MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE
Sumy State University
Faculty of Technical Systems and Energy Efficient Technologies
Department of Applied Hydro-Aeromechanics

«Admitted»
Head of the Department
___________ Mykola SOTNIK
____ ___________ 2025

QUALIFICATION THESIS
for obtaining a master's degree

in specialty 131 "Applied Mechanics",
educational and professional program "Hydraulic machines, hydraulic drives and hydro-pneumatic automation")

on the topic: **Development of a pump for thermal power plants**

Group applicant ГМ.м-31an Xu Yifei

The qualification work contains the results of one's own research. The use of ideas, results and texts of other authors must be referenced to the appropriate source.

____________ Xu Yifei

Head
associate professor of Department of Applied Hydro-Aeromechanics,
PhD, associate professor Vitalii PANCHENKO ___________

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TASK
for the master's thesis for a student

Xu Yifei

1. Topic: **<u>Development of a pump for thermal power plants</u>**
approved by the university order dated 14.11.2024 №*1385-VI*

2. Deadline for student submission of completed work - <u>26.01.2025.</u>

3. Project input data:
pump parameters:
$Q_н$ = 790 m^3ph, $H_н$ = 185 m, $n_н$ = 1470 rpm.

4. Contents of the settlement and explanatory note (list of issues to be developed):

hydraulic calculations of pump flow section elements, electric motor selection calculations, end seal calculation, strength calculations, bearing selection calculations

4. List of graphic material (with precise indication of mandatory drawings):

impeller drawing (A1), pump assembly drawing (A1), theoretical impeller drawing (A1), guide device drawing (A1).

CALENDAR PLAN

№	Name of work stages	Deadline for completion of work stages	Note
1	General characteristics of pumps		
2	Choosing a pump design		
3	Description of the design of the selected pump		
4	Hydraulic calculations		
5	Theoretical drawing of the impeller		
6	Execution of the section "Labor protection"		
7	Implementation of the economic section		
8	Preparation of an internship report		
9	Calculations for selecting an electric motor		
10	Calculation of final compaction		
11	Strength calculations		
12	Bearing selection calculations		
13	Impeller drawing		
15	Pump assembly drawing		
17	Design of the RPP and graphic materials		
18	Presenting the work to the manager. Making amendments.		
19	Checking work for plagiarism.		
20	Time for preliminary defense. Preparation of the report for defense.		
21	Posting the work in the repository. Receiving a review.		
22	Defense of the work in the EC (according to the defense schedule).		The work is allowed to be defended after being checked for plagiarism.

Date of issue of the task - 01.11.2024.

Student ________________ Xu Yifei

Manager ________________ Vitalii Panchenko

Annotation

Explanatory note: 57 p., 7 figures, 5 tables, 35 references.

Thesis project topic «**Development of a pump for thermal power plants**»

Graphic materials: 6 sheets of A1 format.

The purpose of the project is to develop a vertical pump with centrifugal impellers, designed for pumping hot water condensate in thermal power plant systems to the parameters: $Q_{\text{н}}$ = 790 m^3ph, $H_{\text{н}}$ = 185 m, $n_{\text{н}}$ = 1470 rpm.

In accordance with the set goal, the following were done:

- the choice of the pump design scheme was justified;
- a description of the design was made;
- hydraulic calculations were made (centrifugal wheel calculation, guide device calculation);
- strength calculations were made: shaft, keyed joints;
- an electric motor was selected.

In the section on labor protection, standardization and control in the field of labor protection at an industrial enterprise were considered.

In the section on economics, a functional-cost analysis of new equipment was considered.

Keywords: PUMP, CONDENSATE, IMPELLER, DIRECTION DEVICE, HEAD, FEED.

Content

Introduction

Vertical condensate pumps [1] are used to transport water condensate in thermal power plants (TPPs). They play an important role in ensuring continuous circulation of the working circuit and increasing the efficiency of steam turbine equipment that generates electricity.Зазначені насоси мають наступні характеристики та особливості:

Vertical design:

Due to their compact design, the pumps can be installed in confined spaces, which is typical of thermal power plant engine rooms.

Vertical orientation helps reduce vibration and increase operational reliability.

Operate at low inlet pressure:

Designed for pumping low-pressure liquids, in particular condensate from the condenser of a thermal power plant steam turbine.

The special design ensures stable operation at low suction levels, minimizing the risk of cavitation in the flow section of the pump.

Corrosion resistance:

The working components are made of corrosion-resistant materials, which prevents their destruction under the influence of deaerated condensate with a low oxygen content.

High energy efficiency:

Thanks to the optimized hydraulic design of the impellers and other elements of the flow section, a high efficiency is achieved.

Often equipped with frequency control systems, which allows you to adapt the flow rate to changing operating conditions.

Reliability and durability:

Designed for continuous operation.

Modern sealing systems extend the service life of the equipment and reduce maintenance costs.

Multi-stage design:

To achieve the required head under low inlet pressure, pumps are often manufactured in multi-stage design.

Main advantages of vertical condensate pumps [1,3]:

Energy efficiency:

Optimized design reduces hydraulic losses, which allows achieving high pump efficiency.

Modern models are equipped with frequency control, which automatically adjusts the flow rate according to operating conditions, reducing energy consumption and operating costs.

Compactness and space optimization:

Vertical layout reduces the footprint, which is important for TPP machine rooms with limited space.

Convenient location provides flexibility in designing and upgrading hydraulic systems.

Stable operation at low pressure and temperature:

The pumps work effectively with deaerated condensate of low pressure and temperature, minimizing the risk of cavitation.

Special impeller design and precise hydraulic calculations contribute to the durability of the pumping equipment.

Corrosion resistance:

The use of materials such as stainless steel, nickel alloys and anti-corrosion coatings protects the pump from the aggressive effects of water condensate.

This significantly extends the service life of the equipment and reduces the frequency of repairs.

Durability and minimal operating costs:

The robust design and the use of modern sealing systems (e.g. mechanical seals) reduce the risk of leaks and the need for frequent maintenance.

The high wear resistance of components extends the repair intervals, reducing operating costs.

Resistance to water hammer:

The pumps are able to operate under conditions of sudden changes in pressure and flow without the risk of equipment failure.

This increases the overall reliability of the TPP hydraulic system.

Multi-stage design for high head:

To achieve the required pressure in condensate circulation systems, multi-stage designs are often used.

This allows you to effectively pump liquids over long distances or heights with minimal energy consumption.

Flexibility in operation:

The modular design makes it easy to adapt the pump to specific tasks.

Simple connection to modern automation systems provides effective process control and management.

Environmental safety:

Reliable sealing elements prevent leakage of the pumped liquid, reducing the risk of environmental pollution.

High energy efficiency helps reduce CO_2 emissions and improve the environmental performance of thermal power plants.

Ease of maintenance:

The well-thought-out design provides easy access to the main components for inspection, diagnostics and replacement of components.

This reduces equipment downtime and increases maintenance efficiency.

However, these pumps also have certain disadvantages, including [1, 2].

Difficulty of assembly and disassembly.

Execution of upright water-driven mechanisms demands precise alignment and stringent adherence to assembly protocols, elevating the duration and labor-intensiveness of the procedure. Due to the upward arrangement, reaching some parts, like turbines or bearings, poses challenges, hampering maintenance and substitution of these elements. Putting in these pumps can cost more money since they need a strong base or a deep hole to work in. Sensitivity to the quality of the working medium.

The presence of solids in the hydraulic fluid can lead to prompt impeller degradation, reducing efficacy and potentially causing pump malfunctions. High maintenance requirements.

To guarantee extended and dependable functioning, periodical servicing is essential, involving inspecting joints, wheels and blades. Specialized substances increase the cost and complexity of producing spare parts. Pump fixes necessitate total dismantling, causing extended inactivity and further upkeep expenses.

1. Selection and description of the selected design

1.1.1 Purpose and scope

Condensate vertical pumps move the liquid water left over from the power plant heater in sealed systems. They are crucial for steam generators and other applications where the water condenses and its yield needs to be recovered for further use. These pumps are really popular in many industries and energy sectors. They are great for moving the colder, water vapor that has turned back into water.

