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Noise reduction of mechanisms by insulating vibrations

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Abstract

The paper deals with the dependence of parameters of noise generated by vibrating mechanisms. For that purpose analysis is made of the rational diagrams of force elements, which would ensure the reliability of constructions by reducing vibrations, the wave field whereof excites noise. Tasks are set in respect of vibroinsulation and sound insulation. Additional measures are foreseen to ensure the solution of the tasks set. Analysis of solutions performed give the opportunity for engineers, making use of the methods indicated in the paper, to identify measures for reduction of noise generated by mechanisms.
Keywords: vibroinsulation, noise, methods of reduction.

Introduction

Parameters of noise generated by mechanisms are directly related to mechanism vibrations and their wave fields that are generated by these mechanisms.

Reliability of load-bearing elements of the construction may be ensured by the choice of the rational diagrams of force, conditioned by the form of the construction and optimum location of the bearing members, performing mutual reservation [1]. The selected methods allow at the expense of the choice and the diagrams themselves and their elements to achieve the maximum effect at the preset restrictions concerning the mass and the overall dimensions of the construction.

The development engineer seeking to achieve optimum indicators of reliability may select the place of location of the given mechanism, where the levels of external actions are minimum, or use some protective devices.

Elementary methods of vibroinsulation have been widely applied since of old in various fields of technology in the case of protection against external effects and internal sources. The problem of vibroinsulation like of sound insulation is quite simple: on the way of wave energy propagation the additional device, reflecting or absorbing the definite portion of wave energy of mechanical vibrations, is placed. In addition, the main specifications of the construction must remain unchanged. The last requirement commonly is manifested in the form of different restrictions in respect of those mechanical devices concerning mass, overall dimensions, strength or other parameters.

Quite a number of measures exist, allowing to increase vibration reliability at the earlier stage as well, for example, already at the choice of the elements of mechanisms, possessing (owing to accidental circumstances) the higher strength, stability and reliability in respect of those effects, under which the element of the mechanism turned out to be.

Methods for solution of vibroinsulation

Application of various vibroinsulation and vibration absorbing devices at different restrictions of mass and overall dimensions of the construction requires the preliminary evaluation of the achieved effect of vibroinsulation. For that purpose, for consideration it is necessary to introduce the coefficient, which may be the quantitative measure of vibroinsulation [2]. For engineering evaluations, vibroinsulation may be characterized by the relation of energy, delivered to the insulated element of the mechanism, per time unit when using the vibroinsulating device, to the corresponding value, when the insulating device is absent. The shortcoming of such coefficient is its too general view and dependence on time. For practical purposes, it is more convenient to have such constants, with the use of which it is possible to compare different variants of vibroinsulation and to choose the better of them.

Let us review some particular problems relating to the above said. Let us take, that the elastic wave $u_0(x, t) = a_0 \sin(kx - \omega t)$, where k is the wave number, is propagating along some homogeneous section of the elastic tract (for example, along the pipeline of the compressor or ventilator, which is capable of delivering the mechanical energy) [3].

Let it be that in some section of that waveguide, the inhomogeneity of the elastic, inertial or combined character, reflecting the partially wave energy, is established. If the transmitted wave may be written in the form of $u_1(x, t) = a_{10} \sin(kx - \omega t)$, then the relation of the square of amplitudes may be taken as the coefficient of vibroinsulation.

This relation coincides with the relation of energies, averaged within the period of vibrations. Naturally, such averaging is possible only for the given particular case, but that is precisely in such a way that the introduced parameter fully reflects the properties of vibroinsulation and is quite convenient for application in the engineering practice.

In an other case, when random stationary vibrations are maintained, an analogous approach is possible, but it is not so convenient at that. The matter is that after determination of the coefficient of vibroinsulation as the relation of the corresponding statistical characteristics $\alpha = \sigma_1^2 / \sigma_0^2$ (where σ_1^2 and σ_0^2 – intensities of the vibration field in the presence of vibroinsulator and without it accordingly), we do not take into account fully the character of the process of vibroinsulation. Taking advantage in the coefficient α, we may actually lose in the reliability as a result of resonance phenomena in the insulated element of the mechanism on the separate sections of the spectrum where the insulator turns out to be the coordinating link between the waveguide and the element of the mechanism. However, cases exist when this determination of vibroinsulation is, nevertheless, more convenient than the other ones. It is possible, for example, to apply it for the multiresonant system with the quite high density of frequencies of natural oscillations or, taking into account some specific properties of the element of the mechanism, to put on the operation of the vibroinsulating device the additional condition for spectrum density before $(S_0(\omega))$ and after $(S_1(\omega))$ vibroinsulating device $S_0(\omega) > S_1(\omega)$ at all ω.

