Reducing the vibration excitability of a metal plate by applying variable vibrodamping inserts

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Abstract. Thin-walled steel structures of process equipment and vehicles composed of plates are constantly agitated and become powerful sources and conductors of acoustic vibration. To reduce acoustic vibration, special vibrodamping coatings are usually applied to the plate surface by gluing or spraying. However for many engineering applications, concerned with reduction of acoustic vibration, the use of vibrodamping coatings is limited or impossible for a number of reasons, namely, the variability in geometric surface configuration of thin-walled steel structures, stringent restrictions of structure weight (e.g. in hovercrafts) and others. An alternative solution is the use of piece vibrodamping rubber inserts, applied to the plate through perforation. The inserts can be made either solid or with inner channel for free shift of extra weight in the form of a ball. The paper summarizes the results of experimental studies on reducing the loss factor for vibration excitability of plates with inserts.

1 Introduction

Thin-walled metal structures (TMS) are widespread in industry and transport, which include the bodies of transport and special vehicles, shells of self-propelled and stationary agricultural machinery, elements of ship structures, technological doors, protective covers and others. These structures can be the source of intensive noise, which is generated by their vibrational excitation.

The presented study is aimed at solving the problem of reducing the noise from vibration-excited TMS, i.e. structural elements of technological equipment and transport-technological means.

Being sound emitters, TMS, composed of plates, have a number of characteristics. When metal plates are excited, multiple natural resonances occur in them. When the frequency of a periodic excitation force coincides with one of the natural frequencies, resonance vibrations occur, accompanied by intense noise emission [1, 2].

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In an excited plate, the predominantly propagating waves are bending and, to a lesser extent, longitudinal ones.

The propagation velocity of longitudinal waves c_1 has no dispersion, i. e. their phase velocity is constant and independent of frequency [1]:

$$c_1 = \sqrt{\frac{E}{\rho(1-\mu^2)}}, \tag{1}$$

where E is the Young's modulus of the plate material; ρ is density of the plate material; μ is Poisson's ratio.

Bending waves c_2 propagate much slower than longitudinal ones. In bending vibrations, the speed of propagation is dispersive, hence, it depends on the vibration frequency of the bending wave ω [1]:

$$c_2 = \sqrt{\omega}\sqrt[4]{\frac{D}{m}}, \tag{2}$$

where $D = \frac{Eh^3}{12(1-\mu^2)}$ is cylindrical stiffness of the plate; h is thickness of the plate; $m = \rho h$ is mass of the plate per unit area.

The emergence of acoustic vibrations in the TMS is due to propagation of bending waves in them.

The wave pattern in shell-shaped TMS is more complex than in plates.

The nature of sound emission by plates differs significantly when below and beyond its boundary frequency f_b [1]:

$$f_b = \frac{c^2}{2\pi}\sqrt{\frac{m}{D}}. \tag{3}$$

At the boundary frequency of the plate, its bending wave velocity c_2 is equal to the speed of sound in the air c (i.e., $c_2 = c$); below the boundary frequency ($f < f_b$), the bending wave velocity in the plate is lower, and beyond the boundary frequency ($f > f_b$), it is higher. Beyond the boundary frequency, the plate emits sound more efficiently than below it. When the TMS is excited, the dominant sound emission occurs in the frequency range $f > f_b$.

The most rational and effective way is noise reduction in the source by damping the sound vibration in the TMS. This approach results in a reduction of surface vibrations and sound emission by vibrating surfaces both in the immediate vicinity of the object and indoors, as well as in open spaces.

Vibrodamping is the process of reducing sound vibration by converting the energy of mechanical vibration into heat energy. A measure of vibrodamping is the loss factor η, characterized as the ratio of energy, absorbed in the system per vibration cycle (W_2), to the maximum potential energy in the system (W_1) [3, 4]:

$$\eta = \frac{1}{2\pi}\frac{W_2}{W_1}. \tag{4}$$

Vibrodamping coatings (VDC) [3, 4, 5, 6, 7, 8, 9, 10] are widely used to reduce the intensity of sound emission of vibration-excited thin-walled metal structures. A whole scientific trend has emerged, namely, vibrodamping of sound vibrations in engineering structures and vibrodamping materials.

