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# Marine Steering Gears Mechanisms Efficiency: Status and Perspective for Increase

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*ABSTRACT*

The most common problem of modern marine energetic is an energy saving issue. Hydraulic steering gears driving pump units are one of the most significant loads of ship electric plant. For widespread ram-type steering gears the power of these pump units must overcome useful load of stern hydrodynamics torque and friction losses in lever transfer mechanisms. Friction losses in steering gears lever mechanisms can absorb from 10% to 20% of pump units power. Attempt to reduce these losses led to use rollers instead sliders in «ram-tiller» connections. Analysis presented in article illustrates low efficiency of this design solution. Study of friction losses sources demonstrates that its maximum arises from ram gasket friction. Reserve to decrease gaskets losses is transition from ram-type to piston-type stock equipped construction. Increasing of ram-type steering gear technical level need to change the type of mechanism for transformation progressive plunger motion to rotating tiller motion from tangents-type to another.

## 1. Introduction.

Hydraulic steering gears (HSG) are one of the most important and expensive elements of modern ships, the costs of their construction and operation depend on its technical level. Failure-free HSG ensures stable controllability of the ship and safety of navigation, as ship accidents directly caused by HSG failures are not uncommon in maritime practice.

HSG driving is a source of significant load on the ship's power plant, and therefore an increase in the cost of transportation. Steering gears with a ram-type driving, where the load transfers from the rams to the tiller is implemented through a tangential lever mechanism with sliders (Fig. 1) or rollers (Fig. 2), are common on large mainline vessels due to their high loading capacity and rigidity. Such machines are characterized by significant disadvantages, particularly, the presence of mechanical losses in the lever mechanism, and significant weight due to the high load of parts, and therefore the high cost of manufacturing and operation.

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## 2. Literature Review and Problem Statement.

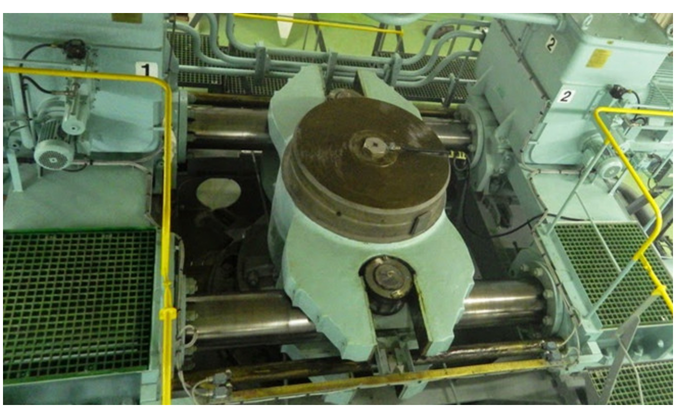
Modern research in the field of energy saving covers mainly the development of new sources and the use of waste energy in various structures [1, 2]. At the same time, an important problem is the reduction of losses in machines that are already in operation, which will allow to reduce the costs of their operation [3, 4]. The known researches of ram-type HSG devices are mainly limited to the problems of reducing the complexity of their installation and assembling [5], improving control [6], studying self-oscillating processes [7], improving the hydraulic system [8]. Other HSG disadvantages, including the imperfection of the lever system, particularly its low mechanical efficiency, remain outside the attention of researchers. The stated facts form the conditions for carrying out new research in this field.

The main goal of the work is to establish the main directions and estimate reserves for improving the technical level of ram-type HSG based on the analysis of the energy efficiency of their lever mechanisms.

Figure 1: General view of YOOWON-MITSUBISHI YDFT-335-2 steering gear with sliders equipped mechanism.

Source: Authors.

Figure 2: General view of FLUTEK-KAWASAKI FE 21-243 steering gear with rollers equipped mechanism.



Source: Authors.

## 3. Materials and methods.

The criterion characterizing the energy efficiency of the lever HSG mechanism is the mechanical efficiency coefficient, which is the ratio of useful and expended work per turn of the tiller.

$$\eta_m = \frac{A_u}{A_u + A_f}, \tag{1}$$

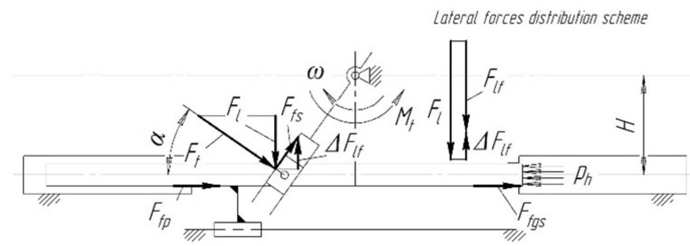
where:

$A_u$ – useful work per turn of the tiller;

$A_f$ – friction resistance forces work at the same period (friction losses).

