JOURNAL OF ENGINEERING SCIENCES

Volume 11, Issue 1 (2024)


*Makivskyi O., Kondus V., Pitel J., Sotnyk M., Andrusiak V., Polkovnychenko V., Musthai M. (2024). The influence of the design features of the submersible pump rotor on the vibration reliability. Journal of Engineering Sciences (Ukraine), Vol. 11(1), pp. D1–D9. https://doi.org/10.21272/jes.2024.11(1).d1*

# The Influence of the Design Features of the Submersible Pump Rotor on the Vibration Reliability

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**Article info:**

| | |
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| Submitted: | August 29, 2023 |
| Received in revised form: | November 20, 2023 |
| Accepted for publication: | December 18, 2023 |
| Available online: | January 5, 2024 |

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**Abstract.** Pumping equipment consumes about 20 % of the electrical energy produced by humankind. A significant, even drastic, reduction in the weight and size indicators of pumping equipment leads to a decrease in the cost price and, therefore, competitiveness of such products in the market. Simultaneously, it makes it possible to use more valuable and high-quality construction materials and technologies that improve the reliability of equipment and its energy efficiency, which in turn is a clear step in solving many UN Sustainable Development Goals (SDGs). According to the research results, it was proved that by increasing the frequency of the drive, it is possible to reduce the mass and size indicators of the submersible pump for the needs of the critical infrastructure sector by reducing the number of stages. Mainly, the amplitudes of oscillations near the rotation frequency are 12–22 % and do not exceed 35 % of the gaps in the seals, as required by the available international standards to ensure the guaranteed vibration reliability of the pump. Overall, using a bearingless support design will significantly increase the reliability of the developed pump.

**Keywords:** vibration reliability, process innovation, life cycle cost, energy efficiency, materials cost, weight and size indicators.

## 1 Introduction

Nowadays, there are clear guidelines for human development, namely the Sustainable Development Goals (SDGs) approved by the United Nations. Compliance with the directions of their implementation is essential for the effective and correct development of human society. According to the SDGs, in the process of industrial equipment development, special attention should be paid to achievements in the following areas: “Clean water and proper sanitation” (goal No. 6), “Affordable and clean energy” (goal No. 7), and “Industry, innovation and infrastructure” (goal No. 9). This is closely related to pumps and pumping equipment because pumping equipment itself consumes about 20% of the electric energy produced by humankind [1].

One of the main indicators that can connect the two parameters of the machine, its mass, and power, can be taken as the specific power of the machine, that is, the electrical power per unit mass of the machine. By this indicator, the technical level of machines can be evaluated, which will give an impetus to the simultaneous increase of their specific energy intensity. Currently, this is possible by using power electronics to increase the speed of rotation of the rotors of such machines [2].

The continuous decrease in the price of current frequency converters – inverters for power electric drives opens up new perspectives in pump construction [3].

All the abovementioned applies to many of the nomenclature of pumping units of the dynamic principle of action, primarily centrifugal ones. The power of these machines, i.e., the flow rate times the pressure, is a function of the rotational speed in cubic degrees. A significant, even drastic, decrease in weight and size indicators leads to a decrease in the cost price and, therefore, competitiveness in the market [4]. Simultaneously, it makes it possible to use more expensive and high-quality construction materials and technologies that improve the reliability of the equipment [5] and its energy efficiency [6].

The power of an AC electric motor with an increase in rotation frequency also increases in direct proportion since its torque is determined by the design of the electromagnetic active part and remains unchanged in the first approximation. Thus, the mass of the active part of the engine at a constant power is inversely proportional to the speed of rotation, which should reduce the dimensions, mass, and cost of the engine when the speed increases, although not as rapidly as the pump of dynamic action (in a linear relationship, not in a cubic one).

## 2 Literature Review

The rotation frequency of the most reliable, cheap, and widespread asynchronous motors is limited by a current frequency of 50 Hz at 3000 rpm [7]. Frequent inverters, gear multipliers, or a turbo drive are used to increase it.

Turbo drive is from a steam, hydraulic, or air turbine. Of course, its application is possible only in the presence of appropriate energy sources and has many nuances, so it is a limited topic in general pump construction. However, it should be noted that in such cases of the absence of restrictions, designers often choose a rotation speed of at least 5000–6000 rpm, giving the optimal ratio of reliability, energy efficiency, and cost.

In cases of extreme necessity, gear multipliers are used, but they are expensive, bulky, and, most importantly, unreliable and durable enough to reduce the final efficiency of the unit, so they have received minimal distribution in pump construction.

Powerful industrial inverters for electric drives have existed for many decades, but the drive of pumping units began to be used widely not so long ago after achieving sufficient reliability and acceptable cost. In most cases, they are used to increase energy efficiency during performance regulation or to ensure a “soft start” of the unit. Only the first isolated cases of the implementation of quite specific projects of high-speed centrifugal pumps with an inverter and a direct drive in the oil production field are known to create high pressure at a relatively small flow of liquid [8].

