Output-only Modal Analysis of a Beam Via Frequency Domain Decomposition Method Using Noisy Data

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P A P E R I N F O

Paper history:
Received 04 March 2019
Received in revised form 23 September 2019
Accepted 07 November 2019

Keywords:
Frequency Domain Decomposition
Modal Parameters
Noise Level
Output-only Modal Analysis
Simply Supported Beam

A B S T R A C T

The output data from a structure is the building block for output-only modal analysis. The structure response in the output data, however, is usually contaminated with noise. Naturally, the success of output-only methods in determining the modal parameters of a structure depends on noise level. In this paper, the possibility and accuracy of identifying the modal parameters of a simply supported beam in the presence of noise has been discussed. The output-only modal analysis method with frequency domain decomposition was used and output data with various noise levels were considered. Initially, finite element modal analysis was used to determine the modal parameters for the beam which were afterwards enforced as the reference modal parameters. Then, appropriate input was applied to the beam and the acceleration signals of different nodes were produced through finite element transient analysis. In order to simulate noisy data, noises with different power levels were generated and added to the signals. Finally, the modal parameters were obtained by frequency domain decomposition method. The results showed that the modal parameters corresponding to the first vibration mode could only be identified with acceptable validity at low to moderate noise levels, whereas for higher modes, the modal parameters can be correctly obtained even at high noise levels.


1. INTRODUCTION

Modal parameters of structures, such as natural frequencies, damping ratios and mode shapes, have been used vastly in the past for numerous civil engineering applications. In order to obtain these parameters experimentally, there are two groups of modal analysis methods: input-output methods and output-only methods. Output-only modal analysis methods are mostly used due to their numerous advantages in comparison with other methods [1, 2]. One of the most common methods of output-only modal analysis is Frequency Domain Decomposition (FDD) which was proposed by Brincker [3]. In FDD method, matrices of Power Spectral Density (PSD) are formed with output data, after that the mode shapes and natural frequencies are extracted via Singular Value Decomposition (SVD) [4-6].

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employed a dynamic noise remover on the basis of the online wavelet concept for ignoring unwanted noise effects.

In spite of all these efforts, because of uncertainties in quality and quantity of noise in reality, it is not possible to entirely eliminate noise [9, 16]. Therefore, output-only methods rely on contaminated data with unknown noise level to identify modal parameters of structures. Although the effect of low levels of noise (up to 60%) were investigated in previous work [17, 18]. There was no study -to the best of our knowledge- in finding the maximum tolerable noise level; or equivalently minimum required value of signal to noise ratio (SNR) to identify the modal parameters via output-only methods. In this research, we will undertake this task. Moreover, we will evaluate the performance of the output-only method of FDD for various SNRs. For this purpose, a simply supported beam was chosen. An appropriate load was applied on the beam and acceleration signals of the beam were obtained using dynamic transient analysis. To generate noisy data, noises with different powers than signal power were added to the signals. Using these noisy data, the modal parameters of beams were obtained at various noise levels using FDD method. These parameters were compared with reference modal parameters of the beam that had been obtained by finite element modal analysis.

2. FREQUENCY DOMAIN DECOMPOSITION METHOD

Among the nonparametric methods in the frequency domain, FDD has been used widely in output-only modal analysis because of its simplicity, dependability, and efficiency [3, 19]. However, FDD always requires the prior-selected natural frequencies as well as its strict hypotheses of uncorrelated white noise excitations and low structural damping. Output PSD matrix can be decomposed via fast-decayed decomposition methods such as SVD. This way, the spectral density reveals contributions of various modes of the structure. By analyzing the singular values, the auto PSD functions conforming to individual modes of a structure can be identified. In this method, the approximated mode shapes are obtained as the singular vectors at maximum location of any auto PSD function conforming to a mode. In what follows, the theoretical basis of the FDD method is summarized [20].

