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EFFECTIVE SENSOR CLUSTERING FOR ONLINE MONITORING IN TRANSMISSION TOWERS

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Abstract

Health monitoring systems are essential to investigate the performances during the service life of a structure such as the three-dimensional transmission tower. However, many real-world sensors and transducers require sensor positioning drafts before a computer based measurement procedure can effectively and accurately acquire the signal. This paper develops a direct physical property adjustment method, named as cross-model cross-mode method. In dealing with spatially incomplete situations, model reduction schemes were used. The selection procedure of the inactive degrees of freedom in process of the model reduction evaluated with a reasonable criterion by using the sensitivity analysis of system response under base excitation. Also, the noisy data measurements are the other crucial factors. The success rates based on the correct detection probability factor were defined in order to evaluate the noise effect on the accuracy of the method. The efficiency of the method is validated by different damage scenarios. The results show that the developed methods are suitable for damage classification. But in the cases of the less used sensors than the degrees of the freedoms, the location of the sensors must be considered an important factor influencing the success rates.

Keywords: Structural health monitoring, Damage detection, Transmission tower, Model reduction, sensor clustering.

1. Introduction

Research on vibration-based damage identification using changes in output signals from the structure has expanded rapidly in last decades [1]. Uhl [2] and Wang et al. [3] can be served as the state-of-the-art reviews on the vibration-based methods. In recent years, several researchers have studied damage detection in transmission towers.

Abbreviations	
CMCM	Cross Model Cross Mode
FE	Finite Element

Heung and Tao [4] discussed structural monitoring in this kind of structures described a fault-detection algorithm using dynamic reduction-based methods. This work provides a detailed overview of the background research and serves as a good reference in this field.

Since the damage may cause the change on the stiffness distribution of the structural system, subsequently the modal properties of the system may be altered as well. Damage detection methods using techniques based on modal parameters can be divided into direct and iterative methods. Using the direct methods the matrices are updated by forming a constrained optimization problem. However, because of changes in the dynamic matrices in the mathematical model, the physical meanings of the original practical structures cannot be preserved. In the iterative methods, the solving procedure follows an optimization problem, in which the inconsistencies between the numerical and practical dynamic characteristics are minimized by adjusting the modal parameters. The major advantage of iterative methods over direct methods is the ability to preservation of the initial correspondence between the degrees of freedom within the dynamic matrices of the practical structures.

The focus of the present paper is on the problems of uncertainties were considered to be the main objectives in developing and evaluating a robust damage detection system. These concepts are investigated by the adaptation of two methods based on different standpoints: experimental modal analysis and time-capture data processing. In addition, one implicit objective of this study is to discuss the inherent difficulties in implementing SHM techniques for complex structures, such as transmission towers. The expansion of such methodologies can be extremely useful in assisting technologies that can be applied to structures in service. In this work, a physical model was constructed for this purpose.

The concept of global monitoring methods is established based on the dynamic responses of systems. These responses can be recorded during excitation of structures by each type of the external dynamic forces. As the considered feature sets of this study are the natural frequencies which are independent of the excitation types, the proposed method is adaptable to each type of input forces and the inherent structural dynamic output response must be considered more carefully which is the significant key point for the related method.

On the other hand, the updating process of the stiffness matrix is extended based on the cross modal cross model method to detect and quantify the severity of damage under the supposition that only the first few modal parameters have been recognized. The change of modal parameters can be used as a basis for these kinds of the fault recognition methods [5].

The global structural health monitoring methods are facing two major problems for the situ towers; the lack of coordination of measurement sensors and degrees of-freedoms of the numerically model, namely the spatial incompleteness [6]. In dealing with spatially incomplete situations, model reduction schemes can be used.

Vibration phenomena have always been a cause of concern to engineers, even more today as structures are becoming lighter and more flexible due to increased requests for efficiency and safety. It is clear that a comprehensive understanding of the existing vibration levels in service is essential. Accordingly, precise analytical models of structures are required to explain the vibration characteristics. A most widely used analytical tool is the Finite Element method based on the modal testing. The Finite Element method is widely used in industry as it can produce a good representation of a factual structure [7]. However, it must be approved that due to limitations in this method, a Finite Element model is always a rough calculation of the prototype. Inaccuracies in the model can occur due to inaccurate estimation of the physical properties of the structure, in individual element shape functions or a poor quality mesh, poor approximation of the boundary conditions and occurrence of additional inaccuracies during the solution phase [8].