Professor A.S. Nikiforov, one of the foremost experts in the field of vibrodamping, predicted the creation of piece vibrodamping devices with simple fixation on vibrating structures in the foreseeable future [4].

Analyzing the advantages and disadvantages of existing vibrodamping devices and coatings, and proceeding from the conditions of the specified problem of reducing noise from special thin-walled metal structures (e.g. the bodies of transport and technological vehicles, protective covers and guards, technological doors, etc.) and the criterial requirements put forward, a new technical solution has been proposed, based on the synthesis of the described vibrodamping means, which consists in the application of special piece vibrodamping inserts

(PVI), manufactured from rubber [11]. The PVI are manufactured as piece-through inserts and are firmly fixed to the surfaces of perforated thin-walled metal structures, without any extra equipment. This is ensured by a circular cross groove on the insert body, which, due to the resilient properties of rubber, firmly covers the edges of the through hole of the TMS perforation. The inserts can be fitted either manually or using special automated equipment.

Before installation of PVI, the surface of the vibroexcited structure is prepared by perforation, or the TMS is perforated during the manufacturing process.

There is no theoretical representation of vibrational energy dissipation for a vibroexcited plate with PVI. The following assumption (hypothesis) is made that vibrational energy dissipation in the «plate-PVI» system is related to:

- visco-elastic friction in the body of PVI, manufactured from rubber;
- dry friction, occurring along the surface of the PVI, contacting the edges of plate perforation [12];
- reducing the energy of longitudinal waves, due to PVI, acting as barriers to their propagation (due to heterogeneity of the medium of longitudinal wave propagation, i.e. alternation of metal with rubber).

It was also assumed that the physical picture of the dissipation of vibrational energy in the PVI plate is largely identical to the one in the system of the vibroexcited plate with a set of point local antivibrators, placed on its surface. The local antivibrator is understood as a vibrating system, consisting of a mass, a friction element and an elasticity element [3]. With the same extent of freedom, such a system possesses an increased mechanical resistance around a certain frequency, which is used for practical implementation of damping.

On the basis of the assumptions proposed, experimental laboratory tests have been carried out to determine the loss factor η_{Σ} in the systems of PVI plate and the one with a vibrodamping coating. Based on the results, a comparative evaluation of their performance was conducted.

2 Methods and materials

2.1 Methods

The main characteristic that determines the measure of vibration energy absorption in structural elements is the loss factor (η).

One of the methods of determining η is the one of measuring the width of the resonance response, which has become quite widespread. The essence of the method consists in determining the width of the resonance peak of oscillations in those points of the dynamic displacement curve, in which the dynamic displacement is a certain fraction of resonance dynamic displacements of the system [3, 13, 14, 15].

Thus, the loss factor value of interest equals to:

$$\eta = \frac{\Delta f}{f_0}, \tag{5}$$

where Δf is the width of the resonance peak at half-power points (with an amplitude decay of up to 0,707 or by 3 dB from the peak apex); f_0 is resonance frequency, Hz.

The parameter η is related to the logarithmic decrement $\overline{d}$ and the quality factor Q of the vibrating system by the following relationships:

$$\eta = \frac{\overline{d}}{\pi}; \tag{6}$$

$$\eta = \frac{1}{Q}. \tag{7}$$

Since systems with distributed parameters behave similarly to the ones with concentrated parameters at resonances, this method is applicable for measuring the loss factor of the plates at all resonance frequencies, except for high ones, where the relative density of resonance frequencies of the plates increases. In this case, the application of this method is impossible [13]. As, while measuring small values η, it was necessary to prevent possible loss of vibrational energy from the measured system through suspensions, the plate was suspended vertically on metal strings, where standing bending waves were excited. When the excitation frequency is changed, the plate resonates at frequencies, determined by its mechanical parameters. When the generator frequency setting is synchronized with the speed of calibrated paper in the data recorder, the levels of the plate's resonance curves will be automatically plotted on paper.

