Optimization of the Lifting Machines' Hoisting Mechanism Design Scheme

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Abstract. Increasing the productivity of lifting machines and their reliability is possible by reducing the dynamic loads that occur at the lifting mechanism starting and braking. Carried out is the analysis of methods to reduce the lifting machines' hoisting mechanism dynamic loads during the acceleration period when load lifting. The method chosen for our study is reducing the mass inertia moments of parts located on the hoisting mechanism's slow-speed shafts. The study aimed to reduce the hoisting mechanism dynamic loads at cargo lifting by optimizing the gear drive mechanism design scheme. One of the promising methods to reduce dynamic loads in machine drives consists of the multithreading principle use. A new design scheme for the gearbox has been developed. This structure represents a multi-threaded two-stage gearbox, where each stage consists of a central gear, intermediate wheels, which axes are fixed in the housing, and a gear wheel with internal gearing. It has been established that using intermediate wheels of different diameters can obtain different gear ratios of the gearbox. It is determined that the minimum gear ratio for a two-stage multithreaded gearbox is 9. With such a gear ratio of the gearbox, the dynamic loads that occur during the lifting mechanism drive start-up are minimal. Accordingly, the maximum efficiency of the lifting mechanism per its operation cycle was obtained. Also, the optimal gear ratio ensures the minimum dimensions of a multi-threaded two-stage gearbox.

Keywords: Gear Ratio · Moment of Inertia · Dynamic Loads · Product Innovation

1 Introduction

In modern mechanical engineering, there are two trends in improving the machine drive:

– ensuring the working body's smooth acceleration and braking by reducing the inertia moment of the drive's moving mechanical parts: clutches, gears, drums, and other parts. It allows to improve the dynamic characteristics and significantly reduce the dynamic load during acceleration and braking and thus decrease the drive's efficiency factor. For example, in [1], the issues of ensuring the crane lifting mechanism reliability are considered. It is especially important when high explosive risk cargo transporting. The reliability of such mechanisms can be improved by slowing down their wear process, eliminating the source of wear, and strictly observing the operation and maintenance rules. The paper does not provide quantitative values on the probability of the lifting mechanism's failure-free operation;

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– the transition from a drive operated with a mechanical gearbox to a gearless drive increases the efficiency factor due to the rejection of parts characterized by a significant inertia moment

To implement the second trend, i.e., the gearbox rejection embodying structure, work is underway to create low-speed electric motors with acceptable mass-dimensional indicators. Synchronous machines with permanent magnets are used at gearless drives in the valve motor mode. Such low-speed motors are not widely used due to the scarcity and relatively high cost of permanent magnets needed to create those machines. In addition, such motors' repair and disposal costs are approaching the cost of a new drive. Also, gearless drives for machine mechanisms are created based on asynchronous electric motors with short-circuited rotors using frequency converters. Frequency converters ensure their torque constancy when the speed of the controlled electric drive decreases. Still, at the same time, there is a decrease in the useful power with a corresponding decrease in the efficiency factor value. Artificially reducing the rated speed of the motor in a wide range lead to a significant decrease in the efficiency factor since at the control range equal to 8, the efficiency factor decreases up to 53%. To increase the energy efficiency of machine drives, it is advisable to engage the method of minimizing the electric motor nominal speed artificial reduction range and, at the same time to provide the required speed of the machine's working body due to the additional use of a gearbox with a high-efficiency factor [2]. Thus, the research and development of an energy-efficient hoisting mechanism for lifting machines is highly relevant, being of actual scientific interest, and has practical value.

This study aims to reduce the dynamic loads of lifting mechanisms when lifting cargo by optimizing the gearbox design scheme.

This study's objectives: development of a gearbox design scheme with improved dynamic properties; determination of optimal gear ratios for the gearbox with improved dynamic properties.

2 Literature Review

Given that the problem of creating highly efficient lifting machines is urgent, researchers and engineers from different countries worldwide are searching for a solution.

The method for increasing the efficiency of the tower crane mechanisms' joint operation is proposed in [2]. A methodology of selecting the mechanism's optimal structure from a variety of possible solutions by introducing a quality criterion for each structure is developed, and a set of parameters optimized for the movements full scope is determined. The method was illustrated for a crane with both lever mechanisms and rope mechanisms.

