Wind Turbine Gearbox Dynamic Characterization using Operational Modal Analysis

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ABSTRACT

The aim of this paper is to characterize the dynamic behavior of a wind turbine gearbox installed on a dynamic test rig to replicate operational conditions. Wind turbines and gearboxes operate under very dynamic and complex conditions, caused by turbulent wind, fluctuations in the electricity grid etc. In those conditions, structural nonlinearities in bearings and gears cause natural frequencies to be significantly influenced by the operational conditions. To verify the dynamic response of a multi-megawatt gearbox, a comprehensive test campaign has been performed in the context of the European project ALARM at the ZF Wind Power test rig [1]. Accelerations have been measured at more than 250 locations on the test rig and for different load levels and operating conditions. This paper focuses on the influence of the torque levels on the identified modal parameters. The acquired time histories during run-ups have been processed using different Operational Modal Analysis techniques. The aim is to provide a modal model that can be used for correlation and updating of a flexible nonlinear multibody model of the whole test rig as well as vibration levels to estimate structure-borne noise in the different operating conditions.

Keywords: Operational Modal Analysis, Wind turbine gearbox, Test campaign, Gearbox test rig, Modal parameters.

INTRODUCTION

The gearbox is one of the key subsystems in a geared wind turbine providing the task to transfer power from the low speed shaft connected to the rotor to the high speed shaft connected to the generator. As turbines become larger, more power is demanded and gearboxes with higher load capacity need to be designed. A deep knowledge into gearbox dynamics becomes of fundamental importance and noise and vibration measurements are demanded [2].

These measurements are mainly quality estimation methods for gear mesh vibrations and overall sound power levels. They are based on standard techniques for the estimation of dynamic characteristic in general applications and not focusing on the wind turbine gearbox case. Since several components properties depend on the applied torque and on the rotational shaft speed, a validation in operational conditions needs to be performed. Building on existing techniques such as "Order Tracking" and "Operational Modal Analysis", a dedicated methodology will be developed for the analysis of operational gearbox dynamic behavior. Particular attention will be reserved to the separation between structural resonances and excitation orders.

Operational Modal Analysis (OMA) is used to derive an experimental dynamics model from vibration measurements in operational conditions. It cannot be applied in a straightforward way due to the self-induced vibrations at several rpm-dependent frequencies (gear meshing orders). These frequencies with high vibration levels can be wrongly considered resonance frequencies of the system. In order to face these problems, an extensive measurement campaign has been performed at ZF Wind Power on a 13.2MW test rig facility. Accelerations have been measured at more than 250 locations on the test rig and for different load levels and operating conditions.

For stationary conditions, that means at constant rpm, the gear meshing orders will give a component at discrete constant frequencies that need to be filtered out from the signal in order to determine the resonances. Run-up tests can be considered as a multi-sine sweep excitation and the resonances can be identified combining advanced order tracking methods with operational modal analysis.

TEST RIG DESCRIPTION

ZF Wind Power's dynamic gearbox test rig is one of the world's largest test facilities in its kind matching the wind turbine power growth in the market. Gearboxes can be tested under representative loading conditions using parameterized load cases that can be programmed into the controller. Potential technical risk can be identified well in advance accelerating the life testing so that it is possible to improve gearbox reliability [3].

The "wind & rotor" side is composed of an electrical machine (motor 1), an optional speed reducer (3:1 gearbox) and a wind turbine gearbox (GB1). The "grid & generator" side is composed of an electrical machine (motor 2) and an optional speed reducer. Analogous to the operation of a wind turbine, the wind side of the test rig is speed controlled, while the generator side is torque controlled. Test gearbox 2 (GB2) is driven at a certain time varying speed corresponding to variable wind speeds and loaded with a certain time varying torque by the generator corresponding to the loading from the grid and thus experiences test conditions very similar to wind turbine behavior [4].

