DYNAMIC CHARACTERIZATION OF CIVIL STRUCTURES USING MOBILE SENSORS

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The state-of-the-art in dynamic characterization for civil structures is limited to the estimation of mode shapes with low spatial resolution. Modal coordinates are identified only at the location of sensors, which are fixed at key points on the structure. Increasing the number of sensors in the structure is an alternative to increase the spatial resolution. Unfortunately, this increases the cost of the instrumentation and the data to be transmitted and processed. Another common alternative is to use numerical algorithms to expand the modal coordinates to non-measured degrees of freedom. However, mode shape expansion techniques could introduce errors in the indentified modes. This document presents the formulation, evaluation and validation of an innovative methodology for modal identification using mobile sensors. The methodology estimates mode shapes with high spatial resolution using at least one mobile sensor. Experimental tests with simply supported steel beams show that the methodology is successful when compared with traditional modal identification methods. This research opens the door to new possibilities of the applications that use modal identification results such as model updating, structural health monitoring and damage prognosis.

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ABSTRACT: The state-of-the-art in dynamic characterization for civil structures is limited to the estimation of mode shapes with low spatial resolution. Modal coordinates are identified only at the location of sensors, which are fixed at key points on the structure. Increasing the number of sensors in the structure is an alternative to increase the spatial resolution. Unfortunately, this increases the cost of the instrumentation and the data to be transmitted and processed. Another common alternative is to use numerical algorithms to expand the modal coordinates to non-measured degrees of freedom. However, mode shape expansion techniques could introduce errors in the indentified modes. This document presents the formulation, evaluation and validation of an innovative methodology for modal identification using mobile sensors. The methodology estimates mode shapes with high spatial resolution using at least one mobile sensor. Experimental tests with simply supported steel beams show that the methodology is successful when compared with traditional modal identification methods. This research opens the door to new possibilities of the applications that use modal identification results such as model updating, structural health monitoring and damage prognosis.

1 INTRODUCTION

Civil infrastructure systems are extraordinarily important for our society. Our economy, security, health and comfort depend directly or indirectly on the adequate transportation, habitat and communication systems. Nowadays, in addition to the impact of strong natural and human made events, infrastructure's deterioration caused by natural use and aging is a concern of the engineering community around the world. As an example, the American Society of Civil Engineers (ASCE) gave a point grade average of D to the infrastructure of the United States for more than ten years, with only a slight improvement in the 2001 report (American Society of Civil Engineers 2009). Additionally, the last report states that 2.2 trillion dollars should be invested in the next five years to improve the current infrastructure.

Strategies to maintain or improve existing structural systems usually require numerical models of the structure to analyze its behavior (Zárate and Caicedo 2008). These numerical models are used to evaluate structural performance under specific conditions such as heavy loading (Schlune, Plos et al. 2009), earthquake motion (Alyami, Rouainia et al. 2009), wind loading (Kim, Yu et al. 2009) or human activity (Racic, Pavic et al. 2009). Therefore, developing accurate models of existing structures is key to evaluate the vulnerability (Galati, Nanni et al. 2008), detect damage (Teughels and De Roeck 2004), study retrofit alternatives (Stehmeyer and Rizos 2008) and predict the remaining useful life of structures (Fritzen and Kraemer 2009). The accuracy of the numerical model refers to the ability to reproduce the response of the real structure having

parameters with a realistic physical meaning. This requires an accurate experimental characterization of the static and/or dynamic behavior of the structure to compare it with numerical results and tune the model.

