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Sound transmission of sandwich beams with the dynamic vibration absorbers
Transmisja dźwięku poprzez płyty warstwowe z dynamicznymi absorberami drgań

Key words: sound transmission, composite layered beams, dynamic vibration absorber, Timoshenko beam, sound absorption properties, optimization of beams – DVA's system
Słowa kluczowe: transmisja dźwięku (pochłanianie hałasu), belki kompozytowe warstwowe (płyty), dynamiczny absorber drgań (DAD), belka Tymoszenki, właściwości dźwiękochłonne, optymalizacji systemu płyta – DAD

Introduction

Noise and vibration are of concern with many mechanical systems including industrial machines, home appliances, transportation vehicles, and building structures (Randall, 2009, Roozen et al., 2009, Tuma, 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 et al., 2008). In some cases of forced vibration, the passive damping 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 (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 (Conlon et al., 2009).

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 et al., 1994). Magnetic and particle vibration dampers can have considerable weight penalties. Passive damping using viscoelastic materials (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 et al. 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, reactive passive devices can increase the broadband (15–1000 Hz) sound transmission loss by about 10 dB. The identification of the elastic properties of laminated plates from the measured eigen-frequencies has been performed.

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 et al., 2008a, b). The elastic constants of laminates have been determined by using an identification procedure based on experimental design, and a multilevel theoretical approach (Diveyev et al., 2009).

Adaptive plate cylindrical bending equation

Let us consider a symmetrical three--layered beam in dynamic bending (Fig. 1) and analyze the cylindrical bending of a symmetric three-layer plate of length (L) and thickness ($2Hp$), assuming the following kinematic approximations $U = U_e \cup U_d$

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

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

$$\sigma_{xx} = C_{xx}\varepsilon_{xx} + C_{xz}\varepsilon_{zz}$$
$$\sigma_{zz} = C_{zx}\varepsilon_{xx} + C_{zz}\varepsilon_{zz} \qquad (2)$$
$$\tau_{xz} = G\gamma_{xz}$$

By substituting Eqs. (1) and (2) into the following Hamilton-Ostrogradsky variation equation

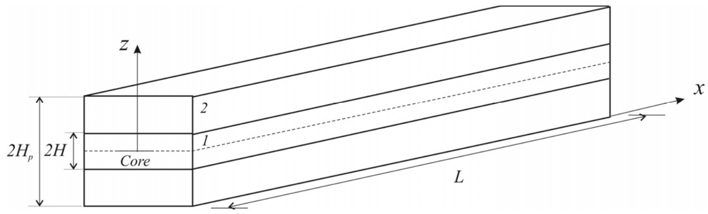


FIGURE 1. Sandwich beam scheme
RYSUNEK 1. Schemat belki warstwowej

$$U_e - \begin{cases} u = \sum\limits_{i,k} u_{ik}{}^e z^{2i-1}\varphi_k(x) \\ w = \sum\limits_{i,k} w_{ik}{}^e z^{2i-2}\gamma_k(x) \end{cases}$$

$$0 < Az < H$$
$$0 < x < L$$

$$U_d - \begin{cases} u = \sum\limits_{i,k} u_{ik}{}^d (z-H)^i \varphi_k(x) \\ w = \sum\limits_{i,k} w_{ik}{}^d (z-H)^i \gamma_k(x) \end{cases} \qquad (1)$$

$$H < z < H_p$$
$$0 < x < L$$

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

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

where:
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.

Assuming also single frequency vibration ($u_{ik}^e = \bar{u}_{ik}^e e^{i\omega t}$, $w_{ik}^e = \bar{w}_{ik}^e e^{i\omega t}$, $u_{ik}^d = \bar{u}_{ik}^d e^{i\omega t}$,

$w_{ik}^d = \bar{w}_{ik}^d e^{i\omega t}$) we obtain the set of linear algebraic equations for the amplitudes (Diveyev et al., 2008, 2009)

$$[A]\bar{U} = \begin{bmatrix} A_1 & A_d \\ A_d^T & A_2 \end{bmatrix} \begin{bmatrix} \bar{U}_e \\ \bar{U}_d \end{bmatrix} = f \quad (4)$$

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

$$-\omega^2 [M]\bar{U} + i\omega [C]\bar{U} + [K]\bar{U} = [A]\bar{U} = \bar{f} \quad (5)$$

This is the traditional frequency domain method which is normally used in linear elastic system investigations (Li et al., 2005).