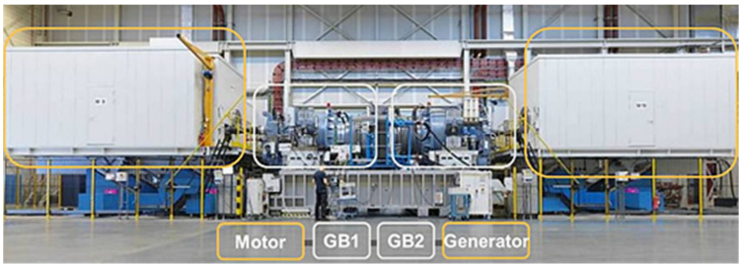

Fig.1 – ZF Wind Power test rig

It has a nominal power of 13.2 MW and a peak power capacity of 16.8 MW. The complexity of applying dynamics is tackled by the concept of load cases [5]. Each load case represents a specific part or phenomenon in the wind turbine behavior. An overview of the test rig is shown in Fig.1.

MEASUREMENT CAMPAIGN

The measurement campaign took place on the 13.2 MW dynamic test rig where a 3.2 MW and a 3 MW prototype gearboxes were placed in a back-to-back configuration with one gearbox (P3) in "generator mode" as in the wind turbine and the other (P2) in "motor mode". The two gearboxes have a slightly different gear ratio so that the motor mode gearbox does not run at nominal speed when the generator mode gearbox is being tested. The measurement campaign was comprised of over 250 measurement points and it included several conditions (constant speed, run up, shaker measurements) at different levels of torque, as shown in Tab.1.
In order to cover all the measurement points in three directions, 7 batches for each configuration were performed because of the limited number of available measurement channels. A global overview of the chosen sensor locations in the FE model is reported in Fig.2, whereas in Fig.3 the test configuration used in LMS Test.Lab is shown. The points were defined under different groups depending on the component as stated in Tab.2.
The X global axis goes from the Low Speed Shaft (LSS) to the High Speed Shaft (HSS) of the main tested gearbox (P3) that means from P2 to P3 in Fig.3; Z axis point vertically up, while Y axis is defined to get a consistent axis system.

Step	Load	Speed
1	0%	Standstill (shaker)
2	33%	run up, 200-1500 rpm (5 rpm/s)
3	33%	constant speed, 1200 rpm
4	33%	constant speed, 800 rpm
5	66%	run up, 200-1500 rpm (5 rpm/s)
6	66%	constant speed, 1200 rpm
7	66%	constant speed, 800 rpm
8	100%	run up, 200-1500 rpm (5 rpm/s)
9	100%	constant speed, 1200 rpm
10	100%	constant speed, 800 rpm

Tab.1 – Measurements load cases

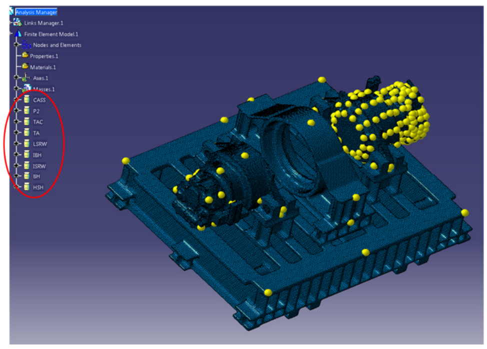

Fig.2 – Test rig FE model

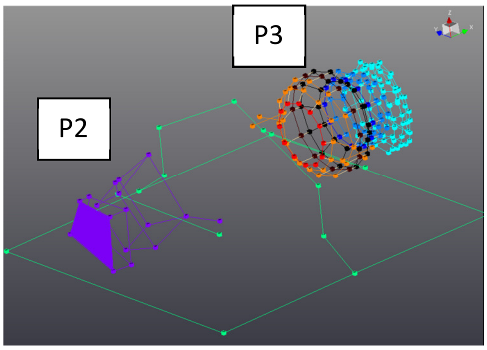

Fig.3 – Test rig measurement points

Component	Abbreviation	Number of measurement locations
Tested gearbox (gearbox 1) = P3		
Torque Arm Cover	TAC	12
Torque Arm	TA	30
Low Speed Stage Ring Wheel	LSRW	12
Intermediate Bearing Housing	IBH	24
Intermediate Speed Stage Ring Wheel	ISRW	12
Bearing Housing	BH	36
High Speed Stage Housing	HSH	76
	Total for P3:	202
Counter gearbox (gearbox 2) = P2		
	P2	27
	Test rig	
Cassette + Motor 1 (tested gearbox, P3) + Motor 2 (counter gearbox, P2)	CASS	27
	Grand total:	256

