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INCREASING THE CONTROL RANGE OF AN AXIAL PISTON SWASH PLATE HYDRAULIC MACHINE

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Abstract

Axial piston swash plate hydraulic machines are widely used in modern volumetric hydraulic drives, which is due to their high energy consumption and simplicity of design. A significant disadvantage of hydraulic machines of this type is the small control range caused by the large value of the minimum stable revolutions of the hydraulic drive shaft. In this research, the possibility of increasing the angle of inclination of the swash palte without increasing the friction forces in the piston- guide bushing pair to reduce the value of the minimum stable revolutions of the hydraulic drive shaft is considered.

Keywords: Hydraulic machines; Axial piston machines; Volumetric hydraulic drive; Control range, Tilt angle of the swash plate, Shaft torque.

1. Object of investigation

The object of research in this work is an Axial piston swash plate hydraulic machine (APGND), since it is widely used in modern hydraulic drives due to its high energy consumption and simplicity of design [1]. An increase in the angle of inclination of the hydraulic machine swash plate reduces the value of the minimum stable revolutions of the drive shaft of the APGND due to an increase in the useful component of the force applied from the shoe to the swash plate of the hydraulic machine [2].

However, an increase in the angle of inclination also leads to a greater misalignment of the piston in the guide sleeve, and accordingly to an increase in friction forces in this kinematic pair. Figures 1 and 2 show diagrams of forces in a piston pair of an APGND of standard and modified design [3].

