JOURNAL OF GUIDANCE, CONTROL, AND DYNAMICS
Vol. 25, No. 3, May–June 2002

Structural Damage Detection Using Virtual Passive Controllers

Jiann-Shiun Lew*
Tennessee State University, Nashville, Tennessee 37203
and
Jer-Nan Juang†
NASA Langley Research Center, Hampton, Virginia 23681

Novel approaches are presented for structural damage detection that use virtual passive controllers attached to structures, where passive controllers are energy dissipative devices and, thus, guarantee closed-loop stability. The use of the identified parameters of various closed-loop systems can solve the problem that the reliable identified parameters, such as natural frequencies, of the open-loop system may not provide enough information for damage detection. Only a small number of sensors are required for the proposed approaches. The identified natural frequencies, which are generally much less sensitive to noise and more reliable than the identified mode shapes, are used for damage detection. Two damage detection techniques are presented. One technique is based on structures with direct output feedback controllers, whereas the other technique uses second-order dynamic feedback controllers. A least-squares technique, which is based on the sensitivity of natural frequencies to damage variables, is used for accurately identifying the damage variables.

Introduction

RELIABLE and efficient techniques for health monitoring and damage detection of large structures, such as spacecraft and aircraft, are essential for safe operation, maintenance cost reduction, and failure prevention. In the last decade, various vibration-based methods have been proposed.[1–6] These methods are more globally sensitive to damage than localized conventional methods such as ultrasonic and eddy current methods.[7] However, the vibration-based algorithms developed so far cannot be considered very efficient and effective. For example, the widely used finite element (FE) model-update techniques[1,2] require many sensors to measure mode shapes, but the number of sensors is limited in practical applications.

In general, the identified mode shapes are much more sensitive to noise and environmental uncertainty than the identified natural frequencies. On the other hand, the natural frequencies are sensitive to structural damage, such as stiffness loss and cracking. Thus, the identified natural frequencies are more reliable for damage detection than the identified mode shapes. In real applications, the identified natural frequencies of the open-loop system may not provide enough information for damage detection because the number of the reliable natural frequencies may be smaller than the number of the possible damage elements. To solve this problem, some researchers have proposed the use of the "twin" structures, where a structure is attached to the tested structure, for damage detection.[8,9] The concept of physical attachment of structures may limit the application of this technique.

In this paper, we use the natural frequencies of the closed-loop systems with virtual passive controllers[10,11] for structural damage detection. Recently, various techniques for the passive controller applications have been developed.[10–15] When a virtual passive controller is applied to a flexible structure, the system is almost always augmented with damping regardless of the system size. In theory, no matter what happens, the controller, which resembles a mass–spring–dashpot, will not destabilize the system because it is an energy dissipative device. In performing the damage detection, the virtual passive controller only uses the existing control devices, and no additional physical elements are attached to the system. The proposed techniques have the advantages of flexibility of controller design and placement.

In this paper, two damage detection techniques based on different control techniques are proposed. First, consider the direct output feedback, implying the absence of the dynamics in the feedback controller. In this circumstance, the number of the natural frequencies of the closed-loop system is the same as that of the open-loop system. Second, assume that the feedback controller contains a set of second-order dynamic equations. It is equivalent to visualize a virtual passive damping system, that is, the feedback controller, that is linked side by side to the real flexible body. In other words, two sets of second-order dynamic equations are coupled to generate a closed-loop system. The number of natural frequencies of the closed-loop system is the summation of the order of the open-loop system and the order of the controller.

In online health monitoring, first, system identification techniques are used to process experimental data to obtain the identified natural frequencies of open-loop and closed-loop systems. Then, a least-squares technique, which is based on the sensitivity of natural frequencies to the variables of damage, is used for detecting the damage variables. Examples are given to demonstrate and verify the presented approaches.

Direct Output Feedback

In this section, we present a damage detection algorithm that is based on a system with a direct output feedback controller. In the analysis and design, the second-order dynamic equation of structural vibration is used,

$$M\ddot{\boldsymbol{x}} + D\dot{\boldsymbol{x}} + K\boldsymbol{x} = B\boldsymbol{u} \tag{1}$$

$$\boldsymbol{y} = C_a\ddot{\boldsymbol{x}} + C_v\dot{\boldsymbol{x}} + C_d\boldsymbol{x} \tag{2}$$

