Frequency Domain Techniques for Operational Modal Analysis

Svend Gade, Nis B. Møller, Henrik Herlufsen, Hans Konstantin-Hansen
Brüel & Kjær Sound and Vibration Measurements A/S,
Skodsborgvej 307, DK2850 Nærum, Denmark

ABSTRACT
Operational Modal Analysis, also known as Output Only Modal Analysis, has in the recent years, been used for determination of modal parameters of civil engineering structures and is now becoming widely used also for mechanical structures. The advantage of the method is that no artificial excitation needs to be applied to the structure, or force signals to be measured. In this paper, the non-parametric technique based Frequency Domain Decomposition (FDD), as well as the more elaborate Enhanced Frequency Domain Decomposition (EFDD) identification technique are discussed. The methods are illustrated by measurements on a wing from a wind turbine acoustically excited by a loudspeaker in the Brüel & Kjær laboratory.

INTRODUCTION
Operational Modal Analysis, also known as Ambient Modal Analysis and Output Only Modal Analysis is presented here. Traditionally, measuring the input forces and the output responses for a considered linear, time-invariant mechanical system performs a modal test of a structure.

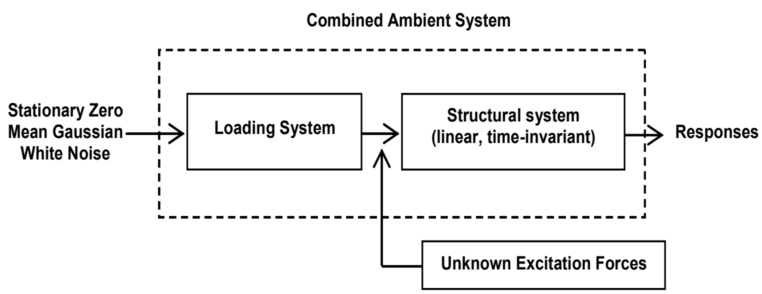

***Figure 1**, Combined Ambient Model*

The excitation usually used is transient (Impact Hammer testing), or random, burst-random, or sinusoidal (Shaker testing). The advanced signal processing tools used in the Operational Modal Analysis technique allow us now to determine the inherent properties of a mechanical structure (Resonance Frequencies, Damping, Mode Shapes), by measuring only the response of the structure, without using an artificial excitation. The advantage of this technique is that it provides a complete modal model under operating conditions, meaning within true boundary conditions, and actual force and vibration levels. The measurement technique is similar to the "Operating Deflection Shapes" type procedure, where one or more accelerometers are used as reference(s), and a series of roving accelerometers are used for the responses at all the Degrees of Freedom (DOF's), or all DOF's are just measured simultaneously. Figure (1) shows a schematic description of an ambient response system.

The measurements were taken using the Brüel & Kjær PULSE™ Multi-analyzer system (Type 3560), and the Modal Test Consultant™ (Type 7753) to create the geometry, assign the measurement points and directions (DOF's), and capture the data. The analysis was then done using the Brüel & Kjær Operational Modal Analysis™ software (Type 7760), where all the advanced signal processing and modal extraction procedures were performed.

MEASUREMENT PROCEDURE
In this paper the use of the Operational Modal Analysis method for a 1:5 scale wind turbine wing is described. The object is a detailed model of one of the blades from a 675 kW wind turbine. The wing has been made for lab investigations of static as well as dynamic parameters. Figure (2) shows a picture of the set-up used for the measurements. The wing itself is supported by a console which is regarded as stiff compared to the wing itself.

24 accelerometers Brüel & Kjær Type 4507 B4 with a sensitivity of 10mV/ms^{-2} are mounted in two rows along the wing. Two time recordings were taken, one with the accelerometers perpendicular to the surface (Z-direction) and one pointing in the direction of wing rotation (X-direction). To determine the combined modes in this two-dimensional model the results from the two recordings are linked together using a tri-axial accelerometer Type 4506 with a sensitivity of 10mV/ms^{-2} as a reference. The wing is considered as stiff in the length direction, Y, so vibrations in this direction are disregarded. Thus Type 4506 was used as a biaxial accelerometer. The total numbers of DOF's are 50. The data acquisition system used was a portable PULSE™ platform Type 3560D composed of a 31 channel front-end for the hardware, and a laptop computer for the software.