Thermal Power Plants (TPP) - provide water circulation in cycles -condensate cycles. Nuclear power projects are similar to nuclear power sites. They have cycles to keep the plants running. Heat Exchanger Numbers - use in systems to settle condensate from steam generators. Companies in the chemical, petrochemical and paper industries use condensation by sending water back to boilers or heat exchangers. Centralized Heat Generation Networks - circulation of fluid after thermal transfer in the heating apparatus. Shipyards manage steam and heat on board with heat and steam circuits. Cogeneration equipment provides hot water circulation in places where electricity and heat are produced simultaneously.

1.1.2. Device and principle of operation.

The designed pump [13] is centrifugal, sectional type, double-case, vertical, single-flow and multistage. Its outer casing is a welded structure with welded nozzles that provide connection to the suction and pressure pipelines. The inner casing is made in such a way that it can be easily dismantled.

The main components of the pump:

rotor,

stator parts,

final seals (stuffing or face type),

bearings,

electric motor.

The rotor is supported by two supports:

the lower one is a sliding bearing,

the upper one is external rolling bearings lubricated with crankcase or plastic grease.

The axial load on the rotor is reduced hydraulically by using an unloading drum or a hydraulic heel. The pump shaft can have two types of end seals: a face seal (-T) or a gland seal (-C), which are interchangeable.

Working principle

A vertical condensate pump converts the mechanical energy of a rotating impeller into the kinetic and potential energy of the liquid.

The condensate enters the pump through a suction pipe located at the bottom.

The liquid is then directed into a screw impeller, which feeds it into a centrifugal impeller.

The centrifugal impeller rotates at high speed, creating centrifugal force that accelerates the liquid and carries it to the periphery of the impeller.

After that, the liquid enters a guide device that directs the flow to the next impeller.

Passing through all stages of the pump, the liquid reaches the last impeller, after which it enters the pressure pipe for further transportation.

1.1.3. Description of the pump design.

The pump [1] is a centrifugal, vertical, three-stage, including three centrifugal stages and a worm wheel installed in front of them.

Main structural elements of the pump

The pump consists of:

housing parts,

bearing assembly,

impellers,

guide devices,

transfer tubes.

Impellers are the main elements of the pump that generate centrifugal force and transfer energy to the liquid from the rotor. They have the spatial shape of blades, are manufactured by casting with subsequent mechanical processing. Together with the guide devices, they form the working stage of the pump.

The bearing assembly provides support for the shaft and its stable rotation. The pump is equipped with:

radial plain bearings operating on the pumped liquid,

radial-axial ball bearings.

The residual axial force is compensated by an unloading drum.

The pump body parts include:

sections,

guide devices,

suction and discharge covers.

The guide devices located in the sections perform the function of converting the kinetic energy of the liquid into pressure energy, and also ensure a smooth flow direction. To prevent rotation, they are fixed with cylindrical pins.

The pump is driven by an electric motor, which converts electrical energy into mechanical energy and ensures the operation of the unit.

2.Hydraulic calculation

2.1 Calculation and design of a centrifugal impeller

The calculation is performed according to the recommendations [10]

1. Determination of the main geometric parameters.

The meridian section of the impeller with the main geometric parameters is shown in Fig. 2.1.

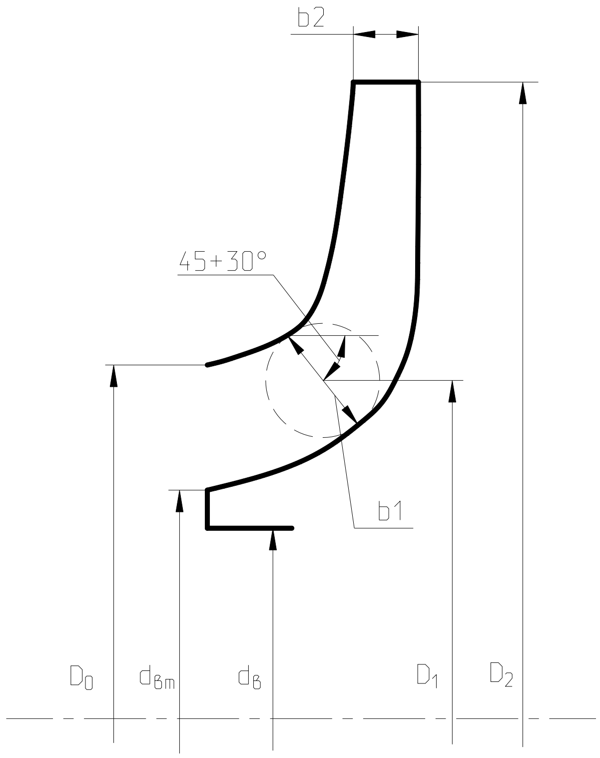

Figure 2.1. Meridian cross-section of the flow part of the centrifugal pump impeller

Speed coefficient:

$$n_s = \frac{3{,}65 \cdot n \cdot \sqrt{Q}}{\left(\frac{H}{i}\right)^{0,75}}, \tag{2.1}$$

$$n_s = \frac{3{,}65 \cdot 1470 \cdot \sqrt{790}}{60 \cdot (185/3)^{0,75}} = 114$$

The given impeller diameter is determined by the Sukhanov formula:

$$D_{1пр} = K_{вх}\sqrt[3]{\frac{Q}{n}}, \quad (2.2)$$

$K_{вх}$=3,5 ÷ 5,0 – coefficient of the inlet funnel of the impeller. We accept $K_{вх}$=4,5.

$$D_{1пр} = 4,5 \cdot \sqrt[3]{\frac{790}{3600 \cdot 1470}} = 0.239 m$$

Total pump efficiency:

$$\eta = \eta_о \cdot \eta_г \cdot \eta_{мех'} \cdot \eta_{мех}, \quad (2.3)$$

$\eta_о$ – volumetric efficiency of the pump;

$\eta_г$ – hydraulic efficiency;

$\eta_{мех'}$ – internal mechanical efficiency;

$\eta_{мех}$ – external mechanical efficiency.

Volumetric efficiency:

$$\eta_о = \frac{1}{1 + 0,68 \cdot n_s^{-2/3}}, \quad (2.4)$$

$$\eta_о = \frac{1}{1 + 0,68 \cdot 114^{-2/3}} = 0.971$$

Hydraulic efficiency:

$$\eta_г = 1 - \frac{0,42}{\left(\lg D_{1пр} - 0,172\right)^2}, \tag{2.5}$$

$D_{1пр}$ – given impeller diameter, mm.

$$\eta_г = 1 - \frac{0,42}{\left(\lg 239 - 0,172\right)^2} = 0.914$$

Internal mechanical efficiency:

$$\eta_{мех'} = \frac{1}{1 + 820 \cdot n_s^{-2}} \tag{2.6}$$

$$\eta_{мех'} = \frac{1}{1 + 820 \cdot 114^{-2}} = 0.941$$

We define the external mechanical efficiency as: $\eta_{мех}$=0,98.

$$\eta = 0,971 \cdot 0,914 \cdot 0,98 \cdot 0,941 = 0.818$$

Power consumed by the pump:

$$N = \frac{\rho_{\max} \cdot g \cdot Q_н \cdot H_н}{\eta}; \tag{2.7}$$

$$N = \frac{970 \cdot 9,81 \cdot 790 \cdot 185}{3600 \cdot 0,818} = 472263 W$$

Pump impeller feed

$$Q_{рк} = \frac{Q}{\eta_o} \tag{2.8}$$

$$Q_{рк} = \frac{790}{3600 \cdot 0,971}$$ =0.226 m^3/s

Theoretical pump impeller head (single stage)

$$H_T = \frac{H}{\eta_г}, \tag{2.9}$$

$$H_T = \frac{61,67}{0,914}$$ =67.47 m

$$d_в = \sqrt[3]{\frac{16M}{\pi \cdot [\tau]}}, \tag{2.10}$$

M - torque on the pump shaft, N•m;

[τ] – reduced allowable torsional stress, N/m^2.

