Experimental identification of smart material coupling effects in composite structures

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Experimental identification of smart material coupling effects in composite structures

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Abstract. Smart composite structures have an enormous potential for industrial applications, in terms of mass reduction, high material resistance and flexibility. The correct characterization of these complex structures is essential for active vibration control or structural health monitoring applications. The identification process generally calls for the determination of a generalized electromechanical coupling coefficient. As this process can in practice be difficult to implement, an original approach, presented in this paper, has been developed for the identification of the coupling effects of a smart material used in a composite curved beam. The accuracy of the proposed identification technique is tested by applying active modal control to the beam, using a reduced model based on this identification. The studied structure was as close to reality as possible, and made use of integrated transducers, low cost sensors, clamped boundary conditions, and substantial, complex excitation sources. PVDF (PolyVinyliDene Fluoride) and MFC (MacroFiber Composite) transducers were integrated into the composite structure, to ensure their protection from environmental damage. The experimental identification described here was based on a curve fitting approach combined with the reduced model. It allowed a reliable, powerful modal control system to be built, controlling two modes of the structure. A Linear Quadratic Gaussian algorithm was used to determine the modal controller-observer gains. The selected modes were found to have an attenuation as strong as -13dB in experiments revealing the effectiveness of this method. In the present study a generalized approach is proposed, which can be extended to most complex or composite industrial structures when they are subjected to vibrations.

Keywords: Smart structures, Experimental Identification, Modal active control, Curved structure, Composite structure, PVDF, MFC.

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1. Introduction

In aerospace, civil and mechanical engineering applications, smart structure technologies using materials such as piezoelectric are commonly used for active vibration control and structural noise reduction. Indeed, such structures have considerable advantages in terms of vibration attenuation, strength, reliability, integration and low energy consumption, when compared to other technologies such as electromagnetic actuators (EMA) or magneto-rheologic transducers (MR). The transition from an academic to an industrial structure is the next step to be achieved in smart structure development. The use of piezoelectric composites has opened up interesting and promising possibilities in response to this challenge. Active composites such as Micro Fiber Composites (MFC) consist of thin PZT (Lead Zirconate Titanate) fibers imbedded in a polymer (Epoxy), covered with an interdigitated electrode pattern. Due to the specific design of MFC's, this type of actuator has high flexibility, high electromechanical coupling, and makes it possible to achieve distributed solid-state deflection. Contrary to the case of a standard monolithic PZT, these properties allow such actuators to be used in curved structures. Moreover, MFCs can be unobtrusively integrated into the composite structure, can be operated as sensors for structural health monitoring [1] and strain measurements [2] and can, of course, serve as sensors and actuators in actively controlled structures [3-7]. Several active control strategies can be used, such as co-located control, as proposed in [8]. In this case system stability is guaranteed and, advantageously, this technique does not require the use of a model. Conversely, this type of control cannot be selectively focused on the control of specific modes. Active modal control is a solution for the targeting of control energy only, in specific modes, such that the on-board amplifier mass and volume can be minimized. Little research has been carried out on the design of light, complex structures such as the curved composite beams discussed in the present study. However, most publications in this field discuss the active modal control of simple one-dimensional structures such as straight cantilever beams. Various examples can be cited, such as Bailey and Hubbard [9] who studied a distributed piezoelectric-polymer for the active vibration control of a cantilever beam. Meyer and Collet [10] used straight piezo-composite beams for the active isolation of electronic components. Modal control also makes it possible to minimize the number of control components, as described in [11]. Further research has been carried out in the case of on-board structures, to reduce the energy consumed by nonlinear modal control systems [12, 13], or by nonlinear one-dimensional structures using adaptive modal control [14]. Some recent studies have investigated combined active modal control and identification techniques. The aim of this approach is to construct an experimental model to feed the controller, thereby avoiding the difficult process of producing a reliable theoretical model of the system [15-16].

Nevertheless, there are various coefficients, such as the

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electromechanical coupling of the piezoelectric transducers, which still remain very difficult to measure. Many studies have focused on the characterization of these coefficients. Initially, Hagood and Flotow [17] defined the generalized coupling coefficients by measuring frequency changes resulting from variations in the stiffness of the transducers, from their short-circuit to their open-circuit values. Several complementary studies were carried out, in particular for the case of composite structures and MFC transducers, with the aim of combining theoretical approaches, numerical simulations, and experimental measurements [18-20]. These studies highlight the fact that the modeling of smart structures equipped with such transducers is problematic, due to the lack of relevant information provided on the suppliers' datasheets. Moreover, these estimations can also be difficult to implement, as a consequence of the shape of the transducers [21]. When associated with manufacturing dispersion (such as transducer location, gluing effects, glass fiber distributions, …), these difficulties make the construction of a reliable smart structure model a problem in its own right.