Here $\boldsymbol{x}$ is an $n \times 1$ displacement vector, and M, D, and K are mass, damping, and stiffness matrices, respectively. In the measurement equation, $\boldsymbol{y}$ is the $q \times 1$ measurement vector, and C_a, C_v and C_d are acceleration, velocity, and displacement influence matrices. The measurement equation may be used either directly or indirectly for a feedback controller design. Here we use the direct feedback, and the input vector $\boldsymbol{u}$ is

$$\boldsymbol{u} = -F\boldsymbol{y} = -FC_a\ddot{\boldsymbol{x}} - FC_v\dot{\boldsymbol{x}} - FC_d\boldsymbol{x} \tag{3}$$

Substituting Eq. (3) into Eq. (1) yields

$$(M + BFC_a)\ddot{\boldsymbol{x}} + (D + BFC_v)\dot{\boldsymbol{x}} + (K + BFC_d)\boldsymbol{x} = 0 \tag{4}$$

Received 4 May 2000; revision received 1 May 2001; accepted for publication 21 January 2002.

*Research Professor, Center of Excellence in Information Systems. Senior Member AIAA.

†Principal Scientist, Structural Dynamics Branch. Fellow AIAA.

In this paper, the changes of the identified natural frequencies of the tested system are used for damage detection.

To illustrate the approach, consider the eigenvalue problem of a second-order dynamic system without a damping term. The eigensystem of the open-loop system can be written as

$$\left[\omega_{0i}^2 M(z) + K(z)\right]\phi_i = 0 \tag{5}$$

where ω_{0i} is the ith natural frequency corresponding to the ith eigenvector ϕ_i and z is the vector of damage variables such as the stiffness losses of elements. The r-dimensional damage vector z is defined as

$$z = [z_1 \quad z_2 \quad \ldots \quad z_r]^T \tag{6}$$

where z_i is the ith damage variable, for example, the value of z_i is 1 when the ith element has 0% stiffness loss and the value of z_i is 0.5 when the ith element has 50% stiffness loss. The natural frequency vector of the open-loop system is defined as

$$\omega_0 = [\omega_{01} \quad \omega_{02} \quad \ldots \quad \omega_{0n}]^T \tag{7}$$

There are n natural frequencies for this second-order dynamic system. The number r of the damage variables may be larger than n. In this situation, the use of the natural frequencies of the open-loop system may not provide enough information to identify the r-dimensional damage vector z. To solve this problem, we include the identified natural frequencies of the m closed-loop systems with different direct feedback controllers. The eigensystem of the jth closed-loop system is expressed as

$$\left[\left(\omega_{ci}^j\right)^2 M_t^j(z) + K_t^j(z)\right]\phi_i = 0 \tag{8}$$

where ω_{ci}^j is the ith eigenvalue of the jth closed-loop system and M_t^j and K_t^j are mass and stiffness matrices of the jth closed-loop system, respectively. Each closed-loop system has n natural frequencies. The natural frequency vectors of these m closed-loop systems are computed as

$$\begin{aligned}
\omega_c^1 &= \left[\omega_{c1}^1 \quad \omega_{c2}^1 \quad \cdots \quad \omega_{cn}^1\right]^T \\
\omega_c^2 &= \left[\omega_{c1}^2 \quad \omega_{c2}^2 \quad \cdots \quad \omega_{cn}^2\right]^T \\
&\vdots \\
\omega_c^m &= \left[\omega_{c1}^m \quad \omega_{c2}^m \quad \cdots \quad \omega_{cn}^m\right]^T
\end{aligned} \tag{9}$$

Then the system natural frequency vector, which includes the open-loop system and the m closed-loop systems, is defined as

$$\omega = \left[\omega_0^T \quad \left(\omega_c^1\right)^T \quad \left(\omega_c^2\right)^T \quad \cdots \quad \left(\omega_c^m\right)^T\right]^T \tag{10}$$

From the Taylor's series expansion, the natural frequency vector ω can be expressed as a function of damage variables

$$\omega(z + \Delta z) = \omega(z) + A(z)\Delta z + \cdots \tag{11}$$

with

$$A = \begin{bmatrix}
\dfrac{\partial \omega_0}{\partial z_1} & \dfrac{\partial \omega_0}{\partial z_2} & \cdots & \dfrac{\partial \omega_0}{\partial z_r} \\
\dfrac{\partial \omega_c^1}{\partial z_1} & \dfrac{\partial \omega_c^1}{\partial z_2} & \cdots & \dfrac{\partial \omega_c^1}{\partial z_r} \\
\vdots & \vdots & \ddots & \vdots \\
\dfrac{\partial \omega_c^m}{\partial z_1} & \dfrac{\partial \omega_c^m}{\partial z_2} & \cdots & \dfrac{\partial \omega_c^m}{\partial z_r}
\end{bmatrix} \tag{12}$$

where A is the $n(m+1) \times r$ sensitivity matrix of natural frequency to damage variables. The sensitivity of the closed-loop system can be enhanced by choosing feedback controllers.[16] The number m is chosen to satisfy the inequality

$$n(m+1) \geq r \tag{13}$$

The linear approximation of Eq. (11) is

$$\omega(z + \Delta z) \approx \omega(z) + A(z)\Delta z \tag{14}$$

Equation (14) can be written as

$$A(z)\Delta z \approx \omega(z + \Delta z) - \omega(z) = \Delta\omega \tag{15}$$

