

# ACTIVE INSULATION OF AEROPLANES MOTORS

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## Abstract

This study aims to deal with the case of a jet aircraft, from whom the vibrations engendered by the residual unbalance from rotors propagate up to the fuselage of the plane and cause discomfort for passengers. We especially studied the active control at the connection between jet-engines with the airfoil. The great rotation speeds cause excitations with the gyroscopic matrix. The end is to actively control these vibrations, in a comfort aim. But the active control concept must be separated in several steps, progressive but indissociable to aim the effective reduction of vibration. Those steps can be separated in two main stages : simulations and implementation. A first feasibility study is presented, to show the possible decrease of vibration expected, and to optimize the actuators' positions. Then, we present an experimental demonstrator with hydraulic actuators on which first control tests have been realised. Those first tests are very important for the next experimental improvements

## Introduction

Since a few decades, structures have been optimised thanks to the emergence of finite element method. In consequence, the problems of dynamic vibrations has become harder and harder. Then, solutions has had to be found to attenuate this phenomenon, as passive isolation. But limits of such solution have been reached quickly. The only exit for such a problem must be a complete change of strategy.

The active control in vibration of turning machines, thanks to the advancement of data processing, has become a current problem.

The general concept of active control applied to an aircraft can be presented with the diagram (fig. 1).

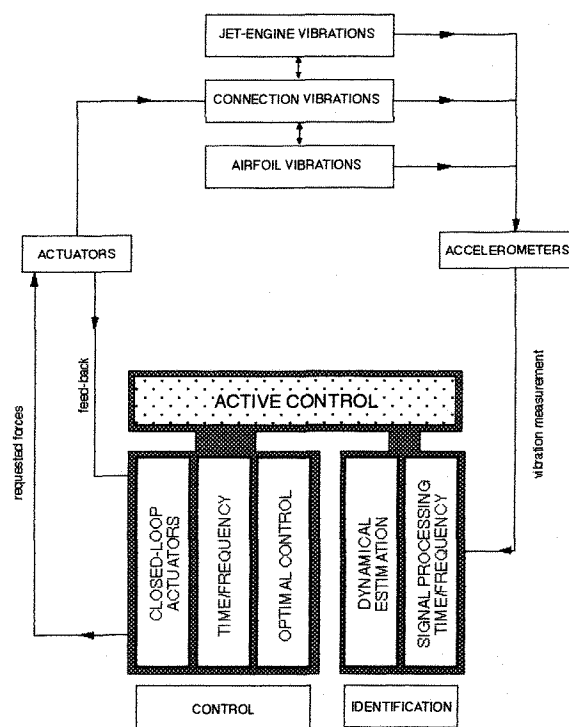


Figure 1: Plane control diagramm

We'll try to present very quickly in this article the different kinds of control algorithms, in time or frequency domain, in open-loop or closed-loop. Then , we'll present the results of a feasibility study for active control of jet-engines on an aircraft, and tests of optimal position of actuators. Then, we'll present the experimental demonstra-

tor and the results realised on it.

## Control law

### Frequency deterministic control

We can represent our mechanical system by the dynamic system equations of motion :

$$M\ddot{q}(t) + [C+G]\dot{q}(t) + Kq(t) = B_0u(t) + p(t) \quad (1)$$

with :

$q(t)_{(n \times 1)}$	nodal displacements vector
$u(t)_{(n_{act} \times 1)}$	actuation forces vector
$p(t)_{(n \times 1)}$	nodal disturbance vector
$M_{(n \times n)}$	symmetrical mass matrix
$C_{(n \times n)}$	symmetrical damping matrix
$G_{(n \times n)}$	gyroscopic skew matrix
$K_{(n \times n)}$	symmetrical stiffness matrix
$B_0_{(n \times n_{act})}$	actuators application matrix

