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Acoustical properties of layered beams with attached dynamic vibration absorbers*
Właściwości akustyczne belek wielowarstwowych z przyłączonymi dynamicznymi absorberami**

Key words: sound transmission, composite, laminated structure, inclusions, Timoshenko beam, stiffness constant, eigen-frequencies, dynamic vibration absorbers
Słowa kluczowe: transmisja dźwięku, struktura wielowarstwowa, kompozyt, belka Timoshenki, stałe sztywności, częstotliwości własne, dynamiczny wibracyjny absorber

Introduction

Noise and vibration are of concern with many mechanical systems including industrial machines, home appliances, transportation vehicles, and building structures [Randal 2009, Rmili et al. 2009, Roozen et al 2009, Thuma 2009]. Knowledge of the sound transmission properties of structures including aircraft, vehicle and ship cabin walls and building walls is also important in order that occupants can be protected from external noise sources. Many such structures are comprised of beam and plate like elements. The vibration of beam and plate systems can be reduced by the use of passive damping, once the system parameters have been identified [Chakraborty 2008, Randal 2009, Rmili et al. 2009, Roozen et al 2009, Thuma 2009, Sakrar and Sonti 2009, Xu and Wu 2009]. In some cases of forced vibration, the passive dam-

*Text was layed out in two-column page, considering complex equations.
**Z uwagi na rozbudowane wzory artykuł złożono jednołamowo.

ping that can be provided is insufficient and the use of active damping has become attractive. The rapid development of micro-processors and control algorithms has made the use of active control feasible in some practical situations [Das and Parh 2009, Elliot 2009, Tanaka 2009]. In most cases, however, passive control is preferred to reduce vibration and sound transmission through structures.

Structures composed of laminated materials are among the most important structures used in modern engineering, especially in the aerospace industry. Such lightweight and highly reinforced structures are also being increasingly used in civil, mechanical and transportation engineering applications. The rapid increase in the industrial use of these structures has necessitated the development of new analytical and numerical tools that are suitable for the optimization of the damping and acoustical properties of such structures with micro and macro inclusions [Thamburaj and Sun 2002, Thompson 2008, Conlon and Hambric 2009, Idrisi et al. w.d.].

The transmission of sound through structures has been investigated extensively for many years. Most studies, until recently, have been limited to the transmission of sound through isotropic materials. It is well known that the mass per unit area, structural vibration damping and structural stiffness are all important parameters that affect the vibration and sound transmission properties of isotropic and anisotropic materials. Only in recent years have studies been made of the transmission of sound through anisotropic materials. Wave transmission theory for elastic bodies is discussed in [Brekhovskikh 1960]. A transmission matrix for the relationship between the velocity and pressure for elastic solid bodies is given in [Allard et al. 1987]. To achieve effective damping over a wide frequency range, various methods are used. Active vibration control techniques can achieve high damping over a wide range of frequencies [Bingham et al. 2001]. However, active damping usually suffers from collateral effects [Hansaka and Mifune 1994, Lin et al. 2002]. Magnetic and particle vibration dampers can have considerable weight penalties. Passive damping using viscoelastic materials [Rao 2003, Li and Crocker 2005] is simpler to implement and more cost-effective than semi-active and active damping techniques. Reactive passive devices have been developed to control low-frequency (< 1000 Hz) noise transmission through a panel in [Carneala 2008]. Re-active passive devices use passive constrained layer damping to cover the relatively high-frequency range (> 150 Hz), reactive distributed vibration absorbers can cover the medium-frequency range (50–200 Hz), and active control can be used to control low frequency noise (< 150 Hz). Overall, re-active passive devices can increase the broadband (15–1000 Hz) sound transmission loss by about 10 dB.

The present paper aims at developing a simple numerical technique, which can produce very accurate results compared with the available analytical solutions and also one which allows one to decide on the amount of refinement in the higher order theory that is needed for accurate and efficient analysis [Diveyev and Crocker w.d., Diveyev et al. 2008a, b]. The elastic constants of the equivalent Timoshenko beam have been determined by using an identification procedure based on experimental design, and a multilevel theoretical approach [Diveyev and Crocker w.d., Diveyev et al. 2008a, b].

