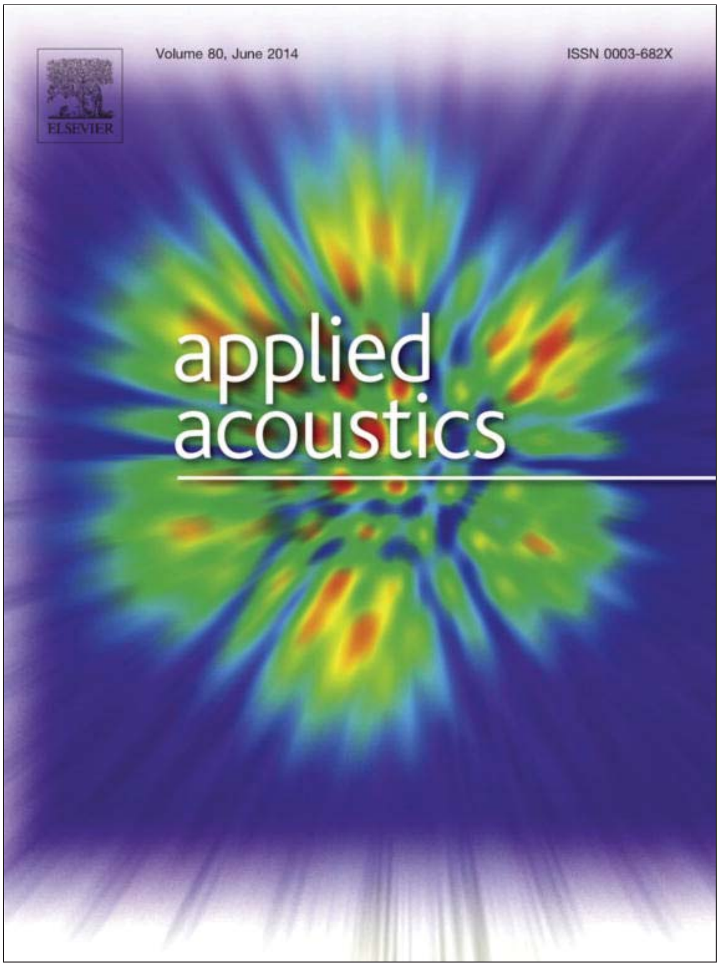
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Acoustic-based damage detection method

V. Arora [a,*], Y.H. Wijnant [b], A. de Boer [b]

[a] Department of Technology and Innovation, University of Southern Denmark, Denmark
[b] Structural Dynamics and Acoustics, University of Twente, Netherlands

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ABSTRACT

Most of the structural health monitoring (SHM) methods is either based on vibration-based and contact acoustic emission (AE) techniques. Both vibration-based and acoustic emission techniques require attaching transducers to structure. In many applications, such as those involving hot structural materials for thermal protection purposes or in rotating machines, non-contact measurements would be preferred because the operating environment is prohibitive leading to potential damage in contact sensors or their attachments. In this paper, a new non-contact, acoustic-based damage detection method is proposed and tested with an objective that the proposed method is able to detect the location and extend of damage accurately. The proposed acoustic-based damage detection method is a direct method. In this proposed method, changes in vibro-acoustics flexibility matrices of the damage and health structure are used to predict the location and extend of damage in the structure. A case study involving actual measured date for the case of a fixed–fixed plate structure is used to evaluate the effectiveness of the proposed method. The results have shown that the proposed acoustic-based damage detection method can be used to detect the location and extend of the damage accurately.

