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VIBROISOLATORS APPLICATION FOR DAMPING VIBRATIONS IN INDUSTRIAL FANS

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Summary: *In this work, authors taking into consideration some aspects connected with using vibroisolators for vibrations damping of simple rotor systems. The good example of such applications are industrial fans. In these machines forces of inertial character triggered by rotor unbalancing acting on hardly rigid structure, exciting oscillations with high amplitudes. Fans, as elements of given technological paths, must have to control possibility of air volume flow rate. This aim is obtained by changing of rotor rotational speed. In practice, that excitation frequencies are in wide band from several to a few dozen hertz. These conditions must be known and taken into consideration during design of vibroisolation systems for it proper efficiency assurance.*

1. Introduction

Fans are commonly applied in industry fluid-flow machines. Main mediums in these devices are: air, air mixture with particles of solids or other gases (fumes) depending on the system character.

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.

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2. Ventilator Mounting on Vibroisolators

For the needs of vibroisolation two kinds of vibroisolators are commonly used: springy and rubber ones.

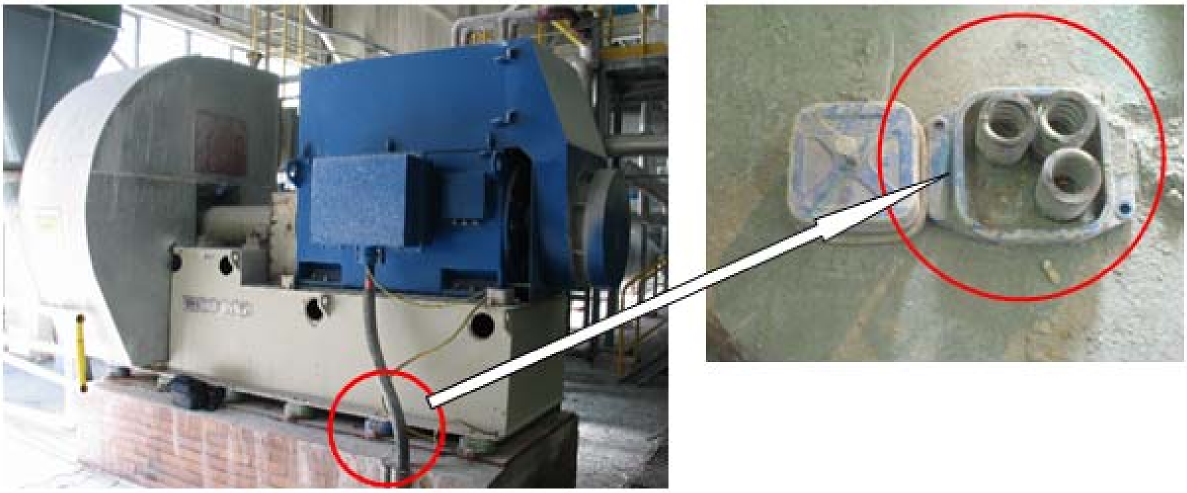


Fig. 1: An example of springy vibroisolator application for mounting of the ventilator

The basic difference between these kinds of vibroisolators lies in their characteristic. The static characteristics of springy vibroisolators W-434, W2-435, W2-482 (GERB producer) and rubber vibroisolator W100/35 determined by first of the authors, have been presented in Fig. 2.

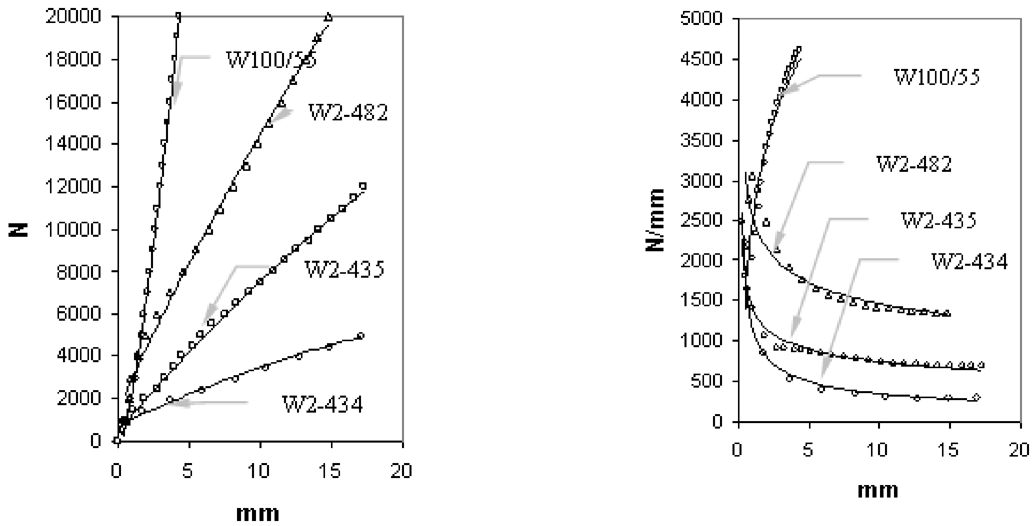


Fig. 2: Static characteristics of springy vibroisolator stiffness and rubber vibroisolator

Static stiffness is defined as a ratio of force acting on the vibroisolator until its deflection

$$k_{stat} = \frac{F_{stat}}{f_{stat}} \tag{1}$$

where: k_{stat} – static stiffness,
F_{stat} – static force acting on vibroisolator,
f_{stat} – static deformation (strain) of the vibroisolator.

