

Operational Modal Analysis on a Wind Turbine Gearbox

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ABSTRACT

A Wind Turbine has many rotating parts making it a real challenge to make modal analysis in operating conditions. The issue is to separate the structural modes from the harmonic components. In addition to the gearbox itself, there are also the generator, the pitch-drive and the yaw-drive producing harmonic vibration for a complete Wind Turbine in operating condition. The 37 ton gearbox tested was running with a rotational speed of 1550rpm and 2900kW load, and mounted in a test rig.

In this paper several approaches and techniques have been used in order to extract the structural modes. A preliminary random hammer testing was applied in order to get some idea of the natural frequencies of the gearbox housing itself. Under operational condition the frequency domain identification techniques applied have been Frequency Domain (Singular Value) Decomposition, FDD and Enhanced Frequency Domain (Singular Value) Decomposition, EFDD, and Curvefit Frequency Domain (Singular Value) Decomposition, CFDD with automatic Harmonic Detection features based on kurtosis calculations. In time domain the Stochastic Subspace Identification, SSI has been applied. In the calculations of the stabilization diagram based on SSI it is possible to set up user tolerances to identify unrealistic low damping, thus being able to separate harmonic components from the structural modes. But also a good deal of common sense and engineering skills is needed, when making manual inspection and identification of the spectra of the singular values.

1. INTRODUCTION

Operational Modal Analysis (OMA), also known as output-only modal analysis, is based on measuring only the output of a structure and using the ambient and natural operating forces as unmeasured input. It is used instead of classical mobility-based modal analysis for accurate modal identification under actual operating conditions, and in situations where it is difficult or impossible to artificially excite the structure. The basic idea and theory of OMA can be found in [6], [7] and [8].

Originally developed for measuring on civil engineering structures like buildings, towers, bridges and offshore structures, OMA is today used extensively on mechanical structures like aircraft, vehicles, ships and operating machinery in general. While civil structures are mainly loaded by ambient stochastic forces like wind, waves, traffic or seismic micro-tremors, the loading forces of many mechanical structures are often more complex. They are typically a combination of harmonic components originating from the rotating and reciprocating parts and broadband excitation originating from either self-generated vibrations from e.g. bearings and combustions or from ambient excitations like air turbulence and road vibrations. However, also civil engineering structures may have responses superimposed by harmonic components from e.g. ventilation systems, turbines and generators. As the input forces to the structure are not measured in OMA, the question of how to separate harmonic components from true structural modes is often asked. This paper starts by briefly describing the consequences of having harmonic components present in the responses. Then it is followed by a short description how the influence of harmonic components are eliminated or reduced in the various identification techniques. Finally the results from Operational Modal Analysis on a Wind Turbine Gearbox are presented.

2. CONSEQUENCES OF HARMONIC COMPONENTS

The algorithms used in Operational Modal Analysis (OMA) assume that the input forces to the structure are stochastic in nature. However, the excitation often consist of strong harmonic components (deterministic signals) superimposed on the stochastic broadband excitation. As the input forces are not measured in OMA, attention must be paid to identify and separate the harmonic components from the structural modes and to reduce their influence on the extracted modal parameters. As described in Jacobsen [1], harmonic components can significantly bias the obtained modal parameters and even prohibit the extraction of the structural modes. The other problem is that harmonic components are potentially mistaken for being structural modes.

The consequences of having harmonic components present in the responses depend on both the nature of the harmonic components (number, frequency and level) and the modal parameter extraction method used. As ref. [1] states, the presence of harmonic components in the responses is not necessarily always a problem. However, it is important to be able to identify potential modes as being either harmonic components or true structural modes and to know the consequences and the pros and cons of the different identification techniques. It should be noticed that harmonic components cannot, in general, be removed by simple filtering as this would in most practical cases significantly change the poles of the structural modes and thereby the estimated values of natural frequency and modal damping.

3. IDENTIFICATION OF HARMONICS USING FREQUENCY DOMAIN DECOMPOSITION METHODS

Either the harmonics must be outside the SDOF bell for the EFFD or CFDD methods thereby potentially narrowing the frequency range for the parameter estimation thus resulting in poorer identification (leakage) or the harmonics must be removed from the data. The harmonics can be identified and removed using kurtosis techniques, followed by linear (EFDD) or polynomial (CFDD) interpolations before modal parameters estimation. See refs. [2], [3], [4] and [5]. The basic idea is that probability density function, PDF for structural mode excited by stochastic excitation differs significantly from the PDF for deterministic excitation such as sinusoidal excitation. It was shown in ref. [3] that the difference is most easily identified when calculating the kurtosis of the measured data. The kurtosis of a stochastic variable provides a way of expressing how peaked or how flat the probability density function is. The kurtosis is defined as the fourth central moment of the stochastic variable normalized with respect to the standard deviation. The PDF of the response of a structural mode stochastically excited will be normal distributed with a kurtosis of 3, whereas the kurtosis for a sinusoidal component is $1\frac{1}{2}$.

