EXPERIMENTAL EVALUATION
OF A NEW ACTIVE VIBRATION CONTROL ALGORITHM

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Abstract: This work deals with little mock up experimentations of a vibration rejection algorithm. In a first time, bases of the algorithm are presented. Next section is devoted to the description of the prototype used to validate the process. Then, the settling of the algorithm using Matlab Simulink XPC Target is described and experimental results are presented and commented. From these results, future works are outlined. *Copyright C 2005 IFAC* .

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1. INTRODUCTION

The improvement of measurements tools such as optic ones needs a drastic reduction of vibrations phenomena in the mechanical structures which carry them. It is also a crucial problem for the next generation of collider (S. Redaelli, 2003).

There are a lot of vibration sources: in particular cultural noise is evocated in the case of ground source which spectrum is almost continuous in the concerned frequency range. Specific installation vibrations which spectrum presents more localised peaks are added (S. Redaelli et al, 2002). The mechanical structure itself finally spreads vibrations in relation with their frequencies. That is for these multiple reasons that passive damping solutions are inadequate to match the required performances.

The work presented in this paper consists in a progressive reduction of vibration effects regarding the most important modes. This approach relies on modal independence. The previous assumption could be discussed when non linear behaviour is considered.

Nevertheless, as far as our goal is to converge towards zero (that is to say stabilization), the process is available.

The principle is the superposition of two excitations for the concerned modes: one is brought by the perturbation, the other one is produced by a controlled excitation such that their combined effects are null or as weak as possible regarding the sensor resolution at the measurement location.

The kinematic behaviour of the structure allows choosing optimum localization of sensors and actuators in order to realize a global efficient attenuation and not only for the nodes of the system.

Section 2 is devoted to the description of the used algorithm to generate the controlled excitation from the sensor measurements (Corduneanu, 2003). For each mode to be attenuated, a generalized observer rebuilds a virtual perturbation which acts on the system and a state feedback control law gives the contribution to be applied. The excitation is evaluated by summing each mode contribution.

Section 3 presents the little mock-up used for experimentations. It consists in a small thickness metal cantilever. A ceramic patch is glued on it and is used to generate the excitation. An accelerometer sensor provides a signal in relation with the vibration amplitude at the free end of the beam. This mock-up is connected to a PC analog I/O board equipped with the Matlab Simulink XPC Target software.

Experimental testing and results are described in section 4 (Cozma, 2004). They allow, on the one hand, validating the process and on the other hand analysing the robustness regarding parameters precision, mode frequencies to be attenuated in particular.

Finally, after a conclusion, several perspectives are suggested in order to improve the proposed process performances.

2. REJECTION ALGORITHM

Different methods aiming at vibration effects attenuation have been proposed in the literature for mechanical structures (Agrawal and Bang, 1996; Cozma, 2004; Fanson and Caughey, 1990). They often rely on a precise modelling of the dynamical behaviour of the system.

The goal of this study is to release from the knowledge of such a modelling. For this purpose, following considerations are made:

- we first make the assumption of the stationary of the perturbation characteristics,
- priority is given to the attenuation of the undamped modes whose frequencies is well known. This information is provided by a parallel spectral analysis of the sensor signal,
- modal components are supposed to be independent and can be processed in parallel. The control variable is then calculated by summing the relative contribution of each of them.

The basis of the algorithm is a PI control scheme which allows cancelling the stationary error even if the dynamical behaviour of the system is unrecognised. To transpose the properties of such a control scheme to present case, following relations are set:

- the set point is zero since as sensor amplitude should tend to zero,
- the dynamical behaviour of the system squares with the setting up dynamic of the amplitude of a sinusoidal signal between the excitation and the measured signal: here, a simple first order model is used which leads to satisfactory results.

But, for each sinusoidal component, amplitude is not sufficient, phase must also be considered. In order to make the process of these two characteristics easier, a sine-cosine based decomposition for each component is rather used. It leads to process a multi-input multi-output problem which is usual in the state space.