Adaptive plate cylindrical bending equations

Let us consider now a symmetrical three-layered plate of thickness $2H_p$ (Fig. 1).

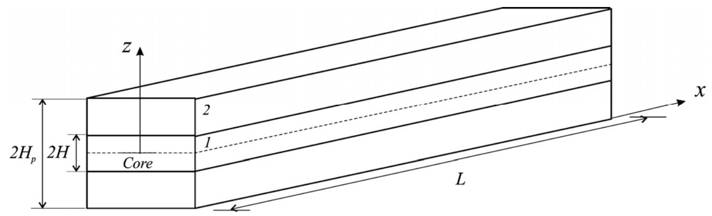


FIGURE 1. Sandwich plate scheme
RYSUNEK 1. Schemat płyty typu sandwicz

Let us consider cylindrical bending of plate by such kinematic assumptions [Diveyev and Crocker w.d., Diveyev et al. 2008a, b]:

$$
\begin{aligned}
U_e - &\begin{cases} u = \sum\limits_{i,k} u_{ik}{}^{e} z^{2i-1} \varphi_k(x), \;\; 0 < z < H \\ w = \sum\limits_{i,k} w_{ik}{}^{e} z^{2i-2} \gamma_k(x), \;\; 0 < x < L \end{cases} \\
U_d - &\begin{cases} u = \sum\limits_{i,k} u_{ik}{}^{d} (z - H)^{i} \varphi_k(x), \;\; H < z < H_p \\ w = \sum\limits_{i,k} w_{ik}{}^{d} (z - H)^{i} \gamma_k(x) \;\; 0 < x < L \end{cases}
\end{aligned}
\tag{1}
$$

here:

$\varphi_k(x)$, $\gamma_k(x)$ – are a priory known coordinate functions (for every beam clamp conditions),

$u_{ik}{}^{e}$, $w_{ik}{}^{e}$, $u_{ik}{}^{e}$, $w_{ik}{}^{e}$ – unknown set of parameters.

The solutions which express Hooke's law with respect to the stress components have the form:

$$\sigma_{xx} = C_{xx}\varepsilon_{xx} + C_{xz}\varepsilon_{zz}; \quad \sigma_{zz} = C_{zx}\varepsilon_{xx} + C_{zz}\varepsilon_{zz}; \quad \tau_{xz} = G\gamma_{xz} \tag{2}$$

By substituting equations (1) and (2) into the following Hamilton-Ostrogradsky variation equation:

$$\int_{t_1}^{t_2}\left(\int_V(\sigma_{xx}\delta\varepsilon_{xx}+\sigma_{zz}\delta\varepsilon_{zz}+\tau_{xz}\delta\varepsilon_{xz}-\rho\frac{\partial u}{\partial t}\delta\frac{\partial u}{\partial t}-\rho\frac{\partial w}{\partial t}\delta\frac{\partial w}{\partial t})dV+\int_{S_K}KU\delta UdS=\int_{S_P}P\delta UdS\right)dt \tag{3}$$

(V – beam volume, S_K – clamp contact surface, S_P – boundary forces surface t_i – arbitrary time moment) for Winkler foundation clamp conditions with the rigidity coefficient K and also assuming single frequency vibration ($u_{ik}^e=\overline{u}_{ik}^e e^{i\omega t}$, $w_{ik}^e=\overline{w}_{ik}^e e^{i\omega t}$, $u_{ik}^d=\overline{u}_{ik}^d e^{i\omega t}$, $w_{ik}^d=\overline{w}_{ik}^d e^{i\omega t}$) we obtain the set of linear algebraic equations for the amplitudes [Diveyev and Crocker w.d., Diveyev et al. 2008a, b]:

$$[A]\overline{U}=\begin{bmatrix}A_1 & A_d\\ A_d^T & A_2\end{bmatrix}\begin{bmatrix}\overline{U}_e\\ \overline{U}_d\end{bmatrix}=f \tag{4}$$

The corresponding frequency equation for the material with the viscous damping should be written such:

$$-\omega^2[M]\overline{U}+i\omega[C]\overline{U}+[K]\overline{U}=[A]\overline{U}=\overline{f} \tag{5}$$

This is the traditional frequency domain method which is normally used in linear elastic system investigations.