The Figure 1 represents a schematic block diagram of the measuring circuit to assess the total loss factor η_Σ in the system of a PVI plate or the one with a vibrodamping coating.

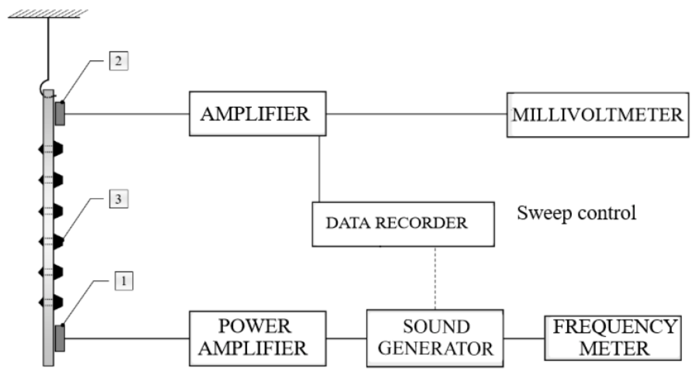

Fig. 1. The schematic block diagram for measuring the total loss factor η_Σ (1 – vibration exciter; 2 – vibration receiver, 3 – piece vibrodamping inserts).

2.2 Materials

According to methodology by Oberst H. [15], the experimental investigations involved the use of steel rods of 220x20x2 mm in size. The material of the rod (as a plate fragment) was «St-2» constructional steel, chosen as the most common one. The rods were manufactured in two versions:

- those with 100 mm diameter through-hole perforation and varying hole spacing for the rubber inserts;
- those without perforation for adhesive application of vibrodamping coatings.

The piece inserts were manufactured in 2 options:

- option 1: solid (Figure 2);
- option 2: with a cylindrical through channel for free inner movement of a metall ball of certain mass with positional friction. In these piece inserts, the channel inlets were closed with a self-adhesive film (Figure 3).

Polyvinylchloride (PVC) and fabric-based linoleum as well as special coatings VD-13 and VD-25 were used as vibrodamping coatings.

To assess the effect of plate perforation and the number of piece vibrodamping inserts on vibroacoustic characteristics, the following indicators have been introduced:

- «perforation rate»

$$P_n = \frac{S_o}{S_n} \cdot 100\%, \tag{8}$$

where S_0 is the total area of perforation in the plate; S_n is the area of the plate.

- «insertion rate»

$$P_b = \frac{S_b}{S_n} \cdot 100\%, \tag{9}$$

where S_b is the total area of perforation, occupied by inserts in the plate.

– «insert usage rate»

$$P_i = \frac{P_b}{P_n} = \frac{S_b}{S_o} \cdot 100\%. \tag{10}$$

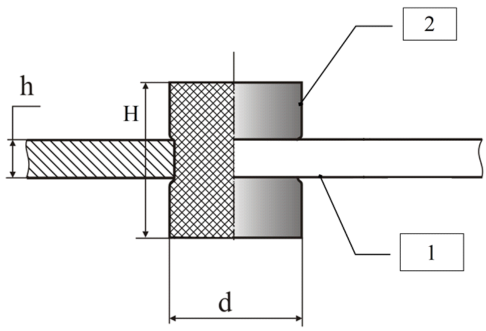

Fig. 2. Solid piece inserts: 1 – metal plate; 2 – vibrodamping insert; H – insert height; h – plate thickness; d – diameter of an insert.

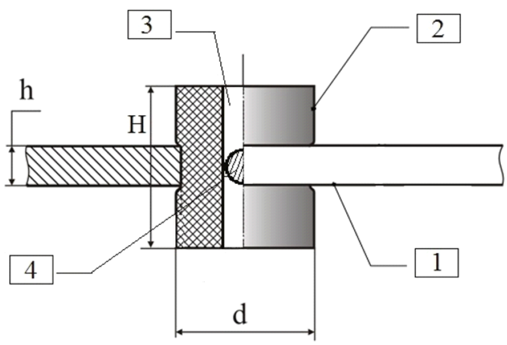

Fig. 3. Piece inserts with cylindrical through channel for free inner movement of a metal ball of certain mass with positional friction: 1 – metal plate; 2 – vibrodamping; 3 – cylindrical channel; 4 – metal ball; H – insert height; h – plate thickness; d – diameter of an insert.