Tab.2 – Measurement points list

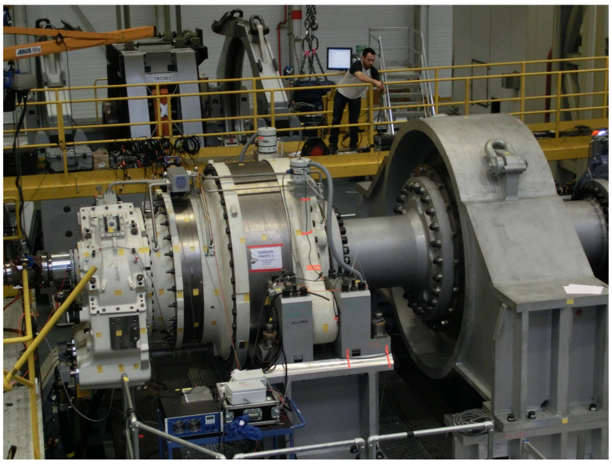

Fig.4 – P3 gearbox

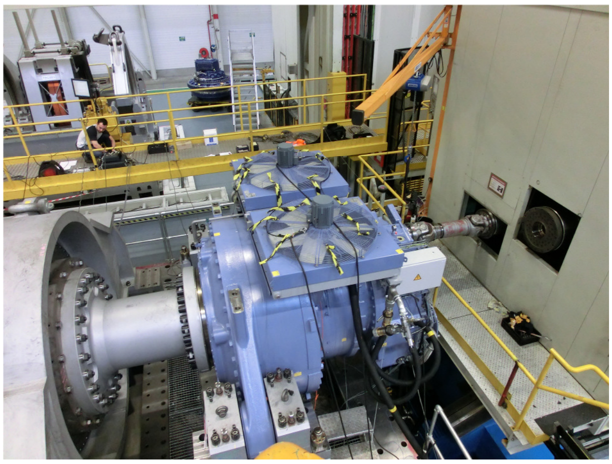

Fig.5 – P2 gearbox

OPERATIONAL MODAL ANALYSIS

Operational Modal Analysis (OMA) technique, also known as output-only modal analysis, allows identifying modal parameters by using operational measurement such as accelerations measured on several points attached to the structure [6]. OMA technique is applied when the input forces cannot be measured and when the system complies with three main assumptions. It must be linear time invariant, the excitation forces must be represented by a flat white noise spectrum in the band of interest and the forces acting on the structure must be uniformly distributed and uncorrelated both temporally and spatially. The better these assumptions are fulfilled, the better the quality of the estimated modal parameters [7]. The identification technique is similar to the classical input-output modal analysis with the substantial difference that, instead than impulse and frequency responses, it uses auto- and cross-correlation and auto- and cross-powers between signals measured simultaneously at different locations. So, there is the need to identify several reference signals that should be as less noisy as possible and that should be able to identify as many modes as possible. Operational PolyMAX [8] and Stochastic Subspace Identification techniques have been applied to the gearbox data for identifying natural frequencies, damping ratios and mode shapes.
In our case, the "natural" flat spectrum excitation provided by the rotation of the shafts inside the gearbox can be used as source. In order to fulfill the white noise spectrum hypothesis, the run up case is considered. In fact, the harmonics (orders), related to the number of revolutions, are sweeping through a broad frequency band and they are useful excitation for estimating the modal parameters. According to Tab.1, the gearbox is rotating from 200 rpm to 1500 rpm, with a speed run up rate equal to 5 rpm/s, which is a huge part of its operational rotation speed range so that the need for white noise excitation is well approximated [9]. During the measurements, the same run up case was performed at three different torque levels (33%, 66% and 100%) that correspond to step 2, 5 and 8 reported in Tab.1.
Accelerations were collected by means of tri-axial accelerometers at a quite high sampling frequency (16384 Hz), so the first step in the data processing is a data down sampling to 1500 Hz in order to focus the attention in a narrow frequency band (0 – 500 Hz). While the most part of accelerometers were moved between a batch and the following one in order to

