THE ROLE OF VIBROISOLATORS IN DAMPING AN RADIAL FAN'S VIBRATIONS

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Summary

In this paper there is analyzed an influence of vibroisolators' stiffness and damping qualities on vibrations level of centrifugal fun. The results obtained from numerical simulations confirmed the regularity that a limitation of force affecting the fan's foundation can cause an increase of its own vibrations' amplitude.

Keywords: springy vibroisolators, rubber vibroisolator, resonance response.

ROLA WIBROIZOLATORÓW W TŁUMIENIU DRGAŃ WENTYLATORA PROMIENIOWEGO

Streszczenie

W pracy analizowany jest wpływ sztywności i własności tłumiących wibroizolatorów na poziom drgań wentylatora promieniowego. Wyniki symulacji numerycznych potwierdziły prawidłowość, że ograniczenie wielkości siły działającej na posadowienie wentylatora może powodować wzrost amplitudy jego drgań.

Słowa kluczowe. wibroizolator sprężynowy, wibroizolator gumowy, odpowiedź rezonansowa.

1. INTRODUCTION

Fans are commonly applied in turbomachinery industry. Main mediums in these devices are: air, air mixture with particles of solids or other gases depending on the system character. Fans are dovces with not complicated designs. Its principal element is a rotor with blades and typical streamlined profile whose diameter may reach a few meters. The width depends on the rotor function. Machines with great volume flow rate have wide rotors while fans with high pressure differences have narrow rotors. Rotating rotor transmits kinetic energy to the medium flowing in chambers between blades. This kinetic energy is changed into dynamic pressure of gas. In a helical chamber the medium is compressed which causes static pressure increase[1].

The fan rotors bases are fixed usually either directly to the foundation using anchors, or on steel frames after being placed in armored concrete. Frequently between the frame and the body vibroisolator is mounted, in which springy-damping element is spring or rubber pad. Using vibroisolators is not always necessary and desirable because their influence on the system dynamic properties. Generally, the principle is suggested, according to which vibroisolators should be used in case when there is a necessity to limit the action of forces on the foundation. Using element with lower stiffness then stiffness of direct connection of the fan body with the frame or the foundation results in decrease in device free vibration frequency. Small damping in the system is factor which favors growing of vibrations parameters amplitude. Sometimes stiffness and body damping are so small, that vibration amplitudes in frequencies both excitation frequency and free frequencies reach values higher then permitted by the norm [2-3].

2. FAN MOUNTING ON VIBROISOLATORS

For the needs of vibroisolation two kinds of vibroisolators are commonly used: springy Fig. 1 and rubber ones. The basic difference between these kinds of vibroisolators lies in their characteristic

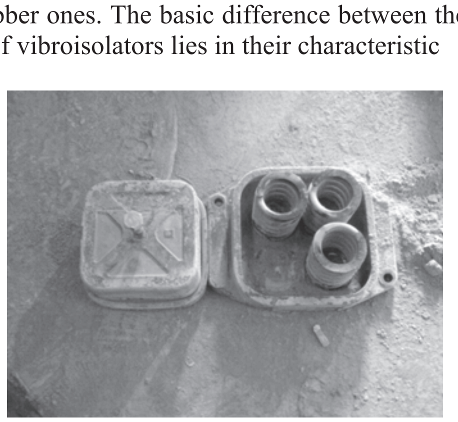


Fig. 1. An example of springy vibroisolator

The static characteristics of springy vibroisolators W2-434, W2-435, W2-482 and rubber vibroisolator W100/35 determined by first of the author, have been presented in Fig. 2.

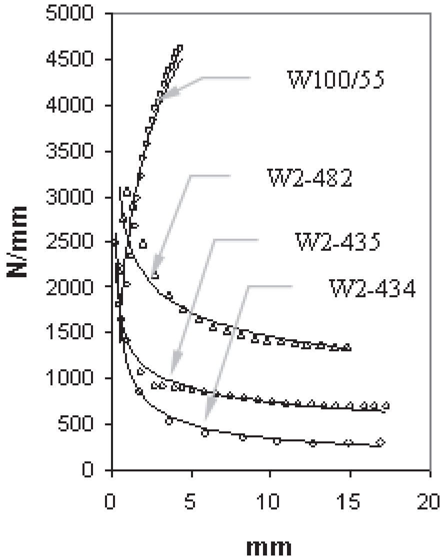


Fig. 2. Static characteristics of springy vibroisolator stiffness and rubber vibroisolator

