Meccanica (2012) 47:1035–1041
DOI 10.1007/s11012-011-9454-9

BRIEF NOTES AND DISCUSSIONS

Experimental evaluation of modal damping in automotive components with different constraint conditions

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Received: 4 January 2010 / Accepted: 16 June 2011 / Published online: 12 July 2011

Abstract The selection of reliable damping values for mechanical components of complex shape is very difficult from a theoretical point of view, above all in the case of complex constraint conditions. This work presents an example of experimental procedure for estimating the modal damping in a family of mechanical components. An Experimental Modal Analysis (EMA) program has been carried out on different types of brackets used on diesel engines, and in different constraint conditions with the aim of collecting the damping values in a database. In particular, the EMA has been carried out on brackets in the freely-supported condition and in the clamped condition actually used in the engine, highlighting the variations in modal damping with the different constraint conditions. The results of the experimental measurements have been processed with different modal analysis algorithms in order to increase the robustness of the solution. The resulting damping plots represent very useful data in the field of numerical analysis.

Keywords Modal analysis · Modal damping · Experimental techniques · Vibration analysis

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1 Introduction

The identification of damping is essential for the computation of stress and strains in vibrating structures, since the actual vibration amplitude is strongly influenced by the damping of the structure being tested. This parameter is often obtained from Frequency Response Function (FRF) measurements processed using curve fitting techniques [1, 2] or via theoretical approaches. Unlikely, in the virtual prototyping phase, the first hard prototype is not obviously available. It is therefore necessary to look for suitable damping values in the literature, but they are not generally available for complex mechanical components. This is the case considered in this paper, concerning brackets used in diesel engines. Moreover, modal damping, as well as the natural frequencies and mode shapes, depends not only on the materials and the geometrical shape, but also on how the bracket is clamped to the surrounding structure (usually the engine block). Therefore, it is useful to collect a database containing modal damping values for different types of brackets in different kinds of constraint conditions. This is necessary in the virtual prototyping phase to analyse the dynamic behaviour of new similar brackets for which the hard prototype is not available yet.

This work presents an example of experimental procedure for estimating the modal damping of a family of mechanical components. An experimental modal analysis (EMA) campaign has been carried out on different types of brackets with different constraint conditions (freely-supported, clamped and clamped with accessories condition) with the aim of collecting the damping values in a database. These brackets are used in diesel engines to support different engine components such as gear pumps for steering systems, water pumps, fuel pumps, oil pumps, gas turbines, alternators. The results of the experimental measurements have been processed with different modal analysis algorithms in order to increase the robustness of the solution. The resulting damping plots represent very useful data for further simulation analyses. These damping values have been analyzed as a function of natural frequency, considering the influence of different bracket shapes, weight and constraint conditions; the data uncertainty has been quantify. As this author is aware, this study is not present in the literature.

The structure of the paper is as follows: Sect. 2—description of the experimental setup; Sect. 3—illustration and discussion of the results of the modal analyses in terms of natural frequencies and modal damping; Sect. 4—concluding remarks.

2 Experimental setup

For modal testing, it is necessary to measure the FRFs. For this reason an impact hammer (PCB 068C04) has been used to excite the different measurement points of the brackets and PCB piezoelectric accelerometers (frequency range 1 to 10000 Hz) have been mounted on the brackets in order to measure the responses. The measurement locations were chosen in order to give an adequate spatial resolution to describe the global structural mode shapes. Measurements were acquired in three orthogonal directions in order to estimate the mode shapes in the 3D space. The procedure used to perform the EMA is the conventional procedure in which both excitation and response are measured simultaneously to obtain the *Inertance*, i.e. the FRF between acceleration and force. In particular, the response points were maintained fixed during the tests, while the excitation moved from one measurement point to another in order to obtain the FRFs among all the considered points. The signals were acquired by using sampling frequency and frequency resolution according with the type of brackets and with the kind of constraint condition. An exponential window for the accelerometer signals and a force window for the force signals are used in order to reduce leakage. The input autopower-spectra, output autopower-spectra and cross-power-spectra are evaluated and stored for each measurement location. Furthermore, the FRFs are calculated by using the H_1 estimator [1]. During the tests, the coherence function [1] is monitored as an on-line check of data quality.