During operation vibroisolators undergo the action of changeable in time force ant therefore its dynamic characteristic is of great importance, especially that the vibroisolator like every vibration system has its own resonance frequency.

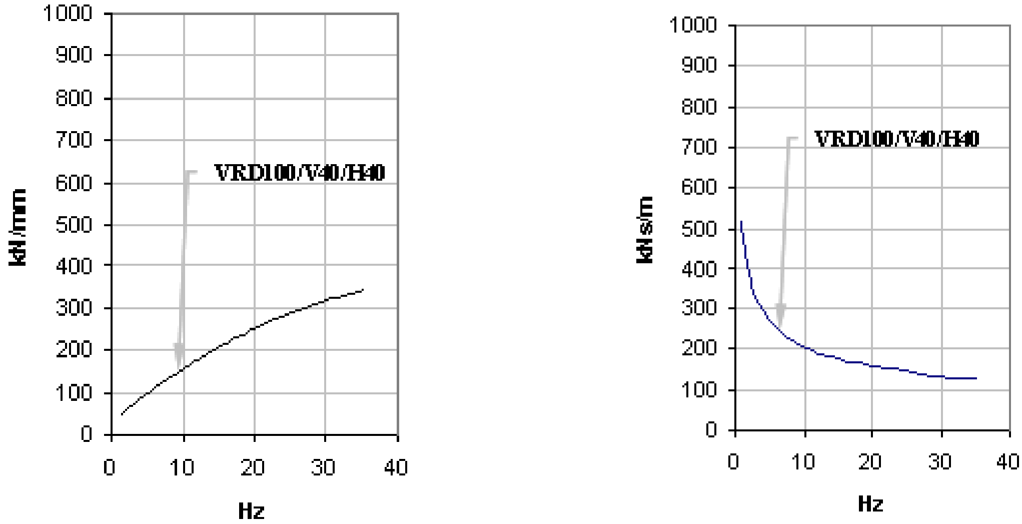
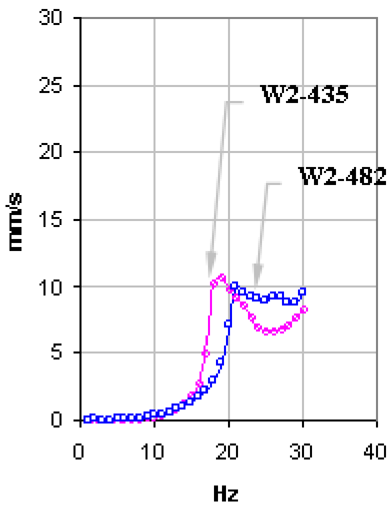


Fig 3: Dynamic stiffness and damping of vibroisolator with viscotic damping VRD100/40/1140 produced by GERB company


(a)



(b)

Fig. 4: Determination of the vibroisolator resonance response

Analogically to definition (1) the vibroisolator dynamic stiffness is defined as:

$$k(\nu) = \frac{F(\nu)}{f(\nu)} \tag{2}$$

where: $k(\nu)$ – dynamic,

$F(\nu)$ – changeable in time force acting on the vibroisolator,

$f(\nu)$ – dynamic deformation of the vibroisolator.

In Fig. 3 there have been showed dynamic characteristics of the vibroisolator VRD 100/V400/H40 stiffness and damping . This is a springy vibroisolator with viscotic damping used usually for damping vibrations of industrial pipelines. It must be emphasized that the vibroisolator stiffness increases when it operates in higher frequencies, whereas its efficiency decreases.

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.) In this way the resonance point shift for vibroisolators (W2-435 W-2-482) and has been defined. Resonance area of vibroisolator W2-482 IS is shifted n the direction to the higher frequencies (Fig.3.b) due to their higher stiffness (Fig.2.1).

