THE INVESTIGATION OF RADIAL OSCILLATIONS OF THE CENTRIFUGAL PUMP ROTOR IN ANNULAR SEALS

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The centrifugal pumps and compressors are obtained a widespread application in modern industry. To a considerable degree, its technical development is defined by vibroacoustic parameters that depend on the vibrational state of the rotor.

In the capacity of the object of investigation of radial oscillations of the rotor in annular seals are chosen a diagram of a single-disk rotor with disk, which is situated between rigid bearings (Fig.1). This diagram is a model of the rotor of single-stage pump with a double-entry impeller.

The disk rotates on a deflection axis of a shaft with constant frequency and fluctuates with small radial and angular oscillations. The analysis of such system with four degrees of freedom represents essential mathematical difficulties. That is why it makes a sense to research easier partial systems performing only radial or only angular oscillations. Present paper is devoted to radial oscillations.

As is generally known, joint radial-angular oscillations occur in real rotors. Although the constructions of rotors, that perform mainly one kind of the oscillations, exist. Particularly, the rotor with localized mass (Fig. 2 a) and symmetric, dynamically stable single-disk rotor (Fig. 2 b) have just two degrees of freedom.

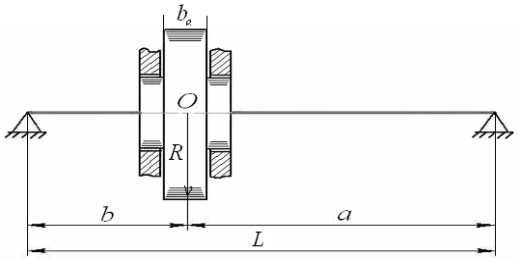

Figure1 – The diagram of single-disk rotor in annular seals with the disk between bearings

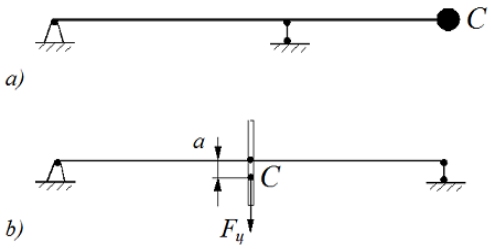

Figure 2 – The models of rotors with only radial oscillations

Methodological and practical interests are exposed by analysis of radial oscillations. As far as it allows to investigate more significant conformity of real rotors motion, assess the impact of hydrodynamic characteristics of annular seals on natural frequencies and critical frequencies, on its stability and amplitudes of forced oscillations.

The equation of free radial oscillations taking into account hydrodynamic forces and moments represents the system of fourth order with real variables.

$$a_1\ddot{u}_x + a_2\dot{u}_x + a_3u_x + a_4\dot{u}_y + a_5u_y = \omega^2 a_x^*$$
$$a_1\ddot{u}_y + a_2\dot{u}_y + a_3u_y - a_4\dot{u}_x - a_5u_x = \omega^2 a_y^* .$$

It should be noted that two variants of problems exist when natural frequencies are determined. The first case is the constant pressure drop in the seal and the second case appears when the pressure drop in the seals is proportional to the square of the rotating frequency.

The determination of critical frequency of the rotation is connected with the resonance that is sharp increase of steady-state forced oscillations amplitude when frequency ω_f of external action on the rotor is approaching S_i any of its natural frequencies. Fig. 3 shows one of the determination methods of critical frequencies of rotation according to frequency diagram.

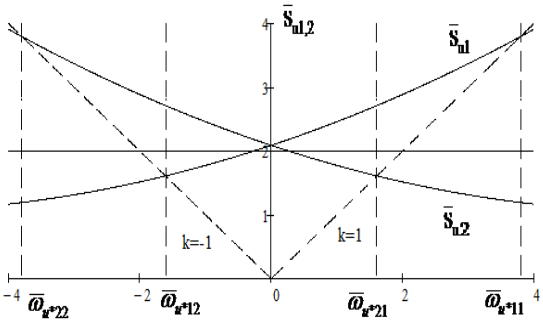

Figure 3 –The determination of critical frequencies of rotation according to frequency diagram

Amplitude-frequency and phase-frequency characteristics were formed for testing model of the rotor for three values of conicity parameter. Amplitude-frequency characteristics have a resonance peak on conditions that match the conditions of critical velocities existence. In other case it increases monotonously.

The investigation in present paper was conducted in Mathcad 14, which is considered to be computer software for engineering calculations. It should be mentioned that all calculations were made analytically.

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