Transition to the Timoshenko beam

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

$$U(x,z,t) = z\,\gamma(x,t), \quad W(x,z,t) = w(x,t) \quad (6)$$

The Timoshenko beam dynamic equilibrium equations are

$$\begin{aligned} & EI\frac{\partial^2 \gamma}{\partial x^2} - SG\left(\frac{\partial w}{\partial x} + \gamma\right) + \rho I \frac{\partial^2 \gamma}{\partial t^2} = 0 \\ & SG\left(\frac{\partial \gamma}{\partial x} + \frac{\partial^2 w}{\partial x^2}\right) - \rho S \frac{\partial^2 w}{\partial t^2} = q \end{aligned} \quad (7)$$

where:
I – denotes moment of inertia,
S – value of the beam cross-section area.

Let us consider a three-layered symmetrical beam (Fig. 1). Details of beam modelling are presented in Diveyev et al. (2008, 2009). 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 module 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 (fibre-composite material: $C_{xx} = 43$ GPa; $C_{xz} = 6$ GPa; $G = 0.6$ GPa; $\rho = 2000$ 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

$$C = \min \sum_{E_T, G_T} \left| f_S^i - f_T^i\left(E_T, G_T\right) \right|$$

in the frequency range (8)

$$f_k - \Delta_k/2 < f < f_k - \Delta_k/2$$

The error function (C) is chosen in the form of deviation of sandwich vibration eigen-frequencies $\left(f_S^i\right)$ from Timoshenko beam values $\left(f_T^i\right)$ of vibration eigen-frequencies. Here E_T, G_T are the Young and shear modules of Timoshenko beam. They change in some intervals

$$\begin{aligned} & E_0 - \Delta_E/2 < E_T < E_0 - \Delta_E/2 \\ & G_0 - \Delta_G/2 < G_T < E_0 - \Delta_G/2 \end{aligned} \quad (9)$$

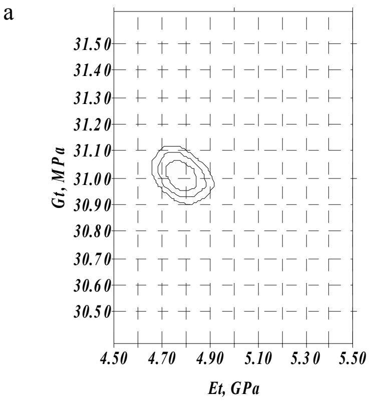
Were E_0, G_0 are a priory values of this coefficients. The result of translation for modules and frequency response function (FRF) is presented in Figure 2.

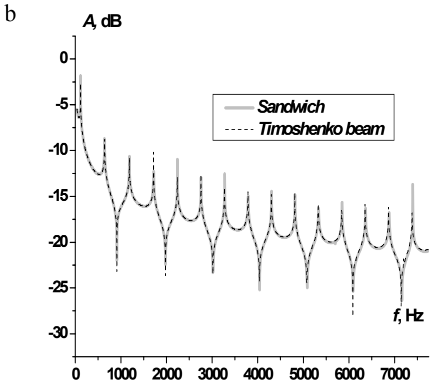
Only in higher frequency range (Fig. 2c) distinctions appear.

$$\eta = \frac{\eta_1 [q]^T |A_1||q| + \eta_2 [q]^T |A_2||q| + ... + \eta_N [q]^T |A_N||q|}{[q\ ]^T |A||q|} \quad (10)$$

Here:

$|A|$ – stiffness matrix,





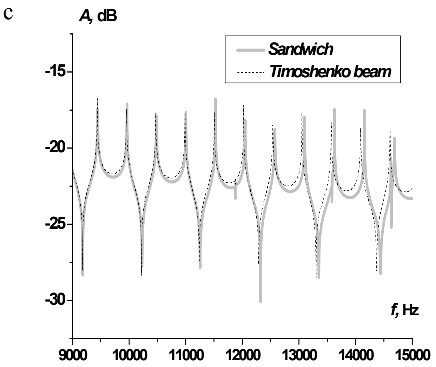


FIGURE 2. The result of sandwich translation to the Timoshenko beam: (a) – equivalent beam modules Et, Gt; (b) – FRF for sandwich beam and the equivalent uniform beam; (c) – the FRF's in higher frequency range

RYSUNEK 2. Wyniki transmisji belki warstwowej do belki Timoszenki: (a) – module ekwiwalentne belki Et,Gt; (b) – wykresy amplituda–częstotliwość drgań (WACD) dla belki warstwowej oraz belki Timoszenki; (c) – WACD dla wysokich częstotliwości drgań

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 $C_i = \eta_i [K_i]$)

$|q|$ – vector of the displacement component,

$|A_i|$ – matrix stiffness component corresponding to the i-th layer ($|A| = \sum_i |A_i|$).

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; $C_{xz} = 40$ MPa; density $\rho = 240$ kg/m^3) and rigid face layers (fibres composite material: $C_{xx} = 47$ GPa; $C_{xz} = 6$ GPa; $G = 0.6$ GPa, $\rho = 2000$ kg/m^3) for various sandwich geometry and various approximation order N_Z, Eq. (1) (in transverse direction) are presented in Figure 3. The corresponding FRF's are also presented.

the damping increases first with the frequency and stay constant after some value of frequency. For rigid core sheet and damping soft face sheets (Fig. 3b) the damping increases linearly.

Acoustical properties

When a panel is excited acoustically, the frequency at which the speed

a b

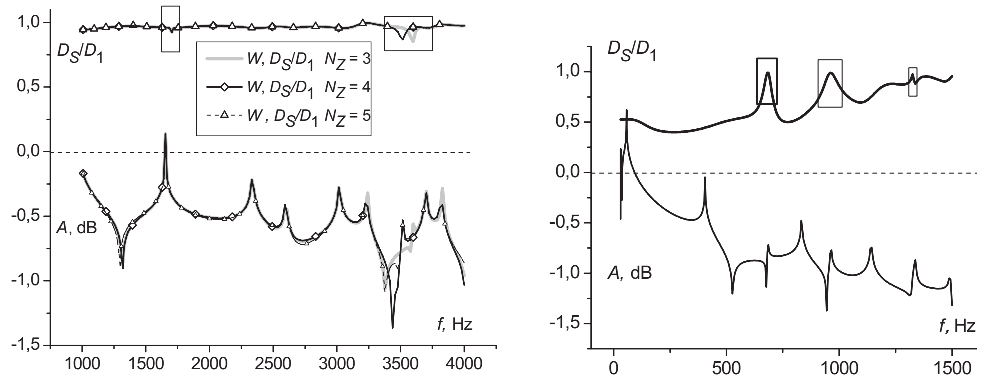


FIGURE 3. Frequency dependent damping for the sandwich beam: (a) – rigid face sheets; (b) – rigid core

RYSUNEK 3. Zależności absorpcji hałasu od częstotliwości drgań dla różnych parametrów belki warstwowej

Here D_S/D_1 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 ($D_S/D_1 = \eta_S/\eta_1$). For $N_Z \geq 3$ the calculating dynamic properties are practically identical, and for $N_Z \geq 4$ also in the higher frequency range.

Damping variations may be seen. For the sandwich with the rigid face sheets and soft damping core (Fig. 3a)

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.

Consider a panel with an incident sound field (Fig. 4).

