Chapter 19
Order Based Modal Analysis Versus Standard Techniques to Extract Modal Parameters of Operational Wind Turbine Gearboxes

S. Manzato, E. Di Lorenzo, A. Medici, F. Vanhollebeke, B. Peeters, and W. Desmet

Abstract Rotating machineries generally operate under very dynamic and complex conditions, during which structural nonlinearities, tonalities or other dynamic related phenomena may arise, affecting the assumptions made in the design phase. Techniques that allow accurately and confidently identifying the dynamic response are then of paramount importance in such complex scenarios. Wind turbine gearboxes are a typical example of such machines, as they operate under strong transient conditions caused by the turbulent and non-stationary wind speed as well as fluctuations in the electrical grid. If models can help predict these interactions, dedicated experiments need to be foreseen to characterize operational response and validate/improve the developed numerical models.

For these reasons, a comprehensive test campaign has been performed at the ZF Wind Power test rig on a multi megawatt gearbox in the framework of a research project. This paper describes the different experiments performed on the gearbox in as many operating conditions as possible (standstill, stationary or transient at different torque levels) as well as the data analysis techniques used to extract a dynamic model from the measured responses. In particular, accelerations acquired in the different conditions are processed using Experimental, Operational or Order Based Modal Analysis. The objective of the paper is to understand how the structural response varies in different operating conditions, and how different methods behave in extracting modal parameters in the analyzed operating conditions.

Keywords Rotating • Gearbox • Operational • Order analysis • Modal analysis

19.1 Introduction

In recent years, increasingly stricter regulations on noise and vibration (N&V) levels are imposing industries to accurately and thoroughly characterize the dynamic response of their machines, being it a washing machine, a car, a wind turbine or an aircraft. As an example, in the wind turbine industry, the installation of new farms quite close to residential areas is making the noise emission levels a matter of public interest and new standards have been developed to regulate and limit as much as possible these emissions. While airborne rotor noise is drastically reduced thanks to new control strategies and tip-speed limitation, the interaction between the drivetrain and the rest of the structure (tower, bedplate and blades) still poses major challenges as it can result in tonal components. In this situation, a root cause analysis requires both the excitation source (drivetrain) and the transmission paths (the rest of the turbine) to be accurately characterized from a dynamic perspective in their operating conditions [1, 2].

Noise and vibration measurements are nowadays mostly performed as quality estimation methods for gear mesh vibrations and or to verify that overall sound power levels comply with the standards. To properly characterize the dynamic behavior of the source and how forces are transmitted to the rest of the structure, however, dedicated and advanced measurements need to be performed. Current practice is to measure vibration levels on the structure, evaluate frequency or order spectra and analyze the so-called Operational Deflection Shape (ODS) at critical frequencies. If this approach generally allows identifying dominant vibration patterns, it fails to characterize whether the high response levels are due to excited resonances, high excitation or a combination of both, limiting the possibility to take corrective actions in the design phase.

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Also, as the accuracy of simulation models is continuously increasing, experimental data needs to be collected to both validate and improve these models and to extend their validity range. In particular, when dealing with rotating machineries, the resulting dynamics may depend on the applied torque and on the rotational shaft speed, thus a full validation can only be performed if also operational conditions are analyzed. Operational Deflection Shapes, providing only overall levels at specific frequencies, are seldom used as correlation metrics.

For these reasons, an extensive measurement campaign was performed on a prototype multi-MW gearbox at ZF Wind Power on their 13.2 MW test rig facility [3]. By using state-of-the-art measurement techniques, data at different operating conditions was collected with the objective to perform modal analysis not only using classical input–output techniques [4], but also more advanced output-only methods which allow estimating modal models in normal operational conditions where input forces cannot be measured [5]. Besides the full characterization of the gearbox, the performed campaign aimed at validating these advanced analysis techniques on a relevant industrial application case as well as providing guidelines on how these measurement can be optimized to be integrated in the current design process.

This paper is organized as follows: first, a theoretical background on the proposed measurement and estimation methods is presented. The soundness of the different methodology in identifying the modal parameters in the different conditions is demonstrated on a simulated example before applying it on the real measured data. Finally, the different methods will be reviewed and possible improvements proposed.

19.2 Theoretical Background

Classical experimental approaches commonly used to solve noise and vibrations problem can in the past fit into two groups:

- Techniques that attempt to characterize the dynamic properties of the system under study (i.e. Modal Analysis)
- Techniques that attempt to characterize the response signals measured while the machine is in operation (i.e. Order Tracking).

Obviously, techniques that attempt to characterize the dynamic properties of a machine are usually preferred as the processed results have true physical meaning with respect to the dynamic properties of the structure. However, the response of the structure in operating conditions might be significantly different than in the closed and controlled laboratory conditions [6]. For these reasons, significant effort have been put in the last two decades in the development of techniques to identify a modal model of the structure in operating conditions. In this paper, a technique combining Order Tracking and Modal Analysis to identify modal models from operating rotating machineries will be discussed and reviewed.

19.2.1 FRF Estimation from Sine Sweep Experiments

Modal identification of large and dynamically complex structures often requires a multipoint excitation to maximize the energy input. Consequently, sine sweep excitation is applied to concentrate more energy on each line of the frequency spectrum. However, as conventional estimation of FRFs generally requires the excitation signals to be uncorrelated, for multi-point sine sweep excitation several runs with altered excitation force patterns have to be performed [7]. In case of two-inputs excitation, two sweeps are required, respectively with the two sources exciting the structure first in phase and then out-of-phase. As the amplitudes and frequencies are the same, the two sweeps, if taken separately, will results in a correlated input matrix. A typical solution to estimate the MIMO FRFs is then to average over the sweeps as:

$$\left[\{\ddot{u}(\omega)\}_S \ \{\ddot{u}(\omega)\}_A\right] = [H(\omega)]\left[\{f(\omega)\}_S \ \{f(\omega)\}_A\right] \tag{19.1}$$

Where $\{\ddot{u}(\omega)\}_S$ and $\{\ddot{u}(\omega)\}_A$ are the dynamic responses due to the excitation runs with a symmetric force vector $\{f(\omega)\}_S$ and with an anti-symmetric force vector $\{f(\omega)\}_A$. By inverting equation (19.1), using the H1 estimator [8] or the on-line estimation feature available in most acquisition softwares, the FRF matrix H can be obtained. In this case, even though the excitation was chosen to excite either symmetric or antisymmetric modes, the FRFs contain contributions of both symmetric and antisymmetric modes. Consequently, the data analysis may be difficult and less clear.

