SOME THEORETICAL ASPECTS REGARDING ACTIVE VIBRATION CONTROL

Cornel MARIN[1], Ionel RUSA[2], Mihai ZDRAFCU[1]

[1]VALAHIA University Targoviste Romania, [2] BIT INVEST Romania:

E-mail: marin_cor@yahoo.com

Abstract: *An active control of vibration for a mechanical system involves the introduction an external source to perform its function to reduce vibration and noise level, such as for example, use of hybrid absorbers with controlled force to reduce the influence of seismic waves or wind power on buildings or mechanical structures, noise reduction in the cabin of an aircraft by reducing vibration of the large panels and thin metal forming booth walls; reduction of vibrations level using piezoelectric or electromagnetic devices mounted on the trailing edge of helicopter blades, devices tuning cars, etc. In this paper we are presented main active vibration control techniques currently employed to control of natural frequency in aim to reduce the effects of vibration by changing damping, stiffness or inertia of the system.*

Keywords: *control, active, vibration,*

1. ACTIVE CONTROL METHODS

The different principles *passive and semi active vibration control* presented in same paper [6, Marin C.] can be summarized in the table 1, which corresponds to different position in relation to the structure.

Table 1. The different devices type passive, semi active and active vibration control

	Passive	*Semi-active*	*Active*
On the interface	Suspensions	Self-tuning	Active suspensions
Inside the structure	Absorbers	Self-adjusting absorbers	Active absorbers
Near the source	Resonators	Self-adapting resonators	Active resonators

So the *active systems of vibration control* should be used in many applications requiring decrease the vibrations level. A vibration control system is called *active* if it uses external power to perform its function. The principle of the internal load control system is to inject a set of dynamic loads into the structure in order to minimize its vibratory response.

The basic system of active vibration control contain an actuator, a sensor, and a microprocessor (Fig.1). The *actuator* applies a controlled force to the mass whose vibration is to be reduced. *The sensor* measures the motion of the mass in terms of displacement, velocity or acceleration, depending on the application. *The microprocessor (computer)* continuously measures and controls the parameters of vibration and modifying the level of forces the elastic and damping system.

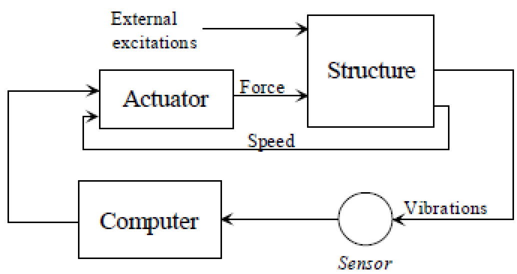

Fig.1.

2. ACTIVE SUSPENSION

The loads depend on the vibratory condition are introduced via hydraulic actuators suitably located on the structure itself. Each actuator introduces in the structure two control forces, F_1 and $-F_1$, which are of opposite sign (Fig.2).

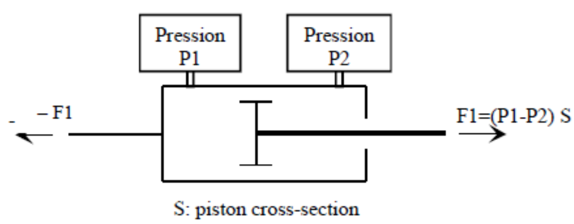

Fig.2

The principle of this basic system is to modify the distribution of internal loads in the structure. The role of the hydraulic actuators is no longer to disperse the vibratory energy of the structure - a dispersing control would increase the reduced damping of different natural modes - *but to modify the distribution of vibratory* *energy for the natural modes of vibration to minimize the effect to the dynamic behavior of structure.*

The global vibratory structure is modeled by a system with *n* degrees of freedom, using for example a finite element method (FEM). In figure 3 is show the block diagram of the mechanical system subjected to *external vector excitation* U the *vector force* V as input and *vector response* X as output.

