The design of responsible details of an axial piston hydraulic machine improving

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Abstract. Methods of improving the designs of the most complex details of the axial piston hydraulic machine (APHM), cylinder block (CB) and face distributor (FD) are proposed in order to increase their strength. The possibility of modernization is demonstrated by the example of APHM 210.25 with a CB inclined. For samples of CB material, bronze CuSn12, tests were performed and fatigue curves are constructed. Based on the results of these tests, the endurance limit of the CB is determined using the theory of similarity fatigue fracture. The calculation of the BC by the finite element method showed that maximum stresses arise in its inter-cylinder wall (jumper). With conserved CB dimensions, a decrease in the maximum stress level by 30% is achieved if the axial cylinders are shifted in the peripheral direction by 2 ... 3 mm. In the proposed design of the CB the calculated stresses became lower than its endurance limit. For FD an increase in the thickness of its peripheral zone was proposed, which also made it possible to reduce stresses by 12%. As a result, the level of acting stresses turned out to be 5.3% lower than the FD endurance limit. The achieved reduction in stress level for the considered details and an increase in their durability is a factor that growing the environmental safety of APHM.

1. Introduction
The advantages of hydraulic machines (compactness, versatility, ease of operation, energy efficiency, high dynamic characteristics) ensure their widespread use in aviation, land and water transport, housing and communal services, the agricultural complex, etc. [1]. Energy efficient determines continuous expansion of the scope of APHM applications, from mobile military equipment to seawater desalination systems by reverse osmosis for domestic use.

In recent years, much attention has been paid to improving energy efficiency. Many energy saving strategies have been successfully developed and implemented [2, 3]. In particular, the use of a hydraulic drive in a wind power installation increases its efficiency by 17% and raises its efficiency to 90.7% [4]. The use of hydraulic components, such as electric pumps, hydraulic motors, electronic control systems, allows in some industrial systems to reduce energy consumption by 50...70% [5]. The experiments showed that the use of a hydraulic drive in a lifting device with optimal control leads to a reduction in energy consumption by 64% [6].

The development of industrial equipment is associated with an increase in its energy and dynamic characteristics. Improving the APHM is primarily aimed at increasing the working pressure and
rotation frequency. The requirement of resource saving obliges to ensure minimum weight and
dimensions. As a result, during the operation of APHM, stresses in their parts increase, temperature
rises, this leads to an increase in the frequency of failure cases [7, 8]. To clarify the causes of failures,
methods for analyzing the kinematics and dynamics of APHM [9–12] are developed and refined, and
mathematical modeling is improved [13–15]. The accuracy of calculations is achieved using the finite
element method (FEM) in a three-dimensional setting [16–18]. Calculation results are checked during
field tests [19]. The increase in pressure causes a growth in the forces of contact interaction and the
intensification of wear during of details relative moving. To wear reducing new anti-friction materials
are developed, working fluids are improved, and methods for cleaning them are improve [20, 21].

Despite said measures, in many cases it is not possible to achieve the required high level of
working pressures and speeds. Therefore, instead of copper alloys, it is proposed to use high-strength
steels as a material for the CB [22]. This decision seems unsuccessful, since with steel pistons the
friction coefficient in the steel-steel pair will be an order of magnitude greater than, for example, in the
tin bronze and steel pair. An increase in friction forces leads to a decrease in mechanical efficiency,
unacceptable overheating of parts and accelerated wear.

In recent decades, technologically more complex composite or precast constructions of increased
strength have begun to be used. When choosing materials for various parts, the APHM follow are
strive to provide with their required local quality and take into account the technological capabilities
of production. If CB is made of steel to ensure strength, then to reduce friction losses for the contact
surfaces it is recommended copper alloys surfacing [23]. Another solution to the problem is to
reinforce the bronze block outside with a shell made of high-strength steel [24]. Cavities can be
created under the shell, into which the working fluid for hydraulic discharging of the CB is fed
through special channels.