The analysis of the method is detailed in (Corduneanu, 2003; Lottin and Corduneanu, 2004). Major elements of the control scheme are given below:

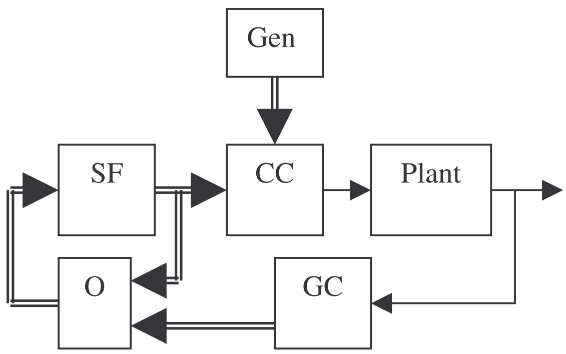


Fig. 1. Overview of the control scheme

- The block denoted GC corresponds with the computation of the sine and cosine components, y_s and y_c, of the sensor signal $y(t)$ for the given frequency.

$$y(t) = y_s \sin(\omega t) + y_c \cos(\omega t) \qquad (1)$$

- The block denoted O is a generalized observer whose function is to estimate the amplitudes of the sine and cosine components of a perturbation which is assumed to act on a particular point of the structure for a given frequency.

$$\begin{cases} \dot{\hat{x}}_e = A_e \hat{x}_e + B_e u + L(y - \hat{y}) \\ \hat{y} = C_e \hat{x}_e \end{cases}, \quad \hat{x}_e = \begin{bmatrix} \hat{x} \\ \hat{p} \end{bmatrix} \qquad (2)$$

with

$$\hat{x}(t) = \begin{pmatrix} \hat{y}_s \\ \hat{y}_c \end{pmatrix}, \quad u(t) = \begin{pmatrix} f_s \\ f_c \end{pmatrix}, \quad \hat{p}(t) = \begin{pmatrix} \hat{p}_s \\ \hat{p}_c \end{pmatrix} \qquad (3)$$

- The block denoted SF builds the sine and cosine components of the excitation whose aim is to compensate the perturbation effect.

$$u(t) = K_x x(t) + K_p \hat{p}(t) \qquad (4)$$

- The block denoted CC builds excitation signal *f(t)* by weighting sine and cosine signals produced by the block denoted Gen according to f_s and f_c values produced by SF.

$$f(t) = f_s \sin(\omega t) + f_c \cos(\omega t) \qquad (5)$$

Simulations have shown a good robustness with respect to bad knowledge of the dynamical behaviour of the system. As far as the frequency value is considered, the robustness has been not so good. That is the reason why experimental testing began.

3. PLATFORM

The following scheme (Fig. 2) describes the whole platform used for testing:

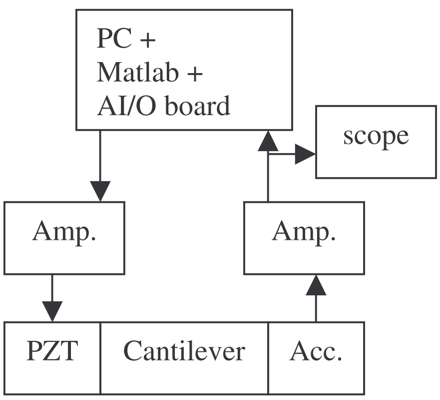


Fig. 2. Experimental set-up (principle)

Its different parts are:

- an instrumented cantilever. Piezoelectric patch is the actuator whereas an accelerometer stands for the sensor,
- associated amplifiers
- a computer equipped with an analog I/O data acquisition board and Matlab software.

By another way, a known perturbation source realized with a loudspeaker has been added in order to make the algorithm implantation easier.

A balance has to be found between resonant modes of the beam, piezoelectric patch bandwidth, sampling frequency.

Of course, in other cases, the actuator would be adapted, but the general process would remain roughly the same. In the present case, in order to perform two frequencies at the same time, XPC Target instead of Real Time Windows Target has been used.