cover all the measurement points, 8 accelerometers were kept at the same position during all the measurement campaign. Those accelerometers can, then, be used as reference channels in the calculation of cross-powers. Two points (named point 5 and point 8), which location is shown in Fig.7 and Fig.8, are considered as the most suitable reference channels since their spectra have a quite good repeatability along the different batches, as shown in Fig.9 for point 8. The smaller the differences, the better the estimation of the modal parameters is. For performing Operational Modal Analysis, a pre-processing is necessary to convert time data to auto- and cross-powers. First of all, auto- and cross-correlation functions are calculated and an exponential window is usually required before computing the FFT algorithm. The exponential window reduces the effect of leakage and the influence of the high time lags, which have a large variance [10].
Some pictures of the test rig configuration are shown in Fig.4 (P3 gearbox), Fig.5 (P2 gearbox) and Fig.6 (whole test rig).

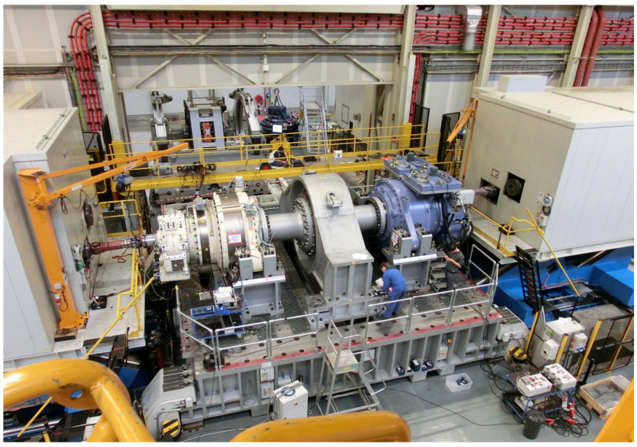

Fig.6 – Whole test rig

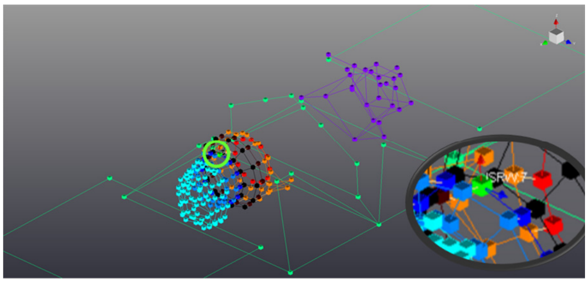

Fig.7 – Reference point 8

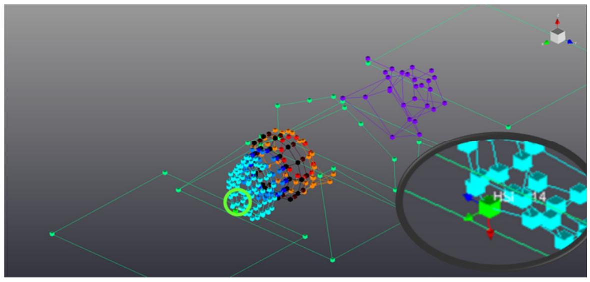

Fig.8 – Reference point 5

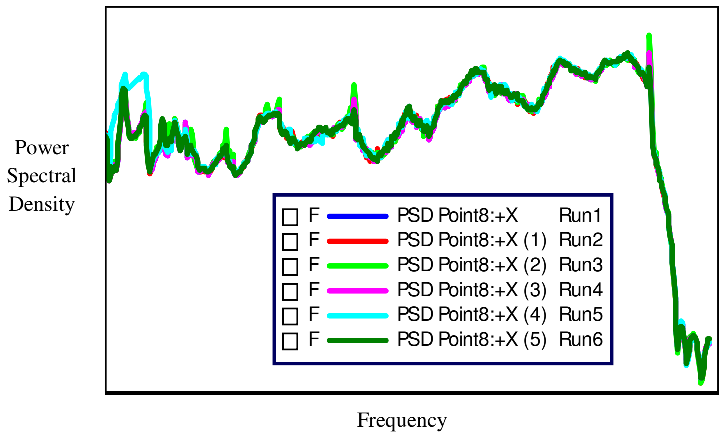

Fig.9 – PSD Point 8 for several batches at the same load case

DATA ANALYSIS

The main gearbox under test (P3) is the one in which more measurement points were considered but, in order to have a better knowledge of the system, the first part of the analysis focuses on the P2 gearbox with only 18 measurement points even if all the reference channels were placed on the other side of the overall system. The different batches are analyzed

separately and then they are merged together by a so-called multi-run analysis in which partial mode shapes are combined to get the global mode shapes. The partial mode shapes are scaled with respect to the common degrees of freedom and the poles are evaluated as averaged poles between the poles coming from the different batches.
Natural frequencies and damping ratios were calculated for the three different torque levels and the results are compared in Tab.3 and in Tab.4.

Natural frequencies comparison					
	33% torque [Hz]		66% torque [% variation]		100% torque [% variation]