1. Introduction

Structural health monitoring (SHM) is a process of damage identification by nondestructive evaluation (NDE). Structural health monitoring techniques can be operated off-line as well as on-line. Presently, periodic visual inspection, vibration-based, acoustic emission (AE) and wave propagation techniques have been widely used for assessing the condition of an infrastructure component. Visual inspection has many limitations such as subjectivity, and lack of ability to capture internal defects. Most of the vibration-based techniques require attaching transducers to structural materials [1,2]. Contact acoustic emission (AE) [3,4] and wave propagation-based techniques [5] have been used to characterize, passively, crack formation and growth in structural materials ranging from metals to concrete. Both AE and wave propagation methods also require attaching transducers to structural materials in order to transmit and receive the acoustic signal and also both the methods are being used for simple structure and damage detection on local surfaces. In many applications, such as those involving hot structural materials for thermal protection purposes or in rotating machines, non-contact measurements would be preferred because the operating environment is prohibitive (e.g., high temperature, high centrifugal force levels) leading to potential damage in contact sensors or their attachments. There are some examples of optical techniques, such as laser-based techniques that do not require contact measurements. These laser-based methods are very expensive with respect to hardware and computational effort and more importantly data acquisition is time consuming [6]. In this paper, the health of the structure is monitored by acoustic means. The pressure distribution of the vibrating structure can be used for health monitoring. For example, in thermal protection system (TPS) applications where acoustic pressures are experienced during launch and reentry, measurements of acoustic pressure behind the TPS panel could be used to identify damage to the panel. Frendi et al. [7] studied the problem of coupling between panel vibration and near and far field acoustic radiation. Ruzzene [8] analyzed the coupling between vibration and sound radiation of a sandwiched beam. Miksis and Ting [9] and Junger [10] introduced the basic concepts of the interaction of an acoustic wave with an elastic panel. Mucchi and Vecchio [11] used the microphones on the outer surface of the helicopter cabin to study its vibro-acoustic behavior due to noise and vibration of jet engine. Very less literature is available of using acoustic pressure for structural health monitoring. For damage detection, Cherng et al. [12] applied an acoustic methodology to detect changes in plate thickness. Jiang et al. [13] proposed a passive acoustic method for material damage detection using non-contact acoustic pressure measurements. Acoustics becomes more feasible for structural health monitoring for the following reasons. (1) It is a non-contact method for monitoring the health of the structure. (2) Fast data acquisition and data processing [13]. The acoustic

* Corresponding author. Tel.: +45 6550 7372; fax: +45 6550 7384.
E-mail address: viar@iti.sdu.dk (V. Arora).

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pressures are measured with tiny microphones (dimensions of a few mm) that are cheap and offer good quality in a broad frequency range. This makes it possible to develop arrays with a large number of microphones, at a low cost of ownership. The use of a large array with tiny microphones will result in the fast data acquisition and real time processing of the data so that the health of the structure can be monitored in real time.

In this paper, a new acoustic-based method to detect a damaged is proposed. The proposed method is a direct method. The proposed method is an extension of the flexibility method using vibrational data given by Pandey and Barai [2]. In the proposed method, the acoustic pressure responses are measured using microphone, which are subsequently used for vibro-acoustic modal analysis to obtain vibro-acoustics eigendata of the vibrating structure. The obtained vibro-acoustic eigendata is then used to detect damage of the structure by calculating changes in vibro-acoustic flexibility matrices of healthy and damage structure. A case study involving actual measured date for the case of a fixed–fixed plate structure is used to evaluate the effectiveness of the proposed method.

2. Theory

In this section, firstly the dynamic properties of vibro-acoustical system are presented which is followed by theory of the proposed acoustic-based structural health monitoring method.

2.1. Vibro-acoustical systems

In this sub-section, the dynamic properties of vibro-acoustical systems are presented. In particular, the vibro-acoustic modeshape and transfer function considering acoustic loading as excitation and response both as structural displacement the acoustic pressure are derived. The equations describing vibro-acoustical interaction between structures and enclosed cavity can be obtained from finite element formulations [15,16]. The second-order coupled equations can be written in a matrix form as follows:

$$\begin{bmatrix} K^s & -K^c \\ 0 & K^f \end{bmatrix} \begin{Bmatrix} x \\ p \end{Bmatrix} - i\omega \begin{bmatrix} C^s & 0 \\ 0 & C^f \end{bmatrix} \begin{Bmatrix} x \\ p \end{Bmatrix} - \omega^2 \begin{bmatrix} M^s & 0 \\ M^c & M^f \end{bmatrix} \begin{Bmatrix} x \\ p \end{Bmatrix} = \begin{Bmatrix} f \\ \rho\dot{q} \end{Bmatrix} \tag{1}$$