3. Determining the diameter of the pump impeller shaft and bushing

Torque:

$$M = \frac{30N_{\max}}{\pi \cdot n}, \tag{2.11}$$

N_{max} - maximum power, W. N_{max} =1,1×N=1,1×472,3=519,5 kW.

$$M = \frac{30 \cdot 519,5 \cdot 10^3}{3,14 \cdot 1470}$$ =3376 N•m

We accept the reduced allowable stress [τ]=$80 \cdot 10^5$ N/m^2

$$d_в = \sqrt[3]{\frac{16 \cdot 3376}{3{,}14 \cdot 80 \cdot 10^5}} = 0.129 \text{ m}$$

Taking into account the design features and specifics of the pump assembly, as well as the shaft calculations for the critical speed, we choose - $d_в$ =0,130 (m).

The diameter of the bushing is chosen approximately:

$$d_{вт} = (1.2 \div 1.25) d_в \qquad (2.12)$$

$$d_{вт} = 1{,}25 \cdot 0{,}130 = 0.163 \text{ m}$$

We accept $d_{вт}$ =0.16 m

4. Determination of the geometric parameters of the impeller inlet

The diameter of the inlet funnel of the impeller is determined from the expression:

$$D_{1пр} = \sqrt{D_0^2 - d_{вт}^2} \qquad (2.13)$$

$$D_0 = \sqrt{D_{1пр}^2 + d_{вт}^2} \qquad (2.14)$$

$$D_0 = \sqrt{0{,}239^2 + 0{,}16^2} = 0.288 \text{ m}$$

flow velocity at the impeller inlet

$$\upsilon_0 = \frac{4Q_{рк}}{\pi \cdot D_{1пр}^2}, \qquad (2.15)$$

$Q_{рк}$ – impeller flow, m³/s.

$$\upsilon_0 = \frac{4 \cdot 0,226}{3,14 \cdot 0,288^2} = 3.47 \text{ m/s}$$

We set the position of the leading edge of the pump blade:

$$D_1 = 0,9 \cdot D_0 \qquad (2.16)$$

$$D_1 = 0,9 \cdot 0,288 = 0.259 \text{ m}$$

In the first approximation, the meridian component of the velocity υ_{1m}', without taking into account the compression of the flow by the blades, is taken to be equal to υ_0. When the flow enters the blades, the meridian component of the absolute velocity increases and is determined from the expression:

$$\upsilon_{1m} = \psi_1 \cdot \upsilon_{1m}', \qquad (2.17)$$

ψ_1 = 1,15..1,3 – the compression ratio of the flow at the inlet.
We accept $\psi_1 = 1,15$.

$$\upsilon_{1m} = 1,15 \cdot 3,47 = 3.99 \text{ m/s}$$

The flow angle at the inlet to the blade is calculated from the velocity triangle:

$$\beta_{1n} = arctg \frac{\upsilon_{1m}}{U_1 - \upsilon_{u1}}, \qquad (2.18)$$

U_1 - transfer velocity at the inlet to the impeller, m/s;

υ_{1u} - circular component of absolute velocity at the inlet, m/s.

Circular speed at the impeller inlet:

$$U_1 = \frac{\pi \cdot D_1 \cdot n}{60} \tag{2.19}$$

$$U_1 = \frac{3{,}14 \cdot 0{,}259 \cdot 1470}{60} = 19.92 \text{ m/s}$$

Circular component of absolute velocity at the inlet

$$\upsilon_{1u} = \frac{0{,}12 \cdot \sqrt[3]{Q_{рк}^2 \cdot n}}{D_1} \tag{2.20}$$

$$\upsilon_{1u} = 0$$

$$\beta_{1n} = arctg \frac{3{,}99}{19{,}92} = 11.33^o$$

The angle of inclination of the blade at the inlet to the pump impeller:

$$\beta_1 = \beta_{1n} + \Delta\beta$$

$\Delta\beta = 3..8°$ - angle of attack.

The angle of attack is introduced to reduce hydraulic losses in the impeller area and improve its resistance to cavitation, we choose $\Delta\beta=5^o$.

$$\beta_1 = 11{,}33^o + 5^o = 16.33^o.$$

5. Determining the number of blades and their thickness.

Most centrifugal pumps of various sizes, which are produced and are characterized by high technical and economic indicators, have the number of blades z = 5..8.

The thickness of the impeller blade is determined taking into account technological requirements, which depend on the material of the wheel, its dimensions, and the capabilities of the foundry. As a rough guide, the blade thickness at the impeller inlet can be assumed to be S1 = 2..10 mm with D2 = 150 - 500 mm. The thickness of the blade at the outlet of the impeller S2 is often taken equal to S1, and towards the middle the blade gradually thickens.

Based on the above, we assume the blade thickness S1=S2= 8 mm, and the number of blades z = 7.

2.1.5. Clarification of the flow compression ratio.

The value of the flow compression ratio is specified by the formula:

$$\psi_1' = \frac{t_1}{t_1 - S_1 / \sin\beta_1}, \qquad (2.21)$$

$t_1 = \dfrac{\pi \cdot D_1}{z}$ - blade pitch at the impeller inlet.

$$t_1 = \frac{3{,}14 \cdot 0{,}259}{7} = 0.116 \text{ m}$$

$$\psi_1' = \frac{0{,}116}{0{,}116 - \dfrac{0{,}009}{\sin 16{,}33^\circ}} = 0.381$$

The condition must be met. $\left|\psi_1' - \psi_1\right| \le 0{,}01 \cdot$

$|1,381 - 1,15| = 0,231 > 0,01.$

Since the condition is not met, we make a second approximation, we set $\psi_1'' = \psi_1' =$ =1.381, then:

$$\upsilon_{1m} = 1,381 \cdot 3,47 = 4.79 \text{ m/s}$$

$$\beta_{1n} = arctg\frac{4,79}{19,92} = 13.52^{o}$$

$$\beta_1 = 13,52^\circ + 5^\circ = 18.52^{o}$$

$$\psi_1'' = \frac{0,116}{0,116 - \dfrac{0,009}{\sin 18,52^\circ}} = 1.373$$

$|1,373 - 1,381| = 0,008.$ The condition is met.

From the continuity equation, we find the width of the impeller at the inlet:

$$b_1 = \frac{Q_{p\kappa}}{\pi \cdot D_1 \cdot \upsilon_{1m}} \qquad (2.22)$$

$$b_1 = \frac{0,226}{3,14 \cdot 0,259 \cdot 4,79} = 0.058 \text{ m}$$

6. Determination of the geometric parameters of the impeller outlet

The preliminary value of the outer diameter of the impeller (with an infinite number of blades z) can be determined in a simplified way from the basic equation of operation of Euler vane pumps for $\upsilon_{1u} = 0$ and $\upsilon_{2u} = 0,5 \cdot U_2$:

$$D_2^{'} = 19{,}68 \cdot \frac{\sqrt{2 \cdot g \cdot \mathrm{H}_{\text{рк}}}}{\mathrm{n}}, \; (n_s > 100), \qquad (2.23)$$

$\mathrm{H}_{\text{рк}}$ – head of the impeller.

$$H_{рк} = \frac{H}{i} \qquad (2.24)$$

$$D_2^{'} = 19,68 \cdot \frac{\sqrt{2 \cdot 9,81 \cdot 61,67}}{1470} = 0.466 \text{ m}$$

Meridian velocity without taking into account compression at the impeller outlet

$$\upsilon_{2m}^{'} = (0{,}5 \div 1{,}0)\upsilon_{1m}^{'} \qquad (2.25)$$

$$\upsilon_{2m}^{'} = 0,52 \cdot 3,47 = 1.8 \text{ m/s}$$

Impeller outlet angle β_2 determined from the velocity triangle depending on the accepted value $\upsilon_{2m}^{'}$ so as to obtain the desired ratio $\frac{W_1}{W_{2\infty}}$.