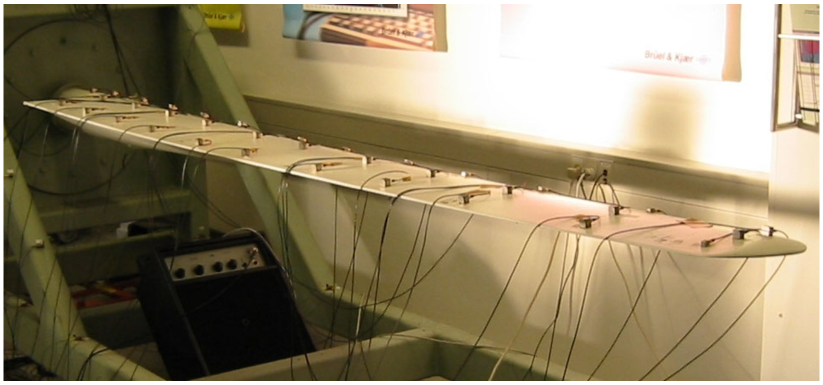

***Figure 2**, Test object, 1:5 scale wind turbine wing*

The wing was exposed to an acoustic excitation by means of a loudspeaker Brüel & Kjær Type 4224 placed beneath the wing. Although the aim is to use Operational Modal Analysis for the test the sound level was measured using a microphone placed in front of the wing and analysed as well. This measurement was done to ensure presence of energy in the frequency range of interest. The spectral distribution does not need to be flat so white noise is not a requirement.

The PULSE system was set up with a Time Capture Analyzer with a frequency range of 200 Hz (sampling frequency of 512 Hz, Nyquist frequency of 256 Hz) corresponding to a sampling interval of 1.953 ms. A record length of 60 s (i.e. 30720 samples) was chosen in order to capture 600 cycles of the lowest frequency of interest, 10 Hz.

All the raw time data, the geometry, and the series of measurements are then directly exported from PULSE to the Brüel & Kjær Operational Modal Analysis™ software for advanced signal processing calculations, and modal extraction.

SIGNAL PROCESSING AND FREQUENCY DOMAIN DECOMPOSITION

The first step of the analysis is to perform a Discrete Fourier Transform (DFT) on the raw time data, to obtain the Power Spectral Density Matrices that will contain all the frequency information. The excitation being broadband and having a continuous type of spectrum, the proper frequency descriptor is the Power Spectral Density (in $(m/s^2)^2/Hz$), that normalizes the measurement with respect to the Noise Bandwidth of the band-pass filters (FFT). Hanning Weighting with 66.6% overlap is used.

In general, for all the series of measurements or datasets, we now have the Spectral Density Matrices calculated. The size of the matrix is n*n, n being the number of transducers (26 in our case - 26 measured DOF's). In this example we have two data sets, i.e. two matrices (of a size 26*26) were calculated for each frequency. Each element of those matrices is a Spectral Density function. The diagonal elements of the matrix are the real valued Spectral Densities between a response and itself (Auto Power Spectral Density). The off-diagonal elements are the complex Cross Spectral Densities between two different Responses. All those matrices are Hermitian (they are complex conjugate symmetric, meaning that they have complex conjugate elements around the diagonal).

FREQUENCY DECOMPOSITION THEORY BACKGROUND

The Frequency Domain Decomposition (FDD) is an extension of the Basic Frequency Domain (BFD) technique, or more often called the Peak-Picking technique. This approach uses the fact that modes can be estimated from the spectral densities calculated, in the condition of a white noise input, and a lightly damped structure. It is a non-parametric technique that estimates the modal parameters directly from signal processing calculations, Refs. [1,2]. The FDD technique estimates the modes using a Singular Value Decomposition (SVD) of each of the data sets. This decomposition corresponds to a Single Degree of Freedom (SDOF) identification of the system for each singular value. The relationship between the input x(t), and the output y(t) can be written in the following form, Refs. [3,4]:

$$\left[G_{yy}(\omega)\right]=\left[H(\omega)\right]^{*}\left[G_{xx}(\omega)\right]\left[H(\omega)\right]^{T} \quad , \tag{1}$$

where $[G_{xx}(\omega)]$ is the input Power Spectral Density (PSD) matrix. $[G_{yy}(\omega)]$ is the output PSD matrix, and $[H(\omega)]$ is the Frequency Response function (FRF) matrix, and * and superscript T denote complex conjugate and transpose, respectively. The FRF matrix can be written in a typical partial fraction form (used in classical Modal analysis), in terms of poles, λ and residues, R

$$\left[H(\omega)\right]=\frac{\left[Y(\omega)\right]}{\left[X(\omega)\right]}=\sum_{k=1}^{m}\frac{\left[R_k\right]}{j\omega-\lambda_k}+\frac{\left[R_k\right]^{*}}{j\omega-\lambda_k^{*}} \quad , \tag{2}$$

with

$$\lambda_k=-\sigma_k+j\omega_{dk} \ , \tag{3}$$

where m being the total number of modes of interest, λ_k being the pole of the k^{th} mode, σ_k the modal damping (decay constant) and ω_{dk} the damped natural frequency of the k^{th} mode.

