Control of braking energy recovery of lifting machines

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Abstract. The increase in the energy consumption of machines when operating in unsteady modes is caused by many reasons, the main of which are deviation from the calculated operating modes and braking of the machine with irretrievable energy losses. When driving cars from electric motors, the recovery of braking energy can be carried out by switching them to the electric generator mode and returning the braking energy to the general power supply network. However, the absence of parallel consumers in the general power supply network requires the presence of an electric energy accumulator to recover braking energy. Effective recovery of braking energy can be carried out in a flywheel energy storage device with a high service life and high specific indicators of stored energy per unit weight of the storage device using differential gear mechanisms having two degrees of freedom.

1 Introduction

The economy of any machine is one of its most important characteristics. The main dynamic and economic properties of lifting and mining machines are laid down in the design process when choosing parameters such as the type and power of the engine, as well as the transfer function of the transfer mechanism. Subsequent calculations based on the criteria of strength and durability no longer have a significant impact on their dynamic qualities and energy consumption in operation, unless there is a significant increase in moving masses [1-4].

The increase in the energy consumption of machines when operating in unsteady modes is caused by many reasons, the main of which are the deviation from the calculated operating modes of the engine and the braking of the machine with irretrievable energy losses [5-6]. An increase in energy consumption and a decrease in the efficiency of lifting and mining machines is usually associated with the frequency of braking modes, therefore, an effective method of increasing their efficiency is the recovery of braking energy [7-8].

2 Materials and methods

When driving cars from electric motors, the recovery of braking energy can be carried out by switching them to the electric generator mode and returning the braking energy to the power supply network. This property of self-regulation, i.e. the possibility of automatic switching

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from the engine operation mode to the generator operation mode, is possessed by numerous types of DC and AC electric motors [9]. DC electric motors have a significant simplicity in controlling the recovery of braking energy, in which this operation of switching from the engine operation mode to the generator mode is carried out simply by changing the current in the excitation winding. However, in the absence of parallel consumers in the general power supply network, the presence of an electric energy accumulator is required for the recovery of braking energy. When using DC electric motors powered by an electric battery in the drive of machines, the recovery of braking energy causes an increase in the frequency of the return of braking energy to the electric battery and a decrease in its service life [10]. Therefore, to accumulate the braking energy of electric vehicles, even in the presence of an electric battery, some foreign companies use a flywheel energy storage device with a high service life and high specific indicators of stored energy per unit weight of the storage device. In order not to increase the frequency of the return of braking energy to the electric battery and reduce its service life, the recovery of braking energy using a flywheel energy storage device can be carried out using differential mechanisms.

For example, the drive of transport vehicles from the internal combustion engine in urban traffic using stepless speed control and regenerative braking significantly increases their efficiency. However, the use of a hybrid vehicle drive from an internal combustion engine and DC electric motors powered by an electric battery for these purposes makes the braking energy recovery control scheme quite complex and expensive. A well-known scientist working in this field, N.V. Gulia, in his writings, justified the method of using the method of energy recovery of braking transport vehicles using flywheel energy storage devices [11-12]. Numerous projects for the recovery of braking energy of special transport vehicles have been implemented in England and other developed countries [13-14].

The main advantages of flywheel drives are:

- high specific power and recovery rate;
- high specific density of stored energy;
- no influence of the number of recovery cycles on the service life and long service life;
- no need for frequent maintenance;
- no harmful effects on the environment.

However, flywheel drives also have certain disadvantages due to the fact that the accumulation of braking energy in the flywheel drive is difficult due to the complication of the mechanical drive due to the need to use mechanical variators. Usually, various types of friction variators are used for this purpose, the disadvantages of which are a narrow range of gear ratio changes and the lack of self-regulation. Self-regulation refers to the process of automatically changing the gear ratio of the kinematic drive chain connecting the flywheel to the transmission when the external load changes. Therefore, in numerous technical proposals, it is proposed to use differential mechanisms with the property of self-regulation in the flywheel drive.

