MAGNETIC MATERIALS FOR ACTIVE CONTROL OF ANTI-VIBRATION SYSTEMS

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MAGNETIC *MATERIALS FOR ACTIVE CONTROL OF ANTI-VIBRATION SYSTEMS*

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ABSTRACT

The need for multi-frequency, multi-point vibration control for heavy structures leads to specific requirements for the actuation and control system. Development of efficient actuators plays an essential role in reaching these targets. The critical requirements are large Force and Strain, Low voltage (for safety], broadband frequency capability, Low weight, small size, high temperature capability and high reliability.

In the frame of this project, a low weight (< 1,5 kg) actuator was designed and built using magnetostrictive material, Terfenol-D. Results have shown that high forces (> 800" N rms), can be generated on a wide frequency range (200-2000 Hz) with low voltage (< 100 V) and power requirements, at temperature from 20"C to 80°C.

Based on these values an electronic controller and a digital control algorithm was designed and developed. With this optimal controller, an efficient darnping of a platform structure vibrations was obtained. Vibration cancellation tests on a real structure using a pure feedback LUG algorithm have demonstrated the capability to achieve significant damping effect up to 20 dB.

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1 - INTRODUCTION

The overall objective of the project was to demonstrate the high potential of magnetostrictive materials for a wide range of industrial applications.

The specific objective of the project was to design and optimise a high energy active system and to demonstrate that a significant reductions (expected 10-15 dB) of vibration levels can be obtained over a wide frequency bandwidth (from 200 Hz to 2 kHz).
The objectives can be defined in more detail as:

1 - To design and manufacture a high energy magnetostrictive actuator and an experimental set-up capable of demonstrating the potential of the actuator for active control of anti-vibration system.
2 - To design and develop the control and feedback system required to achieve real time active control.
3 - To evaluate the actuator performances in active vibration isolation with improved control hardware and algorithms through vibration cancellation tests.

2- ACTUATOR REQUIREMENTS

The selection of the actuator active material is driven by the requirements for actuation forces in pratical structures.

The often critical requirements are large force and strain, low voltage (for safety), broadband frequency capability, small size, low weight, high temperature capability and high reliability. For industrial applications low cost actuator will be also required. The alternatives for an actuator are the solid state extenting materials, Lead Zirconate Titanate (PZT), Lead Magnesium Niobate (PMM) and Terfenol.

Alternatives such as electrodynamic and hydraulic actuators are excluded because of low force to weight ratios in the case of the former and limited life due to mechanical wear in the case of the latter. Capabilities of the candidate materials are summarized in Table 1. While all three candidate materials have good potential as actuator materials in general, Terfenol-D was chosen for the required low voltage and his high energy to weight ratio.

CRITERIA	REQUIREMENT	PZT	PMN	TERFENOL-D
FORCE	1000 N	Qualifies	Qualifies	Qualifies
STRAIN	> 1000 ppm	No qualifies	No qualifies	Qualifies
BROADBAND	2'30-2003 Hz	Qualifies	Qualifies	Qualifies
LOW VOLTAGE	< 100 V	No qualifies	No qualifies	Qualifies

TABLE 1 - CANDIDATE ACTUATOR MATERIALS

3- TERFENOL-D PROPERTIES

Terfenol-D is a rare-earth alloy composed of Terbium (Tb), Dysprosium (Dy) and Iron (Fe), and is a highly magnetostrictive (producing a mechanical strain when applying a magnetic field) alloy that was found to be very accurate for use in many transducer applications [1] [2].

This new rare-earth magnetostrictive alloy presents magneto-mechanical characteristics which allows significant innovative approach for vibration control system. Their high strain (> 1000 micro-strains) and high force associated with a low Young's Modulus, allow the design of devices with interesting performances in the low frequency regime. Compared to piezo-electric ceramics, Terfenol-D generates substantially high strain and the possibility to store 5-10 times of energy per volume unit.
Characteristics of Terfenol-D are shown in table 2. These characteristics are compared with classic magnetostrictive material like Nickel, type III PZT piezo-electric ceramic and PMN-X electrostrictive material,

	TERFENOL-D	NICKEL	PZT III	PMN-X
ε MAX. MAGNETOSTRICTION (ppm)	1000	50	250	750
y: YOUNG'S MODULUS (GPa)	35	35	70	70
ENERGY DENSITY $\varepsilon^2 y/2$ (J/M^3)	17500	50	2200	19500

TABLE 2- ENERGY **DENSITY COMPARISON OF ACTUATOR DRIVER MATERIALS**

Terfenol-D is born in the United States [3] [4] about twenty years ago. Sophisticated facilities are now available in Europe also, for processing rods up to a diameter of 30 mm (shown Figure 1) and for property measurement and characterisation.

FIGURE 1 - "REACTON" TERFENOL-D RODS DEVELOPED BY JOHNSON MATTEY IN UK

Two pre-conditions are necessary for the use of Terfenol-D in actuators. First, as shown Figure 2a, because the applied [positive or negative] magnetic field always causes expansion in the direction of excitation (a square low material), a magnetic bias is required for the actuators to follow a sinusoidal input signal (see Figure 2b).

Second, as shown figure 3, the magnetostrictive strains developed in a given magnetic field also depend on the current state of stress. Typically 50 to 500 bars Impression prestresses are required. For each mechanical prestress level, there is a magnetic bias point where optimum amplitude with minimal harmonic distorsion occurs. The prestress requirement can be accommodated through an independent prestress system which is either part of the actuator or integrated into the structure to be controlled.

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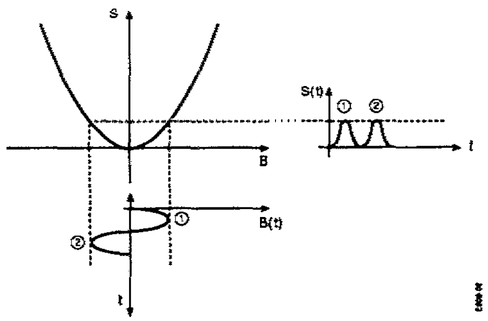


FIGURE 2a - WITHOUT MAGNETIC BIAS BEHAVIOUR

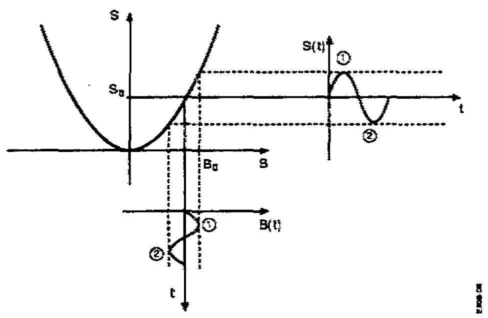


FIGURE 2b - WITH MAGNETIC BIAS BEHAVIOUR

Where:

- B is the magnetization,
- S is the strain,
- (S_0, B_0) is the static working point.

FIGURE 2- MAGNETOSTRICTIVE STRAIN S(t) AS A FUNCTION OF THE MAGNETIC FIELD DRIVE B(t) WITHOUT AND WITH MAGNETIC BIAS B_0

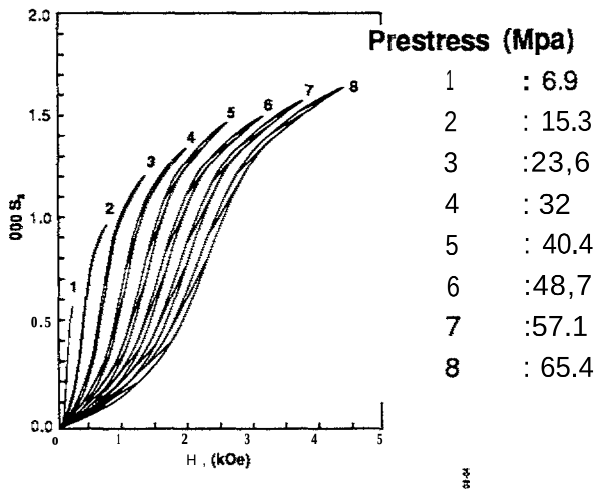


FIGURE 3 - THE STRAIN AS A FUNCTION OF THE MAGNETIZING FIELD FOR DIFFERENT MECHANICAL PI? ESTRESSES [5]

4 - STUDY, MODELLING, DESIGN AND REALISATION OF AN ACTUATOR

Figure 4 shows a cross sectional view of an abbreviated magnetostrictive actuator.