Extended Kurtosis Checking method was described in ref. [2] and it is shown to be a robust harmonic indicator. In this method all frequencies are investigated using narrowband frequency filtering, while the number of channels to be used for the harmonic indicator is reduced from all the measurement channels to only the projection channels. For a system with many measurement channels, the cross spectrum matrix of the measured responses typically consist of much more rows/columns than there are structural modes. Many of the rows/columns are linear dependent of each other. For such a system there is consequently a substantial amount of redundancy, indicating that it might not be necessary to include all of the measurement channels in the algorithm for the harmonic indicator. The basic idea is to find the subset of channels, that provides maximum amount of information. The subset of measurement channels to be used is called the projection channels and found using correlation techniques, ref. [4].

As the Extended Kurtosis Checking method is quite computational intensive, it becomes rather time consuming in case of measurements performed with many projection channels analyzed and using a high number of frequency lines.

In the Fast Kurtosis Check the number of frequency lines are reduced as described in the following. In case of strong harmonic excitation of the structure, narrow peaks will be found in multiple of the singular value curves at the harmonic excitation frequency as several modes have been significantly excited at these frequencies. See figure 11, where the first two singular values are accentuated.

If an abrupt change happens at the same frequency in at least two singular value curves, then we have detected a potential harmonic component. This potential harmonic component should subsequently be tested using the kurtosis check method as described above. In this way it is possible to significantly reduce the number of frequencies to be checked, see ref. [4].

4. IDENTIFICATION OF HARMONICS USING STOCHASTIC SUBSPACE IDENTIFICATION, SSI METHODS

The SSI methods are using State Space Modelling and estimate both harmonics and structural modes, ref. [7]. The modes are estimated correctly even for harmonics close to - or with same frequency as - the modes. Information in the time signal is used both to extract the harmonics and the modes, therefore the recording time should generally be longer than the case where only structural modes exists. The structural modes can be separated from the harmonic components in the stabilization diagram by for example selecting appropriate tolerance values for the minimum damping. The calculated damping is unrealistic low for harmonic components, as well as the obtained damping values for these components also depends on the applied FFT frequency resolution, ref. [2].

5. DESCRIPTION OF WIND TURBINES

The history of wind turbines (See figure 1) which produce electricity can be traced back to the late nineteenth century [10]. The beginning of the modern wind turbine can be set to the mid-1980s where in California a very large number of small wind turbines (< 100kW) were installed. After this first boom the wind turbine market has developed over the last 20 years from selling single prototypes to enthusiastic groups to selling power plants to energy companies consisting of several turbines (Figures 3). Also the turbine size has increased in the last 20 years (Figures 2 & 4).



Figure 1: Wind turbine

During this boom in the '80s a special design appeared on the market which was easier to service and to repair. It was the so called "Danish" concept.

The Figure 5 shows the typical wind turbine design. This concept is based on a three blade rotor connected to a main shaft. The main shaft is supported by one or two main bearings which are mounted the main frame. The main shaft transfers the torque to a linked gearbox. The gearbox transforms the low rotor speed (12 - 25 rpm) to a speed of about 1500 to 1800 rpm for the subsequent generator. In the last years this concept has been modified and improved concerning gearbox types and generator speeds.

There are also wind turbines on the market available with a direct drive, which do not need a gearbox.

The generator is the last part of the drive train. It converts mechanical to electrical energy. The static grid (50Hz/60Hz) and the fluctuation in wind/mechanical power make it different to other generators. In wind turbines you find two different basic concepts of generator. The asynchronous generator is the most common generator for wind turbines. It is an easy, robust and cheap concept. The synchronous generator is a more complex and expensive system. For direct drive system it is the common used generator.

Additional to the drive train there are also some "supporting" systems with rotating parts. One is the pitch system, which is used for rotating the blades around their center line. For the pitch drive, different solutions can be found, which are either an electrical or a hydraulic systems. A second system is the yaw drive. This drive is turning the nacelle with the rotor in the right wind direction. This drive consists of a large gear rim mounted on the tower and four or six electrical drives in the nacelle.

The pitch and the yaw drive are necessary for an efficient work of the wind turbine, but both systems are not working permanently. Therefore they normally do not excite any dynamic vibration problems.

The gearbox (Figure 6) is one of the main components of the wind turbine. It transfers nearly the complete load from the rotor to the generator. The gearbox will also change the speed from the low rotating rotor to the high rotating generator. Most gearboxes for wind turbines are build up in three stages. The first is generally a planetary stage the second is a planetary or a helical stage and the third one is a helical stages.



