ACTUAL PROBLEMS OF MODERN SCIENCE

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CONTENT

DESIGN METHODS FOR REDUCTION OF FORCED VIBRATIONS OF HORIZONTAL ROTARY MACHINES

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1. Introduction

Rotary mechanisms are used in many areas of modern industry, from mechanical engineering to computer and household appliances. Since these mechanisms often have to operate at high speeds, strong vibrations caused by the rotor's center of gravity shift can become a serious problem and even lead to damage to the mechanism. The issues of reducing vibration excited by rotating rotors are among the most important in the design, manufacture and operation of almost all types of modern rotary machines.

The washing machine as an object of study of dynamics and reduction of vibrations and noise is of particular interest due to the constant presence of randomly located and wandering imbalance of laundry in the drum and low requirements for the accuracy of its manufacture and assembly of parts and assemblies so as not to increase the cost.

2. Literature Review

The solution of problems related to the problems of reducing rotor vibration is based on the linear theory of mechanical vibrations. The theory also indicates the main directions of the fight against vibration: vibration isolation [1]; damping [2]; dynamic vibration damping [3]; improvement of methods for balancing rotors [4], including taking into account their flexibility [5]. Many papers investigate electromagnetic balancing devices in the form of electromagnetic bearings [6] and electromagnetic rings [7]. A synchronous radial force can be applied to the shaft to balance the imbalance [8]. Methods of optimal vibration control are widely used [9]. Most of the scientific robots are devoted to the study of the phenomenon of passive automatic balancing using ball, pendulum autobalancers [10] and due to the free movement of fluid.

Although researchers have recently made significant progress in developing active and passive methods and tools to reduce the vibration of washing machines, the level of unwanted vibration can also be reduced by the optimal layout of the components of the machine, which studies this work.

3. Vibration isolation system mathematical model

Consider the oscillations of an elastically suspended tub containing a cantilevered rotating unbalanced drum. Such a scheme is typical of machines with a horizontally located laundry tub, for example, "LG", "Electrolux", etc (Fig. 1, 2). The tub-drum system generally makes six-link oscillations.

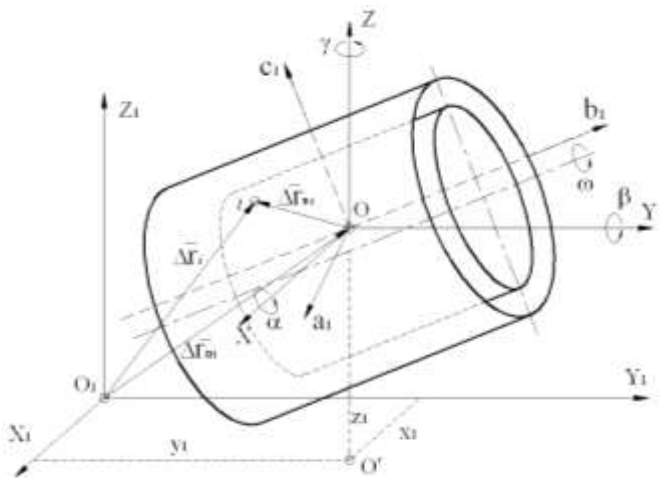


Fig. 1. Scheme of the tub-drum system

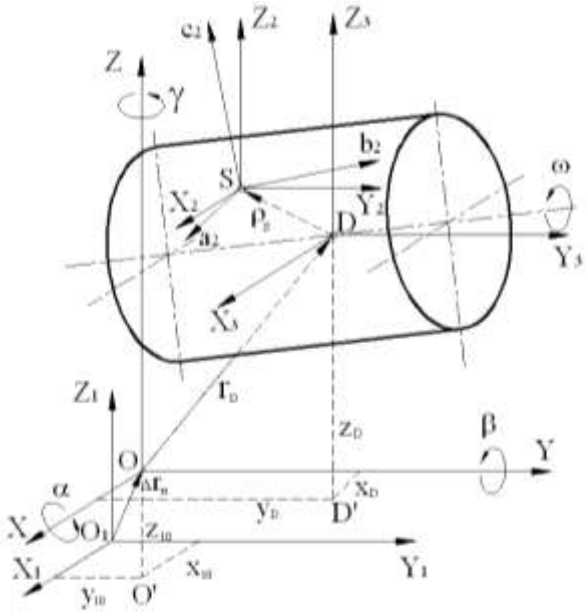


Fig. 2. Moving the drum in general

The system obtained by the authors of six differential equations in matrix form has the form:

$$\mathbf{M}\ddot{\mathbf{q}} + (\mathbf{G} + \mathbf{D})\dot{\mathbf{q}} + \mathbf{A}\mathbf{q} = \mathbf{Q}, \qquad (1)$$

where $\mathbf{M}$ is the matrix of inertial coefficients; $\mathbf{G}$ is the matrix of gyroscopic coefficients; $\mathbf{D}$ is the damping coefficient matrix; $\mathbf{A}$ is the stiffness matrix; $\mathbf{q} = \left[x, y, z, \alpha, \beta, \gamma\right]^T$ is the matrix-column of generalized coordinates; $\mathbf{Q} = \left[Q_x, Q_y, Q_z, Q_\alpha, Q_\beta, Q_\gamma\right]^T$ – is the matrix-column of generalized force factors.

4. Experimental verification

The efficiency, adequacy and accuracy of the developed mathematical model were checked by means of field tests. For this purpose, a laboratory stand containing a household washing machine was used). The theoretically calculated natural frequencies of the oscillating system of the washing machine differed from the experimentally obtained no more than 10%. The calculated amplitude of vertical oscillations at an artificial imbalance of the drum 1800 g·mm was 3.7 mm, and experimentally measured at the same imbalance 4 mm.

5. Dynamic simulation and analysis

The verification of the formulated design requirements was performed by modeling the behavior of the suspended part of the washing machine using the interactive tool *Simulink*. The dynamic model of the machine is presented in fig. 3. The suspended part of the car is fastened by means of two springs and two dampers. The dynamic characteristics of the washing machine are presented in table 1. In practice, due to the random nature of the distribution of loaded laundry in the drum, the center of mass of the drum will always lie not on the axis of rotation. It is established that the greatest eccentricity during the decomposition of linen in the drum is $e = 0.08R_б$, were $R_б$ is the drum radius. For the studied model of the washing machine is accepted $e = 14{,}8$ мм.

In fig. 4 presents the Frequency response function (FRF), obtained for vertical oscillations of the suspended part of the washing machine, taking into account that the center of mass of the tub lies on the axis of rotation of the drum and the drum has only static imbalance, characterized by eccentricity e and $Q_y = Q_\alpha = Q_\beta = Q_\gamma = 0$, and $Q_x, Q_y \neq 0$. Due to the force perturbation along the axes *OX* and *OZ* there are transverse oscillations in the plane *XOZ*.

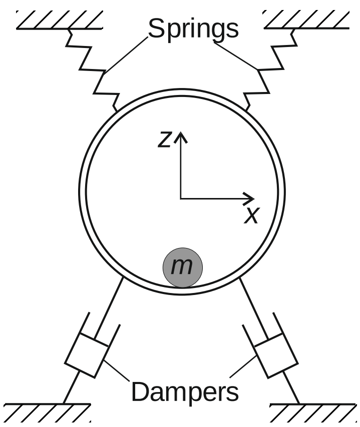


Fig. 3. Dynamic model of the washing machine

Table 1. Dynamic characteristics of the washing machine

Tub weight m_1, kg	48.2
Drum mass m_2, kg	12.5
Stiffness of each of the two suspension springs c, N/mm	6,0
The angle of the springs	75°
The damping factor of each damper h, N·s/m	175
The angle of the dampers	70°
Eccentricity e, мм	14,8
Drum length $L_б$, mm	190

Due to the random arrangement of the laundry, the center of mass of the loaded drum in addition to the eccentricity may have a longitudinal displacement y_D along the axis OY, which causes a imbalance of the rotor. For values y_D=0...90 mm model was simulated in the entire operating frequency range. The increase in angular oscillations causes an increase in the displacement of the suspension part.

