Research Article

Improvement of Transmission Loss Using Active Control with Virtual Modal Mass

V. Lhuillier,^{1,2} L. Gaudiller,¹ C. Pezerat,² and S. Chesne¹

¹ LaMCoS, INSA-Lyon, CNRS UMR5259, F69621, France

² LVA, INSA-Lyon, F69621, France

Correspondence should be addressed to L. Gaudiller, luc.gaudiller@insa-lyon.fr

Received 16 January 2008; Revised 1 April 2008; Accepted 19 May 2008

Recommended by Marek Pawelczyk

This paper deals with an alternative modal active control approach to reduce sound transmission through a structure excited by an acoustic wave. Active control makes it possible to conserve lightness while improving acoustic performances. “Modal mass damping control” is proposed for light and small structures having slight modal overlap. The aim of this control is to modify the modal distribution of high radiation efficiency modes with active modal virtual mass and active modal damping. The active virtual mass effects lower eigen frequencies to less audible frequency range while reducing vibration amplitudes in a broad frequency range. An application of this concept is presented in a simple smart structure. It is harmonically excited on large bandwidth by a normal acoustic plane wave. Results obtained by active modal virtual mass and damping control are compared to other modal control approaches.

Copyright © 2008 V. Lhuillier et al. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

1. INTRODUCTION

The acoustic transmission loss can be improved by passive methods such as mass addition (Mass Law [1]), double panel, and/or the use of sound absorbing materials. Most of these techniques generally involve an increase in mass or volume to provide a good insulation at low frequencies. However, this is not convenient in the transport sector.

Active control can complete the passive methods and reduce the sound produced by vibrating structures at low frequencies. Various active control strategies have been developed.

In the 1930s, research focused on active noise control ANC (Lueg's Patent [2]) in which a secondary field destructively interferes with the primary disturbance. However, these feedforward controllers are tricky to implement because of the difficulty in measuring reference signal and determining the feedback effect of the secondary sources on the reference sensors. ANC was only successfully implemented in applications, where disturbance is tonal such as ducts and exhaust stacks [3].

Active structural acoustic control (ASAC) controllers were then developed to reduce sound radiation by modifying panel vibrations with shakers or piezoelectric patches.

The ASAC feedforward approach has the same drawbacks as ANC. Moreover, for the low-frequency range, the minimization of sound radiation achieved by “modal restructuring” can increase the vibration levels [4]. This method uses coupling between modes below the critical frequency to create destructive interferences in the radiated field.

If the disturbance signal is not available, control strategies are limited to feedback controllers. The type of controller differs as a function of the availability of a model. Elliott and Johnson [5] showed that at low frequencies, reducing the volume velocity of the panel leads to reduced sound power.

Networks of sensors [6, 7] or distributed sensors such as PVDF film [8] have been implemented to evaluate the volume velocity and to supply an SISO controller. System stability is guaranteed [9] when the actuators and the sensors are collocated and dual. An alternative to SISO controllers called “decentralized MIMO” consists in scattering manifold independent control units on the structure [10]. The results

are fairly good but the technical resources needed are so huge that smart structures cannot be easily introduced in everyday equipment such as double-glazed windows, machinery shielding, vehicles. Moreover, active damping provides good performances without a model, but it is only efficient in the vicinity of a resonance frequency.

The model-based control strategy presents several advantages. It allows minimizing the number of sensors with modal reconstruction [11] and reducing control energy by nonlinear modal control [12, 13]. Complex and nonlinear structures can also be controlled with modal adaptive algorithm [14].

In the vibroacoustic field, accurate modeling is also required to adjust the controller. Baumann et al. [15, 16] proposed to compute sound power by using radiation filters. These filters are introduced in the state space model of the structure and the acoustic energy is incorporated in the cost function. Experiments carried out by Bingham et al. [17] and Dehandschutter et al. [18] demonstrated the validity of this method, but this technique is limited by the number of radiation filters that have to be introduced in the state space formulation.

Below the critical frequency, transmission is mass controlled [1]. Alujevic and Gardonio [19] showed that a light panel controlled by an active virtual mass with decentralized MIMO controllers behaves like a noncontrolled heavy panel. Transmission loss is increased but the sound power peaks still remain due to the lack of active damping.

