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REVIEW AND COMPLEMENT OF METHODS FOR CHANGING THE MOVEMENT SPEED OF MECHANISMS AND MACHINES ELEMENTS

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Abstract. In mechanical drives of machines there is a need to control changes in the speed of their actuators. Stepped and stepless gearboxes are used for this purpose. Known speed control devices have many disadvantages that adversely affect the durability and reliability of drive components and machines in general. These include the design complexity, high material consumption, automation complexity, dynamic loads during transitions from one speed to another, intensive wear of parts due to the friction connections use. The purpose of the work is to develop an algorithm for determining the kinematic, power parameters and dimensions when designing speed change devices through the ring gear of a gear differential with a rotary stopper in the form of a closed-loop hydraulic system based on authors' previous computer-theoretical research and classical scientific advices. To solve these problems analytical expressions and graphs have been obtained for the relationship between speeds of gear differential links, the efficiency has been determined by the method of potential power – based on friction losses in each gearing. With the help of computer modeling of analytical expressions, using the MATLAB software, graphical dependences of efficiency have been obtained, which made it possible to evaluate the accomplishment of the gear differential in terms of energy consumption and possible self-braking. Based on Lagrange's theory, a dynamic model of a speed change device with a ring gear control has been constructed and a solution of the obtained system of equations has been proposed. The 3D modeling of the device has been executed and at the final choice of the optimum variant of model, after some specifications, development of technical documentation can be started. The results obtained have practical application at the stage of development and design of new speed control devices through the ring gear, allow to evaluate the operation of gear differentials in terms of energy consumption and self-braking and are the basis for further research. The graphical dependences obtained for the efficiency of the gear differentials clearly allow us to trace the change in the value of the efficiency depending on the angular velocity of the ring gear and the gear ratio. For the first time, analytical expressions were obtained to determine the efficiency of the gear differential of a speed-changing device with a driving sun gear, driven carrier, or vice versa, more accurately. The resulting graphical dependences for efficiency visually allow to trace change of efficiency value depending on angular speed of a ring gear, as a control link, and the gear ratio. Results are recommended for introduction into design and engineering practice at development of designs of speed change devices through differential gears of drives of various equipment and in educational process of higher technical educational institutions in discipline of mechanical engineering. Areas of further research – improvement of speed change devices through gear differentials in the design, manufacture, operation and repair.

Keywords: speed change control device, gear differential, closed-loop hydraulic system, sun gear, ring gear, carrier, planet, energy efficiency, dynamic model, three-dimensional modeling.

Introduction

Execution of technological operations by machines in various industries requires control of changes in the speed of their actuators. Methods and devices for stepped and stepless speed control in magnitude and direction in the form of stepped and stepless gearboxes are widely known in the technology [1], [2]. Known methods and devices for speed changes controlling have many disadvantages. The main disadvantages of stepped speed control are the complexity of the devices design; high material consumption; large dynamic loads during the transition from one speed to another, even with the extensive use of synchronizers, the complexity of automation and others. Stepless speed control is characterized by intensive wear of parts due to the use of friction brakes and clutches, which negatively affects the durability and reliability of parts and machines in general. The need to increase the efficiency and reliability of mechanical drives of machines and mechanisms requires the improvement of methods and devices for speed control. Therefore, there is a need to create new methods and devices to control speed changes, which would eliminate existing shortcomings.

Problem Statement

In this regard, a method of using speed control devices based on gear differentials have been proposed [3], [4] with the control of speed changes through the ring gear connected to the rotational movement stopper in the form of the closed-loop hydraulic system [5], [6] by the gear transmission established instead of friction brakes and couplings. For this method and device, the following research have been performed. The speed changes control possibility has been reasoned [7] and based on the analytical expression of the relationship between the speeds of the gear differential links, by computer simulation, graphical dependencies have been built that clearly confirm such changes. Energy research have been conducted and conclusions have been made about their application considering the real working conditions. Dynamic models of processes in speed change devices with the help of gear differential transmissions have been developed to select the required closed-loop hydraulic system, where the drive shaft is driven by an electric motor and the ring gear control link shaft is driven by a hydraulic motor. Optimization of such devices design using 3D modelling have been proposed. Changing the magnitude and direction of speed in technology is necessary to perform technological operations of machines in various industries. Power research, design and operation of such devices requires knowledge of energy efficiency, which is estimated by the efficiency coefficient. Determining the efficiency for a gear differential, where the speed is controlled by a closed-loop hydraulic system through the ring gear is an urgent task.

Review of Modern Information Sources on the Subject of the Paper

Scientific works [1]–[41] together with the classical technical literature [42]–[52] are recommended for use in determination of kinematic, power and energy parameters and dimensions while making the algorithm of designing the devices for speed change through a ring gear of a gear differential with the rotational movement stopper in the form of the closed-loop hydraulic system.

In [1], [2], methods and devices for speed changes control have been reviewed, which have some disadvantages and the conclusion has been made about the development of a new more rational method – through gear differentials with rotational movement stoppers in the form of closed-loop hydraulic systems instead of using friction belt, pad, disk brakes and couplings.

