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STUDY OF THE EFFICIENCY FACTOR OF SLIDING BEARINGS OF THE NEW CONSTRUCTIVE SOLUTION OF MULTISTAGE REDUCER

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Abstract. The article deals with the issues of quantitative assessment of the internal resistance of oil in sliding bearings of the new constructive solution of two-line three-stage reducer intended for use in transmission mechanisms of machines and devices. Based on the analysis of possible kinematic diagrams of the new constructive solution of reducer, it was determined that the directions of rotation of the driving and driven shafts coincide with the directions of rotation of block gears mounted on sliding bearings. In this case, the resistance in the sliding bearings becomes a useful driving force, which becomes a useful factor characterizing the internal driving force of the oil. This improves the reliability of the mechanism, as well as significantly reduce the energy consumption of the mechanical transmission. Based on the results of calculations, it was determined that the efficiency factor of the new constructive solution of a three-stage reducer is about 2% higher than in classic reducers.

Keywords: *new constructive solution, sliding bearings, double gear, block, driving force, efficiency factor, oil layer, angular velocity, resistance force.*

Introduction. The economic indicators of machines and devices largely depend on the correct choice of kinematic schemes of their constituent mechanisms. Usually 70% of the cost of machines and equipment is the material from which its structural elements are made. One of the components of machines and devices are their transmission mechanisms – reducers (which also known as gearbox) or multipliers [1].

Gear reducers used as a transmission mechanism are widely used in almost all areas of modern technology. The production of reducers that can transmit up to 10 000 kW is estimated at several million pieces per year. Currently, new designs of reducers are being developed in mechanical engineering, as well as methods for calculating the strength of their structural elements are being refined and improved.

At present, in order to save metal used in the production of three-stage gearboxes and to reduce their overall dimensions, the main direction of the development of this mechanical system is the placement of an intermediate shaft under its drive shaft. With this design, the length of the gearbox is reduced and this to some extent affects its price (Figure 1).

Based on research, it was determined, that at designing of machines and devices, including multi-stage reducers (gearboxes), the main geometric dimensions of their constituent structural elements are determined by the existing operating criteria in a chaotic manner without taking into account the influence of exciting factors. And this in turn, leads to different values of the reliability of structural elements, an increase in their metal consumption (and dimensions), as a result of which the technical level of the mechanical system decreases. Therefore, at designing of modern multi-stage gearbox (multiplier) that is resistant to market competition, along with the constructive changes introduced into them, it is necessary to take into account the provisions of the principles of the system approach.

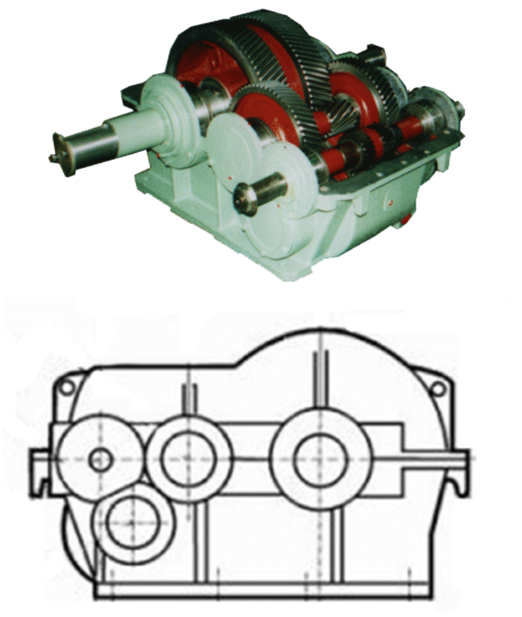


Figure 1. Improved two-line three-stage reducer

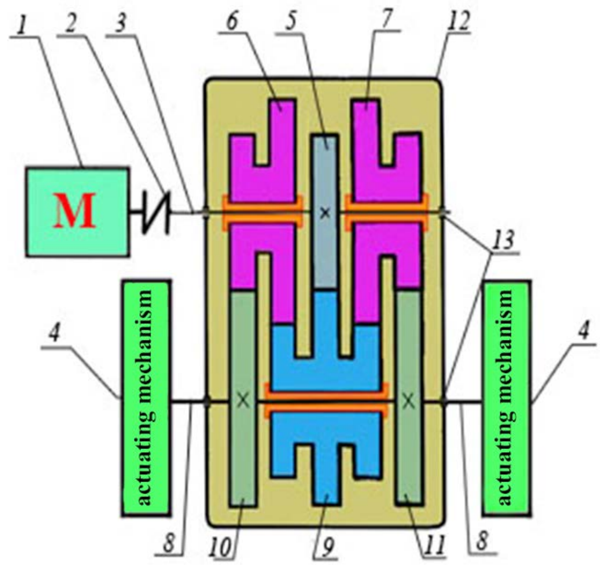


Figure 2. New constructive solution of two-line three-stage reducer which has only two shafts