Figure 2: A 62m long wing of a Wind Turbine

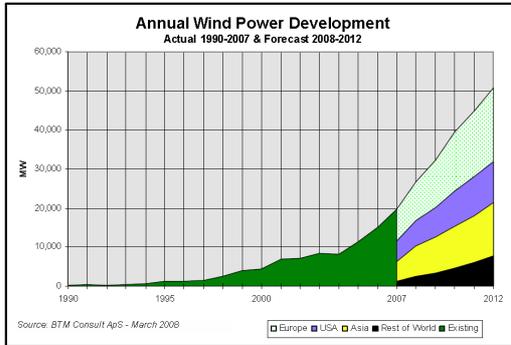


Figure 3: Annual Windpower Development

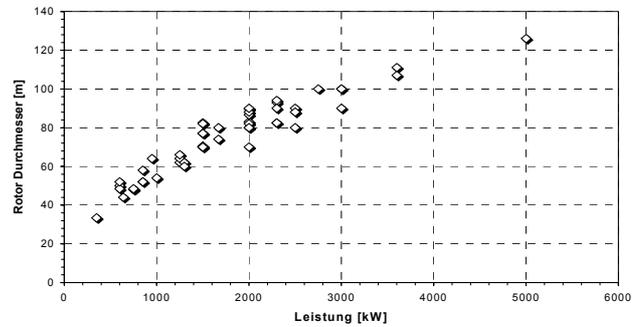


Figure 4: Rotor size development

The gearbox is the part with the most excitations during operation. Due to his construction with a lot of shafts and gears it has several frequencies under operation. Dominating frequencies are the rotating frequencies of the shafts, especially from the high speed side. A second group of frequencies are the tooth mesh frequencies of each stage. For both groups you could find also several harmonics. In a typical FFT- analysis of a gearbox under operation you could find at least round about 10 different frequency peaks in the spectrum. See also reference [9].

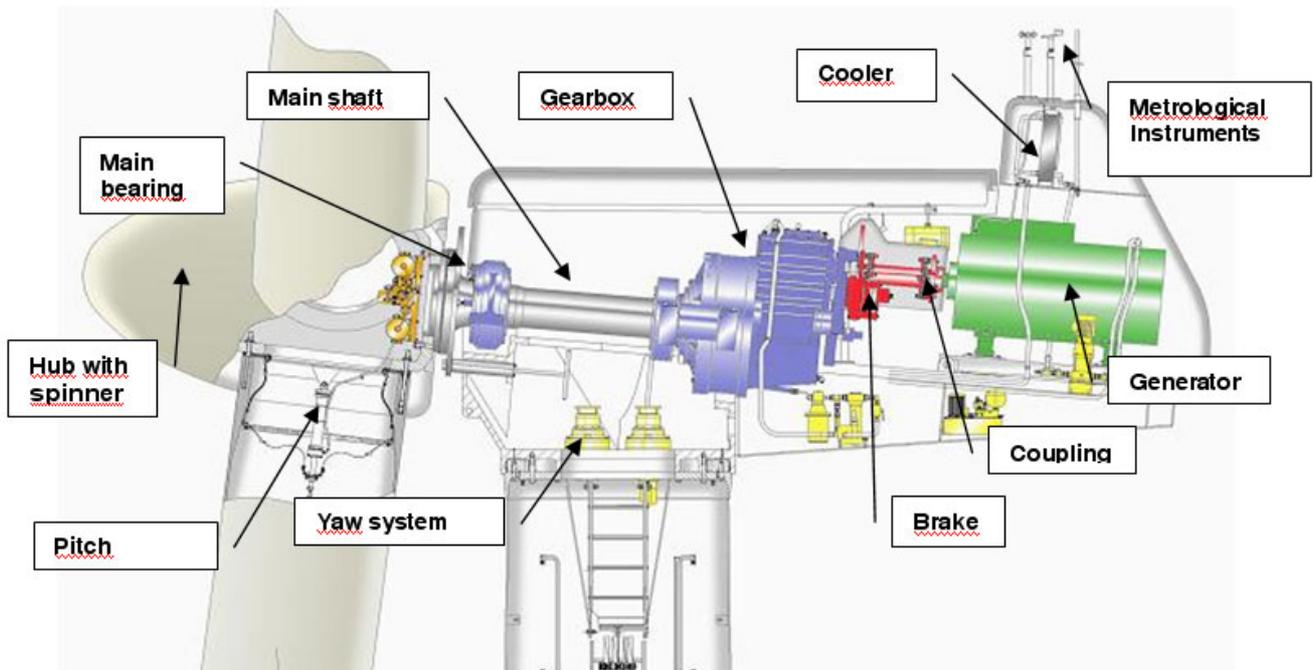


Figure 5: Typical wind turbine design

A Wind Turbine and particular the gearbox has many rotating parts, thus there is a need for testing its structural and vibrational behaviour. A typical type of test could be to make Operating Deflection Shapes (ODS) measurements in order to obtain the vibration pattern, at the running speed or some other significant harmonic components. Another type of test is Operational Modal Analysis, from which we will obtain the modal parameters natural frequency, damping and mode shape for a number of dominating modes. Only the modes that are excited under the given operating condition will be identified.



Figure 6: Gearbox and examples of gears from Wind Turbines

6. INSTRUMENTATION AND MEASUREMENT SETUP

A 12 channel PULSE Type 3560D front end was used. As reference transducers two Miniature Triaxial DeltaTron® piezoelectric ThetaShear® Type 4504-A accelerometers with a sensitivity of 1mV/ms^2 were positioned, one in each end of the gearbox, see the blue arrows on fig. 7a. As roving transducers four monoaxial Piezoelectric Accelerometer Monitoring DeltaTron® Accelerometer Types 8341 with a sensitivity of 10mV/ms^2 were used. See figs. 7 & 8 for location and mounting of the accelerometers.

The PULSE LabShop Type 7700, Modal Test Consultant Type 7753 and Operational Modal Analysis type 7760 software were used.