The least-squares techniques can be used to obtain the approximated solution of Δz as

$$\Delta z \approx (A^T A)^{-1} A^T \Delta\omega \tag{16}$$

To detect the damage variables z_i, $i = 1, \ldots, r$, accurately, we apply the following procedures:

1) Compute the updated natural frequency vector $\omega(z_{\text{new}})$ [from Eqs. (5) and (8)] as a function of the updated z_{new}, where the initial z_{new} corresponds to the healthy structure.

2) Compute the difference between the identified natural frequency vector ω_t, which corresponds to the tested system, and the updated vector $\omega(z_{\text{new}})$ as

$$\Delta\omega = \omega_t - \omega(z_{\text{new}}) \tag{17}$$

3) Compute the sensitivity matrix $A(z_{\text{new}})$.

4) Use the linear approximation to compute the updated variables

$$\Delta z = (A^T A)^{-1} A^T \Delta\omega \tag{18}$$

$$z_{\text{new}} = z_{\text{old}} + \Delta z \tag{19}$$

5) Check if $|\Delta z| \leq$ precision error specified. If yes, stop; if no, go to procedure 1.

The parameters used for damage detection are not limited to the identified natural frequencies. For example, consider an n-degrees-of-freedom spring–mass system with single input and l displacement outputs. The transfer functions of the open-loop system are

$$g_j(s) = \sum_{i=1}^{n} \frac{b_{ji}}{s^2 + \omega_{0i}^2}, \qquad j = 1, 2, \ldots, l \tag{20}$$

where g_j is the transfer function corresponding to the jth displacement sensor. The parameter vector of this open-loop system is defined as

$$p_0 = [\omega_{01} \quad \cdots \quad \omega_{0n} b_{11} \quad \cdots \quad b_{1n} \quad \cdots \quad b_{l1} \quad \cdots \quad b_{ln}]^T \tag{21}$$

The dimension of the parameter vector p_0 is $(l+1)n$. We can also include the parameter vectors p_c^j, $j = 1, \ldots, m$, corresponding to the m closed-loop systems, and then define the system parameter vector as

$$p = \left[p_0^T \quad \left(p_c^1\right)^T \quad \left(p_c^2\right)^T \quad \cdots \quad \left(p_c^m\right)^T\right]^T \tag{22}$$

The augmented parameter vector p can then be used for the identification of the damage vector z.

In the least-squares procedures, the updated variables in Eq. (19) can be computed as[17,18]

$$z_{\text{new}} = z_{\text{old}} + \alpha\Delta z \tag{23}$$

with

$$\Delta z = (A^T A)^{-1} A^T \Delta\omega$$

where α is the learning rate,[17] which is chosen to make the difference between the updated ω and the identified ω_t smaller. In the design process, we choose controllers and the number of the closed-loop systems to make $A^T A$ full rank without ill condition. Other optimization techniques, such as Newton's method and the conjugate gradient method,[17,18] can also be applied to compute the solution of z and solve the problem of the singularity of $A^T A$.

Controller with Second-Order Dynamics

In this section, we present a damage detection method in which the feedback controller is described as a set of second-order dynamic equations

$$M_c\ddot{x}_c + D_c\dot{x}_c + K_c x_c = B_c u_c \tag{24}$$

$$y_c = C_{ac}\ddot{x}_c + C_{vc}\dot{x}_c + C_{dc} x_c \tag{25}$$

Here x_c is the controller state vector of dimension n_c, and M_c, D_c, and K_c are the controller mass, damping, and stiffness matrices, respectively. The quantities M_c, D_c, K_c, C_{ac}, C_{vc}, and C_{dc} are the design parameters for the controller. Let the input vectors u and u_c be

$$u = y_c = C_{ac}\ddot{x}_c + C_{vc}\dot{x}_c + C_{dc} x_c \tag{26}$$

$$u_c = y = C_a\ddot{x} + C_v\dot{x} + C_d x \tag{27}$$

Substituting Eq. (26) into Eq. (1) and Eq. (27) into Eq. (24) yields

$$M_t\ddot{x}_t + D_t\dot{x}_t + K_t x_t = 0 \tag{28}$$

where

$$M_t = \begin{bmatrix} M & -BC_{ac} \\ -B_cC_a & M_c \end{bmatrix}, \qquad D_t = \begin{bmatrix} D & -BC_{vc} \\ -B_cC_v & D_c \end{bmatrix} \tag{29}$$

$$K_t = \begin{bmatrix} K & -BC_{dc} \\ -B_cC_d & K_c \end{bmatrix}, \qquad x_t = \begin{bmatrix} x \\ x_c \end{bmatrix} \tag{30}$$

