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Study of torsion oscillations of pumping unit shafts

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Abstract. The method of detuning from resonant frequencies is an effective method of reducing the vibration activity of the HA at the design stage, which allows selecting the necessary operating mode for the HA, which will provide the necessary reliability of operation. In this connection, works in the field of increasing the vibration protection of NA are relevant. All calculations are based on the dynamic model of free torsional vibrations of the pumping unit, for which the oscillation equation was derived on the basis of Lagrange equations of the second kind, where the angle of rotation of the rotor was taken as the generalized coordinate. Next, expressions were obtained for the determination of the potential and kinetic energies that take into account all the characteristics of the shaft and the disk of the rotor HA; in particular, for the first time the mass of the shaft is taken into account. Then an expression was obtained for determining the frequencies of free torsional oscillations and the equation of motion of the rotor HA. Knowing the parameters of the oscillatory motion makes it possible to estimate the stresses that arise when the shaft rotates. The following is a case of a sudden stop of the shaft, for example, when the support is destroyed, and a graph is obtained from which the main conclusions are drawn.

1. Introduction

In modern conditions of operation and management of oil and gas transportation systems, the relevance of control of the technological process is of great importance. One of the main parameters of technological processes in the oil and gas industry, which requires a high level of control, is frequency of oscillations [1-5].

As operating conditions become more severe, the intensity and frequency of vibrations increase; pressure pulsations have a negative effect leading to the breakdown of pumping units and, as a consequence, disruption of the technological process [6-8].

A method of vibration protection of pumping units (PU) by means of tuning from resonant frequencies implies knowledge of natural oscillation frequencies of PU and frequencies of the disturbing force.

2. Materials and methods

In [1], free torsional vibrations of a cylindrical shaft with *n* disks fixed to it are determined, neglecting the mass of the shaft as compared to the mass of the disk.

Below, a procedure for determining natural frequencies of torsional oscillations of PU is presented, obtained with the help of the Lagrange equation of the second kind taking into account the mass of the shaft [9, 10].

Fig. 1 shows the design scheme of the construction.

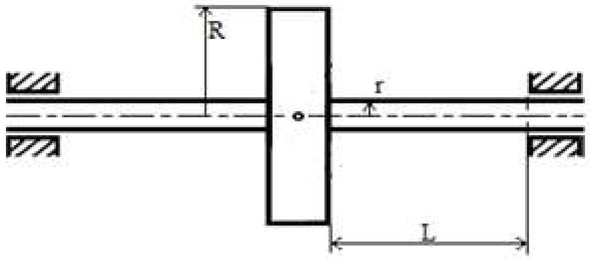

Figure 1. System - shaft and rigidly fixed disk.

The angle of rotation of the shaft - φ is taken as the generalized coordinate.

The Lagrange equation of the second kind for natural torsional oscillations of PU has the form:

$$\frac{\mathrm{d}}{\mathrm{dt}}\left(\frac{\partial \mathrm{T}}{\partial \dot{\varphi}}\right)+\frac{\partial \Pi}{\partial \varphi}=\frac{\partial \mathrm{T}}{\partial \varphi} \tag{1}$$

where t – time, s; T – kinetic energy; Π – potential energy.

The expression for the kinetic energy of the system - shaft and rigidly fixed disk, will be combined from the kinetic energy of rotation of the shaft and the disk:

$$T=\frac{\left(M(R^2+r^2)\quad+mr^2\right)}{4}\dot{\varphi}^2 \tag{2}$$

where M – disc weight, kg; R – outer radius of disk, m; r – shaft radius, m; m – shaft weight, kg.

The potential energy of the system is determined as:

$$\Pi=\frac{GI_p}{L}\varphi^2 \tag{3}$$

where G – modulus of elasticity of the second kind, Pa; I_p – polar moment of inertia, m^4; L – distance from bearing to disc, m.

Substituting the expressions for the partial derivatives in the La Grange equation, let us obtain a second-order differential equation with constant coefficients:

$$\frac{\left(M(R^2+r^2)+mr^2\right)}{2}\ddot{\varphi}+\frac{GI_p}{2L}\varphi=0. \tag{4}$$
$$\ddot{\varphi}+k^2\varphi=0$$

This equation will be a mathematical model of this object.