The most responsible and at the same time the most loaded element of any APHM is the pumping
unit. Its main details are pistons with connecting rods, a drive shaft, bearings, a CB and an FD. The
methods of designing and calculating these details, except for the last two, are well developed. At the
same time, CB and FD account for about 30% of APHM failures [12, 14], a significant part of which
are fatigue fractures. Therefore, when calculating their strength and durability, in order to ensure high
reliability, one should focus on the characteristics of fatigue resistance.

2. The goals and objectives of the study
The purpose of the work is to improve the responsible details of the pumping unit of the APHM in
order to increase their strength without significantly complicating the design.

In the process, the following tasks are solved:

- determination of the characteristics of strength and fatigue resistance of materials of the details
  in question;
- calculation of the characteristics of the fatigue resistance of the details under study based on the
  statistical theory of the similarity of Serensen-Kogaev fatigue failure;
- modeling of details for the purpose of their calculation in the ANSYS Workbench software
  package and analysis of strength using FEM.

Reducing stresses in the CB and FD below the endurance limit will lead to increased reliability and
environmental safety of APHM.

All calculations were performed for the nominal pressure of the working liquid $q = 25$ MPa. Based
on the results of strength calculations of the studied structures, options with the lowest level of
maximum stresses were selected.

With increasing pressure $q$, the maximum stresses will proportionally increase due to the linearity
of the formulation of the problems under consideration.
3. Improving the cylinder block
As a result of relative movements between the CB, FD and pistons, sliding friction occurs. To reduce wear, copper alloys are used as the material for CB, the best of which are tin bronzes. In this work, the object of study is the seven-piston APHM 210.25 (Fig. 1).

![Figure 1. The block of cylinders APHM 210.25.](image)

The most promising material for the CB is bronze CuSn12, which has a low coefficient of friction, high thermal conductivity and wear resistance. Its main components: tin – 12%, lead – 1.3%, nickel – 1%, the rest – copper. This bronze has rather high mechanical characteristics: ultimate strength $\sigma_u=320$ MPa, conditional yield strength $\sigma_{0.2}=180$ MPa, elongation at break $\delta=12\%$, hardness 101...104 HB.

Large volumes of loaded axial cavities and instability of pressure lead to the appearance in the CB of complex, cyclically variable stress fields. Such character of loading causes damage accumulation and, in some cases, leads to fatigue failure. In this regard, CB should be calculated for fatigue resistance. For such a calculation, it is necessary to establish the fatigue characteristics of the CB material bronze CuSn12. To this aim tests of its samples were conducted (Fig. 2).

The tests were carried out at a clean circular bend. The sizes of smooth (corset) and notched samples are shown in Table 1. The values of the theoretical stress concentration coefficients $K_t$ calculated from the Neuber dependences also here are presented.

![Figure 2. The sample for fatigue test.](image)
Table 1. Characteristics of samples for fatigue tests.

| Form of the working part | Sizes, mm | $K_t$ |
|--------------------------|-----------|-------|
| Smooth                   | 100       | 11    | 7.5 | 1.75 | 75 | 1.014 |
| Notched up               | 100       | 11    | 7.5 | 1.75 | 1.0 | 1.737 |

The equation of the inclined branch of the fatigue curve in power form is presented

$$\sigma^mN = \sigma_{-1}N_G = 10^C = \text{const},$$

(1)

where $\sigma$ and $N$ – the amplitude of the alternating stress and the corresponding number of cycles to failure; $m$ and $C$ – the parameters of equation (1); $\sigma_{-1}$ – the material endurance limit for a symmetric stress cycle; $N_G$ – the abscissa of the fracture point of the fatigue curve constructed in double logarithmic coordinates.

According to the results of fatigue tests and correlation-regression analysis, the characteristics of equation (1) are calculated and are presented (Table 2 and Fig. 3).

Table 2. Bronze CuSn12. Characteristics of the fatigue curves.

| Forms of working surfaces of samples | Designation | $m$ | $C$ | $N_{G}$ cycles | $\sigma_{-1}$, MPa |
|-------------------------------------|-------------|-----|-----|----------------|-------------------|
| Smooth (corset)                     | ●           | 7.14 | 21.94 | 2.365·10$^7$ | 110               |
| Notched up, Fig. 1                  | ■           | 7.84 | 22.56 | 1.228·10$^8$ | 70                |

Figure 3. Fatigue curves for tested samples: 1 – smooth; 2 – notched up.