The pole-residue formula of the PSD matrix of output can be written as:

\[ G_{yy}(\omega) = \sum_{k=1}^{m} \left[ \frac{\mathbf{A}_k}{\omega - \lambda_k} + \frac{\mathbf{B}_k}{\omega + j\gamma_k} + \frac{\mathbf{C}_k}{\omega - j\gamma_k} \right] \]  

where \( G_{yy}(\omega) \) is the output PSD square matrix of dimension \( n \) in which \( n \) is the number of output data; \( r \), \( j \) and \( \omega \) are number of desired modes, imaginary unit and natural frequency, respectively. [\( \mathbf{A}_k \)] and \( \lambda_k \) are the \( k \)-th residue output PSD matrix which is an \( n \times n \) Hermitian matrix and pole of the \( k \)-th mode, respectively (Superscript \(^*\) defines complex conjugate). [\( \mathbf{A}_k \)] and \( \lambda_k \) are given by:

\[ [A_k] = [R_k][C]\sum_{i=1}^{n} \left( \frac{[R_i]^T}{\omega_i - \lambda_k} + \frac{[R_i]^T}{\omega_i + j\gamma_k} \right) \]  

\[ \lambda_k = -\sigma_k + j\omega_d k \]  

where \([R_k]\) is the residue of \( k \)-th mode and equals to product of modal participation vector \([\psi_k]\) and that of mode shape \([\phi_k]\). \([C] \) is PSD matrix of input which is constant; \( \omega_d k \) and \( \sigma_k \) are natural frequency corresponding to the \( k \)-th mode of damped system and the modal damping ratio, respectively (Superscripts \( H \) and \( T \) denote complex conjugate and transpose, respectively). Considering only the \( k \)-th mode, Equation (2) turns into:

\[ [A_k] = \frac{[R_k][C][R_k]^T}{2\sigma_k} \]  

If the modal damping ratio is low, Equation (4) is dominant and the residue proportional to the mode shape vector can be achieved by the following equation:

\[ [A_k] \propto [R_k][C][R_k]^T = ([\phi_k]([\psi_k]^{\nu})^T[C][\psi_k][\phi_k]^T = z_k([\phi_k][\phi_k]^T)^T \]  

where \( z_k \) is the \( k \)-th mode scaling factor. For a system with low damping in which at a certain frequency, finite number of modes participate in the structure response, the output PSD matrix can be stated as:

\[ [G_{yy}(\omega)] = \sum_{k \in \text{Sub}(\omega)} \left( \frac{z_k([\phi_k][\phi_k]^T)^T}{\omega - \lambda_k} + \frac{z_k([\phi_k][\phi_k]^T)^T}{\omega + j\gamma_k} \right) \]  

where \( k \in \text{Sub}(\omega) \) is the collection of participating modes at desired natural frequency. The singular value decomposition of the PSD matrix of output at natural frequency \( \omega = \omega_l \) leads the following equation:

\[ [\hat{G}_{yy}(\omega_l)] = [V][\Sigma][V]^H \]  

In Equation (7) the matrices \([V]\), and \([\Sigma]\) are unitary matrix containing the singular vector \([v_i]\) and diagonal matrix of singular values \( \sigma_i \), respectively. Each mode corresponds to a local maximum in the spectrum. When there is only one dominant mode, one term in Equation (6) remains, and the power spectral density matrix approximates to a matrix with rank equal to one:

\[ \hat{G}_{yy}(\omega_l) = s_i[v_{1i}][v_{1i}]^H + \omega_k \]  

Thus, the singular vector related to the first mode \([v_{1i}] \) gives a good estimate of mode shape as follows:

\[ [\hat{\phi}] = [v_{1i}] \]  

In addition, the associated singular value is related to the auto PSD function of the SDOF system of the desired mode. Comparing the approximated mode shape \([\hat{\phi}]\)
with the singular vectors related to the frequency lines near the maximum, the auto PSD function of the equivalent SDOF system is recognized near the maximum of the singular value diagram. Every line distinguished by a singular vector which offers a Modal Assurance Criterion (MAC) value greater than a suitable MAC rejection level, is related to the PSD function of SDOF system. The corresponding PSD function of SDOF system is utilized to obtain estimates of the damping ratios and natural frequencies [21].

3. MATHEMATICAL MODEL OF NOISE

Output-only modal analysis is performed using only the measured response of structure \((NR(t))\). This response contains the structure response \((R(t))\) and the output noise \((N(t))\). The output noise consists of any undesired signals and physically may arise from measurement devices, sensors, etc. [22]. The simplest mathematical model for noise in data is the additive noise model [23]:

\[
NR(t) = R(t) + N(t)
\]

(10)

In this model, the output noise is demonstrated as a white Gaussian process with zero average [23] as shown in Equation (11) [25, 26]:

\[
N(t) = \text{RMS}(R(t)) \times \text{NL} \times W(t)
\]

(11)

where \(\text{RMS}(R(t))\) is the root mean square of signals, \(W(t)\) is a random function with the same dimension as \(R(t)\), and \(\text{NL}\) is a constant that determines the noise level. The noise level is defined as the ratio of the root mean square (RMS) of the noise process to the RMS of the noise-free data. Noise level can also be represented by Signal to Noise Ratio (SNR):

\[
\text{SNR} = \frac{1}{(\text{NL})^2}
\]