For turning machines in steady state, we can assume sinusoid excitation, of pulsation  $\omega$ , so we can take:

$$\begin{aligned} q(t) &= \Re[\hat{q}e^{j\omega t}] \\ u(t) &= \Re[\hat{u}e^{j\omega t}] \\ p(t) &= \Re[\hat{p}e^{j\omega t}] \end{aligned}$$

where hat  $\hat{\phantom{x}}$  indicates complex number,  $\Re$  the real part of value. We have then<sup>[4] [1] [3] [6]</sup> :

$$\hat{q} = \hat{q}_0 + \hat{T}\hat{u} \quad \text{with} \quad (2)$$

$$\begin{aligned} \hat{q}_0 &= (\omega^2 M + j\omega[C+G] + K)^{-1} \hat{p} \\ \hat{T} &= (\omega^2 M + j\omega[C+G] + K)^{-1} B_0 \end{aligned} \quad (3)$$

We then define the performance index :

$$\mathcal{J} = \hat{a}^* Q \hat{a} + \hat{u}^* R \hat{u} \quad \text{with} \quad \hat{a} = -\omega^2 \hat{q} \quad (4)$$

$\hat{a}$  is the complex acceleration,  $*$  is the transconjugate operator,  $Q$  et  $R$  are positive semi-definite weighting matrix. The optimal control law  $\hat{u}^*$  for such a problem is given by :

$$\hat{u}^* = -(\hat{T}^* Q \hat{T} + R)^{-1} \hat{T}^* Q \hat{q}_0 \quad (5)$$

The main problem consists in finding the good weighting matrix.

We obtain here an open-loop control that allows only the control of harmonic vibrations. But as it is an open-loop control, characteristics of state equations are not changed (natural frequencies).

$\hat{q}_0$  can be seen as the system vibratory state with no control, or the transfert function between unbalances from rotors and the structural response. To determine  $\hat{q}_0$ , a classical estimator is needed, least mean square or Kalman estimator for instance.

### Stochastic time domain control LQG

In this section, the disturbance force is considered as a gaussian white noise  $w(t)$ , for a theoretical point of view.

#### State equation

From :

$$M\ddot{q}(t) + [C+G]\dot{q}(t) + Kq(t) = B_0u(t) + w(t) \quad (6)$$

We then define state vector  $x$ ,

$$x(t) = \begin{Bmatrix} \dot{q}(t) \\ q(t) \end{Bmatrix} \quad (7)$$

From (eq. 6), we obtain :

$$\dot{x}(t) = Ax(t) + Bu(t) \quad (8)$$

with :

$$\begin{aligned} A &= \begin{pmatrix} -M^{-1}[C+G] & -M^{-1}K \\ I_n & 0 \end{pmatrix} \\ B &= \begin{pmatrix} M^{-1}B_0 \\ 0 \end{pmatrix} \end{aligned}$$

#### Optimal control

We then<sup>[2]</sup> define a performance index to minimise again :

$$\mathcal{J} = \int_0^\infty (x^t(t) Q x(t) + u^t(t) R u(t)) dt \quad (9)$$

with  $Q$  and  $R$  positive semi-definite and definite weighting matrix. The control command  $u^*(t)$  is classically given by :

$$u^*(t) = -R^{-1} B^t S x(t) \quad (10)$$

with  $S$  Riccati symmetric matrix, minimal solution of stationary Riccati equation:

$$A^t.S + S.A + Q - SBR^{-1}B^tS = 0 \quad (11)$$

## Modeling stage

For a preliminary study of active control of vibrations on aeroplanes motors, we only took into account the part of the plane around an engine, that is to say the engine, the mechanical part joining engine to airfoil and the airfoil (fig. 2).

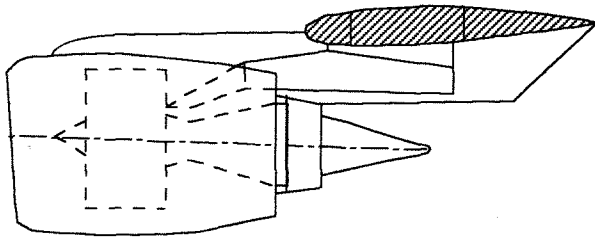


Figure 2: Studied part

To realise the feasibility study for active control of jet-engines on an aircraft, we had first to build a model of the structure to test the different types of control algorithms, frequencial or temporal, and to optimise the position of sensors and actuators. The model is to be adjusted to the real system. We realised the model of the system (jet-engine + airfoil) presented figure (fig. 3).

