SOME THEORETIC AND PRACTICAL ASPECTS REGARDING PASSIVE AND SEMI ACTIVE VIBRATION CONTROL

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

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

email: marin_cor@yahoo.com

Abstract. *The vibrations most often leads to undesirable effects such as mechanical failure, costly maintenance of machines, worsening positioning performance of machines tools and also human discomfort. Theoretically, the vibrations can sometimes be eliminated. However, due to the high manufacturing equipment cost that may be involved in eliminating or reduction of vibration, we need a compromise between an acceptable level of vibration and the supplementary manufacturing cost of devices. In this paper have been presented some classical techniques of passive and semi active vibration control in order to reduction the vibration level, the control of natural frequencies through damping modification, using of different isolators and absorbers systems.*

Keywords*: passive , semi active, vibration, control*

1. PASSIVE VIBRATION CONTROL

The passive vibration isolation methods are used to reduce the levels of vibration and that involves the insertion of a resilient element called an *isolator* between the vibrating mass and the source of vibration. We obtain that a reduction the dynamic response of the system under specified conditions of vibration excitation.

An isolation system is active or passive depending on whether or not external power is required for the isolator to perform its function. A passive isolator consists of a resilient member, i.e. mass-stiffness, and an energy dissipater, i.e., damping. Typical examples of passive isolators include metal springs, cork, pneumatic absorbers and rubber isolators.

These different principles passive and semi active vibration control 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

2. PASSIVE DEVICES IN VIBRATION CONTROL

2.1. Absorbers without damping

If a mechanical system is subjected to force whose excitation frequency nearly coincides with its natural frequency of system, the stationary vibration can be reduced by using the absorber, the devices *inside of structures.* The simply dynamic absorber is an auxiliary mass m_2 attached through a spring of stiffness k_2 to main system (Fig.1).

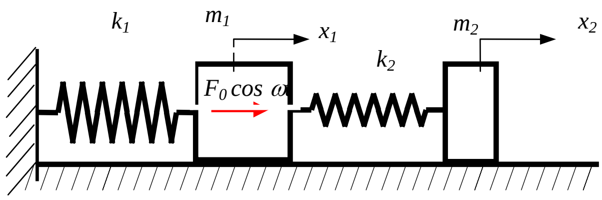

Fig. 1

The equations moving of the m_1 and m_2 masses are:

$$\begin{cases} m_1\ddot{x}_1 + (k_1 + k_2)x_1 - k_2x_2 = F_0 \cos \omega t \\ m_2\ddot{x}_2 + k_2x_2 - k_2x_1 = 0 \end{cases} \quad (1)$$

where x_1 and x_2 are displacements of masses m_1 and m_2, respectively:

$$\begin{cases} x_1 = a_1 \cdot \cos \omega t \\ x_2 = a_2 \cdot \cos \omega t \end{cases} \quad (2)$$

The dynamic steady-state amplitudes of the masses m_1 and m_2 are given by [6 - Marin, C.]:

$$\begin{cases} a_1 = \dfrac{F_0\left(-\omega^2 m_2 + k_2\right)}{\left(-\omega^2 m_2 + k_2\right)\left(-\omega^2 m_1 + k_1 + k_2\right) - k_2^2} \\ a_2 = \dfrac{F_0 k_2}{\left(-\omega^2 m_2 + k_2\right)\left(-\omega^2 m_1 + k_1 + k_2\right) - k_2^2} \end{cases} \quad (3)$$

Equation (3) implies that if :

$$\omega^2 = \frac{k_2}{m_2} \Rightarrow a_1 = 0 \quad (4)$$

the amplitude a_1 of the mass m_1 will be zero.