The aim of this paper is to describe the reduction of sound transmission by modifying the modal distribution. When structures are relatively small and light, the panels have low modal overlap, so in this case, modal control appears to be adapted. It enables limiting the number of active components and concentrating control energy on high radiation efficiency modes. Active modal damping squeezes sound power peaks and controls low radiation efficiency modes which may be excited and transfer vibrating energy to structure bounds. Moreover, above the critical frequency, when the incidence angle is close to the coincidence angle, the behavior of the panel is controlled by damping. Consequently, transmission loss drops dramatically for a lightly damped structure. The effect of adding modal masses shifts the resonance frequencies to a less audible frequency range. Also, the mass control of one eigen mode can be considered as an addition of virtual modal mass. Thus, the acoustic characteristics of the structure should be improved in the upper frequency range.

This paper presents a state feedback controller which acts on modal mass and modal damping. After the state-of-the-art presented previously, Section 2 deals with modeling the structure and the method used to compute the sound power. Section 3 introduces the vibroacoustic controller driven by modal accelerations and modal velocities. The principle of this new “mass damping vibroacoustic modal control” approach is shown in Section 4 on a simple one-dimensional structure excited by a normal incident plane wave. It is then compared to other vibroacoustic modal controls in simulations.

2. MODELING

2.1. Structural modeling

The modal control enables concentrating control energy on high-radiation efficiency modes and limiting the number of active components. Moreover, the effectiveness of this model-based strategy depends on the accuracy of the modeling. The first step consists in building a model that includes the mass and the stiffness of the actuators. The structure is divided into a finite number of elements. The equation of motion governing the dynamic of the controlled system is

$$(M_s + M_a)\ddot{\delta} + C_s\dot{\delta} + (K_s + K_a)\delta = F_d + F_c, \quad (1)$$

with F_d being the disturbance force, F_c the control force, δ the displacement, M , C , K the mass, damping, and stiffness matrices, respectively. The subscripts a and s denote the actuators and the structure, respectively. The control effort driven by the displacement, the velocity, and the acceleration acts on the stiffness, the damping, and the mass of the system, respectively. When the structure is lightly damped and the modes are sufficiently decoupled, the linear system is described by a set of decoupled modal equations after a change of variable:

$$\delta = \phi q, \quad (2)$$

$$\ddot{q}_i + 2\xi_i\omega_i\dot{q}_i + \omega_i^2q_i = \phi_i^T(F_d + F_c) = f_d(i) + f_c(i), \quad (3)$$

with q_i being the modal amplitudes, ξ_i the modal viscous damping ratio, ϕ_i the modal shapes, and ω_i the frequency of the i th mode. The response v of the structure can be given as a linear combination of the modes:

$$v(x, y, z, t) = \sum_{n=1}^N q_n(t)\phi_n(x, y, z). \quad (4)$$

The corresponding modal state variable form is given by

$$\begin{aligned} \dot{x} &= Ax + Bu + Ew, \\ y &= Cx + Du, \end{aligned} \quad x = \begin{Bmatrix} q \\ \dot{q} \end{Bmatrix}, \quad (5)$$

with x being the state vector, u the control vector, w the disturbance noise, y the output vector, A , B , C , D the state matrices and E the disturbance input matrix. From this formulation, the transfer function of the modal velocities can be easily computed and introduced into the following acoustic modeling.

2.2. Acoustic modeling

To predict the acoustic performances, it is necessary to model the sound radiation of the vibrating structure. The acoustic power W can be calculated from the modal amplitudes and from the frequency dependent radiation resistance matrix of structural modes M [5]. The diagonal terms of M are the self-radiation resistance and the off-diagonal terms are the mutual radiation resistance. The global sound power W can

be obtained by

$$W = \dot{q}^H M \dot{q}, \quad (6)$$

where the subscript H denotes the Hermitian. For complex geometries, M can be calculated from the radiation resistance matrix of elemental radiators and modal shapes. Each element acts as an elementary radiator, where the specific acoustic transfer impedance at position y on an infinite plane to an observation point x is given by

$$z(x, y) = \frac{p(y)}{v(x)} = \frac{j\omega\rho S_e}{2\pi r} e^{-jkr}, \quad (7)$$

with r being the distance between x and y , ρ the fluid density, k the acoustic wave number, and S_e the radiator area. It is assumed that the elementary radiators radiate into free space and are small compared to the acoustic wave length. The sound power radiated by only one single element W_e is equal to