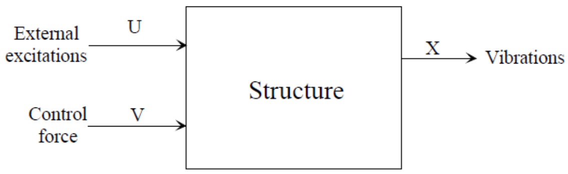

Fig.3

The dynamical behavior of the mechanical system is given by matrix equation:

$$M \cdot \ddot{X} + C\dot{X} + K \cdot X = F \quad (1)$$

where: X is the vector response;

M, C, K are respectively: the mass, damping and stiffness matrices,

F - the excitation vector.

The excitation consists of *external excitations* denoted by vector U and *control excitations* by vector V.

Then: $F = T \cdot U + S \cdot V \quad (2)$

where T is the local matrix of external excitations,

S - the local matrix of control excitations.

For a particular harmonic excitation:

$$F = F_0 \cdot e^{i\omega t} \quad (3)$$

it define the *isochronous transfer function* of the structure as follows:

$$X = H(\omega) \cdot F_0 \cdot e^{i\omega t} \quad (4)$$

where $H(\omega)$ is the transfer matrix (complex flexibility of mechanical system):

$$H(\omega) = \frac{X}{F} = (K - M\omega^2 + iC\omega)^{-1} \quad (5)$$

The *control excitation* $S \cdot V$ must sense a relative modal stiffness of the modes to be controlled, which should be as low as possible.

The actuators cannot be placed between the vibration nodes of a structure, because the control would not be efficient. This is the reason why it is necessary to introduce a supplementary flexibility in parallel with the actuator, enables to reduced the local flexibility of the structure, in order to decreasing the control force.

The control system of internal loads consists of three main elements: actuator, computer and sensor (Fig.1). The sensor measures the level of vibrations (stresses or loads) and send the signal directly to the computer.

There are two important types of algorithms associated with this output:

1. *the frequency algorithm* (FFT), which focuses on a first few number of harmonics;

2. the *time-domain algorithm*, often based on the recursive least squares algorithm at a filtered reference - X filtered *Least Mean Squares* (LMS).

The latter type of algorithm is often used in the *active control of noise and vibrations*, when there is a reference signal correlated to the noise or vibrations to eliminate.

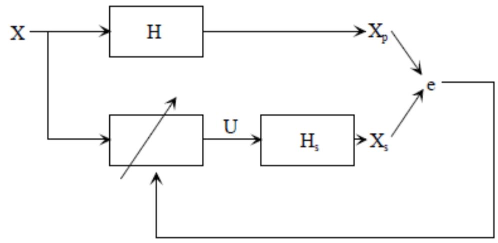

Fig.4

We consider the block-system for Figure 4, where the problem is to determine the *filter coefficients W* that minimize the error *e*.

Because of the scalar and discrete signals, denote by *input* $W(w_0, w_1 \ldots w_{N-1})$ can be sought in the form of a *finite impulse response* (FIR) filter of time N.

Thus, the *output* $U(u_0, u_1 \ldots u_{N-1})$ of the filter at instant time k is written as a convolution integral:

$$u_K = \sum_{i=o}^{N-1} w_i \cdot x_{k-i} \quad (6)$$

where: X_p is the vector response;

X_s is the output of the filter at the instant: $k=0,\ldots N$

The recurrent algorithm is written as follow:

$$w_{i(k+1)} = w_{ik} - \mu \frac{\partial J}{\partial w_{ik}} \quad (7)$$

where: μ is the convergence coefficient,

$w_{i(k)}$ the filter coefficient FIR at a sampling time k

J the square-law function of the coefficients defined by:

$$J = \sum_k e_k^2 , \quad (8)$$

$$e_k = x_k^s + \sum_{j=0}^{N-1} w_j \left(\sum_{i=0}^{M-1} h_i^s \cdot x_{k-i-j} \right)$$

We obtain the recurrent algorithm:

$$w_{i(k+1)} = w_{ik} - 2\mu e_k \sum_{i=0}^{M-1} h_i^s \cdot x_{k-i-j} \qquad (9)$$

The calculation of a given coefficient of the FIR filter requires the product of the value of the filtered reference signal X_s associated with this coefficient, and of a signal X_p proportional to the error.

