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ELASTICITY AND DAMPING OF AVM-2055 RUBBER MOUNTING PADS: THEORETICAL DETERMINATION AND EXPERIMENTAL VERIFICATION

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Abstract: *Critical machine components are monitored in order to evaluate their condition and decide further actions in industry. Vibration measurements are affected from external stimulus, depending on working environment and working conditions. Machinery mounting, aims to isolate a higher proportion than 80 % from external stimulus. In this study, dynamic analysis determinate the signal transportation from environment to machinery and evaluate the theoretically calculated modulus of elasticity and damping of AVM-2055 rubber mounting. The experimental device that simulates rotating machinery, modelled with finite elements method (FEM) in order to analyze its vibration behaviour, under external stimulus. Comparing the theoretical values of natural frequencies with the experimental ones validates the FEM model and the theoretically calculated values of material properties.*

Keywords: *FEM, modal analysis, machine mounting, damping, spring constant, FFT, rubber mounting*

1. INTRODUCTION

In industry, bearings are the most fundamental components of all rotating elements of the machinery. A percentage of 40 – 50 of engine failures due to faults of bearings [1]. The fault diagnosis of rolling element bearings is very important for improving mechanical system reliability and performance. For example in the case of large induction motors, bearing faults can account for up to 44 % present of the total number of failures [2].

Bearings lifecycle can be calculated [3], however not every bearing achieves that useful life. External stimuli have a major impact on bearings and other critical machine components lifecycle. Therefore machinery mounting is important not only to insure production quality but for extension of useful life.

For isolating machines from external stimuli, technical characteristics of mounting material must be selected in order to keep vibration amplitude as small as possible [4].

In condition monitoring as in any diagnostic methodology sensors are used to record various signals from the machine but these signals cannot be directly used because such signals are often very large in size and noisy [5]. Signal processing techniques are hence used to derive useful and compact information about the system from these measurements. These information packets are called features and the process is called feature extraction [6-9].

The same techniques can be used to evaluation mounting in any machinery. Advanced signal processing techniques, such as Hilbert-Huang Transform and Zhao-Atlas Marks Distributions have been used [10] and compared [11]. The results have shown that efficient of mounting to absorb external stimuli can be evaluate through time-dependent amplitudes and instantaneous frequencies.

In this study, AVM-2055 rubber mounting of hardness 65 Shore A has been tested, theoretical and experimental for the ability to absorb external stimulus and for verification of the modulus of elasticity and damping of the material. Natural Rubber/Styrene Butadiene Rubber (NR/SBR) produced from the latex of the Hevea brasiliensis tree, presented from Gent [12] and compared in the scope of hardness from Fediuc et al. [13]. A finite elements method (FEM) model of the experimental device that simulates rotating machinery, in order to analyze its vibration behaviour has been created. The theoretically calculated values of material properties [14,15], are evaluated. Comparison of the theoretical values of natural frequencies with the experimental ones validates the FEM model and the passive transmittance coefficient of mounting material.

2. THEORETICAL DETERMINATION

Material selection for elastomeric vibration isolators' depends on: ultimate loading capacity, sensitivity to environment and internal properties. Additionally modulus of elasticity and internal damping has major impact.

2.1 The modulus of elasticity

The most important factor for the computational treatment of Rubber springs is the shear modulus (shear modulus) G. It is not dependent by the construction form of rubber springs, but only of the material rubber. The shear modulus is a function of the Shore hardness of each rubber.