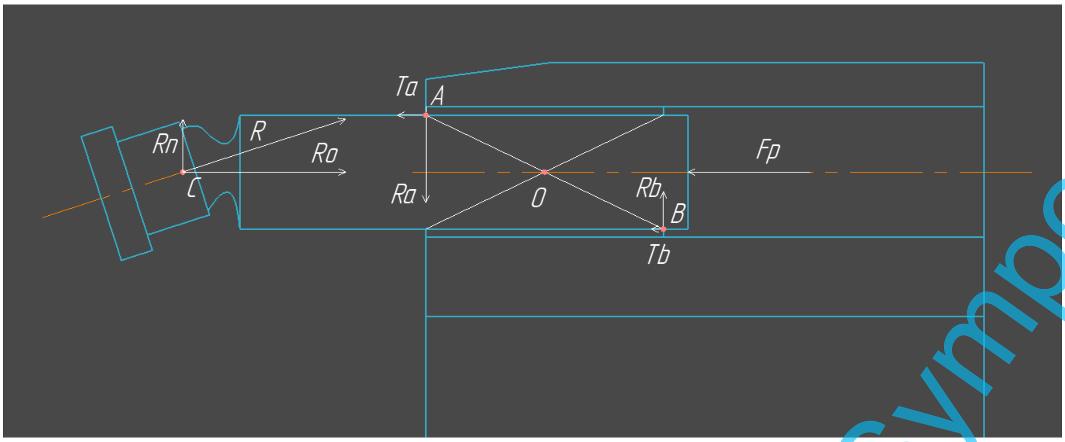


Fig. 1. The diagram of forces in the standard piston mechanism of the APGND

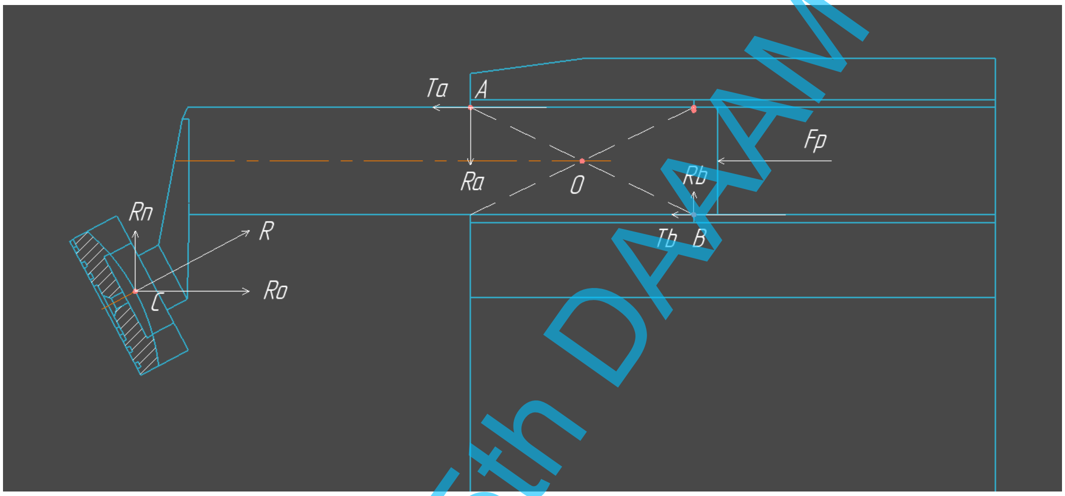


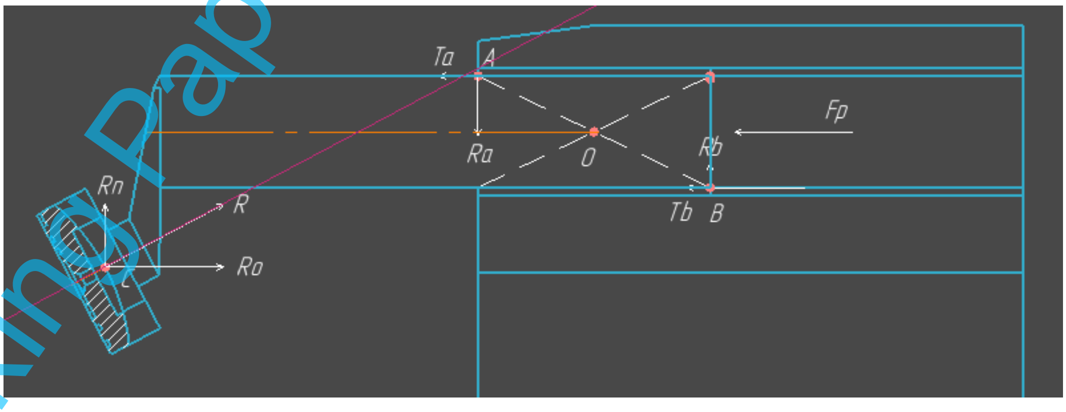
Fig. 2. The diagram of forces in the modified piston mechanism of the APGND

Analyzing the force patterns, it can be seen that when the hydraulic machine is operating, the Fp force causes a reaction R in the piston joint C. This reaction decomposes into two components of the force: Ra and Rb [4].

The magnitude of these reactions affects the friction forces Ta and Tb, the combination of these two friction forces and fluid pressure will balance the horizontal component of the Ro force R. Similarly, the perpendicular component of the force Rn. Rn also balances the resultant force of the two pressures Ra and Rb.

In nominal operation mode, the total torque and force of the entire system are balanced. Considering any power point (taking the point O as the center of the piston core), we have 3 cases depending on the position of the piston inside the guide sleeve [5]. The force diagram corresponding to each of these cases is shown in Figure 3.