The modal measurements are taken directly from a physical structure without any assumptions about the structure and as such they are considered to be more reliable than their Finite Element counterpart. Inaccuracies in the empirical attitudes may occur due to errors owing to noise, the assumption of linear response while there can also be non-linear responses and nonlinearities in the measurements. It is generally believed that more confidence can be placed on experimental data as measurements are taken on the true structure. Therefore, the analytical model of a structure is usually updated on the strength of the experimental model. In this study for the sake of improvement of the well-known cross-model cross mode method we evaluated selection procedure of the slave DOFs by using the sensitivity analysis of system response under a base excitation. This performance leads to faster convergence of iterative algorithm.

2. Cross Model Cross Mode and Model Reduction terminologies and representations

One main scope of the experimental modal analysis is extraction of the frequency response functions (FRF's). In the first step of an experimental modal analysis, the elements of at least one full raw or one full column of the FRF matrix should be measured and then the natural frequencies can be identified using a variety of different methods such as Rational Fraction Polynomial (RFP) method. Another very important aspect of modal testing is the correlation and correction of a numerical model such as a finite element models.

The main outline of the methods being used in this study is based on the methodology similar to cross-model cross mode [10]. So the method is introduced briefly. The equation of motion of an un-damped dynamic system is given as follows:

$$M\ddot{V} + KV = 0 \tag{1}$$

where, M and K are the mass and stiffness matrix, respectively. Also, V and $\ddot{V}$ vectors denote the displacements and accelerations. The i^{th} eigenvalues and eigenvectors are expressed as:

$$K\emptyset_i = \gamma_i M\emptyset_i \tag{2}$$

where γ_i and $\emptyset_i$ is the i^{th} eigenvalue and eigenvector corresponding to baseline K and M matrixes which can be obtained from a finite element model. The stiffness matrix K^* of the experimental model is formulated as a modification form of K:

$$K^* = K + \sum_{n=1}^{N_e} \alpha_n K_n \tag{3}$$

where K_n is the stiffness matrix corresponding to the n^{th} element, N_e is the number of elements and α_n are unknown correction factors must be determined. The jth eigenvalue and eigenvector associated with K^* and M^* are formulated as follows:

$$K^* \Phi_j^* = \lambda_j^* M^* \Phi_j^* \tag{4}$$

Here, it will be assumed that a few of λ_j^* and Φ_j^* are known measurements available from modal testing then with pre multiplying by $\emptyset_i^t$ yields:

$$\emptyset_i^t K^* \emptyset_j^* = \gamma_j^* \emptyset_i^t M^* \emptyset_j^* \tag{5}$$

Substituting Eq. (3) into Eq. (5) yields:

$$C_{ij}^l + \sum_{n=1}^{Ne} \alpha_n C_{n,ij}^l = \gamma_j^* D_{ij}^l \tag{6}$$

Index ϵ is used to replace ij:

$$C_\in^l + \sum_{n=1}^{Ne} \alpha_n C_{n,\in}^l = \gamma_j^* D_\in^l \tag{7}$$

Then

$$\sum_{n=1}^{Ne} \alpha_n C_{n,\in}^l = f_\in^l \tag{8}$$

when N_i modes are taken from the baseline finite element model, and N_j modes are measured from the damaged structure, totally $N_\varepsilon = N_i \times N_j$ equations can be formed from Eq. (8). Equations formed based on Eq. (8) are named cross model cross mode:

$$C_{N_v * N_\in}^l \alpha_{N_\epsilon * 1} = f_{N_v * 1} \tag{9}$$

If N_v is greater than N_e then more equations are available than unknowns and to gain the parameter of α, the least squares solution can be taken as follows:

$$\alpha = (C^t C)^{-1} C^t f^l \tag{10}$$

2.1. Spatial Incompleteness and Guyan Model Reduction Technique

The problem of vibration analysis consists of determining the conditions under which the equilibrium condition expressed by Eq. (1) will be satisfied. It would be assumed that the free vibration motion is simple harmonic and be expressed as follow:

$$V(t)=V\sin(\omega t+\Theta) \tag{11}$$

where, $\hat{V}$ represents the stationary shape of the system and θ is a phase angle. Therefore, the accelerations in free vibration can be derived:

$$\ddot{V}=-\omega^2\hat{V}\sin(\omega t+\theta)=\omega V \tag{12}$$

and with the considering of Eq. (1)

$$-\omega^2 m\hat{V}\sin(\omega t+\theta)+k\hat{V}\sin(\omega t+\theta)=0 \tag{13}$$