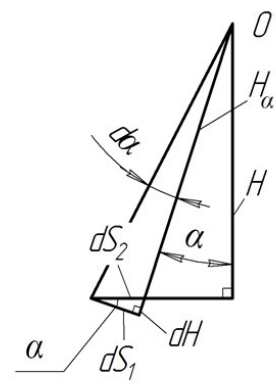
To determine the relevant works (1,) at the first stage, we will draw up a force diagram of the HSG mechanism with a slider (Fig. 3) and use the method of possible movements. When providing the tiller with a possible movement at an angle $d\alpha$, the forces acting in the HSG mechanism will perform work on the movements $dS_1$, $dS_2$, $d_H$ (Fig. 4).

Figure 3: Main parts force interaction scheme of steering gear sliders equipped mechanism.




Source: Authors.

Figure 4: Possible movements determination scheme.




Source: Authors.

Useful work is done by the tangential force on the tiller

$$F_t = \frac{M_t}{2H} \cos\alpha \tag{2}$$

where:

$\alpha$ – tiller rotation angle (stern angle position);

$M_t$ – tiller torque (taking into account the hydrodynamic resistance to turning the stern and losses in the baler bearings).

The list of friction resistance forces, expressions for their determination and possible movements on which they perform work are given in table. 1.

When compiling the table. 1, we take into account that the friction force $F_{fs}$ (Fig. 3) between the slider and the tiller partially relieves the rams from the action of the lateral force, and the force is transmitted to them determinates as:

$$\begin{aligned} F_{lf} &= F_l - \Delta F_{lf} = F_t \sin\alpha - F_t f_s \cos\alpha = \\ &= F_t(\sin\alpha - f_s \cos\alpha). \end{aligned} \tag{3}$$

Table 1: Forces performing work in the HSG sliders equipped mechanism.

| Force name and designation | Calculation expression | | Possible movement | |
|---|---|---|---|---|
| ***Useful resistance force*** | | | | |
| Tangential force $F_t$ | (2) | | $dS_1 = H_\alpha d\alpha = \frac{H}{\cos\alpha} d\alpha$ | (4) |
| ***Friction resistance forces*** | | | | |
| Slider in tiller groove friction force $F_{fs}$ | $F_{fs} = F_t f_s$ | (5) | $dH = dS_1 tg\alpha$ | (6) |
| Ram in cylinder sleeve friction force $F_{fp}$ | $F_{fp} = F_f f_p = = F_t f_p(\sin\alpha - f_s \cos\alpha)$ | (7) | $dS_2 = \frac{dS_1}{\cos\alpha}$ | (8) |
| Ram in cylinder gasket friction force $F_{fgs}$ | $F_{fgs} = \pi d_p h p_{gs} f_{gs}$ | (9) | $dS_2$ | |
| Ram in cylinder sleeve weight initiated force $F_{fG}$ | $F_{fG} = G f_p$ | (10) | $dS_2$ | |
| Slider on axe friction force $F_{fj}$ | $F_{fj} = F_t f_j$ | (11) | $0{,}5 d_j d\alpha$ | (12) |
| Notations in expressions | $f_p$, $f_{gs}$, $f_s$, $f_j$ – friction coefficients, respectively, in pairs: «ram-cylinder sleeve», «ram-gasket», «slider-tiller groove», «slider-axe»; $h$ – ram gaskets height, $p_{gs}$ – ram gaskets tightening pressure (equals 80% of pressure $p_h$ in hydro system circuit, $p_{gs} = 0{,}8\ p_h$); $G$ – ram weight; $d_p$ – ram (plunger) diameter; $d_j$ – slider (roller) axe diameter. | | | |

$$\eta_m = \frac{F_t dS_1}{F_t dS_1 + F_{fs} dH + F_{fp} dS_2 + F_{fgs} dS_2 + F_{fG} dS_2 + 0{,}5 F_t f_j d_j d\alpha}; \tag{13}$$

$$\eta_m = \frac{1}{1 + \left[ f_s tg\alpha + \frac{f_p}{\cos\alpha}(\sin\alpha - f_s\cos\alpha) + \frac{\pi d_p h p_{gs} f_{gs} + G f_p}{F\cos\alpha_t} + \frac{0{,}5 f_j d_j \cos\alpha}{H} \right]}; \tag{14}$$