I



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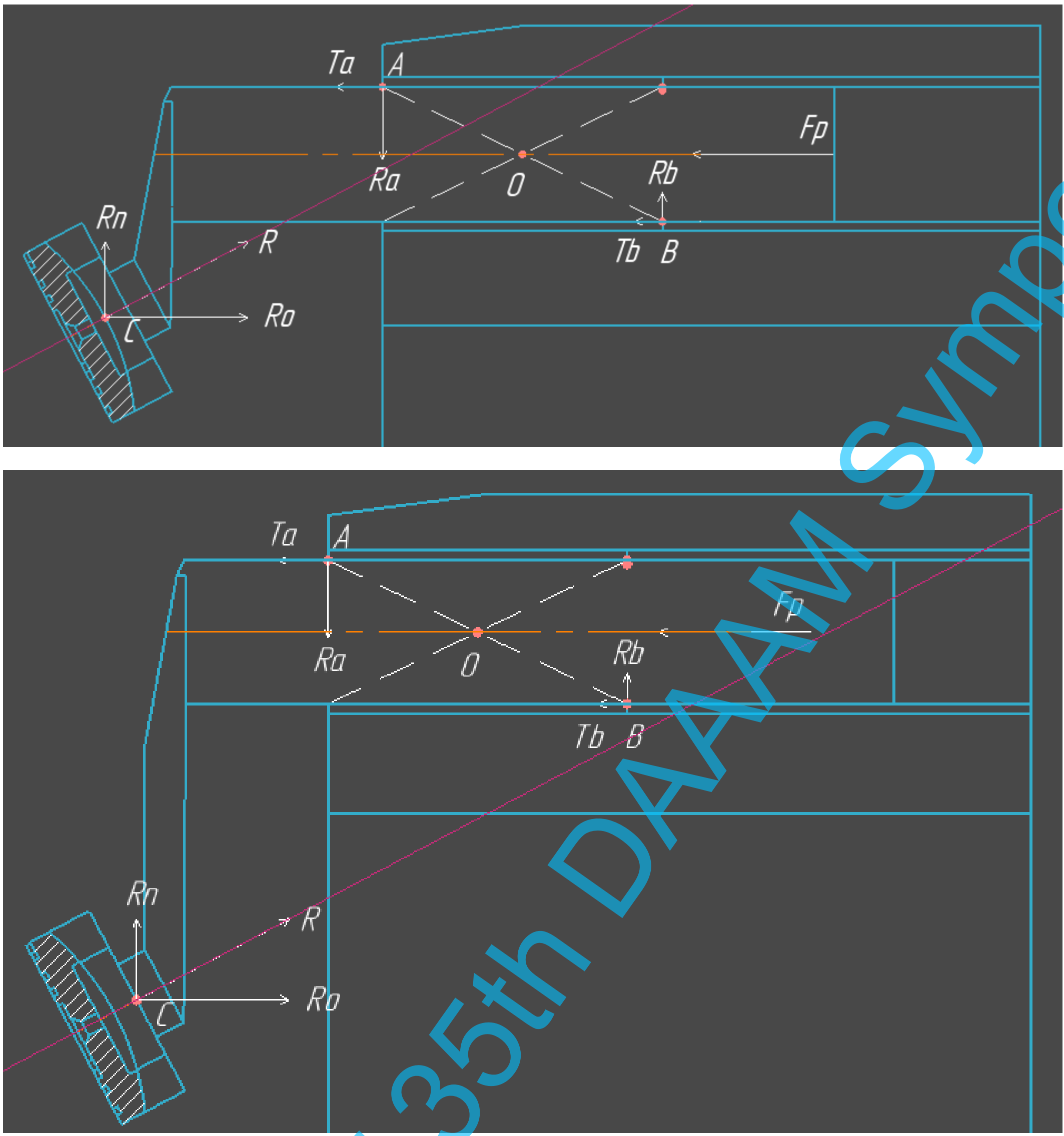

Fig. 3. Force diagram for 3 cases of piston position in a modified piston mechanism of the APGND

The equation of moments for each of these cases will take the form:

$$
\begin{aligned}
&M_{Rn} + M_{Ro} + M_{Ta} + M_{Tb} + M_{Ra} + M_{Rb} + M_{Fp} = 0 \\
&\Leftrightarrow Rn \cdot (l_{CO}) + Ro \cdot 0 - Ta \cdot r + Tb \cdot r - Ra \cdot l_{AO} - Rb \cdot l_{BO} + Fp \cdot 0 = 0 \text{ – I case} \\
&\Leftrightarrow Rn \cdot (l_{CA} + l_{AO}) - (Ta - Tb) \cdot r - (Ra + Rb) \cdot l_{AO} = 0 \text{ – II case} \\
&\Leftrightarrow Rn \cdot (l_{CA} + l_{AO}) - f \cdot (Ra - Rb) \cdot r - (Ra + Rb) \cdot l_{AO} = 0 \text{ – III case}
\end{aligned} \tag{1}
$$