(12)

4. FINITE ELEMENT MODELING

As shown in Figure 1, a beam with simply supported condition was considered to study the noise level effect on output-only modal analysis. Finite element modeling was done in ANSYS\(^1\) software using BEAM 188 element which has six degrees of freedom at any nodes. A beam with a square cross section of 10 mm x 10 mm was modeled in 2-D space with 10 elements, each 100 mm long. The properties of material, namely elastic Young modulus, Poisson’s ratio and mass density of the beam were considered \(2 \times 10^{11}\) N/m\(^2\), 0.3 and 7850 (N.s\(^2\)/m)/m\(^3\), respectively according to typical structural steel. Geometrical properties of the beam were selected so that the vibration modes of the beam were sufficiently separated.

4.1. Finite Element Modal Analysis

The beam’s modal parameters were obtained by finite element modal analysis and were considered as its reference modal parameters. In this type of analysis, the following Eigenvalue problem is solved [27]:

\[
[K - \omega^2 M]\{\varphi\} = 0
\]

(13)

In Equation (13) matrices \(K\) and \(M\) are stiffness and mass, respectively; \(\omega\) and \(\varphi\) are Eigen-frequency and Eigenvector of structure, respectively. The reference modal parameters (natural frequencies and mode shapes) of the beam for the modes from 1\(^{st}\) to 5\(^{th}\) were calculated and have been demonstrated in Tables 1 and 2.

4.2. Applying Load and Transient Analysis

To obtain the modal parameters corresponding to the first to fifth modes of this beam using FDD method, the frequency content of the applied load should include all frequencies of these modes. For this purpose, the load shown in Figure 2(a) is considered in which the load begins from zero at the time \(t = 0\) and linearly reaches 0.3 N at 0.0006 s. It then returns to zero at 0.0012 s and then stays at zero until \(t = 1\) s. Fast Fourier Transform (FFT) of the load can be observed in Figure 2(b) where due to symmetry, only one-half is displayed. In order to assure excitation of the first five modes of the beam, the first frequency with zero FFT value must be greater than the values of natural frequency of the fifth mode. Since this frequency is reversely proportional to time step in transient analysis, the time step should be sufficiently small. Time step of 0.0001 s was considered that corresponds to first zero FFT frequency of 1667 Hz, which is greater than the natural frequency related to the fifth mode of this beam, i.e. 773.8 Hz (Table 1).

In addition to appropriate frequency content for the applied load, the load position must not be located on the nodal points of the considered modes [28]. As can be observed in Table 2, appropriate locations to excite are nodes 2, 4, 8 and 10. The load was applied to this beam on node 4 in vertical direction. Through standard transient analysis in ANSYS, the vertical acceleration of nodes 2 to 10 were obtained.

![Figure 1. Finite element model of beam with node numbers](www.ansys.com)

<table>
<thead>
<tr>
<th>Mode 1</th>
<th>Mode 2</th>
<th>Mode 3</th>
<th>Mode 4</th>
<th>Mode 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural frequency (Hz)</td>
<td>23.2</td>
<td>95.9</td>
<td>228.7</td>
<td>442.5</td>
</tr>
</tbody>
</table>

\(^1\) www.ansys.com
TABLE 2. Mode shape values of the beam obtained by finite element modal analysis

<table>
<thead>
<tr>
<th>Node Number</th>
<th>Mode 1</th>
<th>Mode 2</th>
<th>Mode 3</th>
<th>Mode 4</th>
<th>Mode 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
</tr>
<tr>
<td>2</td>
<td>0.309</td>
<td>0.618</td>
<td>-0.809</td>
<td>-1.000</td>
<td>-1.000</td>
</tr>
<tr>
<td>3</td>
<td>0.588</td>
<td>1.000</td>
<td>-0.951</td>
<td>-0.618</td>
<td>0.000</td>
</tr>
<tr>
<td>4</td>
<td>0.809</td>
<td>1.000</td>
<td>-0.309</td>
<td>0.618</td>
<td>1.000</td>
</tr>
<tr>
<td>5</td>
<td>0.951</td>
<td>0.618</td>
<td>0.588</td>
<td>1.000</td>
<td>0.000</td>
</tr>
<tr>
<td>6</td>
<td>1.000</td>
<td>0.000</td>
<td>1.000</td>
<td>0.000</td>
<td>-1.000</td>
</tr>
<tr>
<td>7</td>
<td>0.951</td>
<td>-0.618</td>
<td>0.588</td>
<td>-1.000</td>
<td>0.000</td>
</tr>
<tr>
<td>8</td>
<td>0.809</td>
<td>-1.000</td>
<td>-0.309</td>
<td>-0.618</td>
<td>1.000</td>
</tr>
<tr>
<td>9</td>
<td>0.588</td>
<td>-1.000</td>
<td>-0.951</td>
<td>0.618</td>
<td>0.000</td>
</tr>
<tr>
<td>10</td>
<td>0.309</td>
<td>-0.618</td>
<td>-0.809</td>
<td>1.000</td>
<td>-1.000</td>
</tr>
<tr>
<td>11</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
</tr>
</tbody>
</table>

