Damping of layered porous composites and an application in machinery

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Abstract. The structured composite can involve the porous materials providing the important internal damping that is usable in mechanical engineering applications. The stiff base material is equipped by layer/s of softer porous material/s covered by constrained layer. Such layered structure is characterized by increased internal damping in vibration process. Paper analyses some either measured or simulated mechanical properties of designed layered porous composites. Novelty provided in paper is a specific application of structured composite for damping the high frequency vibrations in resonance frequency obtaining the beneficial results, e.g. 40-48% of maximum amplitude reduction.

1. Introduction Amplitude reduction efficiency
The main parameters that affect the level of vibration of mechanical systems and thus the emission of noise are mass, stiffness and damping. Increasing the mass is one of the design approaches, but because of requirement of reduction of energy consumption, the priority, especially for moving parts of structures, is to use lightweight materials with high internal damping.

The layered structures can be divided in two groups: unconstrained layer dampers (ULD) and constrained layer dampers (CLD), i.e. sandwich structure. Large metal or fibre composite panels is accomplished typically by applying one of the following techniques: (i) a damping polymeric layer is attached to the outer surface of the host structure – referred to as unconstrained layer damping treatment; (ii) a damping polymeric layer is either sandwiched between two layers of the structure or attached to the host structure and constrained by an additional thin layer – referred to as constrained layer damping treatment [1,2]. Multiple constrained damping layers can also be used in order to improve damping properties over a wider frequency and temperature range [1,2]. The loss factor is not generally increased through the use of more of the damping material in a single constrained layer, either by increasing thickness or length. Thus, attempts to produce greater added damping have turned to the use of more than one constrained layer [3]. Passive or semi-active dynamic vibration absorbers are known approaches in change the dynamic characteristics [4].

Vibration of the mechanical system is caused by various influences, mainly caused by inaccuracy of production (geometric and dimensional inaccuracy), subsequent assembling, wear, mutual friction in contact surfaces and excitation forces of various character - impulse, shock, cyclic, etc. from various sources.
Vibration is the movement of a body around an equilibrium position and the wave is the transmission of an oscillating motion of a source through an environment. The oscillation of the system and the propagation of the acoustic wave are physically connected issues and are inseparable from each other. Discontinuities (e.g. change of component geometry, change of material properties), contact surfaces in joints or the presence of special devices (vibration absorbers) using high-damping materials or active dampers with an external energy source contribute to the change of vibration amplitude. Generally, the main goal is to reduce the amplitude of vibrations in the structure, which contributes to reducing the intensity of emission into the space and thus to reducing noise.

In addition to quantities that characterize mechanical vibration (displacement, velocity and acceleration of vibration), we also know quantities that characterize the transmitted energy of mechanical vibration, e.g. mechanical impedance, mechanical mobility (reciprocal value of mechanical impedance), intensity of mechanical vibration, power of mechanical vibration, etc. The mechanical impedance changes due to discontinuities, which can be the dimension changes in cross-sections and changes in material properties.

Under dynamic loading, the materials and discontinuities (gaps) between components convert some mechanical energy into another form of energy. However, real materials are not perfectly elastic bodies and have different ability to dissipate mechanical energy, which is experimentally proved by a hysteresis loop, which expresses the amount of dissipated energy (converted to another type of energy - heat) in a unit volume of material per cycle under dynamic stress. The area of the hysteresis loop increases with the degree of viscoelastic properties of the material. The amount of dissipated energy is a measure of the internal damping of the material. Vibration attenuation depends on material damping as an important factor \[5,6\].

The similar principals can be found in civil engineering in building structures, e.g. the clamps as passive dampers between stones in historical buildings strengthened the historical wall structure and increase the seismic energy dissipation and protection of the structure from collapse \[7\]. Numerical simulation of damping ratios of stress waves for high velocity shock waves in fiber composites shows the increase of damping up to 37% (fiber radius 0.5mm and fiber volume 35%) comparing with model without fibers \[8\]. Vibration sources in mechanical systems are various and their modelling is not trivial task using the commercial software as it is presented in \[9\].

In the presented paper, we applied the approach of increasing vibration energy dissipation by layered porous composites and the viscoelastic properties of that materials and by using several layers of porous materials to obtain the desired vibration reduction performance. The passive reductions in oscillations (followed by noise reduction) and the transmission of waves from the source to the adjacent components of the mechanical system and the surrounding environment is used.