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

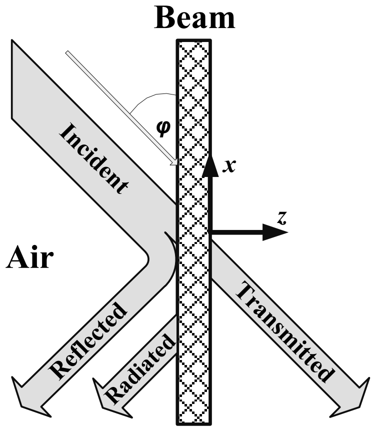


FIGURE 4. Beam in plane sound field (plane waves)
RYSUNEK 4. Belka w polu dźwiękowym płaskim

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 $\tau = I_i / I_t$.

Let us consider study state harmonic vibrations of Timoshenko beam

$$\begin{aligned} \gamma &= \gamma_0 e^{i\omega t} e^{ikx\sin\varphi} \\ w &= w_0 e^{i\omega t} e^{ikx\sin\varphi} \\ q &= q_0 e^{i\omega t} e^{ikx\sin\varphi} \end{aligned} \tag{11}$$

By translating (7) we obtain in the frequency range

$$\begin{aligned} &\left(\frac{\left(SGk_s\right)^2}{EIk_s^2 + SG - \rho I \omega^2} - SGk_s^2 + S\rho\omega^2 \right) w_0 = q_0 \\ &k_s = k \sin\varphi \end{aligned} \tag{12}$$

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

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 $TL = 10 \log (\tau^{-1})$.

The net sound pressure – q is (Renji, 2005)

$$q = p_i + p_r + p_{rad} - p_t \tag{13}$$

where:
p_i – incident,
p_r – reflected,
p_{rad} – radiated,
p_t – transmitted wave pressure.

We assume the same form for all of the sound pressures and for the displacements (on the plate surfaces):

$$\begin{aligned} p_i &= p_{i0} e^{i\omega t} e^{ikx\sin\varphi} \\ p_r &= p_{r0} e^{i\omega t} e^{ikx\sin\varphi} \\ p_{rad} &= p_{rad0} e^{i\omega t} e^{ikx\sin\varphi} \\ p_t &= p_{t0} e^{i\omega t} e^{ikx\sin\varphi} \\ w &= w_0 e^{i\omega t} e^{ikx\sin\varphi} \end{aligned} \tag{14}$$

Since the medium present on both sides of the plate has the same properties, the sound powers radiated to both sides of the panel are equal and hence $p_t = p_{rad}$. The requirement of the continuity of the particle velocity necessitates that $p_i = p_r$. Using the above results and substituting Eq. (14) in Eq. (13), the external force on the plate due to the acoustic excitation becomes

$$q = 2(p_i - p_t) \tag{15}$$

The amplitude of the displacement of the plate is related to the amplitude of the transmitted sound wave by the expression (Renji, 2005, Thamburaj and Sun, 2009)

$$w = \frac{p_t \cos\varphi}{i\omega\rho_a c_a} \tag{16}$$

where:
c_a – sound velocity,
$\rho_a = \rho H$ – beam density,
$i = \sqrt{-1}$.

Substituting Eqs (13)–(16) into Eq. (12) yields:

$$\tau = \left| 1 - i\frac{\Phi\cos\varphi}{2\rho_a c_a \omega} \right|^2$$

$$\Phi = \frac{\left(SGk_s\right)^2}{EJk_s^2 + SG - \rho I\omega^2} - SGk_s^2 + S\rho\omega^2 \tag{17}$$

Let us now consider some numerical examples. Let E = 200 MPa, G = = 50 MPa, ρ = 200 kg/m^3, h = 0.254 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.

In the Figure 6 the TL is presented for various values of E/G. The influence of damping is presented in Figure 7.