In some cases and under very strong assumptions, so-called symmetric and anti-symmetric FRFs can also be derived. A simplified approach consists in assuming that the applied force and the contribution from the two inputs at all points are the same. Considering then the two-point excitation and assuming $|F_1| = |F_2|$, the following set of FRFs can be calculated:

$$\begin{aligned} \{\ddot{u}(\omega)\}_S &= [H(\omega)_{1S} + H(\omega)_{2S}]\{f(\omega)\}_S = [H(\omega)]_S\{f(\omega)\}_S \\ \{\ddot{u}(\omega)\}_A &= [H(\omega)_{1A} + H(\omega)_{2A}]\{f(\omega)\}_A = [H(\omega)]_A\{f(\omega)\}_A \end{aligned} \tag{19.2}$$

Where $[H(\omega)]_{i,S}$ and $[H(\omega)]_{i,A}$ are the FRFs between the input i and the considered output from the symmetric and the antisymmetric sweep cases. It can also be observed that, by combining the two equations in Eq. 19.2 and assuming that the absolute values of the input forces from the 2 shakers are the same, the MIMO FRF (Eq. 19.1) can be reconstructed as:

$$\begin{aligned} [H(\omega)]_1 &= 0.5\,[H(\omega)_S + H(\omega)_A] \\ [H(\omega)]_2 &= 0.5\,[H(\omega)_S + H(\omega)_A] \end{aligned} \tag{19.3}$$

Once the SIMO (Eq. 19.2) or MIMO (Eqs. 19.1, 19.2, and 19.3) FRFs are derived, standard modal identification algorithms can be applied to derive the modal model [4, 8]. In general, while the MIMO FRFs allow identifying in a single analysis all the modes, from the SIMO case only the symmetric or antisymmetric modes can be identified. The two models can then be of course put together to obtain the complete modal model.

19.2.2 Operational Modal Analysis

Experimental Modal Analysis techniques using standard input–output measurement and identification methods are widely used to estimate a modal model of the structure which is then used on one side to dynamically characterize the response of the structure and verify the agreement with the design assumptions and on the other side to correlate and update numerical models. However, during operation, the system properties may be different from those applied in the lab and the dynamic response may vary significantly. For these reasons, in the last 20 years Operational Modal Analysis methods have been developed to identify the structure in its operating conditions, without having to apply and measure a known excitation but relying on the environmental one. In [5] a reference description of the method, assumptions and processing steps are presented. Briefly, the weighted correlogram approach is used, to compute the power spectrum form the estimated correlation functions as:

$$S_{yy}(\omega) = \sum_{k=-L}^{L} w_k R_k \exp(-j\omega k\Delta t) \tag{19.4}$$

Where L is the total number of time lags used for the estimation of the correlations. As the correlation samples at negative time lags ($k < 0$) contain redundant information, it suffices to consider only the positive ones when computing the spectra. This leads to the so-called half spectra, of which even autospectra have a non-zero phase:

$$S_{yy}^{+}(\omega) = \frac{w_0 R_0}{2} + \sum_{k=1}^{L} w_k R_k \exp(-j\omega k\Delta t) \tag{19.5}$$

The power spectra can then be used for the estimation of the operational modal parameters using the following representation:

$$\left[S_{yy}^{+}(\omega)\right] = \sum_{i=1}^{N} \frac{\{\upsilon_i\}\langle g_i\rangle}{j\omega - \lambda_i} + \frac{\{\upsilon_i^{*}\}\langle g_i^{*}\rangle}{j\omega - \lambda_i^{*}} \tag{19.6}$$

where λ_i are the system poles (from which natural frequencies and damping are derived), $\{\upsilon_i\}$ are the mode shapes and $\langle g_i\rangle$ the operational reference factor, which replace the modal participation factors in case of output only data.

19.2.3 Order Tracking

The most commonly used methods for characterizing the response signal acquired on operating rotating machinery fall under the classification of order tracking. Order tracking is the estimation of the amplitude and phase of a response of a machine in function of its operational speed. Several techniques have been developed in the last decades to perform order tracking and two examples are discussed here.

The first method is a constant angle order tracking. In general, the analysis time interval to adequately characterize and extract orders should be large for low rotational speed and reasonably small for higher speeds. By resampling the acquired signal in the angle domain, a constant interval can be used as the angular increment per revolution is constant. The algorithm relies on a reference tacho signal to track the instantaneous rotational speed and perform the angular transformation. When DFT or FFT is used to process the resampled signals, the spectral analysis lines will represent constant-order components since the transform is based on angular domain rather than time domain. The order resolution can be found similarly to the frequency domain resolution of the normal Fourier transform. Defining o_s as the angular sampling rate and $\Delta\theta$ the angular increment, with of course $o_s = 1/\Delta\theta$, and N the total number of angular samples considered, the order resolution is given by:

$$\Delta o = \frac{1}{N \cdot \Delta\theta} = \frac{o_s}{N} \tag{19.7}$$

Also, similarly to the frequency domain, the angular Nyquist rate (or maximum order) that can be processed is defined as:

$$o_{Nyq} = \frac{o_s}{2} \tag{19.8}$$

An alternative methodology to extract orders is the Time Variant Discrete Fourier Transform (TVDFT), which is a combination of tacho-based resampling and DFT algorithms, with the advantage of significantly reducing the required computational time. The method is based upon a DFT with a kernel whose frequency varies as a function of time defined by the rpm of the machine. This kernel is a cosine or sine function of unity amplitude with an instantaneous frequency matching the one of the tracked order at each instant in time. It is generally formulated as:

$$a_n = \frac{1}{N}\sum_{n=1}^{N} x\left(n\Delta t\right)\cos\left(2\pi\int_0^{n\Delta t}\left(o_n \bullet \Delta t \bullet rpm/60\right)dt\right) \tag{19.9}$$

$$b_n = \frac{1}{N}\sum_{n=1}^{N} x\left(n\Delta t\right)\sin\left(2\pi\int_0^{n\Delta t}\left(o_n \bullet \Delta t \bullet rpm/60\right)dt\right) \tag{19.10}$$

where o_n is the analyzed order, a_n and b_n are the Fourier coefficients of the cosine and sine terms for o_n, $x(n\Delta t)$are the data samples, N is the number of data samples within an analysis block. The main difference with the previous method is that the order resolution varies as the rotational speed increases. However, the method is computationally more efficient and, in general, is able to more accurately track the phase of the order [9].

19.2.4 Order-Based Modal Analysis

The relationship between the structure modal model and an order can be investigated assuming the structure is excited by an unbalance-like force rotating at increasing frequency [10]. This can be modeled by two perpendicular forces of equal amplitude and in quadrature. The measured response $Y(\omega)$ in the frequency domain is:

$$Y\left(\omega\right) = H_{(:,fx)}\left(\omega\right)F_x\left(\omega\right) + H_{(:,fy)}\left(\omega\right)F_y\left(\omega\right) \tag{19.11}$$

where F represents the force and H the corresponding column of the transfer function matrix. Taking into account the relation between the two correlated forces, the response of the structure can be written as (considering only the positive frequency axis):

$$Y\left(\omega\right) \propto \omega_0^2\left(H_{(:,fx)}\left(\omega\right) - jH_{(:,fy)}\left(\omega\right)\right)\delta\left(\omega-\omega_0\right) \tag{19.12}$$