The brackets under test are identified with capital letters A, B, C and D. Bracket A is used in marine diesel engines to support the water pump for the cooling system; bracket B and bracket D are used in automotive engines to support the pump for the steering system, and finally bracket C is used in automotive engines to support the alternator, fuel pump and water pump. The bracket material is aluminium and the weights of the brackets and accessories are given in Table 1. The total mass of the additional accessories of the brackets is 5 kg (water pump) for bracket A, 4.2 kg (pump for steering system) for bracket B and D, 19 kg (water pump, fuel pump, alternator, two bearings and one tension pulley) for bracket C (see also Table 1). As said, the brackets were tested in three different kinds of constraint conditions. The freely-supported condition was approximated by suspending the brackets using soft bungee cords. The natural resonances of the bungee cords are much lower than the lowest natural frequencies of the brackets, so let assume that the modes of the bungee cords do not influence the analysis of the brackets. The brackets on the clamped conditions without accessories were analysed by screwing them to their engine block as in the actual condition. Finally, in the last scenario the accessories were fixed upon the brackets, clamped to the engine block. In addition, photos of the brackets in the different constraint conditions are shown in Tables 2 to 5. The different brackets were fixed on different engine blocks, 6 or 4 cylinder engines, as in the actual configuration. An EMA of these two engine blocks was also performed in order to estimate their modal

Table 1 Weights of brackets and relative accessories

	Bracket A	Bracket B	Bracket C	Bracket D
Bracket weight [kg]	0.91	0.78	4.50	0.95
Accessories' weight [kg]	5	4.2	19	4.2
Ratio bracket to accessories' weight	0.182	0.186	0.237	0.226

properties. Such an EMA is important for the evaluation of the effective mode shapes of the brackets. As a matter of fact, some resonance frequencies of the brackets in clamped conditions were observed as being close to natural frequencies of the engine block; moreover, the corresponding mode shapes of the brackets do not show deformation but only rigid displacement in all the measurement points. Therefore, these modes are not own modes of the brackets since they include the only deformation of the engine block at which the brackets are fastened. Furthermore, it has to be underlined that we are interested in evaluating the modal damping of brackets for the analysis of new prototypes; thus, the data which are strongly influenced by the modal properties of a specific engine block are not to be considered. For this reason, in the following section, the above-mentioned modes will be neglected because they do not deal with the brackets, but they are due to engine block dynamics.

3 Results and discussion

Once the experimental modal tests and analyses of the brackets have been performed, natural frequencies, modal damping and mode shapes are available for all modes in the frequency band of analysis. Tables 2 to 5 show the natural frequencies (f_n) and modal damping (ζ) obtained by averaging the values coming from the Least Square Complex Exponential (LSCE) [1, 2] method and PolyMAX method [3], the LSCE method that works in the time domain and the frequency domain algorithm PolyMAX. The results of the analyses show that small differences in natural frequencies and modal damping occur in the same modal test by using the two methods. Thus, the Modal Assurance Criterion (MAC)—a technique to determine the degree of correlation between the mode shapes [1, 2]—has been applied to the corresponding eigenvectors, estimated by LSCE and PolyMAX: the MAC values are close to one, indicating similar mode shapes. Therefore, it is correct to use mean values of f_n and ζ for the evaluation of the natural frequencies and modal damping in all the constraint conditions being tested. In general, it was observed that the first mode shape of the brackets in clamped condition and clamped condition with accessories is clearly a bending mode. Figure 1 depicts the mean modal damping (ζ) as a function of frequency, obtained by averaging the modal damping estimated by LSCE and PolyMAX methods, for the four brackets in all the tested constraint conditions. The results obtained in freely-supported conditions show values of ζ between 0.06 and 0.71% in the frequency range from 470 to 5000 Hz; the results in clamped conditions show values of ζ between 0.1 and 1.45% in the frequency range from 170 to 1600 Hz; finally, the results carried out in clamped conditions with accessories show values of ζ between 0.2 and 3.4% in the frequency range from 17 to 1000 Hz. It is interested