[3]–[8] have described the structure of speed change devices and their components at the level of patents for inventions and utility models and substantiated the possibility of speed change control, if we take, for example, the driving link – a sun gear, driven - a carrier, and the speed control link - ring gear.

In [9]–[14] kinematic studies have been performed related to the control of velocity changes through the ring gear using computer simulation of analytical expressions and obtaining graphical relationships between the speeds of the links of the gear differential.

In [15]–[23] energy studies have been held by determining the efficiency by the method of potential power to evaluate the speed change device in terms of power loss and possible self-braking.

In [24]–[29] the structure and principle of operation of the rotational motion stopper in the form of a closed-loop hydraulic system have been described. The choice of the external gear pump for the closed-loop hydraulic system has been reasoned. Based on classical analytical kinematic and force dependences with the use of computer modelling of analytical expressions graphical dependences have been obtained, a dynamic model has been built and thermal calculation has been performed. When designing a closed-loop hydraulic system and optimizing its size, it has been proposed to initially perform 3D models of its components.

In [30]–[33] dynamic models of processes in devices for speed change control with the help of gear differentials have been developed in order to select the required closed-loop hydraulic system in the case when the control link is a ring gear, where one shaft is driven by an electric motor and the other (hydraulic motor) is used to obtain the necessary law of motion on the driven link. To model the motion of a mechanical system (gear differential plus ring gear drive), it has been recommended to use, in a formalized form, the Lagrange equation of the second kind, which includes the kinetic energy of the speed change device and consolidated moments of resistance.

In [34]–[40] to initially optimize the size of the speed change devices it has been recommended to perform 3D modelling, choose the best option, and then start making technical documentation.

In [41] the stages of designing a speed change device through a ring gear of a gear differential with a rotational movement stopper have been described and reported at a scientific-practical conference.

The considered scientific sources [1]–[41] together with classical [42]–[52] ones related to this topic have been proposed to be used in the algorithm for determining kinematic, energy and force parameters and dimensions in the design of speed change devices through the ring gear of the gear differential with rotational movement stopper in the form of a closed-loop hydraulic system.

Objectives and Problems of Research

The aim of the article is to compile an algorithm for determining the kinematic, energy and force parameters and dimensions in the design of speed change devices through the ring gear of a gear differential with a rotational movement stopper in the form of a closed-loop hydraulic system based on authors' own computer-theoretical research [1]–[41] and classical scientific advices related to this topic [42]–[52] and others. Try to achieve size optimization using 3D modelling.

Main Material Presentation

The initial data for the design of such devices should be the power of the drive motor P_m, the angular velocity of its shaft ω_m, the gear ratio of the device u and technical and technological requirements. According to the initial data on the design of the speed control device, the process begins with the choice of the kinematic scheme of the gear differential, the choice of the control link, the justification of its energy efficiency through efficiency coefficient. Fig. 1 shows the basic schemes of single-stage gear differentials with cylindrical gears, given in classical technical sources, for example, [42]–[44] and others.

The choice of the scheme of gear differentials can be made from the set gear ratio, proceeding from efficiency, weight, dimensions, and other additional conditions of synthesis. In the general case, the choice of scheme can be made only by a detailed comparison of different options. For example, we choose scheme from Fig. 1 *a*, where the single-stage single-row gear differential contains a sun gear 1, planets 2, ring gear 3 and carrier 4 mounted in the housing 5 and the rotational movement stopper 6 is connected to the shaft of the ring gear 3 by gearing 7, as shown in Fig. 2, taken, for example, from [9], [10]

Let us consider the case when the driving link is the sun gear, and the driven - the carrier. The change in the speed of the driven link is carried out through the ring gear. If we take the angular velocity of the driving link for ω_1 = *const*, then by changing the speed of the ring gear (ω_3 = *var*) with the help of a rotational motion stopper, you can smoothly change the speed of the driven link - the carrier (ω_4). We have the fact that the ring gear through the gearing 7 drives a gear hydraulic pump, which is part of a closed-loop hydraulic system and pumps fluid when the control valve is open. If the control valve is closed, then the gear hydraulic pump is stopped and, at the same time, the ring gear is stopped ($\omega_3 = 0$). Thus, depending on the capacity of the control valve, the speed of the ring gear (ω_3) varies from ω_{3max} to 0 and, at the same time, the speed of the carrier (ω_4) changes.

