STATIONARY AND TRANSIENT VIBRATIONS OF DAVI VIBROINSULATION SYSTEM.

Jerzy Michalczyk, ul.Zagorze 1c, 30 –398 Cracow, Poland
Piotr Czubak, ul. Szujskiego 11/4, 31-123 Cracow, Poland – address for correspondence
Grzegorz Cieplok, ul.Graniczna 27a, 41-408 Myslowice, Poland.
e-mail: czubak@uci.agh.edu.pl
DEPARTMENT OF MECHANICS AND VIBRO-ACOUSTICS
University of Mining and Metallurgy, Cracow, Poland.

ABSTRACT:

Analysis of the new, patented, hydraulic dynamic vibroinsulator system of the DAVI type is presented in the paper. The described device applied to the classical vibroinsulation system, subjected to harmonic load, significantly improves its efficiency by means of generating additional forces shifted in phase by π angle against forces exerted by elastic suspensions. This vibroinsulator, contrary to the Frahm's one, does not increase the number of degrees of freedom of the system.

The analytical solution of a stationary motion as well as computer simulations of transient resonance phenomena together with experimental confirmations of the results are included in the paper. Semiactive system designed for variable angular velocity of exciting force is also analysed.

Key words: dynamic antiresonant vibroinsulation, vibratory machine, semiactive systems.

1.INTRODUCTION

An idea of the dynamic antiresonant vibroinsulation [1] in relation to the force vibroinsulation is based on the generation of additional dynamic forces being in anti-phases and of equal values with the forces transmitted to the base by a classic vibroinsulation system.

The task can be solved by passive elements if only the vibroinsulating system performs sinusoidal motion and is not heavily dumped. By adding a certain inert system to supporting springs the theoretical balancing of forces transmitted from the vibrating object to the base can be achieved. This will not limit the static stiffness of the system. The effect occurs in the system shown in Fig.1 [2].

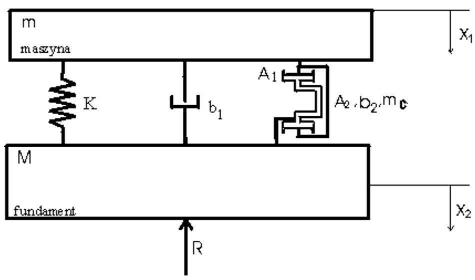


Fig.1 Analysed model of the vibroinsulating system.

The machine body, m, in a steady state, performs harmonic vibrations $x_1(t)$. The vibroinsulation system of elasticity and dumping constants, K and b_1 – respectively, is influencing the foundation M not only by a static component originated from the machine weight but also by a dynamic component connected with cyclic deformation of springs K and the damper b_1.

This force causes vibrations $x_2(t)$ of the foundation M, on which the reaction R(t) acts from the side of the base. Into this system a container with liquid was attached in such a way that its body was affixed to the foundation. The container consists of two chambers, upper and lower, closed by pistons (membranes) of surface A_1. Both chambers are connected by a channel of diameter A_2 containing liquid of a mass m_c. Membranes are stiffly affixed to the machine body. Fig. 8 shows the diagram of the DAVI rubber-oil vibroinsulator. Assuming machine body motion as $x_1(t)=X\sin(\omega t)$, the equation of the foundation motion derived on the bases of Lagrange's method is as follows:

$$\ddot{x}_2\left[M+m_c\left(1-\frac{A_1}{A_2}\right)^2\right]+\dot{x}_2\left[b_1+b_2\left(\frac{A_1}{A_2}\right)^2\right]+x_2K= \tag{1}$$
$$=-w^2X\sin(wt)m_c\left(\frac{A_1}{A_2}\right)\left(\frac{A_1}{A_2}-1\right)+wX\cos(wt)\left[b_1+b_2\left(\frac{A_1}{A_2}\right)^2\right]+X\sin(wt)K-R$$

where: b_2 - viscous damping coefficient in channel A_2.

It has been assumed that there is a laminar flow in the channel.

The condition for the rest of the foundation M, in a steady state, is the zeroing of the right-hand side of equation (1). Since our aim is that the reaction transmitted into surroundings would be zero, R = 0, finally the condition for non-transmitting of vibrations to the base is given by the formula:

$$-w^2X\sin(wt)m_c\frac{A_1}{A_2}\left(\frac{A_1}{A_2}-1\right)+wX\cos(wt)\left[b_1+b_2\left(\frac{A_1}{A_2}\right)^2\right]+X\sin(wt)K=0 \tag{2}$$

Two cases should be considered:

1. $b_1 \cup b_2 \neq 0$
2. $b_1 \cap b_2 \approx 0$

In the first case none of the parameter value will satisfy the equation.