Using the expression (1) for the matrix $[G_{yy}(\omega)]$, and the Heaviside partial fraction theorem for polynomial expansions, we obtain the following expression for the output PSD matrix $[G_{yy}(\omega)]$ assuming the input is random in both time and space and has a zero mean white noise distribution, i.e. its PSD is a constant, i.e. $[G_{xx}(\omega)] = [C]$:

$$\left[G_{yy}(\omega)\right]=\sum_{k=1}^{m}\frac{\left[A_k\right]}{j\omega-\lambda_k}+\frac{\left[A_k\right]^{*}}{j\omega-\lambda_k^{*}}+\frac{\left[B_k\right]}{-j\omega-\lambda_k}+\frac{\left[B_k\right]^{*}}{-j\omega-\lambda_k^{*}} \tag{4}$$

Considering a lightly damped model and that the contribution of the modes at a particular frequency is limited to a finite number (usually 1 or 2), then the response spectral density matrix can be written as the following final form:

$$\left[G_{yy}(\omega)\right]=\sum_{k\in Sub(\omega)}\frac{d_k\psi_k\psi_k^{T}}{j\omega-\lambda_k}+\frac{d_k^{*}\psi_k^{*}\psi_k^{*T}}{j\omega-\lambda_k^{*}} \tag{5}$$

where $k \in Sub(\omega)$ is the set of modes that contribute at the particular frequency and where ψ_k is the mode shape and d_k is a scaling factor for the k^{th} mode.

This final form of the matrix is then decomposed, using the SVD technique, into a set of singular values and their corresponding singular vectors, the singular vectors being an approximation to the mode shapes. This decomposition is performed to identify Single Degree of Freedom Models to the problem

Another way to understand the response signals, y(t) is from their decomposition into participations from the different modes $[\Phi]$ expressed via the modal coordinates $\mathbf{q}(t)$:

$$\mathbf{y}(t) = [\Phi]\mathbf{q}(t) \tag{6}$$

Using eq. (6) in the expression of the correlation matrix, $[C_{yy}(\tau)]$ of the responses we get:

$$[C_{yy}(\tau)] = E\{\mathbf{y}(t+\tau)\mathbf{y}(t)^{T}\} = E\{[\Phi]\mathbf{q}(t+\tau)\mathbf{q}(t)^{H}[\Phi]^{H}\} = [\Phi][C_{qq}(\tau)]\,[\Phi]^{H} \tag{7}$$

Applying the Fourier transform in eq. 7 gives:

$$[G_{yy}(\omega)] = [\Phi][G_{qq}(\omega)]\,[\Phi]^{H} \tag{8}$$

where $[G_{qq}(\omega)]$ is the spectrum matrix of the modal coordinates.

The FDD technique is based upon the SVD of the Hermetian response spectrum matrix at each frequency and for each measurement (data set):

$$\left[G_{yy}(\omega)\right] = [V]\,[S]\,[V]^H \tag{9}$$

where $[S]$ is the singular value diagonal matrix and $[V]$ is the orthogonal matrix of the singular vectors. The singular vectors (the columns in $[V]$) are orthogonal to each.

Eq. (9) has the same form as eq. (8) and it can be understood that the singular vectors present estimations of the mode shapes and the corresponding singular values present the response of each of the modes (SDOF systems) expressed by the spectrum of each modal coordinate The assumptions are that $[G_{qq}(\omega)]$ is a diagonal matrix, i.e. the modal coordinates are uncorrelated, and that the mode shapes (the columns in $[\Phi]$) are orthogonal.

SINGULAR VALUE DECOMPOSITION

The Singular Value Decomposition of an m*n complex matrix A is the following factorization:

$$A = U \Sigma V^H \tag{10}$$

Where U and V are unitary matrices, Σ is a diagonal matrix that contains the real singular values:

$$\Sigma = diag(s_1, \ldots\ldots, s_r) \tag{11}$$

$$r = \min(m, n) \tag{12}$$

The superscript H on the matrix V denotes a Hermitian transformation (Transpose and Complex Conjugate). In the case of real valued matrices, the V matrix is only transposed. The S_i elements in the matrix Σ are called the singular values, and their corresponding singular vectors are contained in the matrices U and V.
This singular value decomposition is performed for each dataset at each frequency. The spectral density matrix is then approximated to the following expression after SVD decomposition:

$$\left[G_{yy}(\omega)\right] = [\Phi]\,[\Sigma]\,[\Phi]^H \tag{13}$$

$$\text{with } [\Phi]^H [\Phi] = [I] \tag{14}$$

where Σ being the singular value matrix, and Φ the singular vectors unitary matrix:

$$[\Sigma] = diag(s_1, \ldots\ldots, s_r) = \begin{bmatrix} s_1 & 0 & 0 & . & . & 0 \\ 0 & s_2 & 0 & . & . & . \\ 0 & . & s_3 & . & . & . \\ . & . & . & . & . & 0 \\ . & . & . & . & s_r & 0 \\ 0 & . & . & 0 & 0 & 0 \end{bmatrix} \tag{15}$$

$$[\Phi] = [\{\phi_1\} \quad \{\phi_2\} \quad \{\phi_3\} \quad . \quad . \quad .\{\phi_r\}] \tag{16}$$

where ϕ_i are approximations of the individual mode shapes. The number of non-zero elements in the diagonal of the Singular matrix corresponds to the rank of each spectral density matrix.

As mentioned earlier, the singular vectors in eq. (16) correspond to an estimation of the Mode Shapes, and the corresponding singular values are the Spectral Densities of the SDOF system expressed in eq. (5).