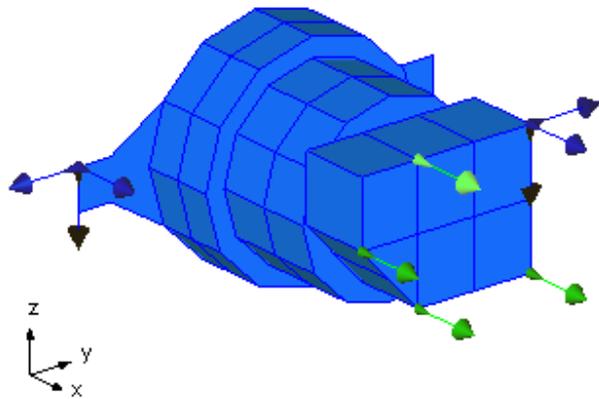


Fig 7a. Locations of transducers for dataset no. 1



7b. Magnet mounting of the roving transducers

Ten dataset were measured obtaining a total of 40 roving DOFS and 6 reference DOFS, see fig 8.

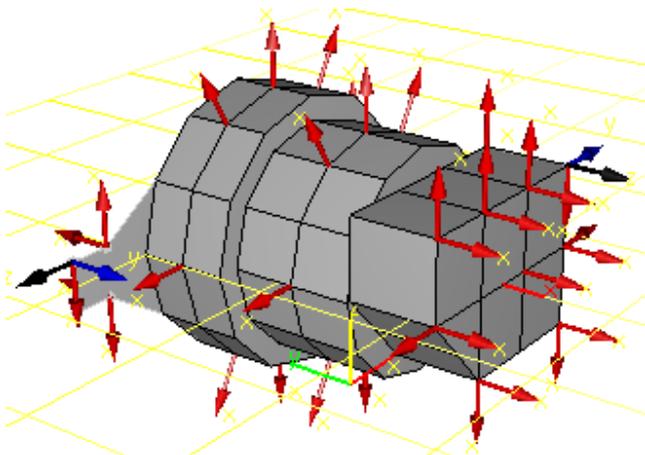


Fig. 8, A total of 46 measured DOFS using 10 data sets

7. PRELIMINARY MEASUREMENTS USING SIMPLE HAMMER EXCITATION

Some preliminary random impact measurements were performed using an ordinary hammer. Triaxial response measurements we made at three locations, each location measuring in horizontal, vertical and axial direction. 100 second long time data recordings were obtained using 3,2 kHz frequency range for post analysis using different frequency ranges.

Fig 9 shows that for example 68,25 Hz (cursor location) is a candidate for being a resonance frequency. The complete list of candidates below 200 Hz is found in table 1. It is reasonable to believe from this that there are resonances at 41 Hz (vertical), 68 Hz (axial), 87 Hz (vertical, very weak), 109 Hz (vertical) and 187 Hz.

In the OMA analysis a frequency range of 400 Hz and a line spacing of 0,4 Hz (resolution of 0,6 Hz by using Hanning Weighting) has been selected using 6 projection channels.

A simple FDD peak picking shows that only modes at 42,4 Hz and 68,8 Hz are to be seen in the SVD plots in fig 10a. Both look like rolling modes with a MAC of 0,76. But also a mode at 212 Hz were found, primarily yaw at the left end (as shown in figs. 4a, 5 and 7b) of the gearbox. Thus more extensive analysis was needed in order to extract more information.