The value of the frequencies of natural torsional oscillations:

$$k=\sqrt{\frac{GI_p}{(M(R^2+r^2)+mr^2)L}}. \tag{5}$$

Thus, an expression was obtained for determining the frequency of natural torsional vibrations, allowing taking into account the characteristics of the shaft and the disk.

Knowing the parameters of the oscillatory motion makes it possible to estimate the stresses that arise when the shaft rotates.

The tangential stresses arising in the cross sections of the shaft are equal:

$$\tau=\frac{M_k}{W_p}\leq[\tau], \tag{6}$$

where M_k – torque in section; W_p – polar moment of circular section resistance.

Let us imagine the case of a sudden stop of the shaft, for example, when the support is broken. The solution of (4) is sought in the form:

$$\varphi = A\sin(kt+\alpha), \tag{7}$$

whence:

$$\dot{\varphi} = kA\cos(kt+\alpha). \tag{8}$$

The initial conditions for this problem can be formulated in the form: $\varphi_0 = 0$, $\dot{\varphi}_0 = \omega$. Substituting it into the solution, one obtains:

$$\dot{\varphi} = kA\cos(kt+\alpha), \tag{9}$$
$$0 = A\sin(\alpha).$$

From here $= 0$, $\omega = kA$, $\varphi = \frac{\omega}{k}\sin(\alpha)$. The maximum value will be reached when $\sin(\alpha) = 1$ whence $\varphi = \frac{\omega}{k}$.

The angle of rotation of a shaft of length L will be:

$$\varphi = \frac{M_k L}{G I_p}, \tag{10}$$

whence:

$$M_k = \frac{G I_p \varphi}{L}, \tag{11}$$

Then the tangential stresses will be:

$$\tau = \frac{G I_p \varphi}{W_p L}. \tag{12}$$

For a rod of circular cross section, the polar moment of section resistance and the polar moment of inertia can be determined from the formulas:

$$W_p = \frac{\pi d^3}{16}, I_p = \frac{\pi d^4}{32}. \tag{13}$$

Then the tangential stresses will be:

$$\tau = \frac{G d\omega}{2kL}. \tag{14}$$

3. Results and Discussion

Fig.2 shows the dependence of the tangential stresses on the angular velocity.

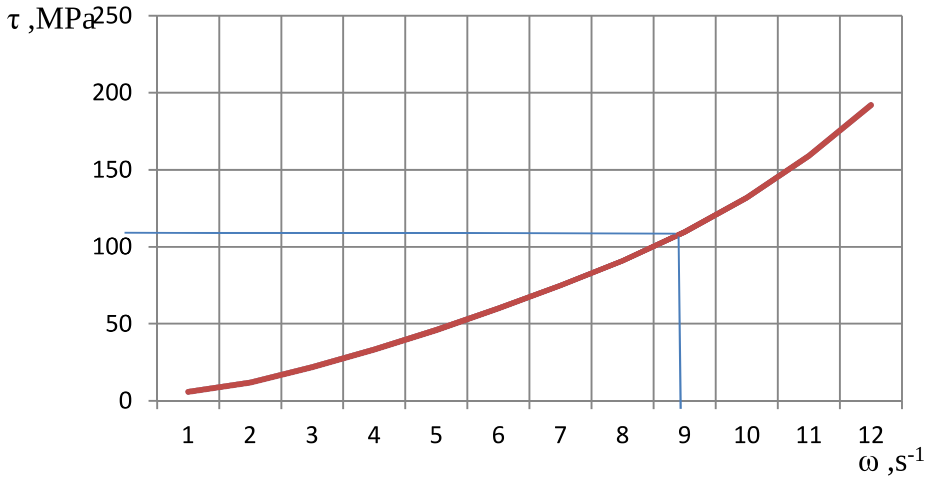

Figure 2. Dependence of tangential stresses on angular velocity.

It can be seen from the graph that even at a speed of about 9-10 rad/s, stresses will exceed the permissible values. Therefore, to prevent accidents, special shock-absorbing devices should be provided.

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Thus, to improve the efficiency of the equipment, the resulting expressions can be used to develop instructions for the design and operation of PU.

4. Conclusion

As a result, a mathematical model based on the Lagrange equations of the second kind was developed for free torsional oscillations of the rotor HA, allowing taking into account the characteristics of the shaft and rotor disk HA. The stresses resulting from the rotation of the shaft are evaluated, and the conditions for the emergence of an emergency situation for a specific task are determined.

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