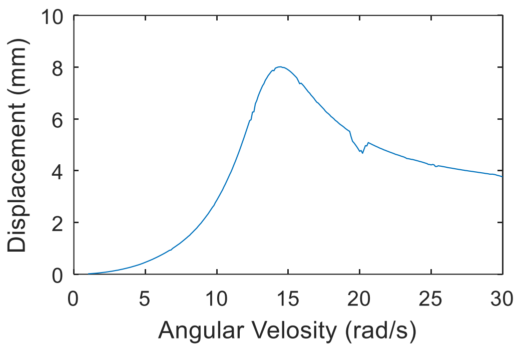


Fig. 4. FRF transverse oscillations in the direction of the *OZ* axis

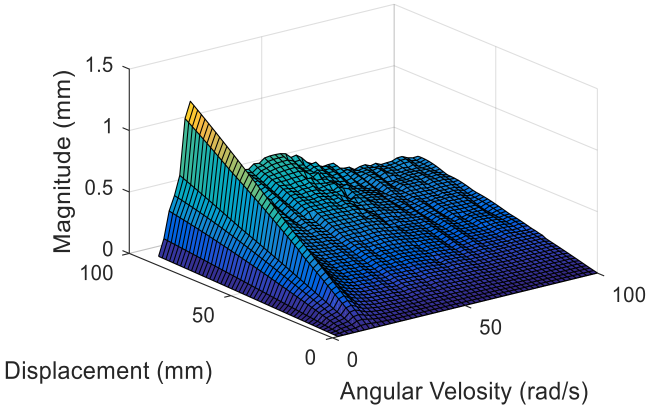


Fig. 5. The magnitude of oscillations in the longitudinal direction *OY*

In fig. 6 and 7 show the results of the simulation of the tub-drum system at different values of the deviation l_c of the position of the center of rigidity of the elastic supports of the suspended part from the center of gravity of the tub (vertical oscillations).

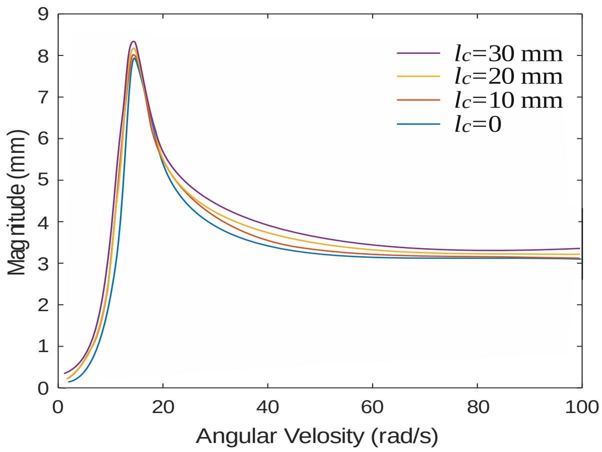


Fig. 6. FRF ($y_D = 0$)

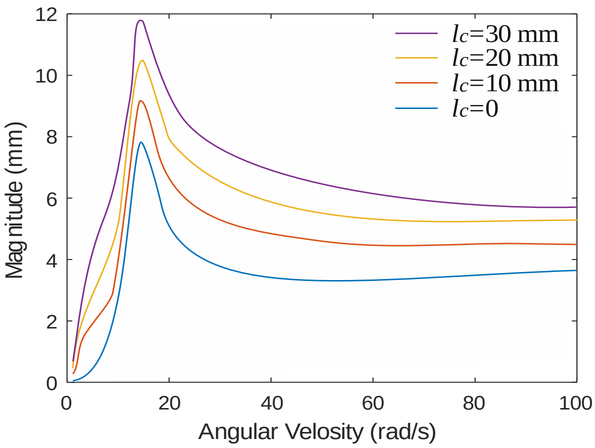


Рис. 7. FRF ($y_D = 90$ mm)

In the case of rotation of the elastic (damping) support at an angle φ relative to the plane XOZ the support system is non-uniformly rigid, and the stiffening matrix (damping) is non-diagonal. For the spring support of the tub with axial stiffness C, the relations are fulfilled

$$C=\sqrt{C_x^2+C_y^2+C_z^2}\,,\; C_x=C\cos\chi\cos\varphi,\; C_y=C\cos\chi\sin\varphi,\; C_z=C\sin\chi\cos\varphi,$$

were χ is the angle of the spring, φ is the angle of rotation of the spring relative to the plane *XOZ* (Fig. 8, 9).