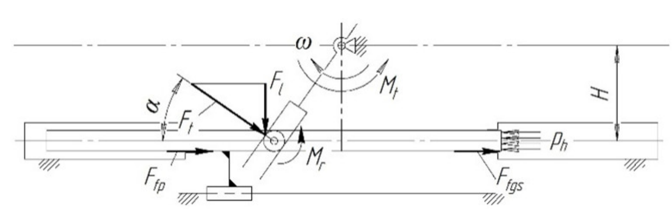
$$\eta_m = \frac{\cos\alpha}{\cos\alpha + f_s\sin\alpha + f_p(\sin\alpha - f_s\cos\alpha) + \frac{\pi d_p h p_{gs} f_{gs} + G f_p}{F_t} + \frac{0{,}5 f_j d_j \cos^2\alpha}{H}}. \tag{15}$$

Source: Authors.

Calculating the work of the forces of useful and friction resistance as the product of these forces and their displacements, we present expression (1) for the mechanical efficiency of the lever mechanism with sliders in the form (13)...(15). Expressions (14) and (15) are convenient for further analysis and are obtained with relation (14) after transformations, where $dS_2$, $d\alpha$, dH are expressed through $dS_1$, the numerator and denominator are divided by $F_t dS_1$ and multiplied by cosα.

For the HSG with rollers equipped mechanism (Fig. 5), the friction resistance forces and possible displacements are given in the table. 2.

Figure 5: Main parts force interaction scheme of steering gear rollers equipped mechanism.



Source: Authors.

Taking into account the above ratios, the expression for determining the mechanical efficiency of the rollers equipped mechanism can be written as (20).

Expressions (21) and (22) are convenient forms of relation (20) for practical use.

Table 2: Forces performing work in the HSG sliders equipped mechanism.

| Force name and designation | Calculation expression | | Possible movement | |
|---|---|---|---|---|
| The rollers in the tiller grooves rolling resistance torque $M_r$ | $F_{ft} = F_t \mu_r$ | (16) | $d\varphi = \dfrac{dH}{\pi d_r} = \dfrac{dS_1 tg\alpha}{\pi d_r}$ | (17) |
| Ram in cylinder sleeve friction force $F_{fp}$ | $F_{fp} = F_l f_p = F_t f_p \sin\alpha$ | (18) | $dS_2$ | |
| Ram in cylinder gasket friction force $F_{fgs}$ | (9) | | $dS_2$ | |
| Ram in cylinder sleeve weight initiated force $F_{fG}$ | (10) | | $dS_2$ | |
| Roller on axe friction force $F_{fj}$ | (11) | | $0,5 d_j d\varphi$ | (19) |
| New notations in expressions | $d_r$ – roller diameter; $\mu$ – rolling resistance coefficient. | | | |

Source: Authors.

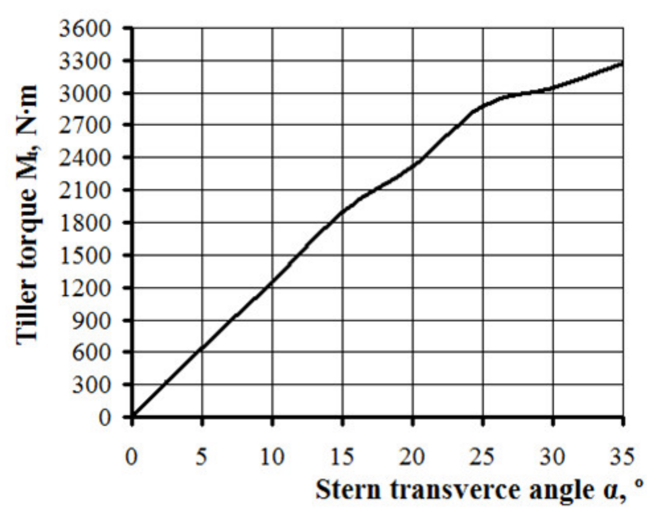
$$\eta_m = \frac{F_t dS_1}{F_t dS_1 + M_r d\varphi + F_{fp} dS_2 + F_{fgs} dS_2 + F_{fG} dS_2 + 0,5 F_t f_j d_j d\varphi}; \tag{20}$$