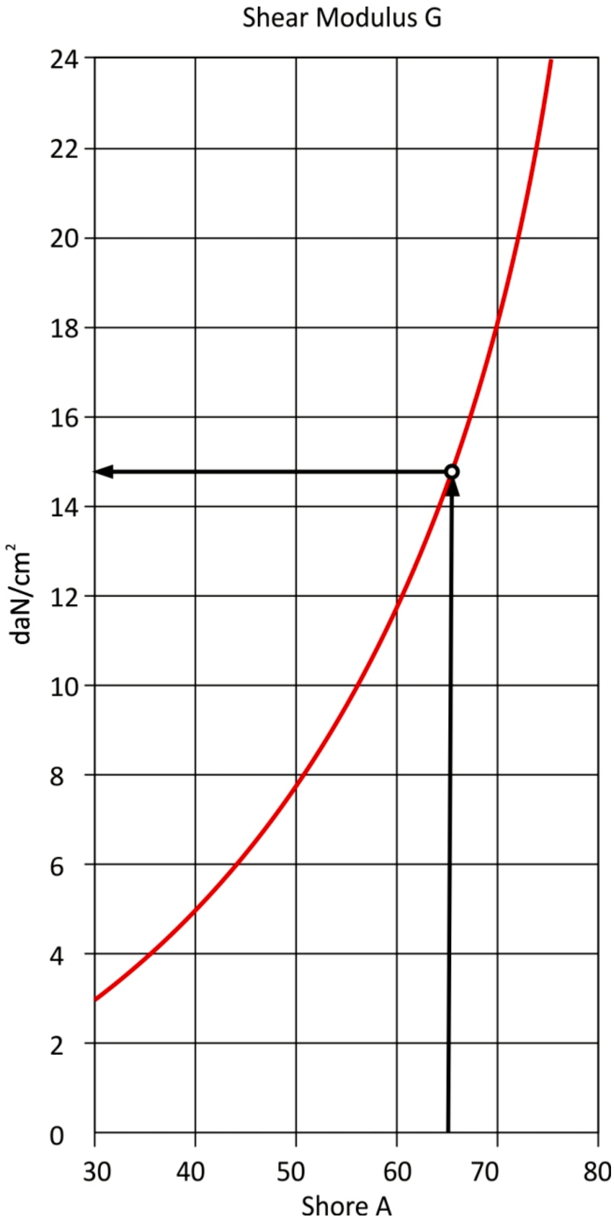

Figure 1. Relationship between hardness and shear modulus

Known from the theory of elasticity is the relationship between the elastic modulus E and the shear modulus G:

$$G = \frac{E}{2(1+\frac{1}{-\mu})}\,. \tag{1}$$

where μ is Poisson's ratio.

Rubber as a volume - elastic and fully incompressible material has a Poisson's ratio µ = – 2, resulting

$$E = 3G\,. \tag{2}$$

This relationship is useless for the calculation of rubber springs, because pressure loaded elastomer springs - primary field of engineering-calculation and design - in their pressure-buckling behaviour are less affected by the material hardness, than on the shape, form and the profile of the elastomer body. From the adherent connection between the metal plate and rubber body (bound rubber spring) results in an obstruction of transverse strain at the end faces and thus an uneven distribution of shear stress as factors influencing the stiffness of stressed rubber body.

For the practical determination of the pressure spring constants, use is made of a substitute bill is introduced in a form-dependent correction factor (shape factor) q, which is not a real material parameter, but it allows applying Hooke's law. The form factor is calculated from the ratio of the loaded body surface to the free surface of the elastomeric element:

$$q = \text{a loaded area / total free surface}$$

The calculation of the elastic modulus E_c can be used for simple geometric bodies with distinct surface pressure as cuboids, cylinders and hollow cylinders according to the formula

$$E_c = 3G(1+q+q^2)\,. \tag{3}$$

From the definition of the form factor follows

$$q = \frac{D}{4h}\,. \tag{4}$$

By derived from the shear modulus and the form factor replacement module E_c is calculated on the compressive stiffness k_c

$$k_c = \frac{F_c}{f_c} = \frac{AE_c}{h}\,. \tag{5}$$

Table 1 shows the theoretical calculated values of elasticity.

2.2 Damping

Damping is the dissipation of energy, usually by releasing it in the form of low-grade heat. In virtually every situation where bodies or masses are moved or accelerated there are repercussions on the system, which take the form of vibrations. These vibrations can lead to disturbances as a result of noise or movement, or to additional dynamic strain on the components. Vibration can occur in dynamic systems, i.e. systems which contain a mass or a spring. Whereas in practice the mass is usually easy to recognise, the spring is often more difficult to identify at first.

Table 1. Theoretical calculated values

65 Shore A		
G	15	daN/cm^2
Factor q	0.25	
E_c	59.0625	daN/cm^2
k_c	185.550316	daN/cm

The aim of the vibration damping or the vibration isolation is to minimise the amplitude of the vibration by using additional specially designed dampers or springs [16].