Meanwhile, the prices of inverters have fallen by half in recent years, to approximately 50 EUR per 1 kW of power, and continue to decrease. Under such conditions, some high-value categories of pump units (corrosion-resistant, submersible, some other multistage) have already reached a level where we should expect an actual positive technical and economic effect due to increasing the rotation speed with the help of an inverter. Despite all the apparent advantages, the following assumptions can explain insufficiently pronounced activity in this direction [9]. First and foremost, production inertia is too high: structures tested for decades brought to perfection and existing corresponding production capacities. Second is the high cost of research and design work, which is necessary to solve several specific technical issues in high-speed hydraulic machines. Finally, there is a specific situation in the electric drive market: compact high-speed motors, instead of the expected reduced prices, have a higher price than conventional motors. This anomaly can be explained by the low seriality associated with the small needs. Thus, we have a closed circle: there are no high-speed pumps because there are no cheap, mass-produced motors, and there are no motors because there is not a sufficient level of orders that these same pumps can supply.

Therefore, it is possible to hypothesize that the time has come to actively introduce the inverter direct high-speed drive into the practice of general pump construction.

The most significant and fastest technical and economic effect can be expected from the modernization of those pumping unit types with the highest specific indicators of the price per 1 kW of power [10]. One of the prominent examples of such units are submersible well pumps, which, due to the lack of transverse space for location and specifically low consumption at relatively high pressures, represent a thin elongated structure with many operating stages. Such a layout feature reduces the pump’s reliability and energy efficiency [11] and the electric motor.

Increasing the rotation frequency of the submersible pump, for example, twice, from 3000 to 6000 rpm, leads to a reduction of the active part of the engine by 2 times and a reduction in the number of pump stages by 4 times [12].

Using an asynchronous motor of a standard design for a submersible pump unit is not without alternatives. Using an inverter to increase the current frequency makes it possible to use a synchronous motor, which immediately gives advantages in energy efficiency. First, the efficiency of a synchronous motor is, by definition, higher by several percent (depending on power) due to the absence of slip losses. Secondly, the synchronous motor’s rotor has smaller dimensions, which reduces hydraulic friction losses in a “wet rotor” motor, which is traditional for units of this type. Thirdly, there is even an opportunity to switch to a gas-filled engine since the main factor of rotor heating-electrodynamic is absent in a synchronous engine, and its rotor does not require liquid cooling [9].

The creation of a high-speed pump requires a change in essential composite solutions due to a significant increase in the energy saturation of the pump’s operating stage, a drastic decrease in their number, and an increase in radial hydrodynamic forces in the flowing part, which determine the vibration state and reliability of the structure. Nevertheless, at the same time, these circumstances lead to new opportunities for increasing the pump’s energy efficiency, firstly – due to the optimization of constructive solutions and secondly – due to the implementation of higher quality and expensive materials and technologies. Such possibilities are connected, first, with a drastic reduction in the number of stages and mass-size parameters of the pump [1].

Thus, introducing inverters increases submersible pump units’ technical and economic levels, but with a particular correction of permanent technical solutions for the pump part and the electric motor. According to rough calculations, the increase in energy efficiency of the unit should be from 5 to 10 %, and the cost reduction of the mechanical part should be from 20 to 50 %. Under such circumstances, the additional costs of including the inverter unit in the unit's composition are covered in a relatively short period of operation [9].

The vibration state of the pump is the primary indicator that characterizes its reliability and durability [13]. According to statistics, most of the problems in the pump are related to vibration. The main source of vibration in centrifugal pumps is an unbalanced rotor.

There are several reasons for vibration [14]: misalignment of the drive and pump rotors, unbalanced radial forces occurring on the connecting couplings, which deform the rotors, and rolling and sliding bearings. Mainly, this applies to rolling bearings, which are a source of oscillation due to elastic deformations of parts. Oscillations are especially noticeable in the case of improper installation.

The other reasons are imbalance of the rotor, non-coincidence of the main axis of inertia of the rotor with the axis of rotation, and hydrodynamic forces caused by the inhomogeneity of the flow at the outlet of the impeller. This problem is partially solved by the correct ratio of the number of blades of the impeller and the outlet blades; hydrodynamic factors acting in the flow part of the pump, mainly in slotted seals (hydrodynamic forces in slotted seals significantly impact rotor stabilization and dynamics.

Since the rotors of centrifugal machines in the process of operation inevitably carry out forced radial-angular oscillations, it is possible to solve the resulting vibration problems in two ways: either to achieve a reduction in the level of vibration activity of the pump unit by carefully balancing and centering the pump rotor or to provide the rotor with the possibility of free self-installation in the support sealing units, which must function in all operating modes of the unit from start-up to full stop. In the second case, the weight and dimensions of the pump are reduced significantly, and bulky and expensive external bearing assemblies are eliminated. In particular, the drive shaft of the torque from the drive to the working wheels can be made much thinner. This leads to an improvement in the vibroacoustic characteristics of the centrifugal machine and a decrease in its cost [15].

While developing the promising ECW 8-63-150 submersible pump [16], the authors considered the second method of solving the emerging problems related to the vibration state of the rotor. For this, it is necessary to refer to the literature and determine the optimal sealing parameters for this flow part. It was necessary to consider the leakage through the seals, which are affected by the pressure differences and the average cross-sectional area [17]. Gap seals are small gaps between the rotor's rotating parts and the corresponding stator housing's parts (Figure 1).