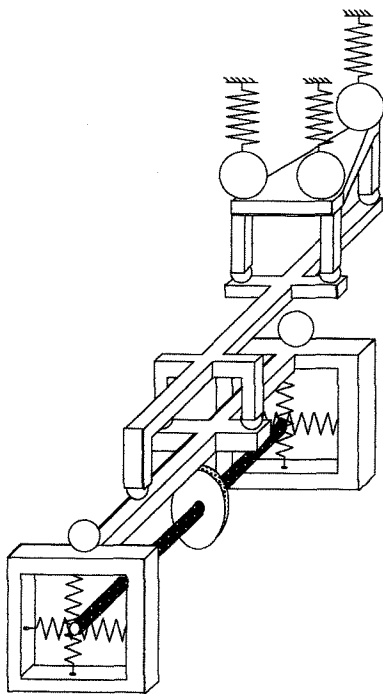


Figure 3: model of the system

We then tried to optimise the parameters of the model to reach a Campbell diagram as near as the Campbell diagram of a finite element method ob-

tained from the airframe manufacturer. We then have the two following diagrams presented at the figure (fig. 4).

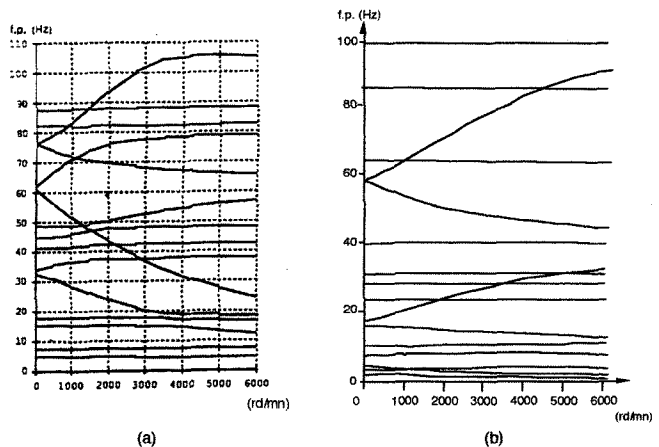


Figure 4: Campbell Diagram: (a) finite element model (b) beam model

In order to realise this feasibility study, we first assumed the perfect knowledge of the whole system continually, and perfect actuators to deliver ideal forces instantaneously. Displacements in the lenthwise of the engine had been neglected with regard to the other displacemet to reduce the number of degrees of freedom to 55. As the major vibrations are due to the Low Pressure rotor, we took into account as a generative force a single unbalance force of 1000 g.cm, that is to say a significant but operationaly acceptable.

### Placement of actuators

In an optimisation aim, we then tried several placements of actuators near the jet-engine or near the airfoil, and tested the case of two and three actuators.

### Frequencial control

We first simulated a frequencial optimal control whoses results are presented below. Here we took in regard for weighting matrix only the control of the three degrees of freedom on the wing. Other tests added rotor and shaft bearings d.o.f for weighting. We tried top minimize the 3 d.o.f. with no weight on the actuator forces, only to see theoritically what intensity of force is required to minimize vibrations. The performance index be-

comes :

$$\mathcal{J} = \hat{a}^* Q \hat{a} \tag{12}$$

The optimal control law  $\hat{u}^*$  for such a problem is then given by :

$$\hat{u}^* = - \left( \hat{T}^* Q \hat{T} \right)^{-1} \hat{T}^* Q \hat{q}_0 \tag{13}$$

The figure (fig. 5) shows that for our model and in stationary domain, between 40 and 60 Hz, we can reduce vibrations of 20 dB thanks to deterministic frequencial control.

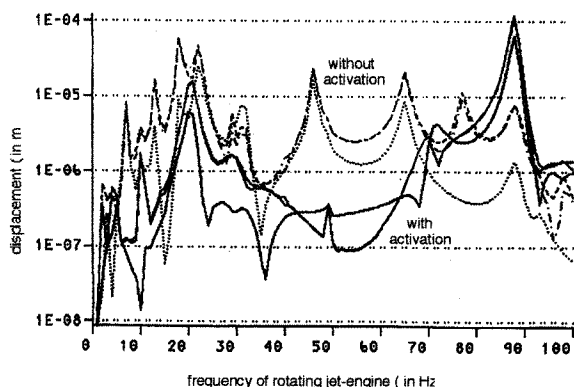


Figure 5: stationnary displacement of a point of the airfoil with and without frequencial deterministic control

On the figure (fig. 6), the level of control forces is presented. The dashed line figure intensity of the generative forces, an unbalance force of 1000 g.cm. The figure shows that for our model and in stationary domain, the level of forces is similar to the generative force.