Many model updating methodologies use modal parameters as input for the updating process. The state-of-the-art in modal identification of civil structures use sensors placed at strategic points on the structure. The coordinates of the mode shapes are calculated at the sensor locations only, resulting in sparse identified mode shapes. However, many applications require modal coordinates at degrees of freedom that have not been measured. The expansion of the identified coordinates or the reduction of the numerical model being updated are two approaches commonly used in the literature (Friswell and Mottershead 1995). However, in general, mode shape expansion methods can introduce errors into the modal identification process due to: i) discrepancy between the location of the sensors and the location of DOF in numerical models, and ii) modeling errors (Balmès 2000; Pascual, Schälchli et al. 2005). In addition, analytical models are based on assumptions that might not correctly represent the actual structure. These assumptions generate modeling errors that need to be considered from a modal expansion perspective.

An alternative to address the low spatial resolution is to install more sensors. In other words, use spatially dense sensor networks. However, the cost of instrumentation and installation of dense sensor networks makes it a sometimes prohibited approach. Smart wireless sensors have been proposed for large instrumentation systems (Spencer, Ruiz-Sandoval et al. 2004; Lynch 2007). The relative lower cost and easy installation of these instruments make them suitable for large sensor networks. However, additional challenges on the network communication due to the large amount of data to be transmitted have to be addressed. In addition, algorithms should be computationally simple to reduce energy consumption on battery powered sensors.

This document proposes the use of mobile sensors to calculate spatially dense mode shapes. The methodology requires the use of one stationary and one mobile sensor to identify mode shapes without any assumption about the shape of the mode (i.e. polynomial). The focus of the paper is in civil structures, where the mass of the sensor is significantly smaller than the mass of the structure. Therefore, the sensor is not expected to modify the dynamics of the structure.

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2 METHODOLOGY

The acceleration response of a one-dimensional system in the $x-axis$ subjected to a dynamic excitation is described by (Chopra 2007)

$$\ddot{u}(x,t) = \sum \phi_n(x)\ddot{q}_n(t) \quad (1)$$

where $\phi_n(x)$ is the $n-th$ natural vibration mode and $\ddot{q}_n(t)$ is the response of the $n-th$ mode in generalized coordinates. If the response of the system is measured with

a mobile sensor travelling at a constant speed v, the response can be written as a function of time only

$$\ddot{u}(t) = \sum \phi_n(vt)\ddot{q}_n(t) \quad (2)$$

Considering the case where the external force is in resonance with the $r-th$ natural vibration mode, the acceleration of the structure can be written as

$$\ddot{u}(t) = \phi_r(vt)A_r\sin(\omega_r t - \pi/2) + \sum n(vt)A_n\sin(\omega_r t - \theta_n) \quad (3)$$

The acceleration response is an infinite summation of sine functions with the same frequency but different amplitudes and phases. The amplitudes of the sinusoidal terms vary with time due to the mode shape (or with the position of the sensor). For each sinusoidal term different to the $r-th$ mode, the phase θ_n is constant and different to $\pi/2$.

Using only the peak values of the $\sin(\omega_r t - \pi/2)$ function to exclude the influence of the non-resonance modes, the $r-th$ mode can be identified as

$$\phi_r(x) = \phi_r(vt) = \frac{u(t)}{\sin\left(\omega_r t - \frac{\pi}{2}\right)}; \quad \text{for } t = \frac{k\pi}{\omega_r}; \quad k = 1, 2, 3 \ldots \quad (4)$$

The methodology has been expanded to impulsive loads and ambient vibration. In general the methodology can be expressed with the equation

$$\Phi = Q^{-1}U \quad (5)$$

Where the matrix Φ contains the modes of vibration, the matrix Q contains the response of the structure in generalized coordinates, and U is a matrix containing the response in geometric coordinates. Notice that the matrix Q does not change with time, can be computed ahead of time and it is dependent on the type of excitation only (i.e. Q for impulse loads is different than for ambient vibration). However, the target natural frequencies and damping ratios must be previously identified to calculate the matrix Q. The matrix U is formed using cross-correlation functions for the case of ambient vibration, based on the fundamentals of the Natural Excitation Technique (James, Carne et al. 1992). Although the proposed methodology is presented for one-dimensional systems, it is also applicable to two dimensional systems such as shells.

