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Predicted resource of the most loaded centrifugal pump bearing

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Abstract: The paper gives a derivation of the formula for calculating the most loaded bearing of a centrifugal pump. The structure, applications, the main types of this centrifugal pump and the comparison of these bearings with hydrostatic are described. The resulting model is well applicable to pumps running on clean, chemically inactive liquids.

Introduction

Currently, centrifugal pumps [1–12] are widely used in various fields of industry and national economy, such as agriculture. The choice of this particular type of pump is due to the fact that it has a simple design, is easy to operate, and can use water, fuel and chemically hazardous compounds as a working fluid.

Unlike hydrostatic bearings, which have a significant resource, the life of pumps with ball bearings is very limited. The life of a ball bearing is largely dependent on the choice of rotational speed of the pump rotor.

In this article, an attempt was made using the method of mathematical analysis to derive a formula for calculating the predicted resource of the most loaded bearing.

Methods

The method is based on solving the equation for an incompressible fluid (ρ=const)

We determine the effect of the shaft speed on the pump resource. To do this, we first determine the value of the equivalent radial force acting on the most loaded bearing. The design diagram of the radial and axial forces acting on the rotor is shown in the figure.

An impeller and two bearings are mounted on the motor shaft. Impeller creates radial force Pr_1, as a result, a radial reaction occurs on the first bearing Pr_2, and on the second bearing, a radial reaction Pr_3. Axial force occurs on the front of the impeller P_{a2}, and axial force on the back of the impeller P_{a1}. The rotor also affects its weight. P_{a4}. The arrangement of the bearings is such that the total axial force from the impeller and the weight of the rotor is compensated by the axial reaction in the second bearing P_{a3}.

From the equations of the equilibrium condition (the sum of the acting moments and forces on the structure is zero $M_o = 0 \sum_i P_i = 0$ we get the following system of equations:

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$$P_{r1} \cdot (L_2 + L_1) = P_{r2} \cdot L_2 \quad (1)$$

$$P_{r1} + P_{r3} = P_{r2} \quad (2)$$

$$P_{a4} + P_{a1} - P_{a2} = P_{a3} \quad (3)$$

$$P_{r2} = P_{r1} \cdot \frac{(L_2 + L_1)}{L_2} \quad (4)$$

$$P_{r3} = P_{r1} \cdot \frac{L_1}{L_2} \quad (5)$$

$$P_{a3} = P_{a4} + P_{a1} - P_{a2} \quad (6)$$

whereL_2 is the distance between the bearings,
L_1 is the distance between the front bearing and the impeller (see figure).