From the velocity triangle:

$$W_{2\infty} = \frac{\upsilon_{2m}}{\sin\beta_2} = \frac{\psi_2 \cdot \upsilon_{2m}^{'}}{\sin\beta_2}, \qquad (2.26)$$

ψ_2 - compression ratio of the flow cross-section by the blades at the impeller outlet, ψ_2 = 1,05÷1,1. We accept ψ_2 = 1,1.

Meridian speed at the impeller outlet:

$$\upsilon_{2m} = \psi_2 \cdot \upsilon_{2m}^{'} \qquad (2.27)$$

$\upsilon_{2m} = 1{,}1 \cdot 1{,}8$ =1.98 m/s

Relative velocity at the impeller inlet from the velocity triangle:

$$W_1 = \frac{\upsilon_{1m}}{\sin\beta_1} = \frac{\psi \cdot \upsilon_{1m}^{'}}{\sin\beta_1} \qquad (2.28)$$

Relation $\frac{W_1}{W_{2\infty}}$ depends on n_s and is determined from the expression:

$$\frac{W_1}{W_{2\infty}} = 3{,}7 - 0{,}054 \cdot n_s + 4{,}0 \cdot 10^{-4} \cdot n_s^2 - 0{,}98 \cdot 10^{-6} \cdot n_s^3 \qquad (2.29)$$

$$\frac{W_1}{W_{2\infty}} = 3{,}7 - 0{,}054 \cdot 114 + 4 \cdot 10^{-4} \cdot 114^2 - 0{,}98 \cdot 10^{-6} \cdot 114^3 = 1.29$$

From the equation for $W_{2\infty}$ after transformation, the following expression is obtained for determining the angle β_2:

$$\beta_2 = \arcsin\left(\frac{W_1}{W_{2\infty}} \cdot \frac{\psi_2}{\psi_1} \cdot \frac{\upsilon_{2m}^{'}}{\upsilon_{1\mathrm{m}}^{'}} \cdot \sin\beta_1\right) \qquad (2.30)$$

$$\beta_2 = \arcsin\left(1{,}29 \cdot \frac{1{,}1}{1{,}373} \cdot \frac{1{,}75}{3{,}36} \cdot \sin 18{,}52^\circ\right) = 16.43^\circ$$

7. Clarification of the outer diameter of the impeller taking into account the finite number of blades. Circular speed at the outlet of the impeller:

$$U_2 = \frac{\upsilon_{2m}}{2tg\beta_2} + \sqrt{\frac{\upsilon_{2m}^2}{\left(2tg\beta_2\right)^2} + g \cdot H_{T\infty} + \upsilon_{1u} \cdot U_1}, \qquad (2.31)$$

$H_{T\infty}$ - theoretical pump head taking into account the finite number of blades.

$$H_{T\infty} = \frac{H_{рк}}{\eta_г \cdot K_z}, \qquad (2.32)$$

K_z - correction for the finite number of blades.

Correction for the finite number of blades K_z we will determine by the Pfleiderer formula:

$$K_z = \frac{1}{1+p} \qquad (2.33)$$

де $$p = 2 \cdot \frac{\psi}{z} \cdot \frac{1}{1-\left(D_1/D_2\right)^2} \qquad (2.34)$$

Coefficient ψ is determined depending on n_s, by n_s < 150:

$$\psi = \left(0{,}55 \div 0{,}65\right) + 0{,}6 \sin\beta_2 \qquad (2.35)$$

$$\psi = \left(0{,}55 \div 0{,}65\right) + 0{,}6 \sin 18{,}52^\circ = 0{,}741 \div 0{,}841$$

We accept ψ = 0,795.

$$p = 2 \cdot \frac{0{,}795}{7} \cdot \frac{1}{1-\left(\frac{0{,}259}{0{,}466}\right)^2} = 0.329$$

$$K_z = \frac{1}{1+0,329} = 0.752$$

$$H_{T\infty} = \frac{61,67}{0,914 \cdot 0,752} = 89.72 \text{ m}$$

$$U_2 = \frac{1,98}{2tg18,52^\circ} + \sqrt{\frac{1,98^2}{\left(2tg18,52^\circ\right)^2} + 9,81 \cdot 89,72} = 32.77 \text{ m/s}$$

$$D_2 = \frac{60U_2}{\pi \cdot n} \tag{2.36}$$

$$D_2 = \frac{60 \cdot 32,77}{3,14 \cdot 1470} = 0.426 \text{ m}$$

8. Clarification of the flow compression ratio ψ_2. The flow compression ratio at the impeller outlet is specified by the formula:

$$\psi_2^{'} = \frac{1}{1 - \dfrac{z \cdot S_2}{\pi \cdot D_2 \sin \beta_2}} \tag{2.37}$$

$$\psi_2^{'} = \frac{1}{1 - \dfrac{7 \cdot 0,009}{3,14 \cdot 0,426 \cdot \sin 18,52^\circ}} = 1.174$$

The condition must be met. $\left|\psi_2^{'} - \psi_2\right| \leq 0,01 \cdot$

|1,174 - 1,1| = 0,074 < 0,1. The condition is met.

9. Determining the width of the impeller at the outlet. The width of the impeller at the outlet is determined from the continuity equation:

$$b_2 = \frac{Q_{рк}}{\pi \cdot D_2 \cdot \upsilon_{2m}} \qquad (2.38)$$

$$b_2 = \frac{0,226}{3,14 \cdot 0,426 \cdot 1,98} = 0.085 \text{ m}$$

2.2 Calculation of the guiding device

Input data

Stage submission	$Q=790\ m^3/h$;
Stage pressure	$H=61{,}67$ m;
Impeller outer diameter	$D_2=0{,}426$ m;
Impeller outlet width	$b_2=0{,}085$ m;
Circular component of velocity flow at the impeller outlet	$V_{u2}=16{,}39$ m/s;
impeller speed	n=1470 r/m.

The calculation is performed according to the recommendations [10]

Diameter of the initial circle:

$$D_3 = 1{,}06 \cdot D_2$$
$$D_3 = 1{,}06 \cdot 0{,}426 = 0.452 \text{ m.}$$

Width of the guiding device in the meridian section:

$$b_3 = 1{,}1 \cdot b_2 + 1.5$$
$$b_3 = 1{,}1 \cdot 85 + 1{,}5 = 95 \text{ mm.}$$

We choose the number of blades of the guide device zна depending on the number of blades of the impeller from the condition of the absence of unbalanced force P and unbalanced pressure pulsations p0.

At zрк=7, unbalanced pressure pulsation forces are absent at zна=9 and zна=12. For technological reasons, a smaller number of blades zна=9 is taken.

We determine the angle of installation of the blade at the inlet:

We preliminarily set the thickness of the blade at the inlet in the first approximation δ=3mm, and the angle at the inlet αзл=50.

The peripheral speed at the inlet in the n.a.:

$$V_{u3} = V_{u2} \frac{D_2}{D_3};$$
$$V_{u3} = 16{,}39 \frac{0{,}426}{0{,}452} = 15.45 \text{ m/s;}$$

Meridian speed:

$$V_{m3} = \frac{Q}{\pi \cdot D_3 \cdot b_3};$$

$$V_{m3} = \frac{790}{3600 \cdot \pi \cdot 0,452 \cdot 0,095} = 1.63 \text{ m/s}.$$

Blade installation angle:

$$\alpha_{зл} = arctg \frac{V_{m3}}{\left(1 - \frac{\delta \cdot z_{на}}{\sin \alpha_{зл} \cdot \pi \cdot D_3}\right) \cdot V_{u3}};$$

$$\alpha_{зл} = arctg \frac{1,63}{\left(1 - \frac{0,003 \cdot 9}{\sin 5^0 \cdot \pi \cdot 0,452}\right) \cdot 15,45} = 7.69^{o}.$$

$$tg\alpha_{зл} = \mu \cdot tg\alpha_{зл};$$

$$tg\alpha_{зл} = 1,4 \cdot tg7,69 = 10.7^{o}.$$

We design a spiral section:

$$R_3^{'} = R_3 e^{\frac{2\pi}{z_{íà}} tg\alpha_{çë}};$$

$$R_3^{'} = \frac{0,452}{2} e^{\frac{2\pi}{9} tg10,7} = 0.258 \text{ m}.$$

Inlet height:

$$a_3 = \left(R_3^{'} - R_3\right) \cos \alpha_{3л} - \delta;$$

$$a_3 = \left(0,258 - 0,226\right) \cos 10,7 - 0,003 = 0.0284 \text{ m}.$$

Determine the dimensions of the diffuser channel:

Diffuser channel length:

$$l_{диф} = 4,9 a_3;$$

$$l_{диф} = 4,9 \cdot 0,0284 = 0.139 \text{ m};$$

Diffuser channel inlet area:

$$F_4 = 2.8 \cdot F_3;$$

$$F_4 = 2,8 \cdot 0,095 \cdot 0,0284 = 0.00755 \text{ m}^2;$$

Equivalent diffuser expansion angle:

$$\psi_{экв} = 2 arctg \frac{\sqrt{\frac{F_4}{\pi}} - \sqrt{\frac{F_3}{\pi}}}{l_{диф}};$$

$$\psi_{экв} = 2arctg\frac{\sqrt{\frac{0,00755}{\pi}} - \sqrt{\frac{0,0027}{\pi}}}{0,139} = 16.14^{o}.$$

We select the number of return channel blades: $z_{пк} = z_{на} = 9$.

The installation angle of the return channel blades is accepted $\alpha_{зк} = 84^{o}$

3. Strength calculations

3.1 Forces acting on the pump rotor

The calculation is performed according to the recommendations [10]

1. Determination of the axial force acting on the pump rotor

The axial hydraulic force consists of the sum of unbalanced forces acting on the pump rotor in the axial direction.

The total value of the axial force is:

$$T_{oc} = T_1 \pm (T_1^{1*} - T_2) + T_{шн} - T_б + T_p$$

T_1 – axial force acting in the suction direction at one stage.

T_1^{1*} - axial force is directed towards the worn seal

T_2 – dynamic force acting along the rotor axis

$T_{шн}$ – axial force acting in the suction direction on the screw

$T_б$ – axial force acting in the direction of discharge on the drum

T_p – axial force resulting from the weight of the rotor.

$$T_1 = \int_{r_{y1}}^{r_{y2}} 2\pi dr \Delta p_i = \rho g \pi (r_{y1}^2 - r_{y2}^2)\left[H_p - \frac{\omega^2}{8g}\left(r_2^2 - \frac{r_{y1}^2 + r_{y2}^2}{2}\right)\right]$$

=19897N

$$T_1 = 970 \cdot 9{,}81 \cdot 3{,}14 \cdot (0{,}151^2 - 0{,}078^2)\left[62{,}17 - \frac{151{,}8^2}{8 \cdot 9{,}81}\left(0{,}2075^2 - \frac{0{,}151^2 + 0{,}078^2}{2}\right)\right]$$

During emergency seal wear, additional axial force occurs T_1*, directed towards the suction side.

This force will be equal to [10]:

$$T_1^* = \pi \cdot \left(r_2^2 \cdot r_{y1}^2\right) \cdot \gamma \cdot \frac{u_2^2}{8 \cdot g} \cdot \left(\frac{r_2^2}{r_2^2 - r_{y1}^2} \cdot \ln\frac{r_2^2}{r_{y1}^2} + \frac{r_2^2 + r_{y1}^2}{2 \cdot r_2^2} - 2\right)$$

From the impeller drawing: r_{y1} = 0,148 м; r_2 = 0,081 м.

Circular speed at the impeller outlet: U_2 =32.77 m/s·

$$T_1'^{*} = 3{,}14 \cdot \left(0{,}213^2 - 0{,}148^2\right) \cdot \frac{970 \cdot 32{,}77^2}{8} \times$$

$$\times \left(\frac{0{,}213^2}{0{,}148^2 - 0{,}081^2} \cdot \ln \frac{0{,}213^2}{0{,}081^2} + \frac{0{,}213^2 + 0{,}148^2}{2 \cdot 0{,}213^2} - 2 \right) =$$

=42781N

Along the pump axis, there is also a dynamic force T2, caused by the inflow of the flow, as well as by the change in the axial direction of its movement to the radial one. The force T2 is equal to [1]:

$$T_2 = B \cdot \frac{\gamma \cdot Q}{g} \cdot v_0,$$

B = 1 - for radial wheels;

v_0 – impeller inlet speed, м/с, v_0 = 3,47 м/с

$$T_2 = \frac{970 \cdot 790}{3600} \cdot 3{,}47 = 738 \text{ N}$$

axial force acting in the suction direction on the screw

$$T_{шн} = S \cdot \Delta p$$

$$T_{шн} = \pi (R^2 - r^2) \cdot \Delta p$$

$$T_{шн} = \pi (0{,}138^2 - 0{,}071^2) \cdot 15381 = 676 \text{ N}$$

R, r – outer radii, respectively, and the inlet radius of the screw, m

Δp - pressure drop across the auger.

Axial force acting in the direction of discharge on the drum

$T_б = S \cdot \Delta p$

S – drum area.

Δp - pressure drop across the drum.

$$T_б = \pi (R_б{}^2 - r_б{}^2) \cdot \Delta p_б$$

$$T_б = \pi (0{,}129^2 - 0{,}062^2) \cdot 1513287 = 60808 \text{ N}$$

axial force acting in the suction direction from the weight of the rotor.

$T_p = mg$

m=413kg - rotor mass

$T_p = 413 \cdot 9,81 = 4052 N.$

Maximum axial force acting on the impeller:

$$T_{ос} = T_1 \pm (T_1^{1*} - T_2) + T_{шн} - T_{б} + T_p$$

$$T_{oc} = 3 \cdot 19897 + (42781 - 738) + 676 - 60808 + 4052 = 15654\ N$$

2. Definition of radial force

To determine the radial force in a centrifugal pump, we use the formula [3]:

$$R = K_R \cdot \left(1 - \left(\frac{Q}{Q_{опт}}\right)^2\right) \cdot \rho \cdot g \cdot H \cdot D_2 \cdot b_2,$$

K_R – dimensionless radial force coefficient;

D_2 – outer diameter of the impeller, D_2 = 0,426 m;

b_2 – the width of the wheel at the exit, which includes the thickness of its rims,

Coefficient K_R depends on n_s. At n_s = 114 K_R = 0,489.

Maximum power will be in mode Q = 0.

$R = 0,489 \cdot 1 \cdot 970 \cdot 9,81 \cdot 185 \cdot 0,426 \cdot 0,094 = 34471$ N

3.2 Shaft strength calculation

The calculation is performed according to the recommendations [10]

1. The shaft strength calculation is carried out at maximum power.

The shaft static power calculation allows you to determine the safety margins and compare them with the minimum permissible values.

Shaft material – Steel 45 DSTU 1050.

Material characteristics.

– ultimate strength $\sigma_y = 600\,MPa$;

– yield strength $\sigma_T = 350\,MPa$;

– bending fatigue limit $\sigma_{-1} = 270\,MPa$;

– torsional fatigue limit $\tau_{-1} = 160\,MPa$.