Consider before the addition of the dynamic vibration absorber that the machine operating near its natural resonance:

$$\omega^2 = p^2 = k_1 / m_1 \quad (5)$$

the stationary amplitudes given by relations (3) becomes:

$$\begin{cases} a_1 = \dfrac{F_0}{k_1}\dfrac{\left(1-\eta^2\right)}{\left(1-\eta^2\right)\left(1+\mu-\eta^2\right)-\mu} \\ a_2 = \dfrac{F_0}{k_1}\dfrac{1}{\left(1-\eta^2\right)\left(1+\mu-\eta^2\right)-\mu} \end{cases} \quad (6)$$

The graphical dependence of two mass amplitudes on the frequency dimensionless was made using MATHCAD for two values of dimensionless stiffness ($\mu=0.5$ and $\mu=0.2$), as show in the figures 2 and 3.

It can be seen from the figures 2 and 3 that with decreasing mass dynamic absorber, the main system stability tapering and the amplitude absorber increase greatly.

If it is necessary to reduce the amplitude of vibration of the absorber over a range of frequencies, a damped dynamic absorber can be used.

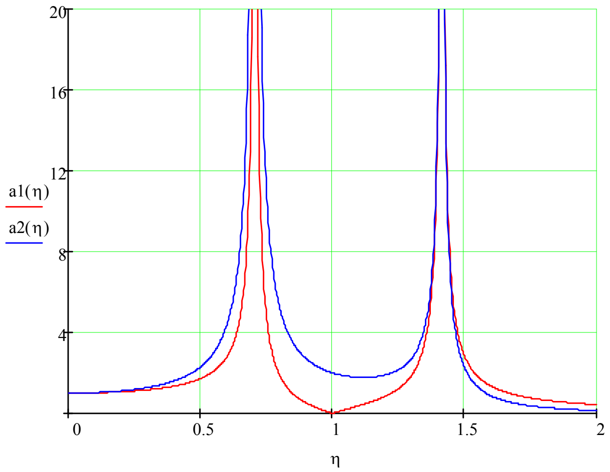

Fig. 2 .($\mu = \dfrac{k_2}{k_1} = \dfrac{m_2}{m_1} = 0{,}5$ **)**

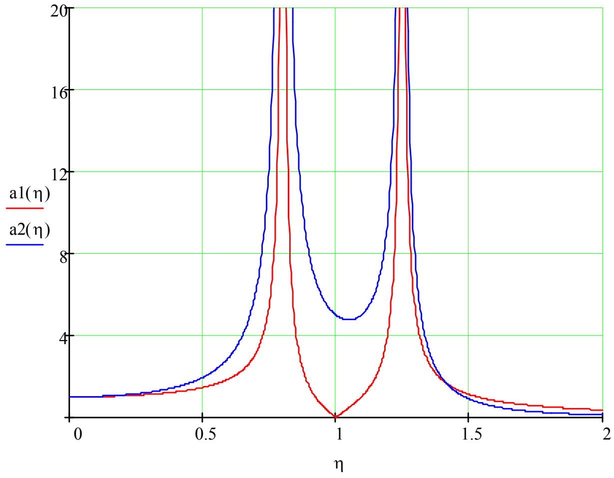

Fig. 3 : ($\mu = \dfrac{k_2}{k_1} = \dfrac{m_2}{m_1} = 0{,}2$ **)**

2.2. Absorbers with damping

The dynamic damped absorber consists a spring-mass system, an auxiliary mass m_2 attached to a mass m_1 through a spring of stiffness k_2 and damp element (c_2 damped coefficient, Fig.4)

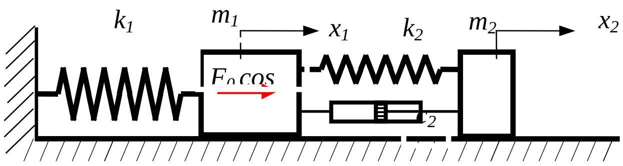

Fig. 4. Damped absorber

The equations moving of the masses m_1 and m_2 are written as:

$$\begin{cases} m_1\ddot{x}_1 + c_2\dot{x}_1 - c_2\dot{x}_2 + \left(k_1 + k_2\right)x_1 - k_2 x_2 = F_0 \cos\omega t \\ m_2\ddot{x}_2 - c_2\dot{x}_1 + c_2\dot{x}_2 - k_2 x_1 + k_2 x_2 = 0 \end{cases} \quad (7)$$

where x_1 and x_2 represent displacements of m_1 and m_2, respectively:

$$\begin{cases} x_1 = a_1 \cdot \cos\omega t \\ x_2 = a_2 \cdot \cos\omega t \end{cases} \quad (8)$$