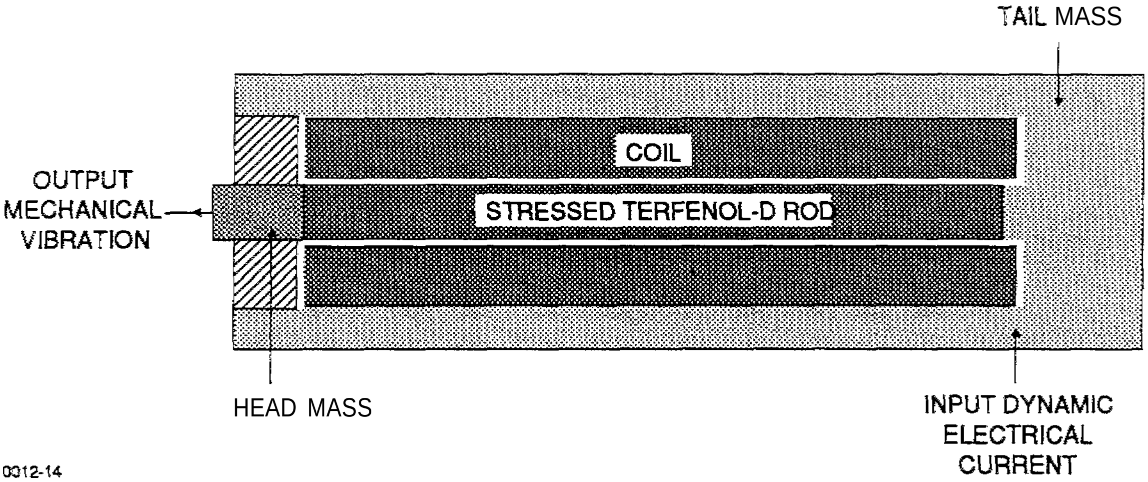


FIGURE 4- CROSS-SECTIONAL VIEW **OF** A **MAGNETOSTRICTIVE** ACTUATOR

The first activities performed by the consortium was:

- to characterise properties of different Terfenol-D,
- to choose the stoichiometry of the material,
- to define the electrical and mechanical parameters to get the optimal behaviour of a magnetostrictive actuator,
- to develop a software for a fast unidirectional design and a finite element magnetostrictive code taking the three dimensional actuator behaviour into account,
- to define the dimensions of the Terfenol-D rod allowing to achieve the required performance,
- to design and optimize a high force actuator.

The main component of this actuator is based on a 100 mm long type 30/70 "REACTON" Terfenol-D (Tb(0,3) Dy(0,7) Fe(1 ,95)) rod from JOHNSON MATTHEY driven by a coil for the generation of an alternative magnetic field.

Approximately, a prestress of 70 bars and a static magnetic field of 60 kA/m are required to achieve a quasi static magnetostriction > 1000 ppm between 20°C and 80°C.

Using AC drive signal (oscillatory application), the bias point is set to the center of the linear portion of the strain versus field curve to maximize bidirectional (push-pull) response. With properly biased and preloaded Terfenol-D rod, there wiii be a push (on the position half of the cycle) and pull (on the other half to the cycle).

The rod diameter is determined by the balance between the effects of eddy current and the required forces.
For a working frequency range [200, 2000 Hz], a 8 mm diameter rod is a good compromise.

In order to optimize the design of this actuator, a numerical code using the variational principle has been derived, following a classical Finite Element Method approach. This model describes the three-dimensional dynamic behaviour of heterogeneous electromechanically coupled structures in air. Using a reduced scalar potential formulation of 'the magnetic field, the model has been implemented within the THOMSON SINTRA ASM three dimensional Finite Element Code "ETTRI".

To develop this model, the following assumptions have been retained.

- The analysis is conducted in the steady-state harmonic cases. The frequencies are low enough to check the magnetostatic hypothesis on magnetic quantities.
- The eddy currents are not taken into account.
- The current intensity in magnetization coil is constant.
- The material behaviour law is linear. The magnetic non-linearity behaviour is not taken into account.
- There are no losses by eddy currents, hysteresis and in magnetization coil.

The assumptions seem quite restrictive. Nevertheless enable to efficiently analyse very low-frequency actuators in their usual environment.
Modelling of the magnetostrictive actuator is shown in Figure 5.

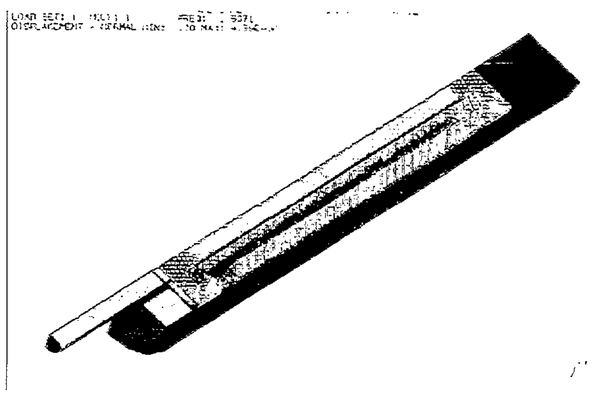


FIGURE 5 - MESH OF THE MAGNETOSTRICTIVE ACTUATOR

Based on modelisations results, an actuator was designed (see Figure 6) and built (see Figure 7} using the magnetostrictive material.

For an optimized coupling factor, the value of the prestress system stiffness should be kept low to avoid clamping of the driving element. The mechanical variable system consists in washers "Belleville". This technology permits to control the value of the prestress in an easy way. Therefore, change in prestress can be obtained by varying the angular position of an hexagonal screw.

The Terfenol-D rod can be magnetically biased by permanent magnets or by a coil excited in DC mode. Using coil gives case for variation of the magnetic bias, but a DC electrical power supply is required to generate DC current.

In general, the use of permanent magnets is more convenient and different solutions are possible:

- bias is supplied by cylindrical permanent magnets, which are also used as flux return path,
- the magnet can be placed at the ends of the rods,
- a stack of alterning magnet with Terfenol-D driving elements can be built.

In the built actuator, a cylindrical ALNICO V magnet provides the necessary magnetic bias {Ho = 30 kA/m), and serves as the return path for the magnetic flux.

The drive magnetic field concentrated in the Terfenol-D is generated by a solenoid coil excited by AC current.
This system is placed inside the cylindrical permanent magnet. It was designed to generate a 30 kA/m peak drive field maximizing the efficiency of the actuator.