3 NUMERICAL EVALUATION

A uniform simply supported beam is considered in this section to numerically evaluate the proposed technique. The $n-th$ mode shape of a uniform simply supported beam and its corresponding natural frequency are described by the equations (Chopra 2007):

$$\phi_n(x) = \sin\left(\frac{n\pi x}{L}\right) \quad \& \quad \omega_n = \frac{n^2\pi^2}{L^2}\sqrt{\frac{EI}{m_l}} \quad (6)$$

where L is the total length of the beam, E is the elasticity modulus of the material, I is the moment of inertia of the cross section, and m_l is its mass per unit length. Assuming $L = 60m$, $E = 25GPa$, $I = 6.75m^4$ and $m_l = 150kN/g/m$, the first five natural

frequencies of the beam are 1.45, 5.80, 13.05, 23.19 and 36.24 Hz. A damping ratio of 5% is assumed for all the modes.

The beam is excited with a sinusoidal load with amplitude of 1 kN and frequency equal to the 1st and 5th natural frequencies for the case of sinusoidal excitation. 10 modes of vibration are used to calculate the response of the beam at $x = vt$, simulating the sensor moving at a constant velocity of 3 m/s. The comparison between the identified 1st and 5th mode shapes with the corresponding theoretical modes is shown in Figure 1. The velocity of the sensor and the resonance frequency determine the number of points in the modal identification. A total of 57 and 1449 points were obtained for the 1st and 5th modes, respectively, with a single 20 seconds acceleration signal sampled at 200 Hz. MAC values (Allemang 2003) of 1.000 and 0.999 were calculated between the 1st and 5th identified modes and the theoretical ones. Similar results were obtained for the other 8 modes of vibration considered on the simulations.

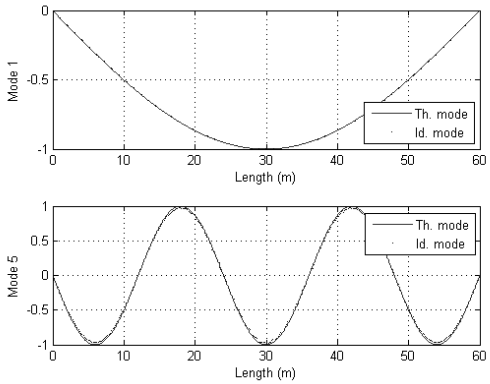


Figure 1. Theoretical and identified first and fifth modes

A distributed force with a random profile defined by zero mean and 1N standard deviation normally distributed random numbers is used to excite the structure for the random excitation case. A total of 2×10^6 points are used to define the load profile.

Theoretically, the methodology allows the identification of a large number modal coordinates for each mode shape. However, noise in the sensors, the speed of the mobile sensor and the natural frequency of the structure play a role on the accuracy of the identified modal coordinates. In this work, only noise-free numerical simulations are used to validate the methodology. In this case, a total of 59 modal coordinates, one each meter, were targeted. The simulation is performed with a stationary sensor at 47.5m from the left support and one mobile sensor with a velocity of 1.67cm/s. Both sensors have a sampling frequency of 600Hz and are resampled to 30Hz before the methodology is applied. At this stage, the exact values of natural frequencies are assumed to be known to verify that the methodology works. The identified first three mode shapes are shown in Figure 2, together with the corresponding theoretical ones. All MAC values for these modes correspond to 1.000.

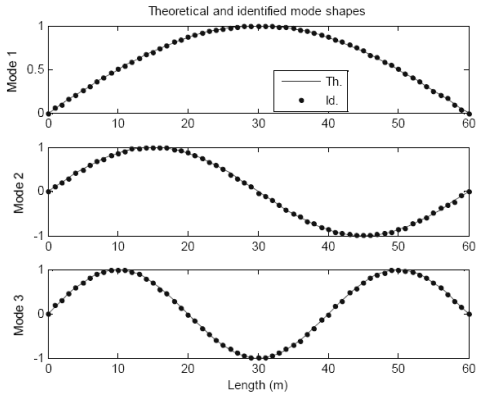


Figure 2. Theoretical and identified first three mode shapes (ambient vibration).

