Rotor Bearing Casing with added Polymer Particle Composite

Analysis of Acoustic Emission Measured Results

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ABSTRACT

The trend of production machines with higher operation speeds brought the issues of vibration amplitude and acoustic emissions to the surface. To solve this problem, a standard approach requiring mass and/or stiffness increase, and utilizing high-damping polymer composite materials, e.g. by adding them to the empty spaces of the original structures is employed. The presented polymer composite application is for rotor bearing casing in which the polymer particle composite is added into the mechanical system by filling the empty space between the rotor bearing casing and the housing body. Before this application, an analysis of four polymer composite samples with different compositions was made. Then the logarithmic decrements were measured by two experimental methods and one composite was selected. The application showed acoustic emission maximum amplitude reduction of 67% and 33% when excitation amplitudes are low (up to 5 g) and large (above 10 g), respectively. In the case of the FFT spectrum of acoustic emissions, the reduction was 85% and 51%.

Keywords-particle composite; damped component; logarithmic decrement; time record; FFT spectrum

I. INTRODUCTION

When increasing the production machines working speed, a state in which resonance occurs can be reached. The most effective way to reduce the vibration amplitude is to increase the mechanical system damping. One way to achieve this is by using material damping. The damping parameters are very low for a standard structural material such as steel. Cast iron and aluminum can be classified as standard structural materials, which have better material damping than steel, but the desired damping effect is not so significant. So, non-standard construction materials and their structures come into consideration. The usual sources of Acoustic Emission (AE) are material failure in the form of fracture (brittle, tough, etc.), plastic deformation, decohesion, inclusions, corrosion, leakage of the operating medium, flow of the operating medium, and the surrounding environment (wind, noise, vibrations). So, AE can be used in fracture mechanics, to determine either the material mechanical state or the presence of corrosion. It is advantageously used for continuous monitoring of complex structures condition in aviation, construction industry, geological engineering, energy production, etc. In production technologies, it can be used to determine tool wear during cutting to check the technological procedure steps. At the same time, AE analysis along with vibration analysis are well-known methods of testing and monitoring the condition, especially of bearings fault diagnosis [1-3], geared shafts and gears [4, 5], oil viscosity [6], production processes [7-9], manufacturing processes [10-12], etc. AE can detect very low energy loss processes due to friction or cavitation. These loss processes create waves of energy in the macrostructure and microstructure due to dislocation and degradation procedures. In the mentioned fields, the AE (noise) is an unwanted effect, however in [13], the sound waves generated by developed acoustic cyclone separator are used for acoustic air cleaning equipment.

Polymers are materials with good damping, but their stiffness and strength are not satisfactory for common engineering applications. However, composites with a polymer matrix find applications in mechanical engineering in the fields of machinery and technology production. Composites of particle-shaped reinforcement (also called polymer-concretes) are the most often ones used in the mentioned fields. Fiber-reinforced polymer composites are also utilized in machinery, mainly due to their high stiffness and strength. They have satisfactory material damping, but it is not as significant as the damping of polymer -particle composites. High damping materials, e.g. lead rubber bearings, are applicable as seismic protection means, even in nuclear power plants. The non-isolated response is 2-3 times larger than the response of base-isolated structures [14]. The industrial bearings of machines and their faults are important vibration sources and AE for which monitoring and evaluation techniques are developed [15].

Each production machine represents a mechanical system consisting of components that naturally have their own mechanical compliance, which is one of the sources of vibration and subsequent AE. When measuring the response in the form of vibrations or AE, it is the response of the entire system, which is being measured rather than that of its individual members. Thus, the excitation point (input) and the output point in the form of a working member (tool) are important in a mechanical system. However, individual components located in the mechanical system between the input and the output as well as their connections or placement in the fluid can either increase or decrease vibrations and AE. To increase damping in mechanical systems with steel components while the contact stiffness is maximum, high material damping can be inserted into the mechanical system. Of course, the weight and stiffness of that system are also increased, but these values are negligible compared to the weight and stiffness of other components. These high-damping materials manifest themselves most significantly in the area of resonant working speeds.