Another determination of the coefficient of vibroinsulation

$$V_\alpha = S_1(\omega) / S_0(\omega) \qquad (1)$$

turns out to be quite convenient for analysis of random vibrations of separate elements, constituting the insulated device, but not so convenient in comparing among different variants of the construction of the vibroinsulation.

It is interesting to note that the idea of vibroinsulation may be transferred directly on the electric circuit of the element of the mechanism. In the definite sections of the circuit (Fig. 1), it is possible to include the filters of frequencies, excited in the same circuit due to of vibrations. In a number of cases it is possible, even though the operator of transformation of vibrational noise into electric one may occur to be non-linear and containing combination frequency of the electric circuit and mechanical vibrations.

Analysis conducted shows that concretization of the properties of the fields and constructions open the possibility for the corresponding engineering solution.

By the first solution we shall find the elasticity of the liner ES/l, with the help of which it is necessary to insulate the rotor of the mechanism from the base. The mechanism is considered the rigid mass M, whereas the base is absolutely rigid. Frequency of rotation of the rotor per time unit Ω, the mass of the rotor m and eccentricity ε are considered as preset.

Method of solution. In the given case it is convenient to measure the vibroinsulation coefficient by the relation of squares of the force amplitudes, acting on the part of the mechanism on the base, with the use of the liner and without it. In the first case the force of reaction will be $F = ESu_1 / l$, where u_1 is found from the equation of the movement of mass on the elastic liner

$$M \frac{d^2 u_1}{d t^2} + \frac{ES}{l} u_1 = m\varepsilon\Omega^2 \sin \Omega t. \qquad (2)$$

In the second case when the mass is established without the liner, directly on the rigid base, we get $F_0 = m\varepsilon\Omega^2 \sin \Omega t.$. Thus, we find

$$V_\alpha = |F_1|^2 / |F_0|^2 = 1 / |1 - \frac{\Omega^2}{\omega^2}|^2, \qquad (3)$$

where $\omega^2 = ES / Ml$.

The characteristic of vibroinsulation is of resonant character (Fig. 2), this showing that at the continuous spectrum of excitation, the amplification of certain components would be possible. As could be seen without difficulty, vibroinsulation in this case (i.e. at the harmonious excitation) starts with the fulfillment of the condition $\Omega > \omega\sqrt{2}$.

The next step of the solution would be the determination of elasticity of the liner, ensuring the given value. For this purpose, it is necessary to solve an equation concerning Ω.

The paper presents a solution which differs from other classical examples and is useful for applications solving engineering problems. Let take a case problem finding the elasticity coefficient κ which ensure proper vibroisolation of mass m from the vibrating base.

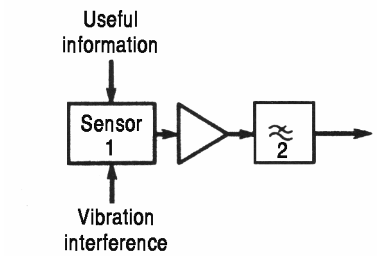

Fig. 1. **Diagram for reduction of vibration interference in the electric circuit of the device: 1 – sensor, sensitive to vibrations; 2 – filter of vibrational frequencies**

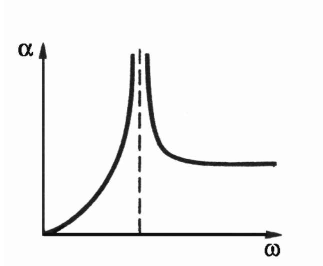

Fig. 2**. Dependence of the coefficient of vibration of the rigid mass on the elastic base from eigen frequency**

Accepted method of solution. In this case it is convenient to determine vibroinsulation as the relation of the squares of amplitudes $|u_0|^2 / |u_1|^2$ of vibrations, where u_1 and u_2 are related by the equation of movement:

$$m \frac{d^2 u_1}{d t^2} + \kappa u_1 = \kappa |u_0| \sin \Omega t. \qquad (4)$$

From here we find $V_\alpha = 1 / |1 - \Omega^2 / \omega^2|^2$, where $\omega^2 = \kappa / m$. The result is analogous to the previous one.

The natural generalization of those two methods of solution is the problem on the vibroinsulation of the system with one degree of freedom at random vibrations. If the spectrum of excitation is located in the sufficiently low-frequency range, the system in this case may be considered with the help of the same equation as in the previous case.

Let the acting spectrum of random stationary vibrations of the base be approximated by the following law:

$$S_0(\omega) = \frac{2\sigma_0^2}{\pi\Omega\left(1+\omega^2/\Omega^2\right)}, \qquad (5)$$

where σ_0^2 – parameter, determining the level of vibrations, and Ω – parameter, determining the band width of the acting spectrum.