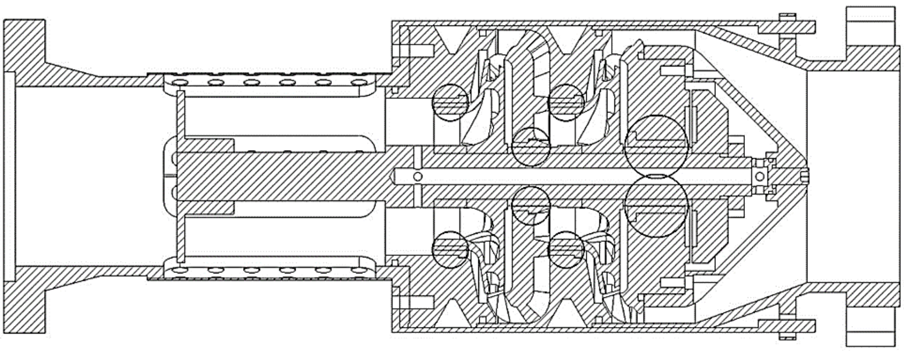

Figure 1 – The developed pump ECW 8-63-150

The shaft rotating in the seal in the presence of a pressure drop from the axial flow of the liquid is subject to significant hydrodynamic forces, which exceed the elastic forces of the shaft during bending and mainly affect the dynamic characteristics of the rotor (Figure 2). It is customary to divide these forces into the following components [18].

The elastic force $F_k = k \cdot e$ is proportional to the transverse displacement (eccentricity $e$, m) of the shaft in the sleeve. It is directed opposite to it and depends on the seal's stiffness coefficient $k$, N/m. The elastic force of the shaft leads to an increase in the total stiffness and natural frequencies of the transverse oscillations of the rotor.

The damping force $F_b = b_1 \cdot v$ is proportional to the speed $v$ of transverse movement of the shaft, is directed opposite to it, and depends on the damping coefficient of the seals $b_1$, N·s/m. Damping forces reduce the amplitude of shaft oscillations, especially at close values of its natural frequencies and rotation frequency. At sufficiently high values of the damping coefficients, "resonant peaks" of the amplitude disappear altogether, and "critical speeds" cease to pose a danger.

The circulating force $F_q = q \cdot e$, like the elastic one, is proportional to the eccentricity of the shaft in the bushing but is directed to it at a right angle and depends on the coefficient of circulating sealing forces $q$, N/m.

During shaft precession, the vector of circulation forces is directed opposite to the vector of damping forces, so circulation forces partially compensate for damping and are destabilizing. In addition, under certain conditions, circulation forces cause rotation (precession) of the rotor with a frequency that is not synchronous with rotation – loss of dynamic stability. In this state, the machine cannot be operated.

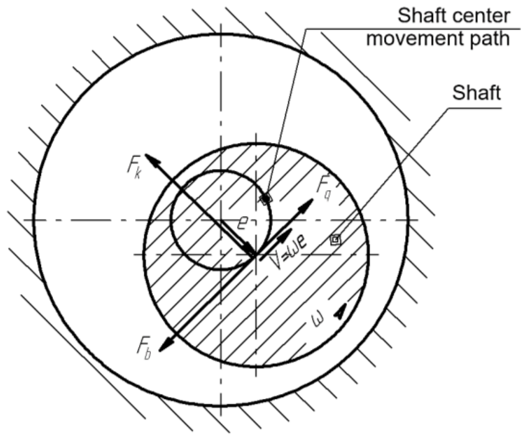


Figure 2 – Hydrodynamic forces in a gap seal:
$F_k$, $F_b$, $F_q$ – elastic, damping, and circulation forces in a gap seal, N; $\omega = \pi\, n/30$ – angular rotation frequency, rad/s

The coefficient of circulation forces depends on the swirl of the flow at the seal inlet. With a typical centrifugal pump relative input twist coefficient 0.5 (half of the shaft rotation frequency), the condition for loss of dynamic stability is that the shaft rotation frequency is twice the first natural frequency of transverse oscillations $\omega > 2\omega_1$, rad/s Suppression of swirling at the seal inlet reduces circulation forces and ensures proper operation of the pump at a rotation frequency of up to $\omega = (3–4)\ \omega_1$ [19].

Significant radial forces with low viscosity of the environment and large gaps are possible due to the axial flow created by high-pressure drops on the seals. All the forces listed above are proportional to the axial pressure drop. In this research, the authors concluded that the centering force in a cylindrical gap arises due to the variable pressure drop at the inlet. Obfuscation and positive skew give a centering force even without considering input losses. This is explained by the dependence of the law of change of static pressure on the length of the ratio of gaps at the inlet and outlet from the channel: in the confusing channel, the pressure curve has a convex shape, and in the diffusing channel – concave, and channels of constant cross-section, the pressure changes in a straight line (Figure 3).