Lastly, we show the major element needed by air-frame manufacturer, that is to say power required for actuators to optimally control vibrations. The figure (fig. 7) exhibits that only very low power is needed, about ten Watts, excepted near the gyroscopic frequency of 90 Hz, but of course that is not the point to control critical speeds.

### LQG control

We then computed time domain LQG algorithm (We chose the good weights for control to have of course realistic stresses of the same order of unbalance forces and powers of few Watts). Despite simulations don't take into account time delay,



Figure 6: stationary force of activation with frequencial deterministic control

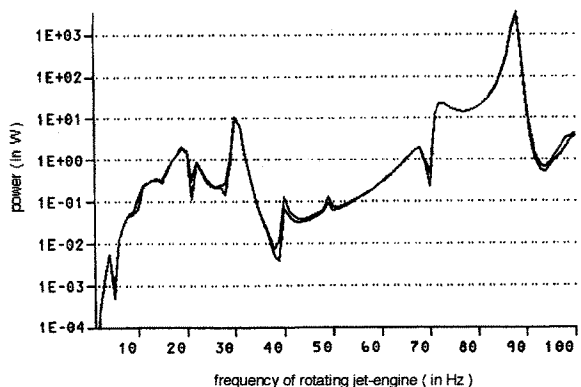


Figure 7: stationary power of activation with frequencial deterministic control

we can see that in time domain (fig8) we reach equally a decrease of 20 dB in vibration in 100 ms, with the same amount of forces and powers as before.

### Conclusions of feasibility study

Thanks to the two kinds of control we presented, with a perfect knowledge of the system and no time delay, we obtained a reducing of vibration of 20 dB, with forces of the same value of the disturbance force of unbalanced rotors, and with powers of about ten Watts.

The more we try to minimize the number of degrees of freedom, the less we succeed to reduce vibration, as we tried to add weights on shaft bearings.

It's impossible in this perfect theoretical feasibility study to distinguish one algorithm from the other, only experiment could.

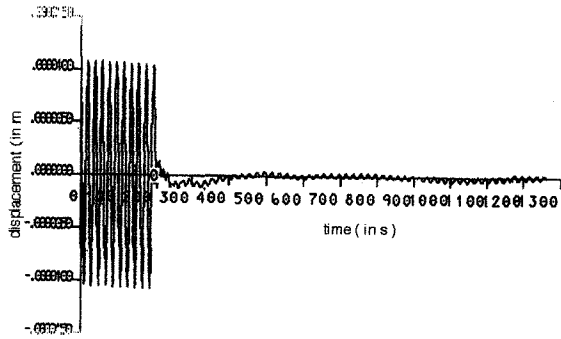


Figure 8: displacement of a point of the airfoil without and then with LQG stochastic control

Finally, the tests realised to optimise the placement of actuators concluded by : the nearer the actuators are from the jet-engine, the better are the results of the control but the more sensitive is the control to the gyroscopic effects.

## Experimental demonstrator

### Introduction

After a feasibility study thanks to simulations, we wanted to validate the active control concept on an experimental demonstrator. We wanted to realise a testing ground that was as near as possible as the structure studied for feasibility, with realistic size, masses and stiffness, and with industrial actuators.

One of the power supply easily available on a jet is the hydraulic supply delivering a pressure of 3000 psi, usual pressure for servovalves and jacks. So, we chose as actuator, obviously, servojacks.

We'll try now in the next section to explain the conceiving of our testing ground.

### Conceiving of testing ground

From our reference system presented in figure (fig. 2), we realised the model (fig.3). This model for a first testing ground is too complex. We first choose not to introduce gyroscopic phenomenon, with no turning rotor. The joining part between engine and airfoil will be considered only as a stiffness element, built with springs. The airfoil is

considered as extremely inflexible, engine is represented with 3 masses, the two extrem (fig. 9) ones as the bending parts of the engines.