Where:
Fp – the pressure force of the working fluid on the end surface of the piston;
R– the reaction force of the swash plate;
Ro – the axial component of the reaction force of the swash plate;
Rn – the transverse component of the reaction force of the swash plate;
Ra – the reaction force of the outer part of the guide sleeve;
Rb – the reaction force of the inner component of the guide sleeve;
Ta – the friction force of the outer part of the guide sleeve;
Tb – the reaction force of the inner part of the guide sleeve;
γ – the angle of swash plate;
C – the center of the hinge;
d - the diameter of the piston;
r – the radius of the piston;
O – the geometric center of the guide sleeve;
l_{CA} – the amount of piston departure;
l_{AB} – the length of the guide sleeve;

l_{AO} – the axial distance from the edge to the center of the sleeve;
f – the coefficient of friction.

The strength of Rn is always proportional to the magnitude of γ. Consequently, by increasing the values of γ, we also increase the force Rn, which, while maintaining the friction force constant, will lead to a decrease in the torque of the hydraulic machine, and, as a result, to a decrease in the value of the minimum stable revolutions of the hydraulic drive shaft.

2. Investigation of the limiting angle swash plate of an axial piston hydraulic machine with a modified piston mechanism

The limit value for the friction force in the piston-guide sleeve pair is the value of this force in the standard piston mechanism of an axial piston hydraulic machine with an angle of swash plate γ > 18°.
Operating pressure of the liquid P = 30 MПa. Then the force of the liquid pressure acting on the piston Fp is equal to:

$$Fp = P \cdot \pi \cdot r^2 = 30 \cdot 10^6 \cdot \pi \cdot 0.007^2 = 4618 \text{ (H)} \quad (2)$$

All parameters were calculated from formula 1, taking into account the two moving piston points on the left and right borders:

	Ra	Rb	Ro	Rn	Ta	Tb	T
l_{CAmax}	3564	1736	5625	1827	677	330	1007
l_{CAmin}	2013	366	5070	1647	382	70	452

Table 1. Initial parameters

In the modified piston mechanism, rc = 12 mm is the maximum possible without changing the diameter of the hydraulic drive shaft.
Taking into account the maximum values of all parameters obtained at different values of γ of the modification APGND, it was found that at a value of γ = 25 °, the values of the friction force T in the modified piston mechanism will be equal to the values of the friction forces in the standard piston mechanism with γ = 18 °.
If the value of γ is more than 25 degrees, the friction value will be greater than the standard mechanis,, which contradicts the criteria, and will not be used.
Figure 4 shows a graph comparing the friction force between the piston and the guide sleeve when the piston is moving. The dotted line is a standard piston mechanism, the solid lines are values corresponding to different angles of inclination of modified piston mechanism with the same rc value (in the case of rc = 12mm on the graph).

