Energy model of transport machines with braking energy recovery

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Abstract. Most of the time, modern transport vehicles operate in unsteady modes. Undoubtedly, the reasons for the decrease in the efficiency of machines are fluctuations in speed and load, their deviations from the optimal values cause an increase in energy losses. Another reason for the increase in energy losses is the process of forced braking when it is necessary to stop the car. A class of vehicles with hybrid propulsion systems that can recover braking energy are currently being developed. Significant advantages among them are transport vehicles with a flywheel energy storage, which have a long service life. This article discusses the energy model of transport vehicles with the possibility of braking energy recuperation.

1. Introduction
The main reason for the growth in energy consumption by machines is their operation in unsteady modes with frequent alternation of acceleration and braking. Therefore, a reduction in energy consumption can be achieved by regenerating the braking energy. Significant advances in the development of braking energy recovery have been achieved in transport hybrid propulsion systems.

2. Main part
Hybrid power plants of hoisting-and-transport vehicles (HTV) are also becoming more and more widespread in the world, since they can significantly reduce the loss of braking energy. In figure 1 shows a diagram of the regenerative drive of the overhead crane movement mechanism:

1. sun gear of the planetary differential 8;
2. satellite;
3. crown wheel;
4. carrier;
5. traction electric motor;
6. flywheel;
7. running wheels.

Before starting the movement, the electric motor 5 is idling, and the flywheel 6 rotates and has a reserve of kinetic energy accumulated during the previous braking. At the beginning of the movement, the traction motor 5 switches to acceleration. Acceleration is intensified by using additional flywheel energy. However, giving up energy during acceleration, the flywheel reduces the rotation speed and
when it drops to zero, the brake 9 is turned on. The steady motion occurs under the influence of the traction motor 5 when the flywheel 6 is braked.

Regenerative braking is carried out in two stages and begins by transferring the electric motor 5 to the generator mode of operation, simultaneously releasing the flywheel 6. At the same time, the speed of the crown wheel 3, connected to the travel wheels 7, decreases, and the speed of the sun wheel 1 and the flywheel associated with it increases. The speed of the sun wheel 1 connected to the flywheel 6 is increased. The kinetic energy of the crane movement is pumped into the flywheel 6 and in parallel is realized by the traction motor 5, operating in a generator mode and transmitting electrical energy to the network.

Figure 1. Scheme of the mechanism of movement of the bridge crane with a mechanical energy accumulator.

The second stage of regenerative braking begins when the speed of the carrier 4, connected to the electric motor 5, drops to zero, the control system switches the electric motor 5 to the motor mode, changing the direction of its rotation. The speed of the flywheel 6 continues to increase, and the speed of the travel wheels 7 decreases. After stopping the crane, the electric motor 5 is turned off or put into idle mode. When the crane is parked, the electric motor 5 and the flywheel 6 rotate without load. The operation of an overhead crane with the developed control system [1] for recuperation has been confirmed experimentally. In the "acceleration - deceleration" cycle, the braking losses without energy recuperation reach 40%, practically becoming commensurate with the share of useful work of 45%.

One of the urgent unsolved problems of designing hoisting-and-transport machines is the issue of optimal control of their drive with recuperation of braking energy into the flywheel accumulator. The only way to control the flywheel battery is to change the gear ratio of its drive. However, the theoretical foundations of the method for controlling the change in the gear ratio for recuperating the braking energy of hoisting-and-transport vehicles have not been fully developed until now [2].

In modern HTVs, various drive schemes for gear mechanisms of flywheel energy accumulators are used. Of particular importance is the decision on the choice of the optimal law of change in the gear ratio of the variator connecting the motor and the HTV actuator during braking energy recuperation [3]. Therefore, in many countries of the world and in Russia, research and development work continues to improve the designs of variators, which harmonize the mechanical characteristics of engines and working bodies of lifting and transport machines.

The structural model of the kinematic chain of the flywheel drive of the hoisting-and-transport machine during braking with a variable gear ratio is shown in figure 2 [4]. It includes an inertialess variator (3) changing the gear ratio of the flywheel drive (1).

Since the acceleration and dynamic loads acting during the braking process exceed the values of similar parameters during the acceleration process, we will focus on the choice of the optimal variator law during braking with energy recuperation. Let us pose the problem of determining the rational law
of changing the gear ratio of the variator according to the criterion of minimum energy losses during braking.

