Scientific bases of control of vibration of hydrodynamic origin in centrifugal pumps

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Abstract. The reliability and efficiency of pumps, like any machines and mechanisms, is determined by a number of factors, including their operating conditions and the ability to adapt to changing operating conditions. This article is devoted to the development of scientific foundations for controlling vibration of hydrodynamic origin in centrifugal pumps. The failure of a centrifugal pump due to fatigue failure of a rolling bearing does not occur suddenly and not instantly. First, some signs of the approach of this process appear, the nature of the vibroactivity of the aggregate changes. Against the background of a stable vibration level, some vibration outliers appear. The study of the sequence and intensity of these emissions is important information not only about the approach of the moment of destruction of the rolling bearing parts, but also about the operating time that the operating personnel still have to take measures to eliminate an emergency and complete catastrophic destruction.

1 Introduction

Reliability, durability and efficiency of pumps, pumping units, like any machines and mechanisms, are determined by a number of factors, including their operating conditions and the ability to adapt to changing operating conditions.

Among the factors that most adversely affect the reliability of pumping units is increased vibration. Centrifugal pump failures due to increased vibration account for 38-45% of all failures. Increased vibration affects the technical condition of mechanical seals and bearings. Failures of these units account for 42-53% of all pump failures.

Reliability and reliability in the operation of rolling bearings is the main link, which is part of a centrifugal pump unit. At the enterprises of the country, those operating centrifugal pumping units account for more than 70% of the total number of units of dynamic equipment under the control of monitoring and diagnostic systems. The operation of rolling bearings differs from plain bearings, they have a constant mechanical contact between the parts of the electric motor. The resource of rolling bearings during manufacture allows for a long time to work without being subjected to significant wear, fatigue phenomena accumulate in the parts of the rolling bearing, which ultimately lead to fatigue damage and failure. At present, the

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state control is carried out without taking into account the possible non-stationary nature of the development of malfunctions, which creates the problem of ensuring the control of the state of a centrifugal pump unit during operation in order to prevent sudden failures of units, assemblies and parts [1].

Vibration of pumping units is caused by causes of mechanical, electrical and hydrodynamic origin. The least studied and predictable is the vibration of hydrodynamic origin, which occurs when pumps operate at low and high flows. Under such operating modes, an intense dynamic effect occurs on the hydraulic part of the pumps, which is perceived by the mechanical part and transmitted to the bearing units, housing and foundations of the units. This type of vibration is accompanied by the destruction of bearings, displacement and beating of shafts, an increase in the vibration of building structures and pipelines adjacent to the pumps. In the most severe cases, the destruction of impellers and bends, volutes of large pumps and pipelines tying the pumps takes place. Vibration can also result in intense hydroabrasive wear of the flow part of pumps, which entails a decrease in cavitation characteristics and efficiency. pumps with a subsequent deterioration in the economic performance of pumping facilities.

The most severe consequences of hydrodynamic vibration of pumps are not only material losses from the destruction of equipment and temporary interruption of the transport of the pumped liquid, but also significant damage to the environment from the spill of environmentally aggressive liquids from destroyed equipment and pipelines. The increased vibration of the equipment also causes significant harm to the maintenance personnel of pumping stations, since, as practice shows, when vibration occurs, it can exceed the permissible values provided for by sanitary and hygienic standards. Severe consequences from vibration of hydrodynamic origin require further deeper study of its nature. Especially acute need for such a study arose in the pipeline transport of water, where, due to the reduction in the volume of water transport, powerful high-performance equipment is forced to operate with an underload in performance and experiences vibration of a hydrodynamic nature [2].

2 Methods

The analysis of the work performed in the area under consideration showed that objectively there is a certain physical relationship between the main design parameters and operating modes of pumps and the level of vibration of pumping units when their operating mode deviates from the optimal one. The same analysis indicates that the existing work on the problem under study, as well as work in related areas, do not reveal the specific causes of technological vibration of centrifugal pumps; the studies available in this direction are relatively fragmentary, with a predominantly empirical bias, and do not have a sufficient generalizing theoretical basis.

Such a state in the study of the problem under consideration requires, in order to achieve more effective results, to approach the solution of the issue under study primarily from theoretical positions based on modern hydrodynamic models and calculation methods. The theoretical approach can be implemented in several ways based on several models that reflect real processes with varying degrees of adequacy. These models include: a quasi-one-dimensional model; a model based on the theory of the boundary layer (spatial inviscid flow in interblade channels, viscous near-wall flow); a model based on the theory of "three models" (spatial quasi-viscous flow in interblade channels, deformation of turbulence characteristics in the found velocity field, viscous unseparated and separated near-wall flow); model based on 3D Reynolds equations.