Comparison of forced frequency function changes of vibroisolators dynamic properties can be evaluated through determination of their resonance response. For this purpose an exciter with controlled forced frequency can be used (Fig. 3)

Fig. 3. Determination of the vibroisolator resonance response

In this way the resonance point shift for vibroisoaltors(W2-435, W2-482) and has been defined. Resonance area of vibroisolator W2-482 is shifted n the direction to the higher frequencies (Fig. 4) due to their higher stiffness (Fig. 2).

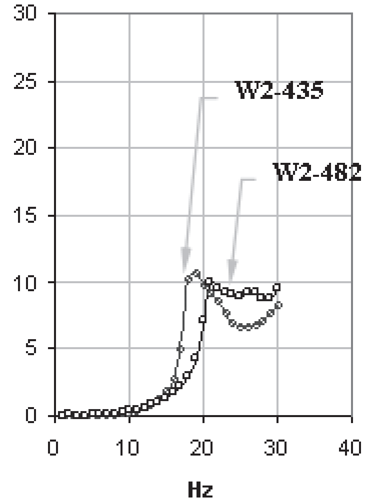


Fig. 4. Resonance point shift for vibroisoaltors (W2-435, W2-482)

3. GENERAL PRINCIPLES OF VIBRO-ISOLATION

The ratio of maximal force transmitted to the foundation in time of vibrations to static force is called transmission coefficient. It is expressed by the following dependence:

$$\varepsilon = \frac{\sqrt{1+4\frac{h^2}{\omega^2}\frac{\nu^2}{\omega^2}}}{\sqrt{\left(1-\frac{\nu^2}{\omega^2}\right)+4\frac{h^2}{\omega^2}\frac{\nu^2}{\omega^2}}} \tag{1}$$

Maximal value of force transmitted from the ventilator to the foundation is:

$$P_{\max} = kx_{st}\,\varepsilon \tag{2}$$

whereas:

ν - forcing frequency,

ω - free vibration frequency,

$h=\frac{1}{2}\frac{c}{m}$ - vibroisolator damping constant,

k- vibroisolator spring stiffness,

x_{st}- vibroisolator spring static deflection,

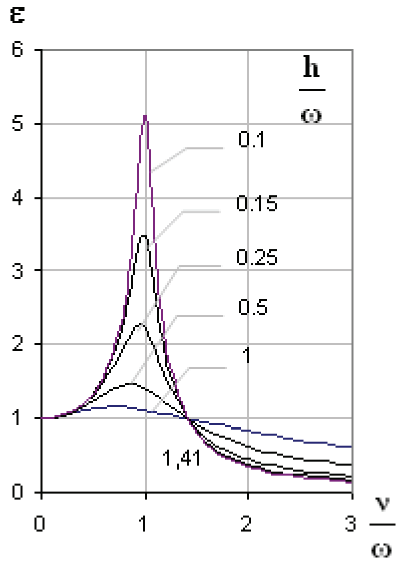


Fig. 5. Dependence of transmission coefficient from forced frequency and free vibration frequency

From Fig. 5 we can read that effectiveness of amortization is correct, when:

$$\frac{\nu}{\omega} > \sqrt{2} \tag{3}$$

Advisability of this condition will be analyzed on a simple example of vibrations of a mass mounted on a vibroisolator affected by forcing with frequency f=6Hz (Fig. 6).