1	[40 - 80]	↓	-0,32	↑	1,77
2	[40 - 80]	↑	0,75	↑	5,99
3	[60 - 100]	↑	0,25	↑	0,26
4	[60 - 100]	↑	0,71	↑	0,82
5	[80 - 120]	↑	0	↓	-0,60
6	-	-	[80 - 120]	↓	-0,37
7	[100 - 140]	↑	-	-	0,25
8	[130 - 170]	↑	1,62	-	-
9	[160 - 200]	↑	3,16	↓	2,83
10	[180 - 220]	↑	0,35	↑	2,14

Tab.3 – Natural frequencies comparison for P2 gearbox at different torque levels. The percentage values are referred to the values in the square brackets.

Damping ratios comparison					
	33% torque [%]		66% torque [% variation]		100% torque [% variation]
1	[0,5 – 1,1]	↑	151,81	↓	142,17
2	[1,7 – 2,3]	↓	-27,15	↑	-20,26
3	[0 – 0,5]	↓	-22,22	↑	85,19
4	[1,5 – 2,1]	↑	4,97	↓	-20,99
5	[0 – 0,5]	↑	40,00	↑	440,00
6		-	[0,5 – 1,1]	↓	-38,82
7	[1 – 1,6]	↓	-	-	-25,00
8	[0 – 0,5]	↑	127,78	-	-
9	[0,5 – 1,1]	↓	-40,45	↑	13,48
10	[0 – 0,5]	↑	56,67	↑	93,33

Tab.4 – Damping ratios comparison for P2 gearbox at different torque levels. The percentage values are referred to the values in the square brackets.

After the P2 is fully analyzed at the different torque conditions and its modal parameters are estimated and compared, the overall system can then be analyzed following the same procedure and including all the measurement points.

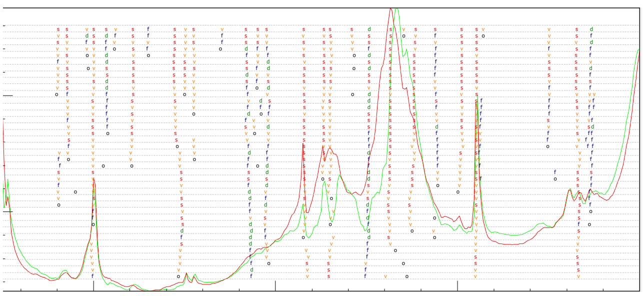

Fig.10 – Stabilization diagram using Operational PolyMAX

The accelerometers placed on the auxiliaries are not included in the analysis. While a shift toward higher natural frequencies (upward arrow) can be seen when the torque is increasing, the same is not always true for the damping ratio that is known to be a parameter which is estimated with more uncertainties [11].
When the full system is considered, not all the modes are identified considering only one reference channel so that two different channels need to be considered at the same time. Fig.10 shows a typical stabilization diagram used to estimate the modal parameters selecting the stabilized poles while increasing the model size.
In Tab.5 a comparison between the natural frequencies identified considering three different reference channel configurations is shown; on the other hand, Fig.11 shows a Modal Assurance Criterion (MAC) comparison between two of the three configurations to show that the identified mode shapes are coherent if two reference channels are considered instead than only one. The frequency resolution is set equal to 2 Hz.