The dynamic steady-state amplitudes of the mass m_1 and m_2 are given by [6 - Marin, C.]:

$$a_1 = F_0\sqrt{\frac{\left(k_2 - \omega^2 m_2\right)^2 + \left(c_2\omega\right)^2}{\left[\left(k_1 + k_2 - \omega^2 m_1\right)\left(k_2 - \omega^2 m_2\right) - k_2^2\right]^2 + \left[c_2\omega\left(k_1 - \left(m_1 + m_2\right)\omega^2\right)\right]^2}}$$

$$a_2 = F_0\sqrt{\frac{k_2^2 + \left(c_2\omega\right)^2}{\left[\left(k_1 + k_2 - \omega^2 m_1\right)\left(k_2 - \omega^2 m_2\right) - k_2^2\right]^2 + \left[c_2\omega\left(k_1 - \left(m_1 + m_2\right)\omega^2\right)\right]^2}} \quad (9)$$

The above dynamic vibration absorber removes the original resonance peak in the response of the main system, but introduces two new resonance finite peaks .

We use the follow dimensionless for:

- relatively excitation frequency $\eta = \dfrac{\omega}{p}$ (10)

- stiffness factor $\mu = \dfrac{k_2}{k_1}$ (11)

- damper factor: $\zeta = \dfrac{c}{c_{cr}}$, $c_{cr} = 2\sqrt{k \cdot m}$ (12)

The dynamic stationary amplitudes a_1 *and* a_2 given by relations (9) becomes:

$$a_1 = \frac{F_0}{k_1}\sqrt{\frac{\left(1-\eta^2\right)^2 + (2\zeta\eta)^2}{\left[\left(1-\eta^2\right)^2 - \mu\eta^2\right]^2 + \left[2\zeta\eta\left(1-\eta^2(1+\mu)\right)\right]^2}}$$
$$a_2 = \frac{F_0}{k_1}\sqrt{\frac{1 + (2\zeta\eta)^2}{\left[\left(1-\eta^2\right)^2 - \mu\eta^2\right]^2 + \left[2\zeta\eta\left(1-\eta^2(1+\mu)\right)\right]^2}} \quad (13)$$

The graphical dependence of two mass amplitudes on the frequency was made using MATHCAD for two values of damper factor (ζ=0.1 and ζ=0.2) and two values of stiffness factor (μ=0.5 and μ=0.2) is show in the figures: 5 , 6, 7 and 8.

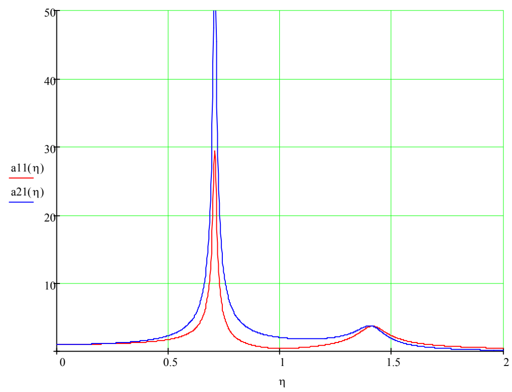

Fig. 5 : ($\mu = \dfrac{k_2}{k_1} = \dfrac{m_2}{m_1} = 0.5$ *and* $\zeta = 0.1$)