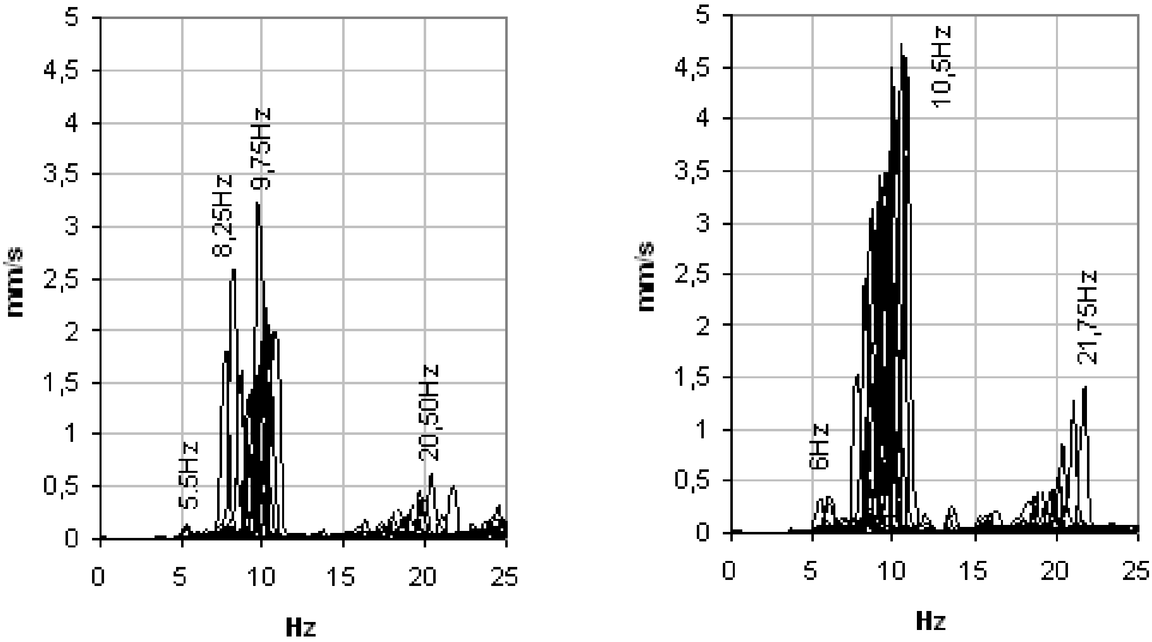
3. Analysis of Ventilator's 436FA76 Dynamics

Definition of the influence of ventilator 436FA76 mounting stiffness on its dynamic properties required carrying out tests both directly on the object and numerical simulation. This ventilator is an important link of the initial milling system. Due to its dimensions and weight ~170kgN it is an interesting object for both experimental and modeling tests.

Fig 5: View of the tested ventilator

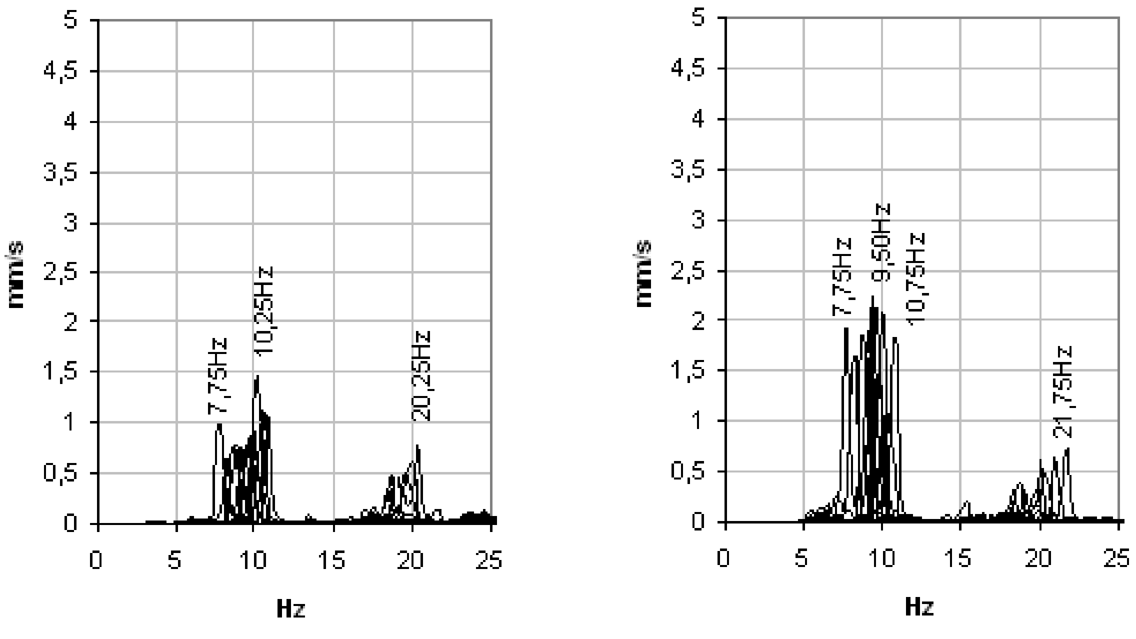
The tested ventilator is driven by an engine with 325kW power. The maximal rotational speed of the engine is 740rot/min. Usually the fan is used with the rotational speed 500-600rot/min.

In order to get to know the fan dynamic properties it is necessary to define the system responses to the excited vibrations. For this purpose, the object response to dynamic excitation with frequencies within 4.5Hz-12.5Hz excited by the rotor unbalancing, has been defined.