Figure 2. (a) Excitation function and (b) its FFT (Not to scale)

5. OUTPUT-ONLY MODAL ANALYSIS

As mentioned in Section 4, the acceleration signals \( R(t) \) were obtained from the transient analysis of the beam. These signals are free of noise (zero noise level). Using these signals, the noise \( N(t) \) was generated with different noise levels \( N_L \) according to Equation (11). By adding \( N(t) \) to \( R(t) \) according to Equation (10), the noisy data \( N_R(t) \) was generated which was used for output-only modal analysis by FDD method. The considered noise levels and the corresponding signal to noise ratios are shown in Table 3. Practically, the amount of noise in recorded response affects the accuracy of the estimated modal parameters [29, 30].

In the noise generation process, each ensemble is statistically independent from others. Moreover, in order to capture the effect of randomness and average out the results, the process should be repeated [24]. At each noise level, 100 noise ensembles \( N(t) \) and consequently 100 noisy data \( N_R(t) \) were generated. Output-only modal analysis by FDD method was performed using each of the \( N_R(t) \)s, and the average of the results obtained for each noise level is presented in the following. The process of noise generation, noisy data generation and Output-only modal analysis by FDD method were all performed in MATLAB\(^1\).

5.1. Natural Frequencies

Table 4 presents natural frequencies of the beam predicted using FDD method for different noise levels from \( N_L=0 \) (noise free case) to \( N_L=200\% \). The prediction error with respect to reference frequencies (Table 1) are shown in the last row of Table 4. As can be seen, FDD method predicted most natural frequencies of the beam accurately. Maximum error is \(-5.2\% \) which belongs to mode 1. Moreover, natural frequency of mode 1 could not be identified at high noise levels; i.e. \( N_L=100\% \) and \( N_L=200\% \). The error of most of the estimated natural frequencies is negative, which shows that they are underestimated.

As shown in Table 4, by increasing the noise level, the values of the natural frequencies identified using FDD method do not change. The results corroborate with those reported by Gkoktsi et al. [31]. In the frequency domain methods such as FDD, the natural frequency of a structure is estimated from the peak values of PSD

<table>
<thead>
<tr>
<th>Noise Level (NL)</th>
<th>5%</th>
<th>10%</th>
<th>20%</th>
<th>50%</th>
<th>75%</th>
<th>100%</th>
<th>200%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal Signal to Noise Ratio (SNR)</td>
<td>400</td>
<td>100</td>
<td>25</td>
<td>4</td>
<td>1.78</td>
<td>1</td>
<td>0.25</td>
</tr>
</tbody>
</table>

TABLE 4. Natural frequencies (Hz) of the beam estimated via FDD method

<table>
<thead>
<tr>
<th>Noise Level (%)</th>
<th>Mode 1</th>
<th>Mode 2</th>
<th>Mode 3</th>
<th>Mode 4</th>
<th>Mode 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>0, 5, 10, 20, 50, 75</td>
<td>22.0</td>
<td>95.2</td>
<td>229.5</td>
<td>439.5</td>
<td>759.3</td>
</tr>
<tr>
<td>100, 200</td>
<td>-</td>
<td>95.2</td>
<td>229.5</td>
<td>439.5</td>
<td>759.3</td>
</tr>
<tr>
<td>Error (%)</td>
<td>-5.2</td>
<td>-0.7</td>
<td>4.0</td>
<td>-0.7</td>
<td>-1.9</td>
</tr>
</tbody>
</table>

\(^1\) www.mathworks.com/products/matlab.html
diagram. Despite the fact that different noise levels lead to different PSD diagrams, the peaks location or the values of natural frequencies of the structure do not change. This phenomenon is shown in Figure 3.