Formulation of the problem. Increasing the number of stages in classic multistage helical gear reducers increases the number of gears, countershafts, and accordingly the number of bearings. In turn, increasing the number of structural elements of the reducer leads to a reduction of its reliability and efficiency, and to increase overall dimensions. To effectively solve this problem, at the department of “Mechatronic and machine design” of Azerbaijan Technical University developed, researched and tested a new principle of designing structural elements of the multistage one- and two-line cylindrical gear reducer system, the novelty of which was approved by the Eurasian Patent (№017053) [2].

Fig. 2 shows a kinematic diagram of a new two-line three-stage reducer. The drive shaft 3 of the gearbox is rigidly connected to the electric motor 1 by coupling 2, and the driven shaft 8 is rigidly connected to effector (disks) 4 of the machine. The drive gear 5 is rigidly attached to the drive shaft 3, the double gear blocks 6 and 7 rotate freely around the axis of the drive shaft. The driven gears 10 and 11 are rigidly attached to the driven shaft 8, and the three-gear block 9 rotates freely around the axis of the driven shaft. The drive and driven shafts are mounted on 15 rolling bearings mounted on the housing 16.

The reducer is equipped with double and triple gear blocks, which are located along the length of the driving and driven shafts, which freely rotate around the axes of these shafts and form the following stages, excluding the intermediate shafts from the design, and their supports (bearings). The principle of operation of the proposed gearbox is as follows: the movement from the drive shaft, which receives rotary motion from the electric motor 1, with the help of the drive gear 5 rigidly mounted on it, is transmitted to the three-crown block gear 9 freely mounted on the driven shaft.

Operation principle of proposed reducer is follows: the movement from the drive shaft, which receives a rotational movement from the electric motor 1, with the help of the drive gear 5 rigidly installed on it, is transmitted to the triple gear block 9, which freely mounted on the driven shaft. In this case, although the angular velocities of the block gears and the corresponding drive and driven shafts are different, their directions coincide and give a push to each other. Triple-gear block 9 with the clutch transmits rotational motion to double-gear blocks 6, 7 which are mounted on the drive shaft. And the double-gear blocks 6, 7 freely rotating around the axis of the drive shaft transmit the rotary motion to the gears 10, 11, rigidly fixed to the driven shaft.

In this case, the direction of their rotational speeds is the same, since both the driven gears 10, 11 and the drive shaft 8, as well as the drive gear 5 and the drive shaft 3 are rigidly connected.

To carry out a comparative analysis of classical multistage gearboxes with a new constructive solution of multistage gearbox, the kinematic characteristics of each of them are preliminarily determined.

Total transmission number for a classic multi-stage reducer [4;5]:

$$u_{\Sigma} = u_1 \cdot u_2 \cdot u_3, \cdots u_k = \frac{z_2 \cdot z_3 \cdots z_k}{z_1 \cdot z_2' \cdots z_{k-1}'} \tag{1}$$

Since new constructive solution of the multi-stage gearboxes $u_1 = u_2 = u_3 = \ldots = u_k$, the total transmission number for them is expressed as follows

$$u_{\Sigma} = u_p^k = \left(\frac{z_2}{z_1} \right) \tag{2}$$

there u_1, u_2, u_3, u_k - gear ratio of the respective stages;

k_p – number of stages.

The new constructive solution of the three stage reducer consists of a drive shaft, a drive gear, two double-gear and one triple-gear blocks and driven shaft. The small gear of the first stage is rigidly fixed to the drive shaft. And double-gear and triple-gear blocks are mounted on sliding bearings on the shaft. These double-gear and triple-gear blocks rotate in the rotation direction of the drive and driven shafts, which in turn, to some extent, reduces the value of the resistance moment.