The median value of the endurance limit for a CB is determined using the statistical theory of the similarity of Serensen-Kogaev fatigue fracture [25–27] through the endurance limit of $\sigma_{-1}$ according to the method of standard GOST 25.504-82

$$\bar{\sigma}_{-1D} = \sigma_{-1}K_FK_A/[2K_t/(1 + \nu^2) + 1/K_F - 1],$$

(2)

where $\theta$ – the similarity criterion for fatigue failure; $K_A$, $K_F$, $K_t$, $\nu^2$ – respectively are the coefficients of anisotropy, the effect of surface roughness, stress concentration and surface hardening; $\nu$ – the coefficient of sensitivity to stress concentration and influence of dimensions of the detail.

Calculations by formula (2) for CB at symmetric cycle of stresses change gave value $\bar{\sigma}_{-1D} = 57.5$ MPa.

In APHM the loading of the cylinders changes according to the pulsating cycle of the alternating stresses. The endurance limit $\bar{\sigma}_{0D}$ with such a cycle was calculated for the CB using the Soderberg formula
\[
\sigma_a / \bar{\sigma}_{-1D} + \sigma_m / \sigma_{0.2} = 1,
\]  
(3)

where \(\sigma_a\) and \(\sigma_m\) – the amplitude and mean stresses of the cycle of variable stresses, respectively,  
\(\bar{\sigma}_{-1D}\) – the median value of the endurance limit of the CB with a symmetric cycle; \(\sigma_{0.2}\) – conditional yield strength.

With the pulsating cycle of stress variation, the average stresses \(\sigma_m\) are equal to the amplitude \(\sigma_a\),  
with \(\sigma_a=0.5\sigma_{\text{max}}\) and \(\sigma_{\text{max}} = \bar{\sigma}_{-1D}\). For these values, according to dependence (3), a formula is obtained for the endurance limit of CB with a pulsating cycle of stress variation

\[
\bar{\sigma}_{0D} = \frac{2\sigma_{-1D} \cdot \sigma_{0.2}}{\sigma_{-1D} + \sigma_{0.2}}.
\]  
(4)

Calculation according to formula (4) gives for CB the value of endurance limit \(\bar{\sigma}_{0D} = 87.1\) MPa.

For calculate the stress state the FEM program of the ANSYS Workbench software is used. The number of cylinders in the injection zone varies from three or four and in the discharge zone, respectively, four or three. The calculations were carried out in a three-dimensional setting with a breakdown of the part into finite elements – tetrahedrons with ten nodes. According to the number of axial holes in the discharge zone, three or four, two calculation options were performed at a pressure of \(q=25\) MPa. These calculations were carried out for each analyzed design option.

It was established that the highest stresses occur in the CB with the location of four axial channels in the injection zone. The maximum of the first main stress \(\sigma_{\text{max}}\) takes place in the inter-cylinder partition (ICP), where its thickness is minimal (Fig. 4, a). The maximum radial stresses acting normal \(x\) to the cross-section of the partition vary in the range (102...104) MPa and are concentrated on a section about 3 mm long. The area of action of radial stresses with a level of (94 ... 104) MPa has a length of about 22 mm (Fig. 4, b). In the middle zone of the inter-cylinder partition the voltage is less than in the peripherals by 30 ... 35%. The calculated value of the maximum radial stress \(\sigma_r=104\) MPa (Fig. 4, b) exceeds the endurance limit \(\bar{\sigma}_{0D} = 87.1\) MPa by 19.4%. Therefore, the CB resource is limited and is approximately \(N=10^7\) cycles.