5.2. Mode Shapes

The first five mode shapes of the beam were obtained by FDD at various noise levels. For instance, mode shape values of the beam obtained at noise level 5% are shown in Table 5. In order to check accuracy of the obtained mode shapes, they will be compared with reference mode shapes obtained by finite element modal analysis (Table 2).

For the purpose of comparison, the MAC criterion is utilized. The MAC as a statistical index, shows a high sensitivity to large variations between two mode shape vectors, but it has relatively no sensitivity to small variations. This statistic indicator produces a good measure and degree of compatibility between two mode shapes. The MAC formulation is [32]:

$$MAC(r, q) = \frac{||\mathbf{\psi}_A^T||^2}{||\mathbf{\psi}_A||^2 ||\mathbf{\psi}_X||^2}$$

where $\{\mathbf{\psi}_A\}_r$ is mode shape number $r$ from source $A$, $\{\mathbf{\psi}_X\}_q$ is mode shape number $q$ from source $X$, and the superscript $T$ denotes transpose. The MAC values vary between zero and one; the one represents full compliance of two mode shapes [33].

Using Equation (14), the values of MAC between the five mode shapes of the beam obtained by FDD method and five reference mode shapes were calculated at various noise levels. For instance, these MAC values at noise level of 5% are presented in Figure 4. (At noise level of 5%, mode shape values in Table 5 and Table 2 were used for calculation of MAC values.) This figure shows that the values of MAC between the same mode shape numbers are equal to one and between different

Table 5 shows the values of MAC between the mode shapes obtained by FDD and the corresponding reference mode shapes at various noise levels (same mode

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>Mode 1</th>
<th>Mode 2</th>
<th>Mode 3</th>
<th>Mode 4</th>
<th>Mode 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
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<td>2</td>
<td>0.312</td>
<td>0.620</td>
<td>0.813</td>
<td>1.000</td>
<td>0.999</td>
</tr>
<tr>
<td>3</td>
<td>0.585</td>
<td>0.998</td>
<td>0.950</td>
<td>0.618</td>
<td>-0.001</td>
</tr>
<tr>
<td>4</td>
<td>0.819</td>
<td>1.000</td>
<td>0.311</td>
<td>-0.618</td>
<td>-0.996</td>
</tr>
<tr>
<td>5</td>
<td>0.956</td>
<td>0.618</td>
<td>-0.588</td>
<td>-0.999</td>
<td>-0.001</td>
</tr>
<tr>
<td>6</td>
<td>1.000</td>
<td>0.003</td>
<td>-1.000</td>
<td>-0.001</td>
<td>1.000</td>
</tr>
<tr>
<td>7</td>
<td>0.960</td>
<td>-0.619</td>
<td>-0.586</td>
<td>0.999</td>
<td>-0.001</td>
</tr>
<tr>
<td>8</td>
<td>0.823</td>
<td>-0.998</td>
<td>0.307</td>
<td>0.618</td>
<td>-0.999</td>
</tr>
<tr>
<td>9</td>
<td>0.593</td>
<td>-1.000</td>
<td>0.951</td>
<td>-0.618</td>
<td>0.000</td>
</tr>
<tr>
<td>10</td>
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<td>0.998</td>
</tr>
<tr>
<td>11</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
</tr>
</tbody>
</table>

Figure 3. The Average Normalized PSD diagram of the beam for various noise levels.
The MAC values equal or close to one indicate that the FDD method appropriately extracted five mode shape vectors of the beam in the presence of noise. As shown in the table, as the noise level increases, the MAC values either decrease or remain constant. The lower values of the MAC for the first mode shape of the beam show that this mode of the beam was not extracted flawlessly in high noise levels. However, for the second to fifth mode shapes, the MAC value remains equal to one even at high noise levels which shows that these mode shapes were extracted with high accuracy.

The identified mode shapes of this beam via FDD method for the various noise levels have been drawn in Figures 5 to 8. As can be observed in Figures 5 and 6, that correspond to noise levels of 5% and 20% respectively, all of the five mode shapes were identified almost accurately. According to Table 5, the MAC values for these mode shapes at the same noise levels are equal to one. At noise levels of 75% and 200% in Figures 7 and 8, respectively, the second to fifth mode shapes are nearly perfect, but the first mode shape is clearly distorted. In Table 5, the values of MAC of second to fifth mode shape for all of the considered noise levels are equal to one. However, for the first mode shape, the MAC value decreases from 1.00 at noise level of 0, 5, 10 and 20% to 0.95 at noise level of 200%. Therefore, for the case being studied, MAC equal to or less than 0.99 means that the obtained mode shape is not reliable.