by omitting the arbitrary sine function:

$$\left[k-\omega^2 m\right]\ddot{V}=0 \tag{14}$$

Now it can be shown by Cramer's rule that the solution of this set of simultaneous equations is of the form:

$$\hat{V}=\frac{0}{\left\|k-\omega^2 m\right\|} \tag{15}$$

Hence, the finite amplitude free vibrations are possible only when:

$$\left\|k-\omega^2 m\right\|=0 \tag{16}$$

The N roots of this equation ($\omega_1^2,\omega_2^2,...,\omega_N^2$) represent the frequencies of the N modes of vibration. Using the MATLAB software, [V, D] = eig (A) produces matrices of eigenvalues (D) and eigenvectors (V) of matrix A. Matrix D is the canonical form of a diagonal matrix A. Matrix V is the modal matrix.

The major problem inherent to dynamic structural analysis is the time-consuming and costly amount of computation required. As practical finite element models can contain tens of thousands of degrees of freedom, the time and expense of computing all of the frequencies and mode shapes are prohibitive. Fortunately, to obtain reasonable approximations of dynamic response, it is seldom necessary to solve the full eigenvalue problem. Two practical arguments underlie the preceding statement. First, the lower-valued frequencies and corresponding mode shapes are more important in describing structural behaviour. Second, when structures are subjected to time-dependent forcing functions, the range of forcing frequencies to be experienced is reasonably predictable. Therefore, only system natural frequencies around that range are of concern in examining resonance possibilities [11].

A well-known model reduction method is a static reduction method introduced by Guyan [11]. This technique partitions the mass and stiffness matrices, and the displacement vector into a set of master and slave degree of freedoms. The Guyan transformation matrix and the reduced Guyan mass and stiffness matrices are presented as follows:

$$\begin{bmatrix}[M_{mm}] & [M_{ms}]\\ [M_{sm}] & [M_{ss}]\end{bmatrix}\begin{Bmatrix}\ddot{V}_m\\ \ddot{V}_s\end{Bmatrix}+\begin{bmatrix}[K_{mm}] & [K_{ms}]\\ [K_{sm}] & [K_{ss}]\end{bmatrix}\begin{Bmatrix}V_m\\ V_s\end{Bmatrix}=\begin{Bmatrix}0\\ 0\end{Bmatrix} \tag{17}$$

Here, the subscripts m and s correspond to master and slave coordinates, respectively. The inertia terms are neglected to obtain the equation:

$$[K_{sm}]\{V_m\}+[K_{ss}]\{V_s\}=[T_s]\{V_m\} \tag{18}$$

This equation may be used to eliminate the slave coordinate to leave the following:

$$\begin{Bmatrix} V_m \\ V_s \end{Bmatrix} = \begin{bmatrix} [I] \\ -[K_{ss}]^{-1}[K_{sm}] \end{bmatrix}\{V_m\} = [T_s]\{V_m\} \tag{19}$$

$$[T_s] = \begin{bmatrix} [I] \\ -[K_{ss}]^{-I}[s_m] \end{bmatrix} \tag{20}$$

T_s is Guyan transformation matrix and I is identify matrix.

The reduced Guyan mass and stiffness matrices are then given by

$$[M_R] = [T_s^i][M][T_s] \tag{21}$$

$$[K_R] = [T_s^i][K][T_s] \tag{22}$$

In dealing with spatial incompleteness, usually applies model reduction schemes. The transformation matrix the master coordinates of the full order coordinates for the baseline model is denoted as *T*. The final relations are produced by applying $\Phi_i = T(\Phi_i)_m$ and $\Phi_j^* = T^*(\Phi_j^*)_m$ the previous equations, where $(\Phi_i)_m$, $(\Phi_j^*)_m$ and T^* are the i^{th} mode shape of the baseline structure calculated only at the master coordinates, the j^{th} mode shape of the damaged structure measured only at the master coordinates and the counterpart of T for the damaged structure respectively.

$$C_v^i = [\emptyset_{im}]^t T^t K T^* (\emptyset_j^*)_m \tag{23}$$

$$C_{n,v}^i = [\emptyset_{im}]^t T^t K_n T^* (\emptyset_j^*)_m \tag{24}$$

$$D_v^i = [\emptyset_{im}]^t T^t M T^* (\emptyset_j^*)_m \tag{25}$$

Here, *T* is equal with the reduced stiffness matrix ($[K_R]=[T^t][K][T^*]$). For the implementation the proposed technique, initially extracted from ANSYS software the mass and stiffness matrices under substructure analysis. Then, is done in *MATLAB* software all calculations including; the calculated frequencies and displacement and eigenvalues vectors, select the master degrees of freedom, calculating the transformation matrix, formation and solving the Eq. (9). Thus, while applying the proposed method, the only source of errors Source from T^*, assuming that $(\Phi_i)_m$ (measured only at the master coordinates) has been a noise-free measurement. Because T^* is unknown originally, an iterative procedure to have $T^* = T$ as its first iteration is proposed.