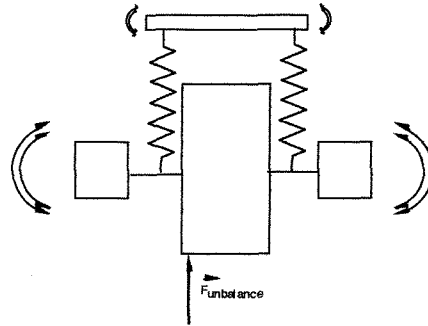


Figure 9: Reduced model

We reduced this element a last time. Bending part of engines are considered as beating elements, with rotational motion around horizontal axe. Finally, the unbalance force is created thanks to a vibrator. Unbalance creates vibration from the engine to the airfoil. But it is very difficult thanks to a vibrator to move a massive part as "jet-engine". We chose to excitate the lighter "airfoil" of our testing ground as presented in the figure (10).

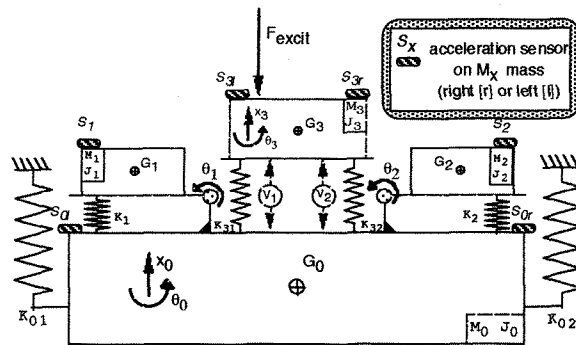


Figure 10: Experimental demonstrator

To reduce the number of d.o.f of our testing ground, we realised it symmetrically according to two vertical orthogonal planes, and the disturbance force was placed in the vertical lengthwise plane. Thanks to those assumptions, the testing ground is composed of 6 d.o.f. It was equipped of two hydraulic jacks ( $V_1$  and  $V_2$ ) and 6 acceleration sensors ( $S_{0l}, S_{0r}, S_1, S_2, S_{3l}, S_{3r}$ ) placed in the vertical lengthwise plane (10).

To realise a industrial ground bed with consistant efforts for actuators, we used the following masses ( $M_0 = 3225kg$ ,  $M_1 = 745kg$ ,  $M_2 = 745kg$ ,  $M_3 = 458kg$ ).

This leads to a demonstrator with the following vibration modes (fig. 11).

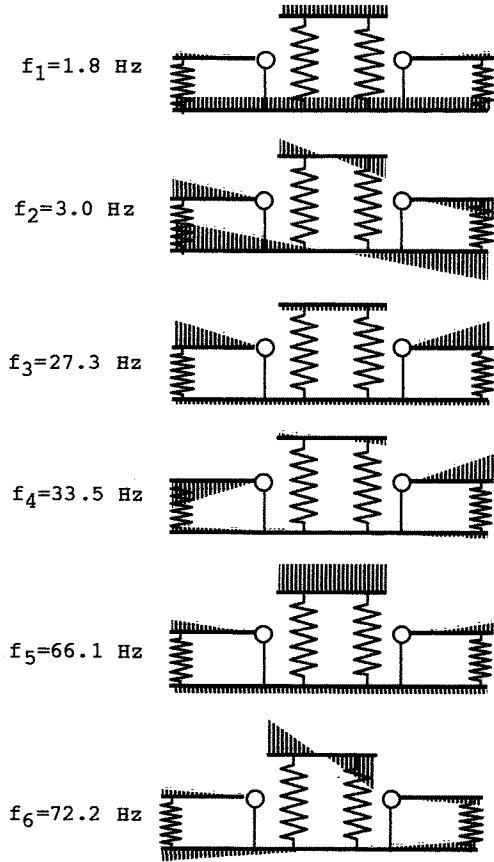


Figure 11: vibration modes

By integrating the 6 accelerations one and two times, we obtain then speeds and displacements at the place of the sensors, and we can easily thanks to the observer vector reconstruct the state vector for stochastic control. We need for this to succeed in discrete Integration. More over, we need to enslave hydraulic jacks, local enslavement at first, adaptive enslavement may be. Identification of jacks would be used.