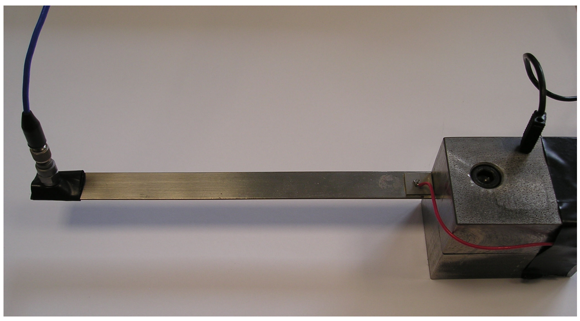


Fig. 3. Mock-up: instrumented cantilever

4. TESTS AND RESULTS

4.1 Plant dynamical behaviour

The first test consists in evaluating the transient behaviour of the sensor output when a sinusoidal signal is abruptly fed to the input. Only the envelope is considered here, since the algorithm is based on links between amplitude of sine and cosine signals of input and output.

Fig. 4 suggests that a first order model is enough to describe the transient behaviour. So a time constant is deduced from the shape of the envelope and is used as starting point to build plant model.

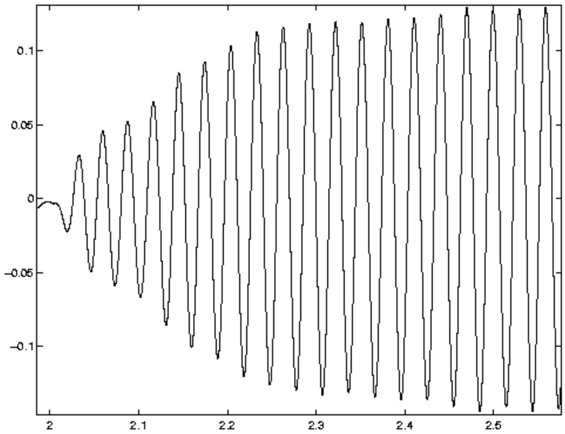


Fig. 4. Step response (amplitude/time in sec)

4.2 Single mode rejection

In order to make the robustness analysis easier, a single mode was first considered. The loud-speaker generates the perturbation. Assuming the knowledge of the frequency, the characteristics (phase and gain) of the sensor-actuator and the sensor-perturbation transfers can be easily found with good precision.

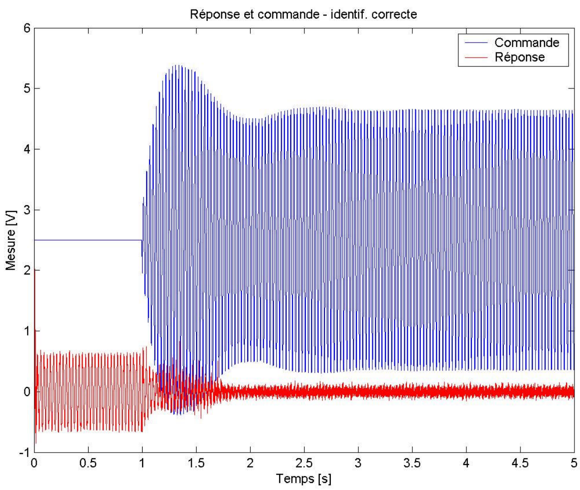
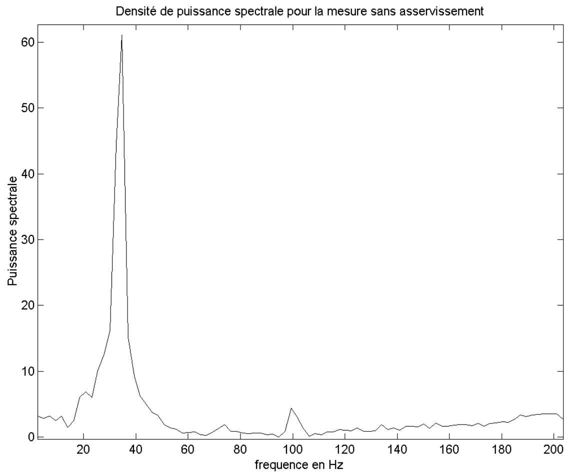