2.2. Selections of Slave Degrees of Freedom Based on the Dynamic Sensitivity Analysis

Sensitivity analysis allows one to evaluate the impact that changes in a certain parameter will have on the structural responses and also it can help the analyser to identify which parameters are the key drivers of a model's results. In this study, spectrum analysis with Single-Point Response Spectrum was used for the sensitivity analysis.

For single-point response spectrum analysis and dynamic design analysis method, the structure is excited by a spectrum of known direction and frequency components, acting uniformly on all support points or on specified unsupported master degrees of freedom. The general process for performing a single-point response spectrum analysis consists of six primary steps. These are consisting of building the model, obtaining the modal solution, obtaining the spectrum solution, expanding the modes, combining the Modes, and reviewing the results.

It must be attending that only the linear behaviour is valid in a spectrum analysis. For this study, the tower model was excited in the range of first mode in the vertical direction. As a result, seismic displacement in the form of equivalent nodal stress was checked as response of the tower. ANSYS offers five different mode combination methods for the single-point response spectrum analysis. Here, the Square Root of Sum of Squares (SRSS) method was applied for the prepared model.

3. Modal analysis and FE model updating based on the modal assurance criterion

Experimental modal analysis is known simply as a process for describing a structure in terms of its dynamic properties. The methods can be classified into Operational Modal Analysis and the Experimental Modal Analysis [13, 14]. When the numerically and experimentally identified dynamic characteristics are compared to each other, some differences between numerical and experimental are found due to various types of uncertainties in the finite element model which can produce the false alarms (in a finite element model updating process for the damage detection purposes, the final experimental modal results are far more acceptable and considered as the objective if the experimental modal analysis was performed satisfactory and the good measurements were obtained). In the mechanical and signal processing laboratories, the measured responses can be obtained from the shaker or hammer impact tests, as shown in Fig. 1 [15].

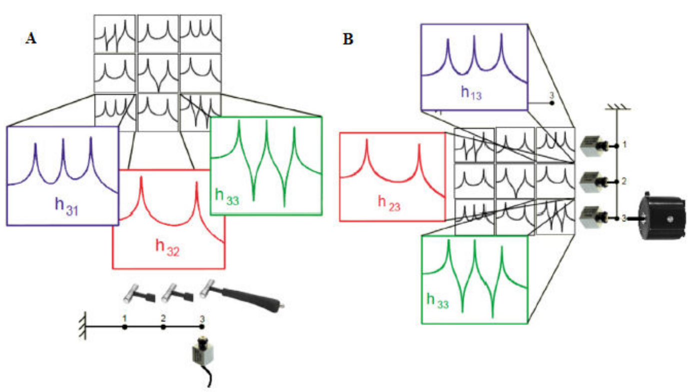


Fig. 1. Experimental modal analysis;
A) moving impact test, B) moving response test [12].

The ANSYS FE package was employed to obtain the numerical modal parameters. The Young's modulus, Poisson ratio and density were 200 GPa, 0.3 and 7850 kg/m^3 respectively. The updating of the initial finite element model is necessary to minimize the numerical model error according to the experimental signatures. The concepts of the "Modal Assurance Criterion" method can be explained as follows: Eq. (26) defines a vector of parameters related to modal properties:

$$w = \begin{Bmatrix} w_1 \\ \cdots \\ w_2 \end{Bmatrix} \quad (26)$$

The parameters in the above equations are defined below:

$$w_1 = \begin{Bmatrix} \cdot \\ \cdot \\ \cdot \\ \left(\frac{\omega_a}{\omega_e}\right)_i \\ \cdot \\ \cdot \\ \cdot \end{Bmatrix}, w_2 = \begin{Bmatrix} \cdot \\ \cdot \\ \cdot \\ MAC_i \\ \cdot \\ \cdot \\ \cdot \end{Bmatrix} \text{ and } MAC_i = \left[\frac{\left({\varphi_e}^T\varphi_a\right)^2}{\left({\varphi_e}^T\varphi_e\right)\left({\varphi_i}^T\varphi_a\right)}\right]_i \quad (27)$$