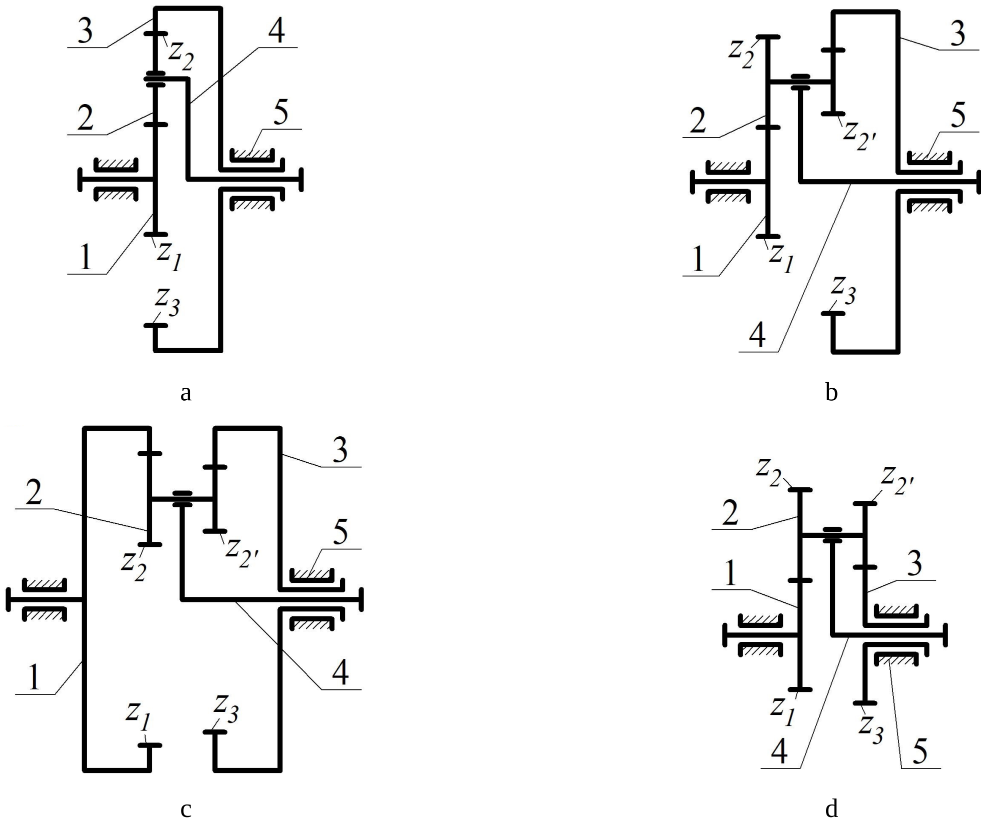


Fig. 1. Basic schemes of gear differentials: *a* – single-row; *b*, *c*, *d* – two-row; and by type of gearing: *a*, *b* – external and internal; *c* – internal; *d* – external

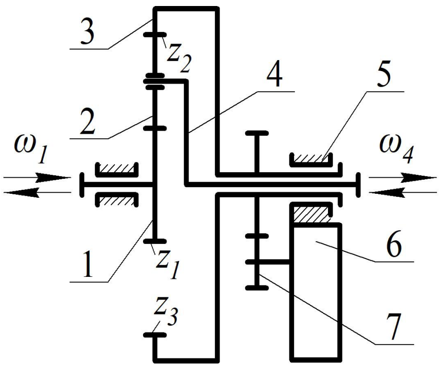


Fig. 2. Scheme of a gear differential with speed control through the ring gear

The dependence of the carrier velocity (ω_4) on the velocities ω_1 and ω_3 is described by Eq. (1):

$$\omega_4 = \frac{\omega_1 + \omega_3 u_{13}^{(4)}}{1 + u_{13}^{(4)}}\,, \tag{1}$$

where $u_{13}^{(4)}$ is the fixed carrier gear ratio of the gear differential (signs taken into account).

Fig. 3 shows the graphical dependences of the carrier speed (ω_4) on the speeds ω_1 and ω_3 and the gear ratio $u_{13}^{(4)}$, obtained by computer programming analytical expression from Eq. (1).

Evaluation of energy efficiency is performed according to the efficiency coefficient [14] using Eq. (2) or graphical dependences shown in Fig. 4, obtained by computer modelling of Eq. (2):

$$\eta_{14} = \frac{(1 + u_{13}^{(4)}\eta_{13})(\omega_1 + \omega_3 u_{13}^{(4)})}{(1 + u_{13}^{(4)})(\omega_1 + \omega_3 u_{13}^{(4)}\eta_{13})}\,, \tag{2}$$

η_{13} – efficiency coefficient for the transmission with fixed axes.