As can be seen in Tables 4 and 6, the natural frequency of the first mode was obtained up to noise level of 75%, but its mode shape was only identified with good accuracy up to noise level of 20%. Therefore, identification of natural frequency of a mode does not necessarily guarantee that its mode shape will be identified correctly by FDD method.

<table>
<thead>
<tr>
<th>Noise Level (%)</th>
<th>Mode 1</th>
<th>Modes 2 to 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>0, 5, 10, 20</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>50, 75</td>
<td>0.99</td>
<td>1.00</td>
</tr>
<tr>
<td>100</td>
<td>0.97</td>
<td>1.00</td>
</tr>
<tr>
<td>200</td>
<td>0.95</td>
<td>1.00</td>
</tr>
</tbody>
</table>

Figure 4. MAC values between all FDD mode shapes (at noise level of 5%) and all FEM mode shapes

Figure 5. The beam's mode shapes obtained via FDD at noise level of 5%

Figure 6. The beam's mode shapes obtained via FDD at noise level of 20%

Figure 7. The beam's mode shapes obtained via FDD at noise level of 75%
In the present work, a method of output-only modal analysis, namely FDD method, was utilized to identify modal parameters of a beam under noisy conditions. A finite element model was prepared for a simply supported beam. Applying a suitable dynamic load to the beam, the acceleration signals of the various points on the beam were obtained by transient analysis. Noise signals with different powers were added to the acceleration signals. Using these noisy acceleration signals, modal parameters of the beam were identified by FDD method. These identified modal parameters were then compared to reference ones that were obtained by finite element modal analysis method.

In noise free case, the FDD method could identify this beam's mode shapes and natural frequencies. The beam's natural frequency corresponding to the first mode was identified with more error in comparison to other modes. MAC values are equal to one in this case, which shows very good agreement between the identified mode shapes and the corresponding reference.

In the case of noisy signals, the beam's natural frequencies of all modes were obtained at all noise levels, except for the first natural frequency which was not identified for noise levels 100 and 200%. The first natural frequency could not be identified at very high noise levels. The natural frequencies of the beam, which were identified, show no change in comparison to the noise free case. The FDD was a powerful method for estimation of the beam's natural frequencies, because it identified the natural frequencies at the reasonably high noise levels. The inspection of the mode shapes obtained using FDD method under noisy conditions showed that all of the mode shapes were in very good agreement with reference mode shapes up to noise level of 20%. Up to this noise level, the calculated MAC values for all of the mode shapes are equal to one. For noise levels greater than 20%, accuracy of the first mode shape decreases (MAC=0.99–0.95), but other mode shapes remain accurate (MAC=1.00). Therefore, the FDD method is more robust for prediction of natural frequencies in comparison to mode shapes. The amount of noise influenced properties of the first mode which is usually the most important vibration mode of a structure in practice. Because noise level is unknown in practice, the higher mode shapes identified by the FDD method are more reliable in comparison with the first mode shape.

7. REFERENCES


Output-only Modal Analysis of a Beam Via Frequency Domain Decomposition Method Using Noisy Data

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\textbf{Paper Info}

\textit{Paper history:}
Received 04 March 2019
Received in revised form 23 September 2019
Accepted 07 November 2019

\textbf{Keywords:}
Frequency Domain Decomposition
Modal Parameters
Noise Level
Output-only Modal Analysis
Simply Supported Beam

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analysis of a beam output-only modal analysis of a beam via frequency domain decomposition method using noisy data.
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Attractor Modal Analysis - One way of analyzing the behavior of a beam is through the use of frequency domain decomposition. The data used in this analysis are usually accompanied by noise. The success of the frequency-domain-only modal analysis method depends on the noise level in the output data. In this paper, the determination of modal parameters of a simply supported beam and their accuracy under the influence of noise is investigated. The analysis of modal parameters of the beam is carried out using the frequency-domain decomposition method and the output data with different levels of noise. The procedure begins with the determination of the modal parameters of the beam using the frequency-domain decomposition method. Then, an appropriate input is applied and signals of acceleration at various nodes are obtained through the analysis of the history of the modal components. To generate the noisy data, different levels of noise are produced and added to the signals. Finally, the modal parameters of the beam are determined using the frequency-domain decomposition method. The results show that the modal parameters of the first mode of vibration can be accurately identified in the low to medium noise level, while for higher modes, the modal parameters are even accurately obtained in high noise levels.


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