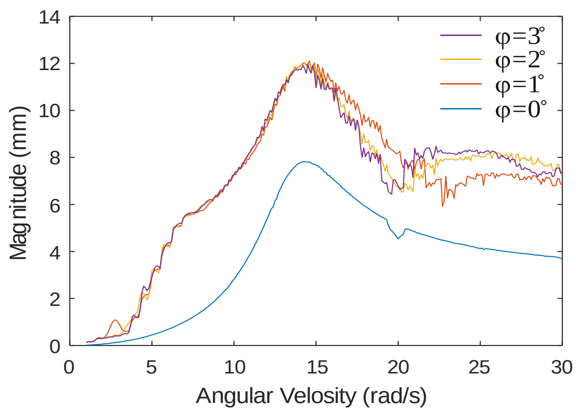


Рис. 8. FRF at $y_{D\max}$ for different angles φ of rotation of springs

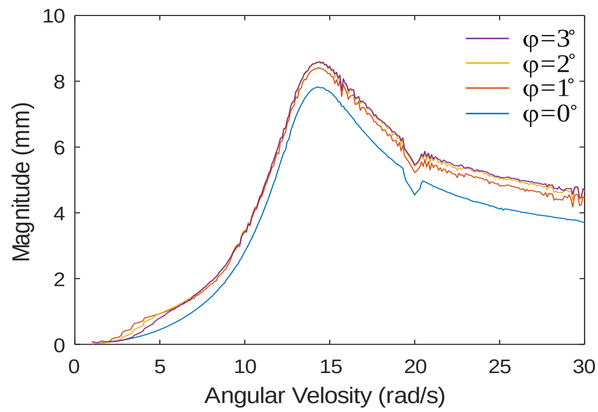


Fig. 9. FRF at $y_{D\max}$ for different angles φ of rotation of the damper

Liquid auto-balancing was used to reduce the vibrations of this machine. To do this, on the basis of mathematical modeling, we will consider the operation of an autobalancer of the Leblanc type.

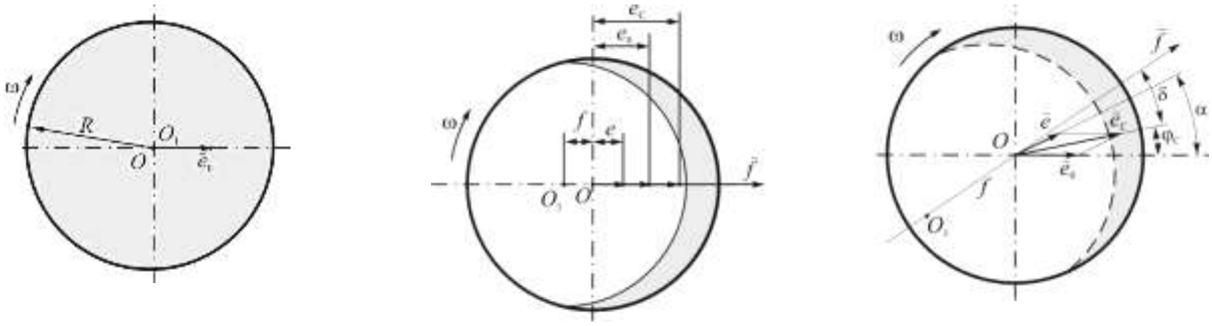
6. Physical content of fluid balancing

In the absence of an Automatic Balancing Device (ABD), the value of the system imbalance remains unchanged in magnitude and direction, and since there is a relationship between the deflection of the shaft, the magnitude of the imbalance and the angular velocity of rotation:

$$f=e_0\left(\omega/\omega_{кр}\right)^2\left(1-\left(\omega/\omega_{кр}\right)^2\right)^{-1}$$

then, obviously, with the increase of the angular velocity ω, the deflection f is continuously increases by subcritical rotation of the shaft.