In the second case the equation will take the form:

$$-w^2X\sin(wt)m_c\left(\frac{A_1}{A_2}\right)\left(\frac{A_1}{A_2}-1\right)+X\sin(wt)K=0 \tag{3}$$

Hence, the condition for zeroing of the force transmitted into the foundation at a weak dumping value can be formulated as:

$$w^2m_c\frac{A_1}{A_2}\left(\frac{A_1}{A_2}-1\right)=K \tag{4}$$

what can be satisfied for $A_1 > A_2$ only. In real situations where $A_1 >> A_2$ the above condition can be written, in an approximation, as:

$$w^2m_c\left(\frac{A_1}{A_2}\right)^2\cong K \tag{4a}$$

If, at the given frequency, ω, the vibroinsulator parameters m_c, A_1, A_2 and K are selected in such a way as to satisfy the equation (4), the force transmitted to the machine foundation will theoretically be of zero value.

This happens, because forces generated by elastic elements and those connected with the inertia of liquid present in the vibroinsulator channel cancel each other. Those forces are in an anti-phase and have equal amplitudes and frequencies.

It can be derived that the transmission coefficient p of the system equals:

$$p = \frac{|R(t)|_{\max}}{P_0} = \frac{1-\left(\dfrac{w}{w_0}\right)^2}{1-\left(\dfrac{w}{w_n}\right)^2} \qquad (5)$$

As can be seen from equation (5) the vibroinsulating system with the damping element described above has only one resonance range, while systems with dynamic dampers have two such ranges, one being in the direct vicinity of the working frequency [3].

2. SIMULATION OF THE RUBBER-OIL VIBROINSULATOR

Previously analyzed model which allows to derive the condition for zeroing of the force transmitted into the base does not illustrate several significant phenomena occurring in real systems. It does not include e.g. resonance phenomena, influence of mass and dumping in liquid on the vibration amplitude in steady state conditions, etc. In order to be able to estimate those phenomena the simulating model of the two-rotor vibratory machine was built (Fig. 2) [4].

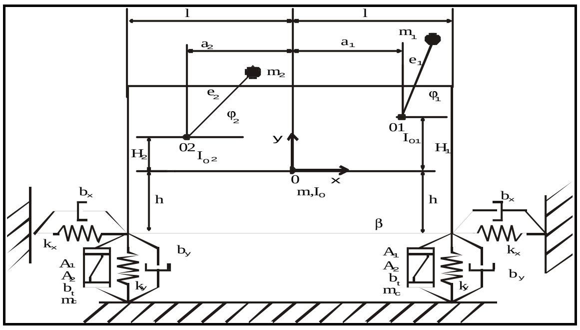


Fig. 2.The model of a vibroinsulated two-rotor machine

The simulated system was symmetrical towards the y axis. Simulation was performed for such values of initial parameters for which the sum of unbalanced forces in the y direction, in the steady state, equals: F = 2468[N].

Paper [4] deals with the simulation of the above system performed for typical parameters. The diagram presented in Fig. 3 gives the comparison of the forces transmitted to the base – in the steady state – for the system vibroinsulated by the rubber-oil vibroinsulator (broken line) and the system not insulated in such a way (solid line). The analyzed type of a vibroinsulator allows to lower the force value transmitted to the base - in the steady working point - to 8.3 % of the value of the classic vibroinsulation (both of the same static stiffness).

Fig. 4 presents the amplitude-frequency characteristics of the forces transmitted to the base. The force transmitted by a classic vibroinsulator is shown by a broken line, while the one transmitted by the oil-rubber vibroinsulator is shown by a solid line. This diagram was determined on the bases of steady states obtained in the time simulations. The resonance amplitude of the machine equipped with a vibroinsulator is more than two times lower than the resonance amplitude of the machine operating without that vibroinsulator – for the case of a stationary resonance.

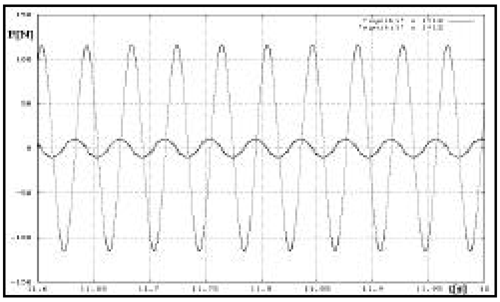


Fig. 3 Forces transmitted to the base by two types of vibroinsulation.