It should be noted that this dynamic model (figure 2) represents only the kinematic chain of one degree of freedom of the drive from the machine transmission to the flywheel using a speed variator. Therefore, this model does not include the main HTV drive from the engine, switched off during braking. The kinetic energy of the main motor is not recovered.

Figure 2. Block diagram of the lifting and transport machine with a flywheel energy accumulator: 1 - the moment of inertia of the flywheel; 2 - the moment from the transmission LCM with a reduced transmission moment of inertia of the machine; 3 – the variator.

When describing the dynamic behavior of the transmission under the influence of external disturbances and from the main engine, the main mover and the recuperation moment when changing the gear ratio, it is necessary to use at least a three-mass system.

The smallest losses of work for braking (part of the kinetic energy cannot be recovered due to the design limitations of the variator) are observed when all the kinetic energy of the machine previously lost during braking is transferred to the flywheel [5]. The law of variation of the variator gear ratio will be optimal if the kinetic losses are equal to zero and the total stock of kinetic energy during braking of the machine with energy recovery has a constant value at any moment of time:

$$T_x = T_{max} + T_f = const;$$

where $$T_x = J_x (\omega_x)^2$$ - the kinetic energy of flywheel, $$T_f = J_f (\omega_f)^2$$ - the kinetic energy of the machine drive, $$\omega_x = \omega_f, \omega_{max} = \omega_1$$ - the current values of angular velocities of transmission and flywheel drive link, $$J_1 = J_{max}$$ - the flywheel moment of inertia, $$J_2 = J_f$$ - the moment of inertia of the machine reduced to the transmission.

Equation (1) allows to solve two problems:

1. Determine the required moment of inertia of the flywheel at the selected maximum speeds of the working element and the flywheel. For this, the initial stock of kinetic energy of the HTV before braking and its stock in the flywheel after braking are equated.

$$\frac{(T_{max})_{max}}{2} = \frac{J_{max} (\omega_{max})^2_{max}}{2} = \frac{(T_f)_{max}}{2} = \frac{J_f (\omega_f)^2_{max}}{2} = const;$$

Relation (2) allows you to determine the moment of inertia of the flywheel $$J_{max}$$, necessary for the full use of the kinetic energy of the machine at a known moment of inertia of the machine reduced to the transmission.

$$\frac{J_f}{J_{max}} = \left[ \frac{(\omega_{max})_{max}}{(\omega_f)_{max}} \right]^2;$$

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2. Determine the necessary law of change of the variator transfer function when braking the hoisting-and-transport machine, taking into account the obtained necessary ratio of the moments of inertia (3), and substituting it into the equation (1):

\[
\frac{U_{\text{var}}(\omega_f)}{K_{\text{var}}} = \sqrt{\left(\frac{(\omega_f)_{\text{max}}^2}{(\omega_f)_{\text{max}}^2 - 1}\right)},
\]

(4)

where \( K_{\text{var}} = \frac{(\omega_{\text{max}})_{\text{max}}}{(\omega_f)_{\text{max}}} \) - the coefficient characterizing the transition process in the variator,

\( U_{\text{var}}(\omega_f) = \frac{\omega_1}{\omega_2} \) - the instantaneous value of the variator transfer function. This rational law of variation of the transfer function of the variator, obtained by modeling in MathCAD, is shown in figure 3, which is also applied the restrictions of the gear ratio of the variator \( (U_{12})_{\text{max}} = Y_{\text{max}} \cdot K_{\text{var}} \) and the range of variation of speed with recovery of braking energy \( D_{\text{rec}} = (\omega_2)_{\text{max}} - (\omega_2)_{\text{min}} \).

![Figure 3. Dependence of the variator transfer function on the ratio of the HTV transmission speed to its maximum value to ensure a constant value of the kinetic energy of the system.](image)

Since almost all known speed variators have a limitation on the maximum and minimum gear ratio, this will affect the economic characteristics of the hoisting-and-transport machine with a flywheel energy accumulator. With a limited maximum gear ratio \( (U_{12})_{\text{var}} \) further transfer of kinetic energy to the flywheel when the machine is braking will stop when it drops below the minimum transmission speed of the HTV:

\[
(\omega_2)_{\text{min}} = \frac{(\omega_1)_{\text{max}}}{(U_{12})_{\text{max}}},
\]

(5)