where ω_i and φ_i are thei^{th} Eigenvalue and mode shape, respectively, and the subscripts a and e denote the analytical and corresponding experimental values. Using the first-order Taylor's series, gives:

$$w_e = w_a + R\Delta x + \epsilon \quad (28)$$

where w_e and w_a are the experimental and analytical function vectors, T is the design sensitivity matrix of w_a, $x\Delta x$ are the changes in x for the least squares minimization, and ϵ is a residual vector. The least squares solution for Δx to minimize $\epsilon^T\epsilon$ is:

$$\Delta x = (R^TR)^{-1}R^T\Delta w = (R^TR)^{-1}R^T(w_e - w_a) = (R^TR)^{-1}R^T(\{1\} - w_a) \quad (29)$$

where the design sensitivity matrix modal functions of the Eigenvalue and the Eigenvector can define as follows:

$$R = \begin{bmatrix} \frac{\partial w_{1a}}{\partial x} \\ \cdots \\ \frac{\partial w_{2a}}{\partial x} \end{bmatrix} = \begin{bmatrix} \frac{\partial[\omega_a/\omega_e]}{\partial x} \\ \cdots \\ \frac{\partial[MAC]}{\partial x} \end{bmatrix} \quad (30)$$

The Eq. (27) is rearranged as follow:

$$MAC_i = \left[\frac{\left({\varphi_e}^T\varphi_a\right)^2}{\left({\varphi_e}^T\varphi_e\right)\left({\varphi_a}^T\varphi_a\right)}\right]_i \equiv \left(\frac{\tau}{\beta}\right)_i \quad (31)$$

where $\tau_i \equiv (\varphi_e^T\varphi_a)_i^2 \; and \; \beta_i \equiv (\varphi_e^T\varphi_e)_i(\varphi_a^T\varphi_a)_i$. If the *MAC* with value of 1 indicates perfect correlation, then the partial derivative of MAC_i with respect to the design variable x_j can be written as follow:

$$\frac{\partial MAC_i}{\partial x_j} = \frac{\beta_i \frac{\partial \tau_i}{\partial x_j} - \tau_i \frac{\partial \beta_i}{\partial x_j}}{\beta_i^2}$$

$$\frac{\partial \tau_i}{\partial x_j} = \frac{\partial (\varphi_e^T \varphi_a)_i^2}{\partial x_j} = 2(\varphi_e^T \varphi_a)_i \left(\varphi_e^T \frac{\partial \varphi_a}{\partial \varphi_j}\right)_i \quad (32)$$

$$\frac{\partial \beta_i}{\partial x_j} = (\varphi_e^T \varphi_e)_i \left(\frac{\partial \varphi_a^T}{\partial x_j} \varphi_a + \varphi_a^T \frac{\partial \varphi_a}{\partial x_j}\right)_i = 2(\varphi_e^T \varphi_e)_i \left(\varphi_a^T \frac{\partial \varphi_a}{\partial \varphi_j}\right)_i$$

Description of the physical model and test setup

For the sake of validation of the methods described above, experimental modal tests were performed on a physical model. The general shape of the model is shown in Fig. 2(a).

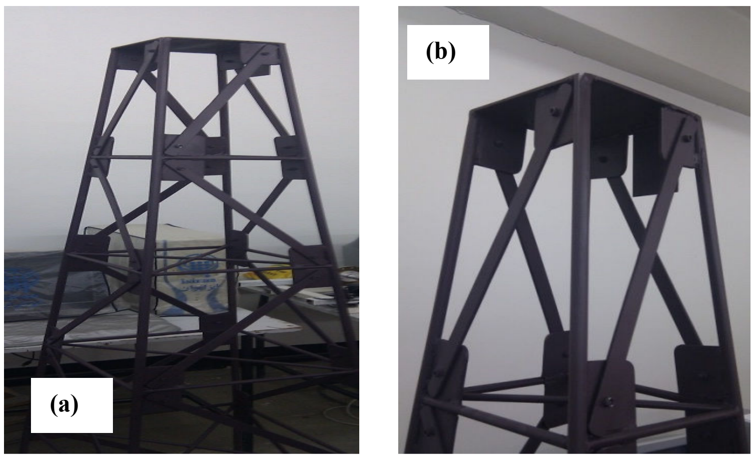


Fig. 2. The physical model description,
(a) general view, (b) Facilities for damage simulation.