In the present paper, it is proposed to avoid such estimations by directly characterizing the Frequency Response Functions (FRFs) of each sensor/actuator pair, without identifying the individual behavior and coupling characteristics of each transducer. With this approach, all of the parameters which need to be estimated are experimentally identified at the same time, through the use of a curve fitting technique on the experimental FRFs. It is shown that, using this technique, the actuator contributions can not be separated from the sensor contribution on the FRFs. In practice, this identification process estimates the global coupling effects between the transducer (actuator-sensor) pairs and the structure. These parameters include the modal shapes, the location and shapes of the transducers, and many other physical characteristics (temperature, glue, etc., …) which can affect the coupling. Consequently, in order to tune the modal control during a simulation, the actuator effects are arbitrarily set to unity. The global dynamics of the FRF are thus associated with the "sensor part". This is a valid approach because the dynamic behavior of the composite structure is characterized by the FRFs measured between its transducers. In the present paper, the proposed identification and control strategies are described in section 2. In section 3, the capabilities of this technique are tested and validated on a complex structure, consisting in a clamped, curved, composite beam. The excitations are produced at the boundaries, simulating the fact that the structure would normally be embedded into a larger structure. The feasibility and efficiency of active modal control is then illustrated by a simple case study. The use of appropriate identification techniques allows the modeling step to be avoided, and is efficient in terms of establishing the dynamic structural model needed to feed the control algorithm.

2. Control and identification strategies

2.1. Modal control

The use of modal control makes it possible to concentrate the control energy into selected modes only, i.e. to implement a strategy to actively control (for example) the most energetic or the most damaging modes [22]. By targeting the control energy into these modes, the required electrical energy is minimized, and the number of active components is limited. This can be very important for on-board applications. The effectiveness of this model-based strategy depends on the accuracy of the model. Usually, the first step consists in building a modal state model of the smart structure, which includes the mass and stiffness of the transducers. An observer is then developed, based on this model. Finally, the control loop can be designed according to the reconstructed state obtained from the observer. The command needed to monitor each mode is optimized, with the controller gains being computed by means of a specific algorithm. When the structure is lightly damped and the modes are sufficiently decoupled, the linear system is described by a set of decoupled modal equations and the corresponding modal state variable form is given by:

$$\begin{cases} \dot{x} = Ax + Bu + Ew \\ y = Cx + Du \end{cases}, x = \begin{Bmatrix} q \\ \dot{q} \end{Bmatrix}, \quad (1)$$

where x is the state vector, $\dot{q}$ and q are the modal velocities and modal displacements, respectively, u is the control vector, w is the disturbance noise, y is the output vector, A is the dynamic system matrix, B and C are the input and output matrices, D is the feedthrough, and E is the disturbance input matrix. The matrix D is chosen to be null, due to the specific choice of light, flexible sensors, which are fully integrated inside the structure. The other matrices are estimated through the use of an identification technique described below. The final modal system can be written as:

$$A = \begin{bmatrix} 0 & I \\ -diag(\omega_i^2) & -2diag(\xi_i\,\omega_i) \end{bmatrix}_{2n,2n},$$

$$B = \begin{bmatrix} 0_{n,N_s} \\ B_i^l \end{bmatrix}_{2n,N_s}, C = \begin{bmatrix} C_i^k & 0_{N_a,n} \end{bmatrix}_{N_a,2n}, \quad (2)$$

where ω_i is the frequency in rad/s and ξ_i is the modal damping of the i^{th} mode. N_s and N_a are the numbers of sensors and actuators, and n is the number of modes under consideration. With this type of control system, the values B^l_i and C^k_i are usually estimated from the generalized electromechanical coupling parameters determined for each transducer (sensor k and actuator l) and each mode.

The classical LQG algorithm is used to determine the controller gains and the control signal *u(t)* which minimize the energy cost function:

$$J = \int_0^\infty (x^t Q x + u^t R u)dt \quad (3)$$

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where Q and R are weighting matrices. The solution to this problem is given by linear constant modal gain feedback:

$$u = -K\hat{x} \qquad (4)$$

where K is the solution to the LQR problem and $\hat{x}$ is the reconstructed state obtained from the classical Luenberger observer, which is designed as

$$\begin{aligned} e &= y - \hat{y} \\ \dot{\hat{x}} &= A\hat{x} + Bu + L(Cx - C\hat{x}) \end{aligned} \qquad (5)$$

where L is the observer gain.
Figure 1 illustrates the control loop design used in the case of the proposed setup.

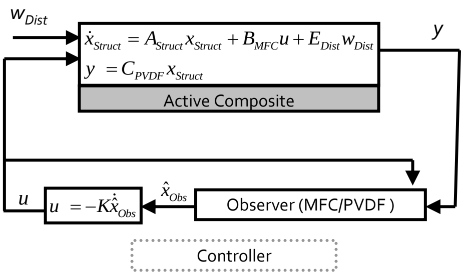

Figure.1. Control loop principle.