This process is repeated continuously until a stable coefficient value w_i is obtained, corresponding to a minimum error. In order to achieve the convergence coefficient μ, will optimize the number k of iterations necessary for algorithm convergence for the set of filter coefficients.

The analytical formulation of the algorithm presented can be applied to determine the dynamic loads and the displacements of the actuator required for the control. This method can be used for the complete specification of the required actuator.

3. ACTIVE RESONATOR

In order to achieve the decrease vibrations in the structure is sometimes used the *active resonator* with dynamic loads, which are controlled by computer with the purpose to minimize the vibratory levels. The technology of generating dynamic loads is fundamental in the active control vibration [1-Thomas Krysinski]. There are two main technologies for generating dynamic loads by:

– *specific dynamic loads* or already-existing load generators;

– *active resonator* using resonant active systems.

The principle of an active resonator is by displacing a mass with an actuator. The resonator uses the dynamic amplification of the mass to generate high loads using minimal energy (Fig.5)[1-Thomas Krysinski].

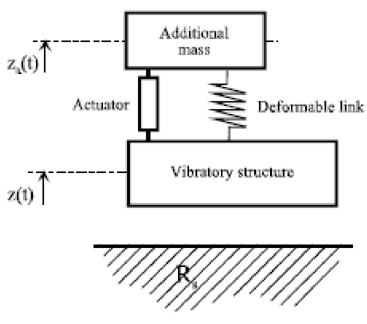

Fig.5.

Different types of actuator technologies have been used to match the fields of application in terms of forces and frequencies. These methods will be developed with: electromagnetic actuator, piezoelectric actuator and hydraulic actuator.

3.1. Electromagnetic active resonator

The mechanical parameters (mass, stiffness) are chosen so that the magnetic loads remain small and compatible with an acceptable energy consumption (operation in the vicinity of the resonance point of the resonator to minimize the control loads). The natural frequency of the mass-spring system must be as close as possible to the frequency of vibrations that we want to control. In figure 6 is presented the principle of *single-stage active resonator system* whose the displacement is obtained with electromagnetic forces [1-Thomas Krysinski]. The mechanical parameters are chosen so that the magnetic loads remain small and compatible with an acceptable energy consumptions. The natural frequency of system must be as close to the frequency of vibrations that we want to control.

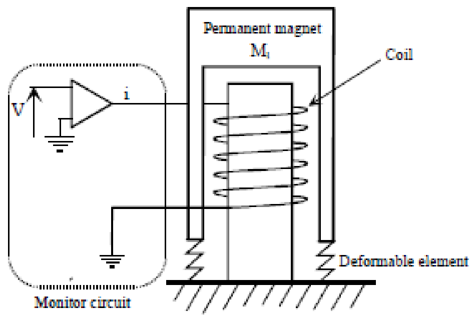

Fig.6

The technology manufacture of single stage *resonator system* is presented in figure 7.

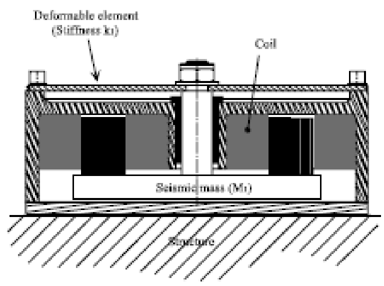

Fig.7

In figure 8 is presented the principle of *two-stage resonator system* where the control forces is obtained between two masses by an electromagnetic load $F_v = kV$ [1-Thomas Krysinski].

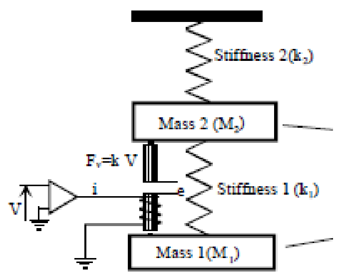

Fig.8

The technology manufacture of *two-stage resonator system* is presented in figure 9 [1-Thomas Krysinski].