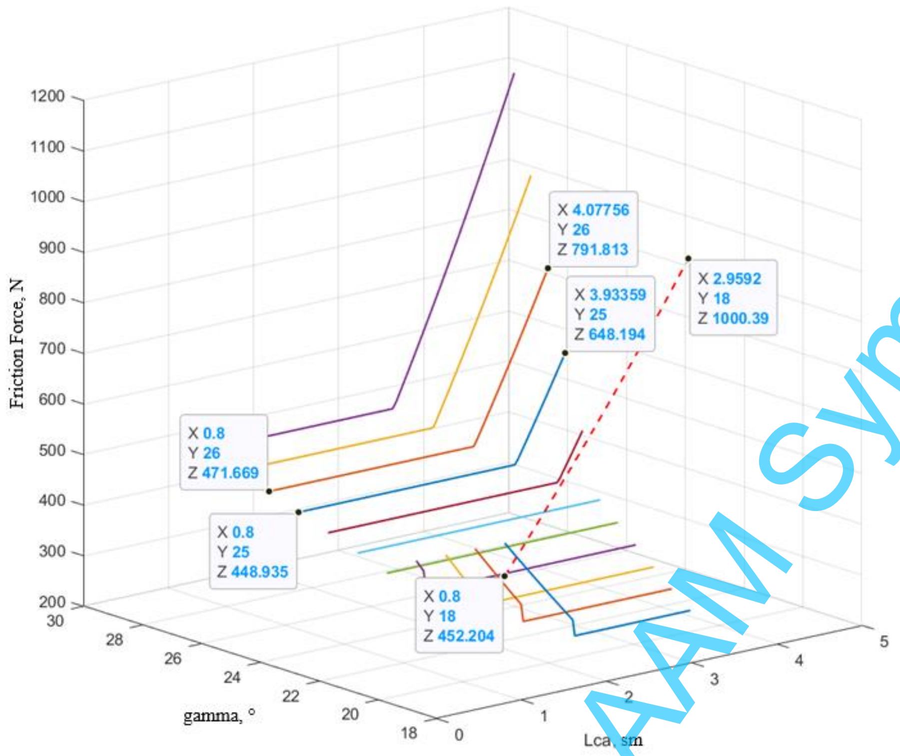

Fig. 4. The friction force in the piston-guide sleeve pair of the modified piston mechanism of the APGND at various γ at rc=12mm

Due to the displacement of the rc piston hinge, a constant value of the friction force is observed for almost all the working stroke of the modified piston mechanism [6].

It can be seen from the graphs in Figure 4 that at an angle of inclination of 25°, the friction force generated by the modified piston is always less than for a standard piston when considering the same piston position.

Figure 5 shows the torque from one piston acting on the main shaft on the work cycle.

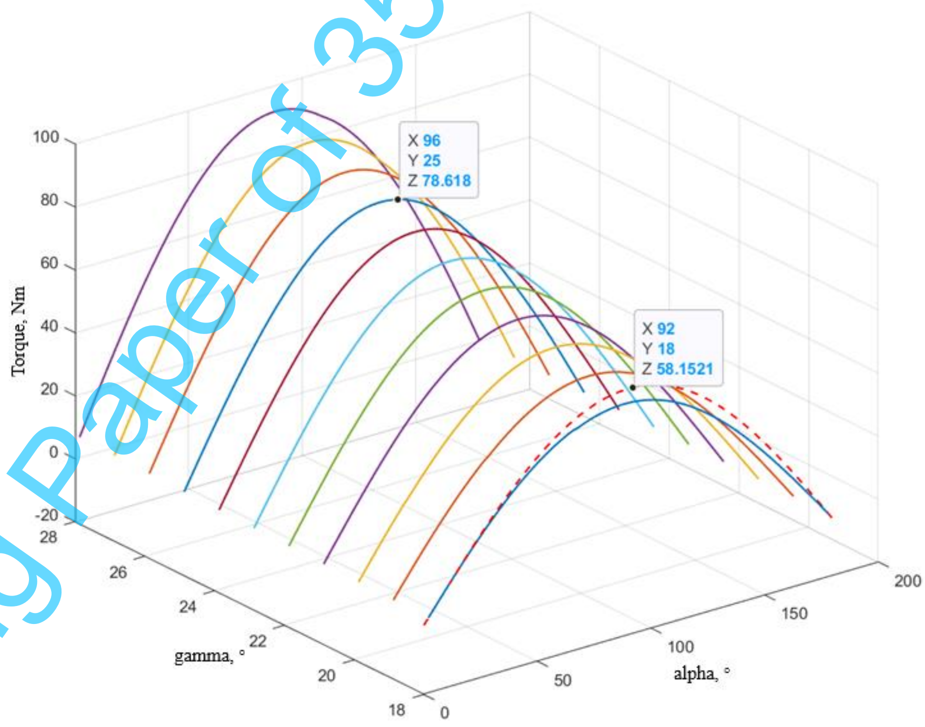

Fig. 5. The torque generated by a single piston mechanism of the APGND

Analyzing the obtained graphs, it can be concluded that when γ changes to 25°, the maximum torque compared to the initial case (γ = 18°) increased from 58 Nm to 78 Nm (35% higher).