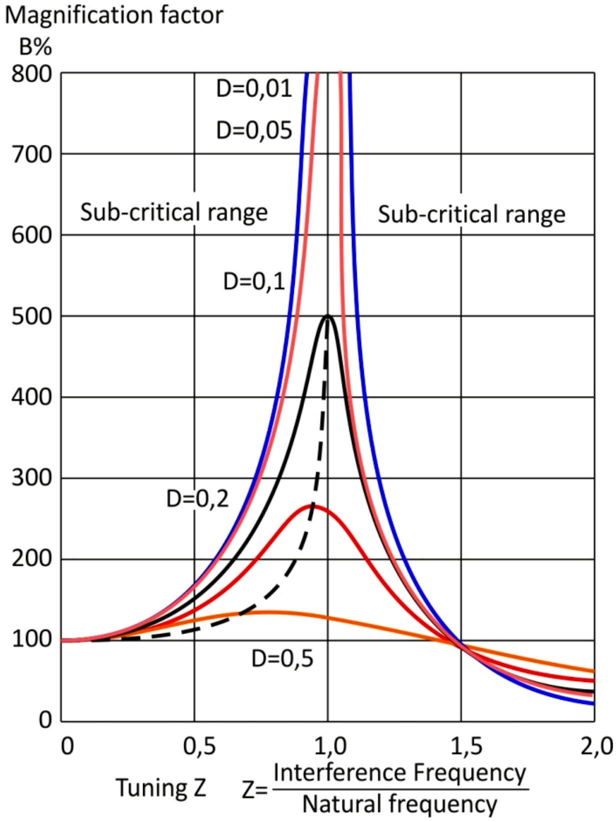

Figure 2. Steady state variation of amplitude with frequency and damping of a driven simple harmonic oscillator

Behaviour of the system depends on relative values of the two fundamental parameters, natural frequency ω_0 and damping ratio Z.

The value of the damping coefficient for current paper, depending on the data of the material is D = 0.04.

3. FEM SIMULATION

Dynamic behaviour of the mechanical system is performed with the finite element method (FEM).

Using the CAD model of the test-rig implemented in the finite element software ANSYS, the FE model is obtained. The materials of the parts and the contacts of the model are defined properly. In the FE model of the mechanism the screws, bolts and ball bearings are replaced with the proper contact conditions. Meshing process is performed by ANSYS automatically creating roughly 598810 nodes and 416308 elements. Meshed FE model and contact region of the mechanism is shown in Figure 3.

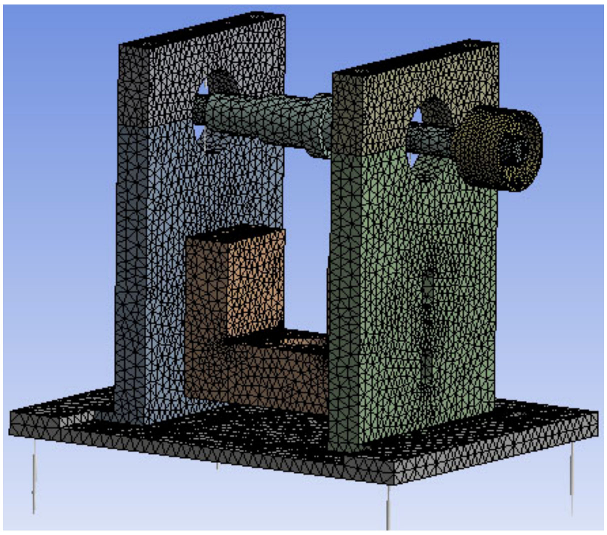

Figure 3. FE Model

4. EXPERIMENT

Test-rig is a mechanical system, shown in fig. 4, consisted of a shaft rotating on two groove ball bearings (of type KOYO 6302) driven by an electric motor controlled by an inverter for rotational speeds adjustment. An accelerometer, mounted close to the bearings, was measuring the developed accelerations. The data acquisition and the subsequent signal processing were performed using software developed on Matlab/Origin.