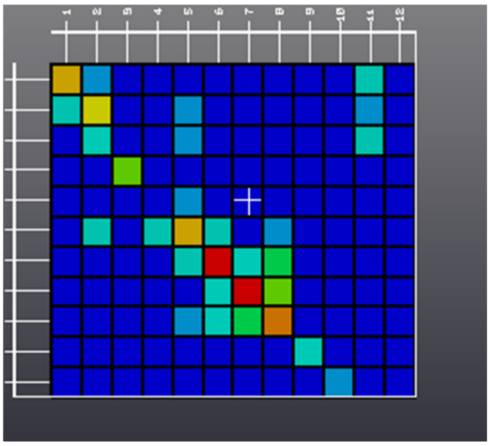

Fig.11 – Modal Assurance Criterion (MAC) comparison using different references (point 5 vs. point 5 + point8)

Natural frequencies comparison (100% torque)			
	Point 5 reference [Hz]	Point 8 reference [Hz]	Point 5 + Point 8 reference [Hz]
1	[60 - 100]	1,52	1,64
2	[80 - 120]	-0,60	-1,20
3	[130 - 170]	-	-0,60
4	[160 - 200]	1,42	0,74
5	[180 - 220]	-0,15	-0,50
6	[200 - 240]		-0,33
7	[200 - 240]	-	0,13
8	[240 - 280]	-	0,19
9	[280 - 320]	-	0,91
10	[350 - 400]	-0,90	-1,35

Tab.5 – Natural frequencies comparison for the full system using different reference points. The percentage values are referred to the values in the square brackets.

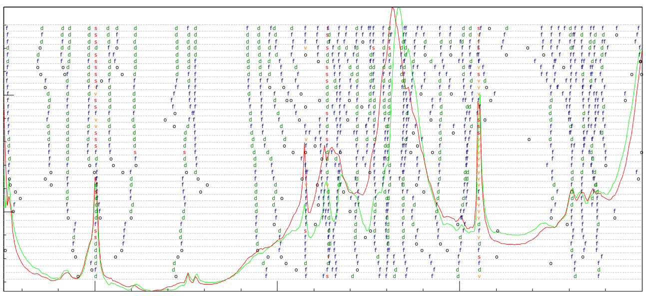

Fig.12 – Stabilization diagram using Stochastic Subspace Identification (SSI)

The natural frequencies and the mode shapes are compared using the two different operational modal analysis techniques mentioned in the previous sections: Operational PolyMAX and Stochastic Subspace Identification (SSI) [12].
Fig.12 shows the same stabilization diagram of Fig.10, with the only difference that it is obtained using the SSI technique. The stabilized poles are not as clear as they were in the previous one, but a better estimation of the mode shapes is possible.