### Actuators modeling

To understand the behaviour of servojacks and realize the identification of actuators, it is usefull to have a model of them. We won't propose here a conception model. We don't aim to conceive

an actuator but to understand it. So, we'll use a simplified model for analyze to described the first order working of servojacks.

We used a servovalve Moog D760-231 with two amplification stages. To describe the first stage of the servovalve, a second order model is usually taken between the control (tension or current) and the slide valve displacement ( $x_t$ ) as :

$$\frac{x_t}{i} = \frac{K_x}{1 + \frac{2\zeta_0}{\omega_{ns}}s + \frac{1}{\omega_{ns}^2}s^2} \quad (14)$$

Constructor data are given for such a model.

The second stage, constituted with a slide valve with four orifices, thanks to the Bernoulli flow equation, by supposing servovalve work near the balance point with low displacements, and with neglecting leak phenomenoms, we obtain :

$$Q_L \approx K_1 x_t \quad (15)$$

where  $Q_L$  is the flow delivered by the servovalve to the jack.

We used a usual double effect jack to realize actuation. Thanks to the flow equation, we obtain :

$$Q_L = S_p \cdot \frac{dx_v}{dt} + \frac{V_0}{2B} \cdot \frac{dP_L}{dt} \quad (16)$$

where  $S_p$  is the efficient piston surface,  $x_v$  the displacement between piston and chamber,  $V_0$  the dead volume of jack,  $B$  the bulk modulus of oil and  $P_L$  the difference of pression between the two chambers of jack. We have to add to those elements an other significant thing, the frition force between the jack piston and chamber. This is a dry friction, but as a simplification, we used viscus friction  $f$ . The whole model of servojack can then be represented thanks to the diagramm (fig. 12).

### Observers

Our testing ground, thanks to symetry of structure and position of excitation have only 6 degrees of freedom. So, thanks to 6 acceleration sensors, we could obtain the full state vector, after 1 and 2 integrations.

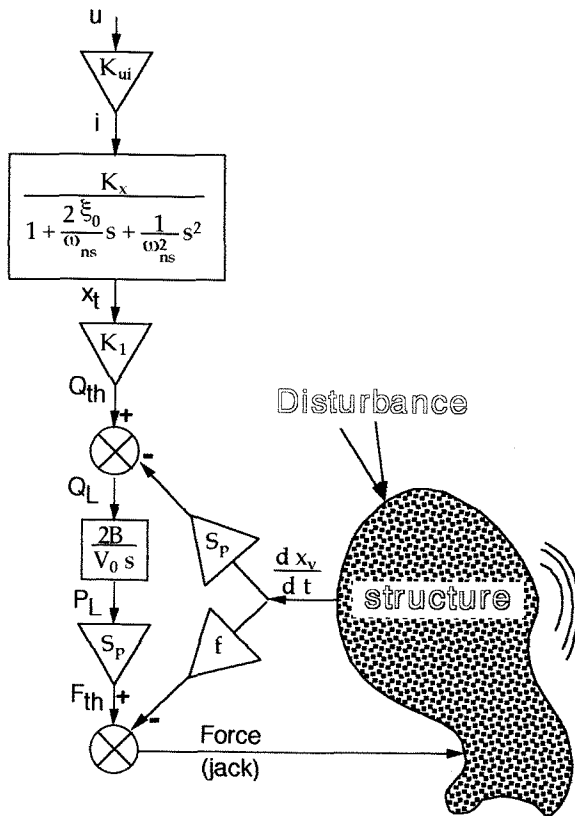


Figure 12: Model of servojack

### Integration

To obtain speeds and displacements from acceleration sensors, we first used trapezoid integration. But we had a data accumulation problem, and secondly the initial condition. To solve the problem, we used the LMS by-pass filtering [7] whose transfer function is given by :

$$H(z) = \frac{z - 1}{z - (1 - 2\mu)} \quad (17)$$

with  $\mu \ll 1$ , and the problem was over. But with our computer means, we only had 4 inputs and 2 outputs. So, thanks to simulations, we chose 4 of the 6 sensors. It could be interesting to use observers, deterministic or stochastic, to obtain the 6 d.o.f. with the 4 sensors, as we can see on the diagram (fig. 13).