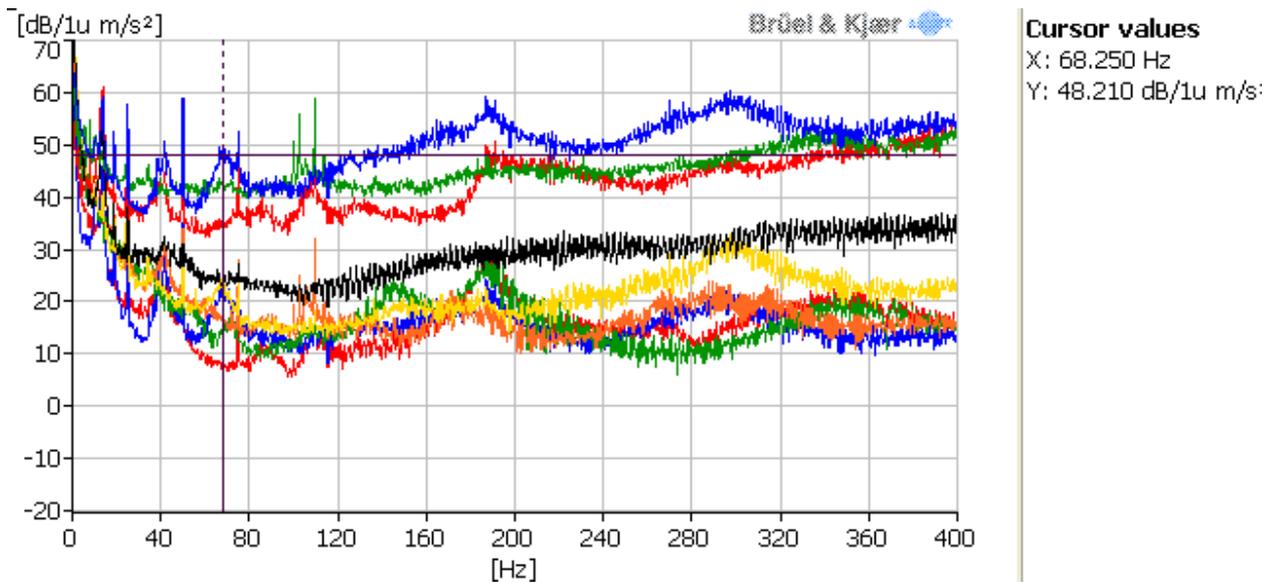


Fig 9. Measurement result from hammer test, 9DOFS

Freq (Hz)	P1 x	P1 y	P1 z	P2 x	P2 y	P2 z	P3 x	P3 y	P3 z
13	X			X		X	X	X	
41	X	X		X			X	X	
69		X			X			X	
72			X						
87	X			X			X		
109	X			X			X		
146			X						
187	X	X	X	X	X		X	X	

Table 1, Resonance inspection of autospectra from 3 points, 9 DOFS

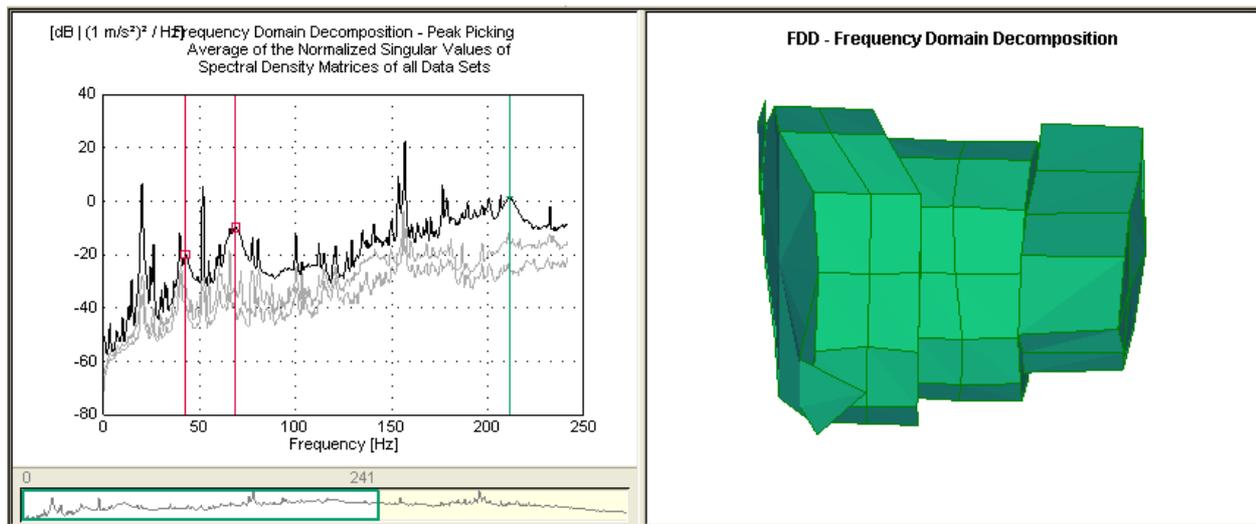


Fig 10a SVD indicates 3 resonances

Fig 10b Mode shape at 211 Hz

8. OPERATIONAL MODAL ANALYSIS

Measurements were performed using a frequency range of 1.6 kHz ($f_s = 4096$ Hz, $f_n = 2048$ Hz) and a recording length of 15 sec (102400 samples). A decimation factor of five has been used in OMA in order to obtain a useful frequency range of approximately 400 Hz. Extended Kurtosis Check has been enabled.

8.1 Frequency Domain Decomposition, FDD

The combination of FDD, Enhanced Kurtosis Checking and Automatic Mode Extraction establish a modal domain around 68,4 Hz only and also around 112 Hz if we accept a high MAC value between the shapes, in this case also a MAC of 0,76. See fig 11. The mode at 42,4 Hz must be picked manually.