Large dynamic loads during the bridge-type cranes' lifting mechanism start-up led to breakdowns of the main parts and sometimes to accidents since the method used to prevent these loads does not provide the necessary accuracy and reliability of overload protection. Developed is a bridge-type crane protection system [3], which allows the implementation of a new method of overload protection, namely, lifting mechanism moving parts disconnecting translationally with simultaneous disconnection of this mechanism drive electric motor. To ensure the required safe operation of the crane, it is necessary to develop constructive schemes of the lifting mechanism capable of excluding so-called systematic overloads efficiently.

It is proposed to decrease peak loads when starting the crane lifting mechanism by reducing the inertia moment of coupling with the brake pulley [4]. Namely, the point is to manufacture a coupling made of composite materials. It is proposed to reduce the inertia moments by increasing the start-up and acceleration time of the lifting mechanism drive. However, reducing the start-up and acceleration time leads to a decrease in the crane's performance.

The influence of rope stiffness on dynamic loads arising in the metal structure of overhead cranes is considered in [5]. When simulating the dynamics of the cargo lifting process with a bridge crane, two dynamic crane models are proposed, rendering it possible to clarify the magnitude of dynamic loads on the metal structure during the lifting mechanism operation.

For all lifting cranes, of key relevance is to ensure high reliability. Therefore, their emergency braking is used in addition to the normal braking of lifting mechanisms [6]. Emergency braking is unavoidable in several situations, for example, when the electrical power is suddenly cut off. The study analyzes the emerging dynamic loads during emergency braking and when braking with a mechanical brake.

To assess the bridge-type crane dynamic loading, a system for crane load monitoring and diagnostics is proposed [7]. This system is developed based on experimental data obtained in various laboratory tests.

In [8], a new system for cargo lifting process control was developed; a controller proposed for this system uses two signals: this one of angular travel and this displacement integral, combining them into a new signal of coupled dissipation. The proposed controller effectively suppresses dynamic load fluctuations at the expense of increasing passivity.

The mathematical model of the bridge crane lifting transmission system is analyzed in [9]. The relationship between the dynamic load stiffness coefficients and the crane drive parameters is established. The bridge crane dynamic model is analyzed for non-stationary modes of lifting mechanism operation [10]. The bridge crane oscillation amplitude is established for these modes, and recommendations on dynamic load reduction are given.

Work [11] is devoted to the study of planetary gears dynamics. Models of the basic structures of planetary gears have been developed using computer software; graphs of carrier angular acceleration have been constructed without account and taking into account the gears' gaps; it is shown that the smallest acceleration of the carrier increases by 1.5 times if taking into account the gaps at the gears.

Studies of the gear-roller planetary gear structure and dynamics have been carried out in [12]. A replacement mechanism was used for this study. The parameters of the replacement mechanism have been considered. The paper [13] presents a methodology for building a mathematical model of a planetary gearbox with rigid links of its planetary series, an example of modeling a dynamic engine – torque converter – planetary gearbox – automobile – road system is given. The simulation results show that the planetary transmission system has three modes of operation [14]. The main frequencies include the motor rotation speed and the frequency of gearing.

The machine drives use planetary gears, which are created on the principle of multithreading, therefore allowing to significantly reduce the size and weight of gearboxes using planetary gear. The principle of multithreading is used not only for transmissions with rotational motion but also for mechanisms implementing the translational one. The roller transfer mechanism [15] is based on the principle of multithreading; in this mechanism, the power flow is divided because it consists of two rows of rollers arranged in staggered order. The analysis of the works under consideration shows that these studies aimed to reduce dynamic loads in machine drives. Still, simultaneously, the problem of creating highly efficient machine drives requires continued research and development of new drive design schemes for various machines, including cargo lifting machines.

3 Research Methodology

3.1 Analysis of Methods for Reducing the Lifting Machines' Hoisting Mechanism Dynamic Loads During Acceleration When Cargo Lifting

With the lifting mechanism's steady movement characteristic for the repeated-short-term operation of lifting machines, the torque on the electric motor shaft must comply with the condition:

$$M_{dv} = M_{st} + M_d \le \psi_{av} \cdot M_n, \tag{1}$$

where M_{dv} is the motor shaft torque, Nm; M_{st}- the static torque on the motor shaft when lifting the nominal load, Nm; M_d- the dynamic torque on the motor shaft determined from the condition of providing the acceleration necessary for speeding up, Nm; M_{H}- the nominal torque of the electric motor, Nm; ψ_{av}- the average starting torque multiplicity.