Based on the nature of the tasks posed in the work, their solution was carried out using two models - a quasi-one-dimensional model and a model based on the theory of "three

models". The first of them is distinguished by ease of use, which is necessary for engineering practice, the second is the necessary scientific level, since it is based on modern achievements, both in the field of physics of hydrodynamic processes and in the field of mathematical description of these processes. At the same time, this option is characterized by relatively low labor intensity. The noted variants of the model are considered in chapters 2 and 3 of this work.

Dedicated to the calculation of hydraulic energy losses in the elements of the flow part of centrifugal pumps. Consideration of only hydraulic energy losses is due to the goals and objectives of this study [3].

An analysis of published works on the assessment of hydraulic energy losses in centrifugal pumps shows that most researchers, whose results are cited, in particular, in the publications of many scientists, when determining pressure losses in vane injection machines, are limited to considering a quasi-one-dimensional flow model based on a system of equations including the continuity equation, the Bernoulli equation, the equation reflecting the law of change of the angular momentum, as well as on the analytically generalized data on pressure losses in the elements of the flow part of machines.

An analysis of previously created calculated dependencies is carried out to find various types of hydraulic energy losses in centrifugal pumps according to their existing classification, and a more advanced method for determining these types of pressure losses is being developed in order to obtain more accurate information to search for specific causes of vibration of hydrodynamic origin [4].

3 Results and discussion

The decrease in the pump head (without energy losses) as a result of the inertial movement of the fluid in the interblade channels of the impeller is found in the form of an identification parameter μc, which takes into account the decrease in the theoretical velocity vector C2mT by the inertial movement of the fluid.

$$\mu_c = \left(1 - \frac{X_u}{C_{2mT} \cdot U_2}\right);$$

$$X_u = \frac{r \cdot \omega^2}{2} \cdot (D_2 - D_1);$$

$$r = \left(\frac{D_2^2 - D_1^2}{4 \cdot Z}\right)^{0,5} \cdot \left(1 - \frac{Q}{Q_{zo}}\right)^{0,5} \cdot \Delta_r;$$

$$Q_{zo} = \frac{k \cdot U_1 \cdot [\pi \cdot l(D + d) - 2 \cdot Z \cdot \delta_1 \cdot l]}{2 \cdot sin\alpha \cdot ctg\beta_{1r}};$$

where: Δr - is an identification parameter that takes into account the inertial movement of the fluid in the interblade channels, which can be taken in the range of 0.92-0.93 for approximate engineering calculations; the remaining quantities are parameters characterizing the geometric dimensions of the impeller and its interblade channels, as well as the quantities included in the L. Euler equation [5].

The parameter μc allows calculating the theoretical head of the pump, taking into account the inertial movement of the liquid in the interblade channels of the HTS impeller according to the formula

$$H_{TS} = \frac{1}{g} \cdot (\mu_c \cdot C_{2mT} \cdot U_2 - C_{1mT} \cdot U_1)$$

To calculate the pressure loss in the impeller from the hydrodynamic processes occurring in it, the dependence

$$h_{g1} = \frac{\mu_D}{2 \cdot g} \cdot [W_n + (r \cdot \omega) \cdot cos\, f \cdot sin\, \beta_{1l}]^2$$

where: μD - is an identification parameter that takes into account the degree of energy dissipation from the interaction of various types of fluid movement in the interblade channels:

$$W_n = cos\, f \cdot sin\, \beta_{1l} \cdot \left[\frac{\pi \cdot n \cdot D_1}{60} - \frac{2 \cdot sin\, \alpha \cdot (ctg\beta_{1l})}{K \cdot [\pi \cdot l \cdot (D + d) - 2 \cdot z \cdot \delta_1 \cdot l]} \cdot Q\right]$$

where: f - is the angle of inclination of the blades to the disks of the impeller;

βx - is the angle between the blade and the direction of the inertial movement of the fluid in the interblade channel, is found from the empirical dependence

$$\beta_x = 34 - 2{,}8\left(\frac{\beta_{2l}}{\beta_{1l}}\right)^{3{,}67}$$

The pressure loss in the branch was found as the sum of the losses from the contact of the flow with the inlet edge of the branch “tongue” and the losses occurring when the flow moves in the expanding branch channel. To calculate the pressure loss from contact with the input edge of the "tongue" of the outlet, the dependence

$$h_{g2} = \frac{C_{2u}^2}{2 \cdot g \cdot z} \cdot \left[\left(1 - \frac{tg\alpha_2}{tg\alpha_{3k}}\right)^2 + \frac{2 \cdot P_{D1}}{\rho \cdot C_{2u}^2}\right]$$

where $C_{2u} = \mu_c \cdot C_{2ur}$; α2 - is the angle of inclination of the absolute flow velocity vector at the outlet of the impeller;

α3k - is the angle of the bevel of the input edge of the “language” of the outlet;

PD1 - pressure loss from the viscous separated fluid flow in the near-wall area at the input edge of the "language" of the outlet.