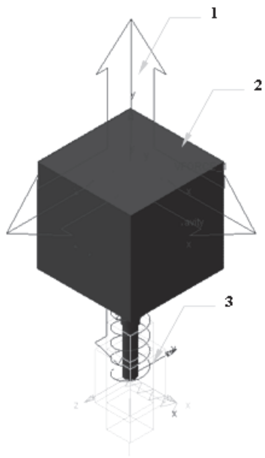
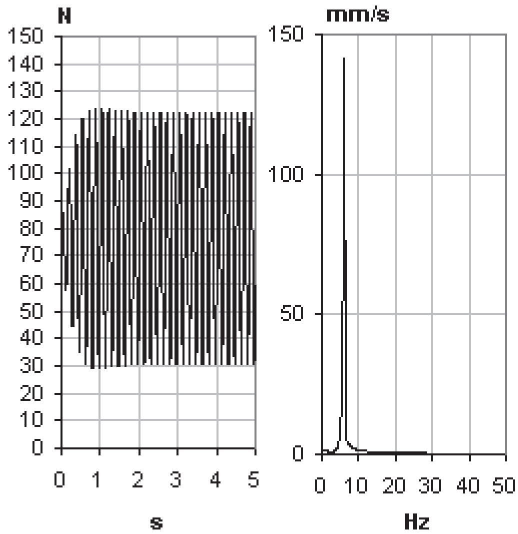


Fig. 6. Idea scheme of vibroisolator operation
1 – harmonic forcings with 10N amplitude and 6Hz frequency, 2 – system mass 7.8kg 3-vibroisoaltor

For vibroisolators with k=10N/mm stiffness the system free vibration frequency is 5.7Hz. For harmonic forcing with frequency 6Hz we can talk about near rezonans vibrations of the system. Thus, the values of the force transmitted onto the foundation (Fig. 7a) and and the vibration high velocity are significant (Fig. 7b). If we assume that the forced frequency can not be moved out of the resonance area it should be recognized that the choice of the vibroisolator has not been proper. Improvement of vibration isolation effectiveness defined by the value of force transmitted onto the foundation, according to condition (3) needs application of a less stiff vibroisolator.

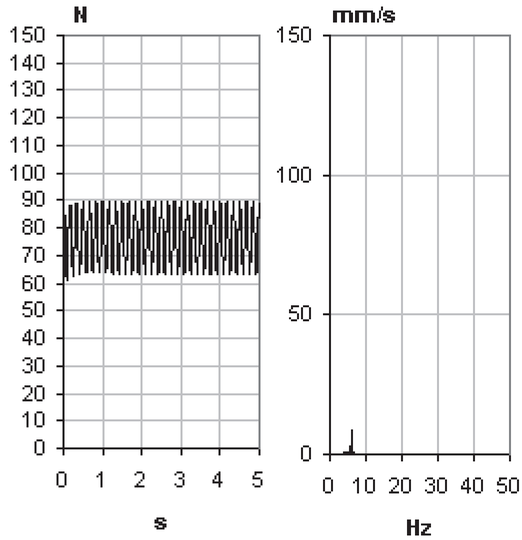
$$k < 5.54 \frac{N}{mm} \tag{4}$$

Vibroisolator with stiffness 3N/mm can really decrease the force value diametrically, almost to a value resulting from static loading (Fig. 9a.).



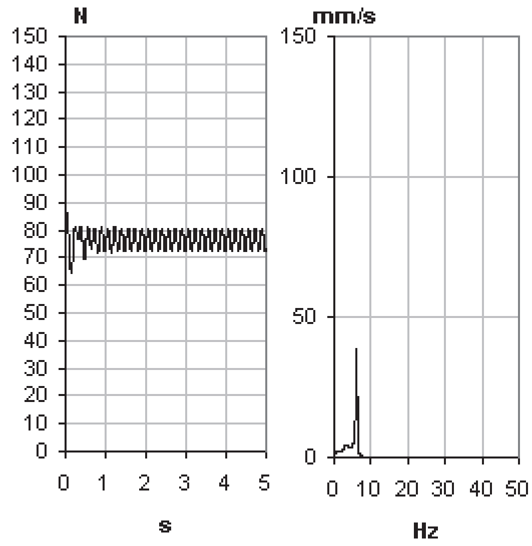
(a) (b)

Fig. 7. Changes in time of force acting on the foundation with vibroisolator frequency 10N/mm (a) and amplitude-frequency characteristic of the system vibration velocity with vibroisolator stiffness 10N/mm (b)



(a) (b)

Fig. 8. Changes in time of force acting on the foundation with vibroisolator frequency 50N/mm and amplitude-frequency characteristic of the system vibration velocity with vibroisolator stiffness 50N/mm



(a) (b)

Fig. 9. Changes in time of force acting on the foundation with vibroisolator frequency 3N/mm and amplitude-frequency characteristic of the system vibration velocity with vibroisolator stiffness 3N/mm

The velocity of system vibartions is still very high (Fig. 9b). In real conditions such a situation could be considered as unsatisfactory. It must be noted that increasing the spring stiffness up to value 50N/mm causes that effectiveness of vibroisolation is related to decrease in (Fig. 8a) the system force action on the foundation, however vibration velocity amplitude value reaches the accepted level. (Fig. 8b).