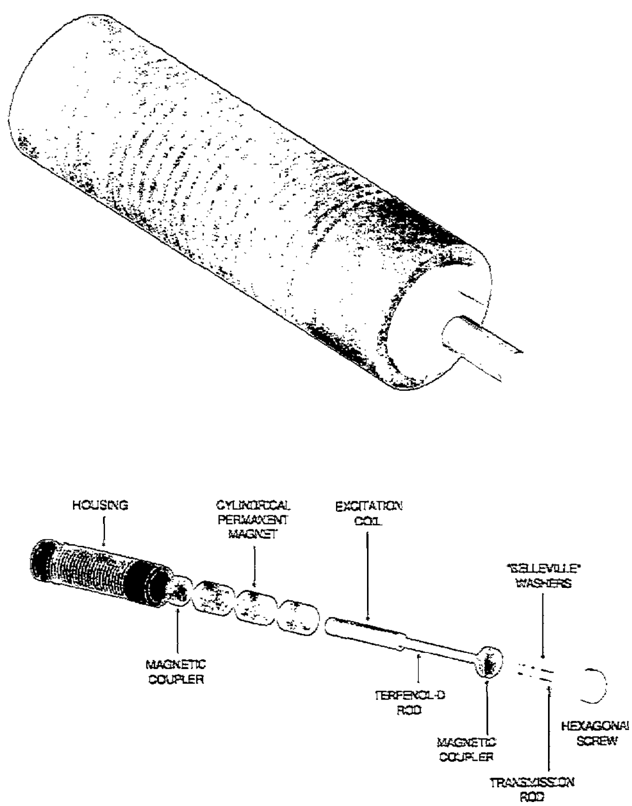


FIGURE 6 - SCHEMATIC VIEW **OF** THE MANUFACTURED ACTUATOR

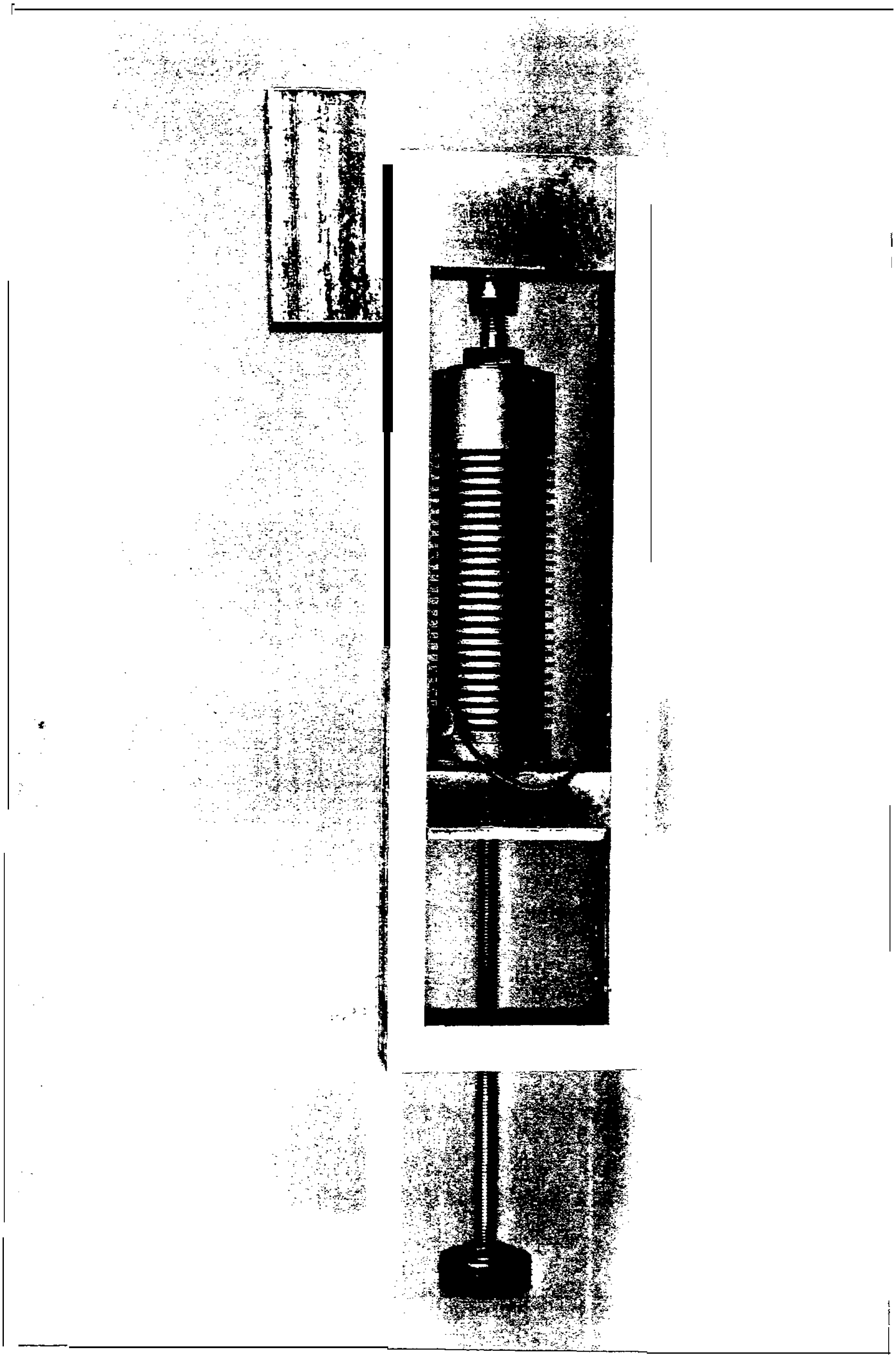


FIGURE 7 - MANUFACTURED ACTUATOR (IN BLUE) MOUNTED IN A TEST MECHANICAL STRUCTURE

So, a 1000 ppm peak-to-peak dynamic strain is achieved with the applied dynamic field of approximatively 30 kA/m peak around the magnetic bias Ho = 35 kA/m (see Figure 8).

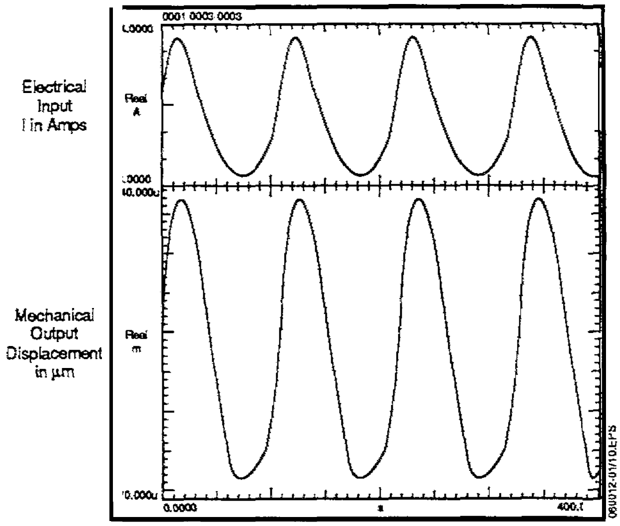


FIGURE 8 - OUTPUT DISPLACEMENT SIGNAL GENERATED BY THE TESTED ACTUATOR - F = 10 Hz, To = 70 bars

Dimensions of the coils are fixed by the rod length and the value of the field required.

Due to magnets and the coil arrangement, the radial size is around 50 mm diameter.

The design of the magnetic circuit for dynamic excitation field requires special attention to eddy currents, they have a determining effect on effective coupling and efficiency of the actuator.
Terfenol-D presents low relative permeability, which is responsible for fringing of the flux lines at the rod ends.
The magnetic circuit should therefore include magnetic couplers between the rod ends and the cylindrical permanent magnet also provides flux feedback system.
In order to ensure a homogeneous and efficient magnetization of the rod, high magnetic permeability coupler material is used.

The couplers must also withstand large mechanical loads and transmit high forces. Its stiffness must be larger than the stiffness of the Terfenol-D rod. In the actuator, couplers are built in soft iron.

In addition, a low weight of 1,5 kg was achieved because an inertial mass is not required with the chosen structure to control.

The actuator packaging was designed in order to get a low weight because inertial mass is already assured by the structure itself.

5- ACTUATOR PERFORMANCE TESTING

Testing of the actuator on a real structure was the first priority of the vibration control study. We sought to experimentally determine the optimal prestress level, the measured displacement (linearity) for small and large signal, the frequency response and the electrical impedance.

5.1- Test mechanical structure and prestress optimization

The team designed and built an experimental set-up, allowing the preliminary evaluation of the actuator.

The mechanical arrangement for testing the actuator and the control system is shown figure 9.