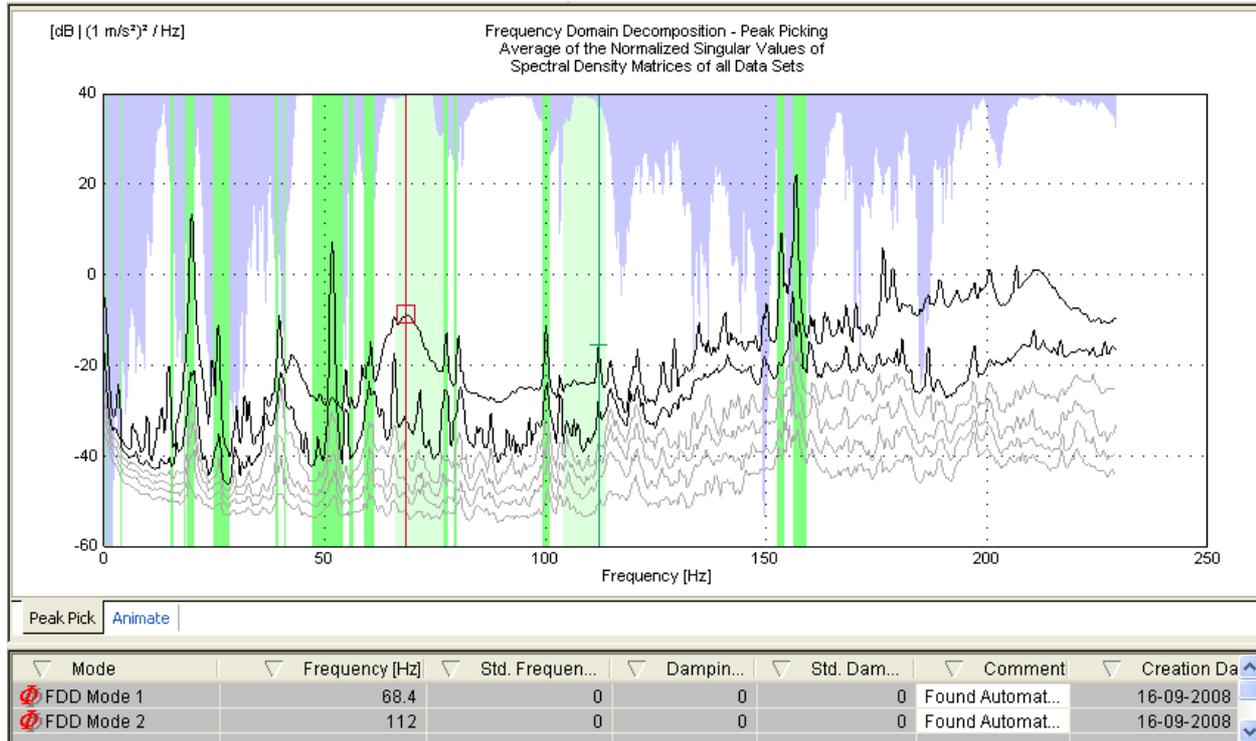


Fig. 11, Modes at 68,4 Hz and maybe also at 112 Hz using FDD

On the other hand, a visual inspection reveals narrow peaks at both first and second singular value at 112 Hz indicating that it could be harmonic excitation although not found by the Harmonic Indicator even though a range of different threshold have been tried.

At each frequency there are 6 Singular values, since 6 projection channels were chosen. Singular values 1 and 2 are made black to make them easier to see. The dark green vertical lines are harmonic indicators. A threshold can be set, the lower threshold the more green vertical lines will appear. Using EFDD or CFDD those singular values will be removed before modal parameter estimation takes place. The light blue graph is a Modal Coherence where neighbouring singular vectors are compared. A frequency region with high modal coherence (above a certain threshold) is indicated with light green colour. Such a region is called a modal domain, since it is dominated by one shape only.

8.2 Enhanced Frequency Domain Decomposition, EFDD

Adjusting the estimation control parameters and inspection all 10 data set for the straight line curvefit process on the magnitude of the autocorrelation function in order to obtain damping, we find a good consistency in the data between data sets as shown in the Standard Deviation, see table 2. The damping at 112,4 Hz has a realistic value. The 42 Hz mode is picked manually.

Mode	Frequency	Std. Frequency	Damping	Std. Damping
1	42,66 Hz	0,20 Hz	1,80%	0,31
2	67,81 Hz	0,37 Hz	3,70 %	0,53 %
3	112,4 Hz	0,21 Hz	1,34 %	0,44 %

Table 2, Results of Enhanced Frequency Domain Decomposition, EFDD

8.3 Curvefit Frequency Domain Decomposition, CFDD

The results are shown in fig 12. As can be seen the SDOF model (blue curve) is more narrow than SVD data (red curve) leading to damping values, which is about half the values from EFDD as well as the consistency of the damping values between data set is also better (smaller). Finally the mode at 42 Hz was manually picked again, once more giving a damping value equal to half the value from EFDD estimation.

