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A STUDY ON A WIDEBAND FREQUENCY PASSIVE DYNAMIC VIBRATION ABSORBER BASED ON VISCO-ELASTIC MECHANICAL SUSPENSION

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Abstract: *This paper proves in experimental terms the high efficiency of a new type of wideband frequency passive dynamic vibration absorber (PDVA) useful in dynamic behaviour correction of mechanical structures. The PDVA uses a mass and a mechanical elastic suspension based on a visco-elastic polymer (e.g. SORBOTHANE). For damping efficiency evaluation was used an experimental set-up based on a low natural damped cantilever beam with a mass and a PDVA placed at the free end. The damping effect (up to 27 dB attenuation and 60 Hz wideband frequency) is generated by the coresonance between the mechanical structure and PDVA.*
Key words: *vibration, passive damping, wideband frequency*

1. INTRODUCTION

In mechanical systems with distributed mass (bridges, tall buildings, etc.) there are often low damped natural modes which can be excited at resonance. A high amplitude (and sometime catastrophic) vibration motion occurs because a high level of modal energy is absorbed from the excitation source (e. g. an earthquake). If the natural damping can not be increased, there are two known ways to reduce the resonating amplification, the elimination of the stored modal energy using active or passive absorbers placed on the structure (so called *skyhook dampers*). An active absorber (Paulitsch et al., 2007) uses an inertial electro-dynamic actuator collocated with a vibration sensor, a feedback regulator (Preumont, 1997), amplifiers and a power supply. A negative force proportional with the vibration velocity is generated and applied on the structure. The active absorbers have a high efficiency in vibration suppression but it needs an expensive equipment and a permanently power supply. The passive absorbers (also called passive dynamic vibration absorber PDVA or tuned mass damper TMD) have a long history started in theory probably with (Ormondroyd & den Hartog, 1928). A PDVA consist basically of a spring-mass-damper system placed on the structure where the vibration amplitude is maximal. The PDVA resonant frequency should be roughly the same with the frequency of the vibration mode to be suppressed (coresonance condition, assured usually by tuning the PDVA's mass). Thus, at resonance, the mechanical structure can not accumulate inside modal energy (supplied by the excitation source) because this energy is transferred to the PDVA. In the PDVA's damper the modal energy is converted and dissipated as heating. A PDVA absorber is a low cost and high efficiency solution in resonance suppression. It was successfully used in 2002 to avoid the collapse of the London Millennium Footbridge (due to the natural sway motion of people walking on the bridge who acted as an excitation source for a low damped lateral bridge's mode). Nevertheless the PDVA in usually design works as a narrowband frequency absorber. If the modal frequency of the structure is changed, the PDVA resonating frequency also should be changed. That is why our paper proposes a new design of PDVA using a mass and a simple spring-damper system based on a visco-elastic polymer (SORBOTHANE). Also the paper presents some experimental research results in dynamic conditions which prove a high efficiency in vibration attenuation (up to 27 dB) on a wideband frequency. The PDVA's mechanical suspension using a visco-elastic polymer (let call it PDVAV) means a good and less expensive opportunity for industrial implementation (Collette et al., 2008).

In the future we intend to increase the PDVAV performances, up to 35 dB attenuation. According to some of our previous researches it is possible to do this with an additional eddy-currents damper (Sodano et al., 2006) placed inside PDVAV. We intend to develop some new techniques for self-generated vibration (chatter) suppression in cutting processes on machine tools, based on PDVAV (e.g. a milling bar with a PDVAV inside).

2. SOME CONSIDERATION ON PDVAV DESIGN

Figure 1 presents an isometric overview on the PDVAV. The absorber consists of an outer and an inner ring made from Aluminium (AlMgSi1 alloy). The mechanical suspension (the equivalent of a spring and a damper) consists of six identical pieces made from visco-elastic polymer SORBOTHANE.

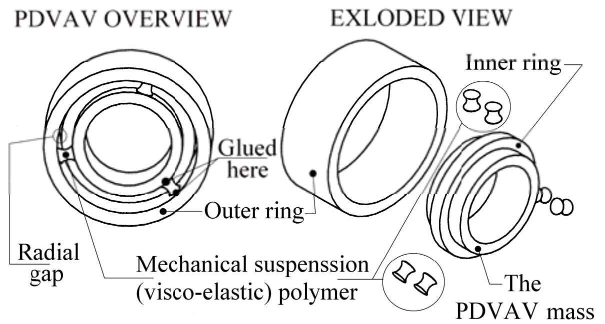

Fig. 1. An overview on PDVAV and exploded view

3. THE PDVAV EXPERIMENTAL BENCH TEST

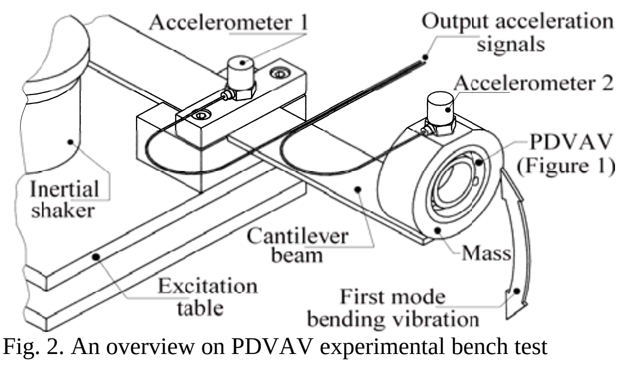

Fig. 2. An overview on PDVAV experimental bench test

Figure 2 presents an overview on the experimental dynamics bench test. It uses a cantilever beam (with adjustable length) clamped on an excitation table, and a mass at the free end. Inside the mass is placed the PDVAV described in Figure

1. An inertial shaker placed on the excitation table is used to excite the cantilever beam in vertical direction. The first bending mode of the beam (excited vibration) can be described using two identical Bruel & Kjær accelerometers (accelerometer1 and 2 on Figure2).