Table 2 Results about bracket A

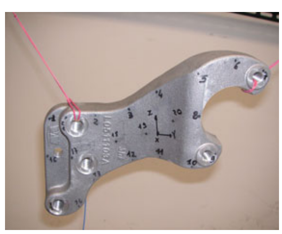
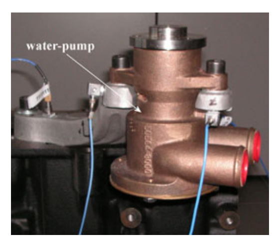
	Freely-supported		Clamped		Clamped with accessory	
Bracket A						
	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]
Modal frequencies and damping (mean value of LSCE and PolyMAX methods)	1452	0.50	510	1.45	157	1.11
	2551	0.09	1258	0.67	209	0.82
	2965	0.13	1577	0.40	414	2.31
	3921	0.08			895	1.46
	5003	0.18			979	0.59

Table 3 Results about bracket B

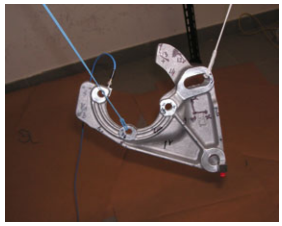
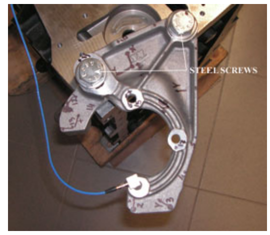
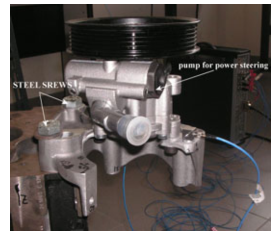
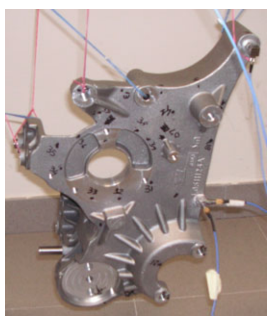
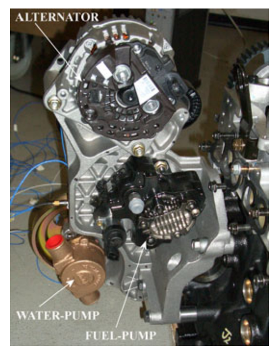
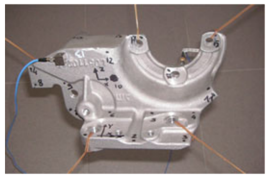
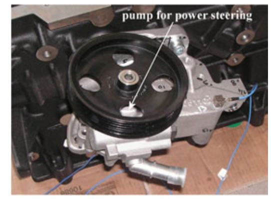
	Freely-supported		Clamped		Clamped with accessory	
Bracket B						
	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]
Frequency and damping (mean value of LSCE and PolyMAX methods)	1043	0.32	312	0.59	119	1.54
	1218	0.51	511	1.00	409	2.05
	2194	0.37	853	0.60	634	1.48
	2994	0.17	1134	0.58	898	3.16
	3088	0.39				