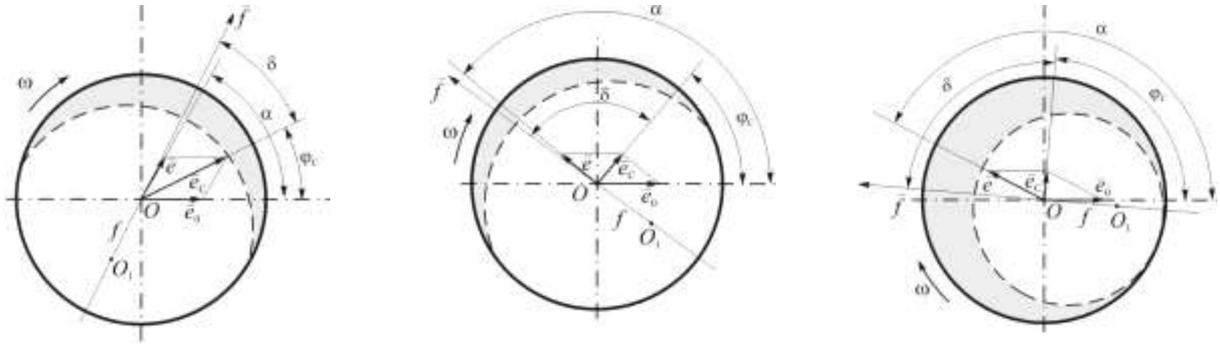
In the case of an ABD, the imbalance of the system $\overline{e}_c$ consists of a stationary imbalance created by the unbalance $\overline{e}_c$ of the shaft, and an imbalance created by the liquid $\overline{e}_c$. At ω =0, the liquid imbalance is zero (Fig. 10 a))



a) b) c)

Fig. 10. Vector model of process of auto-balance liquid

At the initial moment of rotation, the unbalance of the shaft results in a deflection f, which coincides in the direction with the rotor imbalance (Fig. 10 b)). The centrifugal forces reject fluid to the walls of the ABD and its bulk is concentrated in the deflection, as shown by the established law of pressure distribution in the liquid. With an increase in the speed of rotation of the rotor, the direction of the deflection begins to lag behind the direction of the imbalance at the angle δ (in the presence of fluid in the ABD chamber, the angle δ should be deducted from the total imbalance of the rotor) (Fig. 10 c)). The liquid tends to occupy the position at the furthest point from the axis of rotation, namely in the trough, and moves along with it from the imbalance. This in turn leads to a change in the total imbalance of the system in terms of direction and magnitude (increases as the value of the total imbalance is defined as the geometric sum of the vectors $\overline{e_0}$ and $\bar{e}$ and depends on the angle between these vectors α). This change leads to an increase in deflection, and an increase in the speed of rotation results in an increase in the angle δ. The fluid following the deflection changes the total imbalance by increasing the angle α and decreasing the total imbalance (Fig. 11 d). That in turn leads to a decrease in the value of the deflection. Consequently, with an increase in the speed of rotation due to the decrease e_C, we have a decrease in the deflection by magnitude even in the subcritical rotor rotation mode. Because $tg\delta = 2n\omega\left(\omega_{res}^2 - \omega^2\right)^{-1}$ then when the speed of rotation approaches the critical δ, it increases and goes to 90 °.



d). e) f)

Fig. 11. Vector model of autobalancing process (continued)

Because δ is the angle between the deflection and the total imbalance, and φ_C is the angle between the stationary and total imbalances, the angle between the deflection and the stationary imbalance when rotating the rotor with the angular velocity is equal to $90° + \varphi_C$ (Fig. 12). In the case where the stationary imbalance and the imbalance created by the liquid are equal to the magnitude of $e_0=e$, then φ_C will be approximately equal to δ (Fig. 13). And so the angle between the deflection and the stationary imbalance will be directed to 180 °.

With a further increase in angular velocity, a balanced system is rotated without changing the relative position of the liquid and stationary imbalance.

With a small imbalance created by the liquid $e<<e_0$, the total imbalance of the system will mainly be determined by the magnitude and direction of the stationary imbalance and will not change significantly with the increase of the angle δ (Fig. 13). Consequently, the size of the deflection of the shaft, due to the total imbalance of the rotor, will not be significantly reduced.