The  $K_E$  matrix can be obtained for example by kalman filtering thanks to noise correlations.

But with our simulations, we saw that we could hope to succeed in controlling the structure with no observers, if we took into account the 4 sensors

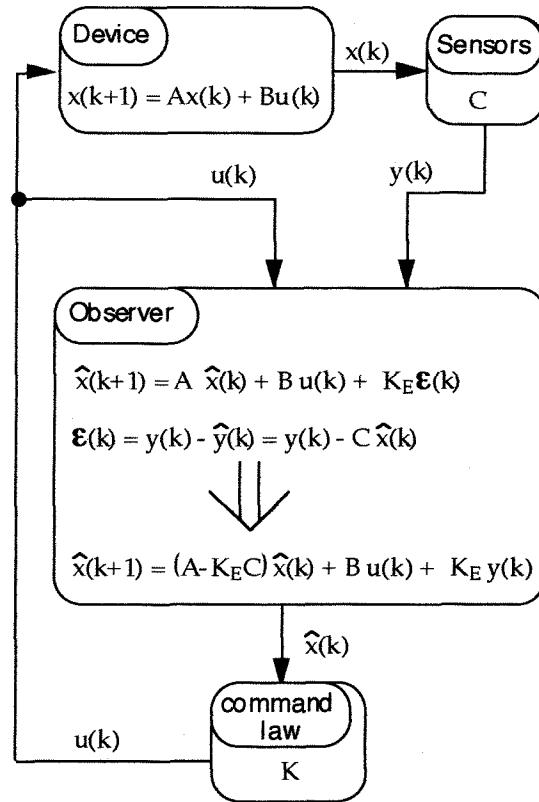


Figure 13: Observer

the farthest of the excitation point. As the determination of observers is not always easy, we chose this option, the two unmeasured d.o.f. were taken to 0.

### Choice of control algorithm

As presented before, we could use 2 control algorithms, a frequential and a temporal one. To use the frequential one, we have to estimate on line the  $q_0$  vector, that is to say the state of the structure would have with no control. It seemed to be difficult to obtain this easily for an aeroplane, and more over, it seemed easier to obtain a representation model of rotor. So, to begin for our testing ground, we chose the LQG algorithm.

### Results

To experimentally succeed in control, we had to study the ground bed thanks to many simulations, with models as close as possible of reality. In a first case, we studied control with ideal actuators only to observe what it was possible to do. It appeared that it was impossible to control the first

two modes. Those two modes are uncontrollable indeed, because of no move between elements. So, we could only try to control the 2 modes around 30 Hz and the two around 70 Hz. As the nominal rotating speed of low pressure rotor is near 70 Hz, we should try to control the 5 or 6 mode. But, as a beginning we preferred to try to control the 3 and 4<sup>th</sup> mode not to mix all the problems, calculation speed, time delay, phase delay for actuators and so one.

For the tests realised at 35 Hz, we were very close of the phase delay limit of 60°. Excitation realised with vibrator was about 500 N, at 35 hz, near the 4<sup>th</sup> mode in an asymmetrical position. The sampling frequency was of 1000 hz. The hydraulic pressure was about 180 bar, upper possible pressure.

The best result was obtained on the sensor 2, at the opposite side of excitation. The registered temporal signal on sensor 2 is showed on the figure (fig. 14).