1.1. Amplitude reduction and loss efficiency due to additional mass

Mass-efficient configurations are identified by the values of the amplitude reduction efficiency \(E_A\) according to \[10\]:

\[
E_A = \frac{A}{m_a}
\]

(1)

where \(A\) is the amplitude reduction relative to the undamped structure and \(m_a\) is the additional mass of the damper as a proportion of the native mass of structure. Moreover, the loss efficiency can be defined similarly as

\[
E_\eta = \frac{\eta}{m_a}
\]

(2)

where \(\eta\) is the modal loss factor of the first mode.

1.2. Amplitude reduction due to internal damping
In general, the material microstructure significantly influences the material (internal) damping. The internal damping is effected by factors as frequency, temperature, amplitude of strain or stress, moreover, corrosion fatigue, grain size and porosity (for metallic materials). Grain boundaries of metals provide higher damping then grains themselves [11]. Thus we can state that more boundaries and its properties directly influence the material damping. In case of metals, the higher porosity produces higher internal damping what is applicable for other materials, e.g. porous materials.

1.3. Amplitude reduction and reduction of vibroacoustic energy by discontinuities
Using the vibration absorbing materials, the benefit of noise reduction is additional advantage. Structural vibration induces the vibration of surrounding fluid that generates the sound wave that is partially absorbed, partially reflected and transmitted. Sound absorbing mechanisms are mainly due to viscous loss of air pumped through the cellular structure and internal damping of material [12,13]. The vibration attenuation is achievable by modifying the vibration path with layers of suitable materials [12].

Polymers are known for their fine material damping. The polymer foams have the higher damping parameters as monolithic polymers. Moreover, the foams are known as excellent sound absorbing (insulation) materials. Thus, we can suppose that the source of the improved material damping of polymer foams are cells, i.e. discontinuities, filled by air.

Material discontinuity is another type of discontinuity. The material damping of layered porous composites is improved by material discontinuities, i.e. change of material properties, at interfaces between individual the layers.

Both interfaces and cells are discontinuities which contribute to increase the internal damping and decrease the amplitude of vibrations and sound absorption, Figure 1. Moreover, the change in geometry, i.e. the cross-section change is considered as discontinuity. Damping at discontinuities improves the material damping.

The interfaces between layers are discontinuities in macroscale level and the cells of foam structures are discontinuities in microscale level. At each discontinuity, the equilibrium of stresses (internal forces) and the continuity of mechanical vibration velocities apply.

Figure 1. Stress and velocity of vibroacoustic wave at discontinuity \( v_i, v_T \) and \( v_R \) and \( v_i, v_T \) and \( v_R \) is stress velocity of incident, transmitted and reflected wave, respectively.

1.3.1. Macroscopic level. The velocities of vibroacoustic wave at discontinuity according to low of momentum are

\[
v_i = v_T + v_R \quad \Rightarrow \quad v_T = v_i - v_R
\]

where \( v_i, v_T \) and \( v_R \) is velocity of incident, transmitted and reflected wave, respectively. Momentum of a particle of an incident wave is sum of momentum of particle of transmission and reflection wave, Figure 1.
The equilibrium of forces and stresses is at discontinuity is that sum of forces in layer 1 is equal to force in layer 2:

\[ \sigma_1 + \sigma_R = \sigma_T \]  

(4)

Using the relation for specific mechanical impedance \( Z \) (with respect per unit area; \( F = \sigma A \))

\[ Z = \frac{\sigma}{v} \Rightarrow \sigma = vZ \]  

(5)

Specific mechanical impedance \( Z \) can be calculated as

\[ Z = \rho c_L \]  

(6)

where \( \rho \) is density of material and \( c_L \) is the longitudinal wave speed in that material.

Intensity of mechanical vibration \( I \) is

\[ I = v\sigma \Rightarrow v = \frac{I}{\sigma} \]  

(7)

Using previous equations and the expression of power transmission coefficient \( d \) is obtained

\[ \frac{4Z_1 Z_2}{(Z_1 + Z_2)^2} = \frac{I_T}{I_1} = d \]  

(8)

where \( I_T \) and \( I_1 \) is intensity of transmitted and incident wave, respectively. It expresses the ratio between the vibroacoustic energy that transmitted through the interface and incident the interface (discontinuity).