2.2. Identification techniques

The smart structure model must be as accurate as possible, to ensure that the control system has adequate performance and stability. The damping and natural frequencies can easily be measured for each mode. However, the generalized electromechanical coupling coefficients of each transducer are usually needed, in order to estimate the input and output matrices. Many approaches have been proposed in academic case studies, for these estimations. In the case of a complex structure with MFC or PVDF transducers, these methods do not appear to be appropriate. For example, in the well known study of Hagood and Flotow [17], it is proposed to obtain the generalized coupling coefficient from a simple experiment, involving the measurement of frequency changes resulting from variations in the stiffness of the transducers, between their short circuit and open circuit values. This method is not well adapted to PVDF sensors, due to their low coupling coefficient and high sensitivity to fluctuations in environmental conditions, and to temperature in particular. Thus, in the present study, it is proposed to use an estimation of the FRFs corresponding to the transducer pairs, combined with the identification techniques used to construct a reduced model. Clearly, one of the main advantages of this approach is the fact that it is based on the use of measured FRFs only. No models are needed to describe the structure, the transducers or their coupling. This method, together with its assumptions and limitations, is described in the following.

Firstly, the frequency response function of each sensor-actuator pair is measured. Using the RFP (Rational Fraction Form) algorithm [23], this function is then fitted by a sum of n rational, second-order fractions:

$$H_{k,l}^{ident}(s) = \sum_{i=1}^{n}\left[\frac{M_i^{k,l}e^{j\phi_i^{k,l}}}{s^2 + 2\xi_i\omega_i s + \omega_i^2}\right], \qquad (6)$$

where $H^{ident}{}_{k,l}(s)$ is a FRF written in the Laplace domain, $M^{k,l}{}_i$ is the modal magnitude, and $\phi^{k,l}{}_i$ is the modal phase of mode i, between sensor k and actuator l. The number of modes n must be sufficient to accurately reconstruct the dynamics of the estimated FRF. However, the modal state space system must not include all of the modes in the control loop. The number of modes selected for the state space model depends on the number of transducers and the purpose of the control system.

When the structure is described by a state space system, the transfer matrix (Sensor Voltage / Actuator Voltage) *H(s)* is given in the Laplace domain by:

$$H(s) = \frac{y(s)}{u(s)} = C(sI - A)^{-1}B, \qquad (7)$$

In practice, the denominator allows the matrix A, containing the dynamics of the system, to be constructed. Theoretically, this matrix remains unchanged for each estimated FRF. Using equations (6) and (7), and considering each mode independently, the magnitude, phase, actuation, and sensing vectors are related by the expression:

$$M_i^{k,l} e^{j\phi_i^{k,l}} = C_i^k B_i^l, \qquad (8)$$

The products $C^k{}_i B^l{}_i$ can thus be estimated by measuring all possible FRFs between the sensor-actuator pairs, and using RFP post-processing to obtain suitable curve fits. It should be noted that although this method allows the products $C^k{}_i B^l{}_i$ to be estimated, it does not allow each vector to be constructed independently. Although it is not possible to separate and clearly identify the contributions from the sensors and actuators in the FRFs, the present study shows that knowledge of this product is sufficient to determine the most suitable controller. This outcome is a key aspect of the proposed method. Indeed, the matrices B or C are usually calculated using an experimental estimation of the generalized electromechanical transducer coupling coefficients. In the case of PVDF sensors, this coefficient is so weak that it is very difficult to measure with precision and repeatability. The proposed approach avoids these estimations and the construction of independent matrices B or C. Indeed, the control law and model are constructed in order to verify equation (8), which represents a condition on the product $C^k{}_i B^l{}_i$. Physically, this product represents a global coupling effect, including the behavior of the structure, actuators and sensors.

In the case study described later in this paper, only one actuator is used. Matrix B has only one column, 2 modes are controlled, and all of its non-zero components are arbitrarily set to unity.

$$B = \begin{bmatrix} 0 & 0 & 1 & 1 \end{bmatrix}^T \qquad (9)$$

Then, using the reconstructed FRFs derived from the curve fitting algorithm, the components in matrix C can be estimated from equation (8). The output matrix can then be written (for two sensors and 2 modes) as:

$$C = \begin{bmatrix} M_1^1 e^{j\phi_1^1} & M_2^1 e^{j\phi_2^1} & 0 & 0 \\ M_1^2 e^{j\phi_1^2} & M_2^2 e^{j\phi_2^2} & 0 & 0 \end{bmatrix} \qquad (10)$$

Once the state space system has been constructed, the observer gains L and weighting matrices Q and R can be tuned by simulation using this reduced model, before the experimental implementation. For practical reasons, only the real part of the C matrix is integrated into the control loop. The phase shifts ϕ_i^k are introduced

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during the signal processing step, using equivalent delays.

Fig. 2 summarizes the proposed approach.