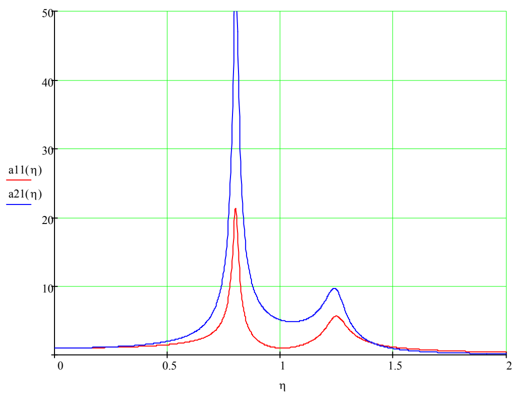

Fig. 6 : ($\mu = \dfrac{k_2}{k_1} = \dfrac{m_2}{m_1} = 0.5$ *and* $\zeta = 0.2$)

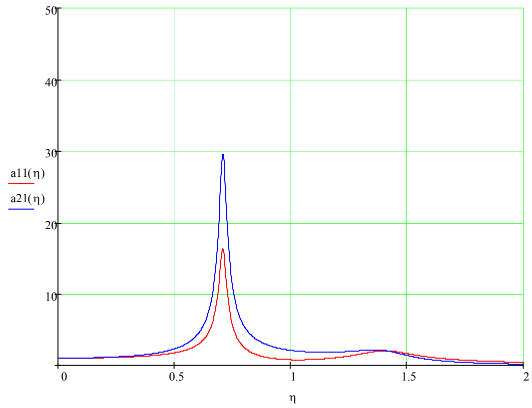

Fig. 7: ($\mu = \dfrac{k_2}{k_1} = \dfrac{m_2}{m_1} = 0.2$ *and* $\zeta = 0.1$)

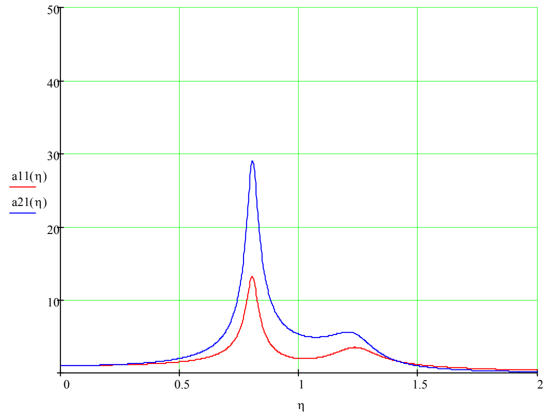

Fig. 8: ($\mu = \dfrac{k_2}{k_1} = \dfrac{m_2}{m_1} = 0.2$ *and* $\zeta = 0.2$)

2.3. Resonators

Resonators are used in certain cases where it is easier and more efficient to locate anti-vibration systems *near the vibration source.* The idea is to create another source of vibrations which will cancel out the original vibration.

The mechanical principle is to use the kinetic energy of resonator where seismic mass m_1 is coupled to the vibratory structure either by a kinematic coupling or via a very flexible coupling (Fig.9) [1-Thomas Krysinski].

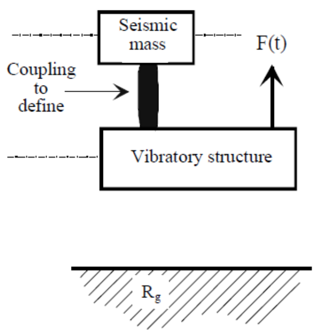

Fig. 9.

2.4. Suspension

The suspensions are used in certain cases where it is more efficient to locate anti-vibration systems *on the interface of vibratory structures.* The suspension is defined as the link between two structures, in order to isolate the structures one from the other. The suspension is a concept for structure isolation that is achieved by canceling the loads transmitted to the support structure.

The characteristics of the suspension are determined by analyzing the dynamic behavior and vibrations without modifying its main function in static characteristics.

In automotive practice (Fig.10), the link characteristics can be modified by varying its stiffness, damping or appropriate positioning of the system natural frequencies and its damping in terms of energy transfer using VOIGT KELVIN suspension type.