Differential mechanisms have two degrees of freedom and are often used in transport vehicles for a stepwise change in the gear ratio by braking one of the links and turning the differential mechanism into a gearbox with one degree of freedom. Braking of another link of the differential mechanism changes the gear ratio of the kinematic drive chain. This property of the differential mechanism is often used in gearboxes of cars and turning mechanisms of tracked vehicles. However, the property of self-regulation of the gear ratio of the drive in the case of a stepwise change is not required.

An example of the use of a differential mechanism for the recovery of braking energy can be the project of a lifting device developed at the Bauman Moscow State Technical University.The flywheel accumulator drive is based on the well-known flat gear differential mechanism with two degrees of freedom. The kinematic scheme of the lifting device developed at the Bauman Moscow State Technical University and shown in Figure 1 compares favorably with the known ones in that it allows you to recover part of the braking energy during descent in a flywheel energy storage and use the accumulated energy after descent to lift the device. The mechanism consists of a cabin 11 with the possibility of descent on a cable 7. On the one hand, the cable 7 is fixed to the supports, and on the other hand, it is wound onto a drum 6 located inside the cabin 11, from which the cable 7 is wound during descent, rotating the drum 6. The latter shaft is connected to a multiplier based on a differential mechanism that increases the speed of rotation of the flywheel 5 compared to the speed of the drum 6.

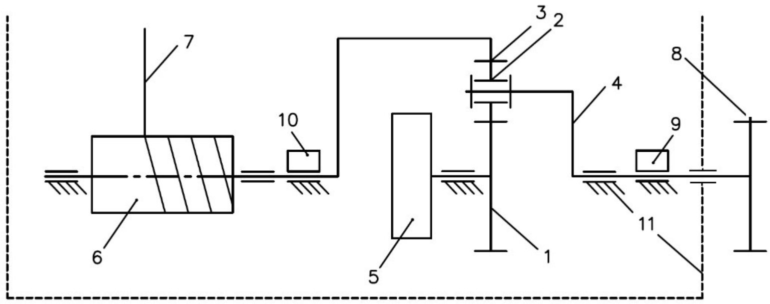

Fig. 1. Kinematic diagram of the device: 1 – solar gear; 2 – satellite; 3 – crown wheel; 4 - differential driver; 5 - flywheel; 6 – drum; 7 - cable; 8 – gear transmission. 9 – brake; 10 – drum brake; 11 – container body.

The lifting device is equipped with a flat single-row gear differential of a well-known design, which consists of a solar gear 1, a satellite 2, a driver and a central gear 3 with internal teeth. The efficiency of energy recovery control increases when the flywheel 5 is connected to the solar gear 1, for this purpose an additional multiplier can be used, which increases the speed of rotation of the flywheel and reduces its required moment of inertia.

The principle of operation of the mechanism is that during the descent of the load, a significant part of the potential energy during braking is pumped into the flywheel storage, which also performs the functions of a descent speed limiter. After lowering the cab, the flywheel has a significant part of the potential energy that it can use for various purposes, transforming it into electrical energy or directly transferring it to the operation of a gear train that allows lifting the cab back.

The control system of the lifting machine provides for the presence of two brakes in the differential mechanism: the brakes of the driver 9 and the brakes of the central gear wheel 3. For the descent of the container, the previously tightened brake 10 of the wheel 3 is released, the brake 9 is tightened and the driver 4 brakes. The internal gear ratio of the differential mechanism when the driver 4 is stopped is equal to:

$$U_{13}{}^{(4)} = -\frac{Z_3}{Z_1} \tag{1}$$

After lowering the cab, the brake of the drum 10 tightens and brakes the crown wheel 3, the brake of the screw 9 is released and with the gear 8 driven by the flywheel 5 working, the cab rises back. In this process, only part of the kinematic chain of the differential mechanism is used when the gear ratio wheel 3 is stopped.

