THE EFFICIENCY OF THE DIFFERENTIAL GEAR TO DEVICES FOR CONTROLLING THE SPEED CHANGE THROUGH A SUN GEAR

O.R. Strilets. The efficiency of the differential gear to devices for controlling the speed change through a sun gear. When performing technological processes, it becomes necessary to control changes in speed in magnitude and direction. Today, special attention is paid to the method of controlling speed changes using single-stage and multistage differential gears with a closed hydraulic system. Relevant issue is the efficiency of such devices. The aim of the work is to obtain analytical and graphical dependencies of the efficiency between the driving and the driven links in single-stage single-row and two-row differential gears with closed hydraulic system, when the driving link is the carrier, and the driven one is the epicycle, and vice versa. The work of a single-row and two-row differential gear was analyzed in cases where the driving link is the carrier, and the driven one is the epicycle, and vice versa. The control link of the speed change is a solar gear that can rotate when fluid is pumped in the hydraulic system, or can be stopped by a closed hydraulic system. An analytical study of the efficiency for such a transmission was performed, and its graphic dependencies on transmission parameters were obtained. Based on the obtained analytical expressions and graphical dependencies, a conclusion was made about the change in the value of the efficiency from the ratio, the speed of the control link, and its estimation from the point of view of self-braking was performed.

Keywords: efficiency, differential gear, sun gear, carrier, epicycle, closed hydraulic system

Introduction. When carrying out technological processes by lifting, transport, construction, road, meliorative machines, cars and tractors, and other equipment, it becomes necessary to control the speed and speed changes in magnitude and direction. To date, methods and devices for stepped and stepless speed control in terms of magnitude and direction are known in the form of stepped gear boxes, belt, chain variators, and the like. Known methods and devices for controlling speed changes have many disadvantages. The main disadvantage of stepped control is the complexity of the design of devices, their large material capacity, the large dynamic loads that arise when switching from one speed to another, even using synchronizers. The main disadvantage of stepless speed control is the high wear of parts due to the use of frictional bonds, as a rule, frictional brakes and locking friction clutches, which reduces the durability and reliability of the details of drives and machines in general. Therefore, there are problems of creating new methods and devices for controlling speed changes, which will eliminate these shortcomings.

Today special attention is paid to the method of controlling speed changes using single-stage and multistage differential gears with a closed hydraulic system [1 – 16]. Control in such devices is realized by means of a sun gear, which rotates the hydraulic pump of a closed hydraulic system through a gear train and pumps the liquid in it with a certain angular velocity of the control link.
Relevant issue is the efficiency of such devices [17 – 21]. Some of these devices are developed at the level of Ukrainian patents and require further theoretical studies of their kinematic, power and geometric parameters [22 – 25].

General concepts of efficiency are widely described in the classical technical literature on the theory of mechanisms and machines, for example in [26], but the data given do not take into account specific cases of operation of specific mechanisms.

The aim of the work is to obtain analytical and graphical dependences of the efficiency between the driving and the driven links in single-stage single-row and two-row differential gears with closed hydraulic system, when the driving link is the carrier, and the driven one is the epicycle, and vice versa.

Materials and Methods. It is well known that the perfection of machines and mechanisms is assessed with help of efficiency.

The efficiency is in the range of $0 \leq \eta \leq 1$ and is a value that is determined by the ratio of the useful power to the spent

$$\eta = \frac{P_{up}}{P_t},$$

where $P_{up}$ – power of useful forces; $P_t$ – total power, supplied to the mechanism,

$$P_t = P_{up} + P_{hp},$$

where $P_{hp}$ – power of harmful forces.

The general definition of efficiency as mentioned above can be specified for individual cases and, importantly, it is possible to obtain formulas for determining the efficiency through other parameters of the mechanisms.

The determination of the efficiency of differential gears is not always necessary. If such a transmission is used as a reducer that transfers power to the machine's actuator for a long time, then to determine its suitability it is necessary to calculate the efficiency. In the case where such transmissions are used as movement control devices for some links, the calculation of the efficiency can be omitted if there is a certainty that the transmission is not self-braking.

In practice, there are mainly three known methods of determining the efficiency [26], which can be taken as the basis for solving the specific task.

The efficiency of the proposed device for controlling speed changes by means of a differential transmission with a closed hydraulic system via the solar gear, in the case where the driving link is the carrier and the driven one is the epicycle, appears as follows

$$\eta = \eta_{43} \eta_6 \eta_7,$$

where $\eta_{43}$ – efficiency of differential gear; $\eta_6$ – efficiency of a drive of closed hydraulic system (gear); $\eta_7$ – closed hydraulic system efficiency (power loss at the pump when transferring liquid in a closed hydraulic system).