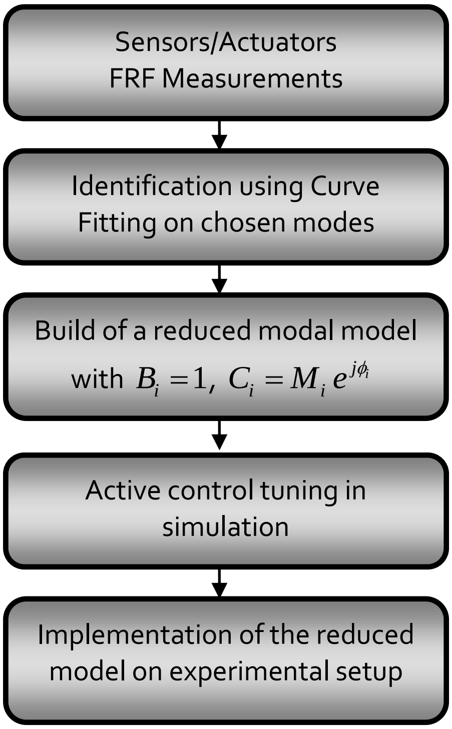

Figure. 2. Diagram illustrating the proposed approach.

3. Experimentation

3.1. Experimental setup

In the following sections, the efficiency of the identification step is demonstrated experimentally, by applying active modal control to a smart structure. The test structure and setup were selected in order to test the proposed method under conditions as close as possible to those of a real composite structure in an industrial context.

The smart structure was manufactured by the M3M laboratory, UTBM – France, and was comprised of several different layers: 4 glass fiber layers and 2 active layers including the active components (two MFC and two PVDF transducers). It was covered by 2 layers of a glass fiber textile, to protect the active components. “Resin Transfer Molding” (RTM) was used to produce the complete structure, in an aluminum mold. Figs. 3 and 4 show the configuration of the various layers, and the positions of the transducers. The MFC actuators were M8507P1 components from Smart Material, and the PVDF sensors were standard low-cost transducers procured from the NEC Company. Table 1 summarizes the characteristic of these transducers.

Table 1: Transducer characteristics according to the supplier’s datasheet.

Transducer	Reference	Capacitance (nF)	d33 (pC/N)	d31 (pC/N)
MFC	M8507P1	1.53	$4.6*10^2$	$2.1*10^2$
PVDF	DT2-042-K/L	1.44	$3.3*10^1$	$2.3*10^1$

It is interesting to note the very low electromechanical coupling coefficient of the PVDF sensors.

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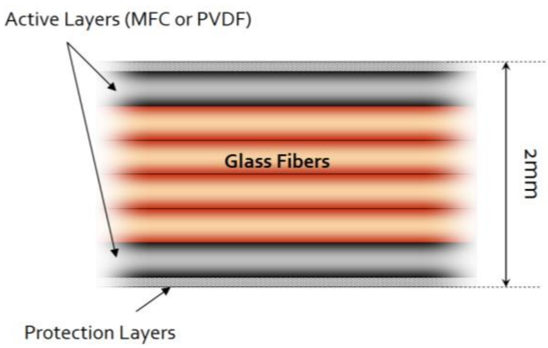
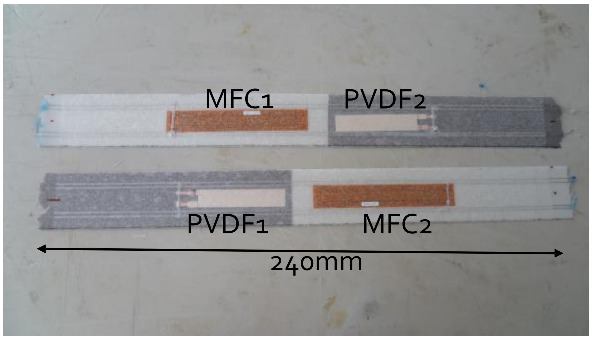


Figure. 3. Scheme of the different layers of the composite.

Furthermore, the beam was perfectly symmetrical and the relative positions of the transducers were chosen to allow the first bending modes to be controlled. Fig. 5 shows the structure when alone, and when mounted on the test setup. The structure was clamped at both ends. To simulate integration of the composite into a moving vehicle, vibration disturbances were applied directly to the peripheral supports by means of electrodynamics shakers. Since the smart structure excitations originate from its periphery, the excitation spectrum is polluted by resonance effects in the supports, thus simulating an on-board application.



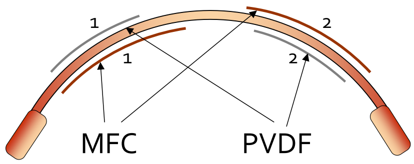


Figure 4. . Locations of the PVDF sensors and MFC actuators inside the glass fiber layer;

The curved beam had a total length of 240mm, with two additional blocks placed on this 45mm edge, in order to achieve the correct clamped boundary conditions. The structure was approximately 30mm wide and had a total thickness of 2mm. These dimensions were chosen, following simplified finite element analysis of the structure, in order to ensure that the modes would be decoupled at low frequencies.