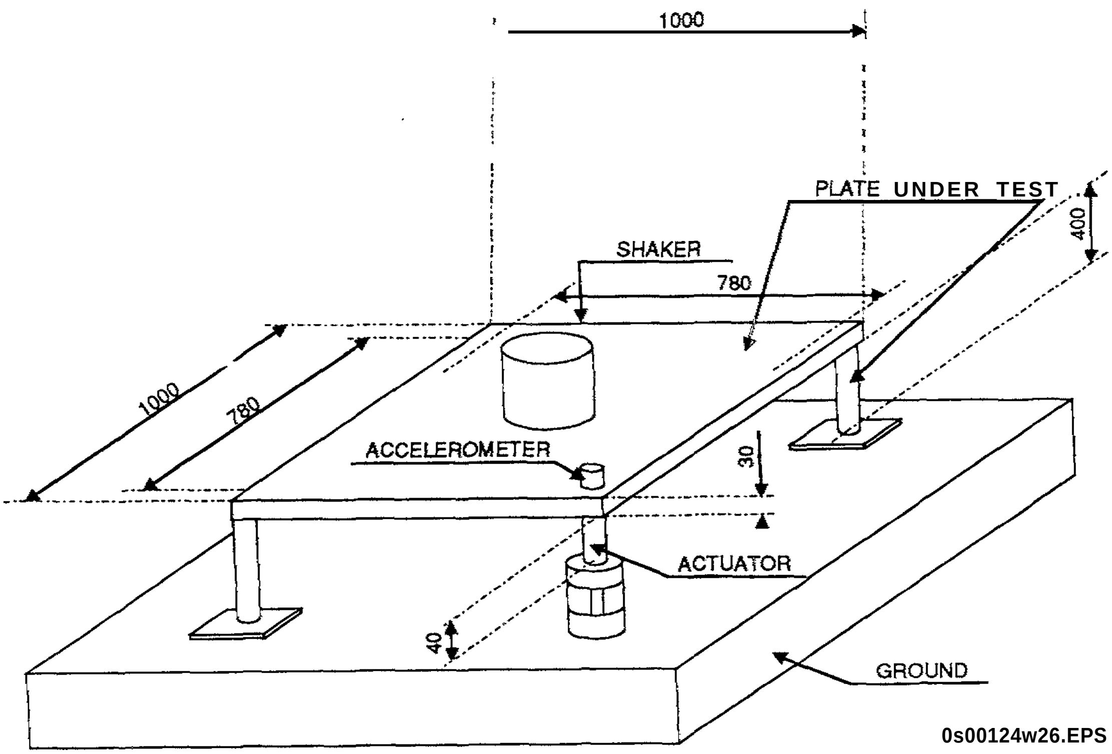


FIGURE 9- MECHANICAL ARRANGEMENT FOR TESTING THE ACTUATOR

The structure used for this investigation is composed by a steel desk with three cylindrical legs. At one end of the plate we place the magnetostrictive actuator and at the center we place a shaker.

The desired level of prestress within the Terfenol-D rod in the actuator can be adjusted with the mechanical variable system placed between the actuator and the bottom table.

The structure is excited with a vibration shaker Bruel and Kjaer type 4809 using a sine/random generator BK type 1027 and a power amplifier BK type 2706.

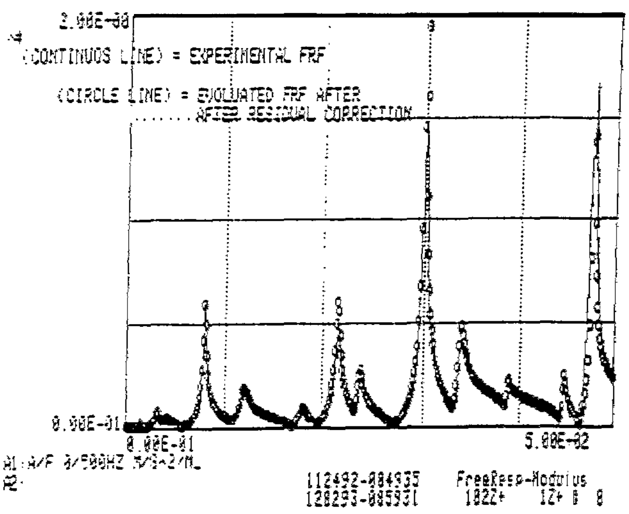
The set-up also includes electrical driving units for magnetostrictive actuator and sensors measuring the structure vibrations.

The structure was designed with a mathematical model in order to generate vibrations in the frequency range from 200 Hz to 2000 Hz.

This Finite Element modelling of the experimental device was realized and takes the plates, the shaker, the sensor and the actuator into account.

The first experimental modal analysis permitted to fii data to give a representative model.

The calculed Finite Element Model data obtained show satisfactory agreement with measured values (Figure 10).



From 0 to 500 Hz

FIGURE 10 - RESPONSE FUNCTION ACQUIRED IN 1.Z LOCATION
Comparison between theoretical and experimental results
(continuos line = experimental datas,
circle line = theoretical datas after residual correction)

5.2- Actuator performance and simple vibration cancellation test

Some measurements were performed in order to:

- investigate the actuator behaviour when mounted on the real set-up,
- define vibratory and electrical characteristics of the designed actuator.

The actuator performance was tested in the frequency range 200-2000 Hz with large and low input signal level. The flat frequency response {figure 11} permit a broad frequency band for control vibration.

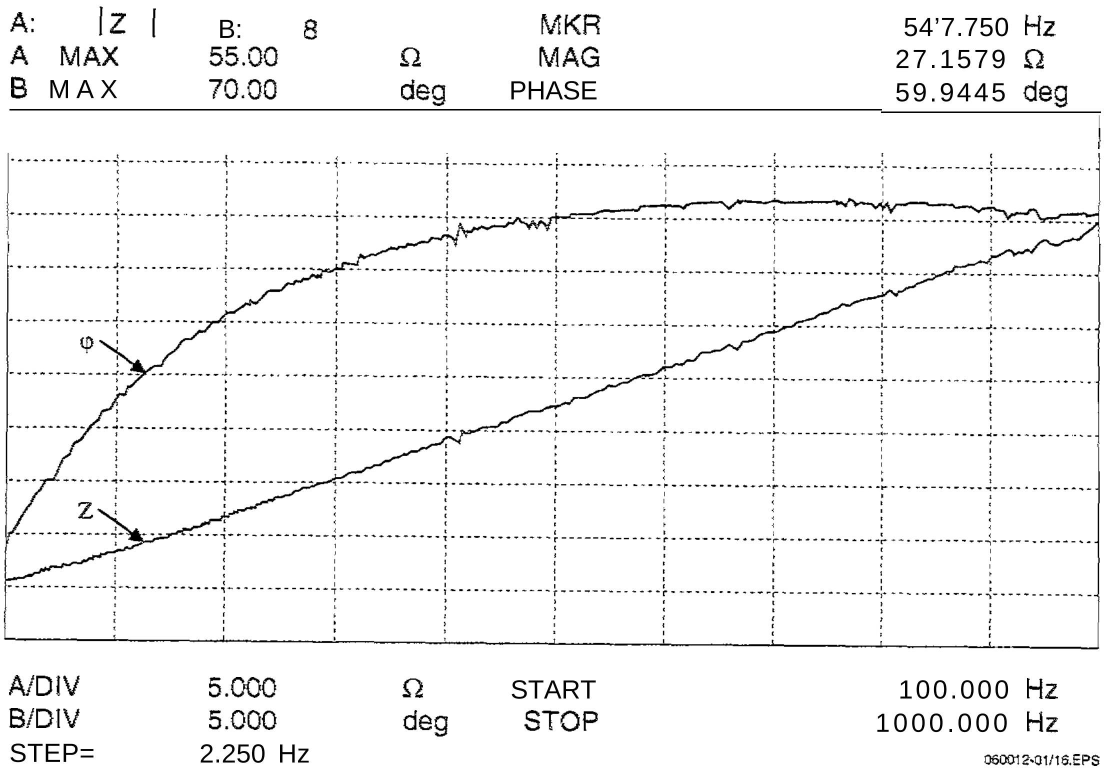


FIGURE 11 - ELECTRICAL IMPEDANCE OF THE ACTUATOR WHEN MOUNTED IN THE REAL FACILITY

Inherent advantages include the capability to generate large vibrations and high forces with low voltage and power requirements.