3. VIBROISOLATION IN CASE OF INERTIAL FORCING

For the rotor rotation centrifugal force appears caused by its unbalancing.

$$F_b(\omega) = m_n \cdot \omega^2 \cdot r \qquad (2)$$

m_n – mass unbalanced,
ω - angular velocity,
r – radius of rotor,

Because of the fact that the amplitude of the forcing force depends on the forcing frequency, we can not talk here about a transmission coefficient in the same terms as we do in the case of the amplitude constant value. The ventilator rotor is an example of this kind of forcing.

An analysis of a radial ventilator vibrations is easy to be performed with the use of the method of multi body systems(MBS). The calculation model is used for simulation, and is presented in Fig. 10.

Depending on the degree of complexity of the model the dynamic properties of the system can be considered taking into consideration an array of factors affecting them. In Table 1 the physical features of a radial ventilator of medium size with a hung rotor have been set up. Also the influence of vibroisolators damping features on the rotor operation stability and in relation to the fact that while being started it undergoes through the resonance area for vibrations in the horizontal direction, vertical and twisting vibrations has been analyzed.

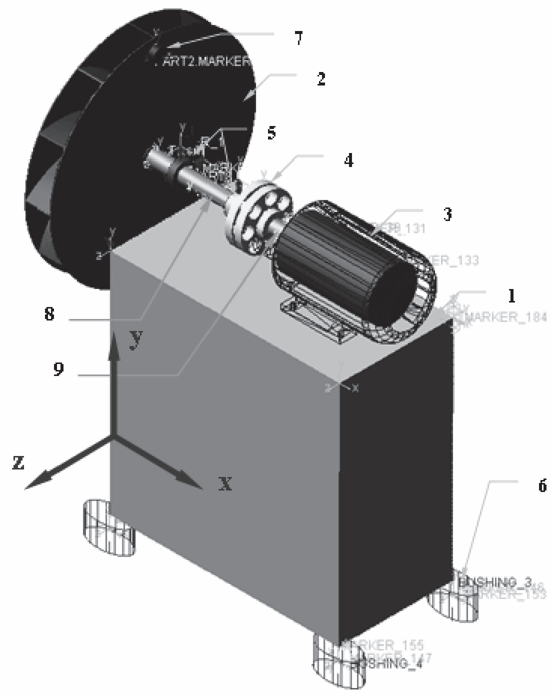


Fig. 10. Computational model of a radial ventilator
1. body, 2. rotor, 3. motor, 4. coupling,
5. bearing, 6. vibroisolator, 7. unbalancing of rotor,
8. rotor shaft, 9. engine shaft

Table 1.
Computational parameters of a ventilator model

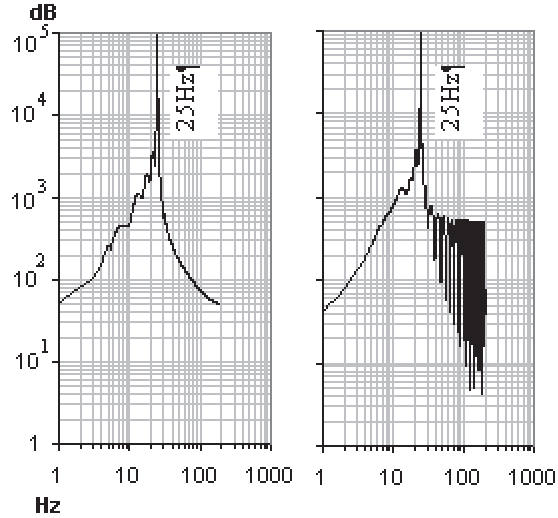
	Mass kg	I_{xx} kg mm^2	I_{yy} kg mm^2	I_{zz} kg mm^2
1.	304	$7.26 \cdot 10^7$	$4.94 \cdot 10^7$	$4.75 \cdot 10^7$
2.	181	$1.42 \cdot 10^7$	$7.67 \cdot 10^6$	$7.67 \cdot 10^6$
3.	151 119	$3.51 \cdot 10^6$ $1.94 \cdot 10^6$	$3.47 \cdot 10^6$ $1.94 \cdot 10^6$	$1.80 \cdot 10^6$ $7.18 \cdot 10^5$
4.	8.3	$4.37 \cdot 10^4$	$2.38 \cdot 10^4$	$2.38 \cdot 10^4$
5.	1.18	2000	1088	1088
7.				
8.	8.8	$1.2 \cdot 10^5$	$1.2 \cdot 10^5$	3970
9.	2	1786	1786	893