The proposed new constructive solution of the three stage reducer is suitable in length and height to fit with coaxial three-stage spur gearboxes. The symmetrical arrangement of the stages in the new construction allows for a more compact design than in the classic three-stage gearboxes. This gearbox has a drive shaft that acts as an axle for the second stage. In this version, the mechanical system excludes two shafts, two “shaft-hub” connections and four rolling bearings, which reduces the number of structural elements and increases the reliability of the gearbox.

The new constructive solution of the gearbox with five or more single stages differs in that even in these gearboxes, despite the increase in the number of double-gear blocks, its previous length and height remain unchanged, but the width increases.

Depending on the number of stages K_p, the number of double-gear blocks K_b is determined as follows:

$$K_b = K_p - 1 \tag{3}$$

As can we see, the proposed new constructive solution of the gearboxes are more compact than classical gearboxes, and in all cases they allow to form more successful gearboxes for a given mechanical system. One of the important tasks to reduce the cost of repair and downtime of a new multi-stage gearboxes is to increase their reliability.

Solution of the problem. Due to structural and economic considerations, sliding bearings are used instead of rolling bearings as supports for the double-gear blocks of the new constructive solution of the gearboxes.

These bearings are embedded under the gear blocks which are placed on the shaft and rotate freely around its axis.

Based on the analysis of possible kinematic diagrams of the new constructive solution of reducer, it was determined that the directions of rotation of the driving and driven shafts coincide with the directions of rotation of block gears mounted on sliding bearings. In this case, the resistance in the sliding bearings becomes a useful driving force, which becomes a useful factor characterizing the internal driving force of the oil. It improves the reliability of the mechanism, as well as significantly reduces the energy consumption of the mechanical transmission.

The pressure created in the fluid layer by friction of one surface against another is determined according to the Reynolds equation as follows [6]:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\rho}{\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3 \frac{\rho}{\mu}\frac{\partial p}{\partial z}\right) = $$
$$= 12\rho V + 6\frac{\partial}{\partial x}(\rho U h) + 6\frac{\partial}{\partial z}(\rho W h) + 12h\frac{\partial \rho}{\partial t} \tag{4}$$

there U, V, W - components of the velocities of one surface on the other surface along the axes OX, OY and OZ, respectively; μ_0 - dynamic viscosity of oil; ρ_y - oil density; h - the thickness of the oil layer in the area under consideration.

If we apply this expression to a normal cylindrical sliding bearing, the expression will be much simpler. Thus, if the lubrication process is carried out by an incompressible oil, then at $t = const$ its density is equal to $\rho_y = const$ and $\frac{\partial \rho_y}{\partial t} = 0$. In addition, if the temperature and pressure of the liquid change insignificantly along the length of the liquid, then the value of μ_0 can be averaged and considered constant. In steady-state operation at a constant rotation of journal, the rotational speed V in the direction of the OX axis will be constant: $V = const$. On the other hand, since the center of the journal does not change its position $e = const$, then the rotation speed V in the direction of the OY axis will be zero. Since the fluid flow velocity is directed along the bearing end surfaces, its velocity in the direction of the OZ axis is less than in the direction of rotation, therefore, $W = 0$ can be taken.