For reasons of simplicity, and to emphasize some of the limitations of the proposed approach, only one (MFC2) actuator was used in this study. For similar reasons, only two (MFC1 and PVDF1) sensors were used in the control loop (their positions are shown in Fig. 4). PVDF2 sensors were not used in this study.

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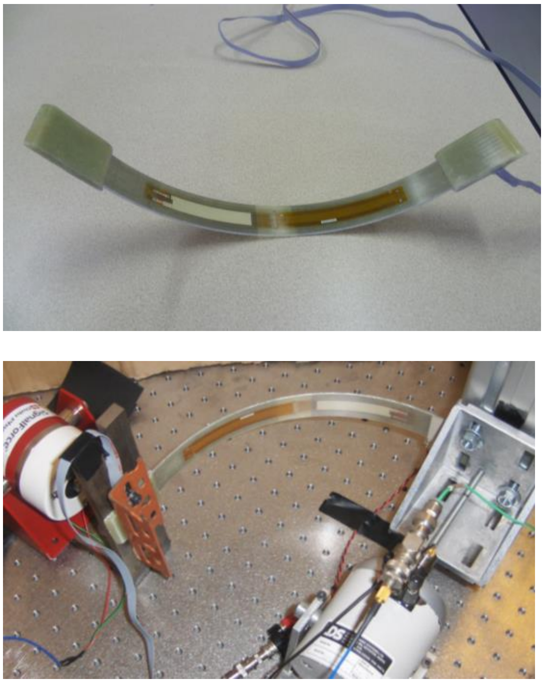

Figure 5. (a) Curved composite beam with integrated PVDF sensors and MFC actuators; (b)) Composite structure installed on the test setup.

A diagram of the test setup is shown in Fig.6. A Dspace interface was used to achieve real-time control of the structure, which involved recovery of the output voltage from the piezoelectric sensor, calculation of the controller gains, and transmission of the command signal to the actuator, via a power amplifier. Various additional sensors were used to gain improved insight into the global dynamic behavior of the setup. An external PVDF sensor glued to the skin of the beam, at its edge, was also used to evaluate the influence of the control system. It is important to note that only the transducers integrated into the composite structure (referred to as "in-loop sensors" in the following) were used as sensors for the control loop, with the other sensors being used to evaluate excitation levels and the performance of the control system.

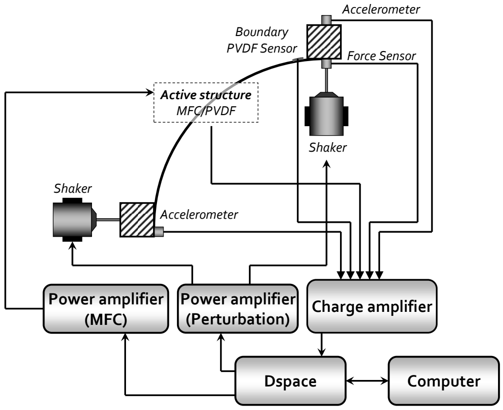

Figure 6. Experimental set up scheme

With this setup, the disturbances are characterized by the fact that the shakers are connected to the peripheral supports. The excitations felt by the beam result from the addition of two uncorrelated random noises generated by the shakers, together with the vibrations produced by the supports. Fig. 7 shows 3 accelerometer measurements recorded on one of these supports: they were measured above, in front of, and below the attachment point. It should be noted that the power spectral density does not remain constant over the full frequency range, because the support resonances create a complex form of excitation, the spectra of which differ from one measurement point to another. The beam is also excited by torsional vibrations. Due to the chosen transducer configuration, these modes cannot be controlled.

As the experimental test setup was designed to be as realistic as possible, the complex excitations felt by the embedded structures, associated with rich spectra, provided a good test environment for the evaluation of the robustness of the control system.

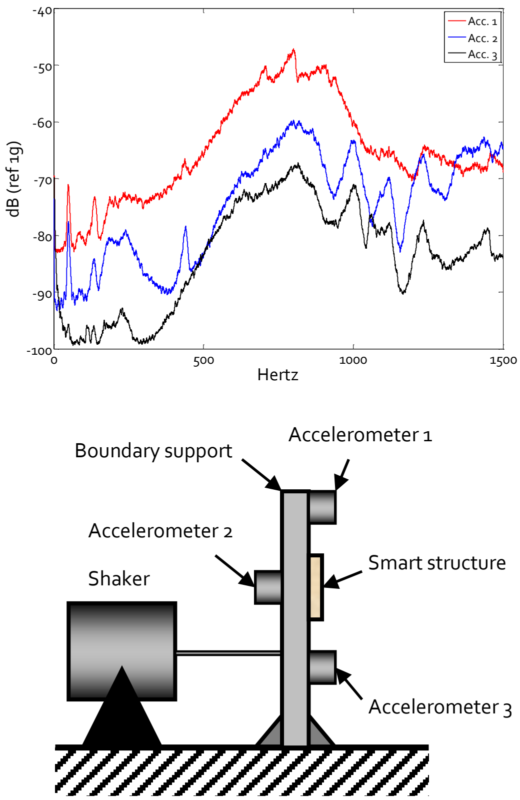

Figure 7 **(a)** Support accelerations (Acc. 1: above the attachment point, Acc. 2: in front of the attachment point and Acc. 3: below the attachment point). **(b)** Diagram of the boundary configuration (the smart structure attachment system is not shown).