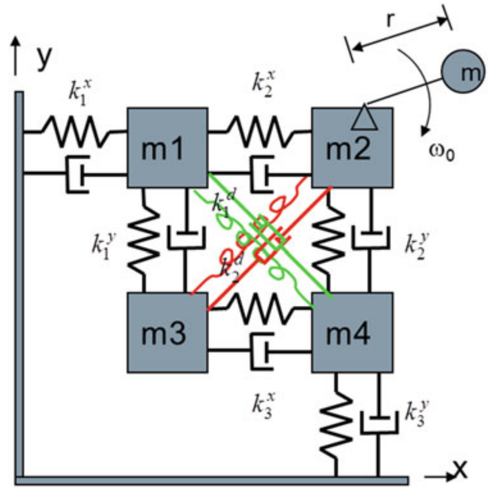


Fig. 19.1 Schematic representation of the anlytical model with rotating excitation

where ω_0 is the rotation speed. From this equation, it is clear that the measured output is proportional to the squared rotation speed and to a complex combination of two structural FRFs related to x and y excitation. A structural FRF can be decomposed in a modal sense as:

$$H_{(:,\bullet)}(\omega) = V(j\omega I - \Lambda)^{-1} L_\bullet + \frac{1}{\omega^2} LR_\bullet + UR_\bullet \tag{19.13}$$

where V, Λ, L are the complex-valued modal parameters (respectively, the mode shape matrix, the diagonal matrix containing the complex poles and the modal participation matrix). $LR_\bullet$ and $UR_\bullet$ are the real-valued lower and upper residuals, modeling the influences of the modes outside the considered frequency band. Finally, combining Eqs. 19.12 and 19.13 we obtain:

$$Y(\omega) \propto \omega^2 \Big(V(j\omega I - \Lambda)^{-1} \left(L_x - jL_y\right) + \frac{1}{\omega^2}\left(LR_x - jLR_y\right) + \left(UR_x - jUR_y\right) \tag{19.14}$$

Equation 19.14 shows that modal analysis can be applied to displacement or acceleration orders taking into account the following statements:

- Displacement orders are proportional to the squared rotating speed and, therefore, acceleration orders are proportional to the 4th power of the rotating speed. In classical modal analysis, acceleration FRFs are proportional to the squared frequency axis.
- Complex upper and lower residuals, while in classical modal analysis they are real.
- Complex participation factor; also in classical modal analysis they can be complex.

It can be concluded that methods like PolyMAX [4] that estimates a right matrix-fraction model are robust against these statements, but methods working on the pole-residue formulation like the LSFD (Least Squares Frequency Domain) method need a new formulation in order to be applied.

19.3 Numerical Validation

To validate the proposed methodologies, an analytical example was built to demonstrate the ability of the different methods to extract the modal model of the structure during run up excitation. The model is shown in Fig. 19.1: it was an 8dof planar system with 4 masses. Forces were applied on mass 2. To simulate the effect of a rotating force, a rotating mass equivalent force was defined. The analytical matrices are converted to a discrete state space model and the transient acceleration responses to the rotating force were calculated. The unbalance-like rotating forces were defined as:

$$\begin{aligned} f_x &= r \cdot m \cdot \omega_0^2 \cos(\omega_0 t + \varphi) \\ f_y &= r \cdot m \cdot \omega_0^2 \sin(\omega_0 t + \varphi) \end{aligned} \tag{19.18}$$

Table 19.1 True modal parameters of the system in Fig. 19.1

Frequency [Hz]	Damping [%]
6.690	1.135
14.785	1.454
18.447	3.387
23.765	3.125
30.883	3.883
34.622	4.040
36.845	3.404
40.956	5.780

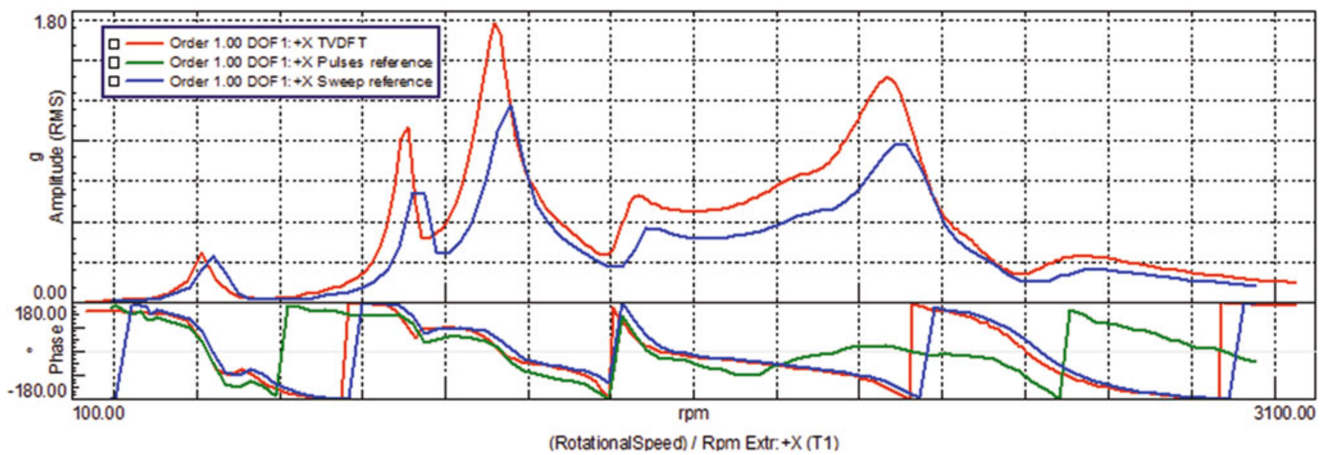


Fig. 19.2 Comparison of different order extraction methods

The response of the structure was simulated for a sweep over 120 s from 0 to 50 Hz. First, PolyMAX was applied on the FRFs calculated using as references the two forces in Eq. 19.18 and the frequency and damping values are listed in Table 19.1. These results are considered the reference for all other methods, as they are exactly the same as those obtained using the system matrices of the numerical model.

As the model was generated to replicate operational measurements, the knowledge of the input forces was then neglected and modal models are derived using the simulated acceleration signals only. Firstly, the data was processed into the power spectral matrix and Operational Modal Analysis was applied. The selected reference signals are the x and y accelerations on mass 2 in Fig. 19.1. In the case of a sine sweep (or a multi-sine sweep if different orders are excited), the excitation can be considered as a broad-band white noise in the frequency range of the sweep. However, as discussed in [10], the so-called "end of order" effect might arise: if any of the orders ends in the frequency range considered for the identification, a spurious peak will show in the spectra and it will be treated by the identification algorithm as a pole. In this case, however, as the excitation order ends at 50 Hz and all modes are below this value, the end-of-order pole falls out of the band of interest and does not pose any problem.

To apply Order Based Modal Analysis (OBMA), orders need to be extracted from the data. As all methods discussed in Sect. 19.2.3 requires the rotational speed as an input, an algorithm was used to extract it from a Time-Frequency diagram. For the constant angle order tracking method, also the tacho moments are required. As the so-called raw-tacho containing this information is generated by the software LMS Test.Lab during the acquisition, even though the data are simulated it needs to be estimated as well using the functions available in the software. For the TVDFT method, on the other end, only the rotational speed is needed. Compared to other applications of order tracking, OBMA requires that the phase is calculated with respect to a reference signal which is synchronous to the excitation (Eq. 19.12). In this case, both a sine sweep with frequency equal to the instantaneous rotational speed and the pulse train signal generated by a tacho sensor were used. An example of the extracted orders is given in Fig. 19.2. The amplitudes estimated with the two methods are quite different, mostly because of the size of the window used to extract the orders. However, it is the frequency shift which is more critical. The error is introduced after the rpm estimation, when additional steps are to be performed to derive also the tacho moments. Because of this, some kind of delay is introduced on the estimated rpm signal and not on the acceleration data, so that the result is a shifted frequency spectrum of approximately 0.6 Hz. TVDFT, relying only on the rpm estimation, is not affected; also, if the rotational speed was measured and the constant angle order tracking performed, this error would not appear.