Table 2 contains a setting up of stiffness and damping of vibroisolators.

Table 2.
Springy-damping properties of vibroisolators have been accepted in the following way

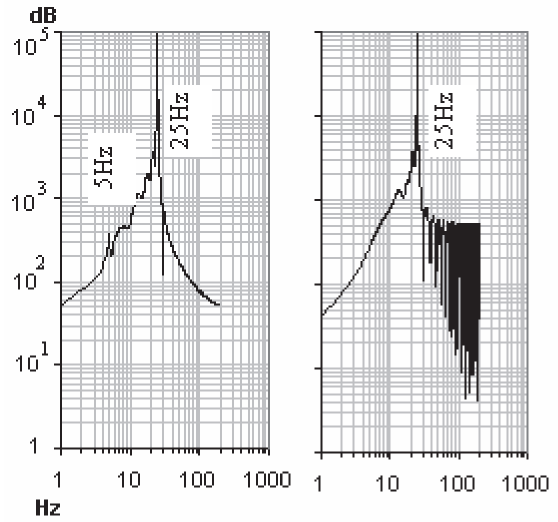
	k_{xx} kN/m	k_{yy} kN/m	k_{zz} kN/m	c_{xx} kNs/m	c_{yy} kNs/m	c_{zz} kNs/m
6	5000	5000	4000	10	10	10
				2	2	2

For damping value 10Ns/mm the system response expressed in decibels that is Bode's characteristics is presented in Fig. 11 and Fig. 12. The first left represents vibrations in the horizontal direction, and the right is connected with the vertical direction.

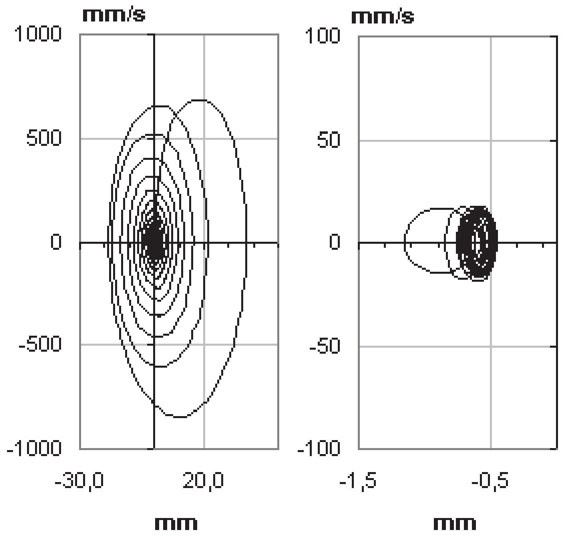


(a) (b)
Fig. 11. Bode's diagram for the system vibrations in the horizontal (a) and vertical (b) direction (damping 10Ns/mm)

The rotor operation stability for such damping is not threatened. Vibrations both horizontal and vertical ones undergo strong damping (Figs. 12 -13).

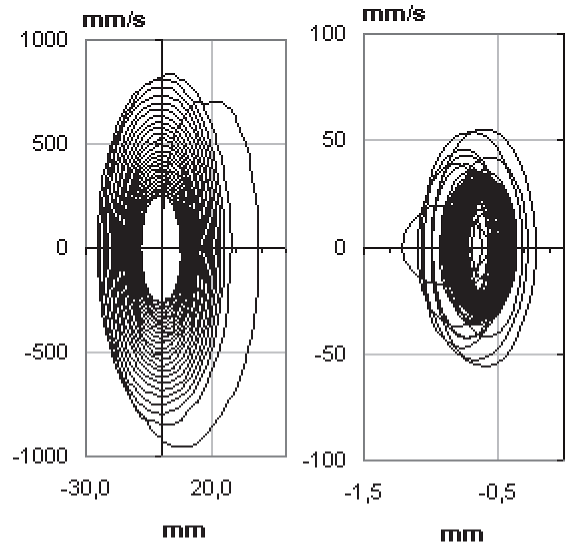


(a) (b)
Fig. 12. Bode's diagram for the system vibrations in the horizontal (a) and vertical (b) direction (damping 2Ns/mm)