Force hammer is used to apply on initial vibration (input signal) on the mechanical system. Vibratory response at the mechanical system (output signal) is detected through an accelerometer/ acceleration transducer with a sensitivity of 100 mV/g (Bruel & Kjaer 4507 B) mounted on the bearing housing, while the sensing cables were kept in a free state, thereby they have little influence on the vibration test.

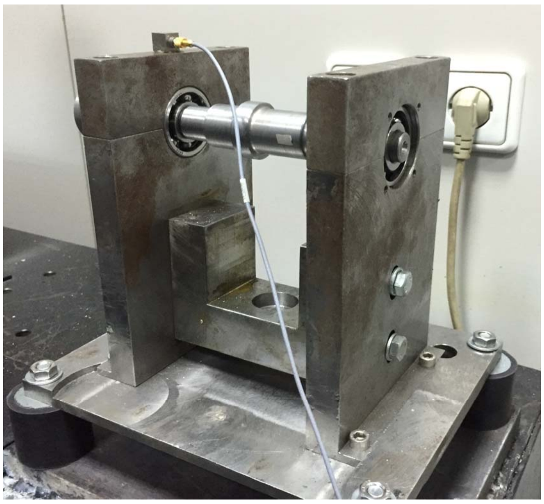

Figure 4. Test-rig

For impact excitation an acquisition and real-time analysis routine provides digital filtering for two-channel acquisition. So both force and acceleration, analogue signals are acquired by an analogue-to-digital converter (FFT analyzer PULSE Bruel & Kjaer), while the corresponding results are recorded on a computer.

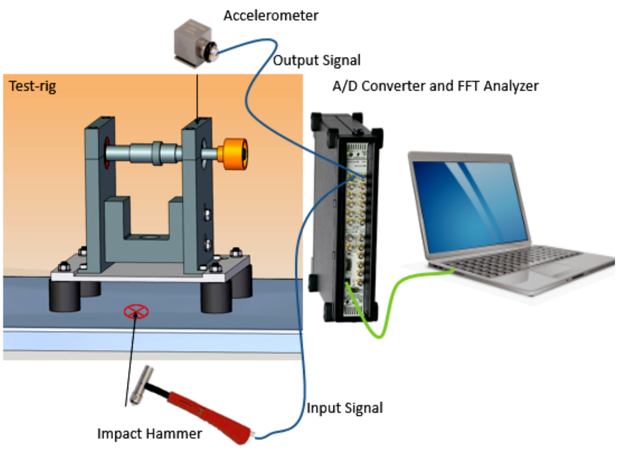

Figure 5. Experimental setup diagram

Sampling frequency of acceleration was 3200 Hz. Each measurement was repeated 10 times and the average value was obtained. Following Figure 5 shown a diagram of the experimental setup, for better understanding.

In order to obtain the mechanical system's properties from the transient vibration, the Fourier Transform of the response signals were calculated. The response can be calculated in terms of displacement, velocity or acceleration and as a result different terms have been used for the ratios of response to force.

5. RESULTS

A modal analysis is performed to determine the natural frequencies of experimental device by ANSYS software. The first five values of the natural frequencies and the corresponding mode shapes are shown in Figures 6-10.