Transition to the Timoshenko beam

Three-layered beam with the soft core (sandwich)

Let us consider a three-layered symmetrical beam (Fig. 1). Details of beam modeling are presented in [Diveyev and Crocker w.d., Diveyev et al. 2008a, b]. Its mechanical properties are assumed to be: length L = 0.6 m and core thickness H = 0.0254 m, face layers thickness h = 0.003 m with damping core (the foam core elastic moduli are assumed to be as follows: $C_{xx}=C_{zz}$ = 180 MPa, G = 35 MPa, and C_{xz} = 40 MPa, density ρ = 240 kg·m^{-3}) and rigid face layers (fiber-composite material: C_{xx} = 43 GPa, C_{xz}= 6 GPa, G = 0.6 GPa, ρ = 2000 kg·m^{-3}).

For translation of the three-layered beam to the uniform Timoshenko beam of equal thickness and linear weight we are taken of use a next criterion:

$$C = \min_{E_T, G_T} \sum \left| f_S^i - f_T^i(E_T, G_T) \right| \text{ in the frequency range } \; f_k - \Delta_k/2 < f < f_k - \Delta_k/2 \tag{6}$$

here f_S^i, f_T^i – sandwich and Timoshenko beam eigen-frequencies, E_T, G_T are the Young and shear modules of Timoshenko beam. They change in some intervals:

$$E_0 - \Delta_E/2 < E_T < E_0 - \Delta_E/2, \; G_0 - \Delta_G/2 < G_T < E_0 - \Delta_G/2 \tag{7}$$

were E_0, G_0 are a priory values of this coefficients. The result of translation is presented in Figure 2. Only in higher frequency range (Fig. 2c) distinctions appear.