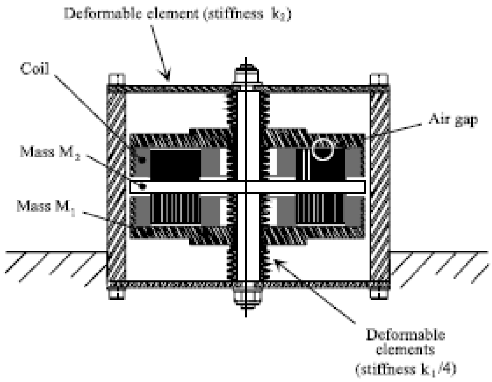

Fig. 9

The two-stage resonator system has the two following advantages: the control force is linear around the control frequency and the dynamic amplification gain generate very high loads with low control energy.

3.2. Hydraulic active resonator

Some actuators cannot provide enough force for larger applications such as vehicle suspension and control of building motions. In building application, hydraulic cylinders are usually used. Active vehicle suspensions use hydraulic devices, electric motors, and magneto-rheological fluid dampers.

The role of the hydraulic actuators is to modify the distribution of vibratory energy for different modes to minimize the structure vibrations, instead of dispersing the vibratory energy of the structure. The design of the hydraulic active resonator is like a passive resonator, but with a hydraulic actuator added. The resonance peak of the structure due to dynamic excitation is replaced by an anti-resonance with two new peaks. The disadvantage of passive resonator is that anti-resonance is very sensitive in the area of natural frequency, such that a small variation of excitation frequency, can shift the system back into a resonance area [6, 7 - Marin C].

So, two active hydraulic resonator solution are used for introducing the energy for vibration control: one with *displacement control* and the others with *load control* (Fig. 10).

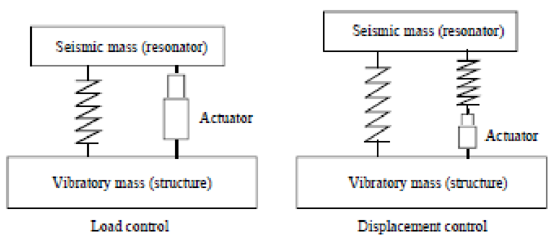

Fig. 10

The displacement control solution has the advantage of not operating at anti-resonance, where the passive resonance system is sufficient to control the vibrations.

The load control requires a non-zero control at anti-resonance. In addition, load control with hydraulic actuator is more technologically complex, therefore the displacement control methode was selected. The principle of *closed loop control* is presented în figure 11.

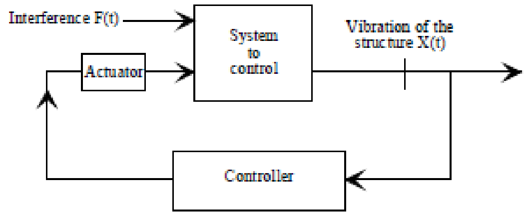

Fig. 11

The results for some achieved tests of the transfer function of acceleration with respect the angular frequency for the open (without active system) and closed loop (with active system) are presented în figure 12 [1-Thomas Krysinski].

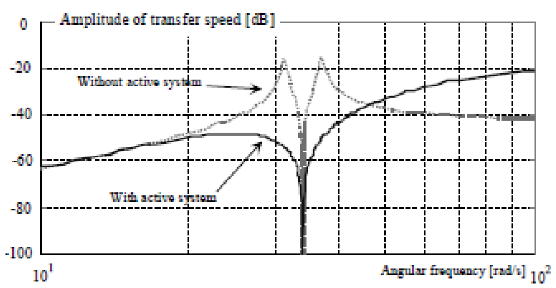

Fig. 12

The experimental results achieved demonstrate the efficiency of active systems compared to passive systems without resonators of the acceleration of the structure with respect the angular frequency. They are presented în figure 13 [1-Thomas Krysinski].

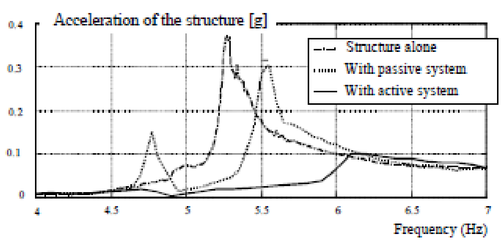

Fig. 13

3.3. External forces generators

For certain application the high level of vibration can be reduced by applying a technology of external load generators. The most used types of generators are centrifugal generators which uses unbalance to generate dynamic loads.