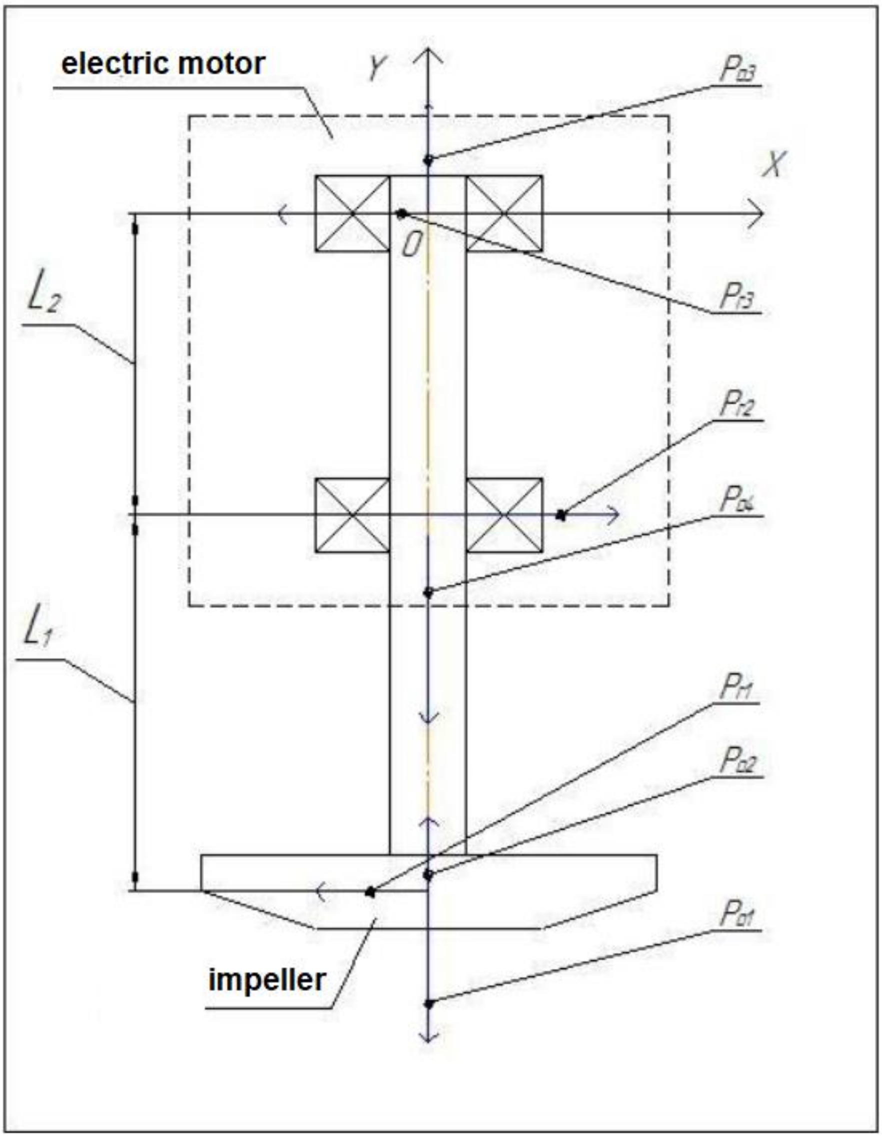

Figure. Design diagram of radial and axial forces, rotor acting

These values are known to us from the design of the electric motor.

Dynamic and static reactions in bearings were not included in the system of equations (4)–(6) due to their smallness for a given pump, while eliminating them from this system allows us to significantly simplify the resulting mathematical model. In order to evaluate their contribution, the magnitude of the static and dynamic reactions can be compared, for example, with the weight of the rotor, which will be calculated below.

Thus, to determine the reaction forces in the bearings, it is necessary to determine the weight of the rotor and the dependence of the axial and radial forces on the shaft rotation frequency. The weight of the rotor is equal to:

$$P_{a4} = M_p \cdot g \quad (7)$$

whereM_r is the mass of the rotor.

The total axial force created by the impeller is:

$$P_{a1} - P_{a2} = \pi\rho g\left(R_s^2 - r_s^2\right)H - $$
$$ - \frac{\pi^3 \cdot \rho \cdot n^2}{8 \cdot 30^2}\left(R_y^2 - r_s^2\right)\left[R_2^2 - 0{,}5\cdot\left(R_s^2 - r_s^2\right)\right] \quad (8)$$

where ρ is the density of the working fluid,

r_s is the radius of the sleeve,

andR_s is the radius of the gap seal. In our case, the design of the impeller is such that $r_{вт}$= 0, $R_s \approx R_1$. As will be shown below, the optimal value of R_1 will be equal to:

$$R_1 = 1{,}125 \cdot \sqrt[3]{\frac{Q \cdot 30}{\pi^2 \cdot n}} \quad (9)$$

Thus, the formula (8) will take the form:

$$P_{a1} - P_{a2} = \pi\rho g R_1^2 H - \frac{\pi^3 \cdot \rho \cdot n^2}{8 \cdot 30^2} R_1^2\left[R_2^2 - 0{,}5\cdot R_1^2\right] \quad (10)$$

The radial force for spiral bends is:

$$P_{r1} = 0{,}4\left(1 - \frac{Q}{Q_{opt}}\right)\rho \cdot g \cdot H \cdot D_2 \cdot b_2 \quad (11)$$

whereQ_{opt} is optimum pump flow.

