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VIBRATION REDUCTION WITH RUBBER-METAL CAB SUPPORTS

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ABSTRACT

This paper investigates the possibility of improving the vibration isolation of a tractor cab by changing the stiffness of its rubber-metal supports. A spectral analysis of the vibration isolation properties of these supports is presented. The research was conducted on a BELARUS 1221 tractor at idling, at the maximum engine speed, with no load. The cab was mounted on rubber-metal vibration absorbers of serial design with different rubber hardness. Measurements were made with a first class certified Oktava 101VM instrument with registration of RMS acceleration values, expressed in dB, in the frequency range of 8-1000 Hz. The instrument was set in "local vibration" mode. The AR2082M probe was attached to the respective measurement points using adhesive. Mathematical relationships are given to optimize the work on the development of a new and upgraded design of the rubber-metal supports in use. It is shown that varying the stiffness of the rubber-metal supports has different effects in the low and high frequencies of the spectrum. A stiff support is effective in damping low-frequency vibrations but degrades the support's properties in damping high-frequency vibrations, and vice versa. A soft support is more effective at higher frequencies and reduces the effectiveness of the support at lower frequencies.

Introduction

The properties of a vibration as a physical factor that has a negative impact on human health (Seidel and Heide, 1986; Hazra and Ghosh, 2009; Jakubczyk-Gałczyńska and Jankowski, 2014; Avdonin, 2015; Alujević et al., 2018) and machine reliability are well known. It is also generally known that power units and chassis are its main sources in mobile machines. A vibration is also a source of the structure-borne noise which reduces the performance of machines (Kurbonov and Sulaymanov, 2022; Hegazy et al., 2000; Kirkham, 1997). The vibration problems exist in all areas of the machine structure. General vibration control measures are to reduce the source vibration levels (e.g. by balancing, counterbalancing, reducing excitatory forces) and to isolate objects from the damaging effects of vibration (Wang and Chen, 2002). The practical solution of the problem lies in the optimal use of both methods to obtain the maximum effect at a minimum cost. This paper did not analyze techniques to reduce source vibration, but the effectiveness of the most common vibration isolating devices - rubber-metal supports. This is the most common type of vibration isolators used in mobile machinery, as they have several advantages, which other types of such devices do not have: the ability to absorb differently directed loads, durability, simplicity, low cost. They are used to support cabs, engines, driveshafts, etc.

This paper uses theoretical developments taken from the works of Razumovsky (1973), Klukin and Borba (1971). These works present estimated characteristics of insulators, give mathematical dependences for calculation of vibration isolators in one-dimensional schemes. Moreover, the authors analyze not only vibration isolators designs, but also schemes of their installation on machines. Despite the different fields of research (shipbuilding, tractor-building), the estimated parameters and mathematical dependences suggested by the authors are very similar. The theoretical calculations of the characteristics of vibration isolators, according to the authors themselves, are applicable at the stage of initial design for approximate calculation of simplified models. The authors point out that the actual effectiveness of vibration isolators can only be determined by experimental investigation. The mathematical dependences presented by the authors (Razumovsky, 1973; Klukin and Borba, 1971) are not fully confirmed by the results of experimental studies given in this article.

The paper by Razumovsky (1973) provides some results of experimental studies of the MTZ-80 tractor cabin vibration. The author conducted research of cab vibration isolators produced serially in the 70-s and gave instructions on their arrangement schemes. However, he considered the influence of rubber stiffness on vibration isolator properties only theoretically. The author's theoretical developments claim that the use of rubber-metal vibration isolators allows to reduce vibration only in the spectrum zones remote from the resonance frequency of the support, the following dependence is given:

$$W=20\cdot\lg\left|1-\left(\frac{f}{f0}\right)^2\right| \tag{1}$$

where:

W – is vibration isolation, dB

f – frequency of the spectrum under study, Hz

f_o – resonance frequency, Hz

This thesis was not confirmed during research on modern BELARUS tractors. It should be noted that over the past time, the tractor design has been significantly upgraded, some systems and their layout have changed fundamentally. The following things have changed: the cab design, the power of the power units, the layout, unit weights, etc. Consequently, the forces and their vectors acting on the vibration isolator have changed. The vibration isolator itself is made from rubber with different qualities, dimensions, and shapes.