$$U_{14}{}^{(3)} = 1 + \frac{Z_3}{Z_1} \tag{2}$$

The need for self-regulation of the gear ratio of the drive of the flywheel energy storage occurs with stepless control of the speed of the machine. For example, it turns out to be useful in automatic transmissions when simultaneously changing the speed and torque by one control body of a transport vehicle. The best case is when the action of this general control of the engine power and the speed of the transport vehicle will also control the recovery of braking energy, i.e. its direction to the flywheel drive and back. Ideally, you can try to provide a constant total value of the kinetic energy of the flywheel and the drive in the mechanical drive system of the machine. In this case, the control system of the lifting machine can be greatly simplified by using a diesel engine equipped with an automatic speed controller of the crankshaft as a drive engine. When creating any control system, it is preferable to use obviously stable elements with self-regulating properties.

3 Results and discussion

Figure 2 shows a block diagram of a single-row plane differential mechanism of a well-known design, which is combined with the velocity plan (on the right) in the coordinate system: the speed of the differential wheel point depends on the radius of the contact point of the differential wheels r. The images of the velocity plans of the planetary differential are presented in the form of rays emanating from the origin, where: V is the linear velocity of a point on the radius of the point r of the planetary differential. The velocity plans of the solar gear 1, the central gear 3 and the driver 4 (links 1, 3 and 4) proceed from the origin, the angular velocities ω of the links of the planetary differential are proportional to the tangents of the angles ψ of the inclination of the corresponding rays of the laws of velocities to the axis of radii G. In the poles of engagement P_1 and P_2 of different gears 1, 2, 3 linear the speeds of the contacting points of different links are equal to each other.

The speed plans correspond to a differential mechanism with a degree of mobility equal to two. For example, the dotted line shows the inclined beam 4* of the law V_{O2} of the change in the speed of the center of rotation point O_2 of the satellite 2 depending on the radius of the point r in the steady-state mode of motion and the inclined beam 1* of the law of the change in the speed of the solar gear 1. The inclined beam of the law of the change in the speed of the link 3 (central gear wheel) coincides with the axis of radius r, which indicates that the angular velocity $\omega_3 = 0$ and the energy consumed from the flywheel is zero. The speed of rotation of the shaft of the diesel engine is maintained constant by an automatic speed controller, therefore, in all modes of movement of the transport vehicle, the angle of inclination of the beam 1 can maintain a constant value.

When the car is braking, the beams 2, 4 on the speed plan (Figure 2) turn counterclockwise and occupy a position marked with continuous lines. The linear velocity V_3 of the gearing pole P_2 takes the maximum value, which corresponds to the maximum value of the angular velocity of the central gear wheel 3 connected to the flywheel of the energy storage.

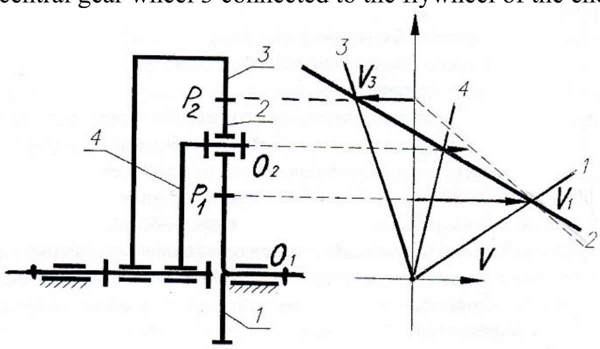

Fig. 2. Block diagram of a single-row flat differential and its speed plans: 1 – solar gear; 2 – satellite; 3 – crown wheel; 4 – driver.