3.2. Identification and control loop

One of the main steps conditioning the success of the proposed method is the identification process. Indeed, manufacturing dispersion effects, involving numerous factors such as resin distribution, transducer locations, composite delaminations, etc… can have an enormous influence on the FRF. Rather than accurately identifying all aspects of the structure's dynamic behavior (in- and out-of-plane bending, torsion…), the novelty of the proposed approach is based on the identification of the dynamics of the measurable frequency response function only, between actuators and sensors, namely the out-of-plane bending frequencies.

Since the tested structure has two sensors and one actuator, two FRFs are used for this demonstration. The corresponding measured FRF are shown in Fig.8, in which four bending frequencies can be clearly identified: 207Hz, 400Hz, 707Hz and 1018Hz.

Fig. 8 also illustrates the reconstruction of these FRF using the aforementioned curve fitting algorithm, with ten numerical modes. As can be seen, when a sufficient number of modes is selected for the identified model, the FRF can be estimated very accurately. Nevertheless, in order to provide the simplest possible illustration of the proposed method, and since only two sensors and one actuator were used, only two modes were included in the control loop. These two modes (208 Hz and 707 Hz) were chosen arbitrarily and the curve fitting technique was applied on either side of each resonant frequency, in order to produce a simple model with two degrees of freedom (DoF). Fig. 9 shows the measured FRFs, together with the FRFs of the reconstructed 2 DoF model. It can

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be seen that the global dynamic behavior obtained with the reconstructed FRF is quite different. However, this difference does not affect the control loop being designed, since this is based on the use of two specific modes

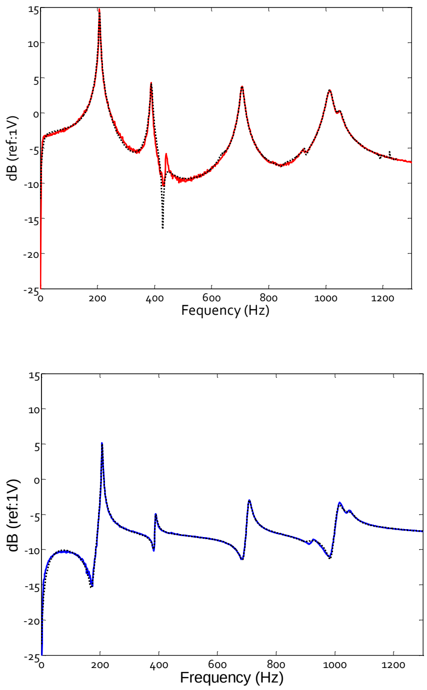

Figure 8. Measured FRF (continuous line) and fully identified model (dotted line, 10 modes). **(a)** MFC1/MFC2 **(b)** PVDF1/MFC2.

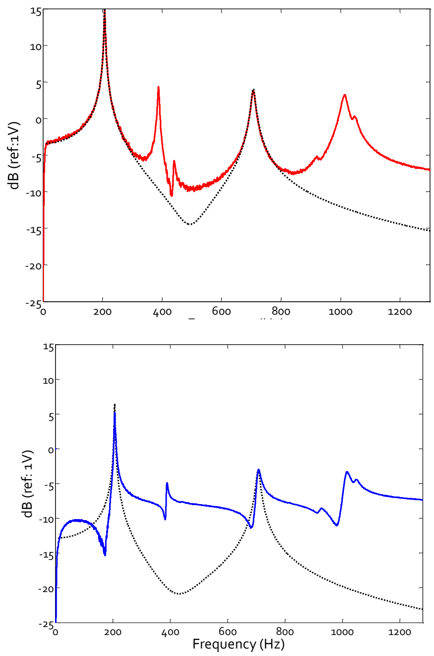

Figure 9. Measured FRF (continuous line) and FRF of the simplified model (dotted line, 2 modes).
(a) MFC1/MFC2 **(b)** PVDF1/MFC2.

Table 2 lists the modal parameters of the model. The modal parameters derived from the identification are inserted into the modal state model defined by equations (2), (9) and (10), leading to the construction of a reduced modal model (see Fig.2). As described above, the phases are generated numerically through the use of delays during signal processing in Dspace. Based on this 2-DoF model, active modal control is established (see Fig.1). Fig. 10

Table 2: Identified modal parameters.