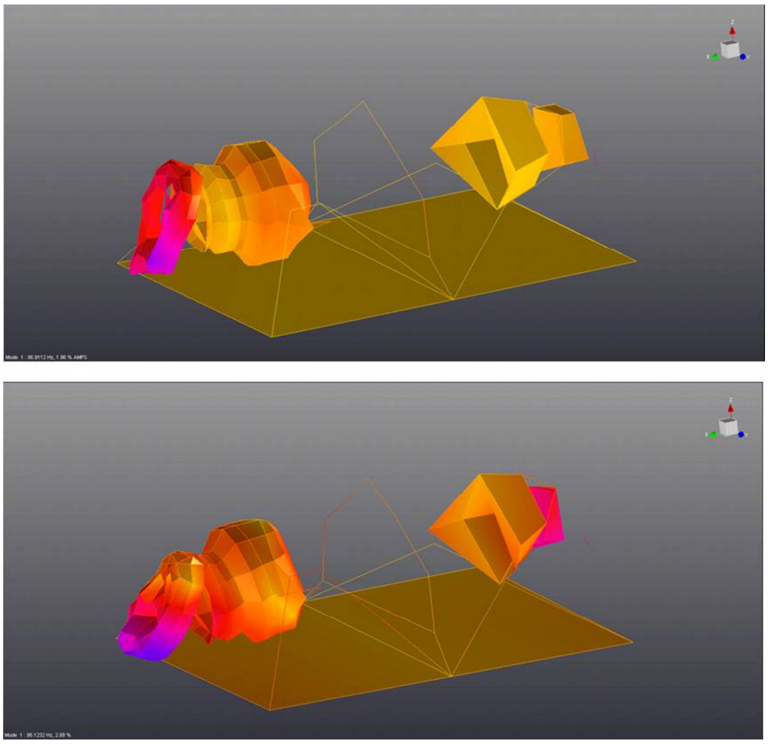

Fig.13 – Mode shapes instantaneous picture comparing Operational PolyMAX result (above) with Stochastic Subspace Identification technique (below)

The natural frequencies and the mode shapes are compared using different torque levels (66% and 100%) increasing the frequency resolution from 2 Hz to 1 Hz.

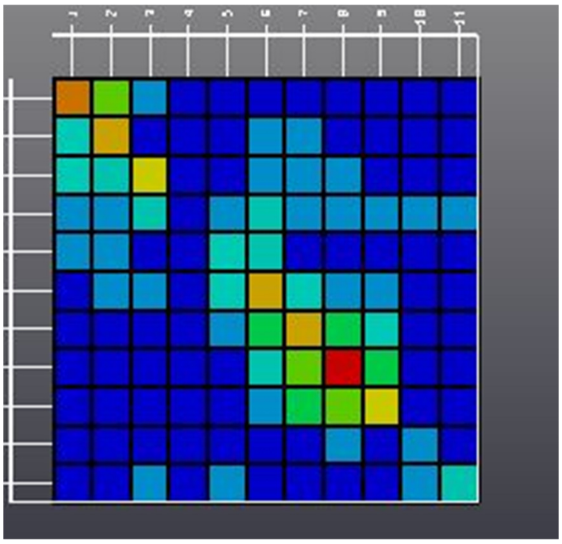

Fig.14 – Modal Assurance Criterion (MAC) comparison at different torque levels (66% vs. 100%)

Natural frequencies comparison		
	66% torque [Hz]	100% torque [Hz]
1	[60 - 100]	0,05
2	[80 - 120]	-0,64
3	[130 - 170]	-0,09
4	[160 - 200]	3,30
5	[180 - 220]	0,48
6	[200 - 240]	0,03
7	[200 - 240]	0,40
8	[240 - 280]	1,22
9	[280 - 320]	-0,12
10	[340 - 380]	2,05

Tab.6 – Natural frequencies comparison at different torque levels

A quite good agreement between the two different torque conditions can be underlined both in the natural frequency table and in the MAC comparison which calculates the correlation between two different sets of modes in order to

compare different processing. The shift toward slightly high natural frequencies can be seen also for the full system respecting the trend that has been found for the single component, as can be seen from Tab.6.
Fig.14 shows the MAC comparison between the full torque condition (100%) and the 66% torque case.

CONCLUSIONS

A huge measurement campaign has been performed in order to characterize the gearbox behavior on the test rig. The set up and the test rig are described and, after that, a first analysis has been done and a first modal model has been obtained using OMA techniques and focusing on the run up measurements at different torque values. First of all the analysis has focused on a single component considering a reduced set of measurement points and then the full system has been considered. In both cases, a positive shift toward higher frequencies with an increasing torque is a consistent trend during the different set of measurements. A consistent set of modal parameters such as natural frequencies, damping ratios and mode shapes has been found.
Building further on existing Operational Modal Analysis (OMA) and Order Based Modal Analysis (OBMA) techniques, a dedicated methodology in the presence of rotating machinery will be developed taking into account the separation between excitation orders (rpm dependent) and structural gearbox resonances. In addition the use of Operational Deflection Shapes (ODS) is an added value in order to optimize these techniques.

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