$$\eta_m = \frac{1}{1 + \left[ f_p tg\alpha + \dfrac{\pi d_p h p_{gs} f_{gs} + G f_p}{F_t \cos\alpha} + \dfrac{tg\alpha}{\pi d_r}(\mu_r + 0,5 f_j d_j) \right]}; \tag{21}$$

$$\eta_m = \frac{\cos\alpha}{\cos\alpha + f_p \sin\alpha + \dfrac{\pi d_p h p_{gs} f_{gs} + G f_p}{F_t} + \dfrac{\sin\alpha}{\pi d_r}(\mu_r + 0,5 f_j d_j)}. \tag{22}$$

Quantitative analysis of the obtained results carried out on the HSG example for the container ship with a 2842 TEU capacity ($DWT$ = 39374 t, $l$ = 222 m, $B$ = 30 m, $V$ = 22,6 kn) for which the moment on the tiller was calculated [9] at angles of stern angle position α = 0...35º (Fig. 6). The main design parameters of HSG are following: $d_p$ = 300 mm, $2H$ = 1700 MM, $h = d_p$ = 300 MM, $d_r$ = 280 MM, $d_j$ = 100 MM, $f_p = f_s = f_j$ = 0,1, $f_{gs}$ = 0,03 [10], $\mu$ = 0,02 mm, ?$_h$ = 25 MPa, $G$ = 1600 kg.

Figure 6: Graph of the dependence of the tiller torque on the angle of stern rotation.



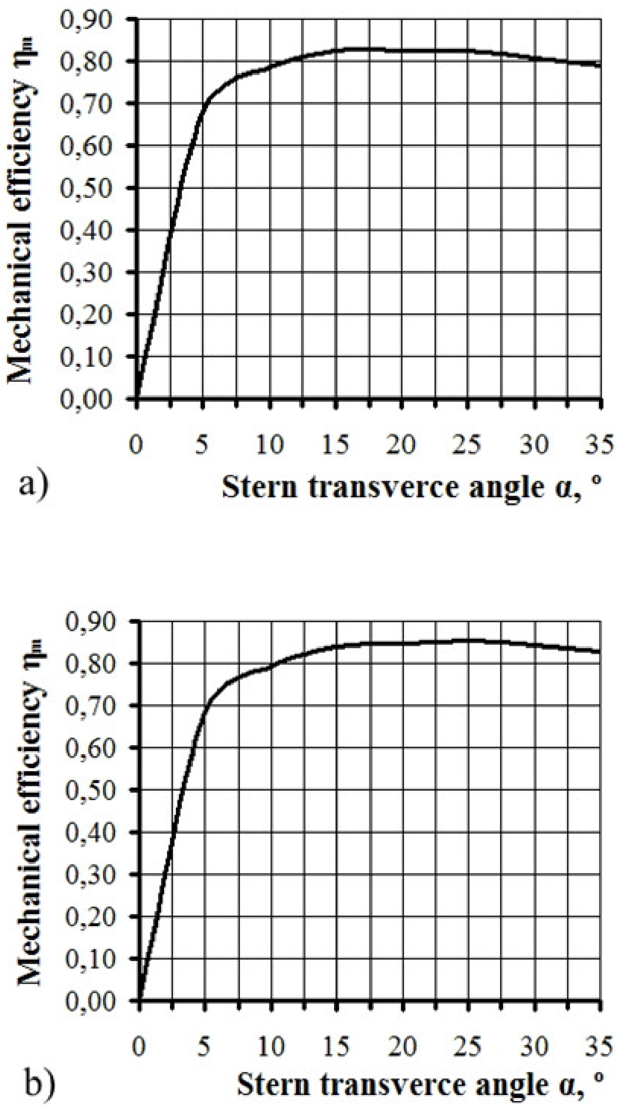

Source: Authors.

## 4. Results.

Taking into account the above results for Mt, graphs (Fig. 7) for lever mechanisms with sliders (a) and rollers (b) were obtained according to expressions (15) and (22).