			FRF *MFC1/MFC2*		FRF *PVDF1/MFC2*	
Mode	Frequency(Hz)	Damping	Phase (deg)	Magnitude	Phase (deg)	Magnitude
1	207	$5*10^{-3}$	160	$7.0*10^{5}$	-13	$7.5*10^{4}$
2	707	$9*10^{-3}$	135	$9.0*10^{6}$	-49	$1.75*10^{5}$

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plots the simulated attenuation which could be expected, i.e. approximately between 8 and 14dB, in the FRF spectrum, with and without control. Fig.11 indicates the locations of the poles of the structure, with and without control, and the poles of the observer.

The dynamics are highly simplified in this case, and it is thus straightforward to tune the control system. The main goal of this study was to validate the proposed approach, and to assess the possibility of implementing active modal control in a complex composite structure. As shown above, the global dynamical behavior of the real smart structure will be far more complex (support resonances, numerous flexural modes and torsional modes, etc...) than those of the simplified model, and the overall control system performance is likely to be lower than that found by the simulations previously described. It was thus decided to select control and observer gains which would not be too high, in order to provide sufficient stability during the tests, during which many uncontrolled and non-simulated phenomena could occur. The weighted matrices and the observer gain are thus:

$$Q = 1e^{5}\begin{bmatrix} 1 & & & \\ & 1 & & \\ & & 2 & \\ & & & 1 \end{bmatrix}, \quad R = 1, \quad L = 1e^{4} \cdot$$

With these parameters, the expected FRF attenuations of the in-loop sensors are -14dB and -8dB, for the first and second controlled modes.

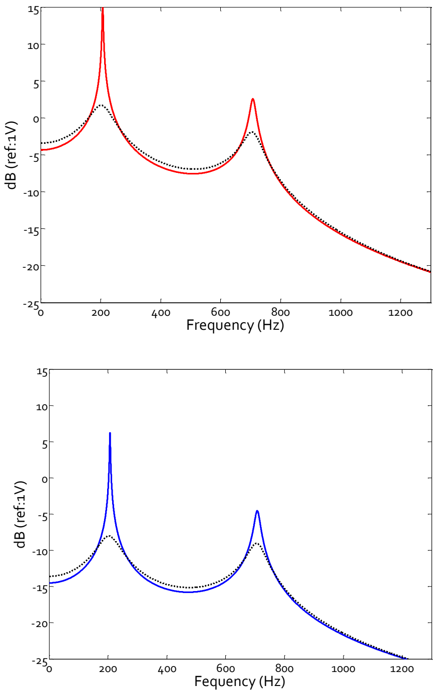

Figure 10 FRF of the control loop with no control (continuous line) and with control (black dotted line)
(a) MFC2/MFC1 **(b)** PVDF2/MFC1
Poles location (Blue circle: uncontrolled structure,

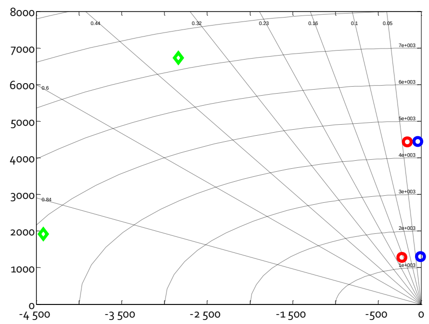

Figure. 11. Red circle: Controlled structure, Green square: Observer).

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3.3. Control results

Disturbances are generated by the 2 shakers, driven by two uncorrelated random noise. Fig. 12 shows the power spectral density (PSD) of the two in-loop sensors. The continuous line represents this PSD without control, and the dotted line shows the PSD with control. As could be expected, the dynamic behavior is considerably more complex than the FRFs of the transducers (Fig. 8). It can be seen that the two modeled modes are very well controlled (attenuations of approximately -12dB and -8dB). These values are similar to the attenuations predicted by simulation (-14dB and -8dB). The two remaining out-of-plane bending frequencies (400Hz, 1018Hz) are still present, but have much lower amplitudes than the modes produced by a realistic excitation. Unfortunately, some spillover occurs, especially around 410Hz and 1150 Hz. In future tests, these modes will need to be added to the control loop, or at least to the observer. Some non-observed modes (120Hz, 650Hz, etc…) remain unaffected by the control system. These are related to the support vibrations, or are modes (e.g. torsional modes) which the MFC actuator is not able to excite. They are nevertheless present in the sensor signal, and can perturb the observer.