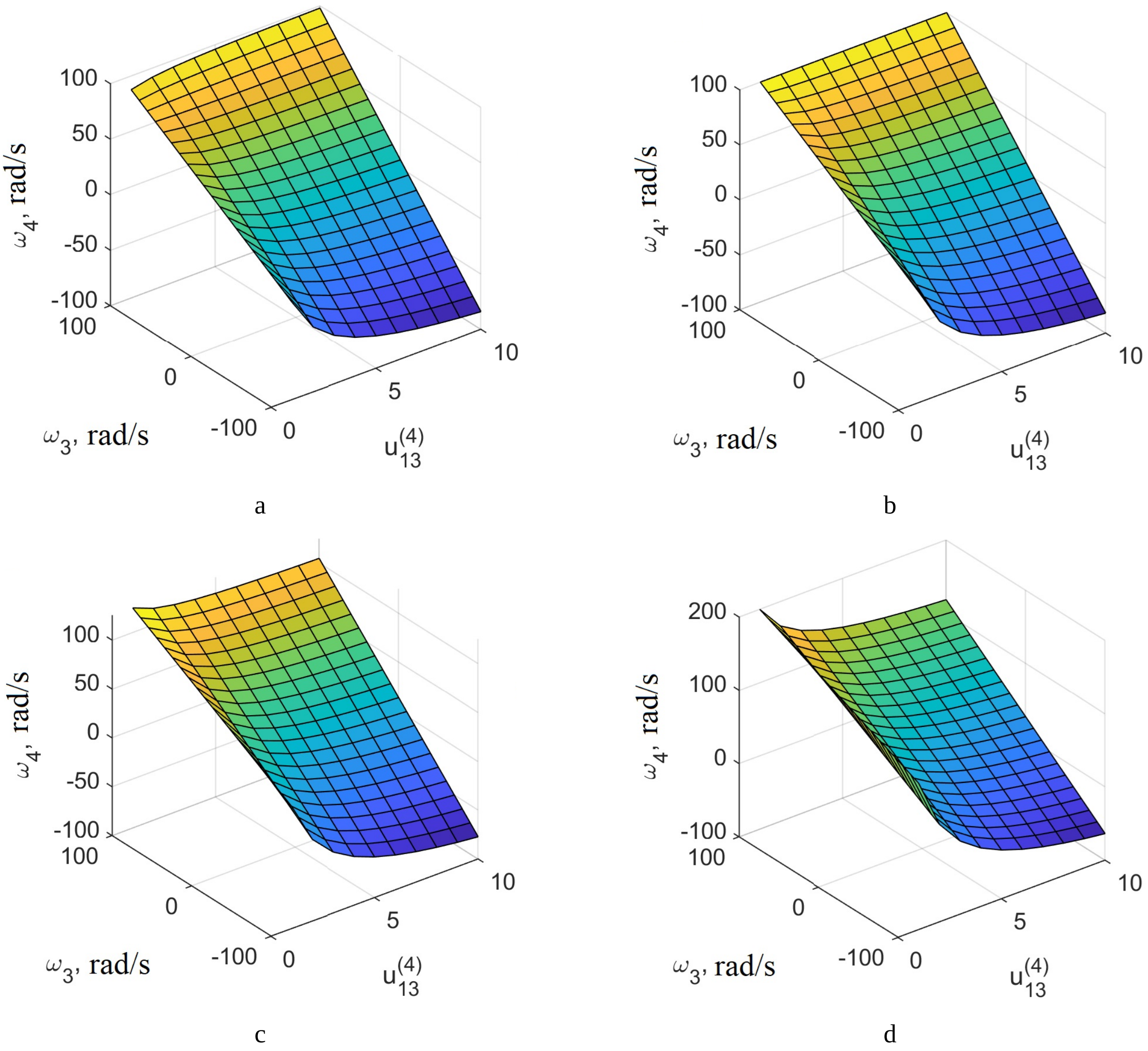


Fig. 3. Graphical dependences $\omega_4 = f\left(\omega_1, \omega_3, u_{13}^{(4)}\right)$ in the single-row gear differential with the ring gear control

Next, we proceed to determining of the number of teeth of the sun gear z_1, planets z_2 and ring gear z_3 and the number of planets k using the classic advises, for example, [42]–[44] and other authors, for which there are three condition equations:

– a given gear ratio:

$$\frac{z_3}{z_1} = u_{13}^{(4)}, \tag{3}$$

– coaxiality of the sun gear and ring gear:

$$z_3 - z_1 = 2z_2, \tag{4}$$

– assemblies;

$$z_1 + z_3 = kA, \tag{5}$$

where A is an arbitrary integer;

– one inequality for the number of planets - restrictions on the condition of the neighbourhood:

$$\sin\frac{\pi}{k} > \frac{z_2 + 2}{z_1 + z_2}. \tag{6}$$

Eqs. (3), (4) and (5) and inequality (6) are solved by selection z_1, z_2, z_3 and k, as indicated in the literature on the theory of mechanisms and machines [42]–[44], or other authors.