From the graphs in fig. 7 it can be concluded that under the most favorable conditions (α = 20°) the mechanical efficiency reaches values of only 82% for the mechanism with sliders and 85% for the mechanism with rollers, that is, 18% and 15% of the power supplied to the HSG mechanism is spent on overcoming frictional losses. The use of rollers instead of sliders increases the mechanical efficiency by only 3%.

Figure 7: Graphs of mechanical efficiency coefficient for HSG mechanism equipped with sliders (?) and rollers (b).

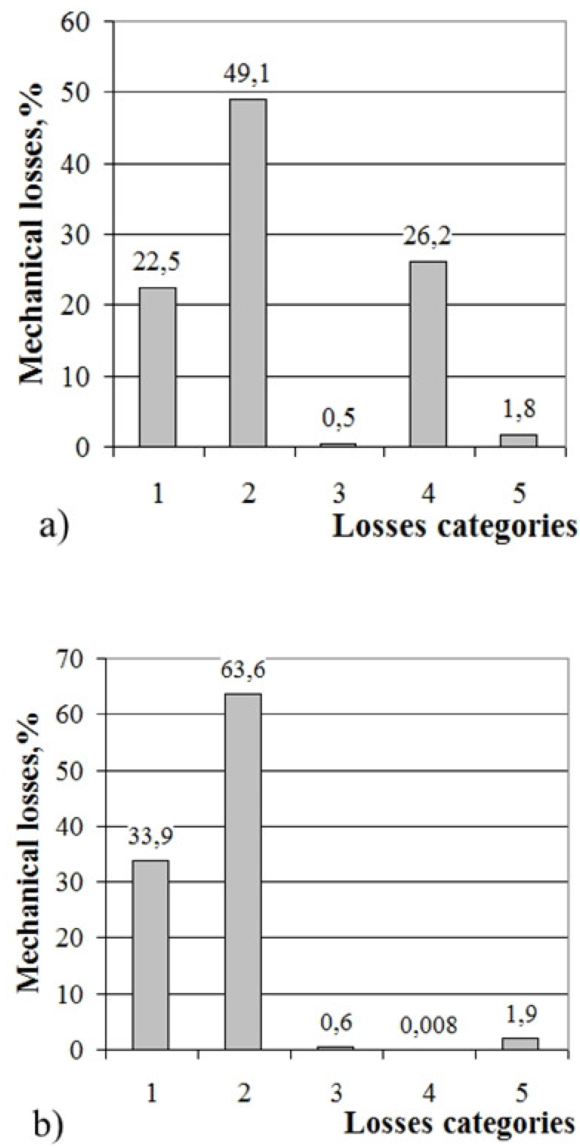



Source: Authors.

In fig. 8 shown the diagram of the distribution of the total volume of work of mechanical losses (100%, calculated according to the expressions in square brackets of formulas (14) and (21)) for mechanisms with sliders (a) and rollers (b) at the angle of stern position α = 35º according to the categories given in the table. 1 and table. 2 (they are marked: 1 – friction losses of plungers in sleeves; 2 – friction losses of plungers in gaskets; 3 – friction losses due to the weight of plungers in sleeves; 4 – friction losses of sliders/rollers in the tiller groove; 5 – losses on friction of sliders/rollers on their axes).

The diagram in fig. 8 shows that the largest volume of mechanical losses is losses due to friction of the plunger in the gaskets (49% and 63%, respectively), to the friction of the plungers in the sleeves (22.5% and 33.9%), which illustrates the directions for increasing the efficiency of the HSG mechanism due to reduction of losses in these categories. Thus, reducing the tightening pressure of the gaskets, for example due to the transition from a plunger design to a piston design (as a result, the pressure in front of the seals can be reduced), makes it possible to significantly increase the efficiency (Fig. 9). Thus, when the pressure pgs is reduced from 20 MPa to 2 MPa, the mechanical efficiency can be increased by 7% for mechanisms with sliders or by almost 10% for mechanisms with rollers.