II. DETAILS OF EXPERIMENT

A. Materials and Procedures

Before the rotor bearing casing application, we made four samples for testing, the detailed description of which is in Table I. The fillers A-D of samples S1-S4 differ mainly in the size of silica sand grains and their composition expressed by volume ratio.

TABLE I. SAMPLES

	S1	S2	S3	S4
Matrix:filler (Volume ratio)	1:3.5	1:3	1:3	1:2
Epoxy resin: Fixative (volume ratio)	100:50	100:16	100:16	100:50
Size of individual fillers (mm)	Filler A: 0.1-0.3 Filler B: 0.3-1 Filler C: 2-4	Filler A: 0.1-0.3 Filler B: 0.3-1	Filler A: 0.1-0.3 Filler C: 2-4	Filler D: 0.1-0.8
Volume ratio of fillers	2:3:5	1:1	1:1	1
Average size of fillers (mm)	1.735	0.425	1.600	0.45

The following measuring devices and software were used: Acoustic emission sensor Vallen-VS45-H with a range of 20 kHz–400 kHz, Dynamic Signal Acquisition device NI-9223 module meter, signal level ± 10 V, resolution 16 bit, sample rate 1 MS/s/ch, software for advanced analysis of the dynamic signal base on LabView Sound and Vibration Toolkit, National Instruments Corporation.

B. Application Description

The rotor bearing driven by a flat belt is a source of vibration for the housing unit (Figure 1), which is part of the textile machine mechanical system. The reduced vibrations, which are the sum of the housing unit reduced vibrations, the bearing in the casing, and vibrations in the bearing individual parts, manifest in reduced AE. The native design of the rotor casing is shown in Figure 2. That design involves a casing and tube in which the rotor is mounted when operating. The native design was designed for a lower rotational speed than the increased rotational speed at which it is expected to operate.

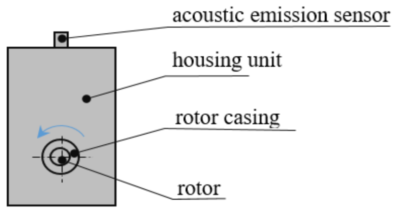


Fig. 1. Housing unit, its parts, and acoustic emission sensor.

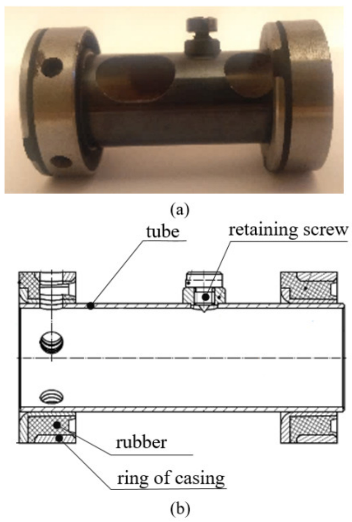
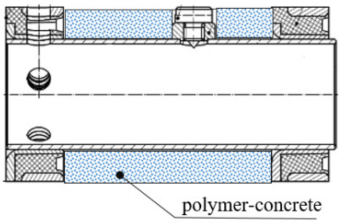


Fig. 2. Rotor casing (a) photo, (b) drawing.

Therefore, the native design stifness is insufficient when the speed is increased to 100,000-130,000 rpm, and the increased deformations contribute to the amplitudes augmentation. The indicated speeds are in the first half of the resonance peak. Several designs of rotor casing were realized, namely those with a reinforced tube and different types of rubber filling at each side of the casing. This article presents the modification of the design by using an additional polymer particle composite (Figure 3), which contributed to increasing material damping in addition to increasing stiffness and mass. Stiffness, weight, and damping are three usual quantities affecting the dynamic behavior of mechanical systems. Dynamic behavior is visible in the form of the time record of vibrations when exciting, which from a theoretical point of view can be considered harmonic.