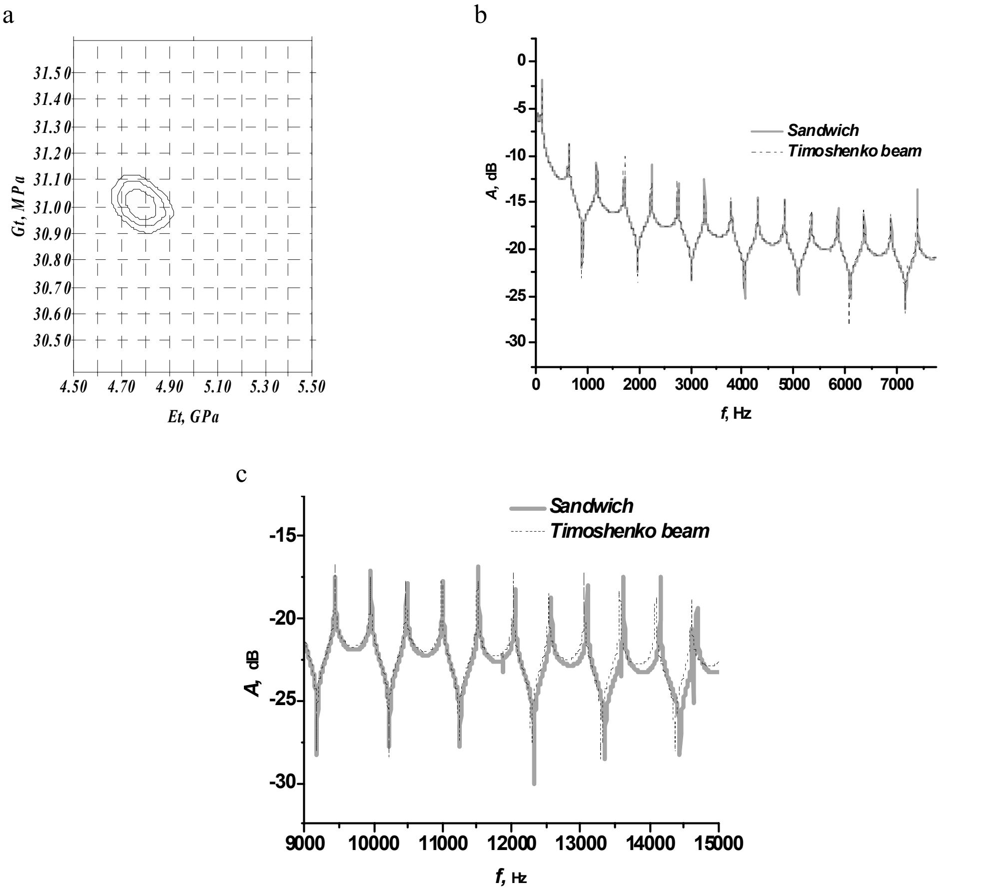


FIGURE 2. The result of sandwich translation to the Timoshenko beam: a – equivalent beam modules E_t, G_t; b – FRF for sandwich beam and the equivalent uniform beam; c – The FRF's in higher frequency range
RYSUNEK 2. Wyniki transmisji z płyty typu sandwicz do belki Timoshenki: a – ekwiwalentne moduły belki E_t, G_t; b – funkcja amplitudo-częstotliwościowa (FAC) dla belki sandwicza i ekwiwalentnej jednolitej belki; c – FAC w diapazonie wyższych częstotliwości

Three-layered beam with the rigid core

Let us consider now transition of a three-layered symmetrical beam with the soft thick face layer and thin rigid core: length $L = 0.6$ m and thickness $H = 0.02$ m, face layers thickness $h = 0.008$ m) (the foam face layers elastic moduli are assumed to be as follows: $C_{xx} = C_{zz} = 150$ MPa, $G = 33$ MPa, and $C_{xz} = 40$ MPa, density $\rho = 2000$ kg·m^{-3}) and rigid face layers (fiber-composite material: $C_{xx} = 50$ GPa; $C_{xz} = 6$ GPa; $G = 1.5$ GPa, $\rho = 2000$ kg·m^{-3}). In this case the equivalent beam can be found only for separate frequency range. In Figure 3 this equivalent module E_t, G_t are found for the different frequency regions.

In the Figure 3 the equivalent Young module (E) decreases and shear module (G) increases with the increasing frequency. This Timoshenko beam FRF's also approximate the sandwich FRF, but only with various moduli in the frequency range.

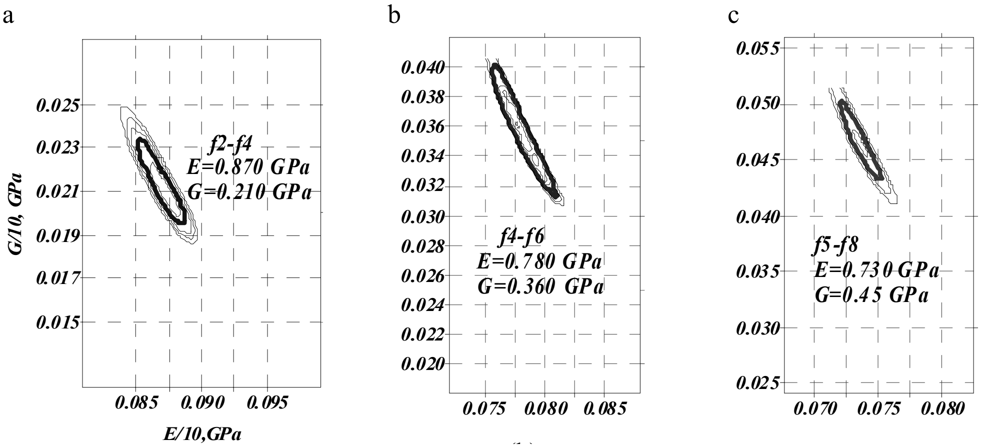


FIGURE 3. The result of sandwich translation to the Timoshenko beam, equivalent beam modules E_t, G_t: a – for $f_2 < f < f_4$; b – for $f_4 < f < f_6$; c – for $f_5 < f < f_8$
RYSUNEK 3. Wyniki transmisji sandwicza do belki Timoshenki – ekwiwalentne moduły belki E_t, G_t: a – dla $f_2 < f < f_4$; b – dla $f_4 < f < f_6$; c – dla $f_5 < f < f_8$