The insert usage rate (P_i) was constant at 100%, while the perforation rate $P_п$ and the insertion rate P_b varied from 1 to 10%.

The diameter of perforation holes was constant and equalled 10 mm. The perforation holes were evenly distributed across the surface of the metal plate.

To investigate the impact of temperature upon variation in the loss factor of the PVI plate, a thermal chamber was used. Based on actual operating conditions of vibrodamping inserts for thin-walled metal structures of vehicles and stationary equipment, the operating temperature range was selected as -20 ºC - +60ºC.

3 Experimental results

3.1 The impact of vibration frequency on damping effectiveness

The results of the experimental study of vibration frequency (f, Hz), impacting the damping properties of the plate with fixed surface inserts are presented in Figure 4 (for various values of insertion rate, P_b, %). The research has been conducted with the inserts, manufactured according to option 1 (Figure 2).

From the analysis of the experimental results obtained, the evident dependence of the total loss factor (η_Σ) upon frequency (f, Hz) across the whole frequency region under study is clearly marked. Moreover, η_Σ increases with rising insertion rate (P_b) (Figures 4, 5), however, the type of curves does not change, depending on frequency only.

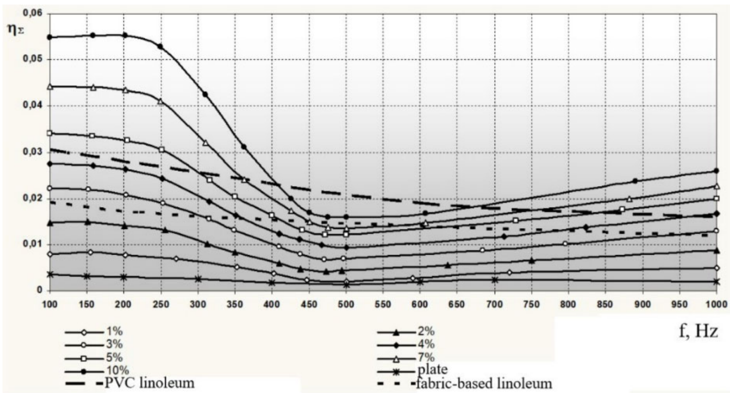

Fig. 4. The dependence of the total loss factor of the plate with damping inserts (η_Σ) upon frequency (f) and insertion rate (P_b).

In the frequency range presented ($100 \leq f \leq 1000$ Hz), the curves η_Σ with various values P_b reach their maximum values at frequencies of 100 – 250 Hz, and their minimal values are reached at frequencies of 450 – 550 Hz.

It can be seen that the character of the curves of η_Σ variation for piece inserts is identical. For vibrodamping coatings, the variation of η_Σ as a function of frequency is smoother, with a tendency to decline with the rise in the excitation frequency.

3.2 The impact of geometrical dimensions of vibrodamping inserts upon damping effectiveness

It is well known that the mass of a structure is one of the significant characteristics of vibrating processes. As far as damping inserts are concerned, their mass depends on the material density and their own dimensions.

For the whole series of experimental studies, the density of rubber as a material for PVI manufacture was assumed constant, equalling 1400 kg/m^3, and the diameter of the perforation hole in the plate was selected to be fixed, equalling 10 mm. Given the cylindrical shape of the PVI, their height (h_b) was their only variable parameter. The entire series of experiments in this phase of study was conducted on perforated steel plate with an insertion rate (P_b) of 5%.

The samples of vibrodamping inserts of 8, 12 and 16 mm in height h_b have been selected.

The variation of damping properties of the plate for various values of h_b as a function of the vibration frequency (f) is presented in Figure 6.