Figure 6 shows the total torque on the drive shaft of the APGND generated by all (9) piston mechanisms.

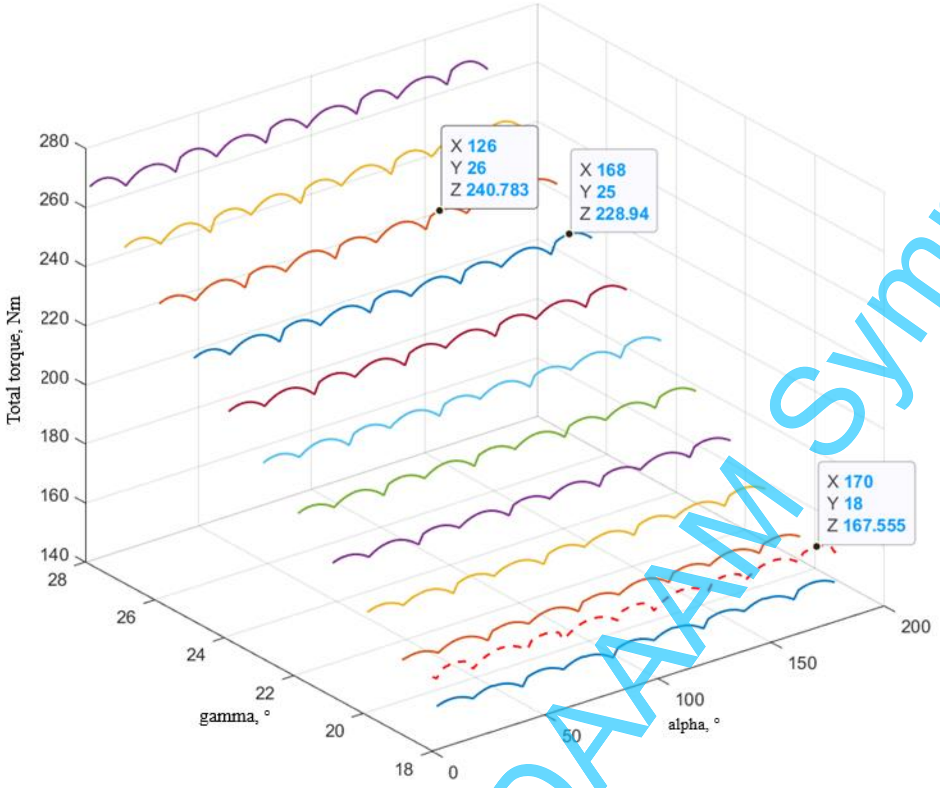

Fig. 6. The total torque generated by all (9) piston mechanisms of the APGND

The total torque also increased by 35% (from 167 Nm to 228 Nm).

3. Conclusion

From the results of the research, it can be concluded that with an increase in the γ to 25 °, the total torque on the drive shaft of an APGND with modified piston mechanisms increased by 35%, which in turn, with constant friction forces compared with piston mechanisms of standard design, will reduce the value of the minimum stable revolutions of the hydraulic drive shaft by approximately 30%.

Due to the decrease in the value of the minimum stable revolutions of the hydraulic drive shaft, the control range of the APG with modified piston mechanisms and the γ = 25 ° will increase by 43% compared with the APGND with standard piston mechanisms and the γ = 18 °.

In the future, it is planned to develop and investigate the design of the entire undercarriage of the APGND with modified piston mechanisms and the γ = 25 °. Also, to develop a set of design documentation for the manufacture of a prototype of an APGND with a modified piston mechanism.

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