$$W_e = \frac{S_e}{2} \Re\{v_e^* p_e\}, \quad (8)$$

where p_e and v_e are, respectively, the pressure and the complex velocity of the elemental radiator. $\Re$ denotes the real part and $*$ the conjugate. The global acoustic power W radiated by a set of elemental sources is calculated with

$$W = \frac{S_e}{2} \Re\{v^H p\}, \quad (9)$$

where v and p are, respectively, the velocity vectors and pressure vectors of the radiators. Next, the matrix of acoustic transfer impedances Z linking the pressure at each element to the velocity at each element is introduced. r_{ij} denotes the distance between elemental sources. R_{ij} is the radiation resistance matrix of the elemental radiators. W becomes

$$W = \frac{S_e}{2} \Re\{v^H Z v\} = \frac{S_e}{4} v^H (Z + Z^H) v = v^H R v, \quad (10)$$

with

$$R_{ij} = \frac{S_e}{2} \Re\{Z_{ij}\} = \frac{\omega^2 \rho S_e^2}{4\pi c} \frac{\sin kr_{ij}}{kr_{ij}}, \quad (11)$$

with c being the speed of sound in the medium, ω the frequency, and ρ the mass density of air. The acoustic power can be obtained from the radiation resistance matrix of the elemental radiators and the velocity of structural modes after a change of variable (2),

$$\begin{aligned} W(\omega) &= v^H R(\omega) v = \dot{q}^H \phi^H R(\omega) \phi \dot{q}, \\ M(\omega) &= \phi^H R(\omega) \phi. \end{aligned} \quad (12)$$

Then, for any structures and for any set of frequencies, the radiation resistance matrix of structural modes M can be calculated and approximated using a Laplace-domain multi-input, multiple output transfer function $M(s)$. $M(s)$ is then factorized into a stable causal radiation filter $G(s)$ (13) and

$W(s)$ is given by (14)

$$M(s) = G^T(-s)G(s), \quad (13)$$

$$W(s) = \dot{Q}(-s)^T M(s) \dot{Q}(s), \quad (14)$$

with $\dot{Q}(s)$ being the modal velocities in the Laplace domain.

3. MODAL MASS DAMPING VIBROACOUSTIC CONTROL

3.1. Principle

When a structure is excited by a diffuse field below its critical frequency, the transmission is “mass controlled” [1]. The effect of mass addition at low frequency slides the resonance frequencies to a less audible frequency range, while reducing the vibration amplitudes. Therefore, the overall transmission loss is raised. Removing the stiffness scales down the eigen frequencies but increases the level of vibration. Reducing eigen frequencies with stiffness control cannot be considered due to the risk of instability.

If the frequency is slightly greater than the critical frequency, the transmission will be “damping controlled” [1]. In case of lightly damped structures, active damping becomes essential to reduce transmission. At higher frequencies, the good sound insulation does not justify extending the control bandwidth.

In the framework of small and lightly damped structures, the modal control can be effective due to low modal overlap. When the panel is excited by an acoustic plane wave, the low-order modes (odd-odd) are highly excited. Controlling only these high-radiation modes may lead to a considerable reduction of sound power.

The aim of the *modal mass—damping vibroacoustic control*— is to modify the modal distribution of the panel. The natural frequencies of high-radiation modes can be lowered by the addition of active modal mass and vibration amplitudes can be reduced at resonances frequencies by modal damping. The active modal mass can be considered as a *virtual mass* concentrated on controlled modes. Consequently, the amplitudes of mass controlled modes are reduced after resonances due to frequency shifts. Active modal damping limits sound power peaks and the vibrations of low-radiation modes. The control gain matrix is computed with an optimal control algorithm. If the cost function considers exclusively sound power, certain nonradiating modes may be highly excited leading to risks of failure and structure-born transmission. The control is driven by modal acceleration and modal velocities from the derivative state. In practice, modal displacements and velocities are calculated from sensors signals with a Luenberger observer [20]. Then, the reconstructed state feeds the controller whose dynamics is nearly independant of the controlled structure. For simplicity, observation problems are not treated in this article:

$$u = -K\hat{x}_c = -\sum_{i=1}^{N_c} K_v(i)\dot{q}_i + K_a(i)\ddot{q}_i. \quad (15)$$

FIGURE 1: Smart structure.