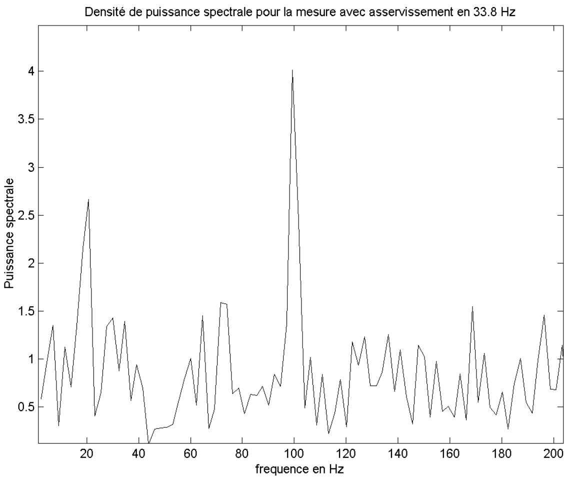
Fig. 5. Time response (ideal case)

Fig. 5 shows the sensor output (lower line) and the control signal (upper line), when the control loop is closed at time t=1s. It is easily seen that vibration amplitude is well reduced but does not converge to zero. The reason is that there is only one frequency which is attenuated, and a spectral analysis gives more information about the effect of the control scheme: see Fig. 6.

Thus, we dispose of the ideal case which is a reference for the following robustness tests.



a: before rejection



b: after rejection

Fig. 6. Comparison of spectra

4.3 Frequency mismatch

As it was already mentioned, the frequency of the mode is a crucial parameter for vibration reduction. Indeed, if frequencies of disturbance on the one hand and excitation on the other hand are not identical, the risk is to introduce beating on sensor output. It is why the first test is carried out in that direction. Before entering the GC block, the sensor output must be filtered by a narrow bandwidth filter in order to get accurate values of sine and cosine components.

This filtering prevents against beating since:

- if frequency gap is high, the vibration is not seen at the output of the filter, and then, there is no reaction by the control loop

- if the frequency gap is small, beating pulsation is low, and then the output of the filter follows this beating and allows vibration reduction, but with poor performances: see Fig. 7, in the case of 0.5% frequency gap.

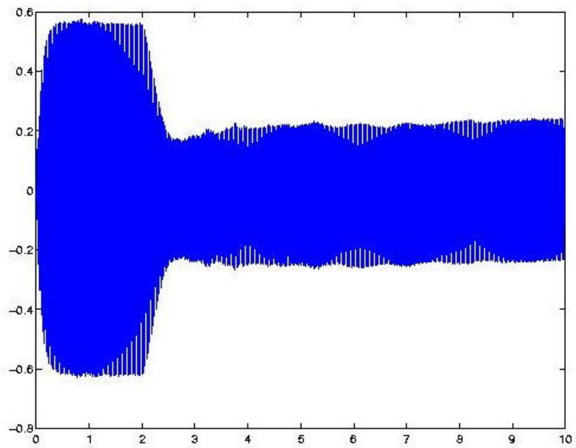


Fig. 7. Imperfect rejection due to frequency mismatch

4.4 Model mismatch

Coefficients in block SF are derived from the characteristics of the transfer between PZT actuator and accelerometer sensor. Usually, they are described by magnitude and phase for the frequency under consideration. In this approach, they are replaced by gains linking sine and cosine components at the input and output (2x2 matrix, but only two parameters are independent)

The aim of this test is to show that the algorithm is robust with respect to these parameters mismatch. To facilitate the analysis, Fig. 8 shows the behaviour of the envelope of the transient response when the control loop is closed at t=2s. Two situations are analysed: magnitude mismatch and phase mismatch.

It can be seen that large deviations of magnitude are acceptable. It is more difficult to quantify acceptable deviations of phase.

4.5 Observer tuning

Another important point is the tuning of observer parameters, namely poles which influence the convergence rate as well as damping of the transient behaviour.

The dimension of the observer is four. The nominal case corresponds to the tuning of observer so that it is about three times faster than the plant itself.