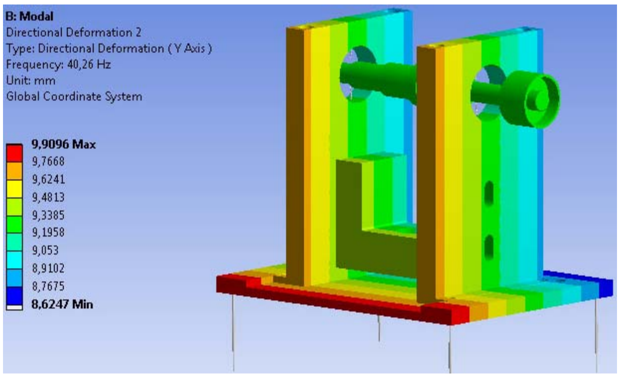

Figure 6. First mode

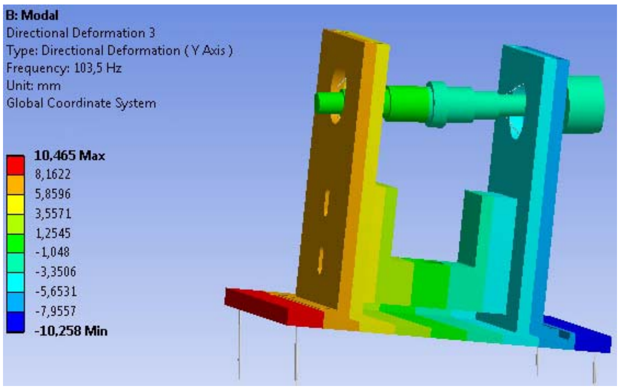

Figure 7. Second mode

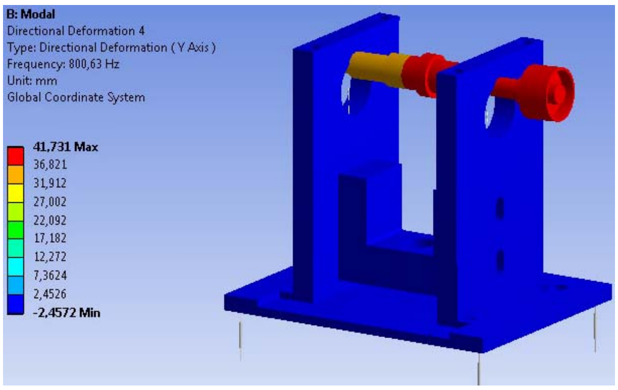

Figure 8. Third mode

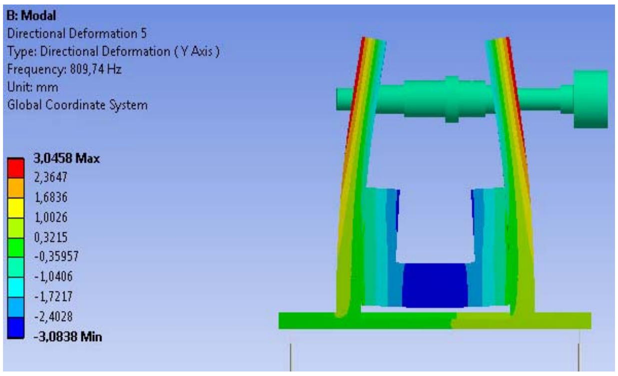

Figure 9. Fourth mode

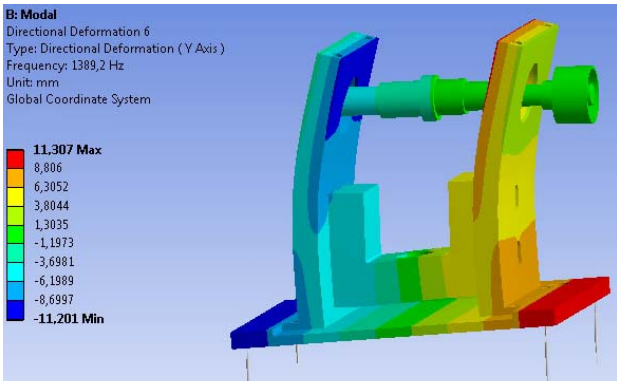

Figure 10. Fifth mode

The measured signals of the acceleration and force are subjected to the FFT analysis and furthermore the natural frequencies of the mechanism are evaluated Figures 11 and 12.