Pump power:

$$N = \frac{\rho_{\max} \cdot g \cdot Q_н \cdot H_н}{\eta};$$

$$N = \frac{970 \cdot 9{,}81 \cdot 790 \cdot 185}{3600 \cdot 0{,}818} = 472263\text{W}$$

Shaft torque,

$$M = \frac{30 N_{\max}}{\pi \cdot n},$$

N_{max} - maximum power, Вт. N_{max} =1,1×N=1,1×472.3=519.5 kW.

$$M = \frac{30 \cdot 519{,}5 \cdot 10^3}{3{,}14 \cdot 1470} = 3376\ \text{N·m}$$

We accept the reduced allowable stress $[\tau]=190 \cdot 10^5$ Н/м^2

$$d_в = \sqrt[3]{\frac{16 \cdot 3376}{3{,}14 \cdot 80 \cdot 10^5}} = 0.129 \text{ m}$$

2. We determine the bending moment:

$$M_{зг} = 0{,}1 \cdot M_{кр},$$

$$M_{зг} = 0{,}1 \cdot 3376 = 338 \text{ N·m}$$

3. We determine the moment of torsional resistance taking into account the weakening by the keyway:

$$W_{кр} = \frac{\pi \cdot d^3}{16} - \frac{b \cdot t_1 \cdot (d - t_1)^2}{2 \cdot d},$$

d – shaft diameter, under the key, m;

b – key width, m;

h – key height, m.

$$W_{кр} = \frac{3{,}14 \cdot 0{,}129^3}{16} - \frac{0{,}01 \cdot 0{,}006 \cdot \left(0{,}12 - 0{,}006\right)^2}{2 \cdot 0{,}129} = 418 \cdot 10^{-6}.$$

Bending moment of resistance:

$$W_{изг} = \frac{\pi \cdot d^3}{32} - \frac{b \cdot t_1 \cdot (d - t_1)^2}{2 \cdot d},$$

$$W_{зг} = \frac{3{,}14 \cdot 0{,}129^3}{32} - \frac{0{,}001 \cdot 0{,}006 \cdot \left(0{,}12 - 0{,}006\right)^2}{2 \cdot 0{,}129} = 208 \cdot 10^{-6}.$$

4. We determine the tangential stress:

$$\tau_{кр} = \frac{M_{кр}}{W_{кр}},$$

$$\tau_{кр} = \frac{3376}{418} = 8.08 \text{ MPa}$$

5. We determine the normal stress:

$$\sigma_{зг} = \frac{M_{зг}}{W_{зг}},$$

$$\sigma_{зг} = \frac{338}{208} = 1.625 \text{ MPa}$$

Equivalent stress:

$$\sigma_{екв} = \sqrt{\sigma_{зг}^2 + 3 \cdot \tau_{кр}^2},$$

$$\sigma_{екв} = \sqrt{(1,625 \cdot 10^6)^2 + 3 \cdot (8,08 \cdot 10^6)^2} = 14.09 \text{ MPa}$$

Static strength margin:

$$n = \frac{\sigma_m}{\sigma_{екв}},$$

$$n = \frac{350 \cdot 10^6}{14,09 \cdot 10^6} = 24.8$$

In relation to: $\frac{\sigma_m}{\sigma_в} = \frac{350 \cdot 10^6}{600 \cdot 10^6} = 0,58$, minimum allowable value

$$[n]_{\min} = 1,4 \div 1,6.$$

Thus, the static strength margin is ensured, since $[n]_{\min} < n$.

The diameters of the shaft under the wheel, bearings, and auger are selected based on the design parameters, provided that it is larger than the calculated one.

3.3 Calculation of the keyed connection of the shaft with the wheel

We perform the calculation according to the recommendations [31]

Initial data for calculation.

Shaft material - Сталь 40Х.

Yield point - $\sigma_{0,2}$ = 750 МПа.

Key material - Сталь 45.

Yield point - $\sigma_{0,2}$ = 345 МПа.

Wheel material - 20Х13Л.

Yield point - $\sigma_{0,2}$ = 435 МПа.

Shaft torque $M_{кр}$ = 2911 Нм.

Key size under impeller, mm b x h x l = 18 x 9 x 62.

When calculating the keyed connection of the shaft with the wheel, the crushing stress is the determining factor.

$$\sigma_{ЗМ} = \frac{2 \cdot M_{КР}}{d \cdot l_p \cdot (h - t_1)},$$

l_p – working length of the key;

t_1 – shaft groove depth;

h - key height;

d - shaft diameter.

d =126 mm, $t_1 = 4,5$ =4.5 mm, $h =$ =9 mm.

$$\sigma_{ЗМ} = \frac{2 \cdot 3376}{0,126 \cdot 0,062 \cdot (0,009 - 0,0045)} \text{=101 MPa}.$$

The allowable crushing stress is calculated for the key material that has the lowest yield strength.

Allowable crushing stress:

$[\sigma_{ЗМ}] = 0,56 \cdot \sigma_{0.2}$;

$[\sigma_{ЗМ}] = 0,56 \cdot 350$ =196 MPa;

The strength condition is fulfilled.

3.4 Bearing durability calculation

We perform the calculation according to the recommendations [31]

We preliminarily assume that the supports are two angular contact ball bearings of the heavy narrow series 52315 DSTU 831, with the following parameters:: d=75 мм; D=162 мм;

- dynamic load capacity – C=151 кН;
- static load capacity – C_o=96,5 кН.

$$\frac{F_a}{V \cdot F_r} = \frac{15654}{1 \cdot 34471} = 0.454$$

e = 0,8

$$\frac{F_a}{V \cdot F_r} \lhd 0{,}8$$

We determine the equivalent load:

$$P_E = V \cdot F_r \cdot K_б \cdot K_T,$$

V – rotation ratio, V=1 during rotation of the inner ring of the bearing relative to the direction of the radial load;

F_r – radial load, F_r=R/2=17236 Н;

$K_б$ – safety factor, $K_б$=1;

K_T – temperature coefficient, K_T=1,1.

$$P_E = 1 \cdot 17236 \cdot 1 \cdot 1{,}1 = 18960\text{N}$$

$$L = \frac{10^6}{60n}\left(\frac{C}{P}\right)$$

$$L = \frac{10^6}{60 \cdot 1470}\left(\frac{151000}{18960}\right)^3 = 41381\text{h}$$

The recommended estimated durability value for machines used around the clock, in particular for pumps, corresponds to 30,000 hours, and the established resource before overhaul is 20,000 hours. Therefore, the condition is met.

4 Calculation of seals

We perform the calculation according to the recommendations [16]

In this calculation, it is necessary to determine the pressure drops in the front and interstage seals of the impeller, as well as the amount of leakage.
Leakage in seals:

$$q = \mu \cdot f \sqrt{\frac{2p}{\rho}}$$

μ - gap flow rate, μ =0,5;
f - gap area, $f = 2\pi r_1 \delta$;
p – pressure drop across the gaps.
Pressure drop across the impeller front seal: Pp=ψ×$H_{пот}$ - Pп ,

ψ - specific gravity of the liquid, N/m^3;
$H_{пот}$ - potential head, m.

$$H_{пот} = H \cdot \left(1 - \frac{g \cdot H}{2 \cdot \omega^2 \cdot R_2}\right),$$

H - pressure of the stage, m.
g – acceleration of free fall;
ω - rotor rotation frequency, rad/s;
R_2 – impeller radius, m.

$$H_{пот} = 56,7 \cdot \left(1 - \frac{9,81 \cdot 61,67}{153,9^2 \cdot 0,213}\right) = 49.9 \text{ m}$$

We determine the static pressure drop in the sinus of the front seal of the first stage:

$$P_П = \rho \cdot \omega^2 \cdot \frac{R_2^2 - R_1^2}{8},$$

R_1 – front seal diameter, m;
ρ - density of the liquid pumped by the pump, kg/m.

$$P_{П} = 970 \cdot 153{,}9^2 \cdot \frac{0{,}213^2 - 0{,}148^2}{8}$$ =67388 Pa

Then the pressure drop across the front seal of the first stage impeller is:

P_p=9,81×970×49,9 – 67388 = 407445 Pa

Leaks in the seal:

$$q = \mu \cdot 2\pi r_1 \delta \sqrt{\frac{2p}{\rho}}$$

$$q = 0{,}5 \cdot 2\pi \cdot 0{,}148 \cdot 0{,}0003\sqrt{\frac{2 \cdot 407445}{970}}$$ =0.004 m^3/h

We determine the static pressure drop in the sinus of the front seal of the intermediate stages:

$$P_{П} = \rho \cdot \omega^2 \cdot \frac{R_2^2 - R_1^2}{8},$$

R_1 – front seal diameter, m;

ρ - density of the liquid pumped by the pump, kg/m.

