Research of stability of rotor of the knife grinding refiners

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Research of stability of rotor of the knife grinding refiners

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Abstract. An object of research is stability of provision of rotor in the knife grinding refiners. Dynamic and mathematical models of rotor in bearing support are developed. Expression for determination of force of a preliminary tightness in bearing support for stabilization of the situation of rotor is received. Amplitude of fluctuations of disk of rotor in bearings with tightness is much less than in bearings with gap. For ensuring stability of an interknife gap of refiners it is recommended to eliminate radial gaps in bearing knots, i.e. to use bearings with a preliminary tightness. It will allow to raise a technical resource of the grinding plate. The technique of researches can be applied in other high-precisionmachines, for example, in the metalworking equipment.

1. Introduction

The knife grinding refiners - the capital processing equipment for grind of fibrous materials in pulp and paper industry [1-4]. The inter knife gap in these refiners makes shares of millimeter [5,6]. Refiner rotor – high-precision knot which has to provide a stable inter knife gap. Now in rotor knot of refiners bearings with gap are used [7].

Article purpose - research of stability of provision of rotor in bearing support of refiner.

2. Dynamic and mathematical models of rotor

The following assumptions are accepted: research is conducted in linear statement; dispersion of fluctuations in rotor design is not considered; rotor of refiner is accepted by rigid; main source of fluctuations is the imbalance of rotor both mechanical and hydrodynamic influence of the ground material. The refiner rotor the mass of m_r and eccentricity of e rotates with constant speed ω (figure 1). Bearings of rotor are modeled by spring c_p and damper of b_p. In rotor model the preliminary tightness in bearings is modeled by springs with coefficient of rigidity of c_s. In bearings with gap of spring of c_s are absent. Radial gap in bearings we will designate s.

The rotor is affected by stabilization forces of F_1 and F_2, normal force of contact in the bearing F_N support, F_T friction force, $m_r g$ gravity, and forces in the bearing Fx and Fz support (figure 2).
Equation of the movement of rotor of refiner

$$m_r \ddot{x} = m_r e \omega^2 \cos \omega t - F_x$$

$$m_r \ddot{z} = m_r e \omega^2 \sin \omega t - F_z - m_r g.$$

Condition of emergence of the mode of pendular fluctuations of rotor of refiner in bearing support it is possible to write down as $F_N > 0$ and $|F_T| = \mu F_N$. Condition of the situation of stability of rotor and lack of pendular fluctuations $F_N > 0$ and $|F_T| < \mu F_N$, where μ - friction coefficient.
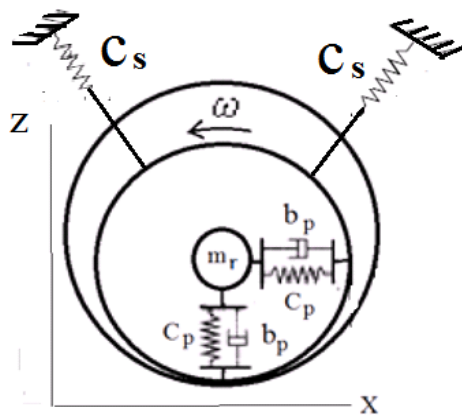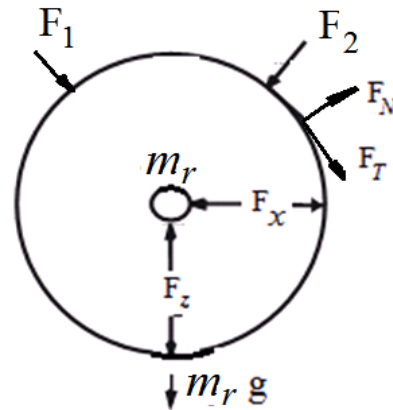
Figure 1. Dynamic model of rotor.

Figure 2. Forces operating on refiner rotor.

3. Results

As result of research of mathematical model expression for determination of force of preliminary tightness in bearing support for stabilization of the situation of rotor and lack of pendular fluctuations is received

$$F_c \geq [(B_3 \sin \widehat{\omega}\hat{t} + B_4 \cos \widehat{\omega}\,\hat{t} - \hat{g})/2 \cos \beta]c_p e,$$

$$B_3 = 2\xi\widehat{\omega}\left(A_1 \mp \left(\frac{1}{\mu}\right)A_2\right) + A_2 \pm \left(\frac{1}{\mu}\right)A_1, \quad B_4 = 2\xi\widehat{\omega}(A_2 \pm \left(\frac{1}{\mu}\right)A_1) - A_1 \pm \left(\frac{1}{\mu}\right)A_2,$$

$$A_1 = \frac{2\xi\widehat{\omega}^3}{\left[\left(1-\widehat{\omega}^2\right)^2 + (2\xi\widehat{\omega})^2\right]}, A_2 = \frac{\left(1-\widehat{\omega}^2\right)\widehat{\omega}^2}{\left(1-\widehat{\omega}^2\right)^2 + (2\xi\widehat{\omega})^2}, \qquad \hat{g} = \frac{gm_r}{ec_p}$$

$$\hat{t} = t\sqrt{\frac{c_p}{m_r}}, \quad \widehat{\omega} = \frac{\omega}{\sqrt{\frac{c_p}{m_r}}}, \quad \xi = \frac{b_p}{2\sqrt{m_r c_p}}.$$

The dependence of force of preliminary tightness in bearings from the eccentricity of rotor of refiner TF-52 is shown in the figure 3. At a nominal eccentricity of rotor of 40 microns force of preliminary tightness will be about 140 kN. It is necessary to explain what eccentricity rotor of refiner is not constant and changes at operation because of uneven wear of plate. Ektsintrisitet of rotor can increase at operation up to 2.5 times in comparison with nominal [8]. Therefore force of preliminary tightness in bearings for refiner TF-52 it is necessary to accept equal or bigger than 350 kN.