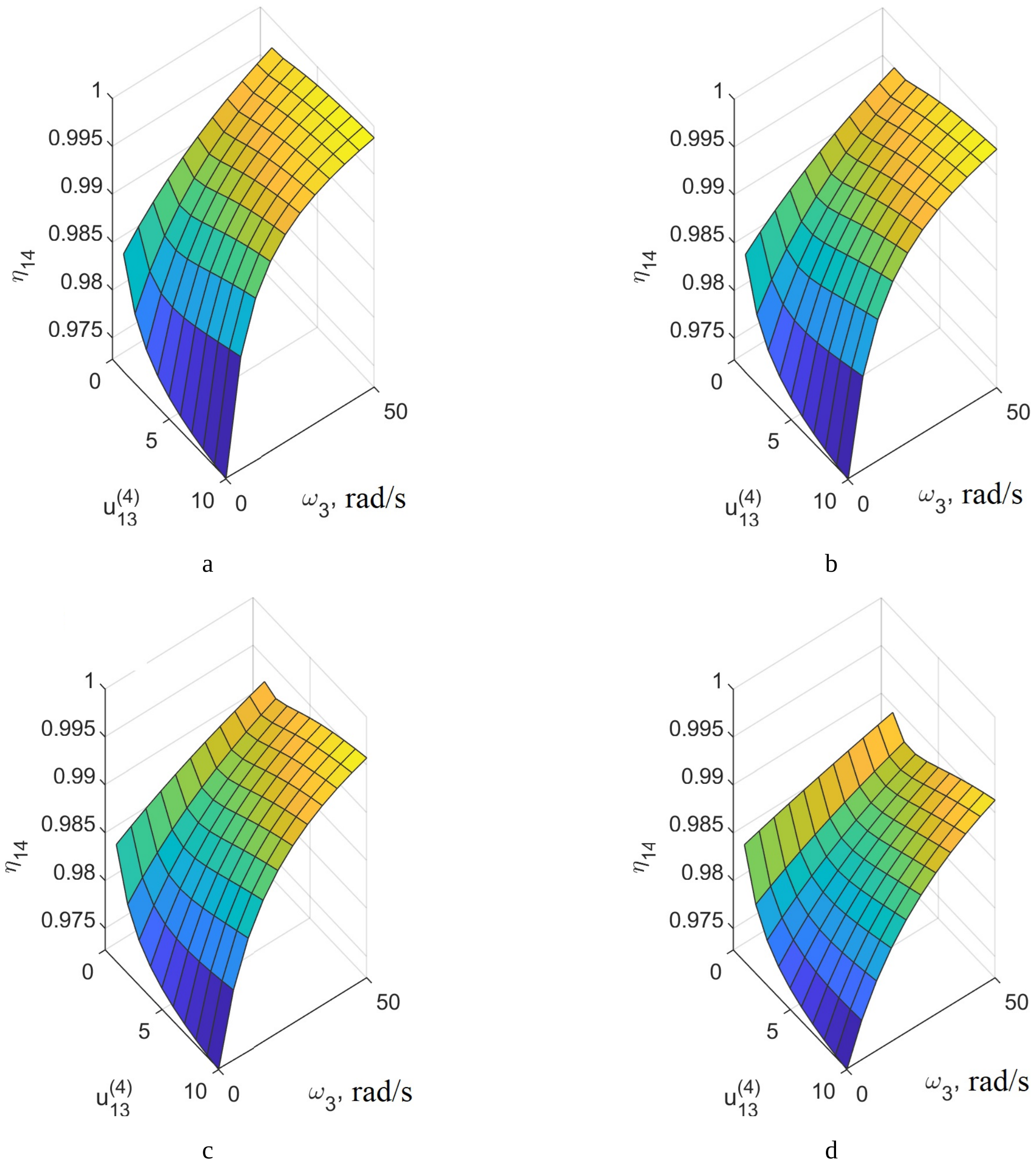


Fig. 4. Graphical efficiency dependences $\eta_{14} = f\left(\omega_1, \omega_3, u_{13}^{(4)}\right)$ in the single-row gear differential, when the driving link is the sun gear, and the driven - the carrier

Kinematic and energy parameters are specified using the results of the kinematic [9]–[14] and energy [15]–[23] studies.

Then we determine the torques on the shafts of the gear differential [45], [46] or other authors. When the power and angular velocity on the drive shaft have been specified, you can determine its torque from the expression

$$T_{вч} = \frac{10^3 P_{вч}}{\omega_{вч}}. \tag{7}$$

Knowing the torque of one shaft in the gear differential, you can determine the required others. For our case it will look like this:

$$T_1 = T_{вч};\ T_2 = -T_1 u_{12} \eta_{12};\ T_3 = -T_1 u_{13}^{(4)} \eta_{13};\ T_4 = -T_1(1 - u_{13}^{(4)} \eta_{13}). \tag{8}$$

Next, we perform preliminary calculations for the strength of the parts of the gear differential. Here determine the geometric dimensions of the gears, shafts and carrier, which will be used in the dynamic model of the speed changes control device. Calculations are performed by classical methods for simple transmissions - with fixed axes. The wheelbase from the condition of contact strength for straight gears is found using the expression:

$$a_{w\min} = 450(u \pm 1)\sqrt[3]{\frac{K_H T_2}{\psi_a u_{23}^2 [\sigma_H]^2}}, \tag{9}$$

where u is gear ratio; T_2 is torque; K_H and ψ_a are coefficients; $[\sigma_H]$ is permissible contact stresses for tooth materials. The values included in Eq. (9) are selected from the reference literature according to the recommendations, for example, [45].