where K, M and C are stiffness, mass and damping matrices respectively. Superscript s, f and c are structural, fluid part and coupling between structure and fluid respectively. x is structural displacement, p is sound pressure, ω is angular frequency, ρ is the fluid density, f is structural force and $\dot{q}$ is the volume acceleration. Eq. (1) can be used for modal analysis. It is clear that Eq. (1) is non-symmetrical and rewriting Eq. (1) in the following compact matrix form makes it clearer:

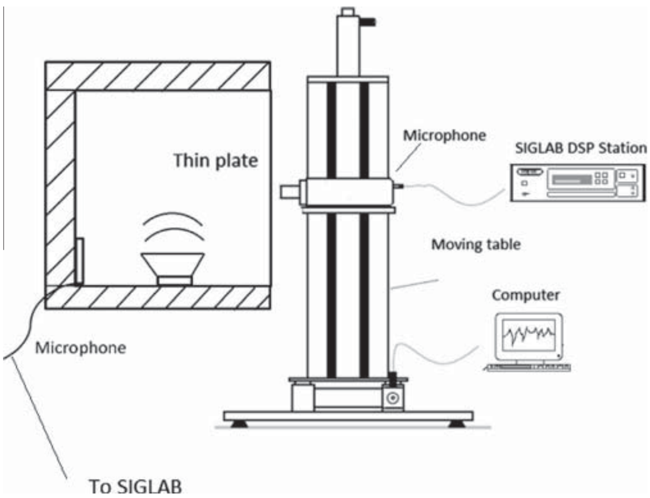

Fig. 1. Schematic diagram of experimental setup.

Fig. 2. Crack on the flexible plate.

$$\begin{bmatrix} A^s & -K^c \\ -\omega^2 K^{c^t} & A^f \end{bmatrix} \begin{Bmatrix} x \\ p \end{Bmatrix} = \begin{Bmatrix} f \\ \dot{q} \end{Bmatrix} = B \begin{Bmatrix} x \\ p \end{Bmatrix} \tag{2}$$

with:

$$A^s = K^s - i\omega C^s - \omega^2 M^s \tag{3}$$

$$A^f = \left(K^f - i\omega C^f - \omega^2 M^f\right)/\rho \tag{4}$$

$$B = \begin{bmatrix} A^s & -K^c \\ -\omega^2 K^{c^t} & A^f \end{bmatrix} \tag{5}$$

From general modal analysis theory [17] the following transfer function matrix can be written as:

$$H(\omega) = B(\omega)^{-1} \tag{6}$$

Due to the non-symmetry of the matrix B, the right and left vibro-acoustic eigenvectors ϕ_r and ϕ_l are different. The relationship between left and right vibro-acoustic eigenvectors is given as [18]:

$$\begin{Bmatrix} \phi_l^{sva} \\ \phi_l^{fva} \end{Bmatrix}_{\lambda_h} = \begin{Bmatrix} \phi_r^{sva} \\ \frac{1}{\lambda_h^2}\phi_r^{fva} \end{Bmatrix}_{\lambda_h} \tag{7}$$

where the superscript sva is the structural response location, fva is the fluid response location and λ_h are the square root of eigenvalue of the system. The transfer functions obtained considering the structural force excitation f_j at the location j. Structural responses x_m at location m and acoustical pressure response p_p inside the cavity at loaction p have been derived in [14]:

$$\frac{x_m}{f_j} = \sum_{h=1}^{n} \frac{(P_h)(\phi_{hm}^{sva})(\phi_{hj}^{sva})}{(z-\lambda_h)} + \frac{(P_h)^*(\phi_{hm}^{sva})^*(\phi_{hj}^{sva})^*}{(z-\lambda_h^*)} \tag{8}$$

$$\frac{p_p}{f_j} = \sum_{h=1}^{n} \frac{(P_h)(\phi_{hp}^{fva})(\phi_{hj}^{sva})}{(z-\lambda_h)} + \frac{(P_h)^*(\phi_{hp}^{fva})^*(\phi_{hj}^{sva})^*}{(z-\lambda_h^*)} \tag{9}$$