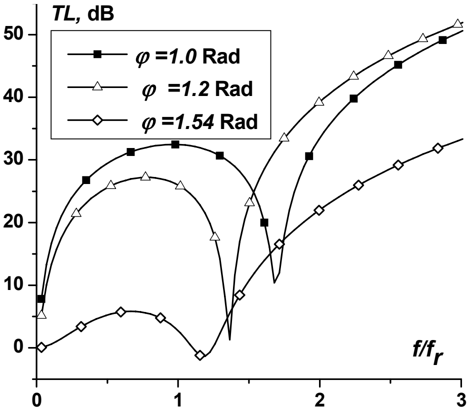


FIGURE 5. TL for various angles (φ) of incident sound wave as a function of non-dimension frequency (f/f_r)
RYSUNEK 5. TL dla różnych kątów nachylenia dźwiękowej fali jako funkcji częstotliwości

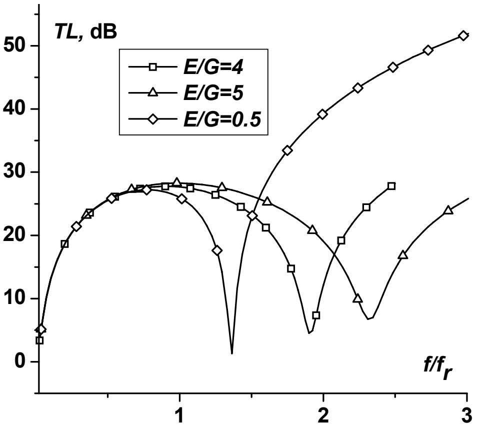


FIGURE 6. TL for various values of E/G (φ = 1.2 rad)
RYSUNEK 6. TL dla różnych parametrów E/G (φ = 1.2 rad)

The independent bending mode and shear mode of viscous damping are considered

$$E = E_0\left(1 + i\omega DempE\right)$$
$$G = G_0\left(1 + i\omega DempG\right) \tag{18}$$

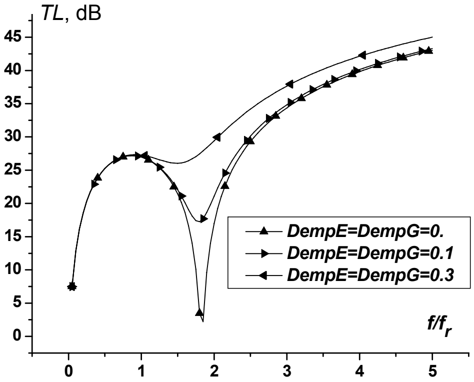


FIGURE 7. Damping dependent *TL*
RYSUNEK 7. Absorpcyjna zależność *TL*

ω DempE, ω DempG – frequency linear depended imaginary parts of complex modules *E* and *G*, presenting viscous damping.

In Figure 8 *TL* is presented for the Timoshenko beam for the Eqs (7) and (13) and for equations without the beam normal angular inertion term. In Figure 9 may be seen the greater influence of anisotropy on the *TL* determined by different Timoshenko beam theories. Near the *TL*

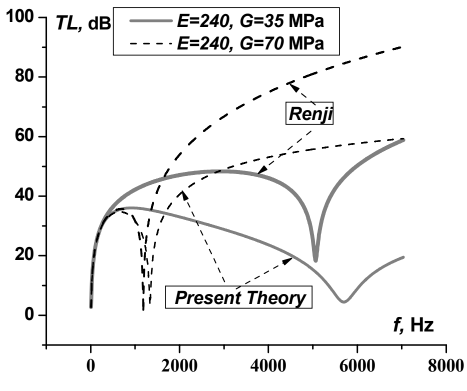


FIGURE 8. *TL* for the uniform Timoshenko beam with the beam angular inertion term (solid lines) and without (dot lines)
RYSUNEK 8. Różnorodne zależności *TL* dla belki Timoszenki

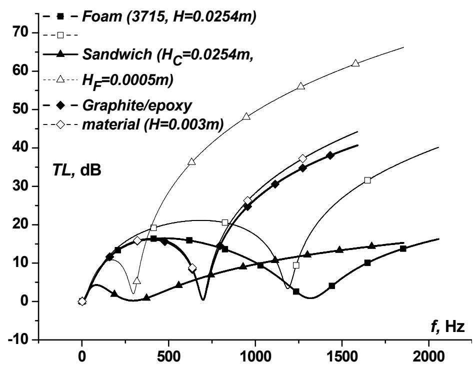


FIGURE 9. *TL* for different Timoshenko beam with the beam normal angular inertion term (solid lines) and without (dot lines)
RYSUNEK 9. Różnorodne zależności *TL* dla belki Timoszenki

is presented for various beam structure and different Timoshenko beam theory.