(a)



(b)

Fig. 3. Rotor casing with polymer particle composite (a) photo, (b) drawing.

The rotor bearing is inserted inside the rotor casing with polymer particle composite (Figure 4). Both components are mounted in the housing unit (Figure 1). The rotor is driven by a belt in contact with a pin that is part of a rotor.

Fig. 4. Tested rotor casing with a rotor bearing inside.

C. Determination of Logarithmic Decrement

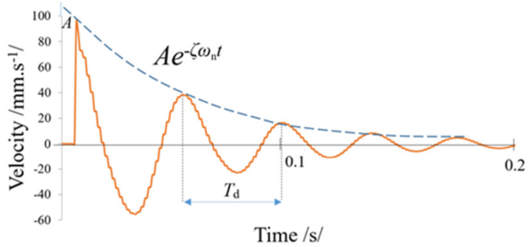
If we compare viscous and hysteresis damping models for a case of a mechanical system with a single degree of freedom and free vibration decaying effect, the model selection practically has no effect [16]. This applies in practice to conventional models with a damping ratio ζ <0.2. The measurement instruments for the determination of the logarithmic decrement were: Polytec PDV 100 contactless vibrometer and LabView software. Plots of time and frequency domain response were evaluated by hand calculation according to (7) and (8).

1) Decrement Determination from the Time Domain Curve

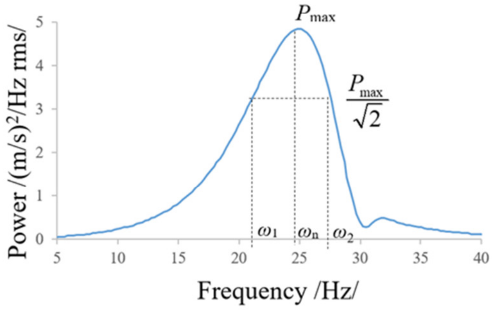
The measurement is based on the transient response of a Single Degree Of Freedom (SDOF) mechanical system on the impulse excitation force, i.e. the free vibration response [17, 18]. The tested material sample represents the SDOF mass-damper-spring vibration system whose dynamic behavior of mass m depends on spring and damper constants k and c, respectively. The equation of underdamped motion (ζ<1) for free damped vibrations of an SDOF mechanical system is:

$$x(t) = X_0 e^{-\zeta\omega_n t} \sin\left(\omega_d t + \varphi\right) \tag{1}$$

where x is the underdamped displacement (amplitude, distance from equilibrium) of the mass for free vibrations effect of mechanical system, X_0 is the amplitude, ζ is the damping ratio, ω_n and ω_d represent the undamped and damped natural frequency, respectively, and ϕ is the phase angle.



(a)



(b)

Fig. 5. Polymer particle composite, sample S1: (a) time domain response curve of 0.2 s, (b) frequency domain curve.

The logarithmic decrement δ, expressing the rate of decay, is defined by:

$$\delta = \ln\frac{x_1}{x_2} \tag{2}$$

where x_1 and x_2 are the amplitudes of two successive cycles (Figure 5(a)). According to (1), for displacements x_1 and x_2, it can be written:

$$\begin{aligned} x_1(t_1) &= X_0 e^{-\zeta\omega_n t_1} \sin\left(\omega_d t_1 + \varphi\right) \\ x_2(t_2) &= X_0 e^{-\zeta\omega_n (t_1+T_d)} \sin\left(\omega_d \left(t_1 + T_d\right) + \varphi\right) \end{aligned} \tag{3}$$

where $t_2 = t_1 + T_d$ and T_d is the damping period.