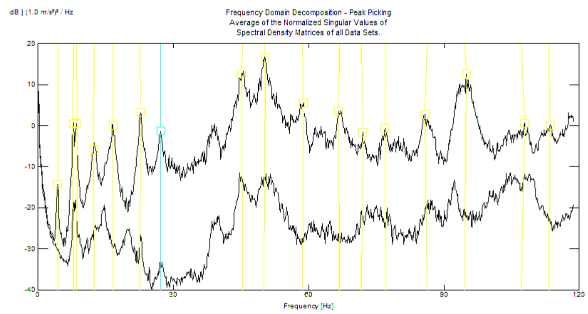

***Figure 3**, Peak-Picking on the average of the normalized singular values of the PSD Matrices for all data sets*

Figure (3) shows the averaged result of the Singular Value Decomposition of the two data sets. We obtain 26 singular values, and 26 singular vectors for each frequency, for simplicity only the two most significant singular values are shown. The singular values and their corresponding singular vectors are ordered in singular value descending order for each of the data sets, meaning the first singular value will be the largest. By peak-picking, applied to he first singular values, 19 significant modes were identified in the frequency range from 4 Hz to 130 Hz.

This technique allows us to identify possible coupled modes that are often indiscernible as they appear on the individual frequency spectra. If only one mode is dominating at a particular frequency, then only one singular value will be dominating at this frequency. In the case of close or repeated modes, there will be as many dominating singular values as there are close or repeated modes.

ENHANCED FREQUENCY DOMAIN DECOMPOSITION

The Enhanced FDD technique allows us to extract the resonance frequency and the damping of a particular mode by computing the auto, and cross-correlation functions. The SDOF Power Spectral Density function, identified around a peak of resonance, is taken back to the time domain using the Inverse Discrete Fourier Transform (IDFT). The resonance frequency is obtained by determining the zero crossing times, and the damping by the logarithmic decrement of the corresponding SDOF normalized auto correlation function. Both parameters and an improved version of the mode shapes are estimated from the SDOF Bell functions. The SDOF Bell function is estimated using the shape determined by the previous FDD peak picking. The latter being used as a reference vector in a correlation analysis based on the Modal Assurance Criterion (MAC). A MAC value is computed between the reference FDD vector and a singular vector for each particular frequency line. If the MAC value of this vector is above a user-specified MAC Rejection Level, the corresponding singular value is included in the description of the SDOF Spectral Bell function. The lower this MAC Rejection Level is, the larger the number of singular values included in the identification of the SDOF Bell function will be.

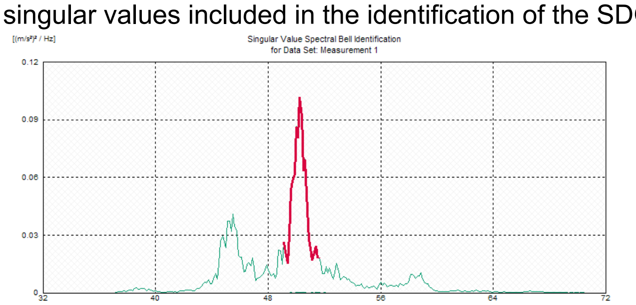

***Figure 4**, Singular Value Spectral Bell identification from one of the Data sets. Linear Y-axis and an expanded frequency axis is used*

Figure (4) shows the estimated SDOF Bell function from data set no. 1, for the most dominating mode found at 50.2 Hz, with a MAC rejection level of 0.91. The value of the MAC rejection criteria has to be adjusted so we obtain a good representation of the Bell function around the chosen peak, and not include any noise around it, often present in an Ambient Modal analysis. Using this SDOF Bell function, we perform an inverse Fourier Transform for the determination of the damping and the natural frequency. The obtained normalized correlation function is shown in Figure (5).

We can see that we have a typical response of a resonating system that decays exponentially. The scattered region indicates the part of the correlation function that is used for the frequency and damping estimation algorithm. The damping is estimated by the logarithmic decrement technique from the logarithmic envelope of the correlation function. The estimation is performed by using a linear regression technique (see figure (6)). The resonance frequency is simply obtained by counting the number of times the correlation function crosses the zero axis per second. The mode shape is estimated as a weighted average of the singular vectors in the user-specified MAC rejection level interval, the MAC values being the weighting factors.