Then, the active mass and damping can be expressed in the modal equation:

$$\begin{aligned} \left[1 + \underbrace{K_a(i) \cdot f_p(i)}_{\text{Active modal mass}} \right] \ddot{q}_i + \left[2\xi_i\omega_i + \underbrace{K_v(i) \cdot f_p(i)}_{\text{Active modal damping}} \right] \dot{q}_i + \omega_i^2 q_i \\ = f_d(i) - \underbrace{f_p(i) \sum_{j \neq i} K_v(j) \dot{q}_j + K_a(j) \ddot{q}_j}_{\text{Excitation of controlled modes}}, \end{aligned} \quad (16)$$

with $f_p(i)$ being the control effort at the i th mode, x_c the state vector of controlled modes, and K_a and K_v the gains relative to modal accelerations and velocities. The state of the controlled system can be calculated from (5) and (15) with the following transfer function:

$$Y(s) = (C - DKs)((I + BK)s - A)^{-1}EW(s). \quad (17)$$

Next, the modal velocities enables computing the sound power. $\dot{q}$ is a highpass filter. Consequently, when the frequency is higher than the eigen frequencies of the mass controlled modes and gains relative to the accelerations are high, the frequency response function (FRF) of the control will be constant. The problem of unwanted excitation (spillover) of modes especially above the control bandwidth has to be considered. In the control bandwidth, spillover can be attenuated with good actuator positions. At high frequency, the passive methods are sufficiently efficient. For instance, in the framework of reducing the sound transmission of the double panel (DP) the consequences of spillover on the higher modes are limited due to their good natural transmission loss (TL) in midrange and high frequencies. The performances of DP can be increased at low frequencies without excessive alteration of the TL at higher frequencies by adding modal masses exclusively to high radiation modes.

3.2. Optimization and adjustment

When the structure is assumed to be linear and its model is available, the optimization method used can be the linear quadratic (LQ) algorithm. The constant gains are obtained after minimization of a quadratic cost function.

The optimal control seeks to minimize both the state function of structural system x , which is usually vibration energy or acoustic energy, and the control cost u . The control gains are computed by solving Ricatti's equation but using the derivative of state vector $\dot{x}$ does not allow computing gains in this way. If the cost function considers the control energy, the optimal gain may favor damping rather than mass control due to the acceleration FRF. During experiments, the maximal acceptable voltage of the transducers is a factor that limits controller performances. Therefore, the cost function computes the acoustic energy and forbids a control voltage higher than the voltage limit. This can be defined as follows:

$$\begin{aligned} \text{if } V \leq V_{\max}, \quad J &= \int_{\omega_{\text{inferior}}}^{\omega_{\text{superior}}} \dot{Q}(\omega)^H M(\omega) \dot{Q}(\omega) d\omega, \\ \text{if } V > V_{\max}, \quad J &= \infty. \end{aligned} \quad (18)$$

4. APPLICATION TO A ONE-DIMENSIONAL STRUCTURE

The aim of this section is to present modal adjustment possibilities. Constraints such as number of components, dimensions, saturation, and measurements are not discussed in this first study. Simulations are performed on a one-dimensional structure using the previously described control approach. The structure is excited by a normal acoustic plane wave to emphasize active modal mass control within the framework of sound transmission. This modal method enables concentrating control energy on high-radiation efficiency modes and limits the number of actuators. Consequently, the actuators are mainly used to act on the first odd modes. The first step of this control strategy requires using system model.

4.1. Modeling

The smart structure considered and shown in Figure 1 is a clamped-clamped beam to which three piezoelectric patches (P1 89 ceramics) are fixed. The dimensions and material characteristics are given in Table 1. An FE Model is developed using Ansys software, where the beam is modeled with Solid45 elements (8 nodes with 3 mechanical dof/node) of $2.5 \times 2.5 \times 1.5$ mm and the piezoelectric patches are modeled with Solid5 elements (8 nodes with up to 6 dof/node including structural and electrical dof) of $2.5 \times 2.5 \times 0.7$ mm.

TABLE 1: Simulations parameters.