Frequency dependent damping

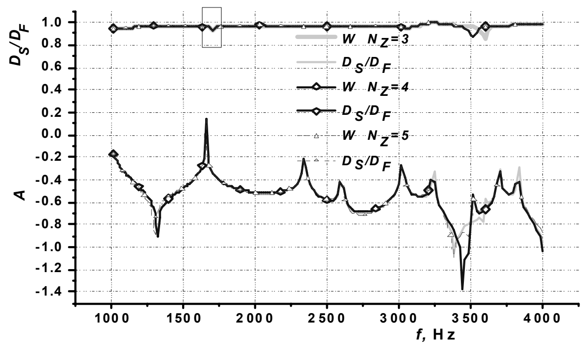
The loss factors in layered beams (plates in cylindrical bending) can be found by comparing their deformation energy. This result may be achieved by direct computation by use of the stiffness matrix if the damping matrix is proportional to the matrix (assuming viscous damping $C_i = \eta_i[K_i]$).

$$\eta = \frac{\eta_1 [q]^T |A_1||q| + \eta_2 [q]^T |A_2||q| + ... + \eta_N [q]^T |A_N||q|}{[q\ ]^T |A||q|} \tag{8}$$

Here: $|A|$ is the stiffness matrix, $|q|$is a vector of the displacement component, $|A_i|$ is the matrix stiffness component corresponding to the i-th layer ($|A| = \sum |A_i|$. The damping coefficients for a three-layered beam (for the following geometrical parameters: length $L = 0.6$ m; core thickness $H = 0.0127$ m; face layers thickness $h = 0.003$ m) with damping core (the foam core elastic modules are assumed to be as follows: $C_{xx} = C_{zz} = 180$ MPa, $G = 35$ MPa, and $C_{xz} = 40$ MPa, density $\rho = 240$ kg·m^{-3}) and rigid face layers (fibers composite material: $C_{xx} = 47$ GPa; $C_{xz} = 6$ GPa; $G = 0.6$ GPa, $\rho = 2000$ kg·m^{-3}) for various sandwich geometry are presented in Figure 4. The corresponding FRF's are also presented (good approximation properties of high order theory for $N_z \geq 3$ are presented).

Here D_S/D_I is the ratio of damping layer deformation energy to the whole deformation energy of sandwich. If other sheets are not damping, this last value present the ratio of damping in the sandwich to the damping value in the damping layer ($D_s/D_1 = \eta_s/\eta_1$).

a



b

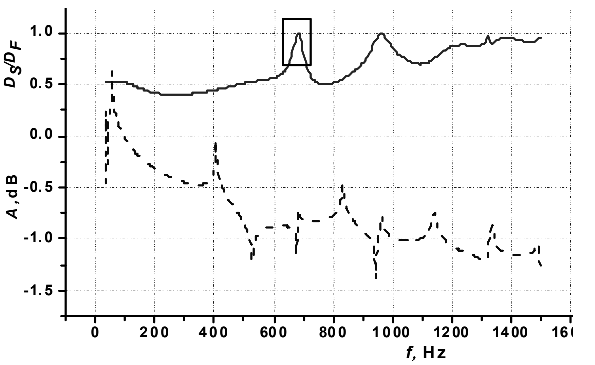


FIGURE 4. Frequency dependent damping for the sandwich beam: a – rigid face sheets; b – rigid core
RYSUNEK 4. Tłumienie drgań w belce typu sandwicz w zależności od częstotliwości: a – twarde warstwy zewnętrzne; b – twarda warstwa wewnętrzna

Damping variations may be seen. For the sandwich with the rigid face sheets and soft damping core (Fig. 4a) the damping is practically constant with some fluctuations. For rigid core sheet and damping soft face sheets (Fig. 4b) the damping increases with some fluctuations.

Transition to the Timoshenko beam in frequency domain

A Timoshenko beam is a particular case of the layered beam model presented in equation (1) only by one terms approximation in the transverse direction. The kinematic analysis is given by:

$$U(x,z,t) = z\gamma(x,t); \quad W(x,z,t) = w(x,t) \tag{9}$$

We obtain the well known Timoshenko beam equations [Timoshenko 1955]. For the case of steady state vibration:

$$\gamma = \gamma_0 e^{i\omega t} e^{ikx\sin\varphi},\ w = w_0 e^{i\omega t} e^{ikx\sin\varphi},\ q = q_0 e^{i\omega t} e^{ikx\sin\varphi} \tag{10}$$

we obtain

$$\left(\frac{(SGk_s)^2}{EJk_s^2 + SG - \rho I\omega^2} - SGk_s^2 + S\rho\omega^2 \right) w_0 = q_0,\ k_s = k\sin\varphi \tag{11}$$

Here E, G are stiffness constants. They are, in general, frequency dependent complex functions.