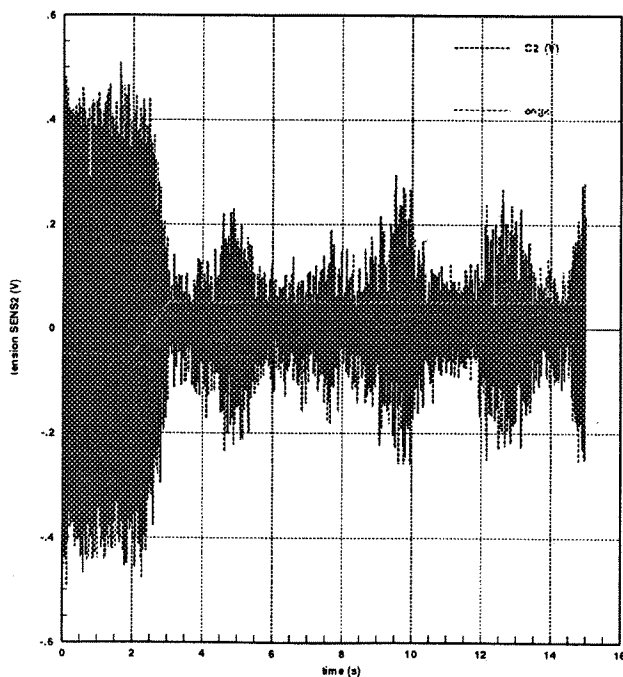


Figure 14: Experimental control recording on  $S_2$

On this test, the control signal is progressively engaged, between the second second and the third second.

We obtain visually that the signal, after control is

divided by more than 5. To characterize the result, we measured the RMS voltage before and during control of sensor 2. We observed a decrease of acceleration of 11 dB. An other interesting component of the signal is the frequencial one. We can see on the figure (fig. 15) the FFT of signal on sensor 2 on 512 spots ( $\approx 1$  s) with use of the Hanning window.

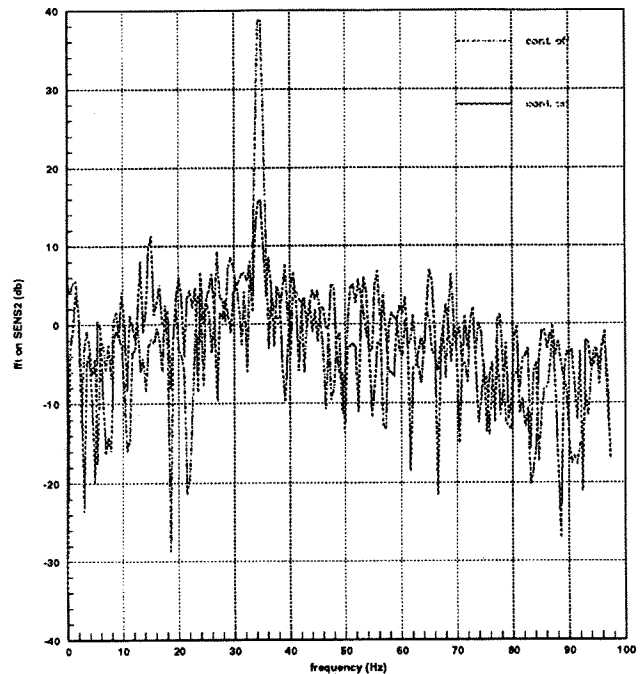


Figure 15: FFT on experimental signal on  $S_2$  with/without control

The decrease of vibration on the excitation peak is between 15 and 25 dB, depending of the recorded part chosen.

Despite these interesting results in this case on sensor 2, we observe no improvements on sensor 1. By exciting the structure perfectly symetrically, at the frequency of 28 Hz, at the third mode, we then obtained symetric results. In the first test, excitation was near the left jack, and the perturbation was so great that the failure of flow that the left jack couldn't control anything. Simulation showed that increase of stiffness between the two part of jacks would improve results, by reducing relative displacements. Servojacks are frequentially good force actuators but bad displacement actuators.



A last test was realised with a symmetrical excitation at 30 HZ, between the third and the fourth mode. We then obtain on sensors a signal rich on all the frequencies. So, the signal is reduced on all the frequencies, not especially on the excitation frequency because of the non frequential performance index.

## Conclusions

For our structure<sup>[5]</sup>, we reach feasibility limit around 40 Hz, because of actuators. Their phase delay is  $60^\circ$  at these frequency, before optimizing structure, especially with reducing relative displacements between piston and chamber of jack by increasing stiffness.

Despite these conditions, we reached a decrease of more than 11 dB RMS or 15 dB on excitation frequency at 35 Hz for vibration on 2 sensors.

We neglected frequential control because the identification of the structure sound difficult. But this would be a good mean to introduce actuators delay in the control loop.

To succeed in controlling aeroplanes motors, airframe designer and motor design will have to work together. The aim, reducing vibrations can only be reached thanks to mechanic, automatic, signal filtering, identification. The aim impose the use of so many technologies that organisation of such a plan will have to make the different people work together, thanks to Concurrent Conceptual Design. The organisation must change to succeed.

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