$$P_{П} = 970 \cdot 153{,}9^2 \cdot \frac{0{,}213^2 - 0{,}082^2}{8}$$ =110982 Pa

Then the pressure drop across the front seal of the intermediate stage impeller is:

P_p=9,81×970×49,9 – 110982 = 363851 Pa

Gap consumption:

$$q = \mu \cdot 2\pi r_1 \delta \sqrt{\frac{2p}{\rho}}$$

$$q = 0{,}5 \cdot 2\pi \cdot 0{,}082 \cdot 0{,}0003\sqrt{\frac{2 \cdot 363851}{970}}$$ =0.0021 m^3/h

The pressure drop across the interstage seal is determined by the formula:

$$P_{Г} = \gamma \cdot \left(H - H_{пот}\right) + P_{м},$$

$P_{м}$ - static pressure drop in the sinus of the interstage seal, Pa.

$$P_{м} = \rho \cdot \omega^2 \cdot \frac{R_2^2 - R_3^2}{8},$$

$$P_{м} = 970 \cdot 153{,}9^2 \cdot \frac{0{,}213^2 - 0{,}083^2}{8} = 110508 \text{ Pa}$$

$P_{Г}$=9,81×970×49,9 – 110508 = 364325 Pa

Gap consumption:

$$q = \mu \cdot 2\pi r_1 \delta \sqrt{\frac{2p}{\rho}}$$

$$q = 0{,}5 \cdot 2\pi \cdot 0{,}083 \cdot 0{,}0003 \sqrt{\frac{2 \cdot 364325}{970}} = 0.0021 \text{ m}^3/\text{h}$$

Pressure drop across the drum:

$$P_{б} = \gamma \cdot H_{н},$$

$H_{н}$ – pump head, m.

$$P_{б} = 9{,}81 \cdot 970 \cdot 185 = 1760405 \text{ Pa}$$

Gap consumption:

$$q = \mu \cdot 2\pi r_{б} \delta \sqrt{\frac{2p}{\rho}}$$

$$q = 0{,}5 \cdot 2\pi \cdot 0{,}141 \cdot 0{,}0003 \sqrt{\frac{2 \cdot 1760405}{970}} = 0.008 \text{ m}^3/\text{h}$$

5 Choosing an electric motor

Based on the power consumption of the pump, we select an asynchronous three-phase electric motor with a squirrel-cage rotor, blown, vertical explosion-proof, VAOV series, designed to drive oil support pumps of the NPV type.

Main technical characteristics of the electric motor:

Type of electric motor	Power kW	Voltage V	Speed (asynchronous), rpm	Efficiency, %	cosφ	$M_{мах}$ M_{nom}	M_{start} M_{nom}	I_{start} I_{bnom}	Motor moment of inertia kg*m2 rotor
BAOB 550 S-4	550	6000	1470	97,2	0,88	1,2	6,1	2,34	8,16

6 Labor protection

Regulation and monitoring of occupational safety at an industrial enterprise.

Standardization in the field of labor protection at an industrial enterprise is the process of establishing and applying standards, rules and requirements directly aimed at ensuring safe and healthy working conditions at a given specific industrial enterprise for the employees of this enterprise (personnel), as well as other persons who may be on the territory of this industrial enterprise during the implementation of technological production processes there [5].

Standardization in the field of labor protection at an industrial enterprise helps to minimize professional risks for employees of this enterprise, as well as improve production conditions. It is necessary to require entrepreneurs (employers) to carefully comply with the adopted procedures, regularly review them with the introduction of the necessary relevant changes and additions in accordance with the updated requirements of the current legislation [8].

Control and supervision of compliance with labor legislation by entrepreneurs-employers is carried out by specialized state bodies (institutions), each of which has its own clearly defined functions and powers. In particular, supervisory bodies may [6]:

- have unimpeded access at any time of the day to the premises (industrial and non-industrial) of various organizations of any form of ownership (including individuals - entrepreneurs);

- require entrepreneurs-employers and their authorized representatives to provide all necessary documents that allow an opinion to be made on the state of the issue of labor protection at a given specific enterprise or institution;

- extract samples of materials and substances that a given specific industrial enterprise uses in its production technological process as raw materials or auxiliary materials for the purpose of conducting analysis through appropriate research and testing;

- conduct an investigation of the circumstances that led to the emergence of a situation at a given industrial enterprise that resulted in an accident and harm to employees of that enterprise while they were performing their immediate official duties, or to other (outsiders) who were on the territory of that industrial enterprise;
- - make demands on entrepreneurs-employers and their authorized representatives to eliminate violations of the labor legislation currently in force that were identified during inspections and that could potentially lead to the emergence of harmful and dangerous situations and, as a consequence, to injuries among employees of that specific industrial enterprise while they were performing their official duties. on its territory;
- - send demands to the courts to eliminate an industrial enterprise that is dishonest or negligent in complying with the requirements of the current legislation to ensure safe and proper working conditions, or to suspend (in general or for a certain period of time - until violations regarding working conditions in production identified during the inspection are eliminated) the activities of individual structural divisions of that enterprise (shops);
- - issue orders to enterprises on whose territory inspections were conducted to remove from their official duties employees of this enterprise who, for one reason or another, have not undergone training in safe methods and techniques for performing technological operations and work at their workplace, occupational safety briefing (primary, refresher, etc.), internship at their workplaces and the construction industry;

- prohibit the use of personal and collective protective equipment for workers and employees and personnel of a given industrial enterprise in the event that these equipment do not fully meet the mandatory technical requirements specified in the relevant regulatory documents, or the requirements for labor protection and safety engineering at a given specific production facility, with mandatory consideration of all the features of the adopted violations in the field of labor safety and safety engineering at an industrial enterprise have been identified, and conduct consideration of cases of administrative violations by employers in the field of labor protection at their own production facility.

Also, specialized government bodies (institutions) have the right to conduct inspections at industrial enterprises regarding compliance with the following requirements [7]:

- - nuclear, radiation, technical and fire safety at industrial enterprises of the relevant sectors (both industrial and others);

- - industrial safety during the performance of work on the design, construction, operation of production lines and equipment, conservation and/or liquidation of hazardous industrial facilities (lines, workshops, sections, individual buildings and premises, etc.), manufacture, installation, debugging, maintenance and repair of technical devices (equipment, devices, sections, process lines, etc.), transportation of potentially hazardous and/or harmful substances (raw materials, materials, etc.) on the territory of production and/or non-production premises during the implementation of the technological process for the production of specific industrial products adopted at a given specific production facility;

- safety in the electric power industry: supply and use of electric power at the enterprise, in particular, by the technological equipment of the enterprise on technological production lines;

- safety of the entire range of works directly or indirectly related to the use of subsoil (minerals);

- fire safety at underground facilities (various industries) and during blasting operations;

- safety rules for the maintenance of hydraulic structures by the owners of hydraulic structures and organizations that operate them;

- Compliance with the terms and requirements of technical regulations in the established area of activity of a particular production or non-production enterprise (organization, institution, etc.).

Specialized government agencies (institutions) may [5]:

- - upon presentation of the relevant service ID, freely reside on the premises (including in production and non-production premises) of organizations, enterprises and institutions, have access to their documents and materials, and verify compliance with and execution of the requirements of laws on occupational safety and health at a given specific industrial enterprise;
- - demand that the management of the enterprise, as well as other official persons, provide the entire set of necessary documents, materials and other information required for a thorough inspection of compliance with the requirements of current legislation on occupational safety and health at the enterprise and its adjacent territories;
- - demand that relevant inspections be carried out, initiated in connection with the receipt of requests and demands from citizens and other interested persons regarding compliance with occupational safety and health requirements at an industrial enterprise and its adjacent territories;

- summon officials and citizens to provide official explanations regarding the information received regarding the presence of violations of the requirements of the current legislation on safety and labor protection at this industrial enterprise and its adjacent territories;

- demand that the owners of the industrial enterprise and their official representatives urgently perform work to immediately and completely eliminate the identified violations of safety and labor protection at this industrial enterprise and its adjacent territories;

- initiate administrative cases on identified violations of the current legislation on safety and labor protection at this industrial enterprise and its adjacent territories;

- demand that appropriate inspections be carried out by other state bodies for control of safety and labor protection in the mechanical engineering industry.