The pressure loss from the expansion of the flow in the branch, according to the studies performed, is determined by the dependence

$$h_{g2} = \frac{C_{2u}^2}{2 \cdot g} \cdot \left(\frac{\overline{S} - \frac{tg\alpha_3}{tg\alpha_2}}{\overline{S} - 1}\right)^2 \cdot \left[\left(1 - \frac{tg\alpha_2}{tg\alpha_3}\right)^2 + \frac{2 \cdot P_{D2}}{\rho \cdot C_{2u}^2} \cdot \frac{tg\alpha_3^2}{tg\alpha_2^2}\right]$$

where α2 – the angle of inclination of the vector of the absolute flow velocity at the outlet of the impeller;

α3 – is the angle of inclination of the wall of the spiral outlet or the guide vanes of the outlet;

$\overline{S}$ - the ratio of the area of the flow section of the discharge pipe of the pump to the area of the flow section of the impeller at its outlet;

P_{D2} - pressure loss due to viscous separated fluid flow in the near-wall region of the outlet.

It is difficult to determine the numerical values of the parameters μD and PD1 and PD2 included in the dependences for calculating hg1, hg2 and hg3 with sufficient accuracy for practice. Therefore, the pressure loss from the viscous separated flow was found immediately for the entire flow part of the pump, that is, integrally. The task was to close the conducted studies on hydraulic losses in the pump on the basis of the energy balance [6].

The studies carried out made it possible to obtain the following general dependence for calculating pressure losses from the viscous separated fluid flow in the near-wall region of the pump flow part:

$$\overline{h_w} = \frac{(1 + \zeta)}{g \cdot F_H^2} \cdot Q^2 - b_T \cdot Q$$

where ζ - is the coefficient of hydraulic resistance of the flow part of the pump;

FH - is the cross-sectional area of the flow (at the branch outlet);

bT - is an identification parameter that takes into account the vortex nature of the fluid flow

$$b_T = \frac{\overline{C}_{2u}}{g \cdot F_H}$$

$\overline{C}_{2u}$- the speed of the "vortex" flow at the inlet to the outlet. To calculate ζ and bT, empirical dependences were obtained, approximating the experimental data for most pumps of the normal series of NM used in main water transport.

Calculation of the pressure characteristics of pumps according to the obtained dependencies based on formulas and subsequent comparison of the calculation results with experimental characteristics

$$H_{fr} = H_{TS} - \left(h_{g1} + h_{g2} + h_{g3} + \overline{h}_w\right)$$

where: Hfr - is the calculated value of the actual pump head;

hg1, hg2 and hg3 - are the corresponding head losses calculated without taking into account losses in the near-wall region, that is, at μD = 1 and PD1 = PD2 = 0 approximating dependences for finding the parameters Δr, βx, ζ and br is no more than 3% [7].

4 Conclusions

On the basis of well-known simplified approaches to the calculation of flow parameters in individual elements of the flow part of pumps, a technique for quasi-one-dimensional engineering calculation of pressure losses in centrifugal pumps of the D6300-80 type used in main water transport has been developed.

The calculation according to the proposed method for pumps of the type (D6300-80, D6300-27, D4000-95, D3200-75, 6NK9KX1) and the pump 16ND 10x1 of the same type with them according to the design scheme showed that, despite the assumptions and assumptions made, the maximum calculation error did not exceed 3%.

The quasi-one-dimensional approach, which gave acceptable results in the assessment of hydraulic losses in centrifugal pumps, cannot give similar results when solving optimization problems related to finding the most perfect geometric shape of individual elements of the flow path of pumps, since it does not take into account the spatial shape of these elements.

The failure of a centrifugal pump due to fatigue failure of a rolling bearing does not occur suddenly and not instantly. First, some signs of the approach of this process appear, the nature of the vibroactivity of the aggregate changes. Against the background of a stable vibration level, some vibration outliers appear.

The study of the sequence and intensity of these emissions is important information not only about the approach of the moment of destruction of the rolling bearing parts, but also about the operating time that the operating personnel still have to take measures to eliminate an emergency and complete catastrophic destruction.

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