Acoustical properties

When a panel is excited acoustically, the frequency at which the speed of the forced bending wave in the panel is equal to the speed of the free bending wave in the panel is called the coincidence frequency. It is expected that the sound power transmission coefficient is very high at the coincidence frequency of the panel. An external excitation in the form of a plane sound wave at the angular frequency ω is assumed to be incident on the first face sheet layer. The sound power transmission coefficient is defined as the ratio of the intensity of the transmitted sound to the intensity of the incident sound. If I_i is the intensity of the incident sound wave and I_t is the intensity of the transmitted sound wave, the sound power transmission coefficient τ is defined by $\tau = I_i / I_t$.

The beam acts as a partition in air of specific acoustic impedance ρc, where ρ and c are the density and speed of sound in air. Also, a sound transmission loss, TL, is defined, which is $TL = 10\log(\tau^{-1})$ [Thambura and Sun 2002, Renji 2005].

$$\tau = \left|1 - i\frac{\Phi\cos\varphi}{2\rho_a c_a \omega}\right|^2; \ \Phi = \frac{(SGk_s)^2}{EJk_s^2 + SG - \rho I\omega^2} - SGk_s^2 + S\rho\omega^2 \tag{12}$$

Let us now consider some numerical examples. Let E = 200 MPa, G = 50 MPa, ρ = 200 kg·m^{-3}, h = 0.0254 m). The transmission loss function TL values are presented here for a light foam material beam as a function of non-dimensionalised frequency f/f_r, $f_r = \sqrt{\pi^2 E_0 H^2 / 3c_a{}^4 \rho}$. The TL is presented for various angles of incident sound waves (the angle φ of incidence is given in radians) in Figure 5.

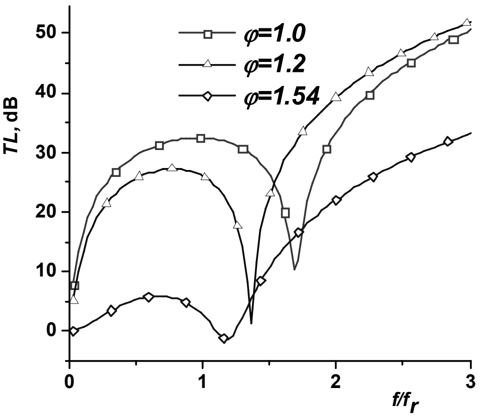


FIGURE 5. TL for various angles of incident sound wave as a function of non-dimensionalised frequency f/f_r
RYSUNEK 5. Zmniejszenie transmisji (ZT) dla różnych kątów φ padającej fali dźwiękowej jako funkcji bezrozmiarowej

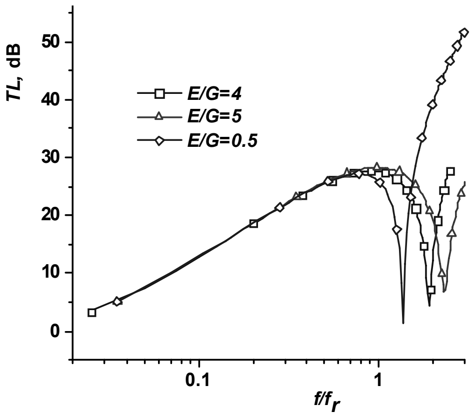


FIGURE 6. TL for various values of E/G (φ = = 1.2 rad)
RYSUNEK 6. ZT dla różnych wartości E/G (φ = = 1.2 rad)

In the Figure 6 the TL is presented for various values of ratio E/G. The influence of damping is presented in Figure 7. The independent bending mode and shear mode of viscous damping are considered

$$E = E_0(1 + i\omega DempE), G = G_0(1 + i\omega DempG) \tag{13}$$

($\omega DempE$, $\omega DempG$ – frequency linear depended imaginary parts of complex modules E and G, presenting viscous damping).

