Universal absorber applied to NVH in EV's powertrain

Calin ITU[1,a)], Sorin VLASE[1,b)], Maria Luminita SCUTARU[1,c)], Consuella Sofia PENA[2,d)], Paul Nicolae BORZA[1,e)], Mircea MIHĂLCICĂ[1,f)]

[1] TRANSILVANIA University of Brașov, Romania, [2] Extremadura University,Badahoz, Spain

[a)] calinitu@unitbv.ro; [b)] svlase@unitbv.ro; [c)] lscutaru@unitbv.ro; [d)] cgragera@unex.es; [e)] borzapn@gmail.com; [f)] mihmi_1@yahoo.com;

Abstract. Inertial dampers ensure, through the structural design, mechanical energy dissipation due to vibration or noise generating source. Also these dampers allow, as result of vibration energy dissipation in volume of plastics on the one hand and as result of dissipative higher power efficiency and stability on the other hand (due to the high energy absorption distributed system) the use of corresponding nonlinear zone flexibility based plastics. Moreover, the spatial distribution of inertial masses ensure a more uniform thermal loading plastic base and can be achieved without breaking conditions thereby avoiding pregnancy and the development of concentrators zones in basic plastic. The same spatial distribution in conjunction with inertial masses form, their surface roughness and especially the flexibility of plastics allow obtaining anisotropic features extremely useful. In this paper a study of a such type of damper is made.

Introduction

In [1] are shown the theoretical equations of motion corresponding inertial mass immersed in basic plastic. In the paper [2] such dampers are included in the class of 'meta-materials', mechanical hybrid structures respectively (composite) that allow "predicting the mechanical properties of the design phase" and damping properties in relation to specific needs in system which the damper is included. Such systems are found in the living world that can fulfill both dissipation function but also amplification and therefore accelerations of vibration transmitted through the damper. First, lower, corresponding to the fundamental frequency applied to the electric motor that is dependent on variable angular speed. The second frequency range of vibration is due to electrical switchgear and circuit characteristics and correspond to higher frequencies such as those produced by electrical switching circuit. This second field will vary very little in relation to the angular speed of the engine or the velocity of the vehicle. This second area is intended to be covered by the damping characteristics (band pass filter) that can perform inertial damper. Alternative solutions, involving the development of an absorber of oscillations in electric commutation type Wiener filter [14], which both in terms of reliability, energy consumption, especially high computing power necessary to implement satisfy only partially the needs of applications involving a rotating as uniform for electric vehicle drive system. Furthermore, if for some reason practically disappears the power control system of the torsional vibration, disappears too the possibility to decrease the vibrations.

Mechanical model and analysis scenario

In Fig. 1 a standard model of an inertial damper is presented. The damper is made by rubber reinforced with balls. In the paper two cases are considered: the balls are made by steel and, alternately, by aluminum (Al). The results are similar, which is why we only present the case where steel balls are. The properties for rubber and steel are those currently used in engineering. A finite element model for the damper is presented in Fig.2. The main goal of the paper is to determine the eigenvalues for a functional structure and for the damaged structure. Making an analysis in the eigenfrequencies changes is possible to identify the moment when the damage is produced. The method was presented in previous paper as [3],[4],[7]. Analytical methods in the field of the eigenvalues can be found in many paper (see [5],[6],[8]-[14]).