4. THE EXPERIMENTAL RESEARCH SET-UP

The experimental research set-up is summary described on Figure 3. It is based on the design already described in Figure 2. The cantilever beam CB (clamped on the excitation table

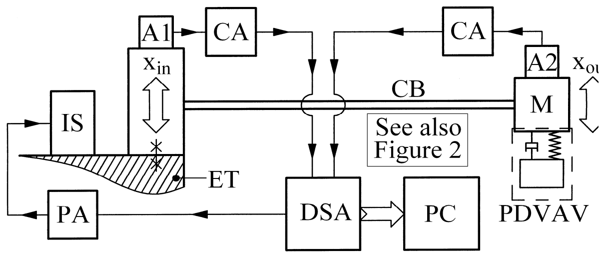

Fig. 3. The summary of the experimental set-up.

ET) has a mass M at the free end. Also on the CB free end is clamped the PDVAV (here sketched as principle).

A dynamic signal analyzer DSA (HP 35670A) is used to describe the dynamics of the mechanical system with and without PDVAV. Especially here we are interested in the transmissibility evolution (the ratio between the $\mathbf{A_{out}}$ amplitude of output motion $\mathbf{x_{out}}$ and $\mathbf{A_{in}}$ amplitude of input motion $\mathbf{x_{in}}$) versus the frequency of excitation. The DSA drive the inertial shaker IS (via a power amplifier PA), so an excitation motion $\mathbf{x_{in}}$ with variable frequency is produced. Two accelerometers (A1 and A2) are used to acquire the input and output motion acceleration $\mathbf{d^2x_{in}/dt^2}$ and $\mathbf{d^2x_{out}/dt^2}$ respectively (via a charge amplifier CA for each one). The transmissibility for the first bending mode in different circumstances (different values of CB length and different resonance frequency values as well, with and without PDVAV) is delivered by DSA to a personal computer PC, used for data processing (according to Figure 4).

5. SOME EXPERIMENTAL RESULTS

Figure 4 presents some results on the transmissibility **T** evolution versus excitation frequency **v**, with the magnitude in

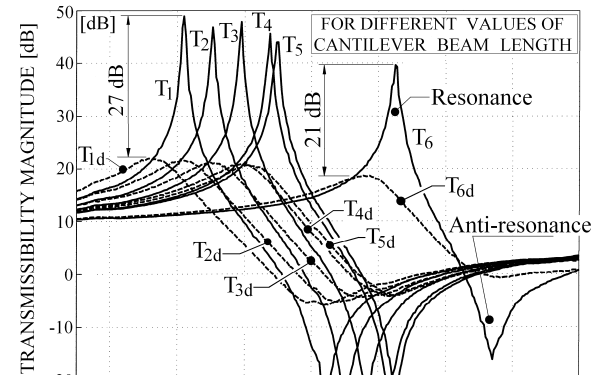

Fig. 4. The transmissibility evolution (with PDVAV – dashed lines, without PDVAV – continuous lines).

decibels (dB) described by:

$$T(v) = 20 \cdot \log(A_{out}/A_{in}) \quad (1)$$

There are two types of curves. The continuous curves (T_1, T_2….T_6) describe the transmissibility without absorber. The dashed curves (T_{1d}, T_{2d}….T_{6d}) describe the transmissibility in the same conditions, with PDVAV. The maximum attenuation (and PDVAV efficiency as well) is around 27 dB (22.3 LM, linear magnitude), for the resonance peak on 91 Hz frequency, see T_1 and T_{1d} curves. The resonance peak on T_{1d} is moved to the left because the absorber increases the mass M, so it decreases the resonance frequency. As is clearly indicated on Figure 4, the absorber still gives a good attenuation (bigger than 20 dB so 10 LM) even if the modal frequency is increased (the length of CB is decreased) up to 122 Hz, see T_6 and T_{6d} curves. Some experiments prove that the absorber keeps the same efficiency if the modal frequency is reduced even at 60 Hz. So the absorber has a bandwidth frequency of 60 Hz around the 91 Hz, the frequency of maximum efficiency .

On Figure 4 the positive peaks describes the poles (resonance), the negative peaks describes the zeros (anti-resonance) of the transmissibility between A2 and A1 accelerometers. On the frequency of a negative peak, $\mathbf{A_{out}} \ll \mathbf{A_{in}}$, the CB works as a vibration isolation system. The PDVAV decreases the transmissibility in poles and increases it in zeros. The absorber works as a moving average filter for the mechanical transmissibility.

6. CONCLUSION

The passive dynamic vibration absorber with visco-elastic polymer (PDVAV) is capable to reduce significantly the resonating amplification in mechanical structures. It works with high efficiency on a wideband of frequency. There are plenty of possibilities to use it, taken into account basically three conditions. Firstly, the PDVAV should be placed on the structure where the vibration amplitude is maximal. Secondly the attenuation at resonance (and the stiffness of mechanical suspension as well) is given by the volume of the visco-elastic polymer. Thirdly the coresonance condition is assured by tuning the PDVAV's mass. Nevertheless we must mention that there are two negative influences against the visco-elastic polymer used in PDVAV, the temperature variation and ageing of material.

7. ACKNOWLEDGEMENTS

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