

Predicted resource of the most loaded bearing of a horizontal centrifugal pump

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Abstract: The paper gives a derivation of the formula for calculating the predicted resource of the most loaded bearing of a horizontal centrifugal pump. A model of a horizontal centrifugal pump, as well as the action of axial forces and radial reactions, are considered.

Introduction

Currently, centrifugal pumps [1–9] 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.

Consider the structure of the horizontal centrifugal pump in Figure Figure 1. On the shaft of the electric motor 1, the impeller 2 and two bearings 3. The working fluid is captured by the blades of the wheel 2, is thrown to the walls of the chamber, creating excess pressure. Water is pushed out under pressure. During the discharge of liquid from the working chamber, a vacuum is created, which ensures the suction of a new portion of the liquid.

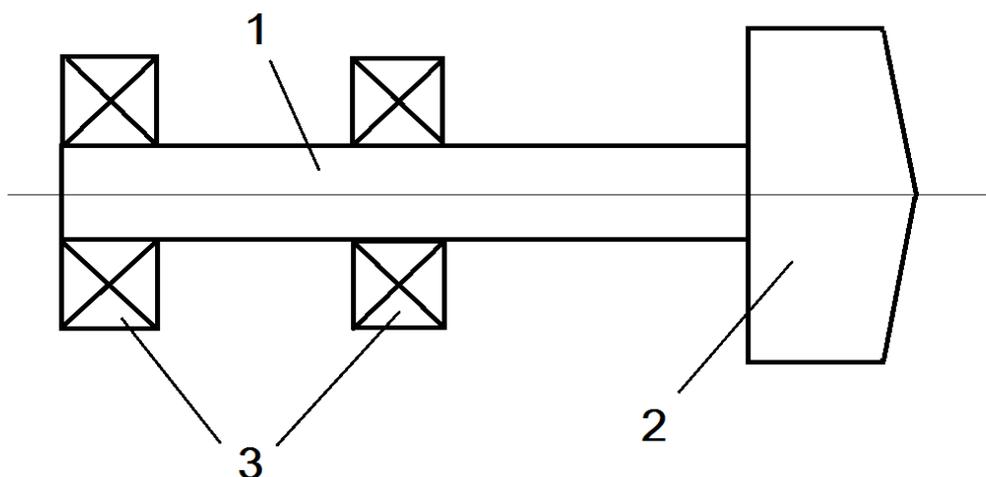


Figure 1. The structure of a horizontal centrifugal pump

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

Consider the figure of Fig. 2. The impeller 2 creates a radial force P_{r1} , as a result of which a radial reaction P_{r2} occurs on the first bearing, and P_{r3} on the second bearing. P_{o2} axial force occurs on the front impeller disk, and P_{o3} on the rear impeller disk. Also, its weight G acts on the rotor. The bearing arrangement 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_{o3} .

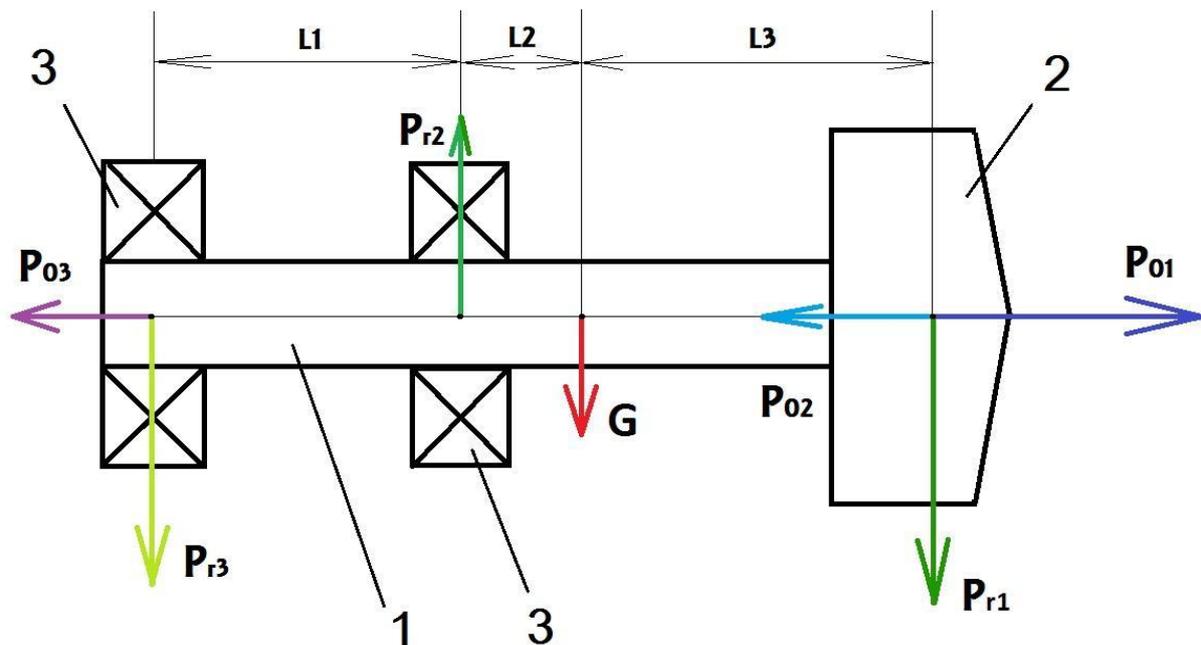


Figure 2. Reactions of a horizontal centrifugal pump

From the equations of the equilibrium condition (the sum of the acting moments and forces on the structure is zero $M_0 = 0$ $\sum_i P_i = 0$) we get the following system:

$$P_{a3} = P_{a1} - P_{a2} = P_{ak} \quad (1.1)$$

$$P_{r3} \cdot L_1 = G \cdot L_2 + (L_2 + L_3) \cdot P_{r1} \quad (1.2)$$

$$G = M_p \cdot g \quad (1.3)$$

where G is the weight of the rotor,

M_p is the mass of the rotor, L_1 is the distance between the bearings, L_2 is the distance between the first bearing and the center of mass of the shaft, L_3 is the distance between the shaft and the impeller (Figure 2).

The total axial force created by the impeller is:

$$P_a = \pi \rho g (R_y^2 - r_s^2) H - \frac{\pi^3 \cdot \rho \cdot n^2}{8 \cdot 30^2} (R_y^2 - r_s^2) [R_2^2 - 0,5 \cdot (R_g^2 - r_s^2)] \quad (1.4)$$

where ρ is the density of the working fluid,

r_s is the radius of the sleeve,

and R_g is the radius of the gap seal.

In our case, the design of the impeller is such that $r_{e1} = 0$, $R_g \approx R_l$. As will be shown below, the optimal value of R_l will be equal to:

$$R_l = 1,125 \cdot \sqrt[3]{\frac{Q \cdot 30}{\pi^2 \cdot n}} \quad (1.5)$$

Thus, the formula (1.4) takes the form:

$$P_a = \pi \rho g R_l^2 H - \frac{\pi^3 \cdot \rho \cdot n^2}{8 \cdot 30^2} R_l^2 [R_2^2 - 0,5 \cdot R_l^2] \quad (1.6)$$

The radial force for spiral bends is:

$$P_{r1} = 0,4 \left(1 - \frac{Q}{Q_{opt}} \right) \rho \cdot g \cdot H \cdot D_2 \cdot b_2 \quad (1.7)$$

where Q_{opt} is the optimum pump flow.

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

$$b_2 = \frac{\pi \cdot n \cdot 0,044 \cdot \left(\frac{Q}{n} \right)^{\frac{2}{3}}}{\sqrt{\frac{H \cdot g}{y \cdot \eta_e}}} \quad (1.8)$$

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

$$P_{r1} = 0,4 \cdot \rho \cdot g \cdot H \cdot 2 \cdot R_2 \cdot \frac{\pi \cdot n \cdot 0,044 \cdot \left(\frac{Q}{n} \right)^{\frac{2}{3}}}{\sqrt{\frac{H \cdot g}{y \cdot \eta_e}}} \quad (1.9)$$

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

Substituting formulas (1.3), (1.4) and (1.9) into the system of equations (1.1)–(1.2), 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:

$$P_{eq} = (V \cdot X \cdot P_{r3} + Y \cdot P_{o3}) \cdot K_{\sigma} \cdot K_m \quad (1.10)$$

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.

We take the indicated coefficients in accordance with [10] according to the given working conditions equal to the following values: $V = 1$; $K_t = 1$; $K_b = 1.2$; $X = 0,56$. Function Y is calculated by the formula:

$$Y = \frac{0,44}{e} \quad (1.11)$$

where the coefficient e is:

$$e = 0,28 \cdot \left(\frac{f_0 \cdot P_{ок}}{C_{or}} \right)^{0,23} \quad (1.12)$$

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

C_{or} is the 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:

$$T = a_1 \cdot a_{23} \cdot \left(\frac{C}{P_{r\text{экв}}} \right)^k \cdot \frac{10^6}{60 \cdot n} \quad (1.13)$$

where a_1 , a_{23} are coefficients depending on the special operating conditions of the bearing and the probability of failure-free operation.

The coefficient k in the formula (1.13) depends on the type of bearing. In this case, it is a ball bearing and coefficient $k = 3$.

Substituting (1.1)–(1.12) into (1.13), we obtain the dependence of the resource on the frequency

$$T = 0,7 \cdot \left(\frac{C}{\left(0,56 \cdot P_{r3}(n) + \frac{0,44}{e(n)} \cdot P_{ок}(n) \right) \cdot 1,2} \right)^3 \cdot \frac{10^6}{60 \cdot n} \quad (1.14)$$

Results

During the study, a working formula was obtained to determine the predicted resource of the most loaded bearing of a horizontal centrifugal pump. As a result, the formula has the form:

$$T = 0,7 \cdot \left(\frac{C}{\left(0,56 \cdot P_{r3}(n) + \frac{0,44}{e(n)} \cdot P_{o3}(n) \right) \cdot 1,2} \right)^3 \cdot \frac{10^6}{60 \cdot n}$$

where C is the basic static radial bearing capacity, P_{r3} - radial reaction on the second bearing, P_{o3} - axial force on the rear disk of the impeller, n is the rotation frequency.

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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