A O-peak force output of greater than 1000 N per actuator was achieved.
These experimentations allow to define the limits of the actuator. Applying high signal, measurements of magnetostrictive transducer show distinct non linearities (output distortions] on his extension-current characteristics as well as evident losses at high magnetic field strengthes.

A simple test (without control unit) consisted to study the feasibility of vibration cancellation, to determine the required conditions and also to verify how the actuator behaves when fixed into structure to control.
The vibration characteristics of the structure used for the experiment were initially identified using a shaker driven by a pseudo random excitation when actuator is not connected.
The resonant frequencies and damping ratios were measured in the 10 Hz to 2000 Hz frequency range (see figure 12}.

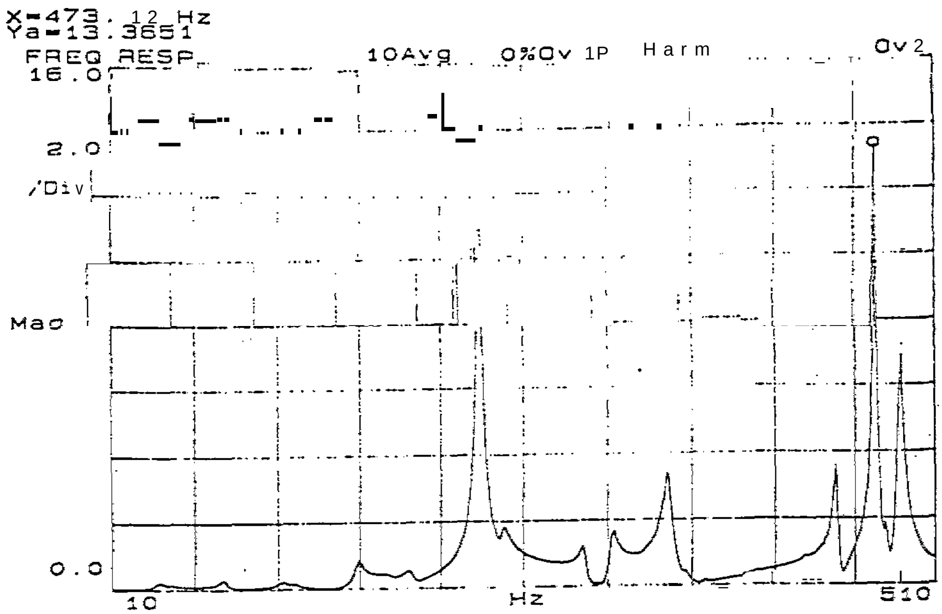


FIGURE 12- THE RESONNANT FREQUENCIES AND DAMPING RATIOS OF THE STRUCTURE MEASURED IN THE 10 Hz TO 500 Hz FREQUENCY RANGE

In the studied frequency range, two modes are dominating below 500 Hz (F_1 = 235 Hz, F_2 = 470 Hz), and above 500 Hz, there are several modes concentrated between 800 Hz and 1500 Hz.
The test consists to apply the superposition principle.
If $a_1(t)$ (respectively $a_2(t)$) is the temporal response of the accelerometer when only the shaker (respectively the actuator) is driven, the total response of the accelerometer a(t), when the shaker and the control actuator are simultaneously driven, is approximately equal to $a(t) = al(t) + a_2(t)$, where $a_1(t) = A_1 \sin(\omega_1 t + \phi 1)$, $a_2(t) = A_2 \sin(\omega_2 t + \phi_2)$.

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So, a vibration cancellation can be obtained in using the appropriate conditions ($\omega_1 = \omega_2$, $A = A_2$, $\phi_2 - \phi_1 = \pi$) as shown figure 13.

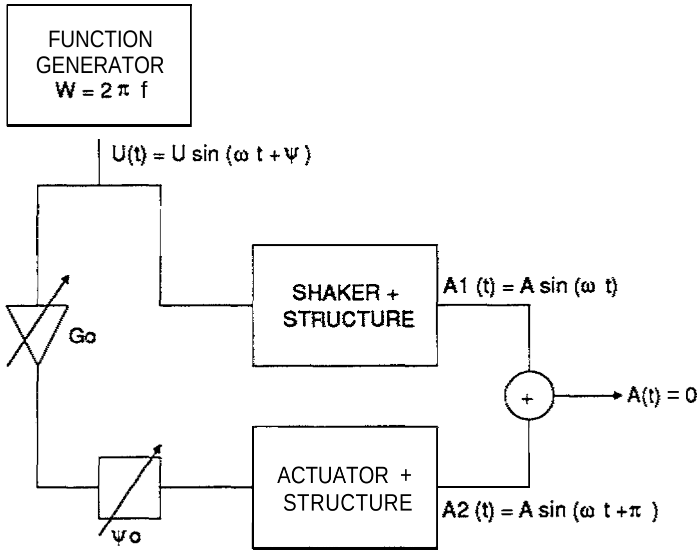

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FIGURE **13 -** APPROPRIATE CONDITIONS FOR **OBTAINING** A **BIBRATION** CANCELLATION

Using this device, structure noise level reduction from 15 to 30 decibels was obtained (see figure 14).

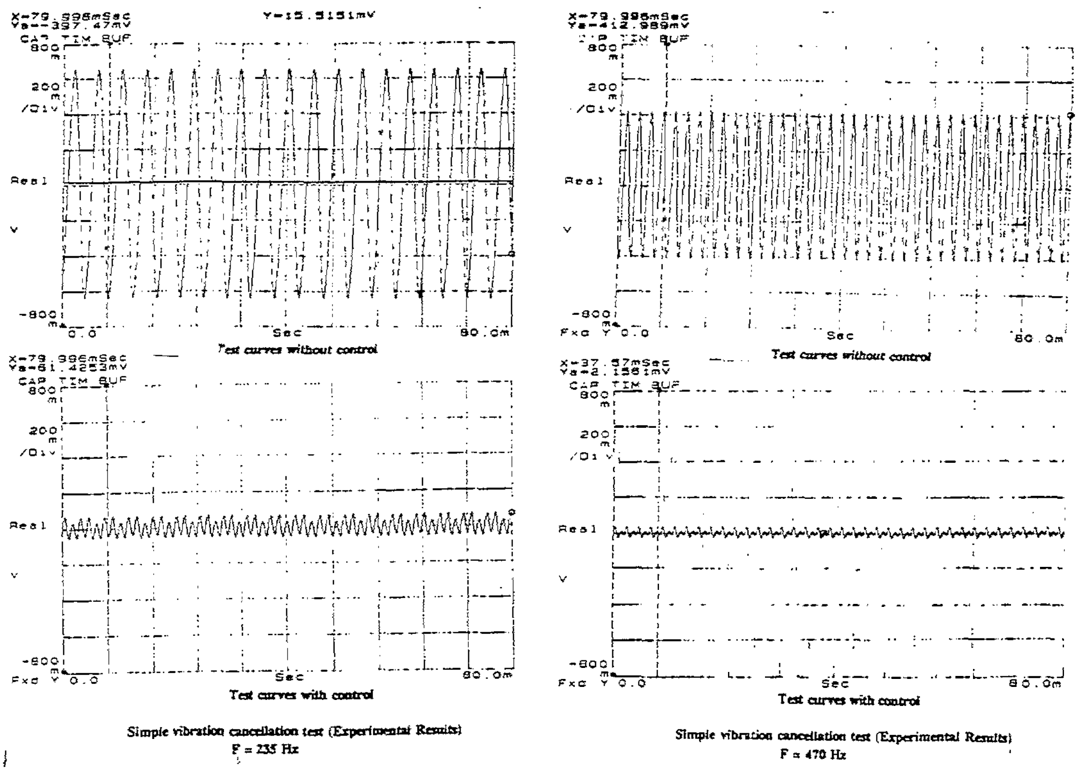

FIGURE 14- SIMPLE VIBRATION CANCELLATION TEST
(Experimental results without control unit)

A low weight actuator was designed and built using the magnetostrictive material, Terfenol-D. Results have shown that it can generate high forces on a broad frequency range (200-2000 Hz), with low voltage and power requirements. The first vibration cancellation simple tests (without control system) on the real structure demonstrated the feasibility of structure noise reduction from 15 to 30 decibels. These experimentations shown the interest of the magnetostrictive actuator in active vibration control problems. Vibratory and electrical characteristics of the designed actuator constituted the inputs of the control system.

