Investigation of the dependence of the axial force acting on the rotor of a vertical centrifugal pump on the area of the discharge holes of the impeller

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Abstract. The article describes a vertical centrifugal pump with a bearing Assembly in the upper part of the rotor. It is shown that for certain values of design parameters, the influence of the so-called discharge holes in the impellers can significantly affect the axial force acting on the rotor, and for certain parameters, for example, with a large weight of the rotor, this influence is insignificant and the manufacture of such impellers is impractical.

Introduction

The issues of modeling working processes in centrifugal pumps are widely illustrated in the literature [1]–[10]. Of particular interest are publications using modern modeling tools [11]–[15]. In a number of constructions of modern centrifugal pumps, discharge holes are used at the impellers.

These holes are used to reduce bearing loads. In this article, the task is to estimate the effect of the area of these holes on the total axial force acting on the rotor.

Methods

The most loaded bearing is the one that perceives the total axial force, because in vertical pumps it is usually maximum and radial ball bearings (see figure 1) do not accept it well. For determination this axial force, it is necessary to consider the rotor equilibrium.

The total axial response perceived by the most loaded bearing from the rotor equilibrium condition is:

\[
P_{\text{pad}} = G - P_{\text{a}} - P_{\text{D}} + P_{\text{o}}
\]

where \(G\) — the weight of the rotor, \(P_{\text{a}}\) — axial force arising from the rotation of the impeller, \(P_{\text{D}}\) — axial force arising from the rotation of the working fluid in the impeller, \(P_{\text{o}}\) — the force due to pressure in the upper part of the impeller from the discharge holes. Let's consider the last force in more detail.

In fig. 3: \(R_{\text{a1}}\) and \(R_{\text{a2}}\) are the Radius of the 1st and 2nd groove seals, \(R_{3}\) is the outer radius of the impeller, \(R_{\text{in}}\) is the radius of the entrance to the impeller, \(R_{\text{u}}\) is the radius of the upper bushing, \(\Delta\) is the radial clearance in the groove seals, it is assumed to be equal for both groove seals in this mathematical model, \(p_{\text{a}}\) is the pressure in front of the upper groove seal, \(\Delta p_{\text{a}}\) is the pressure drop on
the upper groove seal, $\Delta p_o$ is the pressure drop on the discharge hole. Take the pressure at the pump inlet as 0.

Fig. 1. Typical construction of a vertical centrifugal pump.
Then the Po force will be equal to:

$$ P_o = \Delta p_o \cdot (R_{u2}^2 - R_{u1}^2) \cdot \pi $$  \hspace{1cm} (2)

For determination $\Delta p_o$, consider the pressure change on the back of the impeller. The pressure before the groove seal can be found from theory of similarity of centrifugal pumps:

$$ p_{u2} = \rho \cdot g \cdot H_u \cdot \left( \frac{R_{u2}}{R_2} \right)^2 $$  \hspace{1cm} (3)

This pressure is divided into two components:

$$ p_{u2} = \Delta p_o + \Delta p_{u2} $$  \hspace{1cm} (4)

From Bernoulli's relation orifice discharge is equal to:

$$ Q_{u2} = \mu \cdot f_{u2} \cdot \sqrt{\frac{2 \cdot \Delta p_{u2}}{\rho}} $$  \hspace{1cm} (5)
where $f_{u2}$ is the area of the groove seal, and $\mu$ is the discharge coefficient.

From (5) we get:

$$\Delta p_{u2} = \left( \frac{Q_{u2}}{\mu \cdot f_{u2}} \right)^2 \cdot \frac{\rho}{2} \quad (6)$$

Similarly for the next section:

$$\Delta p_o = \left( \frac{Q_{o2}}{\mu \cdot f_o} \right)^2 \cdot \frac{\rho}{2} \quad (7)$$

where $f_o$ is the area of the discharge hole. This is the variable quantity for us in this study. Now we can consider the other power factors in equation (1). The weight of the rotor is calculated using the following formula:

$$G = m \cdot g \quad (8)$$

where $m$ is the rotary mass and $g$ is the acceleration of free falling.

Axial force arising from the rotation of the impeller [16]:

$$P_a = \pi \cdot (R_u^2 - R_{u2}^2) \cdot \rho \cdot g \cdot H_n - \frac{\pi \cdot \rho \cdot \omega^2}{8} \cdot \left[ R_{u1}^2 - R_{u2}^2 \right] \cdot \left[ R_2^2 - 0.5 \cdot \left( R_{u1}^2 + R_{u2}^2 \right) \right] \quad (9)$$

where $H_n$ is the pump head.

Axial force arising from the rotation of the working fluid in the impeller [16]:

$$P_D = \rho \cdot Q_a \cdot \left( \frac{Q_a}{\pi \cdot R_{a2}^2} \right) \quad (10)$$

where $Q_a$ is the pump flow. We introduce the following auxiliary designation.

$$A = G - P_o - P_D \quad (11)$$

$$B = (R_{a2}^2 - R_{a1}^2) \cdot \pi \quad (12)$$

$$D = \rho \cdot g \cdot H_n \cdot \left( \frac{R_{a2}}{R_2} \right)^2 \quad (13)$$

Then from the equations (3), (4), (6), (7) it turns out:

$$D = Q_{a2}^2 \cdot \left[ \frac{E + \frac{I}{f_{a2}^2}}{f_{a2}^2} \right] \quad (14)$$

From which:

$$Q_{a2}^2 = \frac{D}{\frac{E + \frac{I}{f_{a2}^2}}{f_{a2}^2}} \quad (15)$$

Substitute in (7):

$$\Delta p_o = \frac{D}{\frac{1 + f_{a2}^2}{f_{a2}^2}} \quad (16)$$
Substitute (16), (12) and (11) in (1) and get:

\[
\frac{2}{2} \cdot \frac{2}{1} = \begin{bmatrix}
A & B \\
C & D
\end{bmatrix}
\]

\[\text{pod} = u \cdot D \cdot B \cdot f \cdot (17)\]

where A, B, and D are constants if we analyze equations (11)-(15). Thus, the mathematical model is ready.

**Results**

If you substitute the numbers for a typical vertical pump in formula (17), you can get the following dependence of the axial force on the area of the discharge hole:

![Graph showing the dependence of the axial reaction of the most loaded bearing on the area of the discharge hole.](image)

**Fig. 4.** The dependence of the axial reaction of the most loaded bearing on the area of the discharge hole.

Therefore, the axial force depends on the area of the discharge hole.

However, if this model is substituted, for example, a centrifugal pump with a heavy rotor, the dependence does not look so obvious:
Fig. 5. The dependence of the axial reaction of the most loaded bearing on the area of the discharge hole in the case of a heavy rotor.

Conclusions
A mathematical model is obtained that makes it relatively easy to evaluate the feasibility of using impeller discharge holes for the case of vertical centrifugal pumps with ball bearings, two impeller groove seals that are not equal in diameter, and one discharge hole at a radius below the upper groove seal. For the convenience of using the obtained mathematical model, auxiliary constants (11)–(15) were introduced, the values of the design parameters in which did not change during the study. The formula (19) is obtained, which allows you to quickly estimate the effect of using the discharge hole on the value of the axial reaction in the bearing, which perceives the total axial force on the rotor.

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