A block diagram of possible power losses in the device for controlling speed changes by means of a differential transmission with a closed hydraulic system through an epicycle, where the driving link is the carrier, and the driven one is the epicycle, is shown in fig. 1.

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Fig. 1. Block diagram of power losses in the device for controlling speed changes in the case where the driving link is the carrier and the driven is the epicycle, and the control link is the solar gear.
Based on the analysis of power losses in the device, it is concluded that some of the power supplied to the carrier via the satellite is transmitted to the epicycle, and its other part is transmitted via the satellite to the solar gear and the closed hydraulic system.

To determine the efficiency, let’s consider the scheme of the forces acting in the clutches of the sun gear and satellite ($F_{12}$), of the satellite and the epicycle ($F_{23}$), of the satellite and the carrier ($F_{24}$) and write the equilibrium condition of the satellite (fig. 2) in the following form

$$F_{12} + F_{23} + F_{24} = 0.$$  \hspace{1cm} (3)

In addition, the sum of the moments of forces acting on the satellite relative to the axis of its rotation is zero

$$F_{12}r_2 + F_{23}r_2 = 0,$$  \hspace{1cm} (4)

where $r_2$ – radius of the initial circle of the satellite with $z_2$ number of teeth.

From the expressions (3) and (4) follows the dependence of the forces acting on the links of the differential gear, which can be expressed as follows:

$$F_{12} = -F_{23};$$  \hspace{1cm} (5)

$$F_{24} = -(F_{12} + F_{23}).$$  \hspace{1cm} (6)

The obtained formulas show that one of the given forces makes it possible to determine the two others. If there is a given torque $T_1$, then

$$F_{12} = T_1 / r_1,$$  \hspace{1cm} (7)

where $r_1$ – radius of the initial circle of the sun gear $z_1$.

According to the expression (7) for the torque, which is applied to the wheel in the gear with the stopped carrier, we have

$$T_3 = F_{23}r_3 = -\frac{r_3}{r_1}T_1 = -u_{13}^{(4)}T_1,$$  \hspace{1cm} (8)

that is, the rotational moments $T_1$ and $T_3$ without friction are referred to as in a gear with fixed axes. Considering friction on the teeth surfaces, the ratio between the rotational moments can be represented as follows

$$T_3 = -T_1u_{13}^{(4)}\eta_{13}^{k},$$  \hspace{1cm} (9)

where $\eta_{13}$ – transmission efficiency with fixed axes is determined as for the sequential clutch;

- $k = +1$ – when power is transferred from the carrier to the gear wheel $z_3$ and $k = -1$ – when power is transferred from the gear $z_3$ to the carrier;

- $u_{13}^{(4)}$ – differential gear ratio with the driver stopped:

$$u_{13}^{(4)} = -z_3 / z_1.$$
In this case, the efficiency of the differential gear needs to be determined taking into account the fact that significant power is transmitted from the carrier to the epicycle or vice versa. That is, the gear differential transmission is used as a power gear, and the sun gear serves as a link for changing the speed of the driven links of the epicycle in the first case, and the carrier in the second.

The relationship between the rotational moments that act on the links of the differential gear can be established by considering the condition of the balance of the transmission as a whole, namely:

\[ T_1 + T_3 + T_4 = 0, \tag{10} \]

where \( T_4 \) – torque acting on the carrier.

According to [26]

\[ T_4 = T_1 (1 - u_{13}^{(4)} \eta_{13}). \tag{11} \]

When the driving link is the carrier, and the driven one is the epicycle, that is, the torque of the resistance forces is applied to the epicycle, then the expression for the efficiency will have the following form

\[ \eta_{13} = -\frac{T_1 \omega_4}{T_4 \omega_4 + T_1 \omega_1}. \tag{12} \]

Let’s substitute the values of \( T_1, T_3 \) and \( T_4 \) to (12) and express \( \omega_4 \) as \( \omega_3 \), using formula (2) from [10]. After simple transformations we obtain:

\[ \eta_{13} = \frac{(1 + u_{13}^{(4)}) \omega_4 - \omega_1}{(1 + u_{13}^{(4)} \eta_{13}) \omega_4 - \omega_1}. \tag{13} \]

In order to better illustrate the nature of the dependence of the efficiency of a differential transmission on a device in the form of a closed hydraulic system on the transmission parameters, when the driving gear is the sun gear and the driven is the carrier the following is implemented.

Graphical dependences of the efficiency of the differential gear transmission \( \eta_{13} = f(\omega_1, \omega_4, u_{13}^{(4)}) \) for gear ratios \( u_{13}^{(4)} = \{1...50\} \), with angular velocity of driving link \( \omega_4 = 100 \text{ rad/s} \) were constructed. The obtained graphical dependences \( \eta_{13} = f(\omega_1, \omega_4, u_{13}^{(4)}) \) for \( u_{13}^{(4)} = \{1...20\} \) at \( \eta_{13}^{(4)} = 0.97 \), \( \omega_4 = 100 \text{ rad/s} \) and \( \omega_1 = 0...50 \text{ rad/s} \) are shown in fig. 3.