Based on these values a digital control algorithm can be realized.
With an optimal controller, an efficient damping of the structure vibrations could be raised.

6 - VIBRATION CANCELLATION TEST

6.1- Tested algorithms description

Several algorithms for the implementation of an active anti-vibration controller are possible to use. Data provided by the simple tests suggest that the response of the actuator can be no linear with high signals.

A procedure of response linearization can be performed around a particular operating point, that changes with conditions. Variations of actuator parameters have to be considered and treated by the active control system.

The active control system using magnetostrictive actuators must take this situation into consideration and tolerance changes of the actuator response.

The controller must have the ability to Continually monitor input and output information.

Two blocks have to be present in the control system:

- a parameter estimator in order to adapt actuator response to change in the operating conditions,
- a controller' design block for the synthesis of the optimum control algorithms to damp unwanted vibrations.

After a brief overview of works found *in* literature on the subject. It was decided to develop an optimal Linear Quadratic Gaussian (LQG) control algorithm.

Technological and physical reasons motivate the development of such a control strategy which is shown to be the solution of an optimization problem.

For comparison, two types of control strategy were tested:

- an control strategy using the LQG (pure feedback) control algorithm,
- an control strategy using the filtered-X LMS adaptative feed forward control algorithm.

This second control algorithm was tested because the pure feedback controller has to be redesigned when the set-up changes.

6.2 - Control tests using LQG algorithm

The algorithm utilized is the LQG one [6]. This algorithm was developed in C-code and implemented in machine code, including the management of the I/O signals.
A PC program permits the possibility of choose different actions in order to test the system and run the control program.

The following figure shows the test platform structure where positions of the shaker, the magnetostrictive actuator and the error sensor are indicated.

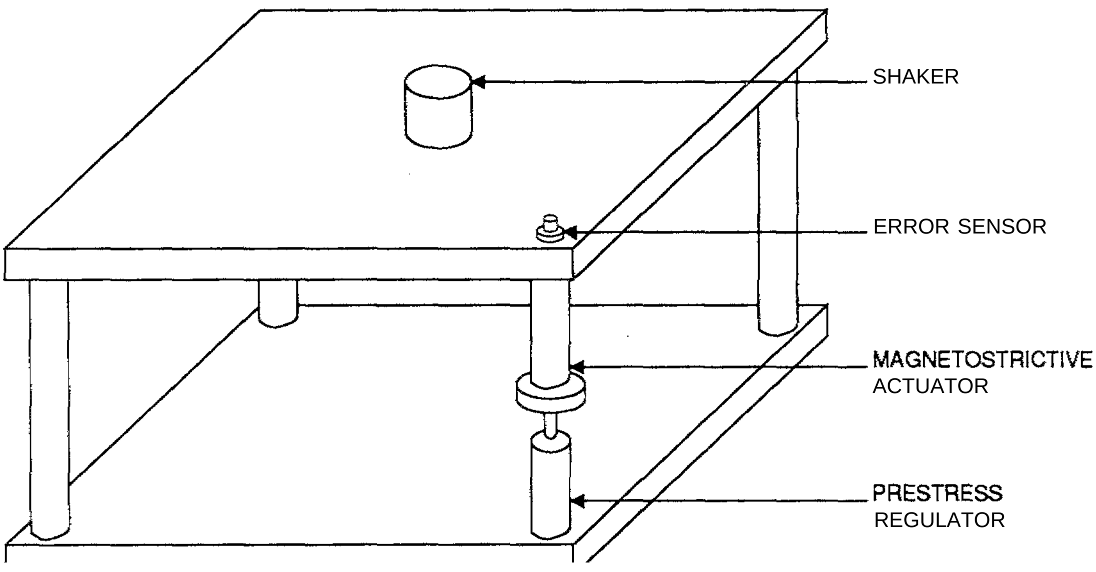

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FIGURE 15- SET-UP CONFIGURATION

An accelerometer was used as the error sensor. The algorithm aims to reduce the vibration locally in the place of the error sensor.

The control boards hosted in the PC are a DSP board with one TI C40 Processor Module installed on board and a I/O board with 16 bit AD and DA converters.

The control system set-up also contains:

- 2 power amplifiers in order to drive the shaker and the actuator,
- 1 power supply and amplifier for the accelerometer.

The following figures report the vibration attenuation achieved at the error sensor, exciting the structure with the shaker driven by random signal in two different frequency ranges. Some LQG controllers were designed in order to compensate the most important resonance peaks of the structure. They were tested and the best achieved results are reported in the following figures 16 and 17.

Vibration cancellation tests on the real structure using the pure feedback LQG algorithm have demonstrated the capability to achieve a significant damping effect up to 20 dB reduction for the more significant frequency peaks.
However we can notice that the vibrating structure generates a noise due to the high frequency component which are not filtered when the system is under control.

Further studies could be made in order to reduce this kind of noise

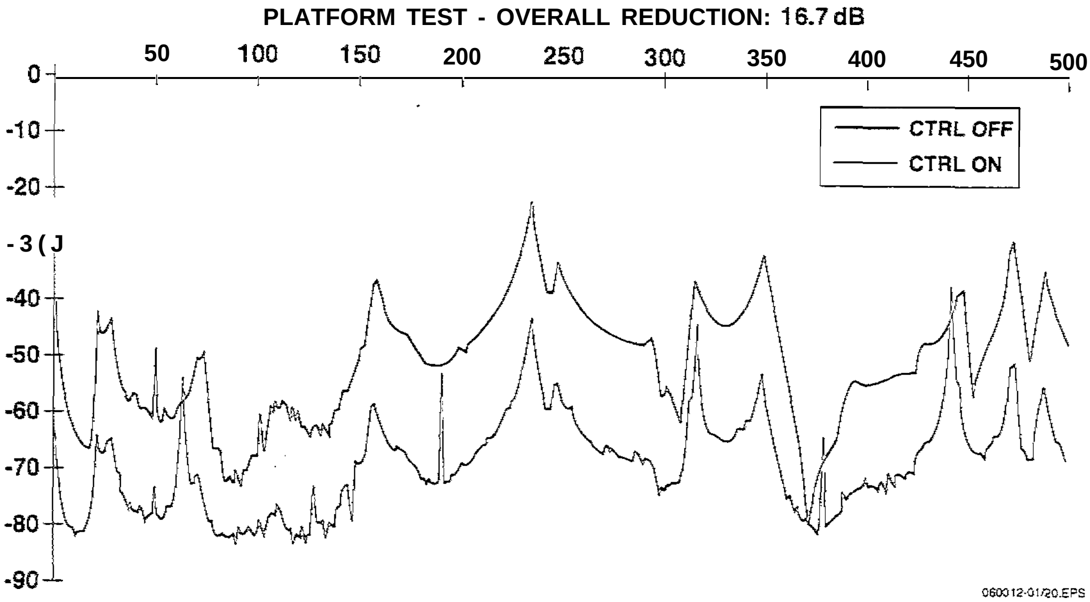


FIGURE 16- VIBRATION CANCELLATION TEST ON PLATE STRUCTURE USING A PURE FEEDBACK LQG ALGORITHM [Frequency range: from 0 to 500 Hz)

