaft vibrations. It was decided to place many accelerometers for this test in order to fully characterize the prototype dynamic behaviour. Figure 43 shows the whole accelerometers setup.

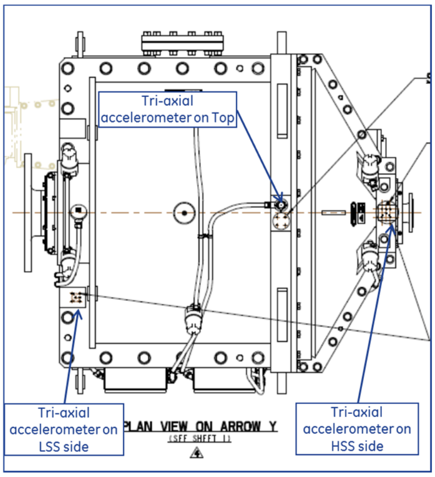


Figure 43 – Casing accelerometers setup.

Figure 44 shows the gear casing vibration trend versus speed and load. As already observed for the shafts rotordynamic behaviour, and bearing temperature, the casing vibration does not show any strong dependence by the transmitted power, being always well within acceptable values.

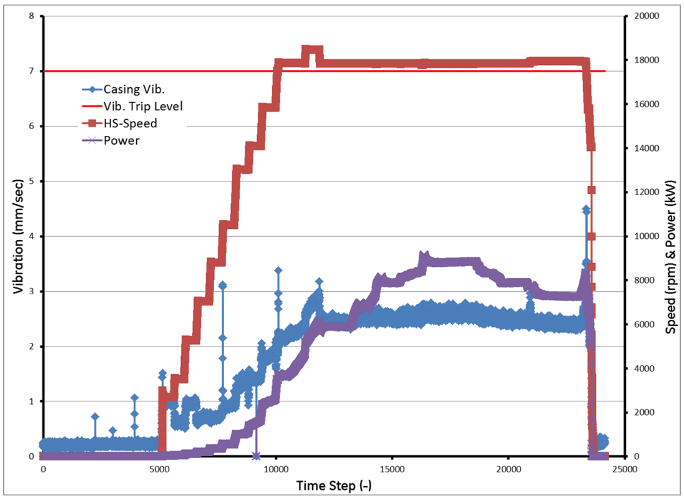


Figure 44 – Trend of casing vibration, speed and power.

Figure 45 shows the steady state spectrum of casing vibration @ 17853rpm, with 10 kHz bandwidth. The major frequency components are:

- 2450Hz → Low Speed Meshing frequency;
- 7350Hz → 3X Low Speed Meshing frequency;
- 8025Hz → High Speed Meshing frequency;
- 9800Hz → 4X Low Speed Meshing frequency;

Also in this case the spectrum does not highlight any unexpected or abnormal vibration signs, providing the typical gearboxes signature.

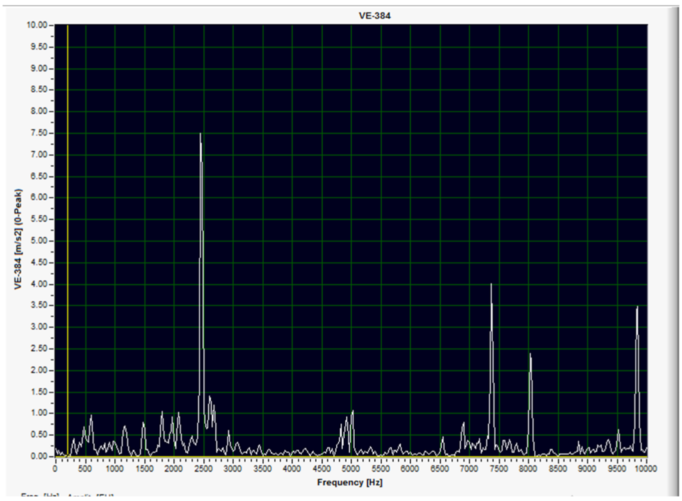


Figure 45 – Casing vibration spectrum.