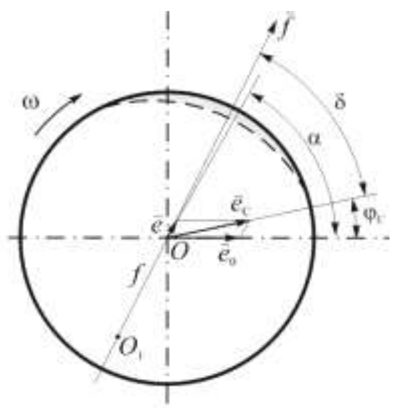


Fig. 12. Vector model of autobalancing process (continued)

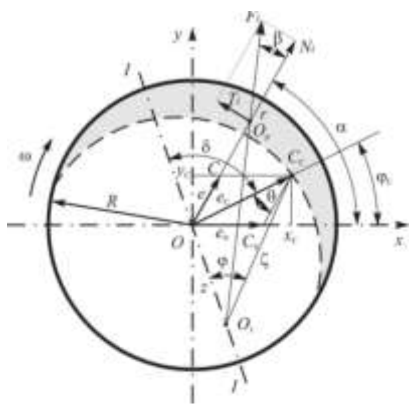


Fig. 13. Forces acting in the SBD in the presence of resistance

The basic idea is that at rotation velocity of the system $\omega < \omega_{rot}$ external resistance (friction forces in bearings, friction forces of cylinder against the air etc.) conditions bend plane (I-I) delay against total imbalance plane (OC_c) for the phase angle δ (Fig. 13) and appearing in this condition tangential force T of centrifugal inertial force F, and specific properties of liquid (fluidity, ability of any liquid volume to freely change its shape being forced by irrespectively low force, viscosity forces are considered only for quite fast motion when liquid displacements change quite fast) enable drawing liquid in ABU chamber to position that reduces the total imbalance of the system even at pre-critical rotation velocity. Analysis of geometric model (Fig. 13) shows that when external resistance is sufficient liquid that is in stable balance can balance rotor at pre-critical angular velocity. It will be shown analytically.

Total rotor eccentricity with liquid is defined as:

$$e_c = \sqrt{x_c^{\ 2} + y_c^{\ 2}} = e\sqrt{1 + 2k\cos\alpha + k^2}.$$

where $k = D_0 D^{-1}$ – relation of rotor and liquid imbalance.

Analysis of geometric model (fig. 15) shows that balance condition that doesn't take into account tension forces is missing tangential force:
either tg β=0, or

$$f\left[tg\alpha\cos\left(\delta + \phi_c\right) - \sin\left(\delta + \phi_c\right)\right] = 0.$$

The last equation is decomposed into two conditions (zero index corresponds to angles for liquid balance ($\beta = 0$)):

$$f = 0; \qquad (2\text{ a})$$

$$tg\alpha_0\cos\left(\delta + \phi_{0c}\right) - \sin\left(\delta + \phi_{0c}\right) = 0. \qquad (2\text{ b})$$

Condition (2 a) corresponds to no deflection condition that contradicts condition of the problem of elastically strained rotor. If liquid is in balance

$$tg\alpha_0 = tg\left(\delta + \phi_{0c}\right)$$

what comes from condition (2 b).

Taking into account values of e, e/e_0, e_c and $\theta = \alpha - \varphi_c$

$$\sin\alpha_0 = k^{-1}\sin\delta\sqrt{1+2k\cos\alpha_0 + k^2}\,. \tag{3}$$

It appears from equation (3) that if external resistance is missing in the system ($\delta = 0$, $k \neq 0$) liquid balance coincides by the angle with rotor imbalance ($\alpha = 0$), what is approved by the conclusion that liquid increases imbalance in system without external damping.