The gear modulus is defined as

$$m_n = \frac{2a_{wmsn}}{z_1 + z_2} = \frac{2a_{wmsn}}{z_3 - z_2}. \tag{10}$$

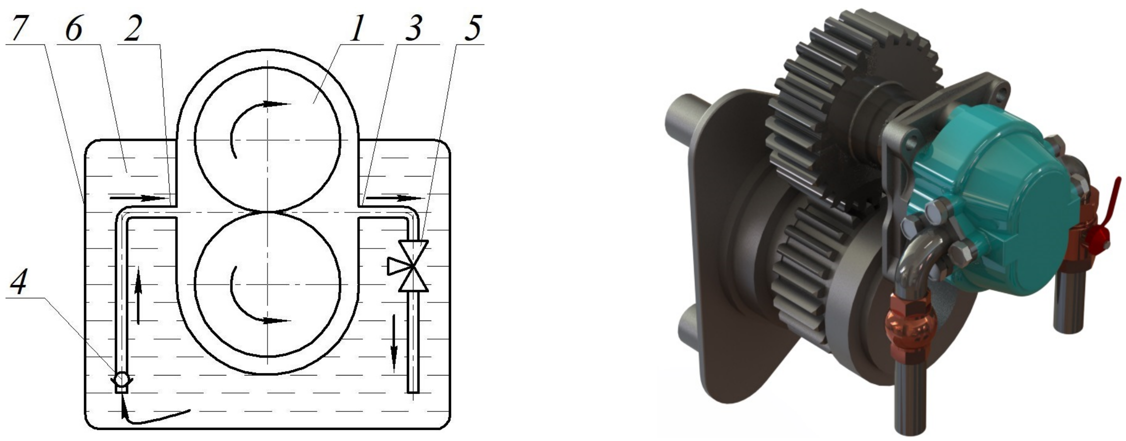
The obtained value of the gear module is rounded to the standard value and so all geometric dimensions of the gears can be calculated.

The minimum diameters of shafts are found from a condition of torsion durability considering that shafts by cross-section can be either solid or hollow

$$d_{\min} = \sqrt[3]{\frac{10^3 T_i}{0,2(1-\xi^4)[\tau_k]}}, \tag{11}$$

where $\xi = d_0 / d_{\min}$ is the coefficient that characterizes the annular cross section of the shaft; $[\tau_k]$ – permissible torsional stresses. Next, choose bearings for shafts and planets supports, other components of the gear differential.

Then we proceed to the calculations of the components of the rotational movement stopper in the form of a closed-loop hydraulic system and its drive from the control link - the ring gear, using the advice of previous research [24]–[29] and the classic advice on this issue [47], [48]. Fig. 5. shows the scheme and model of the rotational movement stopper in the form of the closed-loop hydraulic system. Using the kinematic, power and energy dependences in the closed-loop hydraulic system and the power of the mechanical drive choose the hydraulic pump from the catalog [48], which most rationally corresponds to the application of this speed change device. Focusing on the conditional diameters of the suction and discharge holes of the pump, we accept pipes with the same diameters of holes, choose a reverse valve, control valve and other components of a closed-loop hydraulic system and its drive. The drive of the hydraulic pump is performed, as a rule, in the form of a cylindrical gear, the geometric dimensions of the gears are determined as recommended by [45], [46], or other authors. The structure of the rotational movement stopper is widely described in [6].



a b

Fig. 5. Hydraulic scheme (a) and model (b) of the rotational movement stopper with the drive

In [30]–[33] dynamic models of processes in devices for speed change by a ring gear of a gear differential where the driving shaft – a sun gear is driven by the electric motor, and the second (hydraulic motor) is driven from a ring gear and used for receiving the necessary law of movement on the driven link. To model the motion of a mechanical system (gear differential plus ring gear drive), it is recommended to use, in a formalized form, the Lagrange equation of the second kind, which includes the kinetic energy of the device of speed change and consolidated and resistance moments [49].

Next, to analyse the dynamic model final expressions of the torques of inertia of the driving and driven links have been fond and using the relationships between the moments in the gear differential, you can determine the torques of other links and analyse its operation:

$$M_{i1} = \frac{M_{s4}J_{14} - M_{s1}J_{44}}{J_{14} - \sqrt{J_{44}J_{11}}} \text{ and } M_{i4} = \frac{M_{s4}J_{11} - M_{s1}J_{14}}{\sqrt{J_{44}J_{11}} - J_{14}}, \tag{12}$$

where J_{11}, J_{14}, J_{44} - are the dynamic moments of inertia:

$$\begin{aligned} J_{11} &= J_1 + kJ_2(u_{21}^{(4)})^2 + J_3(u_{31}^{(4)})^2 + J_{76}u_{7'7}^2(u_{31}^{(4)})^2; \\ J_{41} &= 2(kJ_2u_{21}^{(4)}u_{24}^{(1)} + J_3u_{31}^{(4)}u_{34}^{(1)} + J_4u_{7'7}^2u_{31}^{(4)}u_{34}^{(1)}); \\ J_{44} &= kJ_2u_{24}^{(1)} + km_2r_4^2 + J_3(u_{34}^{(1)})^2 + J_4 + J_{76}u_{7'7}^2(u_{34}^{(1)})^2. \end{aligned} \tag{13}$$

Here in Eq. (13) we have: J_i and m_2 are the dynamic moments of inertia of links with respect to the mass centres and mass of the planet; k – number of planets; $v_C = \omega_4 r_4$ – circular speed of the axis of rotation of the planet, r_4 – the radius of rotation of the carrier. This radius is equal to the sum of the initial radii of the sun gear and the planet $r_4 = 0,5(d_{W_1} + d_{W_2})$. Parameters with the corresponding designations relating to components of the speed change device correspond to the scheme shown in Fig. 2.