Table 4 Results about bracket C

	Freely-supported		Clamped		Clamped with accessories	
Bracket C						
	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]
Modal frequencies and damping (mean value of LSCE and Poly-MAX methods)	472	0.06	171	0.35	17	1.62
	735	0.10	439	1.00	46	2.77
	1151	0.09	565	1.02	68	1.94
	1235	0.13	791	1.07	72	1.92
			1116	0.09	141	1.18
					158	1.80
					194	1.09

to note that brackets with the same constraint conditions show similar values of modal damping, even if the brackets themselves have different shapes. In particular, in the freely-supported scenario, the averaged modal damping concerning all the brackets is $\bar{\zeta} \cong 0.27\%$ with standard deviation of 0.19%, while in clamped constraint conditions is $\bar{\zeta} \cong 0.60\%$ with standard deviation of 0.38%. For the brackets with accessories the range is more extensive with $\bar{\zeta} \cong 1.55\%$ and standard deviation of 0.71%. Some difficulties arose

Table 5 Results about bracket D

	Freely-supported		Clamped		Clamped with accessory	
Bracket D						
	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]
Modal frequencies and damping (mean value of LSCE and Poly-MAX methods)	1422	0.60	495	0.34	19.5	3.41
	2077	0.53	847	0.22	138	0.61
	2798	0.45	1055	0.30	152	0.61
	2857	0.25	1152	0.37	243	1.77
	4472	0.69	1252	0.17	290	2.55
	4686	0.20	1380	0.27	562	1.47
			1475	0.26	845	0.16
					923	0.26
					972	1.14

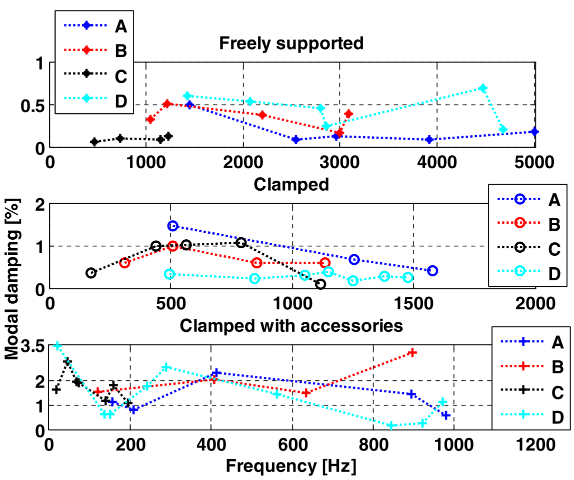


Fig. 1 Modal damping [%] as a function of frequency for the four brackets (A, B, C, D) in all the tested constraint conditions

during these last tests due to the complexity and non-linearity of the structures. In general, the more complex the structure is, the more uncertain the modal damping estimation becomes. Furthermore, the ratios between the brackets and accessories' weights are very low, close to 0.2 for the entire bracket set under investigation (see Table 1); this could produce different damping, leading to high standard deviation values. In particular, the standard deviation is about an half of the mean modal damping $\bar{\zeta}$; such a result can be considered quite unsatisfactory, however when any literature data or hard prototypes do not exist, these mean values of modal damping ($\bar{\zeta}$) are essential. Moreover, it is interesting to calculate the ratio between the standard deviation and the averaged modal damping (namely relative spread) for the three different constraint conditions: it is 0.7, 0.63 and 0.45 for the freely-supported, clamped and clamped with accessories conditions, respectively. Therefore, although the standard deviation, which is an absolute range, is the largest (0.71%) for the clamped bracket with accessories, the relative spread, is the lowest (0.45), leading to conclude that the averaged modal damping for the clamped conditions with accessories is a rather effective and representative damping value. Figure 2 shows the values of the modal damping for the four brackets in the same constraint condition and their interpolation curves. In particular, the interpolation curves are obtained by using a 3rd order polynomial for the modal damping values regarding the freely supported and clamped condition. Due to the high standard deviation for the modal damping values regarding the clamped with accessories condition, the interpolation curve is not calculated but in this case the averaged value, equal to