(a) (b)

Fig. 6: Resonance characteristics: (a) measurement direction 1H, (b) measurement direction 1V



(a) (b)

Fig. 7: Resonance characteristics: (a) measurement direction 2H, (b) measurement direction 2V

Results of these tests revealed the system sensitivity to action of loadings periodically changeable in time. The area of sensitivity is defined by the frequencies 8Hz-11Hz.

Modeling of the fan properties was carried out according to multi body systems (MBS) dynamic rules. The only susceptible elements are vibroisolators and the clutch. Calculations performed for the model with dimensions shown in Fig. 7.

Table 1: Vibroisolator stiffness and damping parameters

Translation stiffness and damping	k_x	N/mm	484	Rotational stiffness and damping	φ_x	Nmm/^{0}	10^6
	k_y		580		φ_y		10^6
	k_z		484		φ_z		10^6
	c_x	Ns/mm	0.4		χ_x	Nmms/^{0}	50
	c_y		0.48		χ_y		50
	c_z		0.4		χ_z		50

The result of simulation is definition of free frequencies and forms of free vibrations corresponding to them.

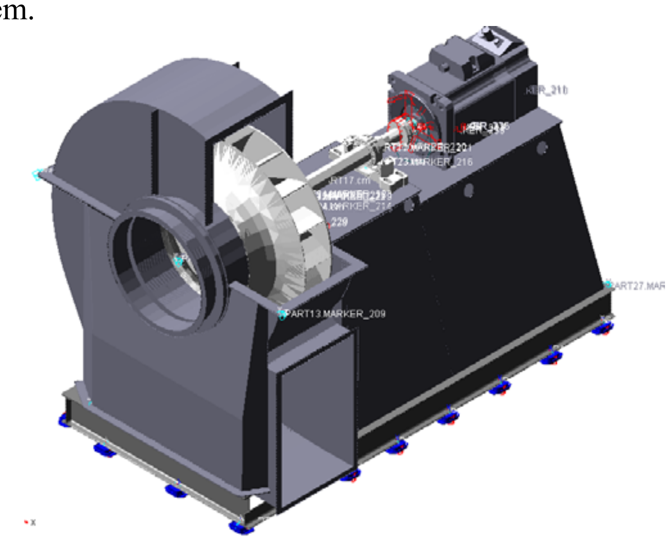


Fig. 7: Scheme of a dynamic model of a fan as a stiff body supported flexibly

Resonance frequencies corresponding to typical translational forms (a-c) are of lower value than 3.9Hz which is higher than the limit of the operation range designed for this type of vibroisolators by their producers.