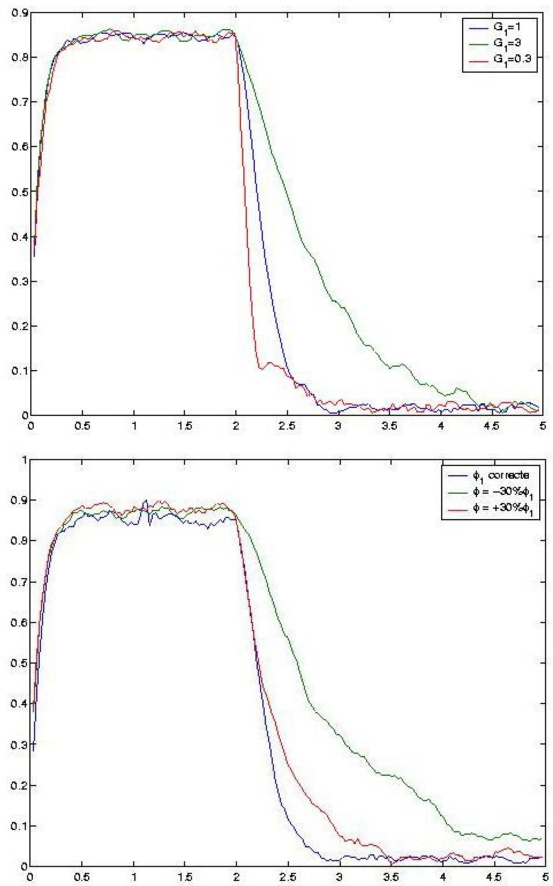


Fig. 8. Comparison of envelopes/time
(top: gain mismatch, bottom: phase mismatch)

Two cases are considered:

- the observer becomes faster: then robustness with respect to model mismatch is decreased.

- the observer becomes slower: then robustness with respect to model mismatch is increased.

Fig. 9 shows transient behaviour for each tuning without model mismatch.

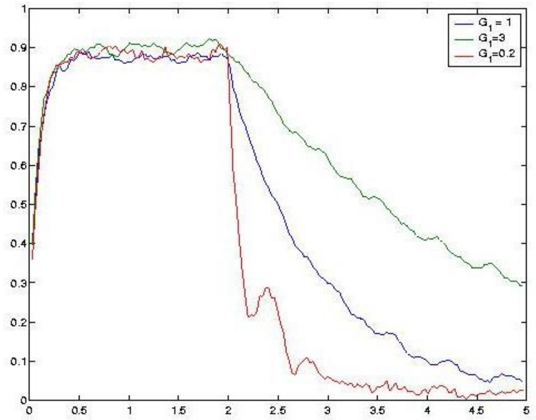


Fig. 9. Observer tuning (envelope/time)

4.6 *Two mode rejection*

The objective of this test is to reduce the effects of two modes simultaneously. It can be seen on Fig. 10 that four modes are candidate. Due to the structure of the GC block, it is not possible to process two modes that are too close. It is why, it was chosen to consider two separated modes. Results are very good. But it becomes necessary to simplify computations in order to increase the number of modes that can be processed in the frequency range up to 100 Hz.

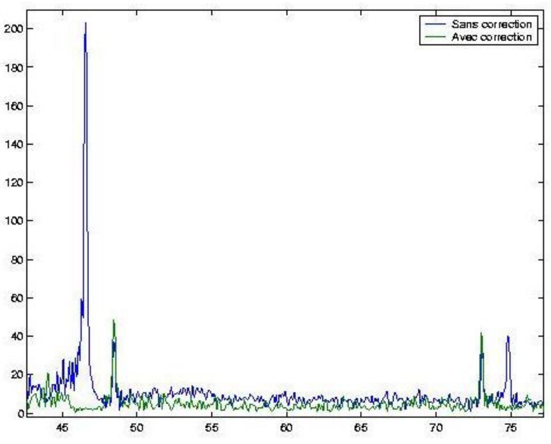


Fig. 10. Spectra comparison (Power/frequency)

5. CONCLUSION

This experimental set-up shows the feasibility of the method. Some improvements are possible in order to reduce computational burden without decreasing the quality of the performance. For example, the GC block was already modified in that way. Obviously attainable performance is related to sensitivity of PZT actuator and accuracy of the accelerometer.

One main interest of this procedure is that it does not require good knowledge of the dynamical behaviour of the plant, except for the frequency of the considered modes. It relies on good spectral analysis of the measured signal.

Another interesting feature of this method is the possibility of making computations in parallel. Indeed each mode is supposed independent and can be treated apart of the others. The final excitation is the sum of all contributions.

However, it is also necessary to deal with continuous spectrum. Work is in progress to extend this method to such cases. Obviously, it is related to inversion of transfer functions. The question is: how to make the system converge by itself to the ideal inverse ?

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