In the book (Klukin and Borba, 1971) some contradictory dependencies can be noted. One expression shows that the vibration isolation of the W, across the spectrum, increases with increasing frequency f:

$$W = 20\lg\frac{2\cdot f}{\pi\cdot\sqrt{C/m}}. \tag{2}$$

And in the same chapter, in another expression, the opposite is claimed:

$$W= 20\lg \frac{1}{E(1+a\cdot f)} \tag{3}$$

where:

C – is the hardness of the rubber seal, Pa·m
m – mass of the object to be damped, kg
a – some positive number

Theoretical studies of vibration damping by rubber plates are of some interest (Au-Yeung et al., 2019; Bashmur et al., 2019), but in this work the authors limited themselves to the study of the vibration damping rate in the zone of the resonance frequency.

Work by Carpineto et al. (2014) is devoted to study of hysteresis phenomena at the resonance frequency in rigid supports.

Papers by Au-Yeung et al. (2019) and Carpineto et al. (2014)) are basic theoretical investigations and their application in practice is limited only to separate phenomena - resonance frequency and hysteresis. The developments given in apply only to ideal models and cover a very narrow area of the vibration isolation problem.

Analysis of the recent works of authors engaged in applied research of vibration of mobile machines shows that they mainly investigate the effect of vibration on noise in the cabin, without conducting any analysis of the effectiveness of vibration isolation devices (Vasil'yev, 2004; Kobzev, 2016; Hao et al., 2022). In Vasilyev's work (2004) the relationship between noise and vibration in the vehicle cabin is considered, but this relationship is considered only under the vehicle layout scheme. Experimental and theoretical studies of vibration in the cab of gantry cranes are given in the articles (Kobzev, 2016). The results given in the articles (Vasil'yev, 2004; Kobzev, 2016) are applicable only for the corresponding types of machines and the authors do not analyze the properties of the dampening devices.

The articles (Wang et al., 2016) attempt to study the vibration characteristics of the truck cabin to improve the driving comfort. A dynamic model of a rigid-flexible coupling of a commercial vehicle is proposed to calculate the contribution of various sources to cabin vibration. The final selection of vibration isolators was made experimentally. The authors classified rubber-metal isolators as ineffective and did not investigate the reasons of their low efficiency.

The purpose of this paper is to analyze the performance of rubber-metal supports of the tractor cab as vibration-damping devices and present mathematical dependencies allowing to optimize the design of rubber-metal supports.

Materials and Methods

When installing the cabs of mass-produced tractors, the rubber metal vibration dampers are used. This type of support is used because, in addition to the function of absorbing vibration from the source it must meet the requirements of safety - to ensure sufficient strength under the influence of transverse forces on the cab.

The research was conducted on a BELARUS 1221 tractor at idling, at the maximum engine speed, with no load. The cab was mounted on rubber-metal vibration absorbers of serial design with different rubber hardness.

The measurement technique was developed based on the recommendations of Brul & Kjaer, a leading manufacturer of vibroacoustic equipment (Serridge and Torben, 1987). Measurements were made with a first class certified Oktava 101VM instrument with registration of RMS acceleration values, expressed in dB, in the frequency range of 8-1000 Hz. The instrument was set in "local vibration" mode. The AR2082M probe was attached to the respective measurement points using adhesive. The selected instrument and transducer allowed real time measurements to be made. All spectrograms below show vibration levels in the form of the vector sum of accelerations measured along three axes in octave bands.

Fig.1 shows the results of the vibration isolation test of a series cab support (rubber hardness 50 Shore).