By nature, inspections are divided into [8]:

- scheduled (comprehensive or thematic);

- unscheduled (targeted or control).

By form, inspections are divided into:

- on-site;
- documentary.

Activities related to conducting scheduled inspections of the state of technical safety and labor protection at an industrial enterprise are conducted no more than once every three years. If such an inspection is comprehensive, then when conducting procedures to control the state of technical safety and labor protection at an industrial enterprise, together with issues directly related to compliance with labor protection requirements, the correctness of maintaining personnel documentation at the industrial enterprise, local regulations and other documentation are also checked. If such an inspection is thematic, then only those issues that are directly related to compliance with the requirements of current legislation regarding safety and labor protection at the enterprise are subject to inspection [6].

A documentary audit is carried out without visiting the territory of an industrial enterprise by analyzing the documentation that the enterprise provided to the institution upon its official request [5].

7 Economic part

Functional cost analysis of new equipment

In order to ensure the specified high level in terms of technical and economic perfection of new industrial (including pump) products, it is extremely necessary to organize work on determining and strict achievement of optimal technical parameters, as well as economic indicators of industrial products that the enterprise produces during the implementation of the approved plan for the production of industrial products. One of the methods capable of effectively solving such a problem is the introduction at the stage of preparation of products for their production at an industrial enterprise of the concept underlying the functional cost analysis (FVA) [35].

During all the activities provided for by the functional-cost analysis, it becomes possible to identify and eliminate additional costs (which are not mandatory and affect the quality of the manufactured products), as well as to create new design solutions for already known industrial products that have a high level of quality and are associated with costs that do not significantly exceed the costs associated with the production of the basic version of this type of product [35].

Functional methods of cost analysis can also be used when performing work related to the design of technological production processes, the organization of production of industrial products for various purposes, arising when mastering new products (equipment) in production.

Functional cost analysis is a method of systematic research of any type or kind of object (for example, a part, product, equipment, process, structure, etc.), which is aimed at achieving a higher level of efficiency in the use of the material (equipment, equipment, raw materials, materials, fixed assets, etc.) and labor available at a given production enterprise. This method, when used consciously, rigorously and correctly, is capable of ensuring an optimal balance between the consumer properties of the manufactured products and the costs associated with their life cycle (from production at the enterprise to the end of operation) [35].

Functional cost analysis can ensure an increase in the efficiency of using material and labor costs in production even at the stage of preparation for the production of industrial products of a mechanical engineering enterprise through the use of the following approaches [35]:

- prevention of occurrence of excessive costs associated with production of industrial products that have a clearly expressed functional nature;

- reduction to an acceptable level or complete elimination of costs for production of industrial products that are unjustified by the features of the technological process or other objective factors and losses that occur during implementation of the technological process for manufacturing industrial products adopted at the enterprise.

It is quite obvious that each new unit of a product that is developed by an industrial product manufacturer should be more rational from the point of view of manufacturing and subsequent use during operation and more economical (both from the point of view of manufacturing - have a minimum cost, and from the point of view of operation - should bear minimum costs for maintaining it in proper condition, including conducting reviews, the functionality and parameters of the purpose correspond to this new unit of an industrial product [35].

An analysis of known designs of industrial products, conducted on the basis of statistical data processing, shows that a significant number of product elements (parts, components, units, their structural elements, etc.) quite often have economic deficiencies, which at first glance often seem quite insignificant, unimportant and have no economic impact. An economic deficiencies is a design decision made during the design of industrial products, which subsequently leads to an increase in costs that are not justified. Examples of such economic deficiencies may be the following [35]:

- unjustified (for example, due to the specifics of the manufacturing process or new functional capabilities) complication of the part shape;

- excessive and unjustified overestimation of the accuracy class of surface treatment of parts (which leads to an increase in the cost of processing work);

- overstated value of surface purity of a part that is not working during its operation directly at the place of its use (in a machine, mechanism, unit, etc.);

- unjustified use of too valuable materials for the manufacture of a part in cases where the use of a cheaper substitute does not lead to a fundamental decrease in the functional capabilities of the part and does not critically affect its operating parameters in terms of strength and durability;

- embedding excessive strength of parts during design (such an increase in strength is ensured by large dimensions of the part elements, which in turn leads to an increase in its mass and, accordingly, the cost of the workpiece);

- use of coatings that are too expensive in cases where cheaper options fully satisfy the requirements of protective coatings and do not critically affect the durability of the product.

With consistent, systematic and conscious application of cost-effectiveness analysis in industrial production, the manufacturer gets the opportunity to eliminate a significant portion of the shortcomings already at the design stage of industrial products (at the very beginning of the product life cycle), as well as during the manufacture of prototypes of these products. The importance of using cost-effectiveness analysis in serial production of industrial products should not be underestimated, when the reduction in the cost of one product will usually be insignificant, but for a batch of homogeneous products such a reduction can be quite significant. The very basis of cost-effectiveness analysis was the principle that the object of research in the implementation of this basic principle should not be the products of labor (individual samples of industrial products manufactured at the enterprise) as such, but the functions that these products of labor perform. This approach is due to the fact that it is the functions performed by the physical object (product of labor) that interest the consumer and satisfy his needs, and not the products of labor themselves as such [35].

All functions inherent in the products of labor (products produced at an industrial enterprise) are distinguished in the theory of functional cost analysis by the following features [35]:

- by the area of application – external functions and internal functions;

- the role they play in the process of satisfying a potential buyer of industrial products – main functions and auxiliary functions;

- by the degree of usefulness of the function for each specific consumer of the products of an industrial enterprise – useful functions, useless functions (useless, useless, unnecessary, redundant, etc.), harmful functions (and sometimes potentially dangerous for the consumer).

In practice, the implementation of functional cost analysis at an enterprise manufacturing industrial engineering products occurs in several important and sequential stages [35]:

Stage 1 - preparatory - this stage of the functional cost analysis involves the following set of actions: selection of the object of study, setting the task for further research, the goal to which this functional cost analysis will be directed, formation of a working group that will directly implement the functional cost analysis. It should be noted that the object of the functional cost analysis can be either a new product (providing for its design "from scratch"), or a modernized version of an already known product (which was previously produced at this particular enterprise, or at other enterprises in the industry);

Stage 2 — informational — this stage of the functional-cost analysis involves familiarizing the working group with the product drawings, technological processes (adopted at this specific manufacturing enterprise for the manufacture of this type of product), passports for the main and auxiliary equipment, which is involved in the process of manufacturing a unit of industrial product. products of this enterprise), advertising brochures of competing firms (manufacturing similar or similar products), scientific articles (containing theoretical and practical information on the calculation and design of this type of industrial product). At this stage, it is mandatory to also study and analyze the economic indicators of industrial production: the cost of manufacturing industrial product samples at this enterprise; market prices for similar or similar products (taking into account the specifics of individual markets for finished industrial products); costs associated with the production and development of new equipment in production;

Stage 3 – analytical – this stage of the functional cost analysis assumes the following goal – conducting an analysis of the object (a unit of new industrial product suitable for its further production at a given machine-building enterprise) as a whole in order to determine specific tasks for the development and formulation of various solutions suitable for further production, each of which should be aimed at;

Stage 4 – creative – this stage of functional cost analysis assumes the following goal – searching for new ideas that could form the basis for creating various options for solving the assigned task, as well as the procedure for selecting the most appropriate options for solving the assigned task (the most acceptable options for solving this specific product for further production at the enterprise);

Stage 5 – research – this stage of the functional cost analysis involves evaluating the options selected at the previous stage, followed by selecting the main option using previously adopted criteria for evaluating and quantitatively comparing various options.

Stage 6 – production and control of results – this stage of the functional cost analysis involves the development of a schedule according to which the implementation of preliminary organizational and technical decisions on the production of new industrial products by a machine-building enterprise will be carried out, as well as the actual implementation of these decisions.

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