The replaceable diagonal bracings were attached separately at the joints of the each space frame spans as illustrated in Fig. 2(b). The external white noise excitation signals were produced by an electro dynamic exciter (type 4809) with a force sensor (AC20, APTech) driven by a power amplifier (model 2706), all made by Bruel & Kjaer. The schematic shape of the model is shown in Fig. 3(a). The test instruments are illustrated in Fig. 3(b). The tests performed on the undamaged structure and then repeated in the same way for the damaged structure.

The MEscope software was used to obtain the experimental modal parameters by polynomial curve fitting of the frequency response functions. The data required to calculate the frequency response functions were recorded by sensors that were fixed on the physical model joints. The resulting numerical parameters were somewhat inconsistent with the experimental values. This inconsistency can be referred to problem of the environmental noises as a main source for uncertainties. The intensities of the sensitivities of mode shapes for the first four modes with respect to the certain damage scenarios are shown in Figs 4 and 5. The updated FE model was used as the baseline finite element model.

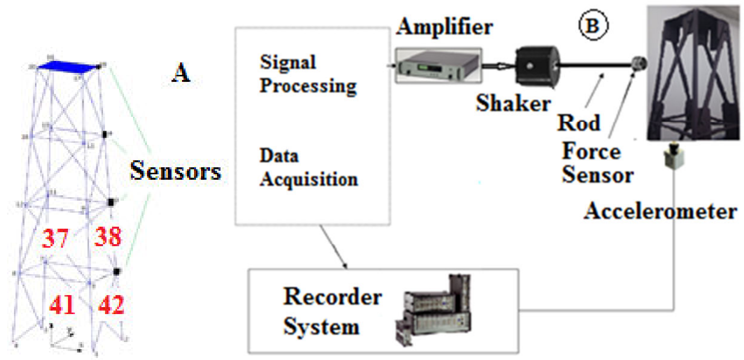


Fig. 3. The plan of the tests,
(a) The sketch of the model, (b) The instrumentations.

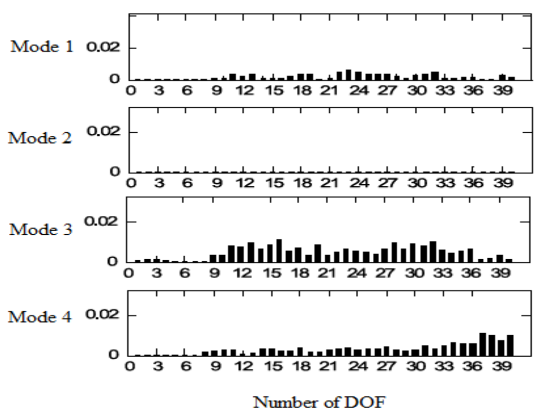


Fig. 4. Intensity of sensitivity for the mode shapes
with the elimination of the member 37.

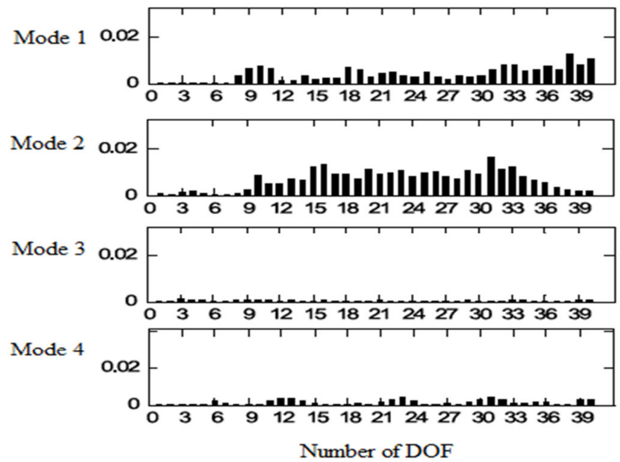
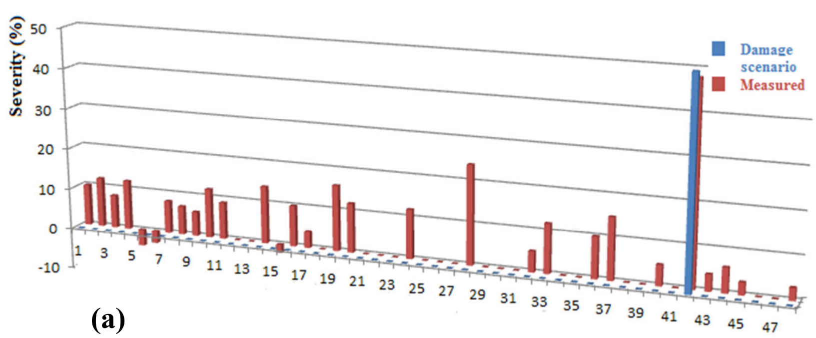


Fig. 5. Intensity of sensitivity for the mode shapes with the elimination of the member 38.