Table 19.2 Overview of modal parameter identification results with the different techniques

OMA sweep			OBMA TVDFT			OBMA Angle pulse ref			OBMA angle sweep ref		
Freq.	Damp.	MAC	Freq.	Damp.	MAC	Freq.	Damp.	MAC	Freq.	Damp.	MAC
6.69	1.13	99.9	6.89	2.35	0.98	7.21	1.27	0.92	7.11	3.15	0.99
14.78	1.45	1.00	14.88	2.56	0.99	15.4	2.01	0.99	15.45	1.62	0.99
18.44	3.39	0.99	18.54	3.37	0.99	19.07	3.05	0.99	19.05	3.31	0.99
23.75	3.12	0.99	23.85	3.12	0.99	24.28	2.73	0.99	24.37	3.06	0.99
30.86	3.88	0.77	30.95	3.93	0.87	31.48	1.43	0.12	31.47	3.94	0.51
34.59	4.05	0.99	34.67	4.04	0.99	35.08	3.51	0.82	35.21	4.07	0.92
36.82	3.41	0.96	36.92	3.34	0.97	37.36	2.22	0.71	34.4	3.27	0.82
40.89	5.78	0.97	40.93	5.75	0.98	41.73	3.64	0.87	41.46	5.64	0.82

The results obtained for the different method are summarized in Table 19.2, where the MAC values are computed with respect to the true modes calculated assuming the input forces are known (Table 19.1). OMA and OBMA applied on TVDFT orders gives results which are very consistent with the reference ones. Constant angle based orders suffer from the frequency shifts related to the preprocessing steps applied to estimate the tacho moments. The effect of the reference signal used is however reflected in the results, with the sweep giving much better results than the pulse train signal. These results were then used used as a starting point for the analysis of the data acquired on the wind turbine gearbox that are discussed in the following section.

19.4 Operational Gearbox Analysis

With the main objectives of characterizing the gearbox dynamic response in different operating conditions and obtain experimental data for model validation and updating, a measurement campaign took place on the 13.2 MW dynamic test rig at ZF Wind Power in Lommel, Belgium (Fig. 19.3). Using this test rig, gearboxes can be tested under representative loading condition using parameterized load cases that can be programmed into the test rig controller. In this case, the following scenarios were tested:

- Shaker sine sweep during standstill and in stationary conditions at 1,200 rpms;
- Stationary conditions at 1,200 and 800 rpms;
- Run up from 200 to 1,500 rpms.

The operational conditions (stationary speed and run ups) were all performed under different torque loading (33 %, 66 % and 100 % of nominal torque).

As can be observed in Fig. 19.3 (right), an extensive grid of 250 points was measured using triaxial accelerometers. To measure all points, the whole test schedule was repeated seven times, roving the majority of the accelerometers while leaving a small group at fixed locations to allow data merging for a more efficient processing.

19.4.1 Gearbox Signature Analysis

Before analyzing the data and try to extract modal models by processing them into FRFs, Power Spectra or Orders, the typical signature of the gearbox and, specifically, of the gearbox installed on the test rig will be analyzed. This will help better understanding the results that will be obtained in the next section. Examples of Time-Frequency plots during a speed run up and stationary operation at rated speed are shown in Fig. 19.4.

For a gearbox in operating conditions, the response is typically dominated by frequency components that are multiples of the fundamental rotational speed and the main source of internal excitation. By knowing the gear ratios between the different stages and the number of teeth meshing at each stage, all these components can be characterized. Also, on top of these frequencies, components coming from pumps, cooling systems and other auxiliaries are also appearing in the response. These characteristics are visible in both diagrams in Fig. 19.4. During stationary operations, the rotational speed is constant as well as all its multiples, so that all orders appear as peaks at discrete frequencies. During run ups, as the rotational speed is increased, also all orders vary and they appear as diagonal lines.

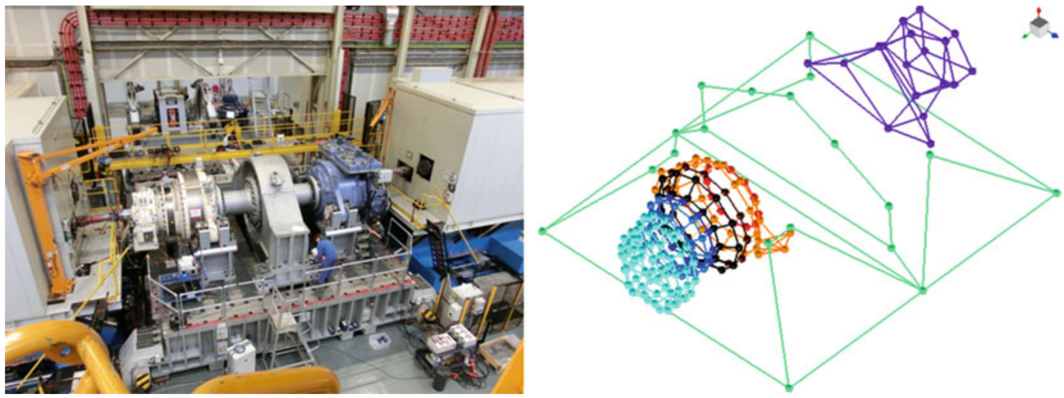


Fig. 19.3 ZF Wind Power dynamic test-rig with the two gearboxes in back to back configuration (*left*) and experimental measurement points (*right*)

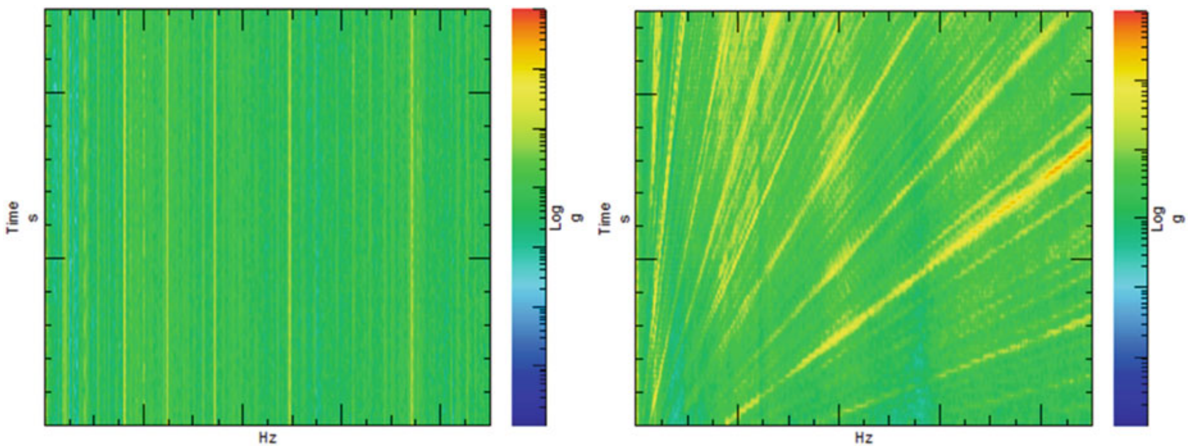


Fig. 19.4 Signature of the gearbox on the test rig in different operating condition. *Left*: stationary speed. *Right*: Run up

What complicates the analysis of the data for this particular case is that on the test rig two gearboxes are tested in a back to back configuration. As in this case the gearboxes were still prototypes, the two tested one had slightly different gear ratios, thus their rotational speeds were different. As vibrations transfer through the test rig and the main shaft from one gearbox to the other, all orders appear in couples at slightly different frequencies. This, as will be discussed, represents an additional challenge when processing the data to derive a modal model.