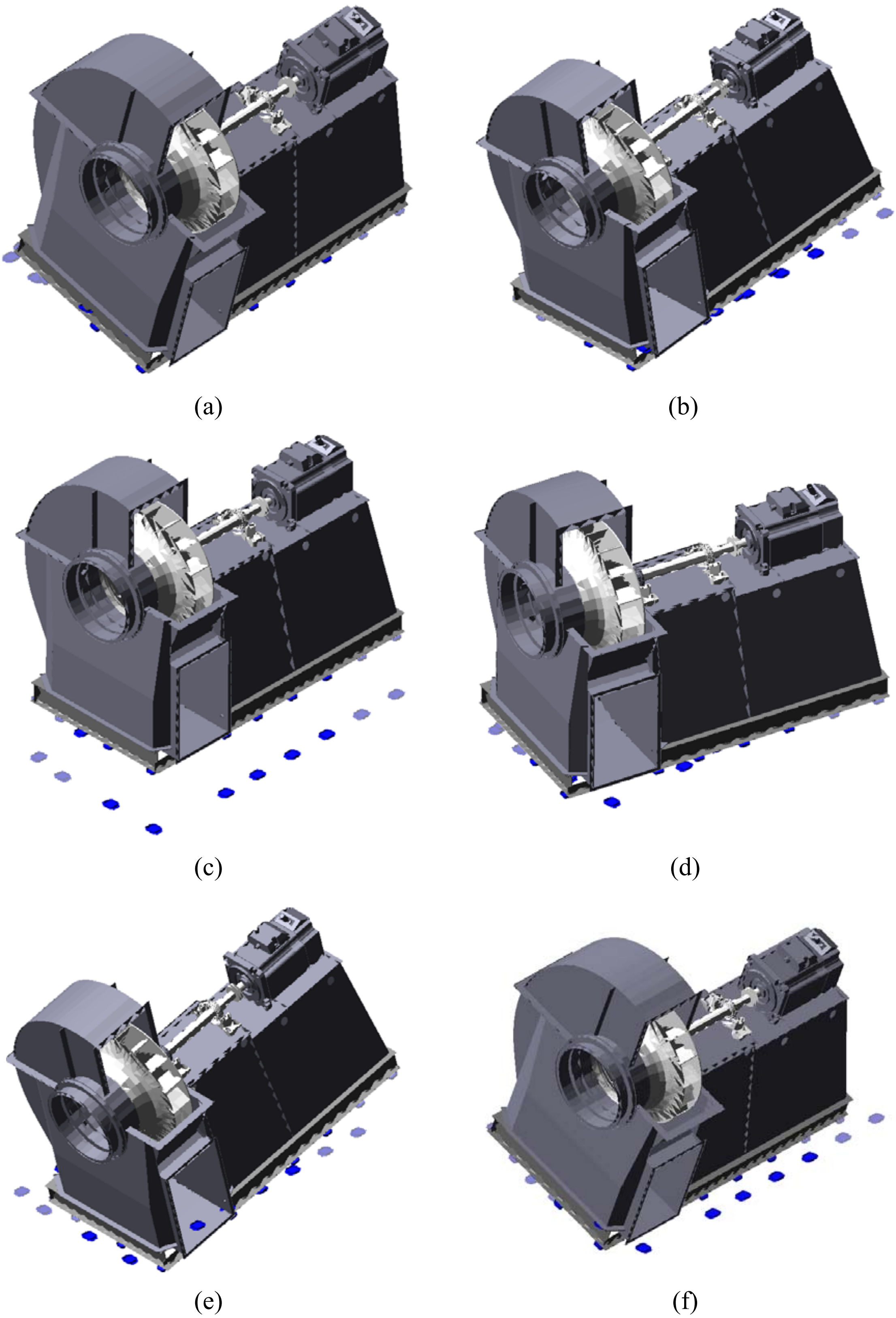


Fig 8: Forms of the fan body free vibrations and frequencies corresponding to them
(a)-1.55HZ, ((b) 1.94HZ, (c) 3.38Hz, (d) 3.9Hz, (e) 5.10Hz, (f) 6.28Hz

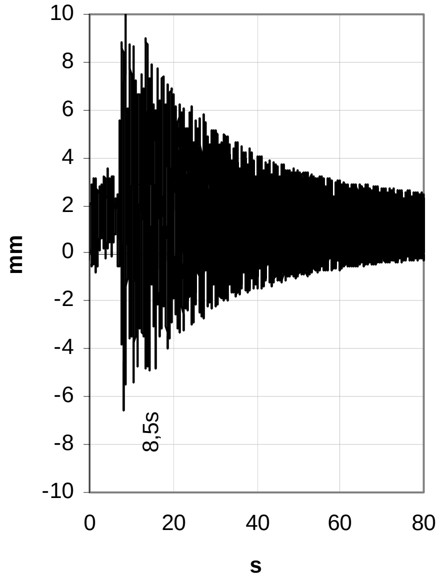


Fig 9: Horizontal displacement of the bearing near the disk during the rotor start running

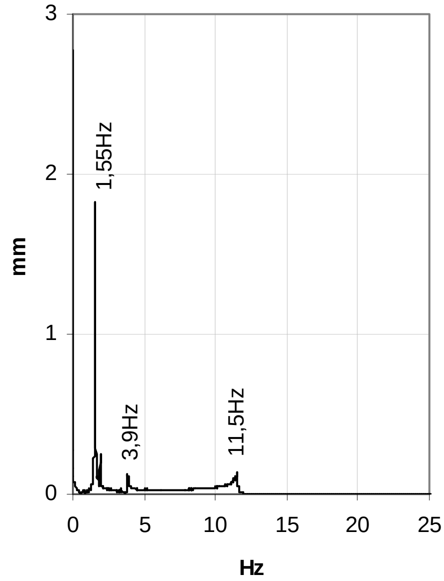


Fig 10: Spectrum of the bearing vibrations horizontal displacement near the disk during the rotor start running

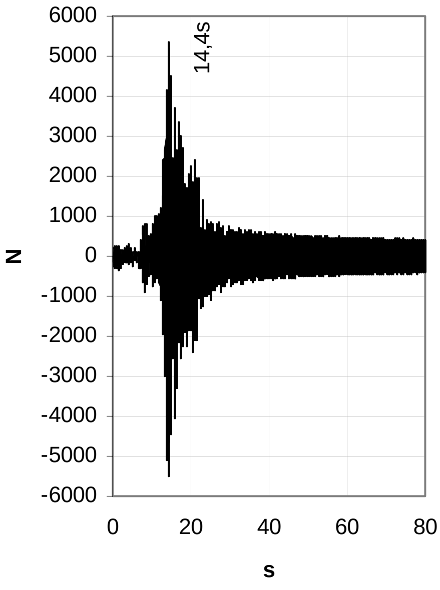


Fig 11: Change in time of the force horizontal component in the vibroisolator springy-damping element