Similarly, transfer functions can be obtained considering acoustical excitation $\dot{q}$ at location k:

$$\frac{x_m}{\dot{q}} = \sum_{h=1}^{n} \frac{(P_h)(\phi_{hm}^{sva})(\phi_{hk}^{fva})}{\lambda_h^2(z-\lambda_h)} + \frac{(P_h)^*(\phi_{hm}^{sva})^*(\phi_{hk}^{fva})^*}{\lambda_h^2(z-\lambda_h^*)} \tag{10}$$

$$\frac{p_p}{\dot{q}} = \sum_{h=1}^{n} \frac{(P_h)(\phi_{hp}^{fva})(\phi_{hk}^{fva})}{\lambda_h^2(z-\lambda_h)} + \frac{(P_h)^*(\phi_{hp}^{fva})^*(\phi_{hk}^{fva})^*}{\lambda_h^2(z-\lambda_h^*)} \tag{11}$$

This modal description given in Eqs. (8)–(11) can be used for obtaining vibro-acoustic eigendata that is eigenvectors, eigenvalues and damping factors.

2.2. Acoustic-based damage detection method

The proposed method of using acoustic pressure response is an extension of the flexibility method using vibrational data given by Pandey and Barai [2]. In this proposed method, pressure response is measured by microphone. The distance between the microphone and the external surface of plate is so short that the very near field assumption is verified that is particle velocities simply represent the velocities of the plate surface in the normal direction. The vibro-acoustic modal analysis of the acoustic pressure data is carried out to obtain the vibro-acoustic eigenvalues and eigenvectors of the plate surface. The vibro-acoustic eigenvalues and eigenvectors are subsequently used to obtain the vibro-acoustic flexibility matrices of damaged and undamaged structures. Flexibility is defined as an inverse of structure's stiffness matrix. Changes in flexibility matrix between damage and undamaged structure can work as an indicator for estimating both the presence of damage and also its location.

The structural vibro-acoustic (*sva*) eigen solution of a system consists of the vibro-acoustic eigenvalue matrix $[\Lambda^{sva}]$, which is a diagonal matrix of the squared vibro-acoustic natural frequency, $diag(\lambda_r^{sva^2})$, and the structural vibro-acoustic eigenvector matrix $[\Phi^{sva}] = [\phi_1^{sva} \quad \phi_2^{sva} \quad \ldots \quad \phi_r^{sva}]$, ϕ_r^{sva} is the rth structural vibro-acoustics mode shape, which is mass-normalized, i.e. scaled such that:

$$[\Phi^{sva}]^T[K^s][\Phi^{sva}] = [\Lambda^{sva}] \tag{12}$$

$$[\Phi^{sva}]^T[M^s][\Phi^{sva}] = [I] \tag{13}$$

in which $[K^s]$ and $[M^s]$ are the structural stiffness and mass matrices respectively, and the superscript *sva* represents structural vibro-acoustic data. Solving Eq. (12), the structural stiffness matrix of the dynamic system can be written in the structural vibro-acoustic eigen system as:

Table 1
Natural frequencies of the damaged and healthy plate.

Mode no.	Analytical healthy plate frequency (Hz)	Experimental		
		Healthy plate frequency (Hz) (acoustic-based)	Damaged plate frequency (Hz) (acoustic-based)	% Change in frequency
1.	277.2	275.1	260.2	5.42
2.	470.1	451.3	443.8	1.66
3.	648.5	626.3	614.4	1.9
4.	786.4	759.4	670	11.77
5.	826.2	802.5	796.3	0.77

$$[K^s] = [\Phi^{sva}]^{-T}[\Lambda^{sva}][\Phi^{sva}]^{-1} \tag{14}$$

Flexibility is defined as inverse of structural stiffness matrix:

$$[F^s] = [K^s]^{-1} \tag{15}$$

Substituting Eqs. (14) into (15) yields the inverse structural vibro-acoustic representation of the flexibility matrix:

$$[F^s] = [\Phi^{sva}]^T[\Lambda^{sva}]^{-1}[\Phi^{sva}] = \sum_{i=1}^{tu}\{\phi_r^{sva}\}^T \frac{1}{\lambda_r^{sva^2}}\{\phi_r^{sva}\} \tag{16}$$

tu is the total number of modes. Since the flexibility matrix $[F^s]$ is defined as the inverse of the stiffness matrix $[K^s]$ and this will satisfy the equation:

$$[F_d^s][K_d^s] = [F_u^s][K_u^s] = [I] \tag{17}$$

where I is the identity matrix and subscript d and u represents damaged and undamaged structures. It can be observed from Eq. (14) modal contribution to structural stiffness matrix $[K^s]$ increases as frequency increases while it is reverse for the flexibility. Therefore the flexibility will decrease by increasing frequency and converges rapidly with low number of modes. Since it is usually more convenient to measure and calculate the lower-order modes, it is more convenient to use flexibility matrices than stiffness matrices for limited information reconstructions of structural properties. Thus, flexibility matrix, $[F^s]$, of a structure can be estimated with sufficient accuracy by using a few lower frequency modes as:

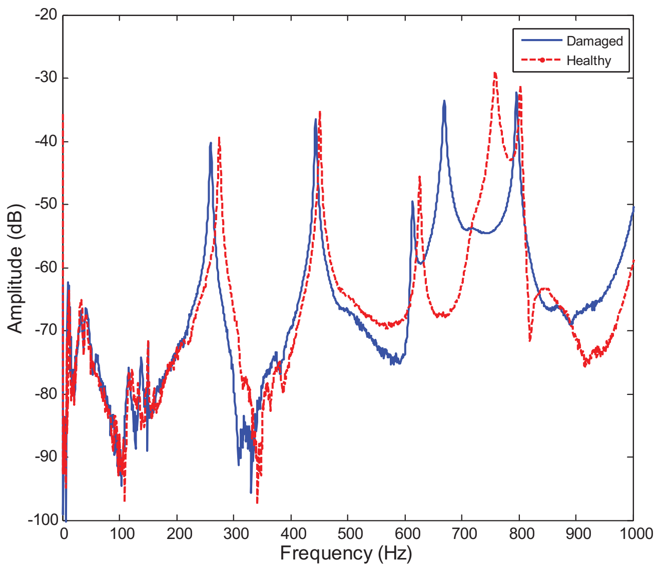

Fig. 3. Pressure response of damaged and healthy structure.

$$[F^s] \approx \sum_{i=1}^{nm} \{\phi_r^{sva}\}^T \frac{1}{\lambda_r^{sva2}} \{\phi_r^{sva}\} \tag{18}$$

where nm is the number of measured modes. Change in structural flexibility which reflects the nature of damage in the structure is defined as:

$$[\Delta F^s] = [F_d^s] - [F_u^s] \tag{19}$$

For each column of $[\Delta F^s]$, the absolute maximum value of the elements in the jth column is,

$$\delta_j = \max |\Delta F^s| \tag{20}$$

The column of matrix, ΔF^s, corresponding to the largest δ_j is the indicative of jth DOF with damage.