THEORETICAL VS EXPERIMENTAL OUTCOMES

The experimental outcomes shown on this paper represents a minimum set of data, relative to a single day run, but the test campaign to validate the HPRC product has been extensively carried out for two and half months, accumulating a significant numbers of running hours.

The compression train has been run at different combination of the operating conditions, like speed, load, oil inlet temperature and pressure, and monitored at different transient operating conditions, like start-up at different acceleration rate, hot re-start, emergency shut-down and system trips. Practically the test campaign was aimed to replicate all the possible cases which can be experienced during the normal site operation.

In all these conditions, the epicyclic gear exhibited regular and stable rotordynamic behaviour, in line with the expectations. Despite the complexity of the gearbox architecture (epicyclic compound with τ = 12.3), and the challenging rated power of 14.5 MW, the unit showed a typical gear vibration firm, as well as a consistent thermal behaviour in all its components, confirming the good correlation between the theoretical and experimental outcomes.

Moreover, the data collected during the string test, represent also the baseline for future equipment, being them a solid and useful reference for potential trouble shooting or data interpretation.

CONCLUSIONS

The industrial community has the responsibility to push forward the technology boundaries to fulfill sustainably the world energy needs. This paper is the final act of an exciting journey, started with a conceptual idea and ended with a product which is ready for service. Authors presented the story of a new product development for the Oil and Gas industry, through mandatory steps like the vision that produces ideas, the analysis of available products, the basic and detailed design, the manufacturing and testing, in a test bench configuration, and then in a full train arrangement. Such challenging task necessarily requires a multidisciplinary Team, committed to execute and deliver.

Definitely, a new product development process requires not only a detailed and analytical study of each single component, but it also requires a system level analysis, in order to consider all the potential interactions like those electro-mechanical and structural.

Finally, the system operating behaviour shall be validated, as done for the specific case, by an extensive and representative test campaign which is able to explore the normal and transient operating conditions.

Results of test campaign confirmed the gear behaviour which was expected on the basis of the detailed engineering analyses.

NOMENCLATURE

HPRC:	*High Pressure Ratio Compressor*
PLV:	*Pitch Line Velocity*
R:	*Gear rating*
***τ**:*	*Gear speed ratio*
P:	*Gear rated power*
n:	*Gear input speed*
JV:	*Journal Velocity*
JVP:	*Journal Velocity Pinion*
JVW:	*Journal Velocity Wheel*
MOS:	*Minimum Operative Speed*
MCS:	*Maximum Operative Speed*
LogDec:	*Logarithm Decrement*
U:	*Unbalance mass*
W:	*Journal static load*
N:	*Operating speed*
Ki:	*Torsional stiffness*
Ji:	*Polar moment of inertia*
VSD:	*Variable Speed Drive*
FAT:	*Factory Acceptance Test*
FEA:	*Finite Element Analysis*
LCC:	*Low Speed Shaft*
CS:	*Compound Shaft*
HSS:	*Hgh Speed Shaft*

REFERENCES

API Standard 613, “Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services”, American Petroleum Institute, 5th edition 2003, Washigton D.C.

API677, “General-Purpose Gear Units for Petroleum, Chemical and Gas Industry services”, American Petroleum Institute, 3rd edition 2006, Washigton D.C.

API684, “Rotordynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsionals, and Rotor Balancing”, American Petroleum Institute, 2nd edition 2010, Washigton D.C.

Brockett L., Barrett T., 1988 (Revised June 1994), Torsional Critical Speed and Forced Response Analysis - A Manual for Computer Program TWIST2 (Version 3.0), ROMAC, http://www.virginia.edu/romac/reports.html.

Orsey R.S., Barrett L.E., June l996, Nonlinear Transient Torsional Response Analysis: A Manual for Use with Computer Program TORTRAN3 - Version l.0, ROMAC, http://www.virginia.edu/romac/reports.html.

ACKNOWLEDGEMENTS

The authors thank colleagues of both Test Departments (Allen Gears and Nuovo Pignone), which have allowed with their outstanding and valuable work, to execute and complete a challenging test campaign, turning engineer’s ideas in reality.