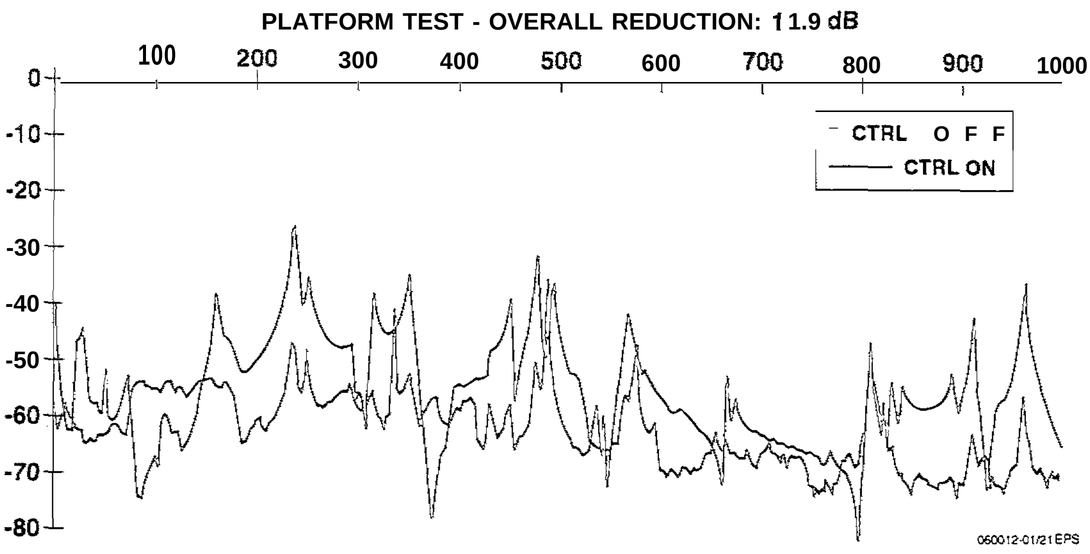


FIGURE 17 - VIBRATION CANCELLATIOPJ TEST ON THE PLATE STRUCTURE USING A PURE FEEDBACK LQG ALGORITHM (Frequency range: from 0 to 1000 Hz)

6.3 - Control test using an adaptive algorithm

For comparison, the consortium performed an adaptative feed forward control algorithm. The algorithm that was tested on the platform is the Filtered-X LSM adaptive feedforward control algorithm. This is due to the fact that the parameters of a non adaptive algorithm, as the previous, have to be redesigned every time the set-up changes.

The following figure represents the block diagram of the algorithm where:

C : adaptive control,
P : secondary path,
P^ : identified secondary path,
LMS : LMS algorithm,
x(k) : reference signal,
y(k) : control signal,
z(k) : anti vibration signal,
d(k) : disturb vibration signal,
e(k) : error signal,
w(k) : modeled anti vibration signal.

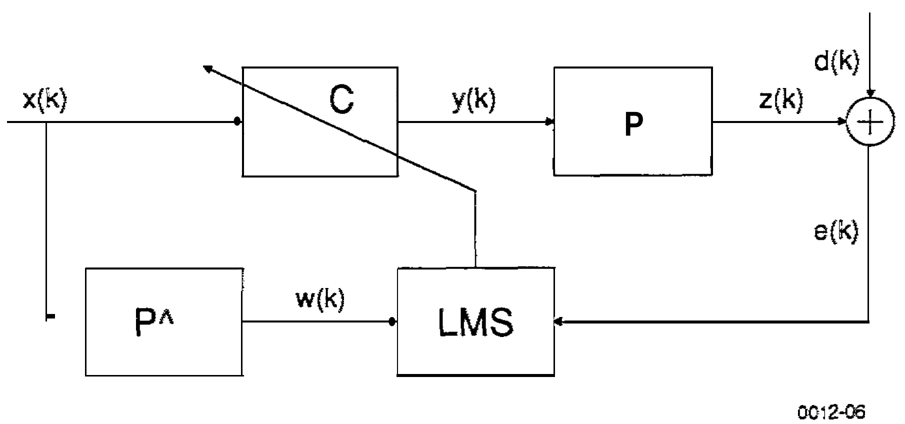


FIGURE 18 - BLOCK DIAGRAM OF ADAPTIVE ALGORITHM

This algorithm takes information from a feedforward signal (reference one) and updates the control signal taking into account the feedback signal.
It can be split into two parts as showed into the following block diagram:

identification,
control.

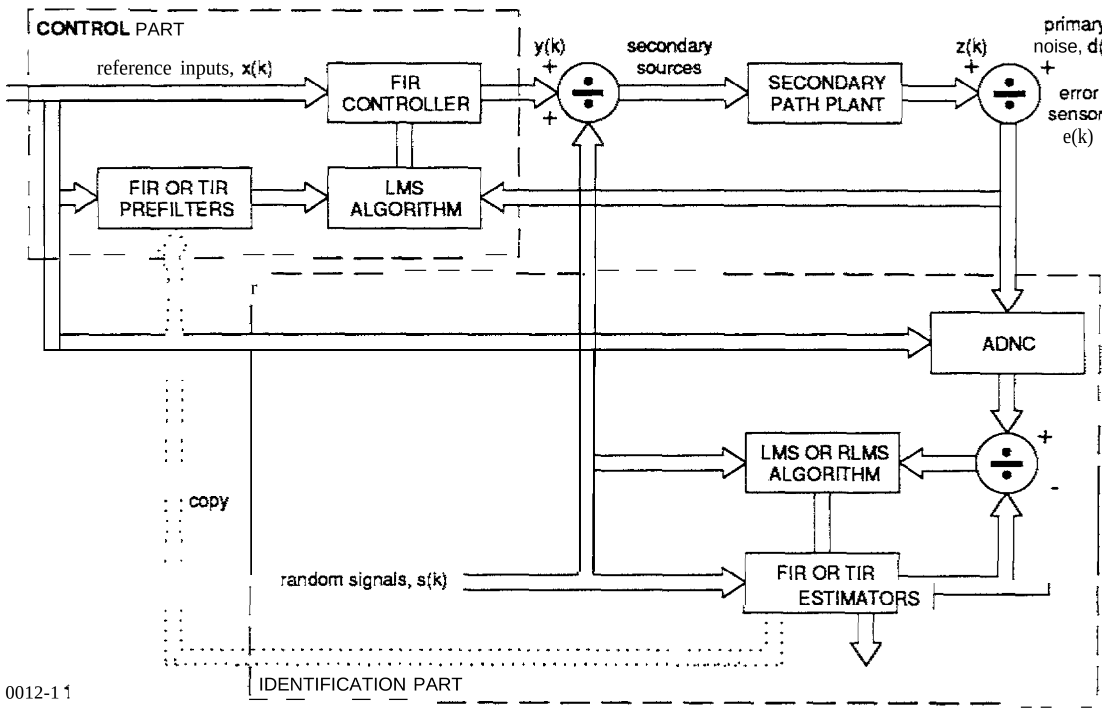


FIGURE 19 - BLOCK DIAGRAM OF THE FILTERED-X LMS ADAPTATIVE FEEDFORWARD CONTROL ALGORITHM

The control and identification programs were implemented in C-Code.
The Filtered-X LMS feedforward control algorithm uses a model of the secondary path (i.e. the transfer function from the control actuator to the error sensor) for each update of the control filter taps. These secondary path is identified during an off-line identification procedure. For the experiments only FIR filters were used for the secondary path model as well as for the control filter. The number of filter taps must be determined for bath types of filters in order to obtain optimal performance.