**Figure 4.** The stress state of the BC under the action of pressure \(q = 25\) MPa in four cylinders:  
\(a\) – distribution of the principal stresses \(\sigma_i\) in the longitudinal section of the ICP;  
\(b\) – the character of the change in radial stresses along the length and thickness of the danger zone of the ICP.
Analysis shows that the stress state of the serial CB is substantially heterogeneous, since in its outer regions the stresses are much less than the maximum stresses in the ICP (Fig. 4, a). To improve the design of the CB, it is proposed to change the location of the axial cylinders so that the stresses are redistributed: decreased in the ICP and increased in the external partitions.

To reduce the stress level in the ICP, a simple design change is proposed, namely, a slight displacement of the axes of the cylinders in the CB diametric planes (Fig. 5). The displacement of the axes of the cylinder holes in the inlet and bottom parts, $\Delta=\Delta_d$ and $\Delta=\Delta_b$, respectively, will be considered positive if they are directed along the radius $r$ from the central axis $Oy$. With a positive offset, the thickness of the partitions between the axial holes increases and the thickness of the outer partition decreases. The minimum values of these thicknesses, respectively, $t_i$ and $t_e$, are determined by the dependences

$$t_i = (D_c + 2\Delta)\sin \frac{\pi}{n} - d_0,$$

$$t_e = 0.5(D_c - D_c - 2\Delta - d_0),$$

where $D_c=110$ mm is the outer diameter of the CB; $D_c=70$ mm is the diameter of the arrangement of the centers of the axial holes in the serial design; $d_0=25$ mm — diameters of these holes; $n=7$ — their number (fig. 5).

The calculation results of the thickness of the partitions according to formulas (5) and (6) at increasing diameter $D_c$, i.e. at positive displacements, are presented (Fig. 6). Established, that the radial displacement of the piston cavities in the positive direction, but not more than 3.5 mm, is preferable. Negative displacements $\Delta_b$ turned out to be irrational, since in this case the maximum stresses in ICP increase.

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Figure 5. Scheme of CB modernization by displacement of axial cylinder bore axes in diametric planes.
As a result of modeling and calculations in the ANSYS Workbench software package, the stresses in the CB of several modernized models were determined. The maximum stresses in the considered models arise in the ICP. Their values are given in Table 3.

Table 3. The maximum stresses in the CB with displacements of the axes of the cylinders.

| Δd, mm | -1.5 | +4  | +3  | +2  | +1  | 0   | +0.5 |
|-------|------|-----|-----|-----|-----|-----|------|
| Δh, mm |      |     |     |     |     |     |      |
| Von Mises stress, σ_{Mmax}, MPa | 165  | 132 | 106 | 110 | 121 | 139 | 214  |
| Principal stress, σ_{1max}, MPa | 137  | 109 | 77  | 79  | 93  | 113 | 204  |

The best of the considered upgrade options turned out to be CBs with axial displacements Δh=+2...3 mm at Δd=0 (Table 3). In these variants, the stresses are reduced in comparison with the initial design with Δd=Δh=0 by 30% (Fig. 7, 8).

Figure 6. Change in the thickness of the partitions: 1 – inter-cylinder; 2 – external.

Figure 7. Distribution of the first main voltage in the advanced CB with Δd=0 and Δh=+2 mm.
In the calculations the number of finite elements and nodes are equal to 400 000...600 000. The calculation time of each option was 3...4 minutes.

![Figure 8. The values of the maximum stresses depending on the displacement $\Delta_V$ at $\Delta_A = 0$:
1 – Von Mises stress, $\sigma_{\text{Mmax}}$; 2 – principal, $\sigma_{\text{1max}}$.](image)

At displacements $\Delta_V > 3$ mm, stresses increase both in ICP and in the external partitions of the CB. The second upgrade option is to change the shape of the transition surface from the cylinder cavities to the exhaust channels in the front of the CB. It is proposed to replace the flat surface with a conical transition with an apex angle of 120 ° ... 150 ° (Fig. 9).

The calculations showed that when replacing the flat form on the conical transition, the stress values are practically unchanged, but in this design the conditions for the flow of the working fluid to the distributor are improved and the noise level will be reduced. The conical transition can be combined with the displacement of the axes which was proposed in the first embodiment.