4. The Results of the Identify the Severity and Location of the Damage

Based on the basic concept of the vibration analysis, the natural is an undamped frequency and in this problem the damping is not considered. Of course, the damping parameter play individual role in behavior of a real structure and can be obtained under various assumptions via software such as the MEscope. But in most of the damage detection problems, in order that the methods would not be affected by damping, the undamped natural frequencies are considered as the desire extracted features via the FE method software. Detection results based on the approach without the iteration procedure and the effects of the iteration procedure with the reduction of number of the Dofs are shown in Figs. 6 and 7. These results obtained based on the using four modes of the damaged and the eight modes of the intact structure. Four tests were performed based on the reported scenarios in Table 1. In this table, the damage severity is defined as below:

$$D = 100\frac{E_{undamaged} - E_{damaged}}{E_{undamaged}} \tag{33}$$



(a)

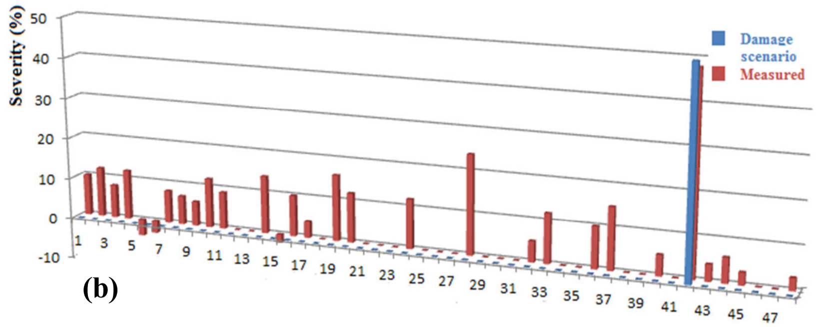


Fig. 6. Classification results with 48 degrees of freedom and without iteration procedure: (a) Damage scenario 3, (b) Damage scenario 4.

Table 1. Description of the experimental tests.

Case	Eliminated members	Damage severity (%)
1	37	100
2	38	100
3	42	50
4	37 and 42	100 and 50

Elements 37 and 38, bracing members between the second and the third floors are poorly estimated. When the iterative procedure is applied, matrix T is always calculated based on the damaged model obtained from the previous iteration. Applying the iterative procedure improves the performance of the detection algorithm.

In practice, for the sake of the saving the costs and due to some implementation issues, the number of the sensors is used less than the degrees of the freedom. Hence, for the examination of the effects of elimination of some degrees of freedom, 24 degrees of freedom considered as slave degrees. At first, they considered on the nodes at the upper floors. Also, during the other test the slave degrees considered on the lower floors. As shown in Fig. 7 errors are great during the first test. As a result, the better performance can be obtained when the available sensors are installed in the upper part of the platform.

During the operational modal measurements, the records always are perturbed by environmental noises. The disturbances in the measuring tools are origin of these errors. The x^{th} polluted mode shape of the damaged structure at the z^{th} DoF, denoted by $\Delta\emptyset_{noisy}^{xz}$, and was simulated by adding a Gaussian random error to the corresponding intact value $\Delta\emptyset$:

$$\Delta\emptyset_{noisy}^{xz} = \Delta\emptyset(1 + u\alpha) \qquad (34)$$

where α denotes a noise level, and u is a Gaussian random number with zero mean and unit standard deviation. In this study, the results were obtained by taking the repeated Monte Carlo simulations. A factor called correct detection probability is defined in order to evaluate the noise effect on the accuracy of the proposed method. If N_n is used to present the number of Monte Carlo simulations for a given level of noise and N_c the number of realizations that an actual damage is detected, the percentage of correct detection probability S_R, known as the success rate will be given by:

$$S_R(\%) = \frac{N_c}{N_n} \times 100 \quad (35)$$

The results for applied noise level distributions from 1% and 5% are shown in Table 2. The averages of the damage estimates from the 500 simulations are observed from this table. Each simulation is based on 1% and 5% error levels and four measured modes that are employed in the cross-model cross mode method. For example, for the Case 1, the detection probability is S_R=81% for a 1% noise level and S_R =68.9% for a 5% noise level.