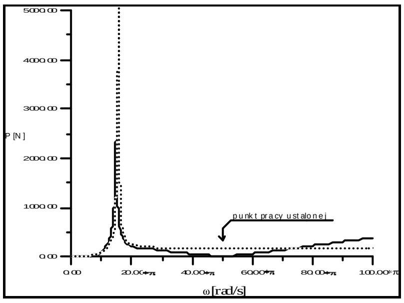


Fig. 4 The dependence of forces transmitted to the base on the forced frequency.

3. STRATEGY OF OPERATING THE RUBBER-OIL INSULATOR.

The inherent defect of the presented above solution is that at the start-up of the machine the suspension system equipped with the vibroinsulator behaves only slightly better than the system of a classic suspension.

Therefore in the presented hereby paper the possibility of controlling the inner diameter of the channel inside the vibroinsulator was investigated. The channel (3) inside the vibroinsulator, shown in Fig. 5, can be constructed in such a way that its inner part has a regulated diameter enabling to control the area A_2 of the channel (3) as a function of an angular frequency of vibrations.

Schematic presentation of the semiactive vibroinsulation is given in Fig.5.

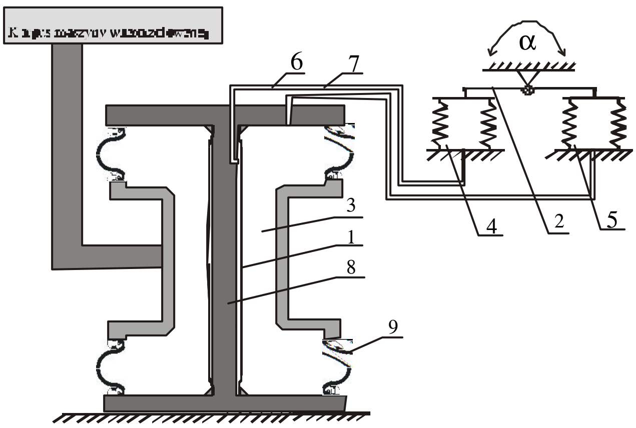


Fig. 5 Diagram of a semiactive vibroinsulator

It is possible to control the inner diameter of the channel (3) by regulating the amount of liquid in the space between the mandrel (8) and a rubber clamping ring (1). By means of that the cross area A_2 of the channel can be regulated. The system of two bellows' (4) and (5) allows to retain the constant amount of liquid inside the vibroinsulator preventing any excessive increase of pressure to act on rubber membranes (9). The amount of liquid in each part of the vibroinsulator is controlled by the change of the position of the lever 2.

Thus in transient states the diameter d_2 can be adjusted to the actual value of an angular frequency of vibration ω in such a way that equation 4 will be satisfied all the time. The semiactive system was applied to the previously simulated system. The forces transmitted to the base during the start-up and the steady operation are shown in diagrams in Fig. 6 and Fig. 7 for the system with the semiactive vibroinsulator and the typical DAVI vibroinsulator – respectively. For the start-up operation of the system with the semiactive vibroinsulator 2.5 times lowering of the force transmitted to the base was obtained.

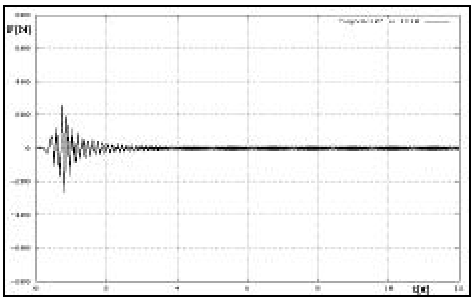


Fig. 6 Forces transmitted to the base by the system with the semiactive vibroinsulator.

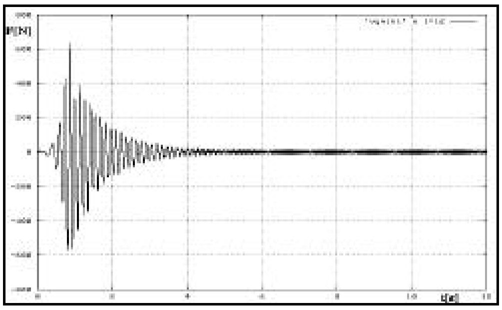


Fig. 7 Forces transmitted to the base by the system with the typical DAVI vibroinsulator

4. EXPERIMENTAL PART

The rubber-oil vibroinsulator with a rectilinear channel was designed and constructed for experimental purposes (Fig.8) in which rubber membranes have elasticity of constant K.