In the controller design, M_c, D_c, K_c, C_{ac}, C_{dc}, and C_{vc} are chosen such that the closed-loop system is stable.[10,11] This closed-loop system has $n + n_c$ natural frequencies. For damage detection, we use the identified natural frequencies of m closed-loop systems with different controllers, where the dimension of the controller state vector x_c is assumed to be a constant n_c for simplicity of presentation. The vectors of natural frequencies of these m closed-loop systems are computed as

$$\begin{aligned} \omega_c^1 &= \begin{bmatrix} \omega_{c1}^1 & \omega_{c2}^1 & \cdots & \omega_{c,n+n_c}^1 \end{bmatrix}^T \\ \omega_c^2 &= \begin{bmatrix} \omega_{c1}^2 & \omega_{c2}^2 & \cdots & \omega_{c,n+n_c}^2 \end{bmatrix}^T \\ &\vdots \\ \omega_c^m &= \begin{bmatrix} \omega_{c1}^m & \omega_{c2}^m & \cdots & \omega_{c,n+n_c}^m \end{bmatrix}^T \end{aligned} \tag{31}$$

Then the natural frequency vector of the m closed-loop systems is defined as

$$\omega = \begin{bmatrix} (\omega_c^1)^T & (\omega_c^2)^T & \cdots & (\omega_c^m)^T \end{bmatrix}^T \tag{32}$$

where ω is a vector of $(n + n_c)m$ dimension. To find the solution of r-dimensional vector z, here m needs to satisfy the following inequality

$$(n + n_c)m \geq r \tag{33}$$

To obtain solutions of the damage variables z_i, $i = 1, \ldots, r$, we use the identified natural frequencies of these m closed-loop systems and apply the least-squares technique in the preceding section. The identified natural frequencies of the open-loop system can also be included for damage detection, and the parameters used for damage detection are not limited to the identified natural frequencies.

Spring–Mass Example

A spring–mass system with two degrees of freedom is used for the study. First, the results with the direct output feedback are presented. Then, the results with the dynamic feedback controller are discussed.

Direct Output Feedback

Consider a spring–mass system with two degrees of freedom shown in Fig. 1. The dynamic equation of this system is

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 1 \end{bmatrix} u \tag{34}$$

Table 1 Case 1 of spring–mass system with direct output feedback

Iteration no.	m_1	m_2	k_1	k_2
1	2.9940	0.9980	49.900	79.840
True	2.9940	0.9980	49.900	79.840

Table 2 Case 2 of spring–mass system with direct output feedback

Iteration no.	m_1	m_2	k_1	k_2
1	4.2087	1.5595	30.733	53.115
2	4.8997	1.9251	30.001	50.324
3	4.9985	1.9980	30.000	50.001
4	5.0000	2.0000	30.000	50.000
True	5.0000	2.0000	30.000	50.000

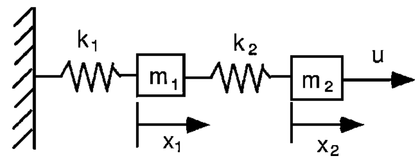

Fig. 1 Two-degrees-of-freedom system.

The values of the four parameters of this system are

$$m_1 = 3, \qquad m_2 = 1, \qquad k_1 = 50, \qquad k_2 = 80$$

When the displacement measurement at x_2 is used, the input u can be expressed as

$$u = -cx_2 \tag{35}$$

Substituting Eq. (35) into Eq. (34) yields

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 + c \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = 0 \tag{36}$$

In this example, the results are based on the analysis of the open-loop system and three closed-loop systems with different output feedback,

$$u_1 = -3x_2, \qquad u_2 = -10x_2, \qquad u_3 = -x_2$$

In the first case, each parameter (m_i or k_i) has a small reduction of 0.2%. To find the solution of these four parameters, we need to use at least two systems because the open-loop system or each closed-loop system has two natural frequencies. Table 1 shows the results when the natural frequencies of the open-loop system and the closed-loop system with the first controller are used. In this minor damage case, each parameter converges to the true one with a negligible error in one iteration. When the natural frequencies of the open-loop system and three closed-loop systems are used, the results are the same as that in Table 1.