The dynamic torque on the motor shaft, determined from the condition of providing the acceleration necessary for speeding up, is determined by the dependence:

$$M_d = I_{pr} \frac{\omega}{[t_n]^2}, \tag{2}$$

where is the I_{pr} inertia moment of the lifting mechanism moving masses and the lifted load brought to the motor shaft during the start-up period when cargo lifting, $kg{\cdot}m^2$; ω - the current angular velocity of the motor rotor rad/s; $[t_n]$ – the permissible duration of the start, s, equals to:

$$[t_n] = \frac{V_f}{[a_n]}, \tag{3}$$

where $[a_n]$ is the permissible acceleration at start-up, m/s^2; V_f is the actual speed of the load-bearing element, m/s.

The design scheme of the lifting machines' hoisting mechanism includes an electric motor, a clutch with a brake pulley, a gearbox, a drum, and a chain hoist. Taking into account the lifting mechanism design scheme and based on the analysis of dependencies (2) and (3), the following methods for reducing dynamic loads can be formulated: method of reducing the motor rotor's moment of inertia; method of reducing the inertia moment of the clutch with the brake pulley; method for reducing the masses inertia moments of parts located on the slow-speed shafts of the lifting mechanism; method of regulating in transients the angular velocity of the lifting mechanism drive's electric motor rotor during the start-up period, that is, thought the use of controlled transient processes.

The motor rotor's inertia moment can be reduced by replacing one motor with two or more while maintaining the total power level. As a rule, the total inertia moment of two half-power motors turns out to be less than the inertia moment of one full-power motor within the same motors series.

The inertia moment of the clutch with the brake pulley can be reduced by using composite materials.

The rational design of the mechanical transmission (selection of the optimal gear ratio) reduces the inertia moments of the masses of parts located on the lifting mechanism's slow-speed shafts.

3.2 Development of a Design Scheme for a Gearbox with Improved Dynamic Properties

One of the promising methods of reducing dynamic loads in machine drives is using the multithreading principle in their design elaboration. The multithreading principle allows to divide the power flow into several parallel branches. As the experience of using gears with power flow separation shows, the mass of multi-threaded gears is 1.5–3 times less than the mass of single-threaded gears. Accordingly, the moment of inertia at such gears is also smaller, and this allows the reduction of the dynamic loads in the machine drives. Multithreaded principles can be arranged as so-called simple gears and planetary gears.

Planetary gears are widely used in mechanical engineering as they have many advantages: small dimensions and less weight; large gear ratios in one stage; it is possible to use gears with internal gearing characterized with increased load-bearing capacity; less noise producing compared to simple gears.

However, planetary gears require increased manufacturing accuracy and a larger number of parts and are more difficult to assemble than gears with fixed axes. The efficiency factor drops sharply with the increase in the gear ratio.

A significant disadvantage of planetary gears is that they have a large moment of inertia at launch because, at drive acceleration, it is necessary to overcome the resistances associated with both the carrier's moment of inertia and the satellites' moment of inertia. To eliminate this drawback, it is proposed to use a simple multithreaded gear, which can be represented as a planetary gear with a stopped carrier.

Figure 1 shows a design diagram of a multithreaded two-stage gearbox, each consisting of a central gear 1, intermediate gears 2, which axes are fixed in the housing, and a gear wheel 3 with internal engagement. Similarly, the second stage is made with a central gear 5, intermediate wheels 4, and a wheel with an internal gearing 6.

In a simple multithreaded gear, the central gear dimensions are smaller than in a simple single-threaded one since each gearing transmits only a part of the entire load.

A simple multithreaded gear, having almost the same advantages as a planetary type one, is easier to assemble, does not require increased manufacturing accuracy, and due to design features, the reduced inertia moment of rotating gears is significantly less than that of a planetary transmission gear.

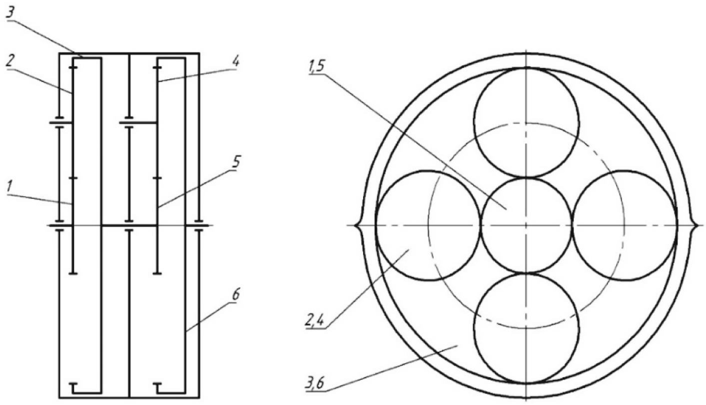

Fig. 1. Design scheme of a multithreaded two-stage gearbox.