The power reflection coefficient is

\[ r = \left( \frac{Z_2 - Z_1}{Z_1 + Z_2} \right)^2 \]  

(9)

The power reflection/transmission coefficients are defined as the ratio between the reflected/transmitted power and the power of the incident wave.

1.3.2. Microscopic level. The surface of the pores becomes another interface. The transmitted wave from the previous interface (pore) then becomes an incident wave and passes through next discontinuities.

Above mentioned relationships do not consider the absorption of vibroacoustic energy, which is expressed by the absorption coefficient, which expresses the degree of absorption of vibroacoustic energy. The applied porous materials in the designed LPC are good absorbers of vibroacoustic energy.

2. Layered porous composites

Three types of layered porous composites (LPC), Figure 2, were designed to reduce vibration amplitudes. The LPC material damping uses material damping mainly of polymers and is enhanced by the presence of pores. The used material structures were solids, foams, and honeycomb made of rubber, PVC (polyvinyl chloride), natural agglomerated cork and aramid. The mentioned materials were arranged in alternating layers, Figure 2, connected together by adhesion layers. The thickness of individual layers is 2.5-4 mm. Table 1 provides mass and density of the designed LPCs.
Table 1. Mass and density of LPCs.

|        | LPC 1 | LPC 2 | LPC 3 |
|--------|-------|-------|-------|
| Mass m (g) | 36.3  | 16.0  | 16.8  |
| Density ρ (g/cm³) | 0.79  | 0.35  | 0.36  |

Figure 2. Materials

Figure 3. (a) rubber foam and (b) PVC foam, scale is up 5 mm and down 1 mm, respectively, (c) cork, scale is up 5 mm and down [14] 0.05 mm

Synthetic rubber is a polymer in form of flexible closed cells foam and solid layers (LPC 1), Figure 3a. PVC is a thermoplastic polymer, in form of rigid cellular structure with preferably closed cells. Although the literature lists cells for rigid PVC foam and flexible rubber foam as closed, a detailed look at PVC foam, Figure 3b, shows that it contains open, closed and partially closed cells. The cells in cork cellular structure are the thin-walled closed cells, Figure 3c, down. The cork layers consist of agglomerated cork granules structure with random orientation, Figure 3c, up. Aramid is aromatic polyamide with molecules characterized by relatively rigid polymer chains. In designed layered structure is aramid in form of aramid over-expanded paper honeycomb consisting of closed cells filled by air (wall thickness is 0.05 mm).

3. Application in machinery

Sources of vibration in application are a textile high-speed bearing (rotor) and vibration of the driving belt, Figure 4. In order to reduce vibrations in the original structure of mechanical system, the
tensioning idler pulley is damped by a prestressed steel plate. In order to reduce the amplitude of vibrations, we applied three types of presented LPCs on a steel plate (steel base).

Figure 4. Schema of a part of a mechanical system

In operation, for rotational speed of rotors up to 100,000 rpm, the magnitude of the vibration was usually below the Warning limit, Figure 5. At increasing speeds, the magnitude of the vibration (shown by the vibration acceleration in Figure 5) increases, and at speeds around 130,000 rpm, the peak of the resonance is for most of rotors. For the operation of actual rotor shown in Figure 5, the limit speed is 119,000 rpm when the vibration acceleration reaches values above 5g corresponding to Warning limit, and from 120,000 rpm, the vibration is above Danger limit (10g), when the bearing life is shortened by an order of magnitude. Similar behavior, as described, is for 90% of rotors.

As mentioned, the goal is to reduce the vibration amplitudes by LPCs in order to achieve a shift of the Warning limit corresponding to higher rotational speed and thus to make the production faster.

Figure 5. Rotor excitation (bearing 095) - time (80 seconds) vs. (a) rotational speed, rpm, (b) acceleration, g PtP, (c) velocity, mm/s rms

3.1. Results

By applying LPC 1, 2 and 3, we achieved the time records shown in Figure 6. The reduced amplitude modulation can be clearly seen on the vibration acceleration waveforms. Maximum amplitude is reduced by 48.0, 46.0 and 40.0% for LPC 1, 2 and 3, respectively, compared to the undamped amplitude. The reduction is significant. Furthermore, the RMS amplitude at resonance is reduced by 29.0, 28.7 and 24.2%, which are also significant values.
Figure 6. Time records of ACC amplitude of 0.05 seconds at resonant rotational speed

The shown reduction of maximum amplitudes and RMS at resonance for the application of individual LPCs result in a reduction of the resonant peak and a change in its shape, Figure 7, for LPC 2 and 3.