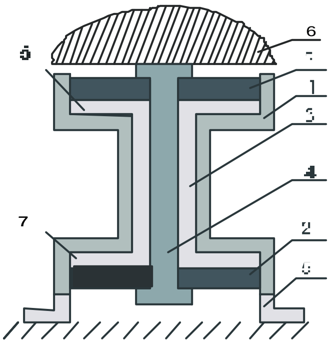
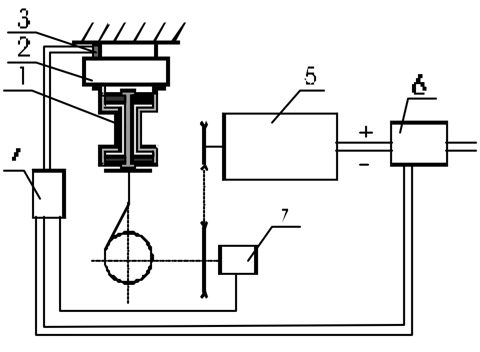


Fig.8. The diagram of the DAVI rubber-oil vibroinsulator.

In accordance with equation (4), it was designed in such a way as to transmit to the base the minimal force at the exciting frequency of 15 Hz. The vibroinsulator was subjected to the kinematic exciting force of the frequency regulated in the range from 3 to 25 Hz and the vibration amplitude equalled app. 2.5 mm.



1- vibroinsulator
2- fixing of vibroinsulator
3- force sensor
4- measuring system
5- direct-current motor
6- supply system
7- tachogenerator

Fig .9. The scheme of the experimental set up.

The vibroinsulator was subjected to the kinematic excitation by a slider crank mechanism. At the opposite side of the vibroinsulator the measurements of the force transmitted were made.

The aim of measurements was to obtain the dependence of the maximal amplitudes of the force transmitted to the base on the frequency of the kinematic exciting .

Experiments were performed for two different cases. When the rubber-oil vibroinsulator is filled with liquid the force connected with its inertia should oppose the one generated by elastic membranes (both forces are in an anti-phase) and this reduces the force value transmitted to the base. When the vibroinsulator is empty the system behaves as a classic rubber vibroinsulator.

Plots of force vs. frequency for the two types of vibroinsulators are shown in Fig.10. The dependence of the force transmitted to the base on the forced frequency for the rubber-oil vibroinsulator and the rubber one are shown by solid and broken lines — respectively. Fig. 11 gives the comparison of the force transmitted to the base versus time (at forced frequency equal 15 Hz) for the rubber-oil vibroinsulator and for the classic one. The force transmitted by the rubber-oil vibroinsulator equals 50 N while the one transmitted by rubber insulator equals 180 N.

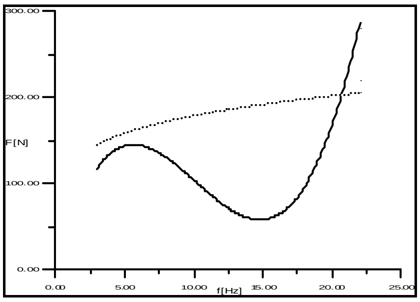


Fig.10. Plot of force vs. frequency

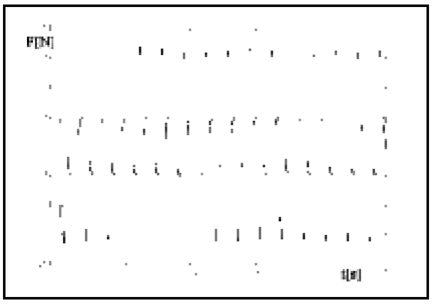


Fig. 11. Plot of force vs. time

5. CONCLUSIONS

1. The analysed type of a vibroinsulator allows to lower the force value transmitted to the base – in the steady working point – to 8.3 % of the value of the classic vibroinsulation (both of the same static stiffness).
2. The resonance amplitude of the machine equipped with a vibroinsulator is more than two times lower than the resonance amplitude of the machine operating without that vibroinsulator – for the case of a stationary resonance.
3. 2.5 times lowering of the force transmitted to the base was obtained for the start-up operation of the system with the semiactive vibroinsulator.
4. Experimental results fully confirmed the correct action of the rubber-oil vibroinsulator. The reduction of the force value transmitted to the base at the working point was 3.6 times larger than in the case of the classic vibroinsulator.

REFERENCES

1. G.Done, 1978, “Vibration Control by Passive Means Other than Using Damping”, Dynamika maszyn, Ossolineum, Wroclaw , 63-116.
2. J. Michalczyk, 19. 08. 1996, “Wibroizolator”, Pat. RP No. 175447.
3. J.Michalczyk, G. Cieplok, 1999, “ Wysokoaktywne uklady wibroizolacji i redukcji drgan”, Coll. Columbinum, Cracow.
4. J.Michalczyk, P.Czubak, 1999,”Analiza teoretyczna i badania symulacyjne antyrezonansowego wibroizolatora dynamicznego”, Mechanika, Tom 18, Zeszyt 1.