Let us assume that the elastic element of the system represents the rod that operates on the extension-compression.

In the framework of the linear theory, instead of Hooke's law we introduce the following operational correlation:

$$F = \frac{S}{l}E\left(1+L\left[\frac{\partial}{\partial t}\right]\right)\Delta u. \qquad (6)$$

Here $L\left(\frac{\partial}{\partial t}\right)$ is some linear integral-differential operator, taking into account the inner losses in the system. Action of that operator on the solution, depending on the time according to

$$\Delta u = \Delta u_0 \exp(-i\omega t), \qquad (6a)$$

must owing to the linearity of the operator L lead to the result

$$L\left(\Delta u_0 e^{-i\omega t}\right) = \Delta u_0 e^{-i\omega t} L(-i\omega), \qquad (7)$$

where $L(-i\omega)$ is some complex function of ω.

Our experiment shows that for many materials in the field of low frequencies there takes place

$$L(-i\omega) = i\eta(\omega) = const \qquad (8)$$

where $\eta(\omega)$ is some real valued function, retaining the approximately constant value. In the literature that function is referred to as the tangent of inner losses in the material.

From these reasonings it follows that

$$S_u(\omega) = \frac{\left(1+\eta^2\right)S_0(\omega)}{\left[\left(1-\frac{\omega^2}{\omega_0^2}\right)^2+\eta^2\right]}, \qquad (9)$$

where $S_u(\omega)$ is the spectrum density of the intensity of vibrations of the insulated mass; $S_0(\omega)$ is the spectrum density of vibrations of the base; ω_0 is the eigen frequency of vibrational system, $\omega_0 = \sqrt{\frac{ES}{ml}}$.

It would be possible to complete the research at this stage by determining vibroinsulation (similarly as in the case of a high frequency) in the form $\alpha = 10\log\frac{S_0}{S_u}$.

However, the mechanism of the harmful action of low-frequency vibrations (differently from the high-frequency ones) is approximately homogeneous, i.e. frequencies, situated in the neighbouring octaves, as to their harmful action are equivalent, and it is more expedient to determine vibroinsulation through the full (integral) intensity vibrations in the form

$$\alpha = 10\log\frac{\sigma_0^2}{\sigma_u^2}, \qquad (10)$$

where

$$\sigma^2 = \frac{\sigma_0^2\left(1+\eta^2\right)}{\pi\Omega}\int_{-\infty}^{\infty}\frac{d\omega}{\left[1+\frac{\omega^2}{\Omega^2}\right]\left[\left(1-\frac{\omega^2}{\omega_0^2}\right)+\eta^2\right]}, \qquad (10a)$$

σ_u^2 is intensity of low-frequency accidental vibrations for the above-indicated low-frequency spectrum.

Conclusions

The analyzed methods of reduction of vibration of mechanisms give the opportunity to calculate and investigate the coefficient of vibroinsulation determined by this method and to select the necessary parameters of the elastic element. In particular, it follows from the given determination that in the case when it is necessary to achieve the higher sound insulation ΔL (dB), the eigen frequency of the system should be selected in accordance with the estimate

$$\omega_0 = 4\Omega\eta/10^{\left(\frac{\Delta L}{10}\right)}. \qquad (10b)$$

References

1. **Butkus R., Deikus J., Gužas D., Šarlauskas A.** The use of rigidity properties in the cylindrical shells. Abstract International conference mechatronis sytems and materials (MSM). Vilnius, LT. 2005. P. 85–87.
2. **Skuchuk E.** Simple and complex vibrational systems. M., Mir. 1974. P. 162.
3. **Gužas D.** Sound radiation and propagation in centrifugal machines piping. Sound and Vibration. Auburn University, USA. 1991. Vol. 3. P. 1219–1226.

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Mechanizmų triukšmo mažinimas izoliuojant vibracijas

Reziumė

Tiriama mechanizmų sukeliamo triukšmo parametrų priklausomybė nuo vibracijų. Tuo tikslu nagrinėjamas jėgos elementų racionaliosios schemos. Jos užtikrintų konstrukcijų patikimumą mažinant vibracijas, kurių bangų laukas sužadina triukšmą. Keliami uždaviniai vibroizoliacijai ir garso izoliacijai. Numatomos papildomos priemonės, kurios padėtų šiuos uždavinius spręsti. Naudodamiesi straipsnyje aptartais sprendimo būdais, inžinieriai galėtų numatyti priemones mechanizmų keliamam triukšmui mažinti.

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