Figure 7. Amplitude vs. Rotational speed

3.1.1. Discussion. The mass of individual LPCs is different (Table 1) and the obtained decreases of RMS amplitudes at resonance are significant. The LPC 1 has the greatest mass, i.e. together with steel base 19.7g + 36.3g = 56.0g and minimum mass is for the LPC 2, 19.7g + 16.0g = 35.7g. Their results in the amplitude decrease are almost the same (29.0 and 28.7%, respectively). An advantage is of the damper 1 due to mass. However, almost the same resulting effect is achieved for both, but with a mass of 56% lower for LPC 2.

The parameter amplitude reduction efficiency $E_A$ according to (1) is in Table 2. The most efficiency in amplitude reduction is for LPC 2.

Table 2. Amplitude reduction efficiency

|       | LPC 1 | LPC 2 | LPC 3 |
|-------|-------|-------|-------|
| Max. amplitude: 50 g Peak |       |       |       |
| RMS amplitude at resonance: 33.1 g PIP |       |       |       |
| Max. amplitude: 26 g Peak |       |       |       |
| RMS amplitude at resonance: 23.5 g PIP | 48.0% | 29.0% |       |
| Max. amplitude: 27 g Peak |       |       |       |
| RMS amplitude at resonance: 23.6 g PIP | 46.0% | 28.7% |       |
| Max. amplitude: 30 g Peak |       |       |       |
| RMS amplitude at resonance: 25.1 g PIP | 40.0% | 24.2% |       |
Amplitude reduction efficiency $E_A (-)$  

|     | 5.2 | 11.1 | 9.4 |

By adding the LPC onto steel base, the stiffness is changed. To determine Young's modulus of LPC 1 or 2 including the steel base, the deflection $w_{LPC+}$ was calculated using computational FEM (Finite Element Method) software. The resulting Young's modulus $E_{LPC+}$ was determined using the following relationship:

$$E_{LPC+} = \frac{F}{3w_{LPC+}L^3}$$

(10)

where $F$ is applied load, $l$ is length of beam (of LPC with steel base), $w_{LPC+}$ is deflection of LPC and steel base and $I$ is area moment of inertia. Bending stiffness $E_{LPC+}L^3I$ is not an advantage for either damper 1 or 2 as they differ only 3%.

The passive dampers change the natural frequencies and mode shapes as shown in Figure 8. The presented models do not involve cell structure having the material properties assigned as for homogeneous material. The approach for determination the material properties regarding the pore size, cell wall and foam porosity from representative volume element using homogenization procedure is described in [15].

![Figure 8. Natural frequencies and natural modes](image)

If LPC 1 and 3 with steel base would have the same mass as LPC 2 with steel base, their RMS amplitudes would be 28.6 and 25.5 g, i.e. 13.4 and 23.0% of undamped RMS Amplitude reduction. In case of same mass, we can compare the damping contribution. LPC 2 with steel base would have 15.3 and 5.7% higher internal damping comparing to LPC 1 and LPC 3 with the steel base. Despite the fact that the LPC 1 and LPC 3 obtained almost the same reduce of resonance amplitude, considering the mass the best damping properties is for LPC2.

4. Conclusion

Use of structural composites to absorb energy becoming the modern approach as the material research provides large possibilities and variability. Increasing energy dissipation by increasing damping capacity is one way to reduce vibration. The application of additional damping in the form of an additional mostly layered viscoelastic material to the existing structure is an effective way to control the transient and steady-state oscillations.

Applying LPCs, the significant decrease of resonance peak and maximum amplitude was achieved – up to 29%. LPC1 and LPC2 obtained almost same result, but mass of LPC2 is 56% lower and
stiffness of both differ only 3%. Considering the same mass and almost same stiffness of both LPCs, material damping of LPC2 is 15.3% larger.

The future direction of research is in designing the LPCs using the specific materials to obtained the improved damping parameters of LPCs, both with support of numerical simulation and experiment.

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