The dynamic moment of inertia of cylindrical gears is defined as $J_i = 0,5m_i r_i^2$, where m_i and r_i are respectively, the mass and radius of the gear.

The gear ratios included in Eq. (13) considering the signs are written as follows:

$$u_{21}^{(4)} = \frac{z_1}{z_2};\ u_{31}^{(4)} = \frac{z_1}{z_3};\ u_{24}^{(1)} = 1 + \frac{z_1}{z_2};\ u_{34}^{(1)} = 1 + \frac{z_1}{z_3};\ u_{7'7} = \frac{z_7'}{z_7}. \tag{14}$$

Consolidated torques

$$M_{s1} = M_1 + M_6 u_{71}^{(4)};\ M_{s4} = -M_4 + M_4 u_{74}^{(1)}, \tag{15}$$

where M_{s1} is the consolidated moment [33] determined from the equality of power moments of forces for fixed carrier. The torque $M_1 = M_1(\omega_1)$ is a function of the angular velocity of the sun gear and is determined as $M_1 = P / \omega_1$, and the torque of the closed-loop hydraulic system is $M_6 = pqu_{7'7} / \omega_3$.

The moment M_{s4}, which is the moment of resistance is applied to the driven link - the carrier, for this example, and is taken from the graphs of typical cases of load change in the form of resistance torque M of the actuator shown in Fig. 6: *a* – the load changes periodically over a long period of time; *b* – the magnitude of the shock load after a sharp increase remains unchanged for a long time; *c* – the magnitude of the shock load after a sharp increase is maintained for a short time; *d* – the actuator stops instantly due to significant overload.

In Eq. (8) to determine the torques we have: P – power; p – pressure in a closed-loop hydraulic system; q – the flow of the hydraulic pump per revolution of the shaft; gear ratio $u_{71}^{(4)} = u_{13}^{(4)}u_{7'7}$.

Next, we determine the torque on all parts of the gear differential and evaluate its operation in the speed change device.

And, finally, 3D models of elements and assemblies of the speed change device are performed according to the advice [34]–[40], [50]–[52] and shown in Fig. 7.

The 3D model is proposed to be implemented in order to optimize the size and placement of the components of the speed change device, because at the initial stage of design you can get a visual idea of differential transmissions with closed-loop hydraulic systems and use a computer to view them from anywhere; increase design accuracy; easy to edit three-dimensional models, i.e. make the necessary changes; to achieve great savings of time and material costs; get a large number of possible design solutions that need to be analysed in detail and in depth and choose a rational one; on the basis of the created basic models of transfers it is possible to receive models of transfers with the different sizes.

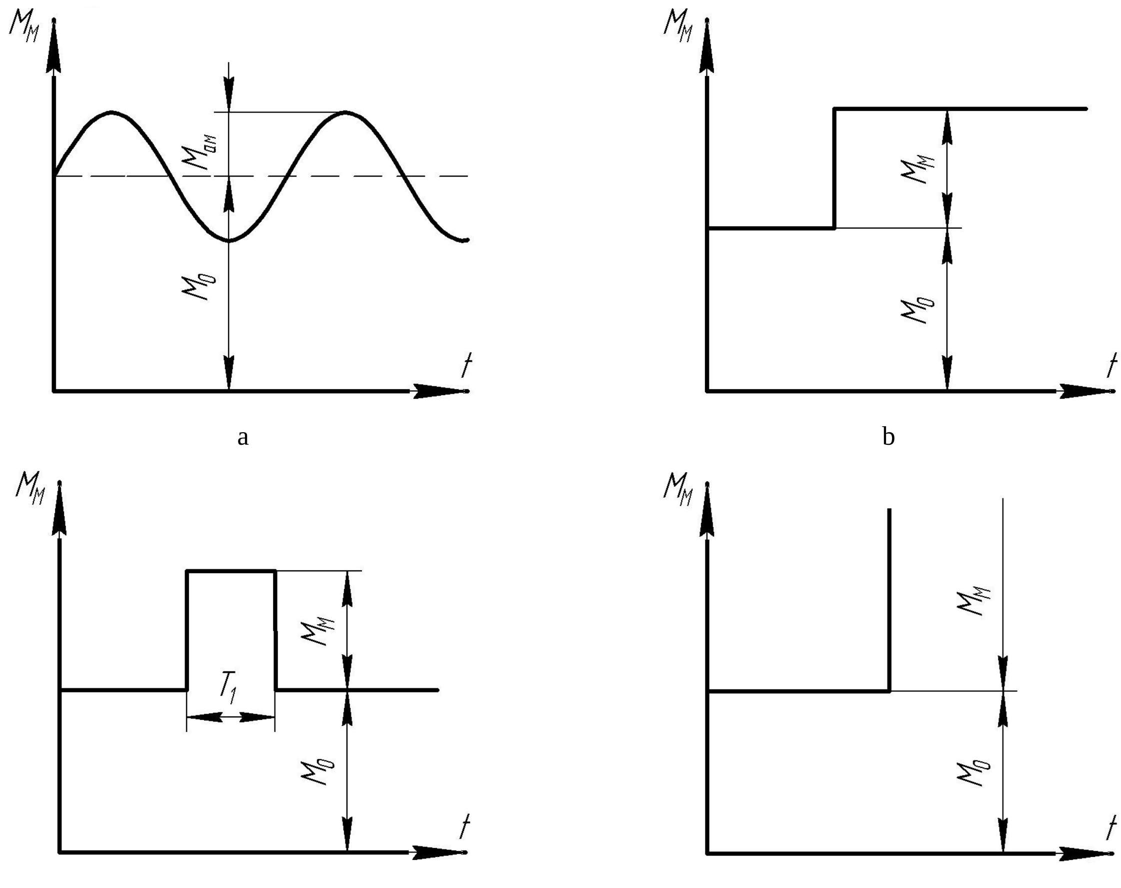
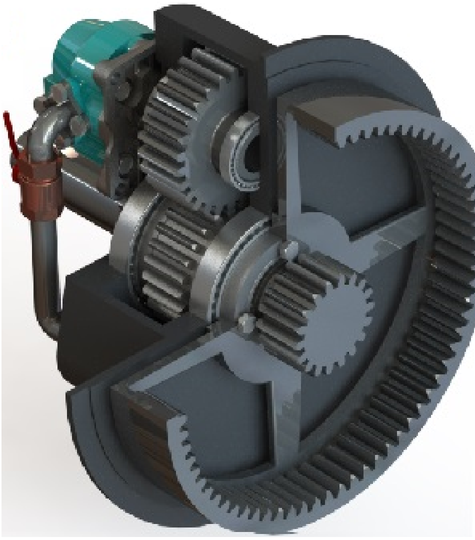