3. Experimental case study

An experimental study on an aluminum plate is conducted to evaluate the effectiveness of the acoustics data for structural health monitoring. The schematic diagram of the experimental setup is shown in Fig. 1. An aluminum box having thickness 5 mm is used to bolt the plate thoroughly to a very stiff aluminum box. The outer dimensions of the box are 300 × 220 × 170 mm. The dimensions of the plate are 210 × 160 × 1 mm. The plate is excited by an acoustic field inside the box, generated by a loudspeaker. A microphone near the loudspeaker has been used as reference. The distance between the microphone and the loudspeaker surface is kept short to assume that pressure is simply proportional to velocity in the normal direction. Pressure responses are measured by microphone on the external surface of the plate. Microphone is placed on X–Y directional moving table making a 9 × 9 grid points on the plate. The distance between the microphone and the external surface of the plate is so short that pressure measurement in the near field of the plate is proportional to velocity in the normal direction. The pressure measurements of the plate is carried out on the undamaged plate and then small crack is induced on the plate, without removing it from the box so that boundary conditions do not change as shown in Fig. 2. The depth of the crack in the plate is 0.5 mm whereas width of crack is 40 mm. The change in first natural frequency of the damage plate is 5.42%. Experimental vibro-acoustic modal analysis in the frequency range 0–1000 Hz is carried out. The pressure responses are analyzed using the curve fitting technique available in visual vibro-acoustics module of ME'Scope software [19] to obtain structural vibro-acoustics eigendata. In this case, first 5 vibro-acoustic modes are calculated using ME'Scope software. The vibro-acoustics natural frequencies of the healthy and damaged plate are given in Table 1. It can also be observed in Table 1 that the analytical natural frequencies of the healthy plate are close to the experimental vibro-acoustic natural frequencies. The difference between analytical and experimental frequencies is due to the fact that fixed conditions can be obtained accurately in reality. The healthy plate and damaged plate acoustics pressure response is shown in Fig. 3. It can be observed from Fig. 3 and Table 1 that because of the damage, the vibro-acoustics natural frequencies of the damaged plate are decreased. The maximum change in the vibro-acoustics natural frequency between healthy and damage plate is observed in the 4th vibro-acoustics natural frequency. The first 5 vibro-acoustics modeshapes of healthy and damage plates are plotted in Fig. 4. It can be observed that from Fig. 4 that 4th vibro-acoustics modeshape is significantly affected by the damage of the plate. First 5 vibro-acoustics modes of the damaged and healthy plate are subsequently used to calculate vibro-acoustic flexibility matrices for the damaged and healthy plate and the difference between them indicates the local position of the damage. The calculated maximum structural flexibility using