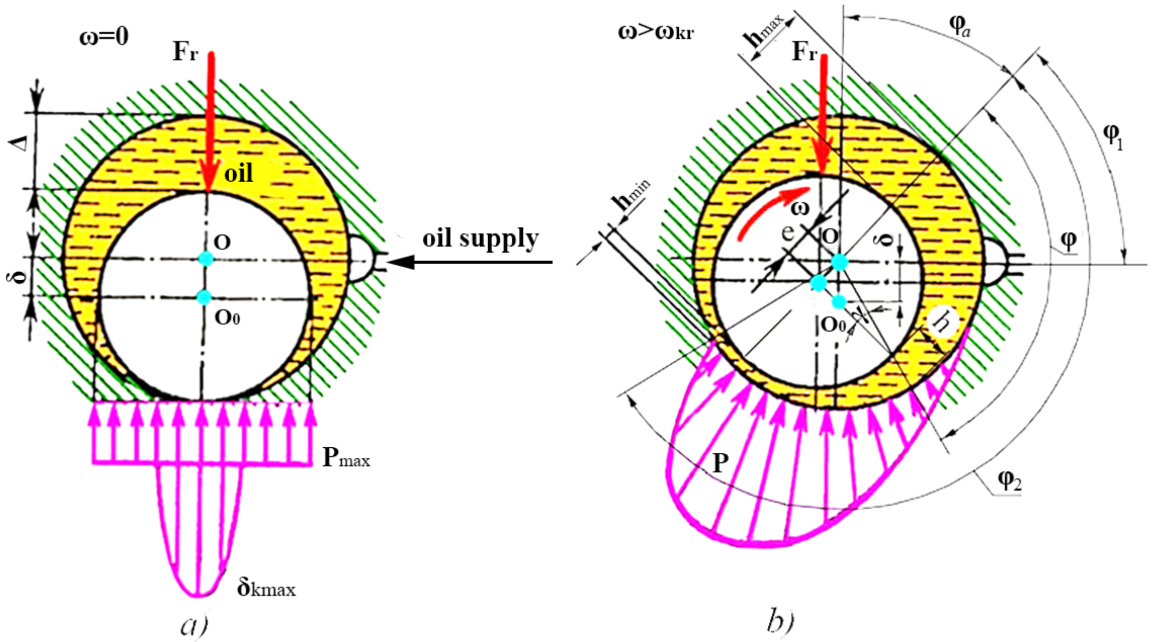


Figure 3. Scheme for determining the pressure at any arbitrary section of the oil layer.

If we take these conditions into account in expression (4), then Reynolds's expression will be greatly simplified as follows [6]:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right) = 6\mu_0 Uh \quad (5)$$

Let's integrate this expression, then:

$$h^3\frac{dp}{dx} = 6\mu_0 Uh + C \quad (6)$$

To determine the integral constant, it is assumed that at a given thickness of the oil layer ($h = h_m$) the pressure takes its maximum value and becomes $\frac{dp}{dx} = 0$ when $p = p_{max}$ (Figure 3). Then from the expression (6):

$$C = -6\mu_0 Uh_m$$

If we consider this expression in (6), then [9]:

$$\frac{dp}{dx} = 6\mu_0 U\frac{h - h_m}{h^3} \quad (7)$$

To make it easier to determine the pressure (p) in an arbitrary section, we substitute $dx = rd\varphi$ and $U = \omega r$.

The thickness of the fat layer between the journal and the backing is determined by the following expression [6]:

$$h = \delta - e\cos(180^0 - \varphi) \quad (8)$$

there $\delta = 0{,}5s$- radial bearing clearance; $s = D - d$ - diametral clearance; D, d - respectively the diameters of the bearing and the journal.

If we change $\frac{e}{\delta} = \chi^*$ in this formula, then

$$h = \delta(1 + \chi^*\cos\varphi) \quad (9)$$

there χ^*- relative eccentricity.

For the minimum thickness of the oil layer:

$$h_{min} = \delta(1 + \chi^* cos\varphi_m) \tag{10}$$

there φ_m- angle corresponding to maximum pressure.

If we consider these expressions in (7), then we get

$$\frac{dp}{d\varphi} = 6\frac{\mu_0\omega}{\psi^2}\frac{\chi^*(cos\varphi - cos\varphi_m)}{(1 + \chi^* cos\varphi)^3} \tag{11}$$

there $\psi = \frac{s}{d} = \frac{\delta}{r}$ - relative clearance.

If we integrate the expression (11) from φ_1 to φ' to determine the specific pressure in the section forming the angle φ' with the center lines, then

$$P_{\varphi'} = 6\frac{\mu_0\omega}{\psi^2}\int_{\varphi_1}^{\varphi'}\frac{\chi^*(cos\varphi - cos\varphi_m)}{(1 + \chi^* cos\varphi)^3}d\varphi \tag{12}$$

The pressure per unit area in the section under consideration is determined by the following expression:

$$\Delta P_{\varphi'} = P_{\varphi'} lr\Delta\varphi' = P_{\varphi'}\frac{ld}{2}\Delta\varphi'$$