![Figure 9. The design of the inner surface of the cylinder cavities in the front of the CB: a – serial; b – modernized.](image)

4. The FD improvement
APHM 210.25 has FD material – low-alloy chromium-molybdenum steel (its counterpart in the European Union is 41CrAlMo7 steel). The main components of this steel: carbon – 0.35...0.42%, chromium 1.35...1.65%, silicon 0.20...0.45%, manganese 0.15...0.25%, molybdenum 0.15...0.25%, aluminum 0.7...1.1%. Chemical heat treatment is normalization and nitriding of the surface.
The mechanical characteristics of this steel are: conditional yield strength $\sigma_{0.2} = 580...590$ MPa, ultimate strength $\sigma_u = 760...790$ MPa, endurance limit $\sigma_{-1} = 420...450$ MPa for smooth samples with a diameter of 7.5 mm.

Machining after nitriding – grinding to the arithmetic mean deviation of the profile of the irregularities $Ra = 1.6 \mu$m.

The main load acting on the FD is the pressure $q$ of the working fluid in the injection window. The nature of loading of serial and modified spherical FDs (Fig. 10).

The stresses arising in the FD under the action of pressure $q = 25$ MPa are determined using the FEM. Their maximum values arise in the boundary zones of the outer wall of the high-pressure window, in the vicinity of points $A$ and $B$. It was found that a rather simple modification of the structure (the thickness of the peripheral zone was increased by 1.5 mm) (fig. 10, c), it decreases the value of the maximum principal stress: from $\sigma_{1\text{max}} = 176$ MPa in a serial FD (SFD) to $\sigma_{1\text{max}} = 157$ MPa in a modified (MFD), i.e. by 12% [18]. The stress distribution $\sigma_{1\text{max}}$ in the radial section of the outer wall passing along the line $Ax$ is presented in table 4.

Table 4. Distribution of the principal stresses in the dangerous section of the FD.

| Coordinate $Ax$, mm | 0   | 1.0 | 2.5 | 4.0 | 5.5 |
|---------------------|-----|-----|-----|-----|-----|
| Serial FD           | $\sigma_{1S}$, MPa | 176 | 140 | 77  | 46  | 21  |
| Modified FD         | $\sigma_{1M}$, MPa | 157 | 114 | 79  | 58.5| 40.5|

A similar picture is obtained in a symmetrically located neighborhood of point B. From the analysis given in table 4 values it follows that, along with a decrease in the magnitude of the principal stress of the MFD, the gradient of these stresses $G = d\sigma_1/dx$ in the vicinity of hazardous points $A$ and $B$ becomes much smaller. The calculations give the following values of the gradients near points $A$ and $B$: for the SFD $G_S = 81.8$ MPa/mm, for the MFD $G_M = 53.0$ MPa/mm. With a cyclic change in stresses a decrease in the magnitude of the stress gradient decreases the rate of fatigue failure.
In work [18] the values of endurance limits were determined for pulsating cycle of stress variation, for SFD $\sigma_{0S}=166.2$ MPa, for MFD $\sigma_{0M}=165.3$ MPa with a 98% probability of distributor non-fracture. It follows from the calculations that the SFD has $\sigma_{0S}<\sigma_{1S}=176$ MPa, i.e. SFD life will be limited. At MFD we have $\sigma_{0M}>\sigma_{1M}=157$ MPa and unlimited durability is ensured.

5. Summary

1. It is shown in the paper that an increase in strength can be achieved without the use of additional reinforcing elements, due to the improvement of the structures of the parts of the pumping unit. Simple methods have been proposed for increasing the constructive strength and cyclic durability of the most complex in form APHM details – the cylinder block and the face distributor.

2. An algorithm for numerically-analytical determination of stress concentration factors has been developed for details of complex configuration, based on the FEM.

3. The possibility of calculating the resource of details with the localization of dangerous stresses in small volumes by the methods of the statistical theory of similarity of fatigue failure is shown.

4. For a serial axial piston hydraulic machine 210.25, if the axial cylinders are offset in the radial direction by 2.5 mm, the maximum stresses are reduced by 30%.

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