Fig. 6. Graphs of possible changes in torque on the shaft of the working mechanism

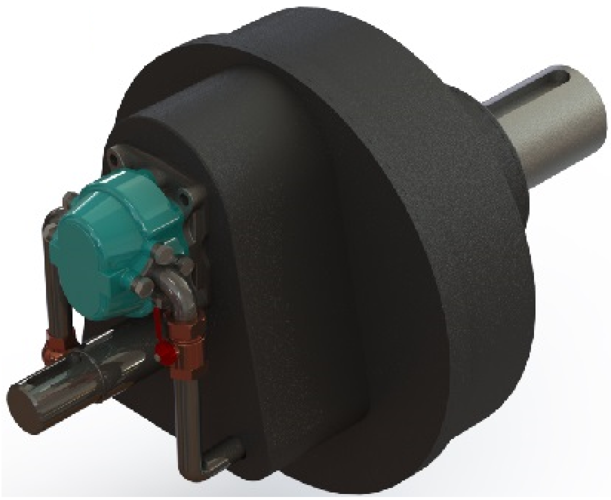
a

Fig. 7. *(Beginning)* Models of drive elements: *a* – custom parts of differential transmission

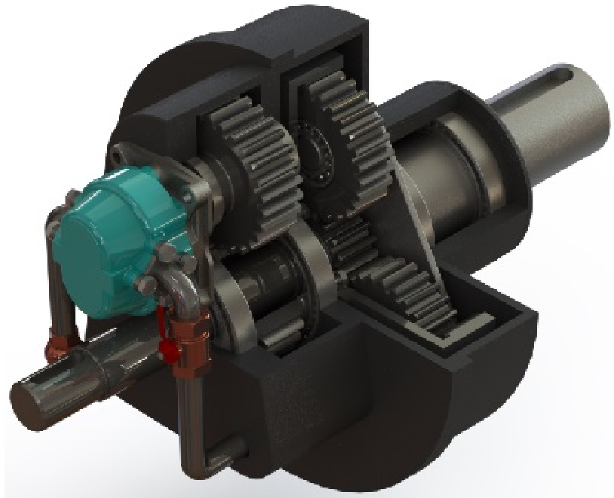


b

c



d



e

Fig. 7. *(Continuation)* Models of drive elements: *b* – assembly of the ring gear with a closed-loop hydraulic system and its drive; *c* – assembly of the carrier, planets and body parts; *d* – complete assembly of the device – closed version and *e* – complete assembly of the device – open version

At the final choice of the optimum variant of model, after various specifications, we start development of technical documentation of the speed change device by means of a gear differential and the rotational movement stopper in the form of the closed-loop hydraulic system.

The calculation steps are performed using computer programming.

Conclusions

1. In the given work the algorithm of practical calculations and components parameters choice of the speed change device on an example of a single-stage single-row gear differential has been offered, when the driving link is a sun gear, driven is a carrier, using classical advice and the results of authors' own research.

2. The given example can be used, as algorithm, for calculations at designing of speed change devices with gear differentials and rotational movement stoppers in the form of the closed-loop hydraulic systems of any schemes and their work in the forward and return directions, as in this example from a sun gear. to the carrier and vice versa - from the carrier to the sun gear.

3. The proposed computer-aided design with modelling can be a significant addition to the existing and previously developed by the authors of theoretical methods for determining the rational dimensions of the drive elements of mechanisms and machines of various branches of engineering and an important basis for further research.

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