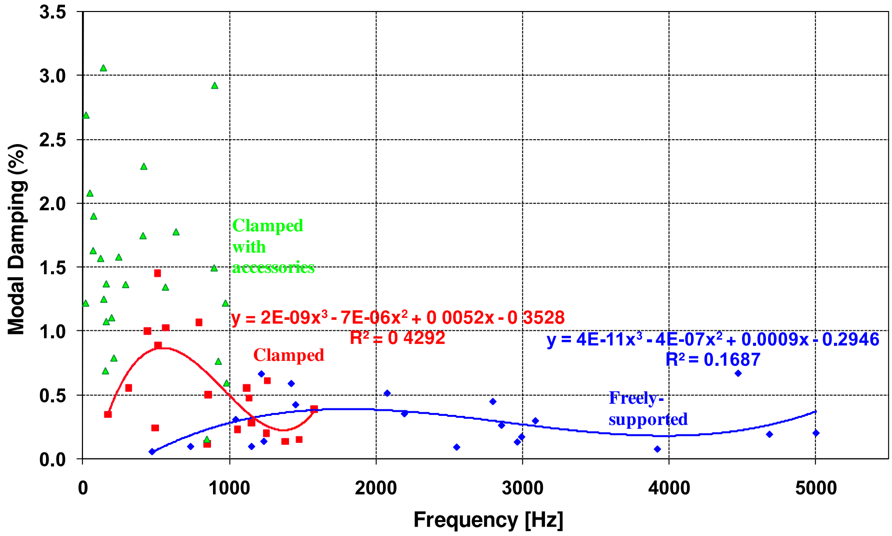


Fig. 2 Interpolation curves of the modal damping values obtained for all the brackets in the three tested constraint conditions

1.55%, seems adequate. Furthermore, Fig. 2 reports the equation of the interpolation curves as well as the correlation coefficients (R^2) as a proof of the interpolation quality. The correlation coefficients are 0.1687 and 0.4292 for the freely supported and clamped condition, respectively. These values could be improved increasing the order of the interpolation polynomial, however a 3rd order polynomial seems an effective solution between the complexity of the curve equation, the correlation coefficient values and the field of application. Such interpolation curves together with the averaged modal damping values can give very useful information for the dynamic analysis (e.g. finite element analysis) of new prototypes for which the hard prototype is not available yet, since the analyst can catch the needed value as a function of frequency.

4 Concluding remarks

This work presents an example of experimental procedure for estimating the modal damping of a family of mechanical components. In particular, an experimental modal analysis campaign has been carried out on different typologies of brackets of diesel engines with different constraint conditions. Tests were performed in order to collect the damping values in a database, useful in the virtual prototyping phase for analysing the dynamic behaviour of new similar brackets for which the actual prototype is not available yet. These damping values have been analyzed as a function of natural frequency, considering the influence of different bracket shapes, weight and constraint conditions; the data uncertainty has been quantify. As this author is aware, this study is not present in the literature. It has to be underlined that the data which are strongly influenced by the modal properties of a specific engine block have been neglected, since they do not deal with the brackets, but they are due to engine block dynamics.

The results show that the modal damping is very sensitive to the constraint condition: brackets with the same constraint conditions show similar values of modal damping, even if the brackets themselves have different shapes. The mean modal damping found are: 0.27% for freely-supported condition, 0.60% for clamped condition and 1.55% for clamped condition with accessories. These values together with the interpolation curves can be very useful data for the dynamic analysis of similar brackets for which no test data are available.

Acknowledgements The author wishes to thank VM Motori S.p.A (Cento, Ferrara, Italy) and their engineers for co-operation and assistance in the collection of experimental data. This work has been developed within the Laboratory of Advanced Mechanics (MechLav) of Ferrara Technopole, realized through the contribution of Regione Emilia Romagna—Assessorato Attivita' Produttive, Sviluppo Economico, Piano telematico—POR-FESR 2007-2013, Attività I.1.1.

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