In the second case, each parameter has a significant change: m_1 changes from 3 to 5, m_2 changes from 1 to 2, k_1 reduces from 50 to 30, and k_2 reduces from 80 to 50. Table 2 shows the results when the natural frequencies of the open-loop system and the first closed-loop system are used. Each parameter converges to the true value after four iterations when all of the parameters have significant changes.

Controller with Second-Order Dynamics

Consider the preceding two-degrees-of-freedom system with a two-degrees-of-freedom dynamic controller as shown in Fig. 2. The second-order controller design in this case is simply

$$u = k_{c1}(x_{c1} - x_2) \tag{37}$$

where the controller dynamic equations of x_{c1} and x_{c2} are solved by

$$\begin{bmatrix} m_{c1} & 0 \\ 0 & m_{c2} \end{bmatrix}\begin{bmatrix} \ddot{x}_{c1} \\ \ddot{x}_{c2} \end{bmatrix} + \begin{bmatrix} k_{c1} + k_{c2} & -k_{c2} \\ -k_{c2} & k_{c2} \end{bmatrix}\begin{bmatrix} x_{c1} \\ x_{c2} \end{bmatrix} = \begin{bmatrix} 0 & k_{c1} \\ 0 & 0 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} \tag{38}$$

Table 3 Parameters of two passive dynamic controllers

Controller	m_{c1}	m_{c2}	k_{c1}	k_{c2}
1	5	3	200	100
2	1	3	50	80

Table 4 Case 1 of spring–mass system with dynamic controller

Iteration no.	m_1	m_2	k_1	k_2
1	2.9940	0.9980	49.900	79.840
True	2.9940	0.9980	49.900	79.840

Table 5 Case 2 of spring–mass system with dynamic controller

Iteration no.	m_1	m_2	k_1	k_2
1	4.6310	1.5974	17.659	72.878
2	4.5499	1.8666	25.804	42.743
3	5.0150	1.9941	29.737	49.870
4	4.9999	2.0000	29.999	50.000
True	5.0000	2.0000	30.000	50.000

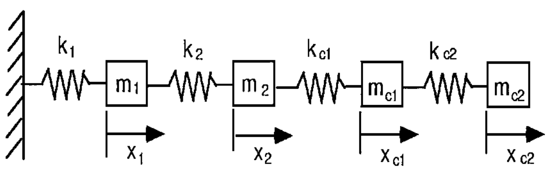

Fig. 2 Two-degrees-of-freedom system with a two-degrees-of-freedom dynamic controller.

The dynamic equation of the closed-loop system is

$$\begin{bmatrix} m_1 & 0 & 0 & 0 \\ 0 & m_2 & 0 & 0 \\ 0 & 0 & m_{c1} & 0 \\ 0 & 0 & 0 & m_{c2} \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_{c1} \\ \ddot{x}_{c2} \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 & 0 & 0 \\ -k_2 & k_2 + k_{c1} & -k_{c1} & 0 \\ 0 & -k_{c1} & k_{c1} + k_{c2} & -k_{c2} \\ 0 & 0 & -k_{c2} & k_{c2} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_{c1} \\ x_{c2} \end{bmatrix} = 0 \qquad (39)$$

This closed-loop system has four natural frequencies, so that only one closed-loop system is required for obtaining the solution of four parameters. The results are based on the analysis of two closed-loop systems with different controllers having the parameters listed in Table 3.

In the first case, the parameter changes are the same as those in case 1 of the preceding direct output feedback example. Table 4 shows the results when the natural frequencies of the first closed-loop system are used. In this minor damage case, each parameter converges to the true one with a negligible error in one iteration. When the natural frequencies of both closed-loop systems are used, the results are the same as that in Table 4.

In case 2 of this example, each parameter has a significant change, which is the same as the one in case 2 of the direct output feedback example. Table 5 shows the results when the natural frequencies of the first closed-loop system are used. All of the parameters converge to the true values after four iterations when all of the parameters have significant changes.

Comparing the results in Tables 1 and 4, both techniques can accurately identify the damage variables in one iteration when the parameter changes are insignificant. From Tables 2 and 5, both techniques can successfully identify the damage variables in a few iterations when parameters have significant changes.