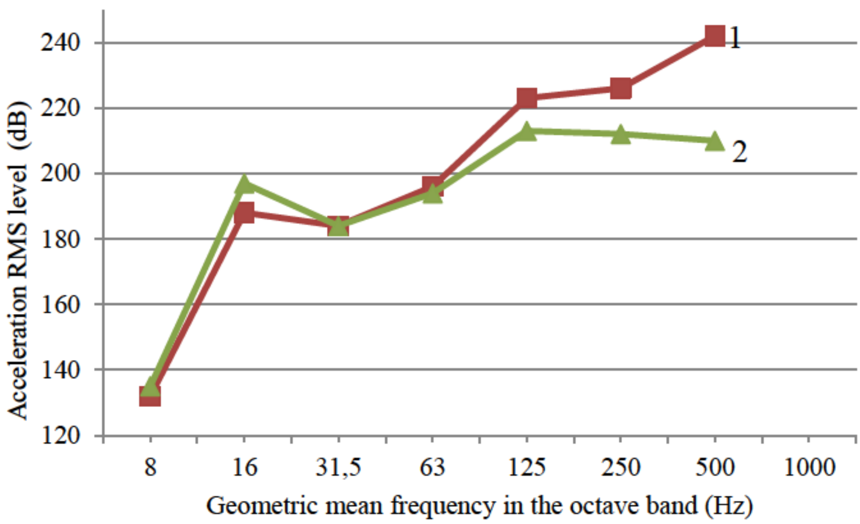

Figure 1. Vibration acceleration spectrum in front of and behind the vibration isolator: 1 – vibration acceleration on the tractor frame in front of the rear cab rubber-metal support; 2 – vibration acceleration on the bracket for attaching the cab to the rear rubber-metal support

As can be seen from the spectrograms shown, the rubber-metal cab mounts dampen vibration at frequencies in the octave bands of 125 Hz and above. In the low frequency range (63 Hz and below), there is no vibration reduction effect, and in the 16 Hz octave band, there is an increase in vibration as it passes through the support.

The reasons of vibration amplification by supports can be: specific properties of rubber, defects in the way of installation on the four support points (Razumovsky, 1973), resonance phenomena or possible combined effect of the above mentioned reasons. Based on the theoretical studies of the above sources, it can be argued that the increase in vibration occurs only as a consequence of the resonance phenomena. In this case, there can be two types of the resonance phenomena: resonance in the rubber pad (wave resonance) and resonance in the cab-vibration isolator-tractor frame system. Wave resonance occurs when the following condition is met:

$$h = n \cdot \lambda = n \cdot \frac{c}{f}\ ;\ n = 1, 2, 3, \ldots \tag{4}$$

where:

c – is the velocity of wave propagation in the medium, $m \cdot s^{-1}$
λ – wavelength, m
f – frequency, Hz
h – one of the geometric dimensions of the support

The wavelength in rubber at a frequency of 16 Hz, depending on the number of external factors, is 40...100 m. By comparing this value with the size of the vibration isolator it can be stated that there is no wave resonance in the support.

Resonance frequency of the system (cab-vibration isolator-tractor frame) is determined by the formula (Klukin and Borba,1971):

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{E \cdot S}{m \cdot h}}\ \text{Hz}; \tag{5}$$

where:

E – static modulus of elasticity, Pa
S – cross-sectional area of rubber pad, m^2
h – height of rubber pad, m; m-mass of the cabin acting on one support, kg

Substituting values corresponding to parameters of the tested tractor “Belarus 1221” into formula (5) we obtain f_0 = 15...22 Hz. i.e. the calculated resonance frequency is in octave band of 16 Hz, and the marked increase of vibration is the consequence of resonance of the cab-vibration isolator-tractor frame system. All oscillating systems have a resonance frequency of their own, at which vibrations from the source can theoretically be amplified to infinity. In this case, the amplification of vibrations is observed, but relatively small, due to the physical properties of rubber, high mechanical losses during deformation.

Using expression (5), it is possible to define support designs with different resonance frequencies. This solves the problem of the increased cabin noise if the resonance frequency of the system coincides with the sound frequency defining the cabin noise.

At high frequencies, in octave bands above 63 Hz, vibration can be represented as a distributed load and its vibration isolating effect is determined by wave resistance of gasket material ρ·c (Razumovsky, 1973). Mathematically, this can be represented by expression (6) (Klukin and Borba, 1971). Here it is assumed that the vibration in the rubber gasket propagates as a sound wave.

$$W=10\cdot lg(1+(\pi\cdot f\cdot(\rho\cdot c)_1\cdot h / E_d)^2) \quad (6)$$

where:

W – is vibration isolation, dB

$(\rho\cdot c)_1$ – a certain wave impedance of the conductor, $Pa\cdot s\cdot m^{-1}$

E_d – a kind of equivalent modulus of elasticity of rubber, Pa

E_d and $(\rho\cdot c)_1$ are physical properties of rubber, exact determination of which is a very problematic issue and, therefore, it is impossible to calculate the W value in a practical application. Expression (6) is useful for designing rubber-metal supports. Using this expression, we can say that at high frequencies the vibration isolation will be higher if rubber spacers are used:

– with a greater height h,
– lower modulus of elasticity,
– higher values of rubber density ρ.