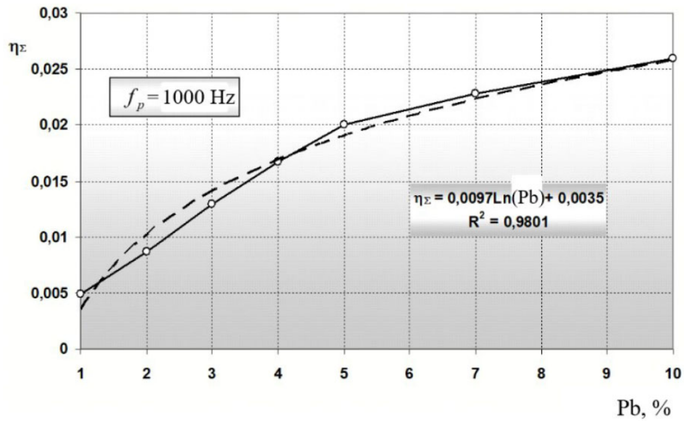

Fig. 5. The effect of the insertion rate (P_b) upon the total loss factor of the plate(η_Σ).

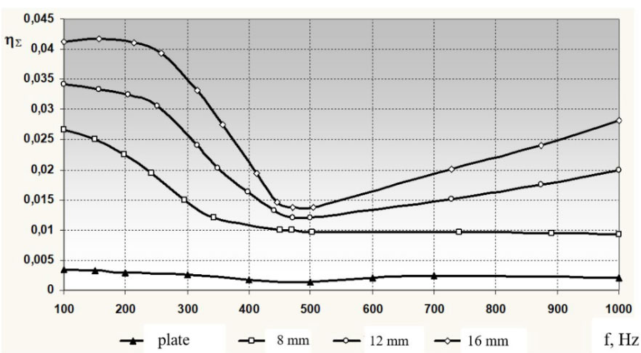

Fig. 6. The dependence of variation in the total loss factor of the plate (η_Σ) on the height of damping inserts (h_b).

It can be seen that the pattern of change in the curves is identical, regardless of the inserts' height. The figure also presents the curve of η_Σ variation for an insert-free plate as a function of frequency *f*. In this case, the value η_Σ increases over the whole range of vibration frequencies, with the rise in the height of inserts h_b and, consequently, in their mass.

3.3 The impact of a dynamically movable ball in the insert channel upon the effectiveness of damping

For this phase of experimental research, the inserts were used, manufactured according to option 2 (Figure 3). The results are presented in Figures 7 and 8.

It can be seen that the behaviour of the graphical dependencies of the loss factor η_Σ on the frequency *f* is practically unchanged.

With the increase in the ball diameter *d* and its mass, the total loss factor η_Σ grows across the entire frequency range. The dependence is almost linear in character (Figure 8).

However, the most significant growth in the η_Σ value across the entire frequency range is observed with increasing number *N* of same-sized balls (1 to 3) in the insert channel (Figure 9).

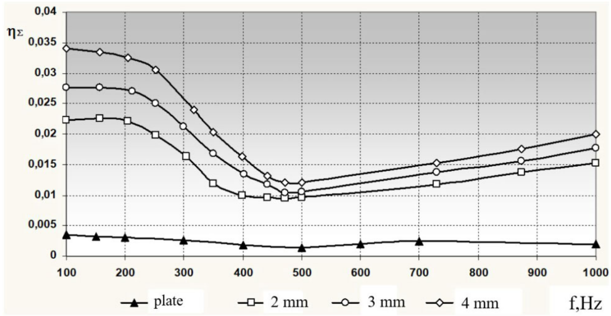

Fig. 7. The dependence of variation in the total loss factor of the plate (η_Σ) on the ball diameter D_s and mass.

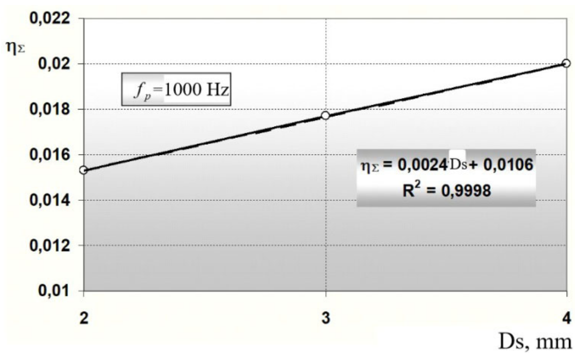

Fig. 8. The impact of the ball diameter (D_s) on the value of the total plate loss factor (η_Σ).