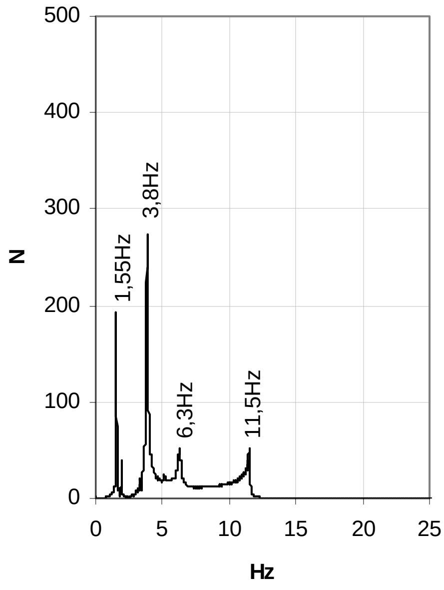
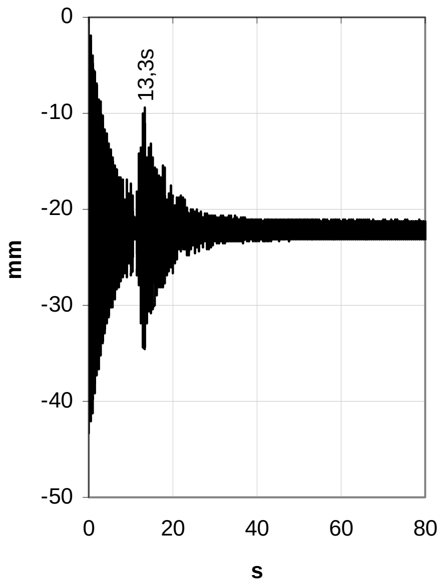


Fig 12: Aamplitude-frequency characteristic of the force horizontal component changes inside the springy-damping element



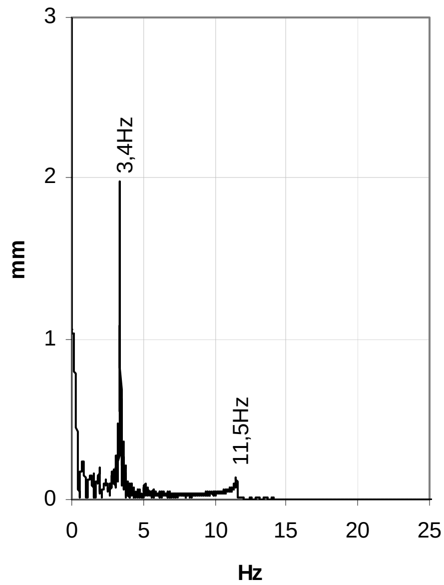
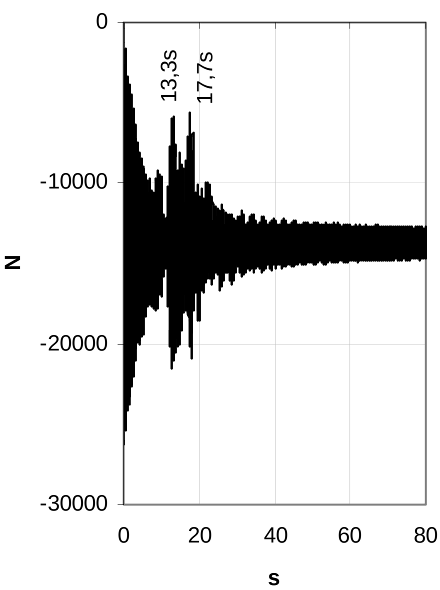


Fig. 13: The rotor vertical displacement near the disk during the rotor start running

Fig. 14: Spectrum of vertical displacement of the bearing vibrations near the disk during the rotor start running



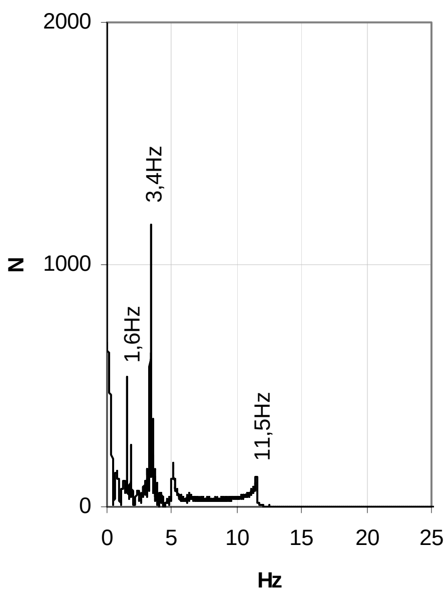


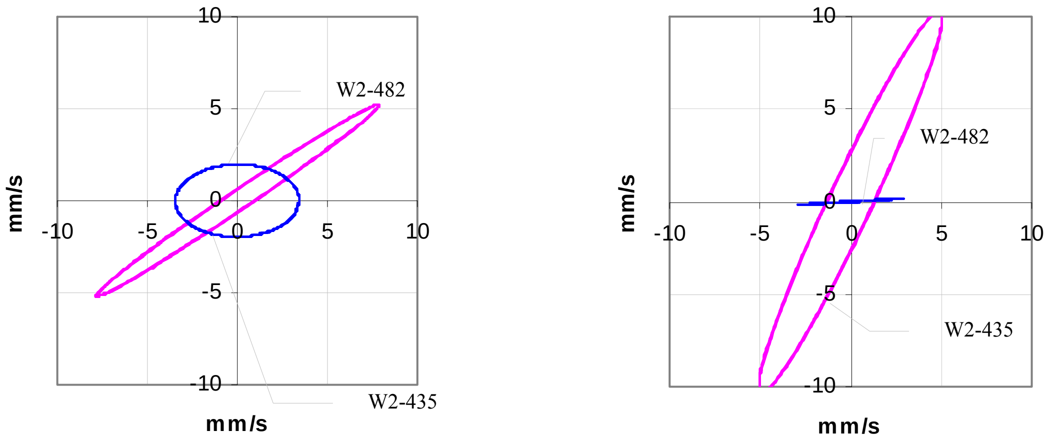
Fig. 15: Change in time of the vertical force constituent in the springy-damping vibroisolator