19.4.2 Shaker Data Analysis

First of all, the data acquired while exciting the gearbox with two inertial shakers are analyzed. The two used shakers were not identical and no force cell was placed in between the shakers and the structure to measure the applied forces. However, by knowing the mass of the shakers and measuring their acceleration, an estimate of the forces was calculated. The two drive signals were defined to obtain two frequency sweeps with the same force magnitude but with the shakers first in-phase and then out-of-phase, as described in Sect. 19.2.1. However, as no online estimation was performed, acquired signals had to be converted to FRFs a posteriori.

In Fig. 19.5 an example of the estimated forces during one of the measured sweeps is shown both in time and in frequency domain. Although the drives were calculated to obtain the same force amplitude on both exciters, measured accelerations (converted to forces) show some small differences. Also, peaks in the time and frequency plots may indicate that the measured acceleration is not only coming from the shaker but also from an interaction with the structure.

The measured accelerations and estimated forces were then processed into system FRFs. From Eq. 19.2 the symmetric and antisymmetric FRFs were derived for the SIMO case, while both Eqs. 19.1 and 19.3 were used for the MIMO FRFs,

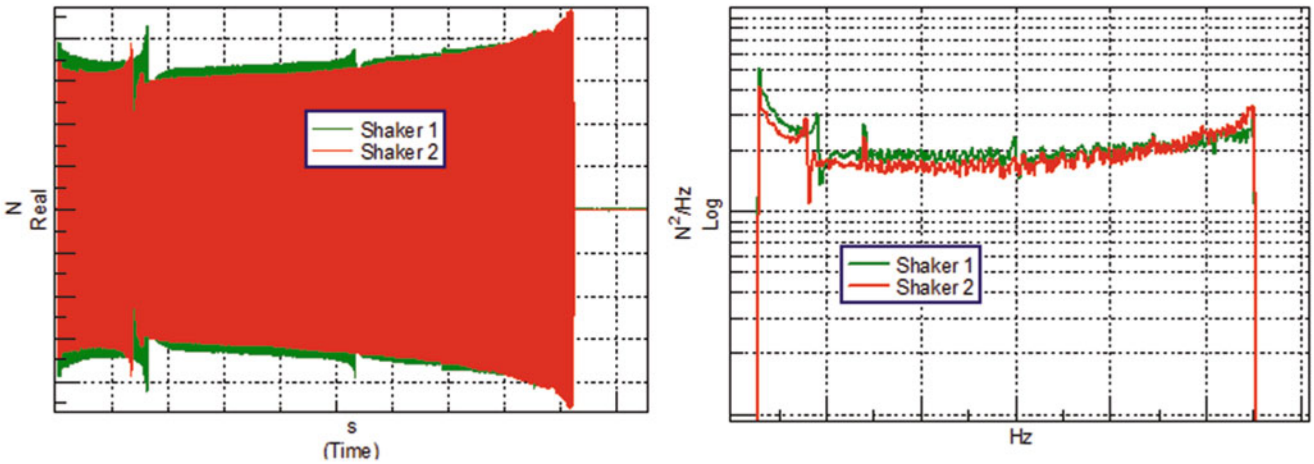


Fig. 19.5 Estimated force signals for the two shakers during a sine sweep in time and frequency domain

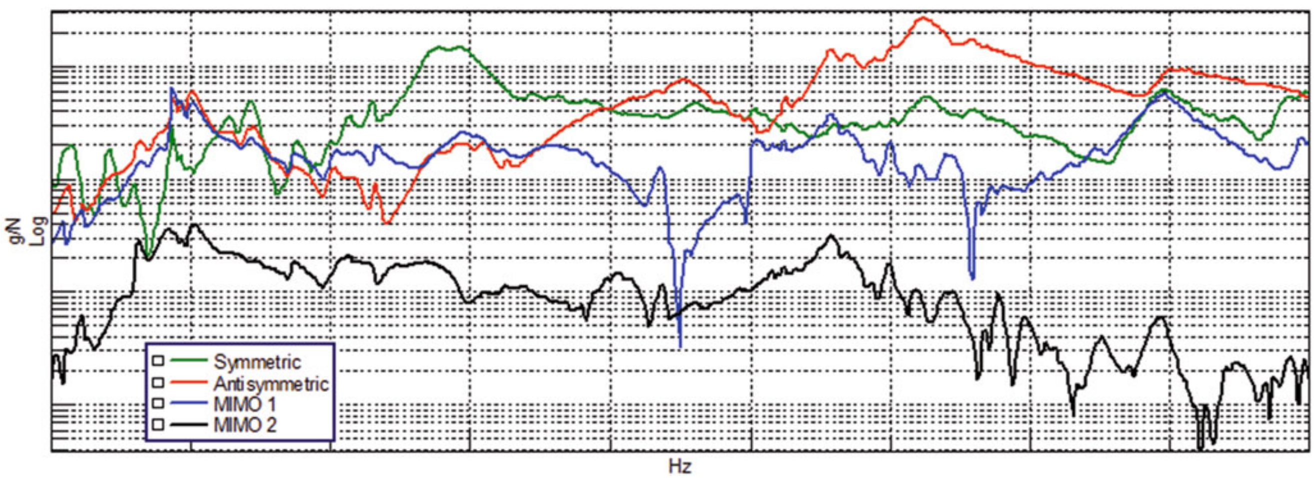


Fig. 19.6 Computed symmetric, antisymmetric and MIMO FRFs for the same point

giving, as expected, the same results. An example of the different FRFs for the same points is shown in Fig. 19.6. The low frequency dynamics is comparable for all FRFs, while less correspondence was generally found at higher frequencies, where however modes are not as well excited because of the low power of the two shakers. Also, the significant difference in level between the two MIMO FRFs is related to location of the analyzed sensor, much closer to shaker 1 than 2.

As they are supposed to be the "true" FRFs, the MIMO data was processed first. All FRFs were processed together but no clear stable poles were identified and the corresponding mode shapes were not clear. As a consequence, the identified modal model couldn't be considered reliable. On the other end, processing the symmetric and antisymmetric FRFs resulted in much clearer and stable poles. After analyzing the results, it was conclude that:

- in the MIMO case, the inability of the identification algorithm of identifying a reliable modal model can be related to a combination of small errors (non-equal forces, different forces in the different runs, see Fig. 19.5);
- in the SIMO case, all these errors are influencing less the results as the two analysis are performed independently;
- SIMO FRFs are derived under very strong approximations that need to be considered when interpreting the results.