In steady-state operation, the external load acting on the bearing unit is defined by the following expression [6; 8]:

$$F_r = \frac{3\mu_0\omega}{\psi^2} ld \int_{\varphi_1}^{\varphi_2} cos\left[\pi - \left(\varphi' + \varphi_a\right)\right]d\varphi' \int_{\varphi_1}^{\varphi'}\frac{\chi^*(cos\varphi - cos\varphi_m)}{(1 + \chi^* cos\varphi)^3}d\varphi \tag{13}$$

If we make a substitution in this formula

$$\Phi_F = 3\int_{\varphi_1}^{\varphi_2} cos\left[\pi - \left(\varphi' + \varphi_a\right)d\varphi'\right]\int_{\varphi_1}^{\varphi'}\frac{\chi^*(cos\varphi - cos\varphi_m)}{(1 + \chi^* cos\varphi)^3}d\varphi \tag{14}$$

then the load acting on the bearing unit is defined as follows: [4;7]:

$$F_r = \frac{\mu_0\omega dl}{\psi^2}\Phi_F \tag{15}$$

there Φ_F- bearing load factor; $l = (0{,}5 \div 1{,}5)d$- length of bearing; ω – summary angular velocity of corresponding shaft (journal) and sliding bearing.

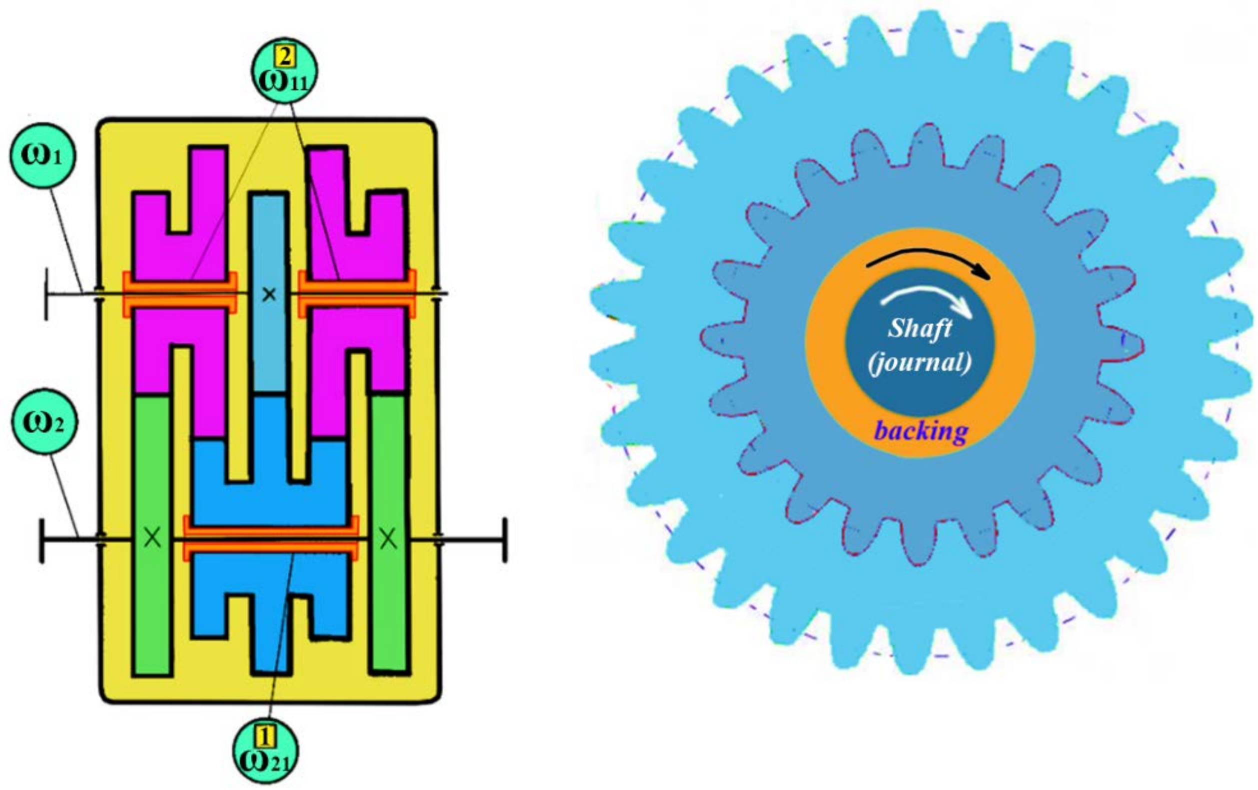
As already noted, the rotation directions of the drive and driven shafts at new constructive solution of the gearboxes coincide with the rotation directions of gear blocks mounted on sliding bearings. Consequently, the total angular velocity will be equal to the sum of the angular velocities of the shaft and the sliding bearings.