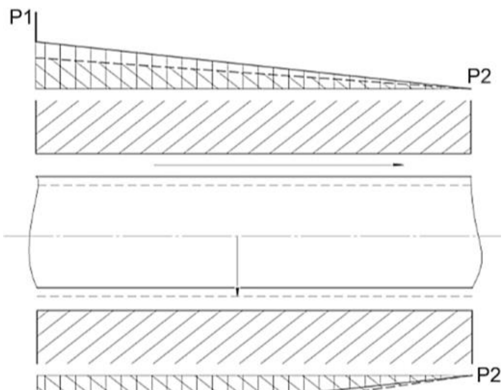


Figure 3 – Pressure distribution in slotted seals

The diagram shows how the pressure in the gap seal changes with the axial displacement of the rotor. As long as there is no eccentricity, the pressure distribution graphs are the same at all points. First, there is a pressure loss at the inlet to the seal; a thick line on the graph indicates this. If, during the pump operation, the shaft moves in a particular direction, then the gap on the opposite side increases. Accordingly, the flow rate increases, and the pressure drops. As the gap towards which the rotor has moved decreases, the flow rate decreases accordingly, and the pressure increases. This is indicated by dashed lines on the diagram and the figure. As the pressure increases, it pushes the shaft into the zone of lower pressure, accordingly trying to center the rotor.

For gap seals to have the highest efficiency and for the pump to operate without the support of bearings, it is necessary to correctly choose the size of the gap and its length.

Based on the results of the literature review, the purpose of the study was determined: determining the vibration reliability of the submersible pump rotor for the development of recommendations for the design of a series of bearingless pumps with increased energy efficiency and reduced weight and size indicators.

In order to achieve the goal, the following tasks were formulated: to ensure the dynamic stability of the rotor of a high-speed submersible pump without bearings (by determination of the frequencies and forms of the rotor's natural oscillations, and amplitudes of forced oscillations in dangerous sections of the rotor); based on research results and the current regulatory framework, draw conclusions about the performance of the proposed pump design; to determine how to reduce the weight and size indicators of the pump.

## 3 Research Methodology

### 3.1 Calculations of dynamic parameters of seals

For calculations on vibration resistance, it is necessary to research the transverse vibrations of the rotor. At the same time, using the relevant provisions of the international standard API 610 / ISO 13709:2009 "Centrifugal pumps for petroleum, petrochemical and natural gas industries" is advisable. This standard applies to pumps in the petrochemical industry, but they do not have fundamental design differences from other centrifugal pumps for other fields of application, while this standard has gained an impeccable reputation over several decades of use. According to this standard, calculations are made for two values of gaps in seals: nominal and doubled (that is, in the extreme pre-repair condition).

To analyze the vibration state of the rotor, the first three natural forms of transverse oscillations (for nominal and double clearances in the seals) are calculated.

### 3.2 Free and forced oscillations

The BEAM module of the ANSYS software was used to construct the amplitude-frequency response and three modes of natural oscillations of the pump rotor. Ansys Mechanical provides an integrated platform that uses finite element analysis (FEA) for structural analysis. Mechanical is a dynamic environment with a complete set of analysis tools, from geometry preparation for analysis to plugging in additional physics for even greater accuracy. An intuitive and customizable user interface allows engineers at all levels to get answers quickly and confidently.

ANSYS provides reliable connectivity to CAD tools, enabling point-to-point button design updates. Fully integrated multiphysics capabilities are available with fluid and electrical solvers.

Several programs are available for studying rotor dynamics. They were considered when choosing the software for our research. After reviewing the literature on this topic, the BEAM188 ANSYS software package was chosen. The main advantage is the possibility of using the student version for free. We also have experience using this software. A comparison of research results obtained from the work of other scientists shows that Ansys is a reliable tool for carrying out dynamic calculations of the rotor and has minor errors compared to others [20, 21].

Before building and entering data into the program, a calculation diagram of the rotor was built (Figure 4), where the shaft is shown as a stepped cylinder, the elements of the rotor (impellers, hydraulic heel disc, clutch, bushings) are represented as point masses, and the seals are represented as point-attached masses and elementary elastic damping links [22].

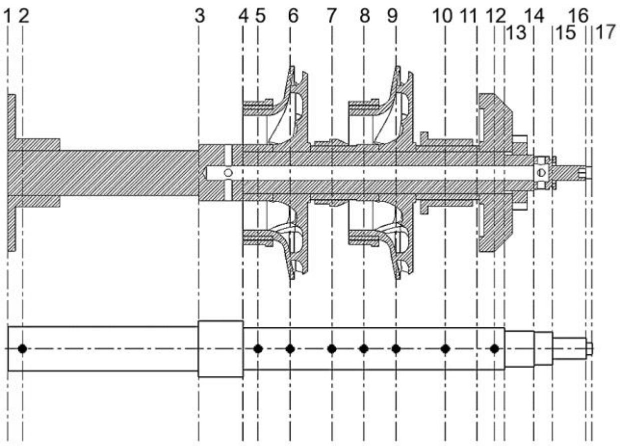


Figure 4 – Graphic view of the calculation scheme

The centers of mass of the rotor elements are chosen as the points of mass attachment. The file in which the geometry of the rotor was specified was named geometry (Figure 5). For the correct execution of the calculation in the Ansys software complex, the file was created without an extension.