Next to the standard FRF measurements at stand-still, a sine sweep was also applied at the gearbox when operating at constant rated speed. This method, generally called OMAX (Operational Modal Analysis with eXogenous excitation) [11] combines the ambient excitation with a known one, allowing to estimate FRFs and the modal parameters in operating conditions. An example of the signal measured in these testing conditions is shown in Fig. 19.7, while an example of the estimated FRFs is shown in Fig. 19.8. In the time-frequency representation, the sine sweep is barely visible on top of the background excitation. Despite the presence of the shakers, the response is still dominated by the rotation frequency and its

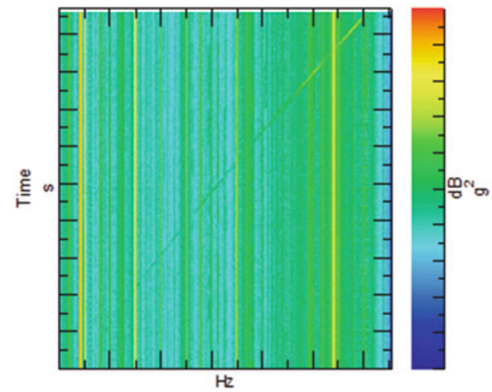


Fig. 19.7 OMAX test data with the applied sine sweep is barely visible on top of the operational excitation

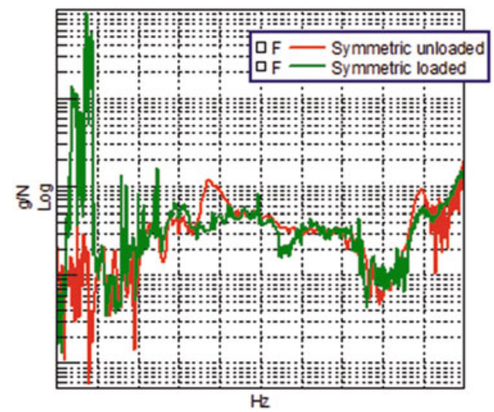


Fig. 19.8 Comparison of loaded and unloaded FRFs in the same conditions

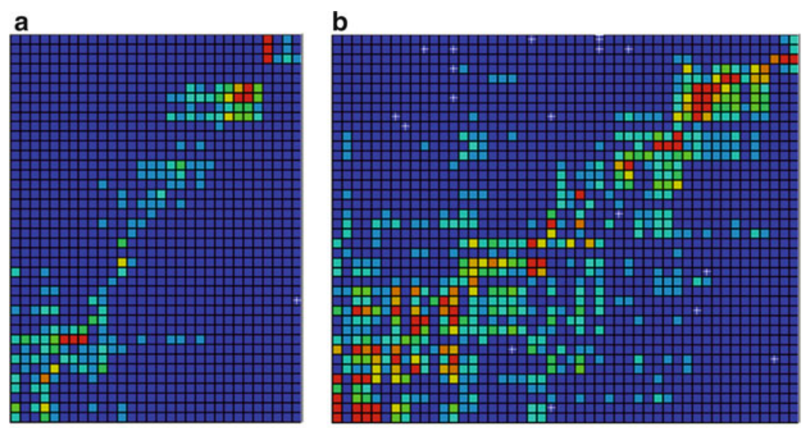


Fig. 19.9 Modal assurance criterion between modal model from different sets of FRFs

multiples, as discussed in Sect. 19.4.1. This reflects In a very poor coherence, in particular at lower frequencies, and in the persistence of the rotor harmonics also in the estimated FRFs.

After analyzing the calculated FRFs, the mode shapes obtained from the symmetric and antisymmetric FRFs at standstill and the symmetric OMAX FRFs are compared in Fig. 19.9 using the MAC. In both cases, modes are quite different as:

- Symmetric and antisymmetric sweeps excite, as expected, different modes, in particular if the structure has some geometrical symmetry;
- standstill conditions are calculated without applied torque, so the gear contact configuration is not repeatable. This problem becomes significant when comparing standstill and operational conditions;
- similarly to OMA, OMAX assumes the background excitation to be stochastic white noise in the frequency band of interest. However, as already discussed, the gearbox signature is dominated by harmonic components that also appear in the estimated FRFs, making the analysis much more complex.

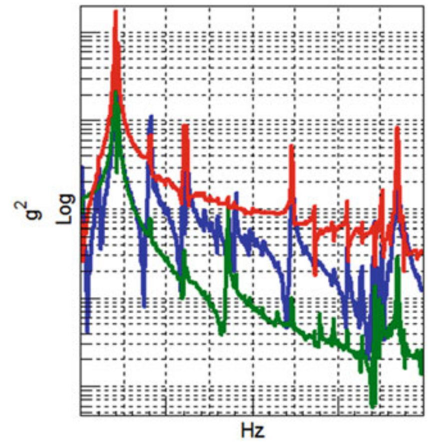


Fig. 19.10 Calculated cross powers during stationary operating conditions

- The MAC diagrams in Fig. 19.9 are shown for a maximum value of correlation of 0.4. Although this is typically considered a very small correlation level, all modes were also visually compared to verify the similarity also from a qualitative perspective. In particular, it was observed that also modes with a 0.2 MAC value where actually showing the same global deformation. This is probably due to the high number of points, the complexity of the test and the measurement noise which is propagated to the results.

19.4.3 Operational Data Analysis

Data in operational conditions were acquired at constant rotational speed as well as during run ups at different torque levels. In this case only data at 100 % torque is discussed. Fig. 19.10 shows a set of cross powers calculated from measured accelerations at constant rated rotational speed. All peaks in the spectrum correspond to harmonic components of the shafts rotational speeds and the gear mesh frequencies at the different stages. Because of the high density of discrete frequencies (and they all appear in pairs as the two gearboxes have different gear ratios) recognizing and thus identifying modes is almost impossible. Furthermore, due to the high number of these rotor harmonics and gear mesh frequencies and their dominance in the response spectrum, application of harmonic filters will not help in identifying more modes.

However, as demonstrated in Sect. 19.3, OMA can also be used on run up data, as soon as the sweeping orders excite the frequency band of interest. In this case, however, the response is dominated by orders ending in the frequency range of interest. This is a problem when applying OMA as the identification algorithm will identify peaks at these frequencies which are however not physical. In a typical gearbox, integer orders are related to the shafts rotation, while gear mesh frequency orders usually appear as non-integer multiples of the fundamental one. In this case, as already mentioned, the two gearboxes have a slightly different gear ratio and all orders are doubled. The number of end-of-orders is thus very high and, similarly to the harmonic component in stationary conditions, will completely dominate the measured response. In Fig. 19.11, an example of the computed cross powers is shown, with the vertical lines indicating some of the end-of-orders frequency lines for one of the two gearboxes. These deterministic components are seen as poles by the identification algorithms, but they actually represent, most of the time, only a forced response at that frequency. However, a modal model could still be obtained and was compared with the one derived from the Symmetric FRF during operation.

The MAC between the two models is shown in Fig. 19.12, where some correlation and similarity between the two modal set is observed. However, it still remains an issue whether correlating mode pairs are true modes or forced vibrations at rotor harmonics / end of order frequencies. Because of the complexity of the system, and the high modal/harmonic density, the general rule stating that non-physical harmonic-related poles show very low damping cannot be applied. Application of processing windows and interactions between modes and deterministic component make the distinction very difficult. Some recurring frequencies were however identified in almost all analyzed dataset giving confidence that these modes are the actual structural ones.

Finally, orders were extracted to apply Order-Based Modal Analysis. During the measurement campaign, 3 tacho signals were measured. One optical sensor was placed on each High Speed Shaft and one on the Low Speed Shaft. However, because of noise on the optical probes, only one of the signals was of sufficient good quality to extract the orders correctly.