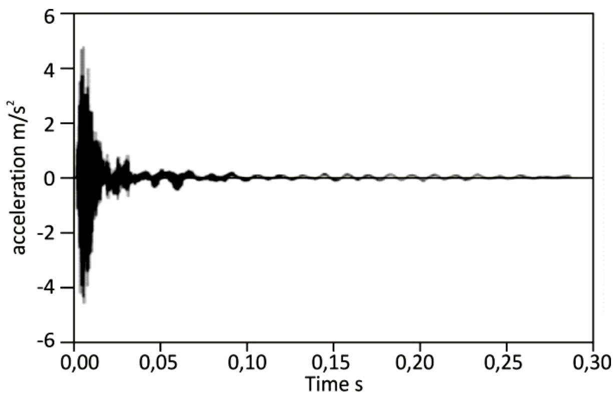

Figure 11. Signal time series

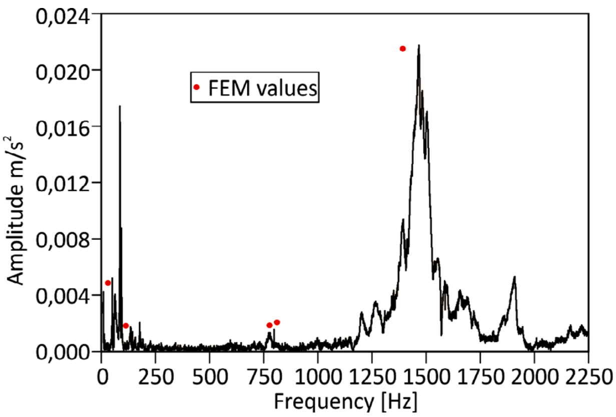

Figure 12. FFT Analysis

6. CONCLUSIONS

In the Table 2 the experimental values of the natural frequencies are inserted. The comparison of the theoretical values with the experimental ones of the natural frequencies shows a good agreement.

It can be concluded that the FE model of the prototype experimental device is satisfactory and can be used for the dynamic analysis.

Table 2. FEM and experimental values of natural frequencies

Finite Elements Method	FFT Analyser		Deviation
40.26	41.5	0.0051	2.9 %
103.5	126	0.00172	17.8 %
800.63	762	0.00131	4.8 %
809.74	780	0.00155	3.6 %
1389.2	1441	0.02118	3.5 %

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REFERENCES

[1] P.F. Albrecht, R.M. McCoy, E.L. Owen: Assessment of the reliability of motors in utility application, Energy Convers. EC-1, No. 1 1986.

[2] P. Zhang, Y. Du, T.G. Habetler, B. Lu: A survey of condition monitoring and protection methods for medium voltage induction motors, in: *IEEE Trans. Energy Convers.*, 2011, pp. 34-46.

[3] T.A. Harris, M.N. Kotzalas: *Essential Concepts of Bearing Technology*, Taylor & Francis, 2007.

[4] K.-D. Bouzakis: *Vibrations and Machine Dynamics*, ZHTH, Thessaloniki, 2010.

[5] J.I. Taylor: *The Vibration Analysis Handbook, Second, Vibration Consultants*, Inc, 2003.

[6] C. Nataraj, K. Kappaganthu: Vibration-based diagnostics of rolling element bearings: state of the art and challenges, in: *13th World Congr. Mech. Mach. Sci.*, 2011, Guanajuato, Mexico, pp. 1-10.

[7] A. Heng, S. Zhang, A.C.C. Tan, J. Mathew: Rotating machinery prognostics: State of the art, challenges and opportunities, Mech. Syst. Signal Process, Vol. 23, pp. 724-739, 2009.

[8] K.R. Mobley: *Maintenance Fundamentals*, Elsevier, 2004.

[9] M.R. Keith, J.W. Darrin, R.H. Lindley: *Maintenance Engineering Handbook*, McGraw-Hill, 2008.

[10] C. Tsiafis, Z.D. Zaharis, M. Xanthopoulou, C. Skeberis, I. Tsiafis, P. Todorovic, et al.: Vibration signal analysis of rubber-mounted roller-bearing mechanical system using hilbert-huang transform, pp. 315-323, 2014.

[11] C. Tsiafis, Z. Zaharis, M. Xanthopoulou, C. Skeberis, I. Tsiafis, P. Todorovic, et al.: Detection of non-linear signal distortions due to external impulse stimulations in rolling bearing experimental device, J. Balk. Tribol. Assoc., Vol. 21, pp. 233-245, 2015.

[12] A. Gent: *Engineering with Rubber*, Hanser, 1986.

[13] D. Fediuc, M. Budescu, V. Fediuc, V. Venghiac: *Compression Modulus of Elastomers*, Bul. Institutului Politeh, Din Iaşi, 2013.

[14] *Techinisce Informationen: Schwingungs-dämpfungselemente*, available at: www.reiff-tp.de.

[15] *Trelleborg, Solutions for Vibration & Shock*, 2003.

[16] General Technical Data and Materials, Simrit Catalogue, Edition 2000.