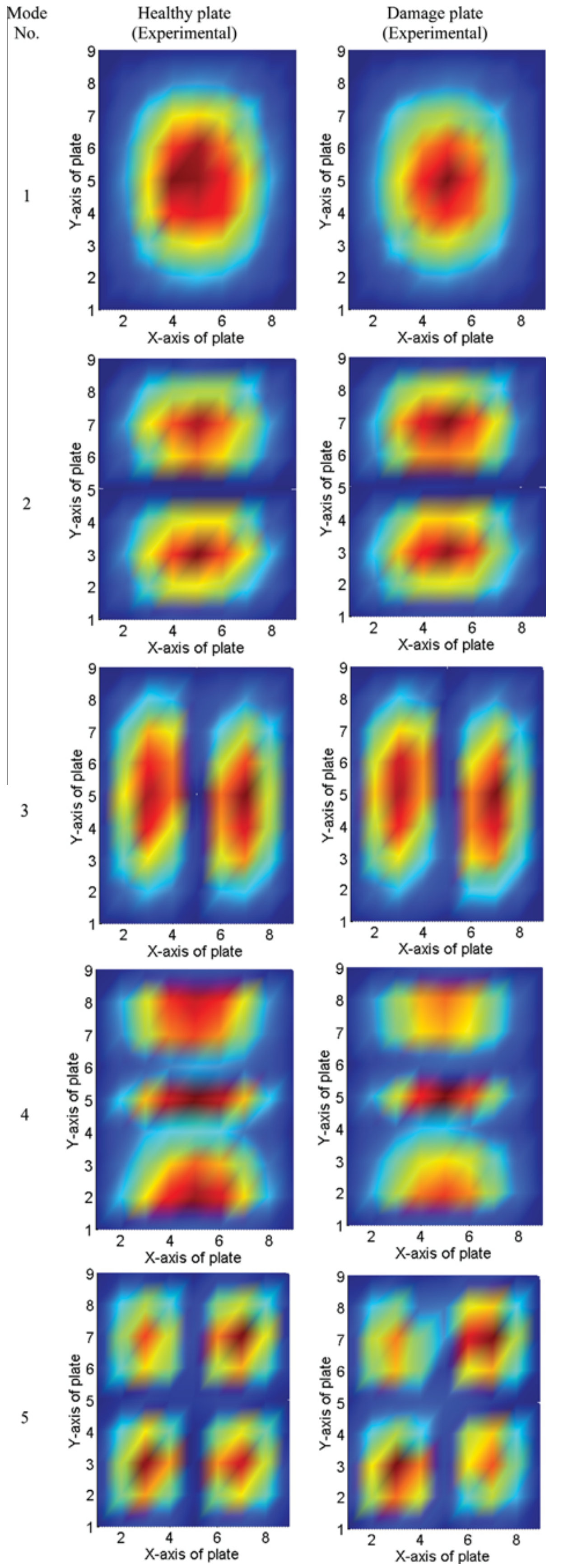

Fig. 4. Experimental first five vibro-acoustic modeshapes of healthy and damaged plate.

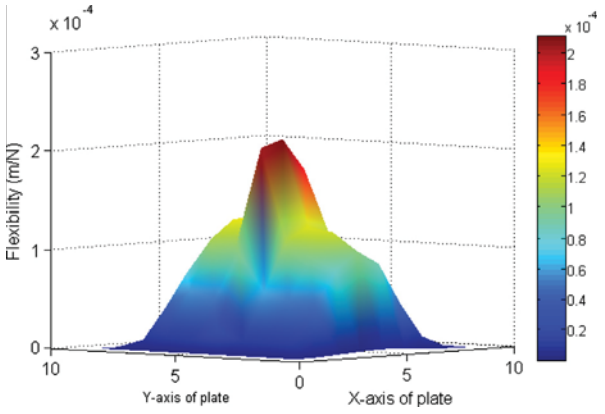

Fig. 5. Flexibility of healthy plate.

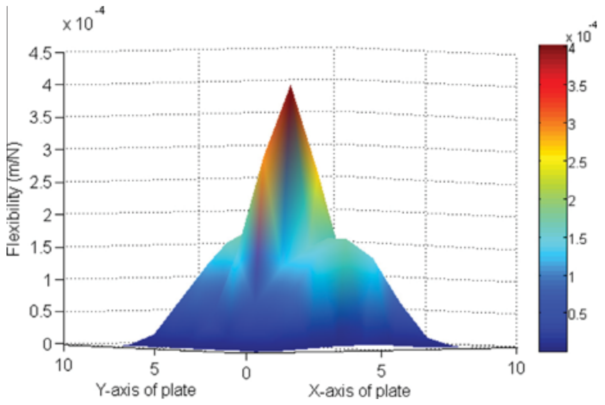

Fig. 6. Flexibility of damaged plate.

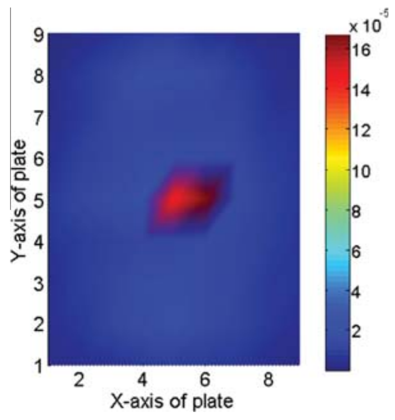

Fig. 7. Damaged detected using 9×9 acoustic grid.

Eq. (20) of the healthy plate and damaged plates using first five vibro-acoustic eigendata are shown in Figs. 5 and 6. It can be observed from Figs. 5 and 6 that vibro-acoustic flexibility is maximum in the center as vibro-acoustic flexibility matrix is significantly influenced by first modes as expected. The maximum values of structural flexibilities for healthy and damage are 2.1×10^{-4} m/N and 4.1×10^{-4} m/N respectively. It can observed from the maximum values of flexibilities of healthy and damage plate that in case of the damage plate maximum value of flexibility is almost 2 times the maximum value of the healthy plate. The difference in flexibility matrices of the damaged and healthy plate is plotted in Fig. 7. It can be observed from Fig. 7 that the vibro-acoustic flexibility method is able to detect local damage using vibro-acoustic data. From the experimental case study, it can be concluded with confidence that proposed acoustic-based damage detection method is able to predict the location of the damage accurately.