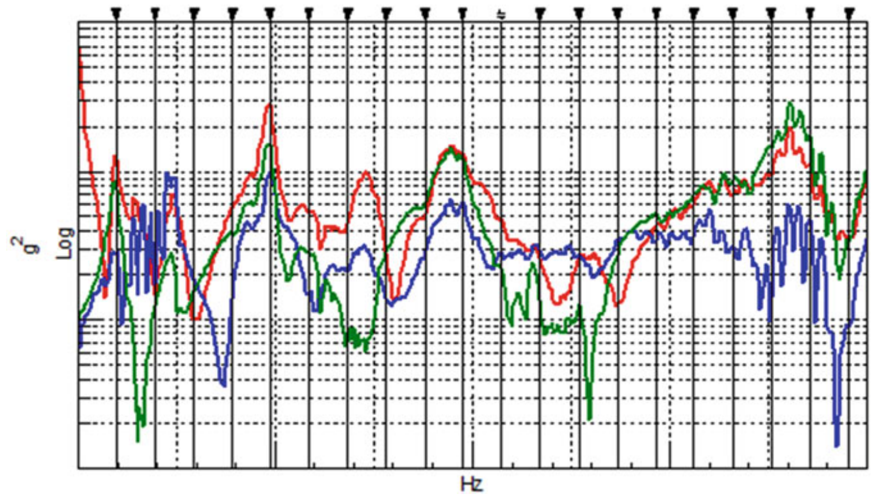


Fig. 19.11 Calculated cross powers during a run up. Vertical thick lines indicate some of the end-of-order frequencies

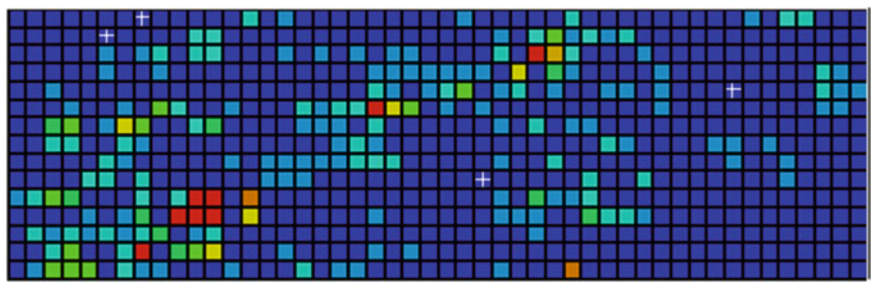


Fig. 19.12 MAC between OMA applied on run up data and EMA on OMAX FRF

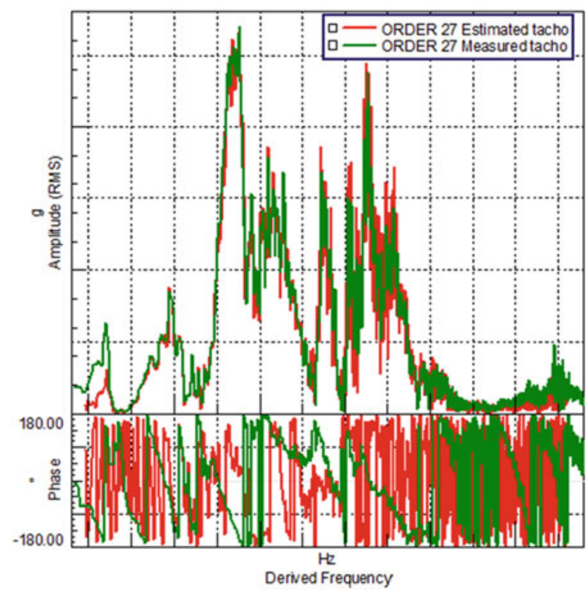


Fig. 19.13 Comparison of estimated orders with measured and estimated tacho signal used to generate the phase reference signal

Based on the results shown in Sect. 19.3, orders were calculated using the TVDFT method. To obtain a proper phase reference, the tacho signal was multiplied by the order to be extracted, the instantaneous angle calculated and finally the corresponding synchronous sine wave extracted. The importance of using the proper signal for the tracking and the phase reference is shown in Fig. 19.13. The phase of the order with the phase reference signal synchronous to the measured tacho is clearer than the one with the estimated tacho signal. Also, the importance of extracting different orders to properly identify the modes in a frequency band as broad as possible is shown in Fig. 19.14. The full modal model can then be assembled combining the models from the different orders together.

The results obtained applying Order-Based Modal Analysis to one of the dominant order are here presented. Fig. 19.15 shows the stabilization diagram obtained by applying PolyMAX on the approximately 700 measured orders. The stabilization

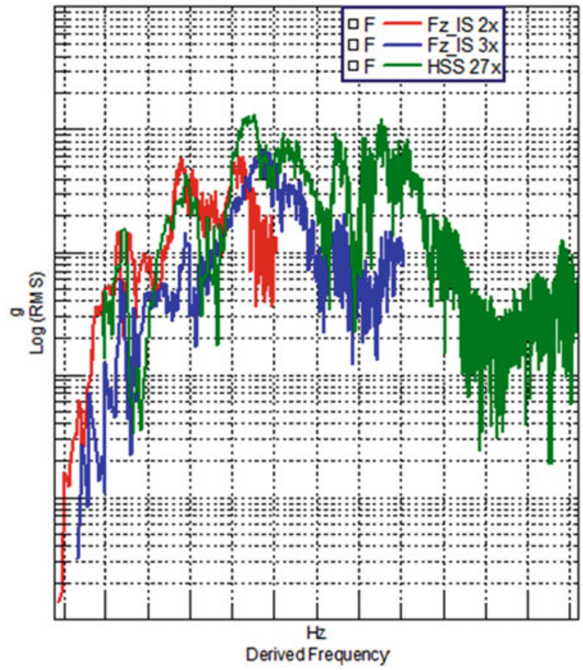


Fig. 19.14 Comparison of different offers extracted for the same point

diagram is quite clear and columns of stable poles can be easily identified. However, because of the noise in the orders, the synthesis is relatively poor (Fig. 19.16), resulting in noisy and non-consistent modes shapes. This can also be observed in Fig. 19.17. The mode shape involving the torque arms bending is compared. In both modes, the dominant fore-aft motion of the torque arms can be recognized, although the phase is slightly different. However, if MAC is computed, a value below 5 % is obtained, because of the general noise in the OBMA modal estimate. In general, however, a set of natural frequencies corresponding to those obtained in the EMA and OMA case were obtained, thus confirming that these frequencies are actual poles of the structure and not forced vibrations.

19.5 Conclusions

In this paper, the problem of identifying a modal model in operational conditions, and in particular in case of rotating machineries, was discussed. Typically, Operational Deflection Shapes at critical frequencies are analyzed to try to understand the cause of high vibrations or acoustic emissions. However, by just analyzing these features, it is impossible to understand whether the high motion is due to the input excitation, a resonance or a combination of both. For this reason, getting a modal model is of paramount importance, not only to identify the root causes of N&V problems, but also to update numerical models using operational information.