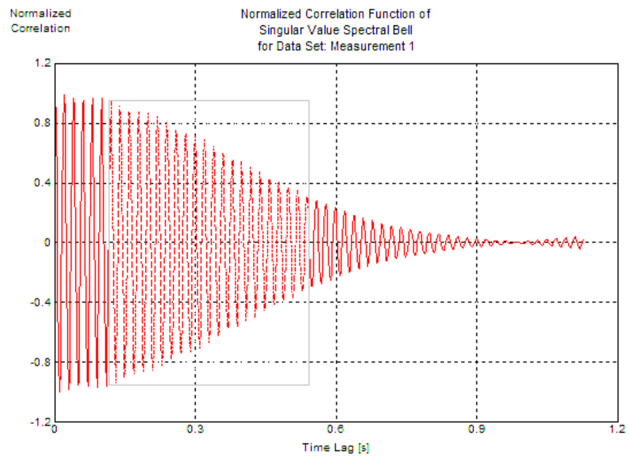

__Figure 5__, Normalized correlation function for mode at 50.2 Hz from one of the Data sets

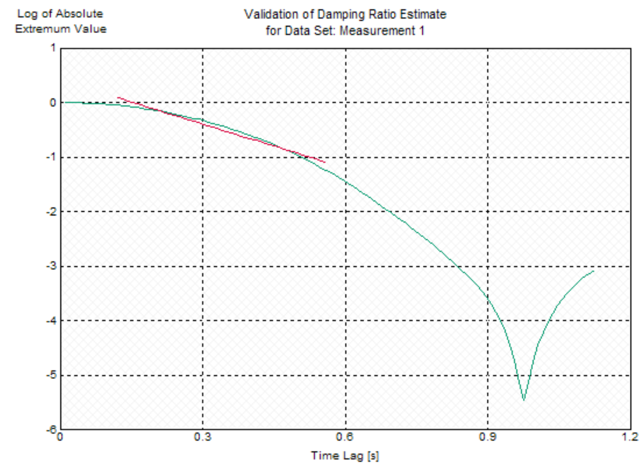

__Figure 6__, Damping Ratio Estimation from the decay curve of the correlation function

It is important to note that the Bell functions, the estimation of the damping and natural frequency is performed for each set of measurement. The final result is then obtained by averaging the results from all the data sets together. Both the average values as well as the standard deviations (std.) are calculated from the data sets. For the most dominating mode the following results were found: Natural frequency 50.2 Hz with a std. of 0.08 Hz. Damping of 1.2 % with a std. of 0.5 %.

USE OF PROJECTION CHANNELS

In the case where a large number of response DOF’s are measured simultaneously (i.e. measurement setups with large channel counts) the parametric model fit suffers from the estimation of many noise modes, compared to the number of physical modes of the system. The main reason for this is that the many channels contain the same physical information but different random errors. A way to reduce the amount of noise modes is therefore to reduce the number of channels in the actual estimation process. The information of the physical modes must not be affected and the selected channels must represent the system.

A simple measure of the amount of information of a measurement channel compared to the other channels can be established from calculation of the correlation coefficients between the different measurement channels.

The first step is to find the channel that correlates most with all the other channels. This channel most likely contains maximum physical information.

The second step is to find the remaining number of requested projection channels. These are found by similar search of the correlation coefficient matrix, as channels that correlate the least with all previously found projection channels. These channels will most likely bring maximum of new information. The only pitfall here is if a channel is dead and only contains noise. In such a case it will have an insignificant correlation with the other channels. To prevent this lower threshold of allowed correlation should be applied.

In case of multiple data sets the first step above is excluded. Instead the user-defined reference channels are applied as initial projection channels. The assumption is that all the modes are sufficiently present in the reference DOF´s. The remaining step is as described above.

Use of projection channels decreases the amount computation time is since the matrix operations are simplified significantly when having many measurement channels. Only a few columns rather than the full output cross-spectrum matrix, $[Gyy(\omega)]$ are used for the decomposition. Proper choice of number of projection channels is found from the initial SVD calculations.

The use of projection channels reduces the SVD-equation (9) to

$$[G_{yy}(:,[p_1\ p_2\ldots])] = [V]_{NxM}[S]_{MxM}\,[V]^{H}_{MxM} \tag{17}$$

where p1, p2.. are the projection channels, N is the number of measurement channels and M is the number of projection channels. The mode shapes are found from the matrix, $[V]_{NxM}$.

MEASUREMENT RESULTS

The wing typically behaves like a cantilever beam with one rigid body mode and 18 elastic body modes found in selected frequency range. Edgewise as well as flap wise bending modes were found up to around 40 Hz. Modes # 7 & 8 were torsional modes. The higher modes were rather complex and typically a mixture of bending and torsion. The frequency and damping results are summarized in Table 1.

Mode	Frequency [Hz]	Std. Frequency [Hz]	Damping Ratio [%]	Std. Damping Ratio [%]
Mode 1	4.41	0.09	2.79	0.83
Mode 2	7.83	0.003	1.37	0.10
Mode 3	8.55	0.0002	1.39	0.56
Mode 4	12.52	0.17	4.04	1.49
Mode 5	16.66	0.01	3.03	0.03
Mode 6	22.71	0.04	1.13	0.06
Mode 7	27.10	0.06	2.54	1.48
Mode 8	45.38	0.07	2.13	0.09
Mode 9	50.25	0.08	1.23	0.54
Mode 10	58.65	0.90	1.78	0.49
Mode 11	66.81	0.58	1.19	0.37
Mode 12	71.72	0.72	1.30	0.38
Mode 13	77.03	0.76	1.23	0.53
Mode 14	86.21	0.36	0.86	0.32
Mode 15	94.51	0.11	1.23	0.002
Mode 16	107.2	0.87	0.99	0.18
Mode 17	113.3	0.03	0.94	0.37
Mode 18	119.1	0.15	1.11	0.50
Mode 19	128.4	0.29	1.43	0.80