Substituting (3) to (2) and using $T_d\omega_d = 2\pi$, we obtain:

$$\delta = \zeta\omega_n T_d = \zeta\omega_n \frac{2\pi}{\omega_d} \tag{4}$$

The relation between undamped and damped natural frequencies is:

$$\omega_d = \omega_n \sqrt{1-\zeta^2} \tag{5}$$

Substituting (5) to (4) and arranging, we obtain the relation for damping ratio ζ:

$$\zeta = \frac{\delta}{\sqrt{4\pi^2 + \delta^2}} \tag{6}$$

2) Decrement from the Frequency Domain Curve

To be able to use a method based on the frequency domain, we need the frequency response function. In general, a Fourier transformation converts a signal of the time domain to the frequency domain, i.e. we obtain the frequency domain response spectrum. In our case, the Fast Fourier Transformation (FFT) is used through the NI Labview software. This method of damping ratio evaluation is named the half-power bandwidth method or the 3 dB method. The values at the resonant peak (Figure 5(b)), determine the damping ratio ζ as follows:

$$\zeta = \frac{\omega_2 - \omega_1}{2\omega_n} \tag{7}$$

where ω_1 and ω_2 are frequencies corresponding to the half-power point, i.e. a $1/\sqrt{2}$ drop from the resonant peak value. On a decibel scale, this would correspond to a 3 dB drop from the peak [19].

In the case of light damping, it is suitable to use the simplified formula $\delta \approx 2\pi\zeta$ resulting from (6).

III. RESULTS

The measured results of logarithmic decrement determined by the method of free-damped vibrations and the half-power bandwidth method are listed in Table II. Based on the measurement and availability of polymer- particle composite components, we created for application the polymer particle composite according to sample S4. To determine the logarithmic decrement, (2) was adjusted to evaluate the set of vibration cycles:

$$\delta = \frac{1}{n} \ln \frac{x_i}{x_{i+1}} \tag{8}$$

where n is number of cycles and x_i is the i-th amplitude of the i-th cycle.

TABLE II. LOGARITHMIC DECTEMENT Δ

Sample	δ Method of free damped vibrations	δ Half-power bandwidth method	Arithmetic mean	Δ (%)
S1	0.8493	0.7555	0.8024	-11.04
S2	1.0790	0.9195	0.9993	-14.78
S3	1.0352	0.7923	0.9138	-23.46
S4	1.0486	0.8828	0.9657	-15.81

A comparison of the existing and new designs was made at a steady excitation rotational speed of 135,000 rpm, which is the area of resonance. AE results were evaluated in the range of 50 to 400 kHz. Two modes of excitation by rotor generating amplitudes were tested: to 5 g and above 10 g, corresponding to excitation by suitable and unsuitable rotors.

Figures 6(a)-(b) show the time records of the AE amplitude response of the native design (black) and the new design (green), respectively.

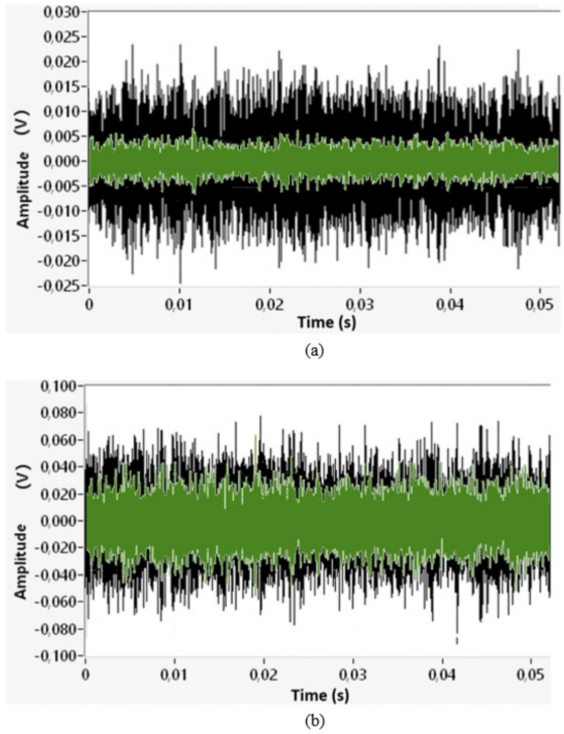


Fig. 6. Time records of AE of native (black) and new design (green): (a) excitation up to 5 g, (b) excitation above 10 g.