Different combinations of experiments and data analysis techniques were first validated on a numerical example and then applied on a gearbox controlled to operate at different conditions. FRFs were extracted using shakers sine sweep, both during stand-still and in operation, and standard Experimental Modal Analysis applied. Then, depending on the operating conditions (stationary or run up), Operational and/or Order-based Modal Analysis are applied. The theoretical examples have shown that all methods are able to properly identify the modes, at least those which are excited with sufficient energy. However, in real conditions, noise, measurement errors (even small errors on sensor orientation angles may cause huge inaccuracies when animating the modes) and the complexity of the test campaign, made identifying a reliable modal model quite complex, in particular in operational conditions. For these reasons, the measurements should be as accurate as possible as the methods are very sensitive to noise and other error sources (electrical disturbances, low excitation levels, inconsistency between the different runs). The importance of accurately measuring (or at least estimating) the rotational speed to perform order tracking was shown, together with the sensitivity of the different order tracking methods to the quality of the rpm signal. Finally, the importance of having a proper phase reference signal for the orders was shown. The most reliable signal was then identified as a sine sweep which instantaneous frequency is equal to the instantaneous rotational speed. In general, Order-based Modal Analysis was shown to be a reliable tool for modal parameter identification from data acquired on rotating machineries

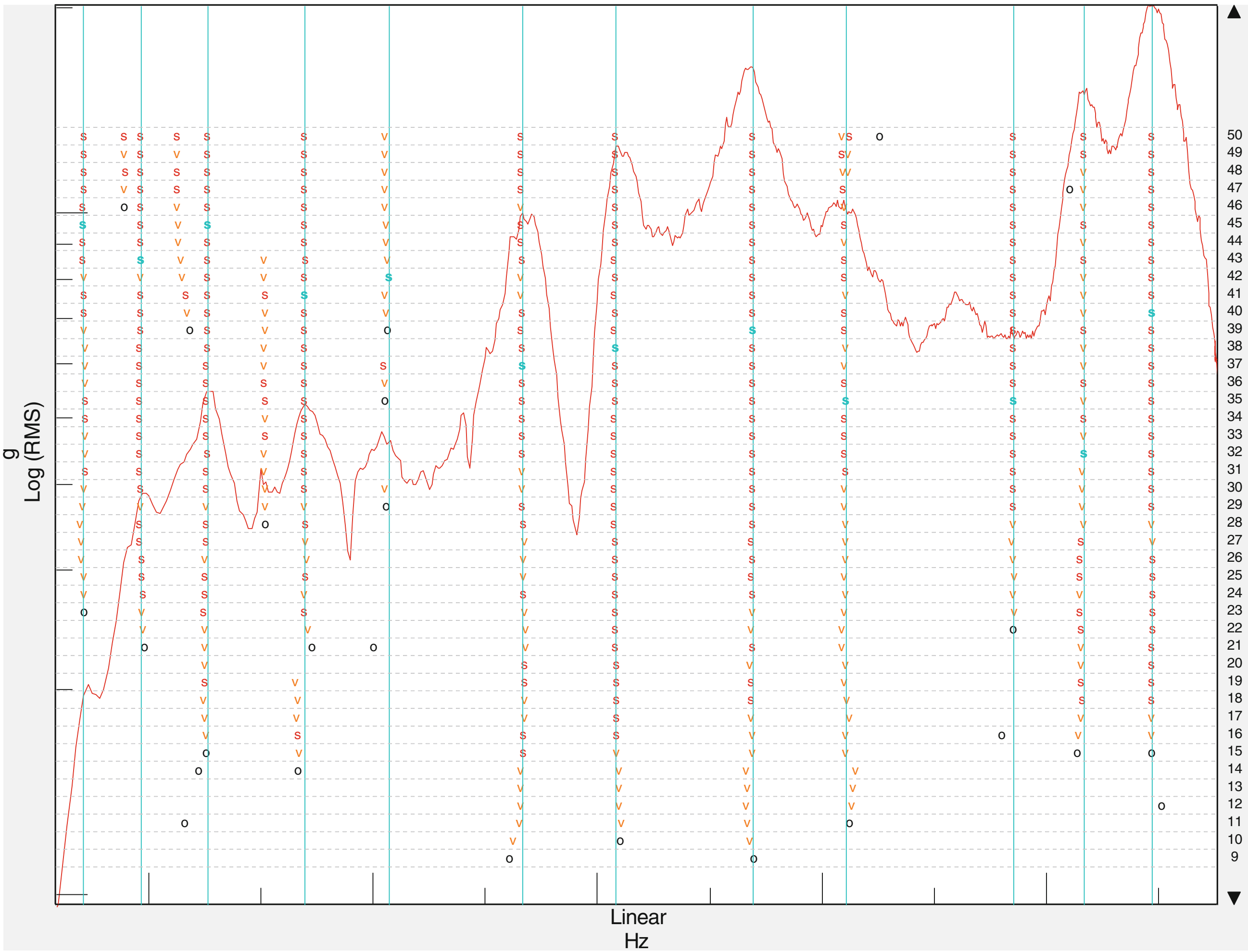
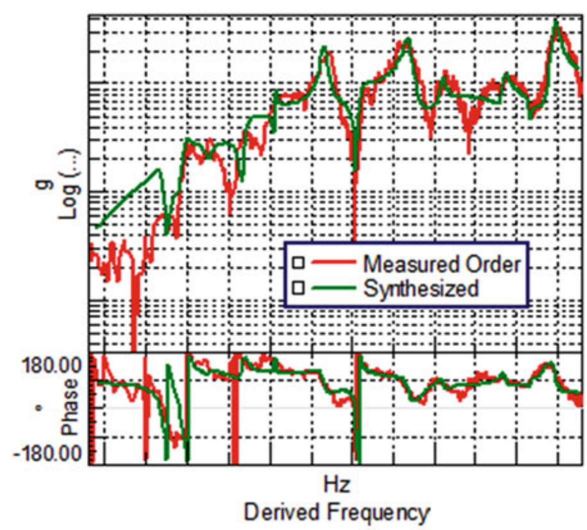


Fig. 19.15 Stabilization diagram obtained for the analyzed order

Fig. 19.16 Comparison of a measured vs. synthesized order



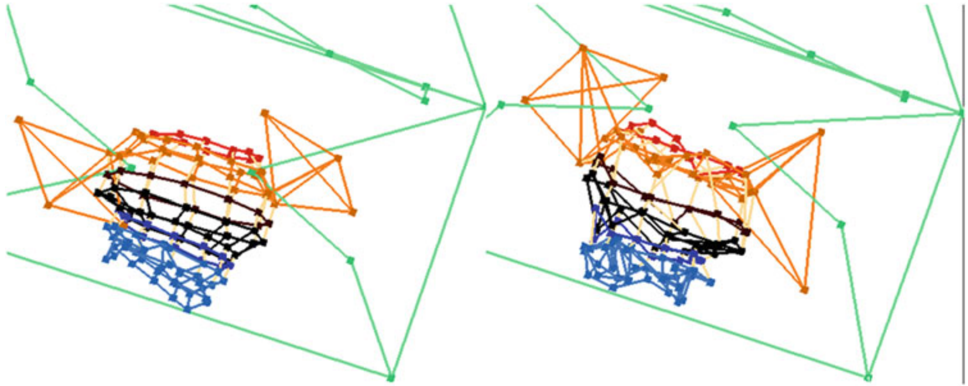


Fig. 19.17 Comparison between EMA mode shape (*left*) and OBMA (*right*)

during operation. Also, while the instrumentation is the same as the one used for Operational Deflection Shape, it allows to derive a modal model with little additional processing that can be used for root cause analysis and/or to update numerical models

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