At the same time, a complete stop of the HTV machine should be provided by a conventional braking system with some energy losses, which is not a significant drawback, since in practice it often becomes necessary to control the transmission speed not only by changing the engine power, but also by the power of the forces of resistance to motion [6]. By limiting the maximum gear ratio, when the speed of the material handling vehicle is at its minimum, some of the kinetic braking energy will be lost. A certain part of the low-potential kinetic energy of the HTV cannot be used, which will cause the appearance of braking energy losses with a coefficient \( d \) (figure 4):
\[ d_{\text{torr}} = \left( \frac{(\omega_2)_{\text{min}}}{(\omega_2)_{\text{max}}} \right)^2; \]  

and limiting the efficiency of the energy recovery process sufficiently high value depending on the limitations of the gear ratio of the variator \( \eta_{\text{rec}} \) (6) or (7):

\[ \eta_{\text{rec}} = 1 - \chi_{\text{cycle}} = \frac{(\omega_2)_{\text{max}}^2 - (\omega_2)_{\text{min}}^2}{(\omega_2)_{\text{max}}^2}; \]

\[ \eta_{\text{rec}} = \frac{2(\omega_2)_{\text{middle}} - D_{\text{rec}}}{(\omega_2)_{\text{max}}^2}; \]  

where \( D_{\text{rec}} = (\omega_2)_{\text{max}} - (\omega_2)_{\text{min}} \) - the range of change of speed of the transport machine at recovery, \( (\omega_2)_{\text{middle}} = \frac{(\omega_2)_{\text{max}} - (\omega_2)_{\text{min}}}{2} \) - the average speed in the recovery range.

Since the optimal law of changing the gear ratio of the variator does not depend on the moments of forces acting in the process of energy recuperation, the machine speed control system can be used to control the variator (figure 5).
When measuring current speed $\omega_2$ of the transmission shaft it is possible to build a closed system control software (CS) $U(\omega_f) = U(\omega_{\text{initial}}) + k_{us}\omega_2$. Parametric synthesis of the CS should start with calculating the required gain $k_{us}$ of CS at the maximum starting and ending parameters of the braking. After that, its calculated dynamic qualities and stability margin should be analyzed by methods of the theory of automatic control. However, it is first necessary to carry out the structural synthesis of the CS, which consists in constructing the structural diagram of the CS and in determining the type of its amplifying elements. A tachogenerator can be used as a speed sensor. The creation of a workable CS will depend on the type of programming device and actuator that transforms the control signal into a gear ratio. At the first stages of the development of diesel acceleration regulators, cam mechanisms acting on the fuel supply organ were used as a programming device. Currently, it is common to use a computer as a programming device, on which numerous auxiliary functions can be assigned. The need for this is dictated by the fact that the initial and final parameters of acceleration and deceleration in operation are random functions associated with the specific conditions of movement of the transport vehicle. Therefore, it is advisable to use electrical signals as control signals from element to element. That is why it is advisable to use an electrical control signal in the actuator [7].

The gear ratio control system can be combined with the vehicle speed control system. It is advisable to use electrical signals as control signals transmitted to the control system from element to element. It is advisable to use a reversible motor-generator in the gear ratio control circuit.

3. Conclusion
The analysis revealed some properties of the proposed rational law of changing the gear ratio $U_{\text{var}}(\omega_f)$ by braking the machine:

- The proposed law of change in the gear ratio $U_{\text{var}}(\omega_f)$ is determined by the change of the speed of the transmission of the machine $\omega_f$. Therefore, the variator can be connected to the automatic control of the machine speed without using a load impulse. In view of the simplicity of including the flywheel battery in the automatic control of the speed of the machine and due to its reliability, its use becomes preferable to an electric energy storage;
- Since all known speed variators have limitations on the maximum and minimum gear ratio, this will affect the economic characteristics of the transmission mechanism with a flywheel energy accumulator. With a limited maximum gear ratio $(U_{12})_{\text{max}}$ (figure 4), further transfer of kinetic energy from the transmission to the flywheel will stop when the vehicle is braking below the minimum transmission speed of the vehicle.

A complete stop of the hoisting-and-transport machine in a given position should be provided by a conventional braking system with some kinetic energy losses.

$$\left(\omega_2\right)_{\text{min}} = \frac{(\omega_f)_{\text{max}}}{(U_{12})_{\text{max}}}.$$  \hfill (9)

A full stop of the transport machine in a given position must be provided by a conventional braking system with some kinetic energy losses.

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