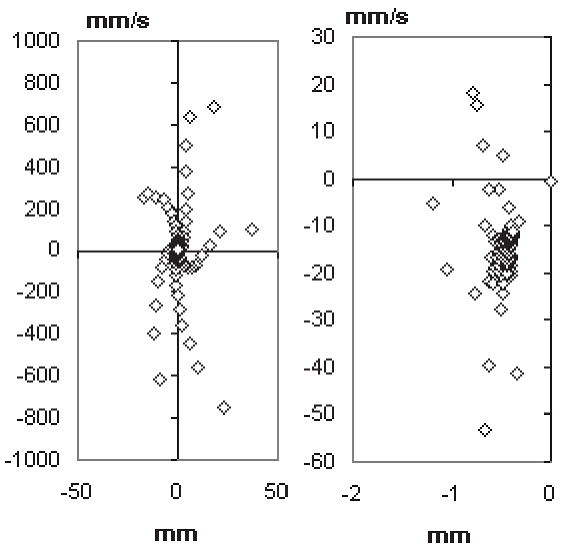
(a) (b)
Fig. 13. The system vibration phase plane in the horizontal (a) and vertical (b) direction (damping 10Ns/mm)

The movement of the body of a ventilator mounted on vibroisolators is of chaotic character, especially while going through the resonance area



(a) (b)
Fig. 14. The system vibration phase plane in the horizontal (a) and vertical (b) direction (damping 2Ns/mm)

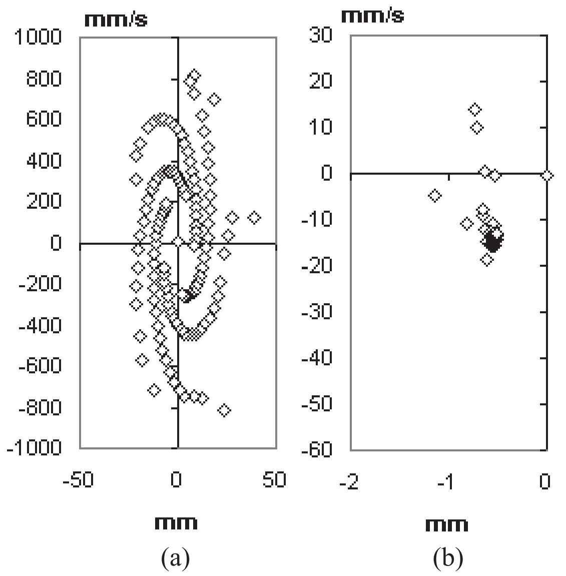
In connection with this it is natural to present a picture of this movement on Poincare diagram.



(a) (b)

Fig. 15. Poincare diagram for the system vibrations in the horizontal (a) and vertical (b) direction (damping 10Ns/mm)

An analysis of the damping quantity influence on the rotor movement stability makes it possible to conclude that for a fan with mass similar to the one that was accepted for the model and similar inertial excitement, vibroisolators with viscotic damping and damping properties -10Ns/mm will provide sufficient damping.



(a) (b)

Fig. 16. Poincare diagram for the system vibrations in the horizontal (a) and vertical (b) direction (damping 2Ns/mm)

4. CONCLUSIONS

The author of the paper presents the fact that issues connected with an application of vibroisolators in industry are not simple ones. A radial ventilator has been chosen as an object of consideration because it is widely used as a turbo machine in many branches of industry. Being an element of a technological line it is often placed in halls' ceilings that is where dynamic excitement with significant amplitude affecting the construction is undesirable. Thus, it is necessary to limit values of the forces transmitted to the machine foundation. The right choice of vibroisolators' damping and stiffness requires deep analyses with the use of knowledge on the machine dynamics and calculation methods allowing to use this method[4-5].

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