The location of the perturbation point was also specified. Moreover, it does not always have to be the middle point of the shaft, which depends on the point for which the calculation must be performed. According to the standard API 610 / ISO 13709:2009, the amount of imbalance is 4 times for the balancing accuracy class G 2.5 according to ISO 21940-11:2016 “Mechanical vibration. Rotor balancing Procedures and tolerances for rotors with rigid behaviour”.

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Figure 5 – View of the geometry file (from left to right): distance from the left end, mm; shaft diameter, mm; stiffness of a seal, N/m; damping coefficient, N·s/m; added mass, kg

To run calculations in ANSYS Mechanical, the following files without extensions were created:

– “scalc” is a file that describes the method of constructing the amplitude-frequency characteristic;

– “mcalc” is a file that describes the method of constructing the self-oscillations of the rotor. These files use geometry file data.

## 4 Results

Based on the research results, the dynamic parameters of seals were determined for nominal gaps and doubled gaps [23] (Table 1).

Table 1 – Calculation results of dynamic parameters

| Nominal clearance | | | |
|---|---|---|---|
| Seal type | Flow rate $Q$, m$^3$/s | Stiffness $k$, N/m | Damping $b_1$, N·s/m |
| Front | $1.3\cdot10^{-3}$ | $1.14\cdot10^{6}$ | $1.21\cdot10^{3}$ |
| Interstage | $0.2\cdot10^{-3}$ | $0.26\cdot10^{6}$ | $0.73\cdot10^{3}$ |
| Hydroheel | $0.5\cdot10^{-3}$ | $0.41\cdot10^{6}$ | $1.94\cdot10^{3}$ |
| Double clearance | | | |
| Seal type | Flow rate $Q$, m$^3$/s | Stiffness $k$, N/m | Damping $b_1$, N·s/m |
| Front | $3.2\cdot10^{-3}$ | $0.61\cdot10^{6}$ | $0.14\cdot10^{3}$ |
| Interstage | $0.6\cdot10^{-3}$ | $0.16\cdot10^{6}$ | $0.19\cdot10^{3}$ |
| Hydroheel | $1.2\cdot10^{-3}$ | $0.27\cdot10^{6}$ | $0.63\cdot10^{3}$ |

It was established that in the case of a two-fold increase in the gaps, the stiffness coefficient of the front seal decreases from $1.14\cdot10^{6}$ to $0.61\cdot10^{6}$ N/m (by 46 %). The stiffness coefficient of the interstage seal decreases from $0.26\cdot10^{6}$ to $0.16\cdot10^{6}$ N/m (by 38 %). The hydroheel stiffness coefficient decreases from $0.41\cdot10^{6}$ to $0.27\cdot10^{6}$ N/m (34 %).

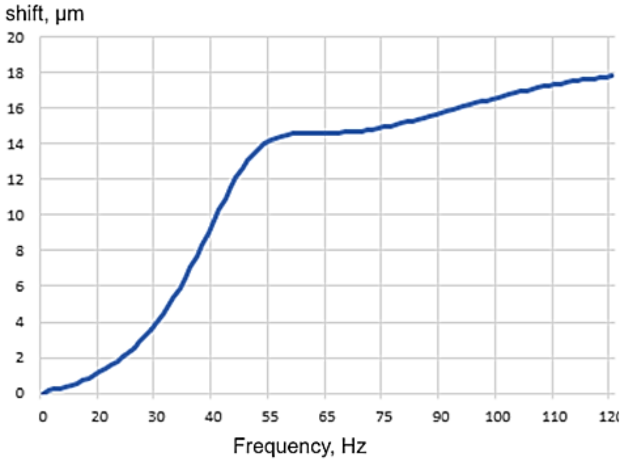
In turn, in the case of a two-fold increase in clearances, the damping coefficient of the front seal decreases from $1.21\cdot10^{6}$ to $0.14\cdot10^{6}$ N/m (by 88 %). The damping coefficient of the interstage seal decreases from $0.73\cdot10^{6}$ to $0.19\cdot10^{6}$ N/m (by 74 %). The hydroheel damping coefficient decreases from $1.94\cdot10^{6}$ to $0.63\cdot10^{6}$ N/m (by 67 %).

Simultaneously, the flow of liquid through the front seal in the case of a two-fold increase in clearances increases from $1.3\cdot10^{-3}$ to $3.2\cdot10^{-3}$ m$^3$/s. Liquid consumption due to interstage sealing increases from $0.2\cdot10^{-3}$ to $0.6\cdot10^{-3}$ m$^3$/s.

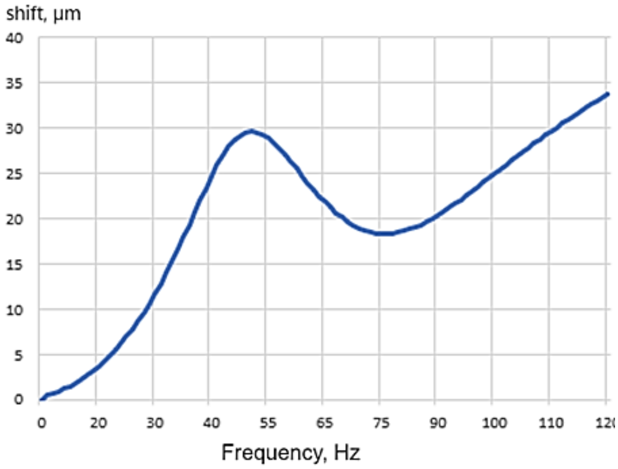
The consumption of liquid due to hydropite increases from $0.5 \cdot 10^{-3}$ to $1.2 \cdot 10^{-3}$ $m^3/s$.