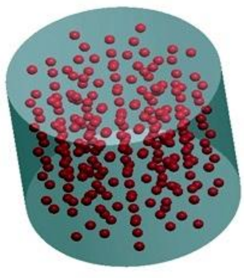

Fig. 1. Physical model of the inertial damper

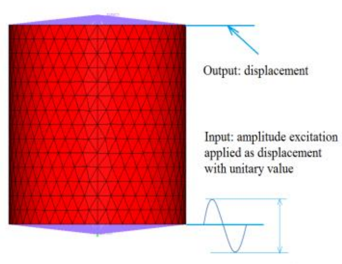

Fig. 2. Finite element model of the damper

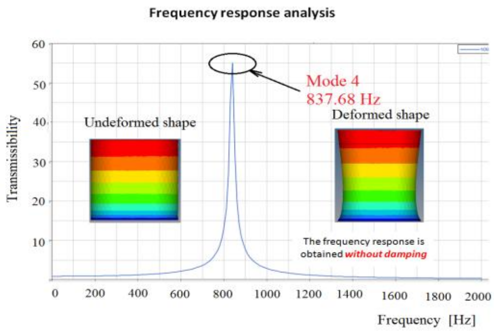

Fig. 3. Frequency response for a rubber model

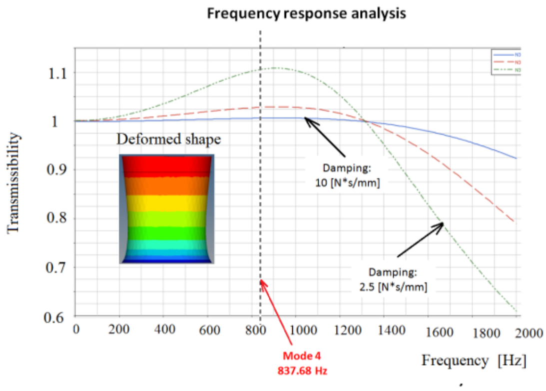

Fig. 4. Frequency reponse for an inertial damper with steel balls

Eigenvalues for the damaged damper

The eigenvalues and eigenmodes were computed using a finite element soft. The result for the eigenvalues computed for the scenarious are presented in Fig. 5. Symmetric modes gives equal eigenvalues [15]- [18] but a crack in the material makes one of the eigenmodes to keep his value while the other will change. That is why it is necessary to consider various eigenmodes each time to make comparison. Some eigenvalues, despite the emergence of crack and damage of the damper, do not change. If we analyze one of the cases, for example, the last case (version 10 – Fig.5) you can see that the eigenmode 7, for example, is not changed although the case study shows a consistent fissure. In contrast to most other eigenvalues are severely affected. It will result so that some eigenvalues are not altered by a change in material continuity while others give some consistent differences. It follows therefore that for detecting a beginning of damage is necessary to consider several vibration modes to be comparable to the initial situation.

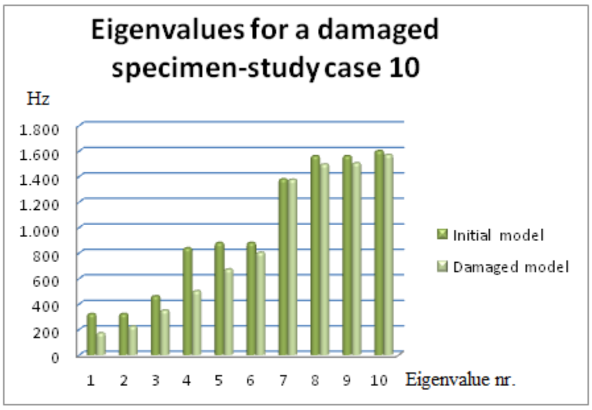

Fig. 5. The first 10 eigenvalues for the scenario nr. 10 (initial and damaged model)

When we use balls made by aluminum the results are presented in Table 2. The conclusions analyzing the values presented in Table 2 offer similar conclusions as for the values presented in Table 2, from a qualitative point of view. The density of aluminum being smaller as the density for steel, the quantitative results differ.

Conclusions

In the operation of a vibration damper, insert mode of the balls inertial in the mass damper rubber makes it vulnerable to damage. It is, of course, be very useful to detect early failures of this damper before damage. It can thus avoid significant material expenses. A periodic visual inspection requires expenses and waste of time. For this reason it can be very useful to find a method of early detection of cracks in the rubber body. In this paper we propose the use of their eigenfrequencies damper measurement and comparison to baseline. If there are significant differences the damper may be replaced. As seen in the presented figures, some eigenfrequencies can differ more or less, depending on the place and dimension of the fissure. For this reason it should be analyzed and compared multiple frequencies simultaneously. We must not confine ourselves to the study of a single frequency of failure because there may be cases where the eigenfrequency corresponding to a particular mode of vibration, not be influenced by the mechanism of destruction, but other frequencies are affected. The analysis was carried out for a single shock absorber, attached at one end. In practice it may be more useful to consider the damper mounted in the mechanical system. This arrangement makes its eigenfrequencies of the damper to change. They can be determined from a more complex calculus but the findings of this study do not change. From the results presented it can be concluded that a change of eigenfrequency more than 5% can pull the alarm on the possibility of cracks in the structure while a difference of more than 10% should lead to a revision of the whole assembly to determine the cause of this change.