The system requires the action generated to be unidirectional force and the amplitude to be adjusted independently with excitation frequency. The centrifugal generator with four rotating masses, whose centers of gravity are offset with respect axis of rotation are prese4nted in figure 14.

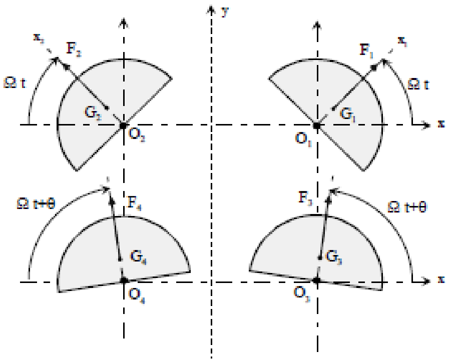

Fig. 14

With this four rotating masses, it is possible to obtain the desired properties by adjustment the rotation direction and their angular phase θ. between the upper and lower counterweights, they are independently of the excitation frequency Ω (Figure 14).

3.4. Piezoelectric actuator

Piezoelectric materials can convert electrical current into motion, and vice-versa. They change shape when an electric current passes through them, and they generate an electric signal when they suffering deformation.

Thus they can be used as actuators to create force or motion, and as sensors. The materials used for high-precision actuation include electro-strictive and magneto-strictive properties. These are Ferro-magnetic materials that suffering deformation when they are subjected to an electro-magnetic field. A voice coil motor VCM actuator has a coil of wire rigidly attached to the structure and suspended in a permanent magnetic field. It is driven when a force is produced to accelerate it radically as a current is passed through the coil.

3. CONCLUSION

Various classical techniques of vibration control were discussed in this chapter and some typical active vibration techniques such as using *active control systems* were introduced individually*: active absorbers*, *active resonator*, or *active suspension*.

The principle of *active control systems* is based on the injection of supplementary energy at the level of the excitation source, or at the level of the interface between the excitation source and the mechanical structure, or in the place where vibrations need to be controlled It is very important to choose the systems according to the place of their implementation on the structure, and also according to the function of the system. As such we differentiate between *active control systems*:

– systems that act at the *interface of two structures*; the function of the system is to isolate one from the other;

– systems that act *inside the structure* and control vibrations exactly in the place where comfort is required. The cushions of a car seat are a good example of this type of system;

– systems that act *near the source* in order to minimize a level of vibrations.

REFERENCES

[1] Tomasz Krysinski, Francois Malburet - Mechanical vibrations: active and passive control, ISTE USA 4308 Patrice Road Newport Beach, CA 92663 USA

[2] Colin Hansen n Scott Snyder , Xiaojun Qiu, Laura Brooks n Danielle Moreau - Active Control of Noise and Vibration, CRC Press , Taylor & Francis Group 6000 Broken Sound Parkway NW, 2013

[3] ANDRÉ PREUMONT - Vibration Control of Active Structures. An Introduction - Kluwer Academic Publishers New York, Boston, Dordrecht, London, Moscow, 2004

[4] S.M. Savaresi, C. Poussot-Vassal. C. Spelta, O. Sename, L. Dugard - Semi-Active Suspension Control Design for Vehicles- Elsevier, Kidlington, Oxford, OX5 1GB, UK, USA, 2010

[5] Chunling Du, Lihua Xie- Modeling and Control of Vibration in Mechanical Systems - CRC Press , Taylor & Francis Group 6000 Broken Sound Parkway NW, Suite 300 Boca Raton, FL 33487-2742, 2010

[6] Marin, C., Rusa, I, Zdrafcu, M. – Some theoretic and practical aspects regarding passive and semiactive vibrations (Same Issue).

[7] Marin, C. – Vibratii mecanice. Aplica ii. Probleme, Ed. Bibliotheca 2008, ISBN 978-973-712-397-8