4 EXPERIMENTAL VALIDATION

A 15.24 cm × 0.32 cm × 121.92 cm (6 in × 1/8 in × 48 in) simply supported steel beam was used for the experimental validation of the methodology for the sinusoidal excitation case. Capacitive PCB 3701D1FA20G accelerometers were used to measure the vibration of the beam, each one with a PCB 478A01 signal conditioner. A Measurement Computing 12-bits USB-1208LS module was used as acquisition system with a sampling frequency of 400 Hz. The baseline natural frequencies and mode shapes of the beam were obtained using the Stochastic Subspace Identification (SSI) method (van Overschee and De Moor 1996; Brincker and Andersen 2006).

Impact excitation tests using a PCB 086C03 general purpose modal analysis impact hammer were initially performed to identify the first two natural frequencies. Five impact tests of one minute length, using a combination of short and long intervals between hits, were performed for each testing location. The mean value of the first natural frequency is 4.681 Hz with a standard deviation of 0.018 Hz. The mean value of the second natural frequency is 18.862 Hz with a standard deviation of 0.171 Hz.

A Brüel & Kjær vibration exciter type 4809, located at 12 inches from the left support, was used to identify the operational mode shape of the beam. The beam was excited with the actuator tuned at a constant forcing frequency of 18.8 Hz, controlled by a Quanser Q8 data acquisition and control board, and a Brüel & Kjær power amplifier. Five tests of 30 seconds length were performed for each testing location. Mean values and standard deviations of the modal coordinates of the operational mode shape were calculated using the SSI method, and are shown in Figure 3.

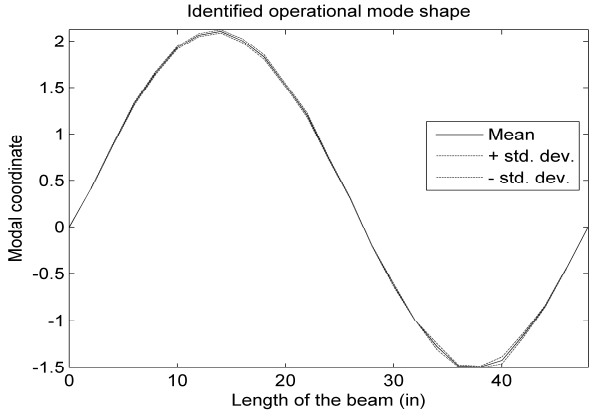


Figure 3. Identified mode shape using the SSI.

The methodology was experimentally implemented with the previously described setup, using a small cart with a constant speed motor (Figure 4). An accelerometer was attached to the cart and it was moved from one end of the beam to the other. The cart was reinforced with an aluminum plate to reduce local vibrations. Four magnetic wheels with 2.395 inches of diameter and 0.25 inches of thickness were used to avoid the cart jumping.

Figure 4. Experimental setup (sinusoidal excitation).

Figure 5 shows the identified operational mode shape using the methodology. A total of 32 points were obtained for constructing the identified mode with a single 12.6 seconds signal. A MAC value of 0.969 was obtained between the operational mode shape identified using the standard approach and the one using the MIMS methodology. Differences between the mode shapes are mostly in the right half of the mode because the mass of the cart is modifying the system dynamics. A smaller effect was found close to the actuator. The weight per unit length of the beam is 0.210 pounds per inch and the weight of the cart is 1.209 pounds, distributed in the front and rear axis which are separated by 2.5 inches.