***Table 1**, Overview of natural frequencies, dampings and their standard deviations*

Figure 7 shows three examples of the extracted mode shapes. Fig 7a show the first bending mode (mode # 2 in table 1) with a node line at the fixing console. Natural frequency is 7.8 Hz and the damping is 1.4 %. Fig 7b show the second bending mode (mode # 5 in table 1) with a node line approximately midway between the fixing console and the free end. Natural frequency is 16.7 Hz and the damping is 3.0 %. Fig 7c show one of the torsional modes (mode # 9 in table 1). Natural frequency is 50.2 Hz and the damping is 1.2 %.

Validation of modal parameters can be done in several ways. It is recommended that modal parameters are estimated using different techniques and their shapes compared using AutoMAC and CrossMAC calculations, where MAC is Modal Assurance Criterion. If different techniques yield similar results it is very likely that a physical mode shape is found. If only one estimation technique is used, like in this paper only AutoMAC calculations are available and it may be useful to use other validation tools such as complexity-plots and phase-scatter.

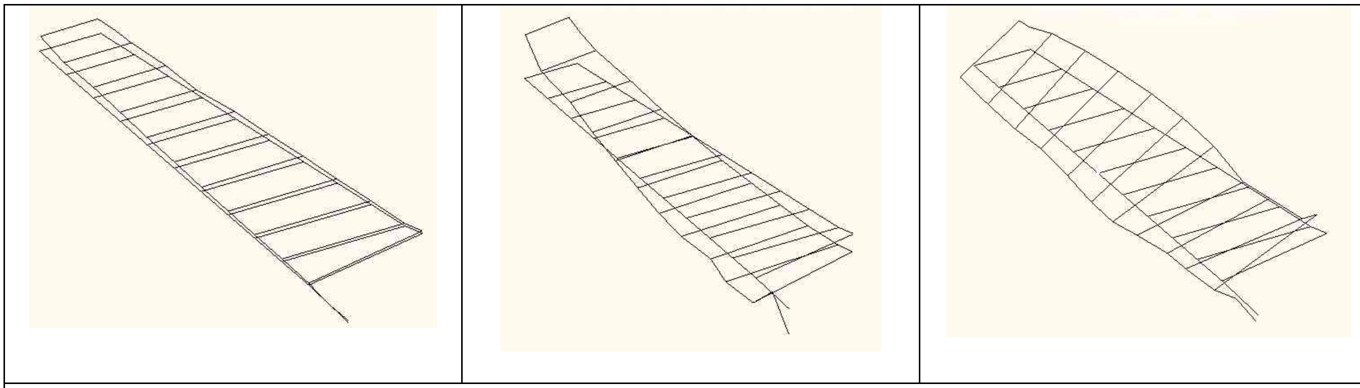

***Figure 7**, **a)** bending 7.8Hz (#2),* ***b)** bending 16.6 Hz (#5),* ***c)** torsion 50.3 Hz (#9)*

Table 2 shows (due to limited space on the page) a part of the AutoMAC table. In general, most of the off-diagonal MAC values are small except for the MAC values between Mode #3 & #4 (MAC=0.66), which is first bending in X-Z direction and between Mode #5 & #6 (MAC=0.63), which is second bending in Z-direction and between Mode #11 & #12 (MAC=0.65), which is torsion with 3 node lines. For all 3 sets of mode shapes it is found that the node lines are found at slightly different locations. See also Refs. [8,9] for more information about Modal Assurance Criterion.

AutoMAC		Mode # 1	Mode # 2	Mode # 3	Mode # 4	Mode # 5
	Frequency	**4,41 Hz**	**7,83 Hz**	**8,55 Hz**	**12,51 Hz**	**16,63 Hz**
Mode # 1	**4,41 Hz**	**1**	0,13	0,58	0,37	0,01
Mode # 2	**7,83 Hz**	0,13	**1**	0,10	0,14	0,37
Mode # 3	**8,55 Hz**	0,58	0,10	**1**	0,66	0,03
Mode # 4	**12,51 Hz**	0,37	0,14	0,66	**1**	0,27
Mode # 5	**16,63 Hz**	0,01	0,37	0,03	0,27	**1**
Mode # 6	**22,66 Hz**	0,00	0,52	0,04	0,25	0,63
Mode # 7	**27,09 Hz**	0,03	0,01	0,01	0,05	0,45
Mode # 8	**45,37 Hz**	0,02	0,07	0,01	0,05	0,17
Mode # 9	**50,16 Hz**	0,11	0,17	0,02	0,01	0,16
Mode # 10	**58,64 Hz**	0,00	0,00	0,00	0,02	0,11

***Table 2**, Part of the AutoMAC table*

Complexity plots may also be very useful in order to check if we have normal shapes or not. A shape is called a "normal" shape if all of its shape components have 0 or 180 degrees of phase relative to one another. In other words, all of the shape components lie along a straight line in the Complexity Plot. A "complex" shape can have arbitrary phases in its shape components, so they will not lie along a straight line. When animated, a normal shape will look like a standing wave, and its Nodal Lines will not move. A complex shape will look like a traveling wave, and its nodal lines will move. Figure 8 shows 3 examples of complexity plots from the measurement data.