Beam density	$\rho_{\text{beam}} = 7800 \text{ Kg/m}^3$	Beam width	$b = 4.0 \times 10^{-2} \text{ m}$
Beam Young modulus	$E = 2.1 \times 10^{11} \text{ Pa}$	Beam height	$h = 1.5 \times 10^{-3} \text{ m}$
Modal damping ratio	$\xi_i = 0.01$	Beam length	$l_b = 6.0 \times 10^{-1} \text{ m}$
Ceramic density	$\rho_c = 7700 \text{ Kg/m}^3$	Patch width	$b_p = 3.0 \times 10^{-2} \text{ m}$
Ceramic Young modulus	$Y_{11} = 6.67 \times 10^{10} \text{ Pa}$ $Y_{33} = 5.26 \times 10^{10} \text{ Pa}$	Patch height	$e_p = 0.7 \times 10^{-3} \text{ m}$
Permittivity	$\epsilon_{11}^s = 1142$ $\epsilon_{33}^s = 668$	Patch length	$l_p = 8 \times 10^{-2} \text{ m}$
Piezoelectricity	$d_{31} = -108 \text{ pC/N}$ $d_{33} = 240 \text{ pC/N}$	Position 1	$A1 = 2.5 \times 10^{-3} \text{ m}$
		Position 2	$A2 = 4.25 \times 10^{-1} \text{ m}$
		Position 3	$A3 = 1.625 \times 10^{-1} \text{ m}$

For the sake of simplicity, 3 identical patches are bonded to the smart structure.

The first 14 modal shapes are extracted and the optimal position of the actuators are sought by maximizing the coupling coefficient k of *one* high-radiation efficiency mode and minimization of coupling coefficient of other modes. The generalized coupling coefficient k_{ij} of the i th mode for the j th piezoelectric patch bonded to the structure is defined by Hagood and von Flotow [21] as follows:

$$k_{ij}^2 = \frac{(\omega_{ij}^D)^2 - (\omega_{ij}^E)^2}{(\omega_{ij}^E)^2}, \quad (19)$$

where ω_{ij}^D and ω_{ij}^E are, respectively, the natural frequencies of the structure when the piezoelectric j th patch is open-circuited and short-circuited. The modal coupling coefficient is defined by Badel et al. [22]:

$$|\alpha_{ij}| = \sqrt{C_{0P} K_{PDi} k_{ij}^2} \quad \text{with } C_{0P} = \frac{2\epsilon_{33}^s l_p b}{e_p}, \quad (20)$$

where K_{PDi} is the equivalent stiffness of the i th mode and C_{0P} is the null strain capacitance of the piezoelectric patch. l_p , b , and e_p are, respectively, the length, width, and thickness of the patches. Next, the modal coupling is introduced in (2):

$$\ddot{q}_i + 2\xi_i \omega_i \dot{q}_i + \omega_i^2 q_i = f_d(i) + \sum_{j=1}^{N_{\text{Actuators}}} \alpha_{ij} V_j, \quad (21)$$

where V_j is the control voltage of the j th patch. The results of optimal placement are presented in Figure 1.

Once the structural modeling has been completed, the radiation resistance matrix $M(\omega)$ is calculated in a frequency band from 10 to 900 Hz and approximated using the Laplace-domain multi-input, multiple output transfer function $M(s)$ (14). The eigen frequencies of the smart structure are presented in Table 2.

4.2. Control gain matrix

4.2.1. Transferring control energy from damping to mass

In order to present the advantages of a derivative state feedback control, the control effort is initially concentrated

TABLE 2: Eigen frequencies.

Mode	1	2	3	4	5	6	7	8
Frequency (Hz)	24	69	130	215	325	451	617	794

on modal damping before being transferred to modal mass. The initial terms of the gain matrix relative to damping can be calculated by solving the Ricatti equation stemming from the minimization of the frequency weighted energy cost function (22). Then, the terms relative to modal mass are introduced and those such as maximal voltage actuators inputs remain constant. The modal mass control is applied here exclusively on the first mode because of its high contribution to the sound power. Indeed, at low frequency and for this type of structure, the sound power is mainly radiated by volumetric modes [5].

In Figure 2, exclusive modal damping control is represented in blue. The considered excitation is a normal acoustic plane wave which is uniform on all the plate. The line color gets hotter when the first mode acceleration gain increases. The left-hand plot shows the evolution of the sound power of the controlled system computed with (14) and (17). When the modal acceleration gain increases, the first eigen frequency scales down and the overall sound power is reduced by the active mass. However, limiting the control effort does not allow still introducing as much active modal damping. Consequently, the sound power peak of the mass controlled mode is not reduced as in the exclusive active damping control.