In the total angular velocity of the sliding bearings integrated under the gear blocks of the new constructive solution of three-stage two-line reducer that we are studying (Figure 4):

$$\left.\begin{array}{r} First\ gear\ block\ \omega_{21} = \omega_2 + \dfrac{\omega_1}{u} = \dfrac{\omega_1}{u_1^3} + \dfrac{\omega_1}{u_1} = \omega_1\left(\dfrac{1+u_1^2}{u_1^3}\right) \\ Second\ gear\ block\ \omega_{11} = \omega_1 + \dfrac{\omega_1}{u_2^2} = \omega_1\left(\dfrac{1+u_2^2}{u_2^2}\right) \end{array}\right\} \quad (16)$$

There - u_1 and u_2 - corresponding gear ratios of the first and second stages of reducer.

Before designing a new multi-stage gearbox taking into account the full use of the permissible load of the high-speed stage and proper lubrication of the gears it is usually assumed that the gear ratio of each stage of the gearbox is $u_1 = u_2 = u_3 = \cdots = u_n$.



b)

a)

Figure 4. The new constructive solution of multistage reducer
a - kinematic diagram; b – gear block

According to the expression (15) the relationship between Φ_F and the main parameters of the bearing under constant load can be determined by the following expression:

$$\Phi_F = \frac{F_r}{ld}\frac{\psi^2}{\mu_0\omega} = \frac{p\psi^2}{\mu_0\omega} \quad (17)$$

The load factor of the corresponding sliding bearings of the reducer:

$$\left.\begin{aligned}\Phi_{F1} &= \frac{(F_{r1}+F_{r2})\psi_1^2}{\mu_0\omega_{21}d_1l_1} = \frac{(F_{r1}+F_{r2})\psi_1^2u_1^3}{(1+u_1^2)\mu_0\omega_1d_1l_1}\\ \Phi_{F2} &= \frac{(F_{r2}+F_{r3})\psi_2^2}{\mu_0\omega_{11}d_2l_2} = \frac{(F_{r2}+F_{r3})\psi_2^2u_2^2}{(1+u_2^2)\mu_0\omega_1d_2l_2}\end{aligned}\right\} \tag{18}$$

Then the total load factor of the sliding bearings of the reducer:

$$\Phi_F = \Phi_{F1} + \Phi_{F2} = \\ = \frac{1}{\mu_0\omega_1}\left(\frac{(F_{r1}+F_{r2})\psi_1^2u_1^3}{(1+u_1^2)d_1l_1} + \frac{(F_{r2}+F_{r3})\psi_2^2u_2^2}{(1+u_2^2)d_2l_2}\right) \tag{19}$$

According to Newton's law, the frictional force that slides the oil layer in sliding bearings:

$$\left.\begin{aligned}F_{f1} &= \pi dl\mu_0\frac{v}{\delta} = \pi d_1l_1\frac{\mu_0\omega_{21}}{\psi_1} = \pi d_1l_1\frac{\mu_0\omega_1}{\psi_1}\left(\frac{(1+u_1^2)}{u_1^3}\right)\\ F_{f2} &= \pi dl\mu_0\frac{v}{\delta} = \pi d_2l_2\frac{\mu_0\omega_{11}}{\psi_2} = \pi d_2l_2\frac{\mu_0\omega_1}{\psi_2}\left(\frac{(1+u_2^2)}{u_2^2}\right)\end{aligned}\right\} \tag{20}$$

Total force used to repel friction in the sliding bearings of reducer:

$$P_f = \frac{\pi\mu_0}{2}\omega_1^2\left(\frac{l_1d_1^2}{\psi_1}\left(\frac{(1+u_1^2)}{u_1^3}\right)^2 + \frac{l_2d_2^2}{\psi_2}\left(\frac{(1+u_2^2)}{u_2^2}\right)^2\right) \tag{21}$$