Complexity plot for mode # 2 at 7.8 Hz shows the results to be nearly a normal shape. For modes # 11 at 66.8 Hz and # 12 at 71.7 Hz we have two of the shapes with a high MAC value – the complexity plots indicates that mode # 11 has a fairly normal shape, while mode shape # 12 is rather complex.

In general shapes # 2, 4, 5, 11 & 15 show low complexity, shapes # 1, 6, 7, 8, 9 & 10 show medium complexity and shapes # 3, 12 – 14 & 16 – 20 show high complexity.

The results has in other papers been verified by using other techniques such as time domain methods as well as classical mobility based modal analysis, see Refs. [5, 6 & 7]. Good agreement has been obtained with respect to both frequencies as well as mode shapes, although discrepancies up to 100% were observed on the damping values. It should be mentioned that most modes were identified using all 3 techniques, while some other modes were only found using one or two of the techniques.

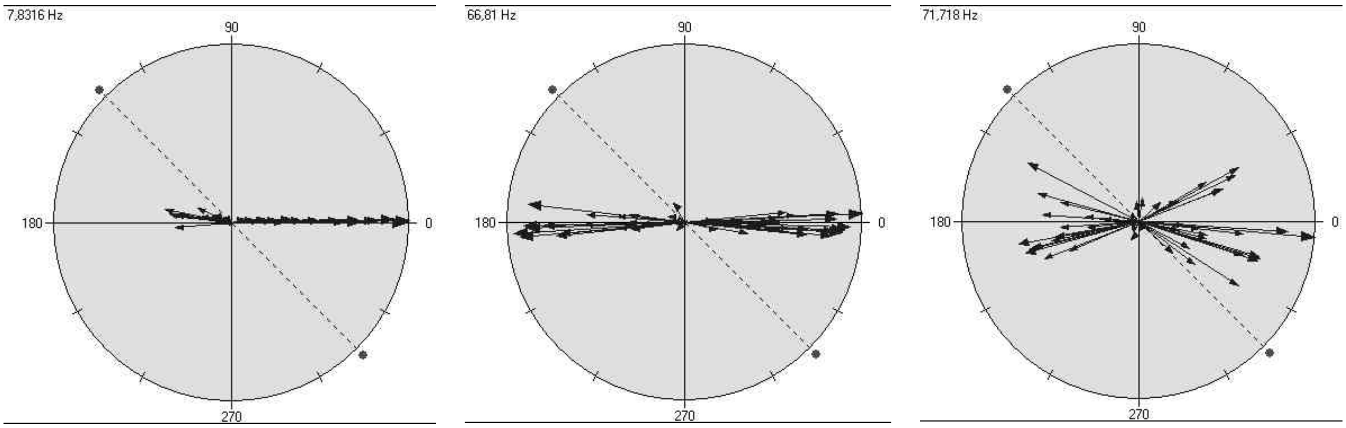

***Figure 8**, Complexity plots for modes # 2, # 11 and # 12*

CONCLUSION

The Brüel & Kjær Operational Modal Analysis, OMA software is an extremely powerful tool that allows the scientist, technician or engineer to elaborate a modal testing procedure very easily, quickly, and accurately. Only measuring the response of the structure subjected to unknown and unmeasured input forces can perform a complete modal determination. The Frequency Domain Decomposition, FDD technique, which is a non-parametric technique, do not rely on any type of curve fitting based on parametric modeling, was explained and applied to a wind turbine wing. It provided very accurate results for natural frequencies and mode shapes. The Enhanced Frequency Domain Decomposition, EFDD technique, which is using simple linear regression curvefitting techniques was performed for damping factors determination, and was proven to be extremely reliable, and accurate. Both techniques can lead to very accurate results even for weakly excited modes, closely spaced modes, and large number of channels and large amount of data.

The advantages of the FDD are that the technique is easy and fast to use. There is a “snap to peak” feature that can be applied on the Averaged Normalized Singular Value function. The corresponding singular vector, which is an approximation to the mode shape, is extracted from all dataset at the selected frequency. In fact, the singular vectors can be extracted from any singular value at any frequency, which may lead to better understanding of the structural behaviour. The disadvantages are that no damping is estimated and the frequency resolution is no better than the FFT line spacing.

The main advantages of the EFDD technique are that both Frequency and Damping are estimated. The “snap to peak” is applied to the maximum Singular Values for each dataset. Then the averaged frequency and damping values are estimated as well as their standard deviations from all data sets. Major disadvantage is that the algorithm does not work properly if no distinguished peak is found in some of the data sets. It is also required to fine tune the three modal estimation parameters, MAC Rejection Level, maximum and minimum correlation (i.e. correlation interval) for each resonance frequency in each dataset, which may be a very time consuming procedure, when there are many data sets and/or many modal frequencies present.