From the obtained results, it can be seen that ensuring minimum clearances in the seals is an urgent need to ensure rigidity and damping of the system.

The amplitude-frequency characteristic is constructed for the nominal (Figure 6) and doubled (Figure 7) gaps in two points of excitation – the radial gap of the hydroheel and the front seal of the first-stage impeller.




a



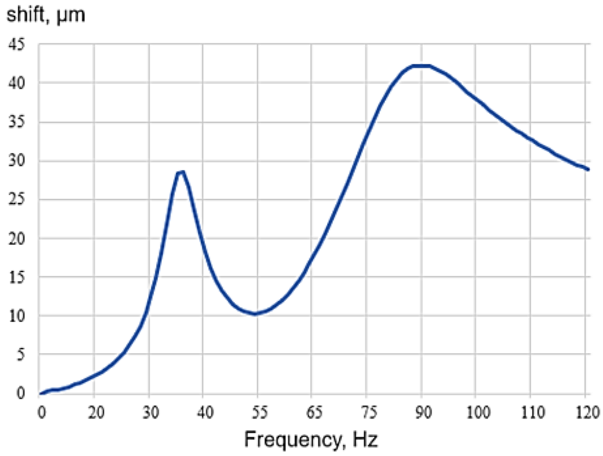

b

Figure 6 – Amplitude-frequency characteristics at the excitation points of the front seal of the first-stage impeller (a) and the radial gap of the hydroheel (b)

As seen from the appearance of the amplitude-frequency characteristic, dynamics and resonance phenomena are not expressed.

The maximum amplitudes of radial oscillations or the radius of precession of the rotor in the range up to 120% of the rotation frequency do not exceed 18 and 42 µm in the cross section under the front seal of the impeller of the 1st stage, as well as 33 and 88 µm – in the cross-section under the radial gap of the hydroheel (respectively for the nominal value of the radial gap of 200 µm and doubled – 400 µm).

Since the second natural frequency is close to the rotation frequency, the "critical" points or intersections of the rotor are the points with the largest amplitudes of oscillations, respectively, according to the second natural form [18]. Those located under the seals are where the gap between the rotor and the stator is the smallest. In our case, it is probably the intersection under the right edge of the radial seal of the hydroheel and the intersection under the front seal of the first stage.

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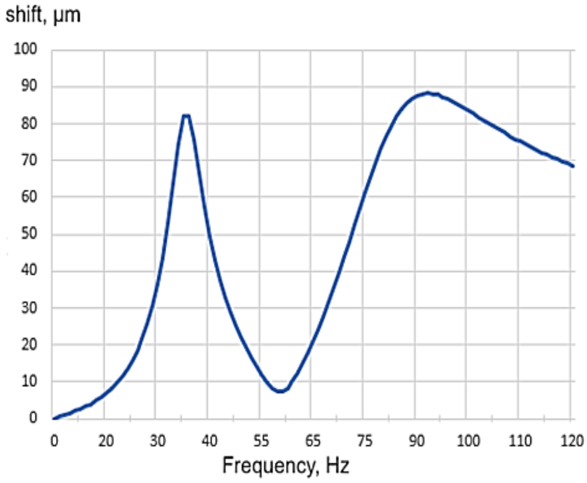
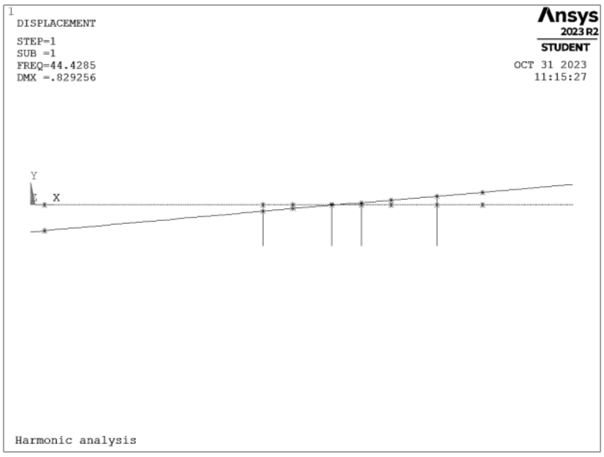



b

Figure 7 – Amplitude-frequency characteristics at the points of excitation of the front seal of the first-stage impeller (a) and the radial gap of the hydroheel (b) at doubled clearances

It is for these intersections that the amplitude of oscillations was calculated in the form of amplitude-frequency characteristics, which allow a better understanding of the essence of dynamic processes (amplitude-frequency characteristics are constructed without taking into account the dependence of the stiffness of non-contact seals on the rotation frequency). The imbalance is distributed over all the parts that make it up.