Amplitude of fluctuations of rotor disk of refiner TF-52 by technique [9] in bearings with gap and tightness (figure 4) is calculated. Amplitude of fluctuations of disk depends on radial gap of the bearing. So, at nominal radial gap of 0.5 mm amplitude of fluctuations of disk is equal in bearings and eccentricity of rotor of 40 microns to 0.75 mm. This amplitude is comparable to an operational interknife gap of refiner TF-52. In design of rotor knot of refiner the possibility of metal contact of rotor and stator at grind of semi-finished products is put. It results in intensive wear of plate and reduces its technical resource [10]. For ensuring stability of provision of rotor and, respectively, the interknife gap is recommended to eliminate radial gaps in bearing knots, i.e. to use bearings with preliminary tightness.
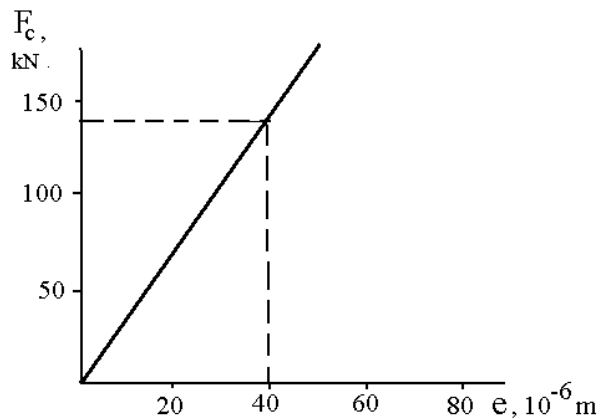
Figure 3. Dependence of force of preliminary tightness in bearings from rotor eccentricity refinerTF-52.
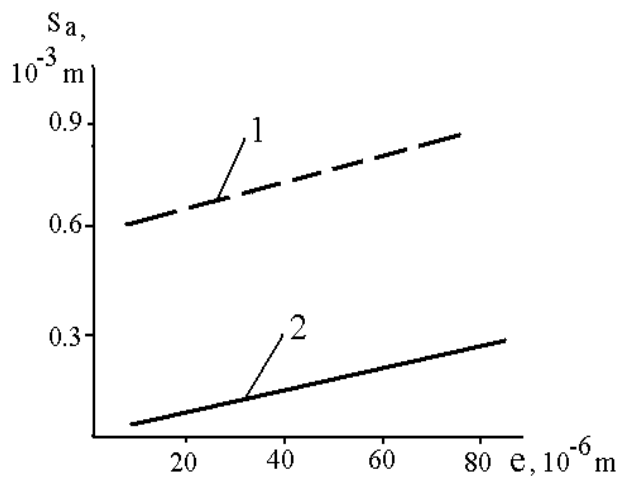

Figure 4. Dependence of amplitude of fluctuations of rotor disk from from rotor eccentricity refinerTF-52:
 1- bearings with gap;
2 – bearings with preliminary tightness.

When using bearings with tightness in rotor hub ofrefiner TF-52 and eccentricity of rotor of 40 microns amplitude of fluctuations of disk of only 0.18 mm. Amplitude of fluctuations of disk in rotor knot with bearings with tightness decreases more than by 4 times in comparison with bearings with gap. Fluctuations of disk of rotor in bearing happen to tightness only due to elastic deformations of elements of system.

4. Discussion

It is necessary to consider that the power of the drive of modern refiners reaches tens of megawatts and at operation, especially at start-up, there are big currents. When using bearings of rotor with tightness these currents will be even more. Therefore it is recommended to design bearing support so that start-up of refiner happened at bearings to gap, and operation – at bearings with tightness.

Other feature of bearing support with tightness is the increased thermal emission. At design of such support it is necessary to provide intensive heat removal from bearings. Also use of system by the managing director which regulates force of preliminary tightness of bearings depending on change of eccentricity of rotor of refiner is recommended.

5. Conclusion

Dynamic and mathematical models of rotor of refiner in bearings with gap and preliminary tightness are developed. The method of calculation of force of preliminary tightness in bearing support is developed for stabilization of the situation of rotor.

Amplitude of fluctuations of disk in rotor knot with bearings with tightness considerably decreases in comparison with bearings with gap. For ensuring stability of provision of rotor and, respectively, stability of an inter knife gap is recommended to eliminate radial gaps in bearing knots, i.e. to use bearings with preliminary tightness. It will allow to raise technical resource of the grinding plate.

However, at use of bearings in rotor knot it is necessary to solve the following problems with tightness:

- the raised electric motor current, especially at its start-up;
- the increased thermal emission in bearing support.

Use of system by the managing director which regulates force of preliminary tightness of bearings depending on change of eccentricity of rotor of refiner is recommended.

The technique of researches can be applied in other high-precision machines, for example, in the metalworking equipment.

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