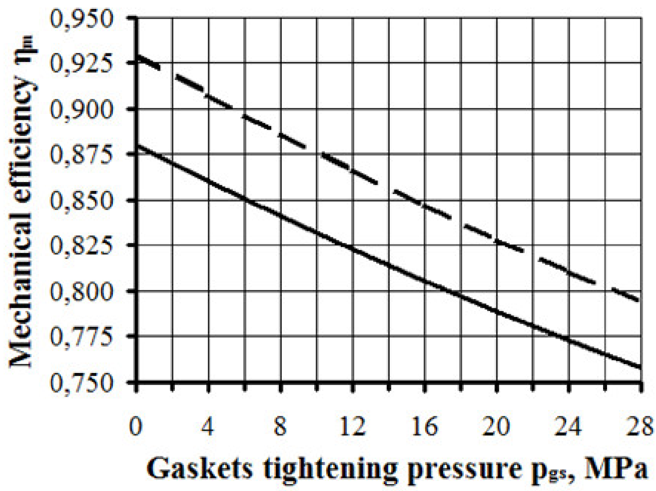
Figure 8: Diagrams of total friction losses by categories distribution for: sliders equipped mechanism (a) and rollers equipped mechanism (b).




Source: Authors.

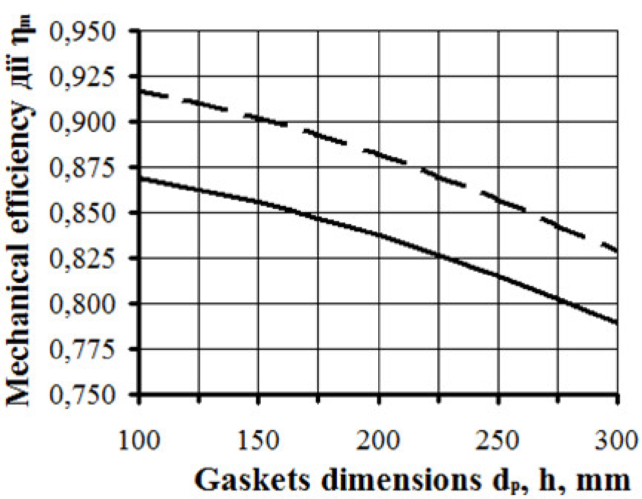
The same losses can be additionally reduced by reducing the diameter and width of the gaskets (when switching to a piston design by reducing the diameter of the piston rod) - by reducing the diameter and width of the gaskets (Fig. 10) from 300 mm to 150 mm (at $p_h$ = 25 MPa), the efficiency can be increased by 7% for mechanisms of both designs.

Figure 9: Mechanical efficiency-gaskets tightening pressure graphs for sliders (solid line) and rollers (dashed line) equipped mechanisms.




Source: Authors.

Figure 10: Mechanical efficiency-gaskets dimensions graphs for sliders (solid line) and rollers (dashed line) equipped mechanisms.




Source: Authors.

The transition from a plunger to a piston design, however, is possible only by reducing the lateral load on the plungers (piston rods), which is impossible with the tangential HSG mechanism used in this design.

## Conclusions.

The main feature of modern ram-type HSG devices, which are common on ships, is the use of a tangential lever mechanism, which ensures the transformation of the translational movement of the rams into the rotary movement of the tiller, while:

- the occurrence of significant lateral forces transmitted from the tiller to the rams and reaching about 53% of the tangential forces on the tiller (at stern angle position $\alpha = 35°$) is ensured, which requires the use of plungers of a significant diameter and mass;

- significant lateral loads require the use of large-diameter rams, which causes perceptible frictional losses of plungers in gaskets - among all frictional losses in the HSG mechanism, these losses make up 49% and 63% for mechanisms with sliders and rollers, respectively.

- it is shown that the reserve for reducing losses in gaskets is: first, the transition to a piston design, which will make it possible to reduce the pressure of gaskets tightening. Thus, when this pressure is reduced from 20 MPa to 2 MPa, the mechanical efficiency of the lever HSG mechanism can be increased by 7% for mechanisms with sliders or by almost 10% for mechanisms with rollers. Secondly, losses in seals can be reduced by reducing their diameter and width (when switching to a piston design by reducing the diameter of the ram and turning it into a piston rod). Thus, when the diameter and width of the gaskets are reduced from 300 mm to 150 mm (at $p_h = 25$ MPa), the efficiency of the lever HSG mechanism can be increased by 7% (for mechanisms of both designs). The transition to a piston design requires a reduction of lateral loads on the plungers.

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