Based on the research results, the forms of the natural oscillations of the rotor in the case of nominal clearances (Figure 8) and doubled clearances (Figure 9) in sealing devices were constructed.

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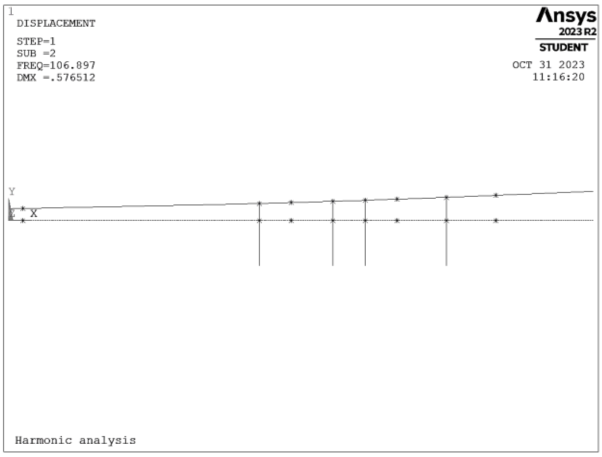


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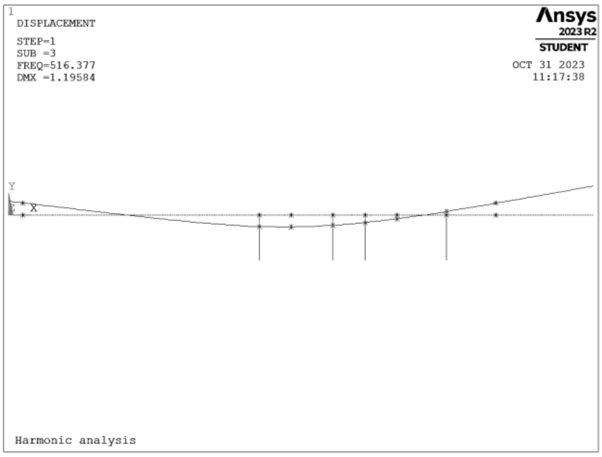
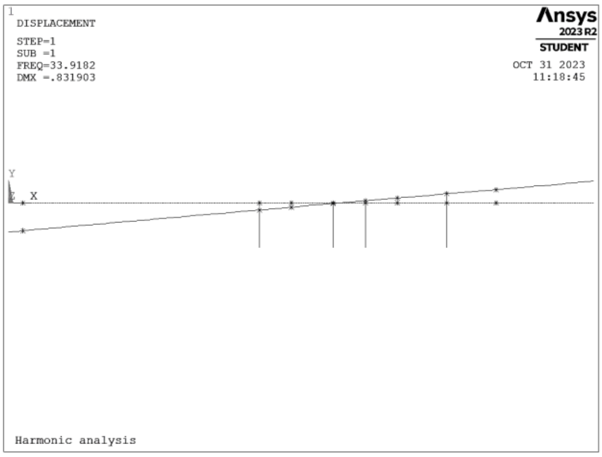



c

Figure 8 – 1st (a), 2nd (b), and 3rd (c) forms of self-oscillations of the pump rotor at nominal clearances

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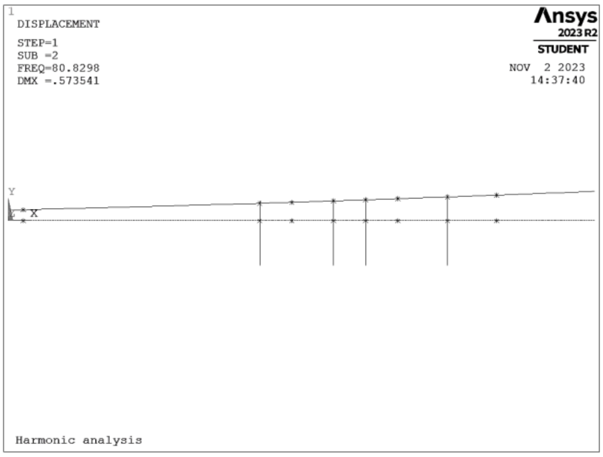


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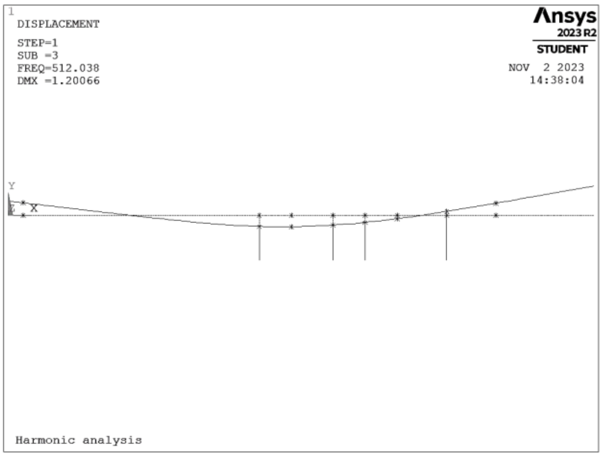


c

Figure 9 – 1st (a), 2nd (b), and 3rd (c) forms of self-oscillations of the pump rotor with double clearances

According to the simulation results, the first form of natural oscillations is observed at a frequency of 44 Hz or 2640 rpm in the case of nominal gaps (Figure 8 a) and 34 Hz or 2034 rpm in the case of doubled gaps (Figure 9 a) in sealing devices. The direction vectors of the imbalance forces are located in opposite directions, which partially compensates for them.