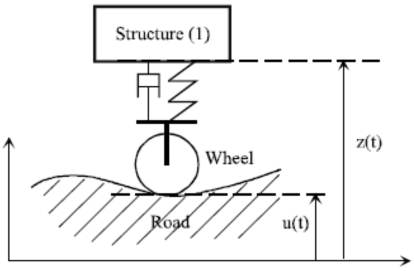

Fig. 10.

In this case, transmissibility factor is defined as the ratio between amplitude of displacement *z(t)* and amplitude of road oscillation *u(t):*

$$T = \frac{z_0}{u_0} = \frac{1}{\sqrt{(1-\eta^2)^2 + (2\zeta\eta)^2}} \quad (14)$$

Graphical dependence of on transmissibility factor (14) to the relatively excitation frequency was achieved using MATHCAD program for three values of factor damper *(ζ=0.1, ζ=0.2 , ζ=0.5,* Fig. 11).

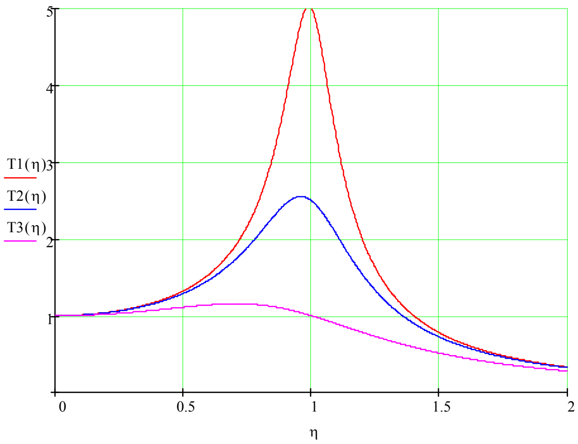

Fig.11

The analysis of the three curves of figure 11 shows that are two possible types of links:

-the link very rigid (i.e. bolted assembly) when the resonance is much higher than the disturbing frequencies. In this case the *damping has very little influence on the behavior of suspension.*

-the link very flexible, when the resonance is lower than the disturbing frequencies. In this case the influence of damping not play an important role on the behavior of suspension.

On the other hand, we observe that it is worthwhile reducing the resonance peak by using damping suspension.

2. SEMI-ACIVE VIBRATION CONTROL

2.1. Stiffness modification

The dynamic behavior of structure is the result from the exchange of mass and spring energy and dissipation energy of damper element. Dynamic forces transfer their energy to the structure by spring couples, which then respond via cinematic of several mechanisms.

Dynamic behavior can be modeled in several ways; the basic of dynamic behavior is Newton's second law of motion. If the external force is static or quasi-static shape, structural stiffness forces develop to create an mechanical equilibrium.

The external dynamic forces are balanced in a complex way with inertial, stiffness and damping forces. Stiffness forces denote the capacity of a spring system to store strain energy. The stiffness force follows Hooke's law: $F_s = k_s{\cdot}x$, where the stiffness constant k_s is expressed in *N/m* for a linear HOOKE model.

Stiffness modification is possible by introducing an *semi-active* vibration isolation: *the proportionally actuator* (Fig.12) link both parts: the vibrating body and the support structure [2, Colin Hansen].

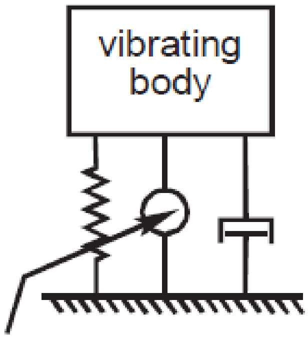

Fig.12

2.2. Damping modification

Damping is the ability of a structure to dissipate mechanical energy. For an oscillatory system, *damping energy* is a measure of how much energy is dissipated by the system during an oscillation cycle. For example, the if we use the structural materials connections between components, there add or remove damping to a structure.

If a system undergoes a forced vibration, its response of vibration near resonance tends to become very large if there is no damping (see Figure 2 and 3). The presence of damping always limits the amplitude of vibration. (see Figure 5 and 6).