References

1. Carotti,A., Turci,E., A tuning criterion for the inertial tuned damper. Design using phasors in the Argand–Gauss plane. Applied Mathematical Modelling, 23(3), 199–217, 1999.
2. Claeys,C. C., Pluymers,B., Sas,P., Desmet,W., Design of a resonant metamaterial based acoustic enclosure, Proceedings of ISMA2014, 3351–3358, 2014.
3. Gillich,G.-R., Praisach,Z.-I., Iavornic,C.M., Reliable method to detect and assess damages in beams based on frequency changes. ASME 2012 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference. pp. 129-137, 2012.
4. Gillich,G.-R., Praisach,Z.-I., Damage-patterns-based method to locate discontinuities in beams. SPIE Smart Structures and Materials. 869532-8, 2013.
5. Neubert, V.H., Design of Damping and Control Matrices by Modification of Eigenvalues and Eigenvectors, Shock and Vibration, Volume 1, Issue 4, Pages 317-329, 1994.
6. Nobari, A.S., Ewins, D.J., On the Effectiveness of Using Only Eigenvalues in Structural Model Updating Problems, Shock and Vibration, Volume 1, Issue 4, Pages 339-348, 1994.
7. Praisach,Z.-I., Gillich,G.-R., Birdeanu,D.E., Considerations on natural frequency changes in damaged cantilever beams using FEM. International Conference on Engineering Mechanics, Structures, Engineering Geology, International Conference on Geography and Geology-Proceedings. pp. 214-219, 2010.
8. Vlase, S., A Method of Eliminating Lagrangian-Multipliers from the Equations of Motion of Interconnected Mechanical Systems, Journal of Applied Mechanics – Transaction of ASME, Volume: 54 Issue: 1 Pages: 235-237, 1987.
9. Vlase, S., Dynamical Response of a Multibody System with Flexible Elements with a General Three-Dimensional Motion, Romanian Journal of Physics, Volume: 57 Issue: 3-Pages: 676-693, 2012.
10. Vlase, S.; Teodorescu, P. P., Elasto-Dynamics of a Solid with a General "Rigid" Motion using FEM Model. Part I., Theoretical Approach, Romanian Journal of Physics, Voume: 58, Issue: 7-10, Pages: 872-881, 2013.
11. Scutaru, M.L., Vlase, S., Some Properties of Motion Equations Describing the Nonlinear Dynamical Response of a Multibody System with Flexible Elements. Journal of Applied Mathematics, Article Number: 628503, 2012.
12. Vlase, S., Eigenvalues and Eigenmodes of an Inclined Homogeneous Truss in a Rotational Field. Romanian Journal of Physics, Volume: 59, Issue: 7-8, Pages: 699-714, 2014.)
13. Modrea, A.; Vlase, S.; Teodorescu-Draghicescu, H.; et al., Properties of advanced new materials used in automotive engineering. Optoelectronic and Advanced Materials – Rapid Communications, Vol.7, Issue 5-6, pp. 452-455, 2013.
14. Shimkin,N., Estimation and Identification in Dynamical Systems. Israel Institute of Technology, Department of Electrical Engineering, Lecture Notes, 2009, Chapter 3, Fall 2009.

15. Vlase,S., Danasel,C., Scutaru,M.L., Mihalcica,M., Finite Element Analysis of a Two- Dimensional Linear Elastic Systems with a Plane "Rigid Motion", Romanian Journal of Physics; 59(5-6), 2014.
16. Vlase,S., Teodorescu,P.P., Itu,C., Scutaru,M.L., Elasto-Dynamics of a Solid with a General "Rigid" Motion Using FEM Model. Part II. Analysis of a Double Cardan Joint. Romanian Journal of Physics, 58 (7-8), 2013.
17. Vlase, S., A Method of Eliminating Lagrangian-Multipliers from the Equation of Motion of Interconnected Mechanical Systems. Journal of Applied Mechanics-Transaction of the ASME, vol. 54, Issue:1, pp.235-237,1987.
18. Vlase,S., Munteanu,M.V., Scutaru, M.L., On the Topological Description of the Multibody Systems. Annals of DAAAM, Book Series, pp.1493-1494, 2008.