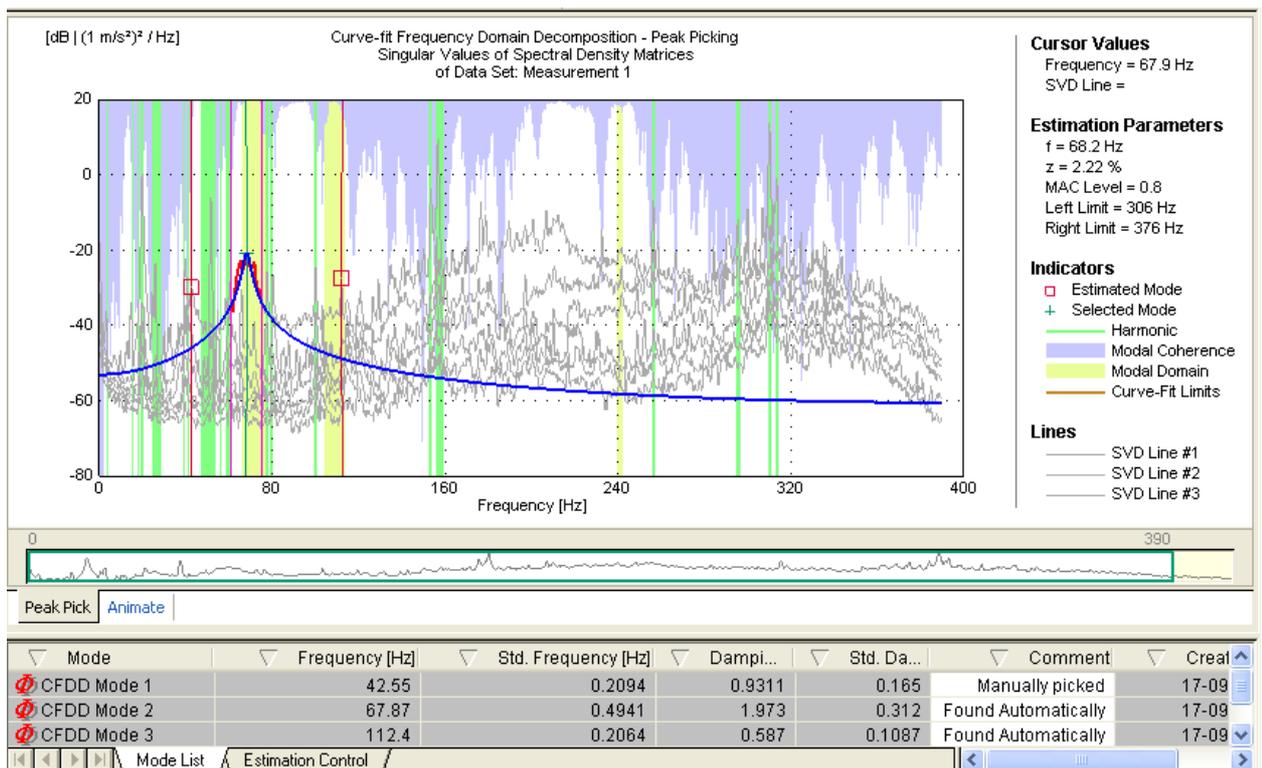


Fig 12, Results of Curvefit Frequency Domain Decomposition, CFDD

8.4 Stochastic Subspace Identification, SSI

Since there is not much difference between the 3 methods, Unweighted Principal Component, UPC, Principal Component, PC and Canonical Variate Analysis, CVA (see refs. [7,8]) only the UPC method has been applied. The results are shown in fig. 13. A 0.5% lower limit for damping has been selected for the stabilization diagram for

this data set (Data set no. 5). All shapes looks reasonable when animated being either a mode shape or an ODS, f.ex. the shape at 19.7 Hz looks basically like a yaw mode. The harmonic indicator used in FDD, EFDD and CFDD suggests that we are dealing with harmonics at 19,7 Hz, 51 Hz and 153,6 Hz, where we also find a low damping value for two lowest frequencies. Zooming and further decimation of the data reveals that there is a mode at 19,7 Hz and also a harmonic around this frequency. This decimation also reveals the mode at 42 Hz. Also by decimation 112,4 Hz are found but with 6 times and 12 times higher damping value than EFDD and CFDD. Thus we believe that we do not have a structural mode at 112,4 Hz. Modes at 67,96 Hz and 210 Hz were also found by manual inspection using the FDD method. The mode at 67,96 Hz were found by the Automatic Mode Estimation using both EFDD and CFDD. The “mode” at 182 Hz requires some discussions. The hammer test suggests that we have a mode at 187 Hz, this frequency is found in 7 out of 9 DOFs, see fig. 9 and Table 1. On the other hand this mode is not excited in operational condition as seen for example in the first singular value in fig. 11. Fig 11 shows 2 narrow peaks at 176,4 Hz and 178,8 Hz, but they look like harmonics and they are indicated as harmonics when adjusting the harmonic threshold to a slightly different value than the one used in fig 11. Animation of the “mode” found at 182 Hz looks non-physical like a compression, where the left side (figs. 7a, 8 and 10b) is moving forwards and backwards. Since only measurements at the left end are made at the two small flanges, where also one of the reference transducers are located, it simple suggests animation interpolation errors so the “mode” at 182 Hz will not be considered as a mode.