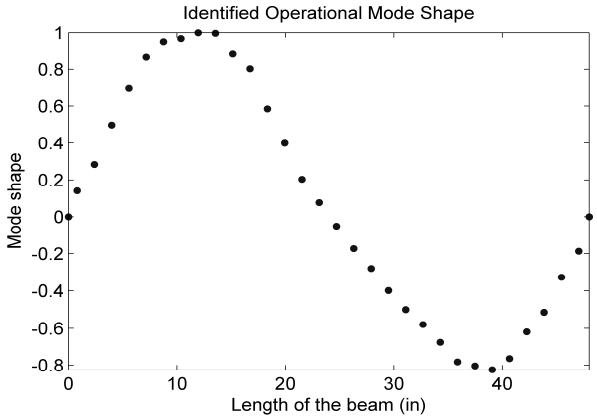


Figure 5. Identified mode shape using the methodology for sinusoidal excitation.

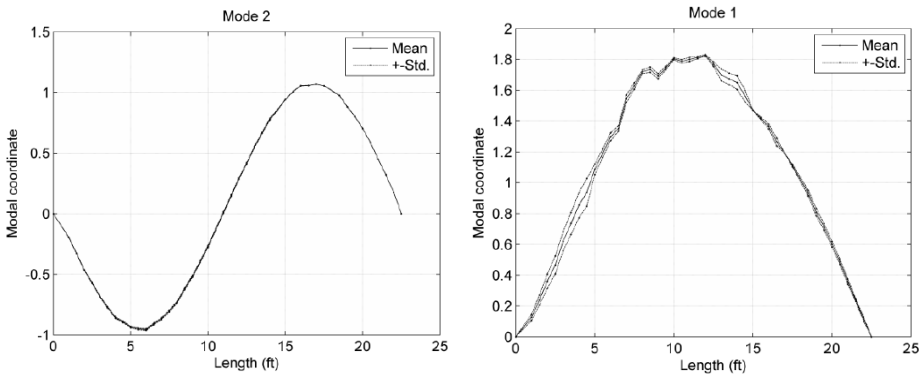
A 7.010m (23ft) length W14x53 simply supported steel beam was used to experimentally validate the technique for random excitation. The beam supports are at 0.152m (0.5ft) from each end as shown in Figure 6. A Brüel & Kjær electro dynamic shaker was used to excite the beam vertically with a 100-Hz band-limited white noise signal generated using a National Instruments 6062E data acquisition card. The shaker was clamped to the bottom of the beam at 1.118m (44in) from end A.

Figure 6. Experimental setup (random excitation).

Table 1. Identified natural frequencies and damping ratios (random excitation).

Mode	Frequency (Hz)	Std. Dev. Frequency (Hz)	Damping (%)	Std. Dev. damping (%)
1	9.448	0.075	1.089	1.455
2	30.758	0.100	0.258	0.214
3	67.959	0.521	0.472	0.253

A baseline modal identification was performed using the SSI method. As described in (Caicedo and Marulanda 2010), 43 locations, one every 0.152m (0.5ft), were selected to place stationary sensors and identify the natural frequencies, damping ratios and mode shapes of the beam. The identification process using the SSI method was repeated several times to calculate mean values and standard deviations for the natural frequencies, damping ratios and mode shapes of the beam. An algorithm to automatically recognize the true poles from the ones created by noise and numerical errors was used (Giraldo, Song et al. 2009). The final results are summarized in Table 1 and Figure 7. Errors in the identification of the first mode shape are due to the 10Hz low limit of the shaker. The asymmetry of the modes along the axis of the beam is caused by the additional mass of the shaker's attachment.



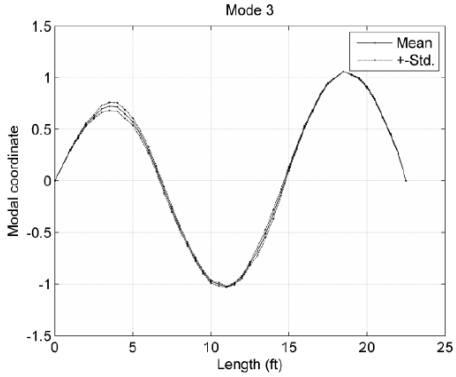


Figure 7. First three identified mode shapes using SSI method (random excitation).