The performances of the controlled systems differ drastically according to the weighting on the mass and damping. If the aim is to reduce the sound power peaks, the control will focus its energy on damping. A good tradeoff between mass and damping can be found when reducing sound power.

The right-hand side plot shows the control frequency response of actuator A1. This frequency response becomes constant when the gains applied to acceleration are high compared to those applied to damping. With full damping, the command is like a pass band filter, whereas it is like a highpass filter in mass control. Consequently, the control energy of the vibroacoustic modal mass control is slightly

FIGURE 2: Transfer of control energy from modal damping to modal mass: (a) sound power, (b) control.

higher than that of the regular state feedback controller. Note that in Figure 2(b), the control voltage is calculated at the initial state which leads to the highest control voltage. So, the first peak of the control appears at 24 Hz.

4.2.2. Minimization of acoustic energy

In Figure 3(a), the cost function computed with (18) between 10 and 900 Hz is drawn with respect to the acceleration gain and velocity gain of the first eigen mode. When the control voltage is higher than the maximal acceptable voltage of the transducers, the cost function reaches infinity. The bold line indicates the border beyond which the cost function tends to infinity. The acoustic energy reaches a minimum for a given gain couple. When the gain applied on acceleration is high compared to that on the velocity, the acoustic energy

FIGURE 3: Evolution of acoustic energy with respect to the acceleration gain and the velocity gain on first eigen mode: (a) sound power with optimal gains (b).

increases because of the peak on the first mode due to the limitation of control voltage. The right-hand side plot shows the sound power of the mass damping controlled system computed with an optimal gain matrix.

The optimal gain matrix is obtained from the results presented in Figure 3. Then, as shown in Figure 4(a) time response simulation of the structure excited by a plane shock normal wave is performed with the optimal gain matrix. The sound power is presented in the left-hand side plot. The high-frequency components of the sound power vanish after a few oscillations due to the active damping on these modes. Consequently, the sound power is mainly created by the first mode which is initially more disturbed. In the right-hand side plot, the first mode velocity shows that the mass effect increases the oscillation period and the amplitude is reduced with active damping.

FIGURE 4: (a) Sound power, (b) first mode velocity.

4.3. Effects of the modal mass control compared to other modal controls

4.3.1. Other approaches

In this section, the modal mass control is compared with two other modal control approaches. The first is a pure vibration control using a regular state feedback control with gains computed by minimizing the frequency weighted (FW) energy cost function (22):

$$J = \int_0^\infty (x^T Q x + \rho u^T u) dt. \quad (22)$$

The second control strategy considers vibration and acoustics. In Baumann et al.' [15] development, the radiation filters (RF) $G(s)$ are converted to state space form (23) and introduced in the augmented state space of the system (24):

$$\dot{r} = A_G r + B_G \dot{q}, \quad (23)$$

$$z = C_G r + D_G \dot{q},$$

$$\begin{bmatrix} \dot{q} \\ \dot{q} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} A & 0 \\ 0 & B_G & A_G \end{bmatrix} \begin{bmatrix} q \\ \dot{q} \\ r \end{bmatrix} + \begin{bmatrix} B \\ 0 \end{bmatrix} u, \quad (24)$$

with z being the result of the passing velocity components through the radiation filter. z^2 is the sound power. In the case of a normal incident plane wave, the general force decays with increasing frequency, which is why an integrator term is added in (26). If the optimal gain matrix is computed by a minimization of z^2 , low-frequency modes will not be controlled because the $M(\omega)$ terms are small when the modes do not radiate independently. Consequently, the cost function that considers the disturbance can be written as follows:

$$J_{\text{Rad}} = \int_0^\infty (z_2^T z_2 + \rho u^T u) dt, \quad (25)$$

$$z_2 = \mathcal{L}^{-1}(s^{-1} G(s) \dot{Q}(s)), \quad (26)$$

where z_2^2 is the weighted sound power.