3.3 Determining the Optimal Gear Ratios of the Gearbox Design Scheme with Improved Dynamic Properties

An important direction of lifting machines improvement is to increase their productivity, on which the overall success of industry, transport, and construction depends to a certain extent.

To increase the productivity of lifting machines, it is necessary to reduce, all other things being equal, their start-up time as well as the braking time. Reducing the start-up and braking time increases the dynamic loads of the machine mechanisms since, to achieve the machine's working speed in this case, it is necessary to increase acceleration (deceleration) during the specified periods.

An increase in dynamic loads at start-up leads to a decrease in the machine's efficiency during this period and, accordingly, a decrease in the machine's efficiency during the work cycle.

By applying the method of reducing the masses inertia moments of parts located on the lifting mechanism's slow-speed shafts, it is possible to reduce dynamic loads during start-up. Since the dynamic moment on the lifting mechanism drive electric motor shaft depends on the inertia moment of the lifting mechanism and the lifted load moving masses brought to the motor shaft during the start-up period, to determine this moment, we consider the calculation scheme shown in Fig. 2.

The equation for the inertia moment reduced to the motor shaft has the form:

$$I_{pr}\frac{\omega_1^2}{2} = I_1\frac{\omega_1^2}{2} + I_2\frac{\omega_1^2}{2} + I_3\frac{\omega_1^2}{2} + I_4\frac{\omega_2^2}{2\cdot\eta_1} + I_5\frac{\omega_3^2}{2\cdot\eta_2} + \\ + I_6\frac{\omega_3^2}{2} + I_7\frac{\omega_4^2}{2\cdot\eta_3} + I_8\frac{\omega_5^2}{2\cdot\eta_4} + I_9\frac{\omega_5^2}{2\cdot\eta_5} + m_c\frac{v_c^2}{2\cdot\eta_6}, \quad (4)$$

where, I_{pr}– the inertia moment of the mechanism brought to the motor shaft at the start–up; ω_1, ω_2, и ω_3– angular velocities, respectively, of the motor shaft, the first

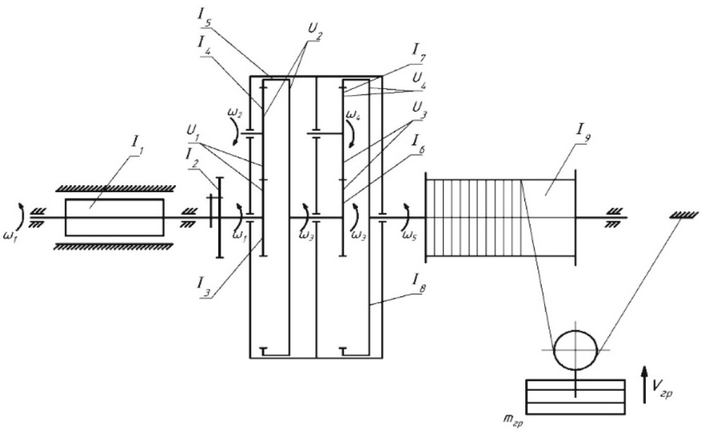

Fig. 2. The cargo lifting mechanism's scheme for bringing moving masses.

and second stage shafts of the gearbox, and the drum shaft; $I_1, I_2, I_3, I_4, I_5, I_6, I_7 I_8, I_9$,– inertia moments, respectively, of the motor rotor, clutch with brake pulley, the first and second stage gears of the gearbox and drum; v_c, m_c- respectively, the speed and weight of the cargo being lifted; $\eta_1, \eta_2, \eta_3, \eta_4, \eta_5, \eta_6$– respectively, the efficiency of the first gear engagement and the second, third, and fourth gear engagements, the efficiency factor of the mechanism from the drum to the motor shaft; the efficiency factor of the lifting mechanism, including the efficiency factor of the chain hoist. Substituting in (4) $\mathrm{v_c} = \frac{\omega_5 \cdot r_6}{\mathrm{a \cdot u_1 \cdot u_2 \cdot u_3 \cdot u_4}}$ $\omega_2 = \frac{\omega_1}{u_1}$; $\omega_3 = \frac{\omega_1}{u_1 \cdot u_2}$; $\omega_4 = \frac{\omega_1}{u_1 \cdot u_2 \cdot u_3}$; $\omega_5 = \frac{\omega_1}{u_1 \cdot u_2 \cdot u_3 \cdot u_4}$, and dividing the left and right parts of expression (4) into $\frac{\omega_1^2}{2}$, we get:

$$I_{pr} = I_1 + I_2 + I_3 + I_4 \frac{1}{u_1^2 \cdot \eta_1} + I_5 \frac{1}{(u_1 \cdot u_2)^2 \cdot \eta_2} + $$
$$+ I_6 \frac{1}{(u_1 \cdot u_2)^2} + I_7 \frac{1}{(u_1 \cdot u_2 \cdot u_3)^2 \cdot \eta_3} + I_8 \frac{1}{(u_1 \cdot u_2 \cdot u_3 \cdot u_4)^2 \cdot \eta_4} + $$
$$+ I_9 \frac{1}{(u_1 \cdot u_2 \cdot u_3 \cdot u_4)^2 \cdot \eta_5} + m_c \frac{r_b^2}{(a \cdot u_1 \cdot u_2 \cdot u_3 \cdot u_4)^2 \cdot \eta_6}, \quad (5)$$

where, $u_1,\ u_2,\ u_3,\ u_4$, – respectively the gear ratios of the first, second, third, and fourth gears; r_b– the drum radius; a – the chain hoist multiplicity.

Considering the power flow from the motor to the drum and taking $\eta_1 = \eta_2 = \eta_3 = \eta_4 = \eta_5 = 1$, after the transformations, we write:

$$I_{pr} = I_1 + I_2 + I_3 + I_4 \frac{1}{u_1^2} + (I_5 + I_6) \frac{1}{(u_1 \cdot u_2)^2} + $$
$$+ I_7 \frac{1}{(u_1 \cdot u_2 \cdot u_3)^2} + I_8 \frac{1}{(u_1 \cdot u_2 \cdot u_3 \cdot u_4)^2}, \quad (6)$$

The reduced inertia moment of the lifting mechanism, according to expression (6), will have the smallest value, all other things being equal, when using a multithreaded two-stage gearbox made according to the design scheme represented in Fig. 1 with gear ratios $u_1,\ u_2,\ u_3,\ u_4,$ if the following condition is met:

$$\left\{ I_3 + I_4 n \frac{1}{u_1^2} + (I_5 + I_6)\ \frac{1}{(u_1 \cdot u_2)^2} + I_7 n \frac{1}{(u_1 \cdot u_2 \cdot u_3)^2} + I_8 \frac{1}{(u_1 \cdot u_2 \cdot u_3 \cdot u_4)^2} \right\} \Rightarrow min, \tag{7}$$

where n is the number of intermediate gears in one stage.

Since the first and second stages of the gearbox are similar, we consider condition (7) for the first stage:

$$\left\{ I_{pr} = I_3 + nI_4 \frac{1}{u_1^2} + I_5\ \frac{1}{(u_1 \cdot u_2)^2} \right\} \Rightarrow min, \tag{8}$$

Let us transform the expression (8) by assuming that the formula determines the inertia moments of each of the gear wheels in this gearbox:

$$I = \frac{G \cdot D^2}{7 \cdot g} \tag{9}$$

where I is the inertia moment of the gear mass, kgm·s^2; D – the gear diameter, m; G – the rotating gear weight, kg; g– free fall acceleration, m/s^2.

The weight of a rotating gear wheel can be found as follows:

$$G = V \cdot \gamma \tag{10}$$

where, $V = \frac{\pi \cdot D^2}{4} \cdot h$– the cylindrical gear volume, m3; h– the gear width, m; $\lambda\gamma$– the specific weight of its material, kg/m^3.