The same mobile sensor used for the sinusoidal excitation case was used here for the validation of the methodology for random excitation. A second stationary sensor was attached to the beam at 5.486 m (18 ft) from end A to use it as a reference sensor. In addition to the excitation provided by the shaker, the beam was also excited using one rubber mallet hitting randomly at the same location of the shaker. This additional excitation was used to increase the signal-to-noise ratio of the mobile sensor. A sampling rate of 280 Hz was used for both sensors.

The first step in the methodology is the identification of the natural frequencies of the structure. In this case the first three natural frequencies were identified using the SSI method with the signal from the stationary sensor. The identified natural frequencies are 9.271, 30.689 and 67.921, which differ 1.87, 0.23 and 0.06% from the results of the baseline identification, respectively. The corresponding mode shapes were identified and are shown in Figure 8, together with the baseline identified modes using the SSI method. Twenty one modal coordinates were targeted for the identified modes using the proposed methodology. MAC values between the baseline and the identified modes using smart mobile sensing units are 0.988, 0.993 and 0.992.

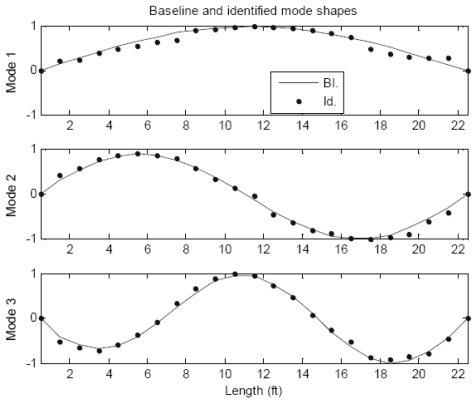


Figure 8. First three baseline and identified modes (random excitation).

5 CONCLUSIONS

A dynamic characterization using mobile sensors methodology was derived and verified using numerical and experimental data. Numerical evaluations were successfully performed for the sinusoidal, impulse and random excitation cases. Experimental tests were performed for the validation of the sinusoidal and random excitation cases using simply supported steel beams. Detailed standard modal identification procedures were

performed to identify the dynamic behavior of the beam and compare the results with results from the proposed methodology.

The number of points in the modal identification depends on the velocity of the sensor and the frequency of the identified mode, for the sinusoidal excitation case. In this particular case, from 57 to 1449 points were obtained with a single 20 seconds simulated signal. MAC values of 1.000 and 0.999 were obtained for the first and fifth mode, respectively. Similar results were obtained for the other 8 modes of vibration considered on the simulation. In the experimental validation, a total of 32 points were obtained for the identified mode shape with a single 12.6 seconds signal. A MAC value of 0.969 was obtained between the operational mode shape identified using the standard approach and the one using the methodology. Differences between the two identified mode shapes are mostly due to the influence of the mass of the car over the beam. This is not expected in full-scale implementations on civil structures.

In the numerical simulations for the random excitation case, MAC values of 1.00 for the first three modes, indicating a perfect match between the theoretical and identified modes were achieved with the methodology. A total of 59 modal coordinates were identified using only two acceleration records. In the experimental validation, twenty one modal coordinates were identified for the first three mode shapes of the beam using two 64 seconds signals. One signal was measured by the mobile sensor and the second signal by the stationary sensor. MAC values of 0.988, 0.993 and 0.992 were obtained between the mode shapes identified using the standard approach and the ones using the methodology. A total of 21 sensors would be required to obtain the same spatial density with the current state-of-the-art modal identification techniques. Furthermore, additional modal coordinates can be obtained by reducing the speed of the mobile sensing unit and/or increasing the signal-to-noise ratio.

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