Low damping efficiency at low frequencies can be explained by the fact that in this frequency range the rubber pad perceives vibration as a concentrated load, i.e., it behaves as elasticity (Razumovsky, 1973). Vibration transmission, in this case, is determined by the parameters of the vibration system, in the example under consideration it is the cabin - vibration isolator - the tractor frame. An approximate calculation of vibration isolation can be made by equation (2). To check this assumption an experiment with supports of different stiffness was performed, presented in Fig. 2

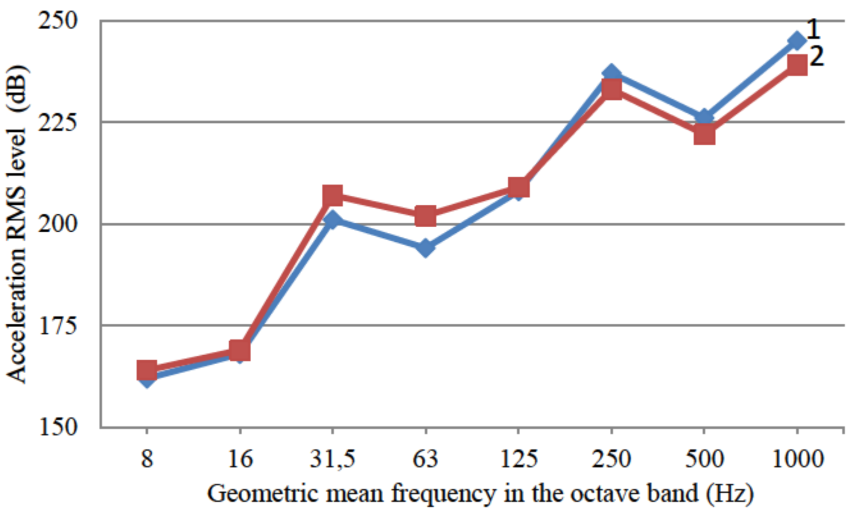

Figure 2. Vibration acceleration spectrum behind the cab vibration isolator when installed on rubber supports of various hardness. 1-rubber hardness 50 Shore units;2-rubber hardness 40 Shore units.

Analysis of the spectrogram indeed confirms that at low frequencies rigid rubber has better vibration isolation. The validity of equation (2) is also confirmed for the low frequency range.

Conclusion

1. Changing the stiffness of the rubber-metal mounts has a different effect in the low and high frequency parts of the spectrum. A rigid support is effective in damping low-frequency vibrations but impairs the support properties in damping high-frequency vibrations. Conversely, a soft support is more effective at higher frequencies and reduces the effectiveness of the support at lower frequencies.
2. By changing the properties of the rubber used in the support, it is not possible to significantly increase the efficiency of rubber-metal supports, across the entire vibration spectrum.
3. The approximate calculation of the support for the low-frequency spectrum is recommended using equation (2), for the high-frequency spectrum using equation (6).

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REDUKCJA WIBRACJI ZA POMOCĄ GUMOWO -METALOWYCH WSPORNIKÓW KABINY

Streszczenie. Niniejsza praca analizuje możliwość ulepszenia izolacji wibracji w kabinie ciągnika poprzez zmianę sztywności gumowo-metalowych wsporników. Przedstawiono analizę spektralną właściwości izolacji wibracji tychże wsporników. Związki matematyczne zostały podane w celu optymalizacji pracy nad stworzeniem nowego i ulepszonego projektu gumowo-metalowych wsporników. Wykazano, iż różnicowanie sztywności gumowo-metalowych wsporników daje różne efekty w niskich i wysokich spektrach częstotliwości. Sztywny wspornik jest skuteczny w wyciszaniu wibracji o niskiej częstotliwości, ale zmniejsza właściwości wyciszania wibracji o wysokiej częstotliwości i vice versa. Miękkie wsparcie jest bardziej efektywne przy wysokich częstotliwościach i redukuje skuteczność wspornika przy niskich częstotliwościach.

Słowa kluczowe: wibracja, wspornik gumowo-metalowy, izolacja wibracji, ciągnik, kabina, spektrum częstotliwości