3.4 The comparative analysis of the impact of the temperature factor on the effectiveness of vibration damping by vibrodamping coatings and piece inserts

The experimental results have shown a much lower dependence of η_Σ in insert plates on temperature (Figure 10). Although there is a decreasing trend in η_Σ with rising temperature, this change is insignificant. In general, vibrodamping inserts demonstrate the stable value of the total loss factor η_Σ in the wide frequency range (-20ºC - +60ºC).

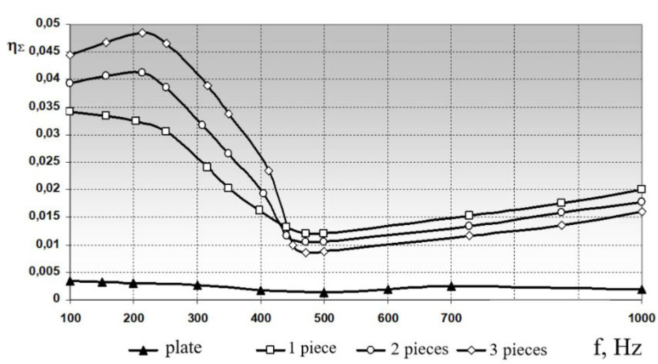

Fig. 9. The dependence of variation in the total plate loss factor (η_Σ) on the number of balls in the insert (N_s).

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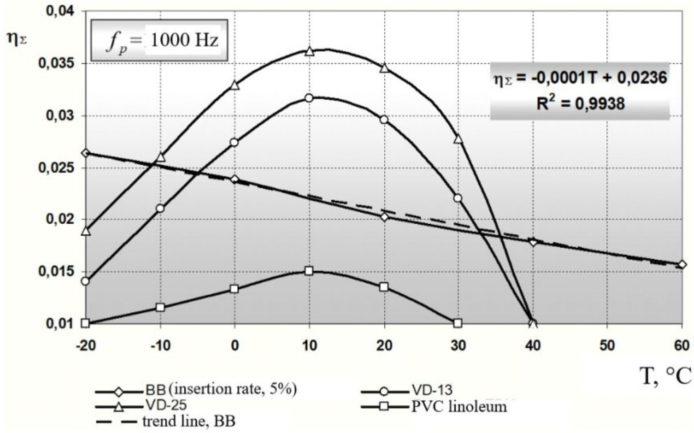

Fig. 10. The dependence of variation in the total loss factor (η_{Σ}) of various damping coatings on temperature (T, ^{0}C).

The effectiveness of vibrodamping coatings is extremely contingent on the temperature factor (Figure 10).

The maximum values of the total loss factor η_{Σ} of the vibrodamping coating are demonstrated at T=10ºC. With rising and lowering temperature, there is a tendency for a sharp decline in η_{Σ} value, and this kind of difference between temperature dependencies of loss factor η_{Σ} for piece inserts and vibrodamping coatings is observed in the entire frequency range of rod vibroexcitation (100 – 1000 Hz).

4 Conclusion

From the results of the laboratory experiments, certain conclusions can be made.

It has been proven that piece vibrodamping inserts can be just as effective in reducing the vibration excitability of thin-walled steel structures as vibrodamping coatings.

Vibrodamping inserts, while in some cases inferior to vibrodamping coatings in effectiveness, possess a number of advantages over the latter.

Firstly, vibrodamping coatings are virtually unaffected by the impact of temperature in the real temperature range of their operation.

Secondly, their use for engineering applications, imposing stringent weight limits for thin-walled metal structures (TMS), is, as a whole, obviously more advantageous.

The weight reduction is achieved by perforating the plate and not depositing the continuous vibrodamping coating on the TMS surface.

Thirdly, the possibility of locating freely moving elements with varying mass inside the piece vibrodamping inserts, hypothetically provides a choice of effective frequency range for their application. This fact enables us to claim a wider range of applications for vibrodamping inserts.

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