Damping modification in the system is often used to control its response, by using the structural materials having high internal damping, such as laminated or sandwiched materials [2, Colin Hansen] (Fig.13).

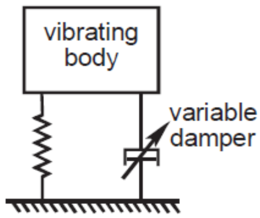

Fig.13

An example is the use of intelligent viscoelastic materials; these materials are used for semi active vibration control, they are subjected to shear stresses or strains. A simplest arrangement is that a layer of the intelligent viscoelastic material is attached to an elastic one. Another arrangement is that a viscoelastic layer is sandwiched between two or more elastic layers. The material with the largest loss factor will be subjected to the smallest stress, while the stress is proportional to the displacement. Hence viscoelastic materials having large *damping factor* are used to provide internal damping for vibration control. The viscous damper is efficient at low frequencies (resonance filters) they are very badly at higher frequencies. The ideal solution would be to apply damping only in the resonance area, for which a fluid should be used. The last techology solution of a high damping system used in the automotive industry is represented in Figure 14 [1,Thomas Krysinski].

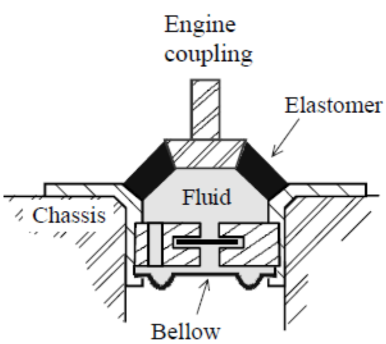

Fig.14

The damping system include an elastic cone-shaped element made of very resilient rubber. A rigid plate delineates a closed chamber fluid under the cone. A second chamber is delineated on the other side of the plate by a very flexible metallic bellows. An orifice through the plate, called *column* enables communication between the upper and lower chambers. The interior part of the support is filled with viscoelastic fluid.

The strain energy is generated by the chamber wall (bellow) deformations and the kinetic energy by the fluid movement through the *column*. The system's natural resonance can be designed to coincide with the vertical excitation frequency produced by the engine coupling. With this system, it is possible to generate efficient variable damping because the column transfers a great part of energy.

A supplementary device, called a *decoupling flap,* is often used to reduce transmissibility at high frequency. It provides a direct connection between the chambers for the low amplitude vibrations that must be filtered. For high amplitude vibrations that must be damped, the flap is blocked by its end stops and the fluid must flow through the column. This guarantees high damping for high amplitude vibrations.

There are thus distinct modes of operation at high and low frequencies, which produce excellent filtering at both the resonance frequency and higher frequencies. This system compensates for the negative effects of the active viscous systems.

3. CONCLUSIONS

It is important to present the different passive and semi active anti-vibration systems that are available and to outline their applications.

The above classification will always make it possible to choose the best anti-vibration system for a particular problem. Therefore, anti-vibration systems were divided, based on experience, according to:

– their mode of action: passive, active or semi-active,

– the strategy of their implementation place on the structure.

The main differences of anti-vibration systems, in terms of efficiency and complexity, are described in according to their mode of action.

So, *the passive systems*, which adjust the structure on the basis of their characteristics (mass, stiffness, damping) have the particularity of not completely adjusting to the excitation changes caused by modifications of the machine functioning conditions, of the machine mass or its configuration.

The semi-active systems work by the same principle as passive systems, but they have the capacity to adjust the setting so that they adapt to modifications either of the amplitude or frequency excitation stresses or of the dynamic parameters of the structure to control (mass, stiffness or damping).

The active systems have the best capacity to adjust the setting so that they adapt to modifications either of the amplitude or frequency excitation stresses or of the dynamic parameters of the structure to control (mass, stiffness or damping) and they will be presented in a separate paper.

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. – Vibratii mecanice. Aplicații. Probleme, Ed. Bibliotheca 2008, ISBN 978-973-712-397-8