The following figure shows the platform where the positions of the reference and error sensors is shown.

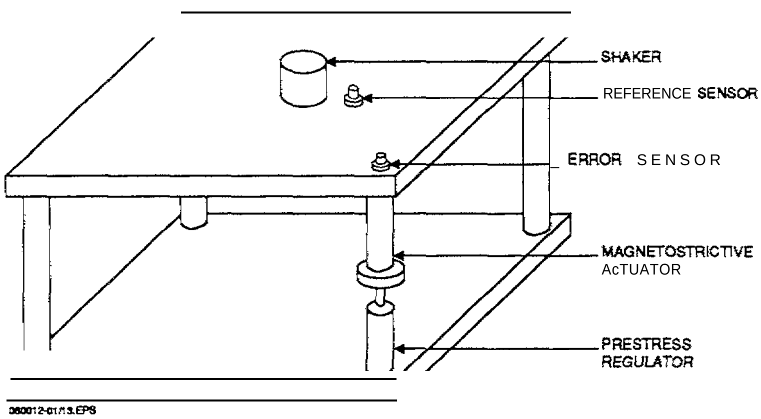
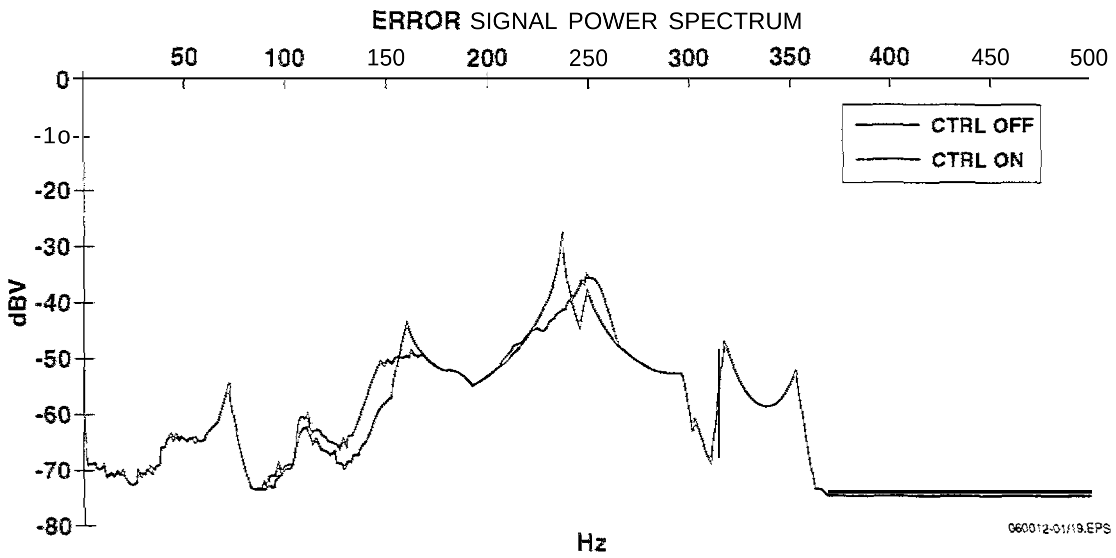


FIGURE 20- SET-UP CONFIGURATION

The control set-up is practically the same of the one used during tests of LQG algorithm. The difference is that another accelerometer is necessary as reference sensor.
Besides, error' and reference signals have been filtered with a 4th order low pass filter in order to avoid the introduction of disturbs at higher frequency (antialias). The cut off frequency is 320 Hz.
The actuator signal was filtered by a band pass filter [40-320 Hz) in order to avoid the resonant behaviour at low frequency of the actuator (that might saturate the actuator) and not to introduce vibrations at higher frequency.

Figure 21 represents the power spectrum of the error signal during the same acquisition with the control system off and on (dB Volts and Linear Volts). The acquisition is averaged with 84 samples.
Due to the complexity of the structure the error power spectrum without control in not a flat spectrum in the frequency range of interest (as it is for an ideal platform), so the control system focus itself into the reduction of the most important resonance peaks of the structure. The control system is able to achieve an overall reduction of 3 dB, with peak reduction of 13.5 dB at the frequency of 233.75 Hz (78 % of reduction of the acceleration measured at that frequency].

The noise reduction results using this type of controller is less significant (less than 15 dB reduction for the more significant peak) and needs optimization. However this adaptative feed forward control has the avantages that it doesn't need a redesign when the set-up changes.



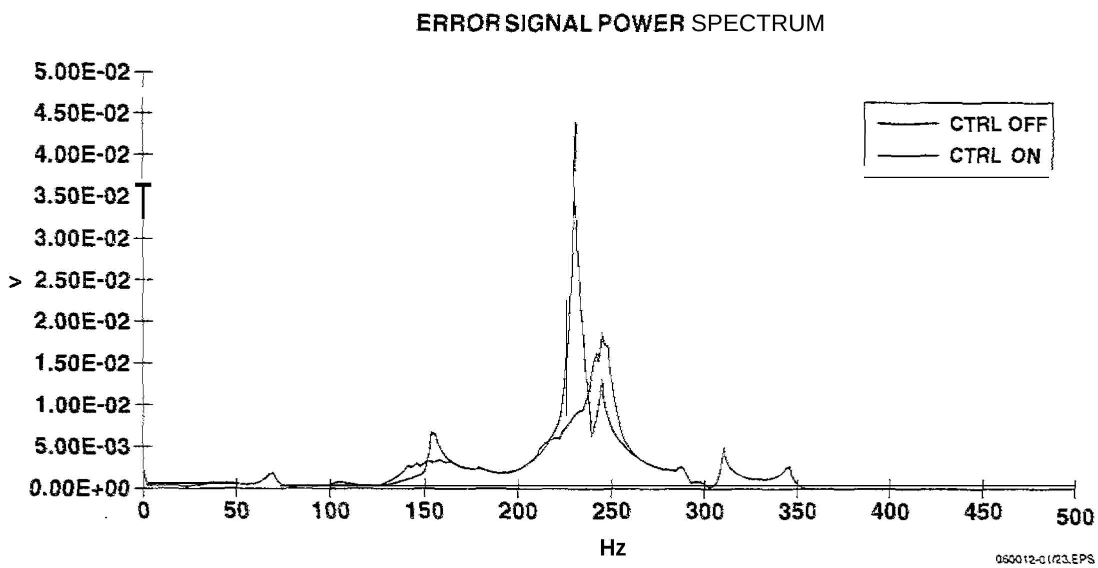


FIGURE 21 - VIBRATION CANCELLATION TEST ON THE PLATE STRUCTURE USING THE FILTERED-X LMS ADAPTATIVE FEEDFORWARD CONTROL SYSTEM

7- CONCLUSION

These experimentations have shown the obvious interest of the magnetostrictive actuator in active vibration control problems.

The Terfenol actuator is a real alternative in this field since control *can* be achieved with much lower voltages than piezo-ceramic actuators usually require.

The potential market is seems large indeed. There are many possible civilian and military application. Some examples are in the fieds of aircrafts, railway vehicles, cabines of helicopters, ships and cars . . . .

The industrial benefits are not only in anti-vibration systems, but also in fields of robonics high speed machinery, and acoustic systems.

However for the future, substantial cost reduction is still required.

The next step is to investigate in order to decrease the cost of the Terfenol-D rod which represents more the 50 % of the total cost of the actuator. This progress calls for a tight cooperation between materials manufacture, actuator manufacturers and industrial end-u sers... [t is clear, that the mst of these devices will continue to decrease thanks to design improvements and increased production volumes.

The resulting improvement in profit margin on active control systems should ensure an increasingly widespread application of these systems as we approach the year 2000. European Community is continuing to play a leading role in these developments.
A further evaluation of the potential market has to be investigated.
This project could be the basis for a type ‘VALUE” EC programme.

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