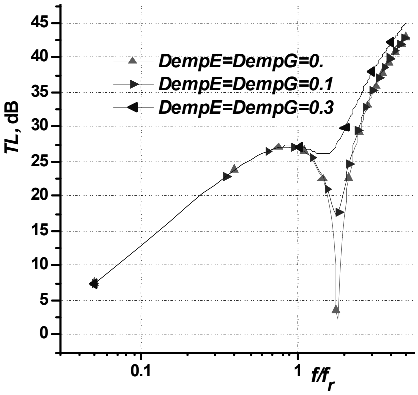


FIGURE 7. Damping dependent TL
RYSUNEK 7. Wpływ intensywności tłumienia na ZT

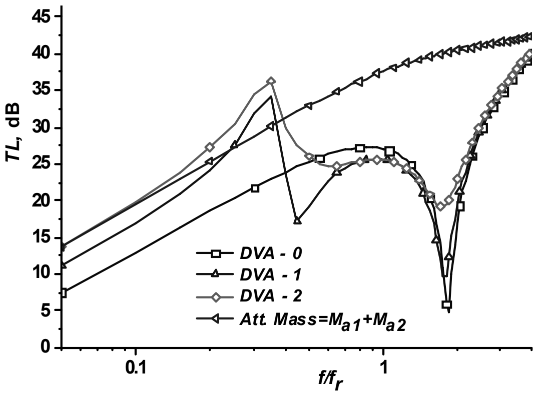


FIGURE 8. Influence of DVA's number on TL
RYSUNEK 8. Wpływ ilości DWA na ZT

Beam with DVA

Let us now consider a layered beam with a locally attached DVA [Korenev and Reznikow 1993]. Taking into account only the first bending mode type of vibration we obtain a similar set of equations as in equation (7). Only one additional equation for the DVA is needed:

$$-M_A\omega^2 w_A + (K_A + i\omega C_A)(w_A - w) = 0 \tag{14}$$

The influence of DVA's number on *TL* is presented (Fig. 8). Four cases are presented: 1 – beam without the DVA's; 2 – one DVA; 3 – two DVA's; 4 – DVA's masses rigidly connected to beam. The last case present "mass law" for the heavy beam. The better sound transmission panel properties may be seen for the DVA's system.

Concluding remarks

The present paper is a first attempt at proposing a novel procedure to derive the damping and sound insulation parameters for sandwich plates with the presence of a DVA. The main advantage of the present method is that it does not rely on strong assumptions about the model of the plate. The parameter dependent FRF and damping are presented for a three-layer beam. The results of this paper have shown that the presence of a DVA causes a decrease in the sound transmission in the low-frequency range. In the future, the extension of the present approach to various layered plates with various micro- and macro-inclusions will be performed in order to investigate various experimental conditions.

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Summary

Acoustical properties of layered beams with attached dynamic vibration absorbers. This study aims to predict the sound transmission properties of composite layered beams structures with the system of dynamic vibration absorbers (DVA's). The effective stiffness constants of equivalent to lamina Timoshenko beam and their damping properties have been determined by using a procedure based on refined numerical schemes and eigen-frequencies comparison. Numerical evaluations obtained for the vibration of the equivalent Timoshenko beam have been used to determine the sound transmission properties of laminated composite beams with the system of DVA's.

Streszczenie

Właściwości akustyczne belek wielowarstwowych z przyłączonymi dynamicznymi absorberami. W artykule zostały opisane wyniki badań dźwiękochłonnych właściwości kompozycyjnych wielowarstwowych struktur z systemem dynamicznych wibracyjnych absorberów (DWA). Stosując procedury usciślonych numerycznych schematów i porównania własnych częstotliwości, wyznaczone zostały efektywne stałe sztywności i tłumienia drgań belki Timoshenki odpowiednej do wielowarstwowej płyty typu sandwicz. Otrzymane dla ekwiwalentnej belki Timoshenki numeryczne oceny wibracji wykorzystane zostały do determinacji dźwiękochłonnych właściwości wielowarstwowej kompozycyjnej belki z systemem DWA.

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