Fig 16. Amplitude-frequency characteristic of vertical force constituent change inside the springy element of the vibroisolator

Simulation of dynamic behaviors of fan during the rotor start running for rotary velocity 690 r.p.m. are presented in Figs. 5 ÷ 12.

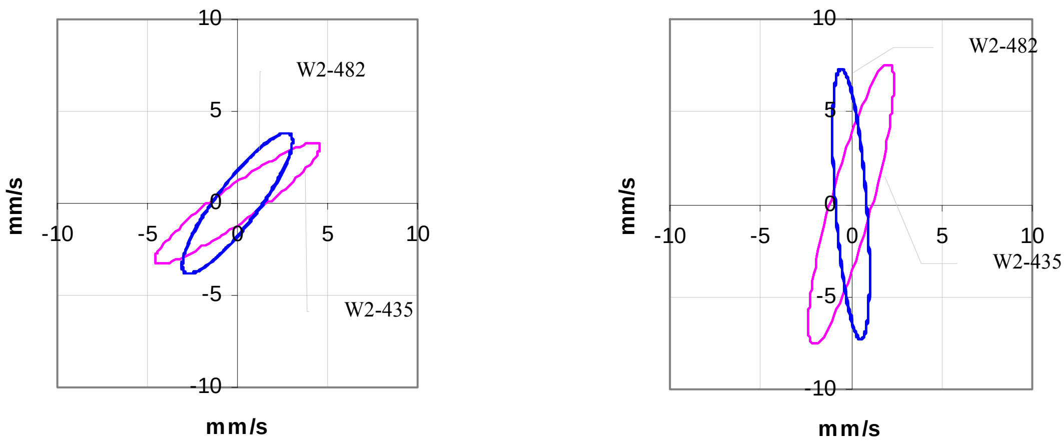
The real character of the 436FA76 ventilator engine bearing in dependence on stiffness of the used fan is defined by phase-amplitude characteristics, presented in Figs 13 ÷ 16.

It can be noticed that change of stiffness of 15 vibroisolators arranged on the frame of the fan mounting changes the character of its vibrations significantly. Generally, a significant effect of vibration amplitude damping is obtained for excitement frequency 9Hz (Fig. 13). With increase of exciting force frequency up to 10Hz the effect of W2-482 vibroisolators vibroisolation is similar to the previously obtained one with a less stiff vibroisolator mounting (Fig. 14). An interesting effect is obtained with rotational frequency 11.7 Hz and 12.5 Hz. The form of the object vibration favors the situation in which the engine part of the fan undergoes damping whereas vibration velocity amplitudes of the body fragment on which the engine is placed increase. (Fig.5.22-5.23).



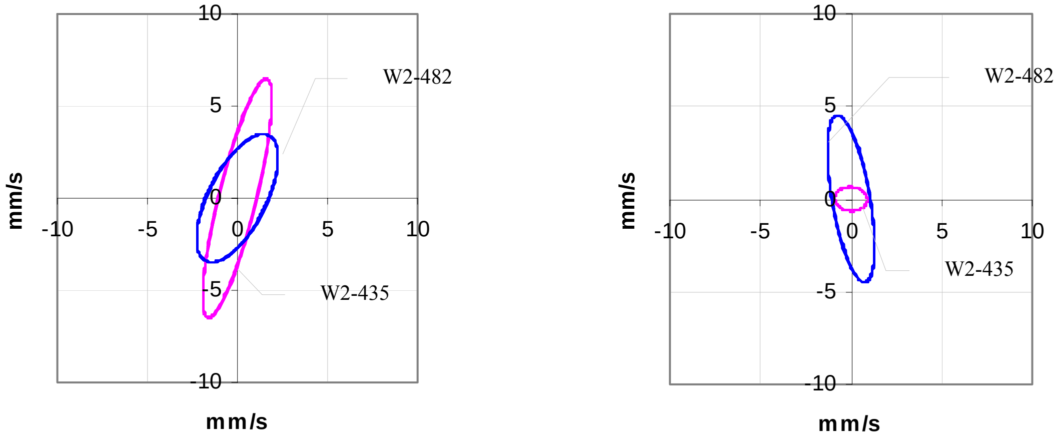
(a) (b)

Fig. 17: Amplitude-phase characteristic of the fan bearing vibration velocity (a) near the engine, (b) rear bearing of the engine with excitement frequency 9.2Hz.