Table 6 Case 1 of Euler's beam with direct output feedback

Iteration no.	1	True
z_1	1.0000	1.0000
z_2	1.0000	1.0000
z_3	1.0000	1.0000
z_4	1.0000	1.0000
z_5	0.9980	0.9980
z_6	1.0000	1.0000
z_7	1.0000	1.0000
z_8	0.9980	0.9980
z_9	1.0000	1.0000
z_{10}	1.0000	1.0000
z_{11}	0.9980	0.9980
z_{12}	1.0000	1.0000
z_{13}	1.0000	1.0000
z_{14}	1.0000	1.0000
z_{15}	1.0000	1.0000

Table 7 Case 2 of Euler's beam with direct output feedback

Iteration no.	1	2	3	4	5	True
z_1	0.9670	1.0101	1.0018	1.0000	1.0000	1.0000
z_2	0.6375	0.6864	0.6980	0.7000	0.7000	0.7000
z_3	0.9802	1.0023	1.0046	0.9999	1.0000	1.0000
z_4	0.5084	0.6012	0.5989	0.6000	0.6000	0.6000
z_5	0.6846	0.7523	0.7981	0.8000	0.8000	0.8000
z_6	1.1144	1.0388	0.9950	1.0000	1.0000	1.0000
z_7	0.9174	0.9801	1.0017	1.0000	1.0000	1.0000
z_8	0.6405	0.6925	0.6993	0.7000	0.7000	0.7000
z_9	1.0857	1.0284	0.9975	1.0000	1.0000	1.0000
z_{10}	0.8670	0.9488	0.9998	1.0000	1.0000	1.0000
z_{11}	0.8522	0.8255	0.7970	0.8000	0.8000	0.8000
z_{12}	1.0392	0.9948	1.0039	0.9999	1.0000	1.0000
z_{13}	0.8776	0.9682	0.9952	1.0000	1.0000	1.0000
z_{14}	0.5773	0.6071	0.5997	0.6000	0.6000	0.6000
z_{15}	0.9764	1.0153	1.0053	0.9999	1.0000	1.0000

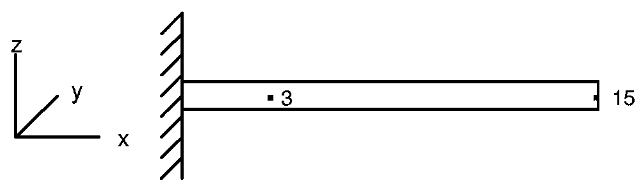

Fig. 3 Cantilevered Euler's beam.

Euler's Beam Example

The second structure used for study is a cantilevered aluminum Euler's beam, as shown in Fig. 3. The length, width, and thickness of this beam are 1, 0.0254, and 0.000635 m, respectively. The study is based on the analysis of the FE model of this beam structure.[19] For the structural damage, we consider the stiffness loss of 15 elements of equal length from the fixed end to the free end. The damage variables z_i, $i = 1, \ldots, 15$, which correspond to the 15 elements, are equal to 1 for the healthy structure. If the stiffness reduction of the ith element is $a\%$, then the value of z_i is $1 - 0.01a$. For example, the value of z_i is 0.5, when the stiffness loss of the ith element is 50%.

Direct Output Feedback

In the direct output feedback example, we use two displacement measurements located at positions 3 and 15, respectively. The first closed-loop system has the collocated output feedback controller at position 3. The second closed-loop system has the collocated output feedback controller at position 15. The natural frequencies of the first 10 modes of the open-loop system and the two closed-loop systems are used for damage detection.

Tables 6 and 7 show the results of damage detection for two different cases. In case 1, elements 5, 8, and 11 each has 0.2% stiffness loss. The solution of each parameter in the first iteration converges to the true one. The results in Table 7 show that all of the

Table 8 Design variables of controllers

Variable	Controller 1	Controller 2
m_{c1}	0.08	0.32
m_{c2}	0.16	0.24
m_{c3}	0.24	0.16
m_{c4}	0.32	0.08
k_{c1}	70	140
k_{c2}	140	70
k_{c3}	140	70
k_{c4}	70	140

Table 9 Case 2 of Euler's beam with dynamic controller

Iteration no.	1	2	3	4	5	True
z_1	0.8916	0.9600	1.0015	0.9999	1.0000	1.0000
z_2	0.9711	0.6987	0.6933	0.7000	0.7000	0.7000
z_3	1.0211	1.0596	1.0036	1.0001	1.0000	1.0000
z_4	0.6541	0.6397	0.5975	0.6000	0.6000	0.6000
z_5	0.9504	0.8848	0.7915	0.8000	0.8000	0.8000
z_6	0.9890	0.9576	0.9887	0.9999	1.0000	1.0000
z_7	0.7979	0.9494	1.0072	1.0000	1.0000	1.0000
z_8	0.4471	0.6455	0.6984	0.7000	0.7000	0.7000
z_9	0.7410	0.8829	1.0012	0.9999	1.0000	1.0000
z_{10}	0.8122	0.9416	0.9989	1.0000	1.0000	1.0000
z_{11}	0.7451	0.7628	0.8034	0.8000	0.8000	0.8000
z_{12}	1.0017	0.9552	0.9916	0.9999	1.0000	1.0000
z_{13}	1.0535	1.0141	1.0042	0.9998	1.0000	1.0000
z_{14}	0.7395	0.6145	0.5922	0.5999	0.6000	0.6000
z_{15}	0.9904	1.0583	1.0335	0.9984	1.0000	1.0000