4.3.2. Transmission loss

Considering weak fluid-structure coupling, the parietal pressure can be described with the so-called *blocked pressure* [1]. The disturbance load corresponds to the incident pressure is given by

$$p(x) = -2p_{\text{in}}, \quad (27)$$

where p_{in} is the amplitude of the wave of the acoustic plane wave. The transmission loss can be defined as follows:

$$\text{TL} = 10 \log_{10} \left(\frac{W_{\text{in}}}{W} \right), \quad (28)$$

with W being the radiated sound power and W_{in} the incident sound power defined by [23, 24]

$$W_{\text{in}} = \frac{|p_{\text{in}}|^2 l_b \cdot b \cdot \cos(\theta_i)}{2\rho c}, \quad (29)$$

where ρ is the fluid density, c the speed of sound in the medium, and θ_i the incidence angle. For a normal incident acoustic plane wave, $\theta_i = 0$.

4.4. Comparison

Figure 5 presents the transmission losses of the uncontrolled and controlled structure with the three different approaches. The structure is still excited by a normal incident acoustic plane wave that exclusively disturbs odd modes. The

FIGURE 5: Transmission loss.

three controllers are tuned so that the maximal voltage inputs are the same. The modal mass damping controller uses the optimal gain matrix determined previously. The transmission losses of the FW and the RF controllers have the same characteristics. They give good isolation at resonance frequencies with damping. Consequently, the peaks disappear but the general behavior of transmission loss does not change. Note that the optimal gain matrix of the RF controller is found without searching weighting factors Q as in the frequency weight function (22). The modal mass damping control reduces the peaks at the resonance frequency with damping in the same way as FW and RF controllers, and it also improves the general behavior of the transmission loss due to the mass effect. This active modal mass is above all a mass addition. The reduction of the first eigen frequency from 24 Hz to 21.6 Hz leads to the diminution of modal velocity after the first resonance. Contrary to damping, the mass effect is not limited to the vicinity of the resonance frequency of the controlled mode.

Also, the three controllers can generate spillover. This phenomenon can be limited by considering residual modes in the patch placement optimization and by introducing them in the cost function [9]. Additional passive components are frequently introduced to reduce the spillover induced by the controllers. For future applications such as double panel control, the spillover problem can be limited by the good natural properties of the controllers in midrange and high frequency.

5. CONCLUSION

A new modal active control is proposed in this paper. It permits adjusting a frequency response template of the controlled structure. This control has been developed within an

acoustic framework designed to reduce sound transmission through a structure. For each mode, the active modal mass addition lowers the eigen frequency of the mass controlled mode and reduces its velocity after resonance. The use of this method on a simple smart structure equipped with piezoelectric patches is described with an explanation of the concept by way of a simulation. The good performances at resonance frequencies obtained with an active damping controller can be completed with this approach which modifies the modal distribution of the structure. The “mass damping vibroacoustic modal control” is compared to other control approaches which mainly use damping to reduce sound transmission.

This preliminary study can be adapted to more complex structures such as double panels. Indeed double panels provide good transmission loss at midrange and high frequencies, making it possible to concentrate control energy on the first modes. Thus, the method proposed is well adapted to this kind of structure. Moreover, the downward shift of the eigen frequencies can be considered as virtual transformations of structures that could be used in the field of sound quality.

REFERENCES

- [1] F. J. Fahy, *Sound and Structural Vibration: Radiation, Transmission and Response*, Academic Press, London, UK, 1985.
- [2] P. Lueg, “Process of silencing sound oscillations,” *Us patent* 2043416, 1934.
- [3] S. Laugesen, “Active control of multi-modal propagation of tonal noise in ducts,” *Journal of Sound and Vibration*, vol. 195, no. 1, pp. 33–56, 1996.
- [4] R. L. Clark and C. R. Fuller, “Experiments on active control of structurally radiated sound using multiple piezoceramic actuators,” *The Journal of the Acoustical Society of America*, vol. 91, no. 6, pp. 3313–3320, 1992.
- [5] S. J. Elliott and M. E. Johnson, “Radiation modes and the active control of sound power,” *The Journal of the Acoustical Society of America*, vol. 94, no. 4, pp. 2194–2204, 1993.
- [6] T. C. Sors and S. J. Elliott, “Volume velocity estimation with accelerometer arrays for active structural acoustic control,” *Journal of Sound and Vibration*, vol. 258, no. 5, pp. 867–883, 2002.
- [7] A. François, P. De Man, and A. Preumont, “Piezoelectric array sensing of volume displacement: a hardware demonstration,” *Journal of Sound and Vibration*, vol. 244, no. 3, pp. 395–405, 2001.
- [8] A. Preumont, A. François, P. De Man, N. Loix, and K. Henriouille, “Distributed sensors with piezoelectric films in design of spatial filters for structural control,” *Journal of Sound and Vibration*, vol. 282, no. 3–5, pp. 701–712, 2005.
- [9] A. Preumont, *Vibration Control of Active Structures: An Introduction*, Kluwer Academic Publishers, Dordrecht, The Netherlands, 2nd edition, 2002.
- [10] S. J. Elliott, P. Gardonio, T. C. Sors, and M. J. Brennan, “Active vibroacoustic control with multiple local feedback loops,” *The Journal of the Acoustical Society of America*, vol. 111, no. 2, pp. 908–915, 2002.
- [11] L. Gaudiller and J. Der Hagopian, “Active control of flexible structures using a minimum number of components,” *Journal of Sound and Vibration*, vol. 193, no. 3, pp. 713–741, 1996.