The Operational Modal Analysis technique provides efficient estimation of modal models, without test rigs, shakers, or even a laboratory. It perfectly suits applications for large structures, very small, or operating structures. It offers a model under real boundary conditions, based on distributed and multiple inputs.

The current implementation only provides unscaled mode shapes, which means that simulations such as Structural Dynamic Modifications, SDM and Forced Response Simulations, FRS are not possible. This may be a minor problem if the OMA is used for refinement of Finite Element Models, FEM. Then simulation may be performed using the FEM model. In Refs. [10,11,12] possibilities for obtaining scaled mode shapes from OMA measurement are described. These are based on the fact that when we have a scaled (calibrated) modal model then mass modifications can be simulated and frequency shifts of natural frequencies can be predicted. For OMA we may make tests with different (but known) mass loadings, which yield (slightly) different natural frequencies but similar mode shapes. From these test results a scaled modal model may be obtained.

It is a challenge for most engineers to use a new technique such as OMA, thus there is a general lack of experience of knowing how to optimize the various analysis and modal parameter estimation parameters. This includes choice of frequency resolution (FFT line spacing) for the FDD method. In addition for the EFDD method the modal estimation parameters, MAC rejection level and the maximum/minimum correlation factor must be fine tuned for the SDOF Bell identification. For other methods like the Stochastic Subspace Identification, SSI method, [7] there is also a range of parameters to chose, where the main choice is the method itself: a) Principal Component, PC, b) Unweighted Principal Component, UPC or c) Canonical Variate Analysis, CVA, and no proper guidelines exist in the literature. For these methods there is a wide choice of parameters to select and adjust: State Space order, deviation of frequency, damping, Mode Shape MAC and modal amplitude MAC between consecutive models as well as the expected range of realistic damping. Finally there is the choice of number of projection channel in order not to calculate the full power spectral density matrices when using a large number of simultaneous measurement channels. The solution for some of these choices is discussed in Ref. [6].

The variety of potential applications for Operational Modal Analysis is very large and diverse: Automotive industry: On-road testing for cars, and motorcycles. Testing of sub-components and body-structures in automotive. Earth-moving equipment and trains may be tested using this method. Aerospace engineering: in-flight testing of aircraft structures. Rotating machinery analysis: turbines, generators, compressors, drilling machines, and transmissions. Civil Engineering and maritime: shipbuilding, platforms, towers, buildings, and bridges.

REFERENCES

[1] Brincker, R., L. Zhang and P. Andersen: "Modal Identification from Ambient Responses using Frequency Domain Decomposition, Proc. of the 18th International Modal analysis Conference, San Antonio, Texas, February 7-10, 2000

[2] Brincker, R., Andersen, P. And N. Møller: "Output-only Modal Testing of a car body subject to Engine Excitation", Proc. Of the 18th International Modal Analysis Conference, San Antonio, Texas, February 7-10, 2000

[3] Bendat, Julius S. and Allan G. Piersol: "Random Data, Analysis and Measurement Procedures", John Wiley and Sons, 1986

[4] Brincker, R. and Andersen, P.: "ARMA (Auto Regressive Moving Average) Models in Modal Space", Proc. Of the 17th International Modal Analysis Conference, Kissimmee, Florida, 1999

[5] Herlufsen, H. and Møller, N.: "Operational Modal Analysis of a Wind Turbine Wing using Acoustical Excitation", Brüel & Kjær Application Note, BO 0500-11

[6] Herlufsen, H., Andersen, P., Gade, S. and Møller, N.: "Identification Techniques for Operational Modal Analysis – an Overview and practical experiences", Proceedings of 1st IOMAC Conference, Copenhagen, Denmark, April 26-27, 2005

[7] Møller, N. B., Herlufsen, H. and Gade, S.: "Stochastic Subspace Identification Techniques in Operational Modal Analysis", Proceedings of 1st IOMAC Conference, Copenhagen, Denmark, April 26-27, 2005

[8] Allemang, R.J.: "The Modal Assurance Criterion (MAC): 20 Years of Use and Abuse", Proc. of the 20th International Modal Analysis Conference, Los Angeles, California, 2002

[9] Ewins, D.J.: "Modal Testing: Theory, Practise and Applications", Research Studies Press, Baldock, England, 2000, p. 422 - 437

[10] Brincker, R.: "A Way of Getting Scaled Mode Shapes in Output Only Modal Analysis", Proc. of the 21th International Modal Analysis Conference, Kissimmee, Florida, 2003

[11] Parloo, E., Verboven, P., Guillaume, P., Van Overmeire, M.: "Sensitivity-Based Operational Mode Shape Normalization", In Mechanical Systems and Signal Proc., 2002, 16(5), p. 757-767

[12] Aennlle, M. L.,Brincker, R.,Canteli,A.F.: "Some Methods to Determine Scaled Mode Shapes in Natural Input Modal Analysis", Proc. of the 23th International Modal Analysis Conference, Orlando, Florida, 2005