Parameters are of beam layers are defined in former chapter. The great difference may be seen for different Timoshenko beam theories, especially for composite or sandwich beams.

Acoustical properties of beam with DVA

Damped DVA's are used to provide energy dissipation, thereby motivating the term "absorber". These realistic absorbers furthermore reduce their sensitivities to parameter variations from optimal values and reduce the primary system motion at its resonance frequencies, while increasing their effective bandwidth, as compared to undamped examples. Consider the viscously damped DVA with elastic and viscous damping elements, used between the masses. Detailed descriptions of the fundamentals of such DVA's are given in.

Let us now consider a layered beam with a locally attached DVA (Fig. 10):

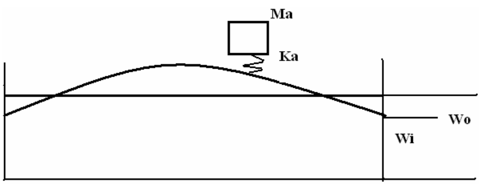


FIGURE 10. Beam with the DVA
RYSUNEK 10. Belka z DAD

MA – DVA mass; KA – DVA clamping rigidity. Taking into account only the first type of vibration we obtain a similar set of equations as in Eq. (7). Only one additional equation for the DVA is needed

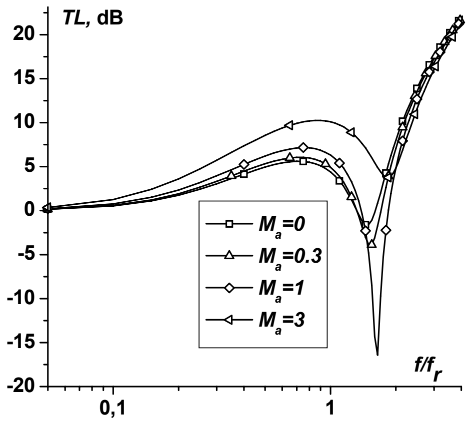
$$-M_A\omega^2 w_A + (K_A + i\omega C_A)\left(w_A - w\right) = 0 \tag{19}$$

The *TL* is presented for various DVA parameters in Figure 11.

The influence of the DVA mass on the *TL* is presented in Figure 11a. The influence of the damping of the DVA is presented in Figure 11b. The influence of DVA's number on *TL* is presented Figure 12.

Four cases are presented: 1 – beam without the DVA's; 2 – one DVA; 3 – two DVA; 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. In Figure 13 DVA constructed for the most dangerous case in beam resonant zone, the frequency of sound and own beam frequency is shown.

a



b

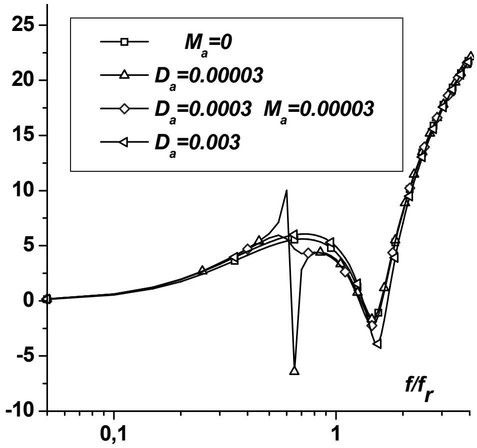


FIGURE 11. (a) – mass influence of the DVA; (b) – damping influence of DVA
RYSUNEK 11. Zależność pochłaniania hałasu od masy DAD (a) i absorpcji DAD (b)

Conclusions

In the present study, theoretical models for investigations into the vibrations and damping of layered composite plates are developed. A rational approximation of the field of displacements is established, which al lows one, at a small number of parameters, to predict the dynamic behavior of a beam. Based on this model, for a three-layer composite beam, not only the damping from the shear deformation in the core, but also the damping associated with the normal and bending deformations of layers, which is of