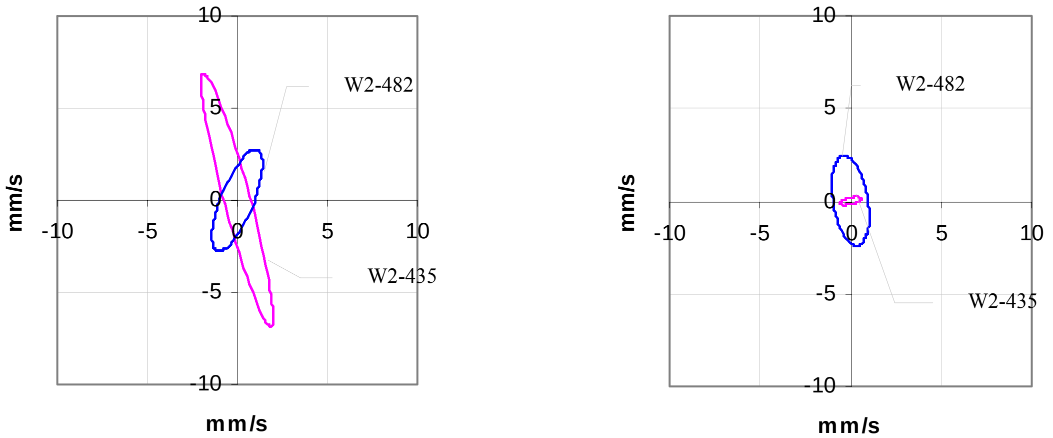
(a) (b)

Fig. 18: Amplitude-phase characteristic of the fan bearing vibration velocity (a) near the rotor, (b) rear bearing of the engine with excitement frequencies 10Hz



(a) (b)

Fig. 19: Amplitude-phase characteristic of the fan bearing vibration velocity (a) near the rotor, (b) rear bearing of the engine with excitement frequency 11.7Hz



(a) (b)

Fig. 20: Amplitude-phase characteristic of the fan bearing vibration velocity (a) near the rotor, (b) rear bearing of the engine for forcing frequency velocity 12.5Hz

4. Conclusions

In presented paper the authors wish to show that problems connected with industry machines vibroisolation are not simple issues. As object taken into consideration was chosen the radial fan, which is fluid-flow machine commonly used in many industry branches. As process line element it is very often placed on workshops cellars. In this place dynamic excitation with high amplitude strongly affects building construction. In this case necessity of limitation forces magnitude transferred at machine foundation is very important [Zachwieja J (2007)]. The proper choice of vibroisolators rigidity and damping requires deep analysis using knowledge on both machine dynamics and calculation methods which allow using this knowledge.

From carried out analysis we can draw following conclusions, which seems to be obvious:

1. Vibroisolators choice on stage machine foundation design must be preceded by analysis of the object dynamics. The proper tool of this aim are numerical methods which make possible multi body systems modeling. In this case one may by both rigidity and damping simulation chose vibroisolators with proper characteristics and also planning their placing on foundation. In case of damping problem optimization for existing objects is necessary to determine resonance characteristics for frequency interval including whole range of rotor rotational speed.
2. Vibroisolator operates effective if ratio excitation frequency and free vibration frequency of damping object meet dependence

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

   Vibroisolator effectiveness is understood as possibility to decrease magnitude of force transferred from fan rotor to foundation. But this condition fulfillment causes increase of amplitude system vibrations, what also unfavorably affects it elements life. Vibroisolators choice must be provided in such way that amplitudes of vibration parameters were in scope considered as acceptable or transitory acceptable.
3. One may be conscious that damping system effectiveness takes place for given frequency. It follows that vibroisolator can decrease system vibration amplitude value if its working point is properly placed on resonance characteristic. While excitation frequency is changing it is possible that in new conditions vibroisolator effectiveness is insufficient. It is unacceptable that rotor rotational frequency is near resonance frequency. In case rotors with long shaft when on one side is placed disk with large mass and large moment of inertia and on the other side clutch disk, very difficult for damping are vibrations in form of cone axode, which rotor runs in the space. For such resonance form rotor balance in it own bearings is very difficult.

References

Feng N.S. & Hahn E.J. (1995) Including foundation effects on the vibration behaviour of rotating machinery, *Mechanical Systems and Signal Processing* 9(3), pp. 243-256.

Fortuna S. (1998) *Wentylatory* (in English: Fans), TECHWENT, Kraków 1999.

Zachwieja J., (2005) Diagnosis of horizontal ventilator rotor with small support stiffness, *Diagnostyka* 35, pp. 63-70.

Zachwieja J. & Peszyński K. (2007) Experimental test of flow parameters influence on ventilators system vibrations, *Proc. of International Conference Experimental Fluid Mechanics*, Liberec (Czech Republic), pp. 166-173.