As will be shown below, the optimal value of the impeller width at the output is:

$$b_2 = \frac{\pi \cdot n \cdot 0{,}044 \cdot \left(\frac{Q}{n}\right)^{\frac{2}{3}}}{\sqrt{\frac{H \cdot g}{y \cdot \eta_г}}} \quad (12)$$

Thus, in the worst case, the magnitude of the radial force is determined by:

$$P_{r1} = 0{,}4 \cdot \rho \cdot g \cdot H \cdot 2 \cdot R_2 \cdot \frac{\pi \cdot n \cdot 0{,}044 \cdot \left(\frac{Q}{n}\right)^{\frac{2}{3}}}{\sqrt{\frac{H \cdot g}{y \cdot \eta_г}}} \quad (13)$$

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The resulting P_{r1} value corresponds to a zero pump flow.

Substituting formulas (7), (8) and (13) into the system of equations (4)–(6), we obtain the reaction values in the rotor bearings. After calculating the axial and radial reaction forces in the bearings of both bearings, we determine the radial forces equivalent to the reaction forces obtained. The worst situation will be observed in the second bearing, as both radial and axial forces act on it. Equivalent radial force is defined as:

$$P_{eq} = \left(V \cdot X \cdot P_{r3} + Y \cdot P_{a3}\right) \cdot K_b \cdot K_t \quad (14)$$

where$V$, X, K_b and K_t are coefficients depending on various conditions, Y is a functional dependence characterizing the fraction of axial forces in the equivalent radial force (see below).

We take the specified coefficients according to the given working conditions equal to the following values: $V = 1$; $K_t = 1$; $K_b = 1{,}2$; $X = 0.56$ and the function Y is calculated by the formula:

$$Y = \frac{0{,}44}{e} \quad (15)$$

where the coefficient e is :

$$e = 0{,}28 \cdot \left(\frac{f_0 \cdot P_{o3}}{C_{or}}\right)^{0{,}23} \quad (16)$$

where f_0 = 15, (determined depending on the design parameters of the bearing,

C_{or} —basic static radial bearing capacity.

Next, we determine the value of the resource of this bearing for various pump operation modes.

The estimated bearing life is determined by the formula:

$$T = aa_1 \cdot aa_{23} \cdot \left(\frac{C}{P_{eq}}\right)^k \cdot \frac{10^6}{60 \cdot n} \quad (17)$$

where aa_1, aa_{23} are coefficients depending on the special operating conditions of the bearing and the probability of failure-free operation.

The coefficient k in formula (17) depends on the type of bearing. In this case, it is a ball bearing and coefficient $k = 3$.

Substituting (4)–(16) in (17) we obtain the dependence of the resource on the frequency:

$$T = 0{,}7 \cdot \left(\frac{C}{\left(0{,}56 \cdot P_{r3}(n) + \dfrac{0{,}44}{e(n)} \cdot P_{a3}(n)\right) \cdot 1{,}2}\right)^3 \cdot \frac{10^6}{60 \cdot n} \quad (18)$$

where C is the basic static radial load bearing capacity,

P_{r3} is the radial reaction on the second bearing,

P_{a3} is the axial force on the rear wheel of the impeller.

Results

Thus, the obtained technique allows us to obtain a working formula for determining the predicted resource of the most loaded bearing of a horizontal centrifugal pump.

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Conclusion

The obtained method allows us to predict the life of a centrifugal pump under the assumption that its operating mode is constant. The disadvantages of the obtained method include the assumption that failure is possible only in the most loaded bearing, which is not entirely true, however, if the pump is operated on a clean, uncontaminated and chemically inactive liquid, the described model can be considered suitable for use in the calculation of centrifugal pumps.

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