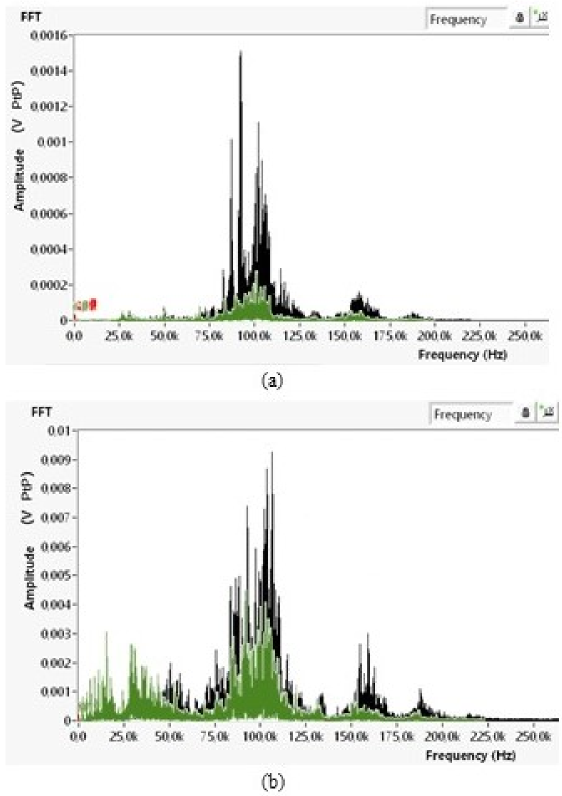


Fig. 7. FFT spectrums of AE of native (black) and new design (green): (a) excitation up to 5 g, (b) excitation above 10 g.

The FFT spectrums in Figure 7 show significant reduction mainly for excitation up to 5 g, where the amplitude reduction is about 85%. The FFT spectrum of the new design (green) shows new frequency peaks. The results are summarised in Table III.

TABLE III. SUMMARY OF RESULTS

		Excitation amplitude up to 5 g		Excitation amplitude above 10 g	
		native	**damped**	**native**	**damped**
Additional mass (grams)		-	70	-	70
Mass (grams)		92	162	92	162
Time record	Maximum amplitude (V)	0.012	0.004	0.036	0.024
	Maximum amplitude reduction	-	66.6%	-	33.3%
	Amplitude reduction efficiency E_a (-)	-	0.876	-	0.438
FFT spectrum	Maximum amplitude (V PtP)	0.00151	0.00023	0.0092	0.0045
	Maximum amplitude reduction (V PtP)	-	84.7%	-	51.1%
	Frequency peaks (kHz)	100, 160, 190	100, 160, 190	50, 100, 165, 190	15, 30, 100, 155, 190
	Amplitude reduction efficiency E_a (-)	-	1.114	-	0.671

IV. CONCLUSIONS

The current paper presents an application of polymerparticle composite in the rotor bearing casing. Before application, analysis of four polymerparticle composite samples with different compositions was conducted, the logarithmic decrements were measured by two experimental methods, and one composition was selected for the considered application. The presented application is for rotor bearing casing in which the polymer- particle composite is inserted into the mechanical system by filling in the free space between the casing and the housing body. The application showed beneficial results for other similar applications in production machines. The study outcomes are summarized as:

- For the time domain, the reduction of acoustic emission maximum amplitudes was 67% and 33% when excitation amplitudes were up to 5 g and above 10 g, respectively.
- For the FFT spectrum of acoustic emission, the reduction was 85% and 51%.

The novelty of the presented application lies in operation with extreme rotational speeds in the resonant area. The polymer particulate composite is filled in a relatively small cavity of the joint, resulting in the ability to increase the operational speed with minimum time of design modification and costs. The mentioned features make the presented approach very effective. The results are consistent with the results of applications based on adding high-damping capacity materials such as aluminum foam sandwiches and carbon fiber-reinforced polymers [20], particulate reinforced polymer composites [21], and particulate damper [22] into machine tool structures.

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