The expression for angle α_0 corresponding liquid balance ($\beta = 0$) is obtained through algebraic transformations of expression (3) when $\delta \neq 0$, $k \neq 0$:

$$\alpha_0 = \pi - \arccos\left(\sin^2\delta + \sqrt{\left(\sin^2\delta - 1\right)\cdot\left(\sin^2\delta - k^2\right)}\right). \tag{4}$$

Analyzing expression (4) gives positive arccosine argument and that's why even in pre-critical frequency of the rotor when $\delta \in \left(0, \frac{\pi}{2}\right)$ angle $\alpha_0 \in \left(\frac{\pi}{2}, \pi\right)$.

Effectiveness of balancing is characterized by relation of misalignment with rotational axis of mass center of the system with liquid and without it λ_0:

$$\lambda_0 = \frac{\sqrt{1+2k\cos\alpha_0 + k^2}}{k}; \tag{5}$$

if $1+2k\cos\alpha_0+k^2 = 0$ balance will be complete, therefrom the only solution will be $k = 1$ if $\cos\alpha_0 = -1$, and if $\alpha_0 = \pi$. Hence, the greatest balancing effect will be reached when imbalance of liquid that is involved into balancing is close to initial imbalance of the rotor ($k \approx 1$). Liquid which takes part in balancing makes concentric circles what has no influence on total imbalance of the system.

Expressions (3) and (5) contain relations of angular velocity to critical one, external resistance factor, relation of initial imbalance to liquid imbalance and relative dimensions of ABU. Obtained analytical relationships between these parameters allow solving applied problems to select optimal parameters of liquid ABU for particular rotor system, find effect of varying parameters of the system rotor – ABD – liquid on effectiveness of balancing. Theoretical results were proved by experimental researches.

Created plant allowed to make experimental researches of self-balancing of the rotor with horizontal rotation axis and to observe fluid behavior in ABD. Research of self-balancing process and fluid behavior in ABD were made on described plant with using next special devices: inductive sensor, marker of turns; - ADC (analog-to-digital converter); - Notebook computer.

By filling in each camera of ABD with 10 ml of water, without changing position or mass of imbalance, the plant was turned on again. Rotor vibrations were recorded on computer in text file in conditions of smooth rotor acceleration from 0 to 2000 rpm. Simultaneously were made video record of fluid behavior in ABD. FRF of rotor when passing through resonance are illustrated on Fig. 14.

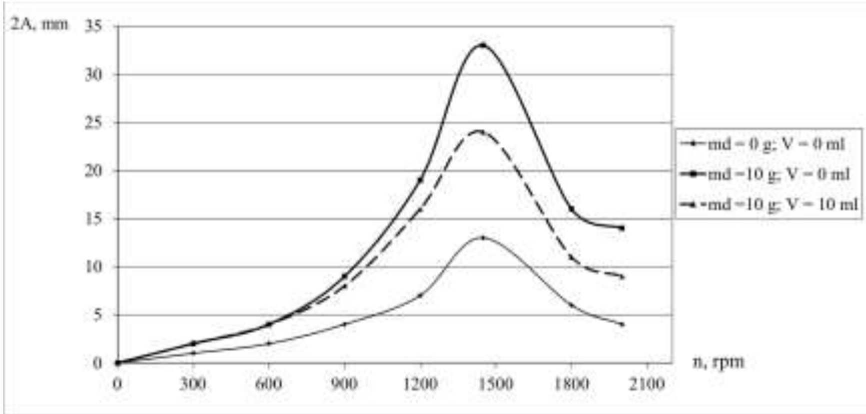


Fig. 14. Frequency response function

7. Conclusions

Theoretically, the basic requirements for the layout of washing machines were obtained and experimentally confirmed. It is proposed to implement it using liquid automatic balancing of the rotor.

The results of experimental and theoretical research of fluid behavior in passive self-balancing devices, which are installed in rotors with horizontal rotation axes when passing the resonance, are given in this article. Received relations showed that automatic balancing by fluid is effective for elastically deformed rotors or (and) rotor on elastic supports, where exists the difference in phases between the direction of centrifugal force and flexure (or movement) of the rotor. As experimental results showed, this difference in phases occurs when rotor reaches resonance speed and increases up to 180º when passing the resonance. In this case the tangential force occurs, under the influence of which the fluid moves to the side of flexure, opposite to imbalance, and further equilibration of the rotor already at resonance rotation frequency.

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