The third natural frequency is 516 Hz or 30 984 rpm in the case of nominal gaps (Figure 8 c) and 512 Hz or 30 720 rpm in the case of double gaps (Figure 9 c) in sealing devices. The vectors of the imbalance forces are located in opposite directions.

The worst option for excitation is observed for the second form of oscillation at a frequency of 107 Hz or 6414 rpm in the case of nominal gaps (Figure 8 c) and 81 Hz or 4850 rpm in the case of doubled gaps (Figure 9 c). In this case, the direction vectors of the imbalance forces are located in the same direction for all parts. However, it should be noted that according to the specifications for the developed ECW 8-63-150 pump, the operation of the pump is limited to a frequency of 100 Hz or 6000 rpm.

Thus, in the case of using seals with a nominal gap, the second form of natural oscillations is not achieved. In the case of using seals with a double gap, the pump can operate with the second form of self-oscillation in the frequency control process at reduced head values, which requires a reduced frequency of rotation of the pump rotor (80.8 Hz or 4850 rpm), as well as short-term pump operation in the output mode at the nominal rotation frequency of 100 Hz (6000 rpm). In order to avoid such phenomena, a requirement has been expressed regarding the size of the gap in the sealing devices - not less than the nominal one. Limiting the imbalance in the pump's rotor parts is also proposed, per the API 610 / ISO 13709:2009 standard. A four-fold imbalance is adopted for the balancing accuracy class G 2.5 according to ISO 21940-11:2016.

## 5 Discussion

The results obtained from the research satisfy and confirm the effective use of non-contact seals in submersible pumps with inverter direct high-speed drives.

It can also be concluded that developing pumps with non-contact seals without bearing units has great prospects.

As can be seen from the appearance of the amplitude-frequency characteristic, dynamics and resonance phenomena are not pronounced, and the amplitudes of oscillations near the rotation frequency do not exceed 35% of the gaps in the seals, as required by API 610 / ISO 13709:2009, with a significant margin.

The first three natural frequencies of transverse oscillations of the rotor were determined, which were 34–44 Hz, respectively, 81–107 Hz and 512–516 Hz. Thus, the frequency of rotation (100 Hz) coincides with the frequency of transverse oscillations according to the second natural form (81–107 Hz). Unlike some rotary machines that do not allow operation in such conditions (e.g., electric motors, generators, gas turbines, and turbocompressors), this condition is not mandatory for centrifugal pumps. Radial forces in internal non-contact seals have a powerful damping component, which, under certain conditions, minimizes or eliminates resonance phenomena and makes it quite possible for the pump to operate at rotation frequencies close to its natural frequencies [20]. In the latest standard revisions, tuning from natural frequencies is no longer mandatory. In the case of insufficient tuning, according to this standard in the 11th edition, a further check for rotor vibration amplitudes is required, considering possible imbalances and damping in the seals.

Switching to the use of pumps without bearing units has several advantages. Firstly, the bearings negatively affect the pump's durability and require regular replacement and lubrication. Secondly, due to the absence of bearing brackets, the weight and dimensions of the pump are reduced, and its energy efficiency is increased while maintaining the vibration reliability of the rotor.

## 6 Conclusions

As a result of the calculations, the amplitudes of oscillations near the rotation frequency are from 12 % to 22 % and do not exceed 35 % of the gaps in the seals, as API 610 / ISO 13709:2009 requires to ensure the guaranteed vibration reliability of the pump.

The 1st mode shape corresponds to angular oscillations of the rotor as a solid body around a certain point in its central part (anti-phase precession of two parts of the rotor) at a frequency of 44.4 Hz, the 2nd – in phase precession of the entire rotor at a frequency of 89 Hz, and the 3rd – already bending vibrations at a frequency of 512 Hz, as in a conventional rotor on rigid supports.

Considering the results obtained when determining the amplitudes of forced oscillations and the frequencies and forms of natural oscillations of the rotor, it can be concluded that the dynamic stability of the rotor of a high-speed submersible pump without bearings is ensured. This pump design ensures reliable operation at frequencies from 0 to 120 Hz.

It has been proven that by increasing the frequency of the drive, it is possible to reduce the mass and dimensions of the pump by reducing the number of stages from 8 to 2 while increasing the efficiency by 5 %.

## 7 Acknowledgments

The study was conducted within the framework of the project "Development of design solutions and layout schemes of a parametric series of high-speed energy-efficient well pumps for the needs of enterprises in the field of critical infrastructure" (No. 2022.01/0096) of the National Research Foundation of Ukraine (NRFU) for the possibility of project implementation. The authors would like to thank the "International Association of Technological Development and Innovation" (IATDI) and the Public Union "Sumy Machine-Building Cluster of Energy Efficiency" (SMBCEEQ) for their support in conducting the study.

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