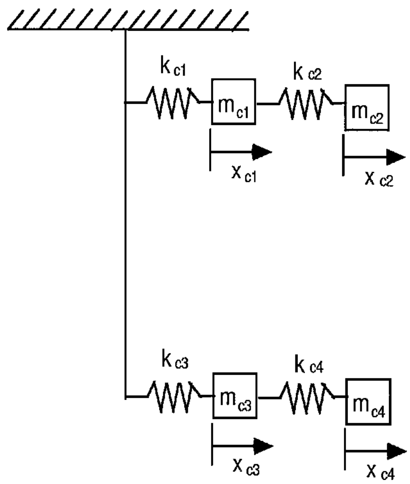

Fig. 4 Cantilevered Euler's beam with passive dynamic controllers.

parameters converge to the true ones after five iterations when six elements have significant stiffness reductions.

Controller with Second-Order Dynamics

Two passive systems, which are spring–mass systems with two degrees of freedom (Fig. 4), are attached to positions 3 and 15, respectively. The results of damage detection are based on the analysis of two closed-loop systems with controllers of different designed variables as listed in Table 8. The natural frequencies of the first 12 modes of two closed-loop systems are used.

In case 1, elements 5, 8, and 11 each has 0.2% stiffness loss. Each parameter converges to the true one in one iteration, and the results are the same as that shown in Table 6. The results in Table 9 show that all of the parameters converge to the true ones after five iterations when six elements have significant stiffness reductions.

Table 10 lists the first 10 natural frequencies of the open-loop system and the four closed-loop systems, which include the preceding two closed-loop systems with direct output feedback and the preceding two closed-loop systems with passive dynamic controllers. The natural frequencies of the first three modes have relatively significant changes when the displacement output feedback is used, while the natural frequencies of the high-frequency modes change little. This may limit the application of the direct output feedback approach because noise and environmental uncertainty may have significant effects on the identified natural frequencies of the high-frequency modes. All of the natural frequencies of the two closed-loop systems with passive dynamic controllers change significantly.

Table 10 Natural frequencies of various systems

ω_i, Hz	Open	Direct 1	Direct 2	Dynamic 1	Dynamic 2
$i = 1$	0.2314	0.4074	0.9297	0.0385	0.0576
2	1.4571	1.3465	1.8906	0.6854	0.5064
3	4.1401	4.3181	4.4665	1.5921	1.5126
4	8.3103	8.2954	8.3300	3.5606	3.5784
5	14.1798	14.2440	14.2698	4.6111	4.7958
6	21.9843	21.9771	22.0034	7.9502	8.1083
7	31.9425	31.9397	31.9570	10.0531	9.9792
8	44.1603	44.1765	44.1789	15.8172	16.7393
9	58.5049	58.5086	58.5024	20.7861	22.0166
10	74.4921	74.5023	74.4978	28.6436	26.3393

The advantage of direct output feedback technique is its simplicity because the feedback controller is directly from the output measurements. The use of the controller with a passive dynamic system has the following advantages: 1) flexibility of adjusting natural frequencies, 2) variety of choice of passive controllers, and 3) increase of the number of the effective natural frequencies, which are reliable in the considered low-frequency range. In real applications, we can combine these two techniques and use the advantages of both techniques for damage detection.

Conclusions

This paper presents novel approaches for structural damage detection by adding virtual passive controllers to structures. The controller is passive in the sense that it contains mechanisms that serve only to transfer and dissipate energy to the system. Stabilization can be accomplished by a controller with gains interpreted as virtual mass, spring, and dashpot elements. Both damage detection techniques, which are based on the direct output feedback and the feedback controller with second-order dynamic equations, can efficiently identify damage in the presented examples when the damage variables have minor as well as significant changes. In this paper only the identified natural frequencies are used for damage detection because the identified natural frequencies are generally more reliable than the identified mode shapes. Only a small number of sensors are required for the presented approaches. The advantage of direct output feedback technique is its simplicity. The technique with the controller of passive dynamic system has the advantages of flexibility and variety. In real applications, one may combine the advantages of both techniques for damage detection.

Acknowledgments

This research is supported in part by NASA Grant NCC5-228 and National Science Foundation Grant HRD-9706268. The support is greatly appreciated.