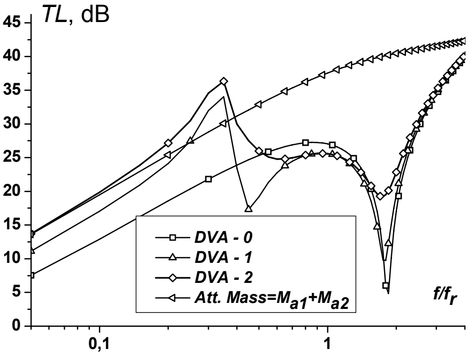


FIGURE 12. Influence of DVA's number on *TL*
RYSUNEK 12. Parametry pochłaniania hałasu w zależności od ilości DAD

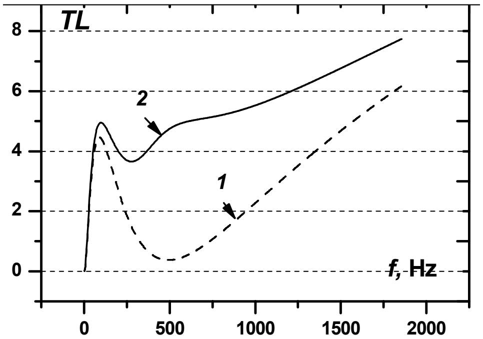


FIGURE 13. Influence of DVA in beam resonant zone on *TL* (1 – without DVA; 2 – with DVA)
RYSUNEK 13. Rezonans belki bez DAD (1) i z DAD (2)

importance for analyzing their damping properties under vibrations with moderate and high frequencies, was investigated. A new procedure for determining the parameters of the dynamic rigidity of three-layer plates is suggested, which was used to find the equivalent values of elastic modules for a Timoshenko beam. We should note that the method presented does not require rigorous assumptions concerning the plate model.

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

Sound transmission of sandwich beams with the 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 multi-level numerical schemes and eigen-frequencies comparison.

The strategy of an anisotropic beam to the Timoshenko beam seem to be such: the raw of models can be applied at different vibration or static conditions of the plate by a suitable analytical ore approximation method, research of sensitiveness in relation to the parameters of fixing and material anisotropy, numerical experiments on identification of elastic modules, practical module identification by exploring different schemes of experimental setup and, finally, posterior analysis of identification quality. The combined method of identification was proposed on the basis of the simultaneous use of information on a homogeneous beam and beam with an internal layer, with identical mechanical properties to the homogeneous beam.

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. The optimization of beams-DVA's system sound absorption properties is performed in the low frequency range.

Streszczenie

Transmisja dźwięku poprzez płyty warstwowe z dynamicznymi absorberami drgań. W artykule zostały przebadane procesy pochłaniania hałasu w kompozytowych płytach warstwowych wyposażonych w dynamiczne absorbery drgań (DAD). Skuteczne współczynniki sztywności belki równoważnej do belki Tymoszenki i jej właściwości absorpcyjne zostały określone analitycznie przy użyciu wielopoziomowych systemów liczbowych i przez porównanie ich własnych częstotliwości drgań.

Porównanie belek anizotropowych z belką Timoszenki przeprowadzono w następujący sposób: modele porównywały się dla różnych dynamicznych i statycznych właściwości płytek stosując metody analityczne i aproksymacyjne, badała się korelacja parametrów mocowania belki i anizotropii jej materiału, doświadczalnie ustalone zostały wartości modułów sprężystości, które uściślono w trakcie badań różnych schematów doświadczalnych instalacji, a następnie analizowano dokładność określania parametrów. Połączona metoda określania parametrów przewidywała analizę porównawczą jednorodnych i warstwowych płyt (płyty z wewnętrzną warstwą) o identycznych właściwościach mechanicznych.

Wyniki liczbowe otrzymane w badaniu drgań równoważnej belki Timoszenki zostały wykorzystane do określenia parametrów kompozytów warstwowych płyt izolacji akustycznej wyposażonej w systemy DAD. Optymalizację właściwości izolacji akustycznej systemu płyta – DAD przeprowadzano w zakresie niskich częstotliwości drgań.

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