The plan drawn in continuous lines in Figure 2 corresponds to both the end of the braking mode and the beginning of the acceleration process. The acceleration mode begins with the maximum rotation speed of the flywheel ω_3 connected to the central crown wheel 3. During acceleration, the energy from the flywheel and the operation of the traction motor connected to the solar gear are summed up and transmitted through the driver 4 and the gearbox to the lifting mechanism of the lifting machine. By the end of lifting the load, the speed of rotation of the flywheel due to its braking is reduced to zero.

Considering the speed change in Figure 2 in the opposite direction, it can be noted that it corresponds to the braking mode with energy recovery. This mode starts with the minimum rotation speed of the link 3 ω_3, associated with the minimum speed of the flywheel.

To determine the speeds of the links of the differential mechanism, the method of reversed motion is used and their movements relative to the driver are considered. It is possible to write down the ratio of the speeds of the links in contact of a flat single-row differential mechanism when the driver is stopped:

$$\frac{(\omega_1-\omega_4)}{(\omega_2-\omega_4)} = -\frac{Z_2}{Z_1} \tag{3}$$

$$\frac{(\omega_2-\omega_4)}{(\omega_3-\omega_4)} = -\frac{Z_3}{Z_2} \tag{4}$$

From the two equations obtained, the third speed can be determined by the known or given absolute rotational speeds of the two links. It is possible to determine the dependence of the gear ratio of any two links by rearranging the resulting equations, for example, multiplying the right and left parts of equations (1) and (2) we obtain:

$$\frac{(\omega_1-\omega_4)}{(\omega_3-\omega_4)} = -\frac{Z_3}{Z_1}$$

or in the form of:

$$\frac{\frac{\omega_1}{\omega_4}-1}{\frac{\omega_3}{\omega_4}-1} = -\frac{Z_3}{Z_1} \tag{5}$$

Solving the resulting equation (3) with respect to the gear ratio $u_{34} = \frac{\omega_3}{\omega_4}$, we obtain an equation relating the ratio of the transmission speed and link 3 of the kinematic chain of the differential mechanism:

$$u_{34} = \left[-u_{kop} + u_{14}{}^{3}\right] \cdot u_{31}{}^{(4)} \tag{6}$$

where $-u_{kop} = u_{14}$ - gear ratio of the gearbox.

The ratio of the rotational speeds of the transmission shafts ω_4 and the flywheel ω_3, depending on the gear ratio, is shown in Figure 3. This linear dependence changes sign at a critical value of the gear ratio of the gearbox equal to:

$$U_{14}{}^{(3)} = 1 + \frac{Z_3}{Z_1}$$

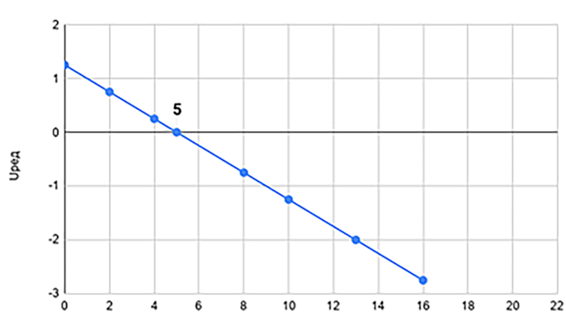

Fig. 3. Dependence of the ratio of the angular velocity of the flywheel to the transmission speed on the gear ratio of the gearbox: $u_{kop} = \frac{\omega_1}{\omega_4}$.

The area of use of a single-row differential mechanism for self-leveling the speeds of the flywheel and the transport machine lies in the negative area of the gear ratios. Earlier studies have shown that the control of the energy recovery of the machine is also possible with the help of a two-row differential mechanism with a negative gear ratio.

4 Conclusions

Based on the results of the study, the following conclusions can be drawn:

- Modeling of the characteristics of the energy recovery control system of a transport vehicle using a single-row planar planetary differential mechanism showed the prospects of its application using examples of a load-lifting device.
- To control the self-leveling of the speeds of the differential mechanism, a two-row differential mechanism with a negative gear ratio can be used.

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