- [12] L. Gaudiller and F. Matichard, "A nonlinear method for improving the active control efficiency of smart structures subjected to rigid body motions," *IEEE/ASME Transactions on Mechatronics*, vol. 12, no. 5, pp. 542–548, 2007.
- [13] F. Matichard and L. Gaudiller, "Improvement of potential energetic exchange using nonlinear control," in *Proceedings of the IEEE/ASME International Conference on Advanced Intelligent Mechatronics (AIM '05)*, pp. 807–812, Monterey, Calif, USA, July 2005.
- [14] L. Gaudiller and S. Bochart, "Adaptive active control of flexible structures subjected to rigid body displacements," *Journal of Sound and Vibration*, vol. 283, no. 1-2, pp. 311–339, 2005.
- [15] W. T. Baumann, W. R. Saunders, and H. H. Robertshaw, "Active suppression of acoustic radiation from impulsively excited structures," *The Journal of the Acoustical Society of America*, vol. 90, no. 6, pp. 3202–3208, 1991.
- [16] W. T. Baumann, F.-S. Ho, and H. H. Robertshaw, "Active structural acoustic control of broadband disturbances," *The Journal of the Acoustical Society of America*, vol. 92, no. 4, pp. 1998–2005, 1992.
- [17] B. Bingham, M. J. Atalla, and N. W. Hagood, "Comparison of structural-acoustic control designs on an active composite panel," *Journal of Sound and Vibration*, vol. 244, no. 5, pp. 761–778, 2001.
- [18] W. Dehandschutter, K. Henriouille, J. Swevers, and P. Sas, "Model-based feedback control of acoustic radiation from vibrating structures by means of structural control," *Flow, Turbulence and Combustion*, vol. 61, no. 1, pp. 239–254, 1998.
- [19] N. Alujevic and P. Gardonio, "Decentralized feedback control systems in double panel," in *Proceeding of the International Conference on Noise and Vibration Engineering (ISMA '06)*, Katholieke Universiteit Leuven, Leuven, Belgium, September 2006.
- [20] D. G. Luenberger, "Observers for multivariable systems," *IEEE Transactions on Automatic Control*, vol. 11, no. 2, pp. 190–197, 1966.
- [21] N. W. Hagood and A. von Flotow, "Damping of structural vibrations with piezoelectric materials and passive electrical networks," *Journal of Sound and Vibration*, vol. 146, no. 2, pp. 243–268, 1991.
- [22] A. Badel, M. Lagache, D. Guyomar, E. Lefeuvre, and C. Richard, "Finite element and simple lumped modeling for flexural nonlinear semi-passive damping," *Journal of Intelligent Material Systems and Structures*, vol. 18, no. 7, pp. 727–742, 2007.
- [23] Q. Mao and S. Pietrzko, "Control of sound transmission through double wall partitions using optimally tuned helmholtz resonators," *Acta Acustica united with Acustica*, vol. 91, no. 4, pp. 723–731, 2005.
- [24] J. P. Carneal and C. R. Fuller, "An analytical and experimental investigation of active structural acoustic control of noise transmission through double panel systems," *Journal of Sound and Vibration*, vol. 272, no. 3–5, pp. 749–771, 2004.