Taking for the central gear $h = D_3$, and for the intermediate wheels $h = 0,25D_3$ and considering that $\frac{D_4}{D_3} = u_1$; $\frac{D_5}{D_4} = u_2$; $D_4 = D_3 \cdot u_1$; $D_5 = D_3 \cdot u_1 \cdot u_2$, we find expressions for moments of inertia:

$$I_3 = \frac{\pi}{4 \cdot 7g} \cdot \gamma \cdot D_3^5;\ n \cdot I_4 = \frac{\pi}{4 \cdot 7g} \cdot \gamma \cdot D_3^5 \cdot u_1^5;\ I_5 = \frac{\pi}{4 \cdot 7g} \cdot \gamma \cdot D_3^5 \cdot (u_1 + u_2)^5, \tag{11}$$

Substituting the corresponding values $I_3,\ nI_4,\ I_5$ into expression (8), after the transformations, we get:

$$\left\{ \frac{\pi}{4 \cdot 7 \cdot g} \cdot \gamma \cdot D_3^5 \left[1 + u_1^3 + (u_1 \cdot u_2)^3 \right] \right\} \Rightarrow min \tag{12}$$

Taking into account the fact that the central wheel diameter D_3 can be found based on the strength of the teeth by contact stresses, expression (12) takes the form:

$$\left[1 + u_1^3 + (u_1 \cdot u_2)^3 \right] \Rightarrow min \tag{13}$$

By expression (13), we can determine at what gear ratios of the multithreaded gearbox stages u_1, u_2 there will be minimal dynamic loads of the cargo hoisting mechanism at lifting machines.

4 Results and Discussion

By replacing one engine with two or more, while maintaining the total power of the drive of the lifting mechanism, a decrease in the moment of inertia of the engine rotor is achieved. Using composite materials can reduce the moment of inertia of the clutch with a brake pulley.

Multi-threading allows you to divide the power flow into several parallel branches; therefore, the mass of multi-thread gears is 1.5–3 times less than that of single-thread gears. One of the options for a simple multi-threaded transmission is proposed in the form of a multi-threaded two-stage gearbox (Fig. 1).

Using intermediate wheels of different diameters, obtaining different values of the gear ratios at the gearbox first stage is possible. Table 1 shows the possible options, taking into account that $u_1 = \frac{D_4}{D_3}$; $u_2 = \frac{D_5}{D_4}$; $D_5 = D_3 + 2D_4$; $D_3 = 34\,mm$.

Table 1. The possible options.

Options	u_1	u_2	$u_1 \cdot u_2$	D_5, mm
$D_4 = D_3$	1	3	3	$D_5 = 3D_3 = 102$
$D_4 = 1{,}5D_3$	1.5	4/15	4	$D_5 = 4D_3 = 136$
$D_4 = 2D_3$	2	5/2	5	$D_5 = 3D_3 = 170$
$D_4 = 2{,}5D_3$	2.5	6/2,5	6	$D_5 = 6D_3 = 204$
$D_4 = 3D_3$	3	7/3	7	$D_5 = 7D_3 = 238$

Analysis of the data given in Table 1 shows that the diametrical dimensions (D5) of the gearbox are relatively small and increase with increasing gear ratio $u_1 \cdot u_2$.

Analyzing the expressions (13) and taking into account expressions (11), we state that the smallest value of the reduced inertia moment and the smallest dynamic loads for the first stage of a multithreaded reducer will be identified at $u_1 = 1$, $u_2 = 3$. The values of the reduced inertia moment and the minimum dynamic loads will also be relevant for the gearbox second stage. For a two-stage multithreaded gearbox, dynamic loads will be minimal at a gear ratio equal to 9.

The studies in this paper offer a new method for designing lifting mechanisms for hoisting machines compared to the method for reducing inertial masses proposed in [4] and the method for reducing dynamic loads in planetary gear [11]. Still, at the same time, the results obtained do not contradict earlier studies.

5 Conclusions

An efficient method to reduce dynamic loads during the start-up period at cargo lifting will be the method of reducing the inertia moment of the masses of the parts located on the lifting mechanism's slow-speed shafts.

One of the promising methods of reducing the machine drives' dynamic loads is using the multithreading principle in the design.

The principle of multithreading has been applied when creating planetary gears. These gears have many advantages, but a significant disadvantage of planetary gears is that they have a significant inertia moment at launch because, during the drive acceleration, it is necessary to overcome the resistance associated with both the inertia moments of satellites and the inertia moment of the carrier.

A new constructive scheme of the hoist mechanism reducer is proposed, which is a simple multi-threaded gear train.

It is established that the reduced inertia moment of the two-stage multithreaded gearbox gears depends on the sum of the squares of the first-stage gear ratios and the gearbox as a whole. The value of the reduced inertia moment and dynamic loads for a two-stage multithreaded gearbox will be minimal when the gearbox gear ratios are equal to 9.

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