The moment of frictional forces under the condition of integration of the driving force of the oil on the journal surface:

$$T_f = \frac{\pi\mu_0}{2}\omega_1\left(\frac{l_1d_1^2}{\psi_1}\left(\frac{(1+u_1^2)}{u_1^3}\right) + \frac{l_2d_2^2}{\psi_2}\left(\frac{(1+u_2^2)}{u_2^2}\right)\right) \tag{22}$$

Total efficiency factor of new constructive solution of reducer [2]:

$$\eta_\Sigma = \frac{P_2}{P_1} = \frac{P_1-P_f}{P_1} = 1 - \frac{P_{dç}+P_{dy}+P_f}{P_1} = \\ = 1 - \left(\psi_{ge}^* + \psi_{rb}^* + \psi_{sb}^*\right) \tag{23}$$

there ψ_{ge}^*- the factor coefficient taking into account the losses in the clutch; ψ_{rb}^* - coefficient taking into account the losses in the rolling bearings; ψ_{sb}^* - the total useful ratio, which characterizes the internal resistance force of the oil in the sliding bearings.

$$\psi_{sb}^* = \frac{P_f}{P_1} = \frac{\pi\mu_0\omega_1^2}{2P_1}\left(\frac{l_1d_1^2}{\psi_1}\left(\frac{(1+u_1^2)}{u_1^3}\right)^2 + \frac{l_2d_2^2}{\psi_2}\left(\frac{(1+u_2^2)}{u_2^2}\right)^2\right) \tag{24}$$

If the power in the drive shaft of the new multi-stage reducer is P_1, and the power in the outlet shaft without the sliding bearings is P_2, then the power in the outlet shaft of the reducer will be as follows, taking into account the positive effect of the internal resistance of the oil in the sliding bearings:

$$P_2^* = P_2 - \psi_{sb}^* P_2 = P_2(1 - \psi_{sb}^*) \tag{25}$$

If we take into account the friction loss when the gears are engaged (ψ_{ge}) and the friction loss in the roller bearings (ψ_{rb}), then the efficiency factor of the current mechanical system:

$$\eta^* = \frac{P_2^*}{P_1} = \frac{P_2(1 - \psi_{sb}^*)}{P_1} \tag{26}$$

$$\frac{\eta^*}{(1 - \psi_{sy}^*)} = \frac{P_2}{P_1} = 1 - \psi_{ge} - \psi_{rb}$$

from there

$$\eta^* = (1 - \psi_{ge} - \psi_{rb})(1 - \psi_{sb}^*) \tag{27}$$

Formulas (26) and (27) allows us to estimate the performance of the new three-stage reducer, as well as the energy loss in the transmission.

It is known that

$$1 - \psi_{ge} - \psi_{rb} = \eta_{ge} \cdot \eta_{rb} \tag{28}$$

$$(1 - \psi_{sb}^*) = \eta_{sb} \tag{29}$$

Then the efficiency factor of the new three-stage reducer:

$$\eta^* = \eta_{ge}^3 \eta_{rb}^2 \eta_{sb} \tag{30}$$

To quantify the internal resistance of the oil in the sliding bearings, we assume that the new constructive solution of reducer uses industrial oil И-30, and for this oil the dynamic viscosity at 60^0C is $\mu_0 = 0{,}014\, Pa \cdot sec$. The power of the electric motor of the converting mechanism of the pumping unit is $P_M = 7{,}5 kVt$, the rotational speed of its shaft is $n_M = 750\, min^{-1}$, so the angular velocity of the drive shaft of the reducer:

$$\omega_1 = \frac{\omega_M}{u_b} = \frac{\pi \cdot n_M}{30 \cdot u_b} = \frac{3{,}14 \cdot 750}{30 \cdot 1{,}6} = 49{,}06\, sec^{-1}$$

there $u_b = 1{,}6$ - is the transmission number of the belt drive applied in the converting mechanism of the pumping unit.