4. Conclusions

In this paper, a new method of damage detection using acoustic pressure response is proposed. The proposed method is a direct method. In this proposed method, changes in vibro-acoustics flexibility matrices of the damage and healthy structure are used to predict the location and extend of damage in the structure. The acoustic pressure responses are measured using microphone, which are subsequently used for vibro-acoustic modal analysis to obtain vibro-acoustics eigendata of the vibrating structure. The obtained vibro-acoustic eigendata is subsequently used to calculate changes in vibro-acoustics flexibility matrices of the damage and healthy structure. The difference between the vibro-acoustic flexibility matrices is used to detect the location and extend of damage in the structure. A case study involving actual measured date for the case of a fixed–fixed plate structure is used to evaluate the effectiveness of the proposed method.

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References

[1] Adams RD, Cawley P, Pye CJ, Stone BJ. A vibration technique for non-destructively assessing the integrity of structures. J Mech Eng Sci 1978;20:93–100.
[2] Pandey PC, Barai SV. Multilayer perceptron in damage detection of bridge structures. Comput Struct 1995;54:597–608.
[3] Gudmundson P. Acoustic emission and dynamic energy release rate for steady growth of a tunneling crack in a plate in tension. J Mech Phys Solids 1999;47:2057–74.
[4] Ballad EM, Vezirov SY, Pfleiderer K, Solodov IY, Busse G. Nonlinear modulation technique for NDE with air-coupled ultrasound. Ultrasonics 2004;42:1031–6.
[5] Toutountzakis T, Mba D. Observations of acoustic emission activity during gear defect diagnosis. Non-Destr Test Eval Int 2003;36:471–7.
[6] Pai PF, Oh Y, Lee SY. Detection of defects in circular plates using a scanning laser vibrometer. Struct Health Monit 2002;1:63–88.
[7] Frendi A, Maestrello L, Ting L. An efficient model for coupling structural vibrations with acoustic radiation. J Sound Vib 1995;182:741–7.
[8] Ruzzene M. Vibration and sound radiation of sandwich beams with honeycomb truss core. J Sound Vib 2004;277:741–63.
[9] Miksis MJ, Ting L. Panel oscillations and acoustics waves. Appl Math Lett 1995;8:37–42.
[10] Junger MC. Acoustic fluid-elastic structure interactions: basic concepts. Comput Struct 1997;65:287–93.
[11] Mucchi E, Vecchio A. Acoustic signature analysis of a helicopter cabin in a steady-state and run up operational conditions. Measurement 2010;43:283–97.
[12] Cherng JG, Chen XF, Peng V. Application of acoustic metrology for detection of plate thickness change. Measurement 1996;18:207–14.
[13] Jiang H, Adams DE, Jata K. Material damage modeling and detection in a homogeneous thin metallic sheet and sandwich panel using passive acoustic transmission. Struct Health Monit 2006;5:373–87.
[14] Farshidi R, Trieu D, Park SS, Freiheit T. Non-contact experimental modal analysis using air excitation and a microphone array. Measurement 2010;43:755–65.
[15] Wyckaert K, Augusztinovicz F, Sas P. Vibro-acoustical modal analysis: reciprocity, model symmetry and model validity. J Acoust Soc Am 1996;100:3172–81.
[16] Pierro E, Mucchi E, Soria L, Vecchio A. On the vibro-acoustical operational modal analysis of a helicopter cabin. Mech Syst Sig Process 2009;23:1205–17.
[17] Heylen W, Lammens S, Sas P. Modal Analysis Theory and Testing. Leuven, Belgium: KUL Press; 2007.
[18] Ma ZD, Hagiwara I. Sensitivity analysis methods for coupled acoustical–structural systems. Part I: modal sensitivities. AIAA J 1991;29:1787–95.
[19] ME'Scope, Vibrant Technology, 2008.