References

[1]Jauregui, D. V., and Farrar, C. R., "Comparison of Damage Identification Algorithms on Experimental Modal Data from a Bridge," *Proceedings of the 14th International Modal Analysis Conference*, Society for Photo-Optical Instrumentation Engineers, Bellingham, WA, 1996, pp. 1423–1429.

[2]Zimmerman, D. C., and Kaouk, M., "Structural Damage Detection Using a Subspace Rotation Algorithm," *Proceedings of the 33rd AIAA/ASME/AHS/ASC Structures, Structural Dynamics, and Materials Conference*, AIAA, Washington, DC, 1992, pp. 2341–2350.

[3]Lew, J.-S., "Using Transfer Function Parameter Changes for Damage Detection of Structures," *AIAA Journal*, Vol. 33, No. 11, 1995, pp. 2189–2193.

[4]Pandey, A. K., Biswas, M., and Samman, M., "Damage Detection from Changes in Curvature Mode Shapes," *Journal of Sound and Vibration*, Vol. 145, No. 2, 1991, pp. 321–332.

[5]Smith, S. W., and Beattie, C. A., "Secant-Method Adjustment for Structural Models," *AIAA Journal*, Vol. 29, No. 1, 1991, pp. 119–126.

[6]Lew, J.-S., "Damage Detection Using Neural Networks and Transfer Function Correlation," *Proceedings of the 18th International Modal Analysis Conference*, Society for Photo-Optical Instrumentation Engineers, Bellingham, WA, 2000, pp. 1783–1789.

[7]Ness, S., and Sherlock, C. N., *Nondestructive Testing Handbook, Vol. 10, Nondestructive Testing Overview*, American Society for Nondestructive Testing, Columbus, OH, 1996.

[8]Trivailo, P., Plotnikova, L. A., and Wood, L. A., "Enhanced Parameter Identification for Damage Detection and Structural Integrity Assessment Using 'Twin' Structures," *Fifth International Congress on Sound and Vibration*, 1997, Univ. of Adelaide, Adelaide, SA, Australia, pp. 1733–1741.

[9]Trivailo, P., and Plotnikova, L., "High Sensitivity Damage Identification in 3D Modular Structures: Improved Modal Analysis Technique," *International Conference on Application of Modal Analysis, Recent Advances in Modal Analysis Practice*, 1999, pp. 1–14.

[10]Juang, J. N., and Phan, M., "Robust Controller Designs for Second-Order Dynamic Systems: A Virtual Passive Approach," *Journal of Guidance, Control, and Dynamics*, Vol. 15, No. 5, 1992, pp. 1192–1198.

[11]Bruner, A. M., Belvin, W. K., Horta, L. G., and Juang, J. N., "Active Vibration Absorber for the CSI Evolutionary Model Design and Experimental Results," *Journal of Guidance, Control, and Dynamics*, Vol. 15, No. 5, 1992, pp. 1253–1257.

[12]Juang, J. N., Wu, S. C., Phan, M., and Longman, R. W., "Passive Dynamic Controllers for Nonlinear Mechanical Systems," *Journal of Guidance, Control, and Dynamics*, Vol. 16, No. 5, 1993, pp. 845–851.

[13]Morris, K. A., and Juang, J.-N., "Dissipative Controller Designs for Second-Order Dynamic Systems," *IEEE Transactions on Automatic Control*, Vol. 39, No. 5, 1994, pp. 1056–1063.

[14]Lee-Glauser, G., Juang, J.-N., and Sulla, J. L., "Optimal Active Vibration Absorber: Design and Experimental Results," *Journal of Vibration and Acoustics*, Vol. 117, No. 2, 1995, pp. 165–171.

[15]Chuang, C.-H., Courouge, O., and Juang, J.-N., "A Robust Controller for Second-Order Systems Using Acceleration Measurement," *Journal of Dynamic Systems, Measurement, and Control*, Vol. 119, No. 2, 1997, pp. 350–354.

[16]Ray, L. R., and Tian, L., "Damage Detection in Smart Structures through Sensitivity Enhancing Feedback Controller," *Journal of Sound and Vibration*, Vol. 227, No. 5, 1999, pp. 987–1002.

[17]Hagan, M. T., Demuth, H. B., and Beale, M., *Neural Network Design*, PWS Publishing, Boston, 1996.

[18]Reklaitis, G. V., Ravindran, A., and Ragsdell, K. M., *Engineering Optimization: Methods and Applications*, Wiley, New York, 1983.

[19]Craig, R. R., *Structural Dynamics: An Introduction to Computer Method*, Wiley, New York, 1981.