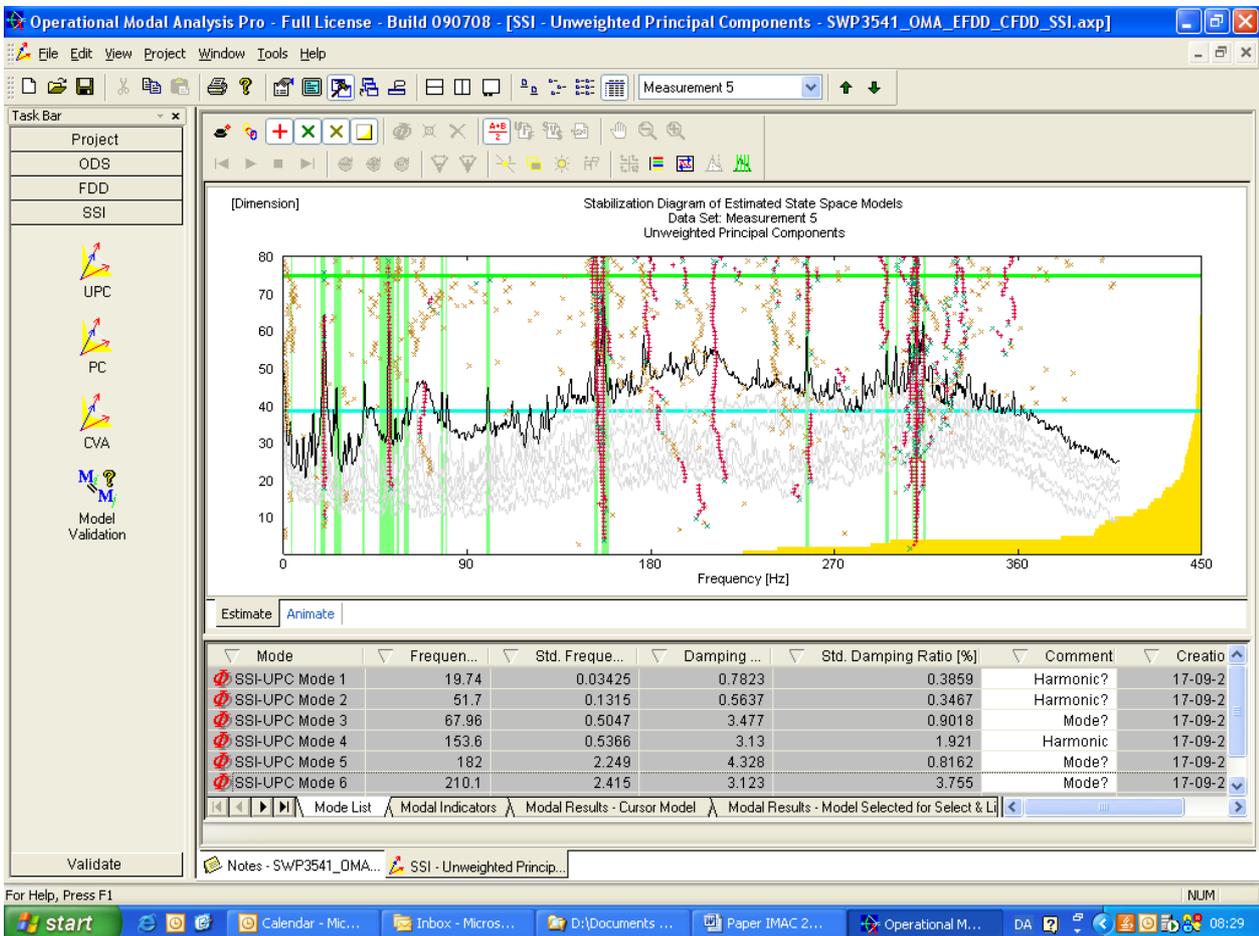


Fig. 13: Stabilization diagram for data set no 5 and list of “modes” found by the SSI method

Modal Assurance Criterion, MAC is shown in table 3 for 4 modes. Modes at 43 Hz and 68 Hz has a MAC value of 0,74 since they both are dominated by rolling behaviour. Modes at 20 Hz and 210 Hz also shows a high MAC value although it is clear in the animation that one is a rigid body mode and the other is a bending mode. This

may be due to undersampling of the structure and also the fact that the gearbox is in a test rig, not in free-free condition.

MAC	20 Hz	43 Hz	68 Hz	210 Hz
20 Hz	1	0,003	0,057	0,48
43 Hz	0,003	1	0,74	0,0012
68 Hz	0,057	0,74	1	0,007
210 Hz	0,48	0,0012	0,007	1

Table 3, Modal Assurance Criterion, MAC

CONCLUSION

The presence of dominant harmonic components in the measured responses is unavoidable in many applications of Operational Modal Analysis. The consequences can be quite drastic and can result in significantly biasing of the modal parameters obtained or even prohibit the extraction of the structural modes.

This paper has using some recent method based on the original EFDD technique, where the harmonic components are identified using a Kurtosis Check method and then removed by first performing a linear interpolation across the harmonic components in the SDOF function and secondly applying a curve-fitting using the CFDD technique. Also the SSI method has been used by adjusting the tolerances for the minimum allowable damping for the mode or harmonic indication in the stabilization diagram. In this case the SSI method seems to be most informative/useful for mode identification in combination with the harmonic indicator, which is part of the FDD/EFDD/CFDD methods. It is also confirmed, as stated in ref. [8], that the SSI method cannot cover the full frequency range in a single analysis. Different degree of decimation as well as filtering is needed in order to reveal all the poles (structural, harmonics, noise and computational modes) in the data.

The methods have been demonstrated on measurements a gearbox of Wind Turbine Wing. The gearbox was mounted in a test rig. Four modes were found with sufficient confidence at 19,7 Hz (primary rigid body mode, yaw), 42 Hz, 67 Hz (primary rigid body mode, roll) and 210 Hz (elastic mode, bending). In this case it is not enough to rely on automatic procedures. The human interaction was necessary in order to get meaningful results.

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This paper was presented at:

IMAC XXVII, A Conference and Exposition on Structural Dynamics

Orlando, Florida USA (February 9 – February 12, 2009)

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