Assuming that the efficiency factor of the belt drive is $\eta_b = 0{,}96$, then the force on the gear shaft of the reducer:

$$P_1 = P_M \cdot \eta_b = 7{,}5 \cdot 0{,}96 = 7{,}2\, kVt$$

The new constructor assumes the diameter of the gear shaft under the bearing $d_1 = 50mm$, its length $l_1 = 50mm$, relative distance $\psi_1 = 0{,}002$, diameter of the drive shaft under the bearing $d_2 = 50mm$, its length $l_2 = 100mm$ and relative distance $\psi_2 = 0{,}003$. The coefficient of the total internal resistance of the oil in the sliding bearings is as follows:

$$\psi_{sy}^* = \frac{P_f}{P_1} = \frac{\pi \mu_0 \omega_1^2}{2P_1} \left(\frac{l_1 d_1^2}{\psi_1} \left(\frac{(1 + u_1^2)}{u_1^3} \right)^2 + \frac{l_2 d_2^2}{\psi_2} \left(\frac{(1 + u_2^2)}{u_2^2} \right)^2 \right) =$$

$$= \frac{3{,}14 \cdot 0{,}014 \cdot 49{,}06^2}{2 \cdot 7{,}2 \cdot 10^3} \left(\frac{0{,}05 \cdot 0{,}05^2}{0{,}002} \left(\frac{(1+4^2)}{4^3} \right)^2 + \frac{0{,}1 \cdot 0{,}05^2}{0{,}003} \left(\frac{(1+4^2)}{4^2} \right)^2 \right) =$$
$$= 0{,}007348(0{,}00441 + 0{,}0941) = 0{,}000724$$

If we take into account efficiency factor of gear drive $\eta_{ge} = 0{,}98$, efficiency factor of a pair of rolling bearing $\eta_{rb} = 0{,}99$, then the efficiency factor of new three-stage reducer:

$$\eta^* = \eta_{ge}^3 \eta_{rb}^2 (1 - \psi_{sb}^*) = 0{,}98^3 \cdot 0{,}99^2 (1 - 0{,}000724) = 0{,}9218$$

efficiency factor of reducer with classic construction:

$$\eta = \eta_{ge}^3 \eta_{rb}^4 = 0{,}98^3 \cdot 0{,}99^4 = 0{,}9041$$

Comparison of the efficiency of the new three-stage reducer with the efficiency of the classic three-stage reducer as a percentage:

$$\Delta\eta = \frac{\eta^* - \eta}{\eta} 100\% = \frac{0{,}9218 - 0{,}9041}{0{,}9041} 100\% = 1{,}96\ \%$$

Conclusions. As we can see, since in the new constructive solution of the gear-box the rotation direction of the sliding bearings and the rotation of the shafts coincide, the mutual displacement of the oil layers in the clearances of the sliding bearings, the partial contact of the roughness of the surfaces in contact with each other, and the displacement forces associated with the viscosity of the oil have a positive effect on the movement shafts and the efficiency factor of the mechanical system. Based on the results of calculations, it was determined that the efficiency factor of the new constructive solution of a three-stage reducer is about 2% higher than in classic reducers.

REFERENCES:

[1]. Vodeyko V.F., Efros D.G., Gear reducers (Russian). Moscow, MADI, 48 p., (2014).
[2]. Abdullaev A.H., Najafov A.M., Three-stage double-flow cylindrical gear, Patent №: 017053 B1, F16H 1/20 The Eurasian Patent Organization (EAPO), Moscow/Russia, Bulletin № 9, 4 p., (2012)
[3]. Najafov A.M., Abdullaev A.I., *About results of industrial testing of a three-stage double-flow reducer of the СКД 3-1,5-710 pumping unit*. Bulletin of NTU "KhPI". № 40, pp. 89-93, (2013).
[4]. Ivanov M.N., Finogenov V.A. Machine Elements (Russian). Moscow/Russia: “Visshaya shko-la“, 408 p., (2008).
[5]. Matlin M.M., Mozgunova A.I., Lebskiy S.L., Shandybina I.M., Basics of calculating parts and units of transport machines. Volgograd: VOLGTU, 251 p, (2010).
[6]. Chernavsky S.A., Sliding bearings. MASHGIZ, Moscow, 245 p., (1963).
[7]. Aristov A.I., Malysheva E.B., et al. Calculation of landings with a clearance (by the example of sliding bearings). MADI, 28 p., (2015).
[8]. Nazarenko Y.B., Hydrodynamics of sliding bearings and critical rotor speeds. Science, №3 (111), pp.16-18, (2017)
[9]. Balyakin V.B., Vasin V.N., Machine parts. Samara, 151 p., (2004)

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