![Fig. 3. Efficiency dependences \( \eta_{13} = f(\omega_1, \omega_4, u_{13}^{(4)}) \) in differential single-row transmission in the case where the driving link is the carrier, and the driven one is the epicycle](image)

The efficiency of the device for controlling speed changes by means of a differential transmission with a closed hydraulic system through the solar gear, where the driving link is the epicycle, and the driven one is the carrier, appears as follows...
\[ \eta = \eta_{34} \eta_6 \eta_7, \]  
(14)

where \( \eta_{34} \) – differential transmission efficiency;

\( \eta_6 \) – efficiency of drive of closed hydraulic system (gear);

\( \eta_7 \) – efficiency of a closed hydraulic system (loss of power for pump operation when pumping liquid in a closed hydraulic system).

A block diagram of possible power losses in a device for controlling speed changes by means of differential transmission with a closed hydraulic system through a solar gear, where the driving link is the epicycle, and the driven one is a carrier, is shown in fig. 4.

If the torque \( T_4 \) is the moment of resistance of the working machine (that is, the carrier is the driven link), \( T_3 \) is the driving epicycle torque and \( T_1 \) is the steering torque (sun gear), then the efficiency with the driven carrier can be expressed by the ratio of the usable power to total power

\[ \eta_{41} = - \frac{T_4 \omega_4}{T_1 \omega_1 + T_4 \omega_3}. \]  
(15)

If we substitute the values to the formula (15) \( T_1, T_3 = -T_4 u_{13}^{(4)} \eta_{13}^{-1} \) and \( T_4 = -T_1 (1 - u_{13}^{(4)} \eta_{13}^{-1}) \) to the formulas (9) and (11) and express \( \omega_4 \) through \( \omega_3 \), using formula (3) from [10], after simple transformations we obtain an expression for efficiency with entered carrier as:

\[ \eta_{34} = \frac{(\eta_{13} + u_{13}^{(4)})(\omega_3 + \omega_3 u_{13}^{(4)})}{(1 + u_{13}^{(4)})(\omega_3 \eta_{13} + \omega_3 u_{13}^{(4)})}. \]  
(16)

For this case, similar to previous graphs, we obtained dependences of efficiency \( \eta_{34} \) from \( u_{13}^{(4)} = 1 \ldots 20 \) for \( \eta_{13}^{(4)} = 0.97 \); \( \omega_3 = 100 \text{ rad/s} \); \( \omega_1 = 0 \ldots 50 \text{ rad/s} \). The results are shown in fig. 5.

Fig. 4. Block diagram of power losses in the device for controlling speed changes in the case where the driving link is the epicycle, the driven one is the carrier, and the control link is the solar gear.

Fig. 5. Dependence of efficiency of \( \eta_{34} = f(\omega_3, \omega_1, u_{13}^{(4)}) \) in differential single-row transmission in the case where the driving link is the epicycle and the driven one is the carrier.
For a two-row differential gear (fig. 6), the ratio value $u_{13}^{(4)}$ will be calculated as

$$u_{13}^{(4)} = \frac{z_2 z_3}{z_1 z_2'}.$$  \hspace{1cm} (17)

In addition, the sum of the moments of forces acting on the satellite relative to the axis of its rotation is equal to zero:

$$F_{12} r_2 + F_{23} r_2' = 0,$$  \hspace{1cm} (18)

where $r_2$ and $r_2'$ – radii of the initial circumferences of the gears of the satellite with the number of teeth $z_2$ and $z_2'$ respectively.

**Conclusions.** The following conclusions can be drawn from the study:

The obtained analytical and graphical dependences of the efficiency between the driving and driven links (carrier and epicycles, and vice versa) in single-stage single-row and two-row differential gears with closed hydro systems (fig. 3 and 5) show that the efficiency value varies and give the possibility to estimate it from self-braking point of view.

It is shown (fig. 3) that in the gear differential transmission in the case where the driving link is the carrier and the driven one is the epicycle, the efficiency is higher than for the simple transmission and significantly increases with the increase in the gear ratio in the range of 0.1...5, and further is almost constant and depends little on the angular velocity of the control link.

It is shown (fig. 5) that in the gear differential transmission in the case where the driving link is an epicycle and the driven one is a carrier, the efficiency is higher than in the simple transmission and decreases somewhat with the increase in the gear ratio within the limits of 0.1...0.5, and in the range of 0.5...5 increases and further is almost constant value and depends little on the angular velocity of the control link. Analysis of expression (17) shows that the efficiency does not decrease to zero for a given quadrant and self-braking is not possible.
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