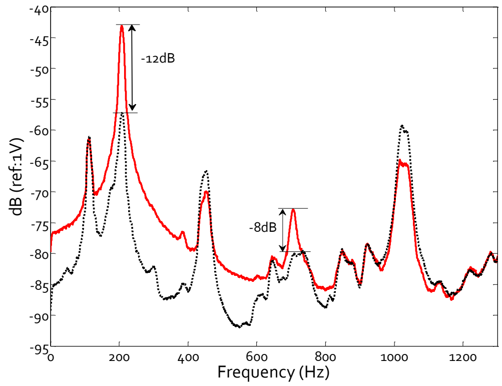

Figure 12. Power Spectral Density of in-loop sensors **(a)** MFC1 and **(b)** PVDF1 without (continuous line) and with control (black dotted line)

Fig. 13 shows the power spectral density of the external PVDF sensor. This external sensor allows the dynamics of the smart structure and its environment to be recorded. The attenuation and spillover are different to those observed with the in-loop sensors. However, it should be noted that the vibrations are reduced on the focused modes (-4dB and -7dB), and that very good performance is still achieved with the second controlled mode, which has almost disappeared. The weak attenuation of the first mode can be explained by the fact that this sensor is poorly coupled to the actuator at frequencies close to the frequency of the first mode. This is shown in Fig. 14, which represents the coherence function between these transducers. From Fig. 14 it can be assumed that over this frequency range, the measurements recorded by the peripheral PVDF sensor are not clearly representative of the smart structure vibrations. It is very likely that the measurements are significantly perturbed by the support vibrations, which lie on the paths followed by the external excitation. It is nevertheless important to note that the sensors inside the structure, which make more direct measurements of the smart structure vibrations, reveal the control system's efficiency in attenuating the first controlled mode (Fig 12).

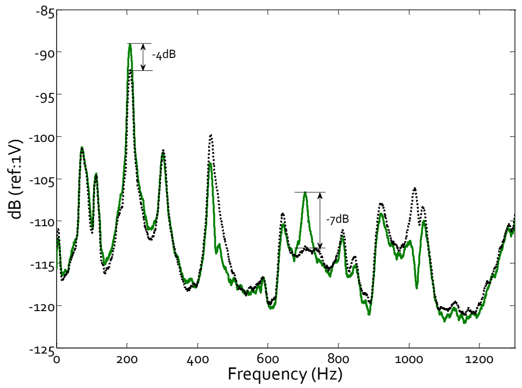

Figure 13. Power Spectral Density of the external PVDF sensor without (green continuous line) and with control (Black dotted line)

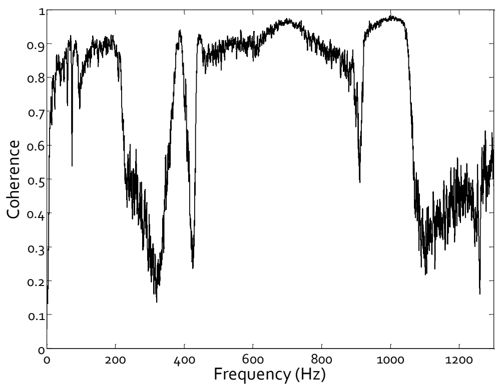

Figure 14. Coherence function between the boundary PVDF sensor and the MFC actuator

For on-board applications, it is important to consider the power supply voltage needed to obtain these results. Fig. 15 provides a plot of the PSD control voltage at the MFC actuator and its value, over a period of 0.3 seconds. Throughout the duration of all control system measurements, this voltage remained very low. This is one of the advantages of modal control, since the main part of the energy is focused on the controlled modes. Nevertheless, it is important to note that the vibration excitation levels at the boundaries were quite small (Fig.7), due to the specific configuration of the shakers.

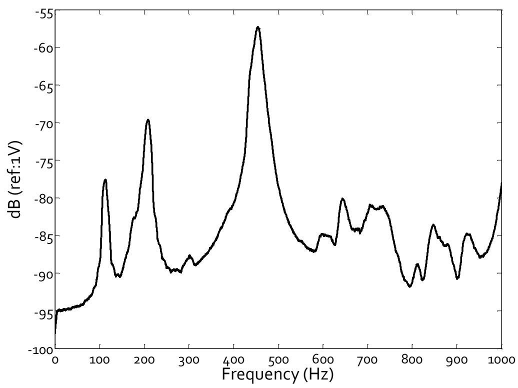

Figure 15. Power Spectral Density of the Control Voltage, and its value during 0,3 seconds

4. Conclusions

An experimental identification technique is proposed, for the estimation of smart material coupling effects for integrated transducers. This approach uses curve-fitting techniques on measured frequency response functions. Active control based on this reduced experimental model has been successfully implemented on a curved composite beam. The transducers were fully integrated into the structure. No external source was required for this characterization. This model was used to supply the controller and the observer. It has been shown that the usual modeling step, which is difficult to use for such structures, can be avoided. Moreover, the electromechanical coupling coefficients of the sensors and actuators are not required with this approach. The non-zero values of the state space model input matrix are forced to unity. Indeed, the influences of the actuators or sensors are not considered independently, since they are related through the identified FRF. In view of these practical advantages and simplifications, this approach could clearly be applied to many different structures. The effectiveness of this simple method depends on the extent to which the FRFs of the actuator/sensor pair are representative of the structure's global dynamic behavior.

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