Also, the standard deviations of measured severities for 1% noise levels are illustrated in Fig. 8. The column member 37 which is located on the upper floor implies the larger σ values that are indicated to this fact that these members are most sensitive to measurement errors. This observation can be explained with considering the vibration mode of the dominant first mode. Since the structural members at the lower floors are more constrained, therefore the top side members supply less modal strain energy from bending effect than the lower ones. Consequently, the damage detection at lower floor can be more accurate with less sensitivity to environmental noises.

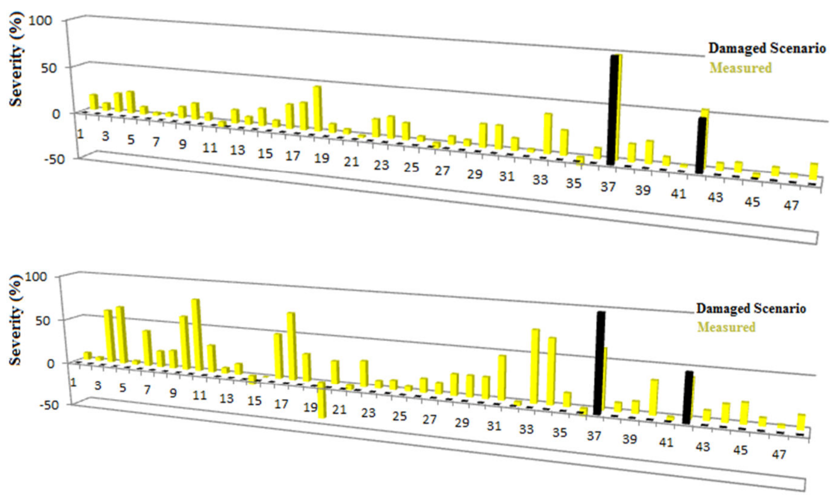


Fig. 7. Results for damage scenario 4, with considering of the 24 degrees of freedom and iteration procedure: (a) Sensors at the upper floors, (b) Lower floor.

Table 2. Success rates for different applied noise levels.

Damage Scenario	Success Rate ($\alpha = 1\%$)	Success Rate ($\alpha = 5\%$)
Case 1	81	68.9
Case 2	83.1	67.8

Case 3	85	73
Case 4	92.3	86

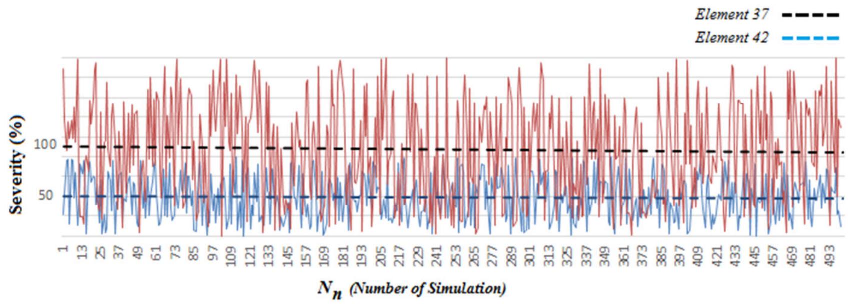


Fig. 8. Detection of damage severity with 48 degrees of freedom without iteration procedure under $\alpha = 1\%$.

5. Conclusions

A classification algorithm to classify damages in an apace frame model of a transmission tower, named as cross-model cross-mode method is presented and inspired by modal analysis and data processing. The global structural health monitoring methods are facing a major problem for the situ towers, namely the spatial incompleteness. In dealing with spatially incomplete situations, model reduction schemes were used. On the other hand, the noisy data measurements are the other crucial factors. To address the problem, the correct detection probability factors were defined based on the Monte Carlo simulations in order to evaluate the noise effect on the accuracy of the method. On the other hand, an initial FE-model is modified through updating the analytical model with consideration of the experimental modal analysis results based on a physical model.

In this process, the parameters of the elastic modulus and the stiffness of the supports at the base of the structure are considered as the more efficient factors. Moreover, the reflection of the sensitivity analysis on the updated model played an important role as a perspective to reduce the model for assessment the improved cross-model cross mode method via the application of the appropriate criterion to select the degrees of freedom. The development of such methods would be extremely useful to save costly amount of computation required, both in time and cost.

Also, it was observed that the main columns of the towers are most sensitive to the applied noise. By removing of some points of the sensing, it was observed that sensors which were located on the top floors of the tower possess the most roles in terms of performance and influence at the process of damage detection.

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