# Evaluation of Dynamic and Energy parameters of a Tower Crane with a Frequency-Controlled Drive

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**Abstract.** The purpose of the study is to evaluate the energy parameters of tower cranes when using a frequency-controlled drive mechanism. The relevance of the study lies in the fact that existing tower cranes quickly become obsolete. However, there is an alternative to buying new cranes – the modernization of existing ones with the improvement of their parameters and the scientific justification for increasing the efficiency of work. The following methods are used for research: method of equivalent power and the methods of metal fatigue theory. The main scientific novelty is the approach to assessing the possible increase in the service life of crane elements when using a frequency drive based on the theory of fatigue. Also presented are examples of equivalent power calculations for the working cycles of the turning mechanism with various forms of speed change in acceleration and deceleration modes. Numerical estimates are given for increasing the service life of the main types of welded joints with different values of stress concentration factors. Showing ways to further improve the efficiency of tower cranes and their mechanisms due to more optimal transients. The proposed approaches can be used when choosing the type of drive for the designed crane or to evaluate the economic efficiency of replacing the drive.

## 1. Introduction

Tower cranes have a significant spread. However, most of them have used part of the resource of their metal structures or have an outdated control system. At the same time, the continued use of existing cranes may be economically feasible compared to replacing them with new cranes in case of ensuring energy efficiency and further safe operation. This can be achieved, for example, by reducing dynamic loads and rational control laws. Suboptimal operating modes of the mechanisms also lead to excessive energy consumption. First of all, this is relevant for the turning mechanisms of tower cranes. In addition to the economic component, the saving of electrical energy is an important part of the general trend of environmental protection. Thus, to justify the expediency of continuing the operation of tower cranes, it is necessary to ensure smooth operation of their mechanisms and energy efficiency. After modernization, it is necessary to quantitatively assess the increase in the service life of the metal structure units and mechanisms and the reduction of energy consumption. Determining the dependencies on which these calculations are based is the purpose of this article.

## 2. Literature Review

In the article [1], the main attention when performing a dynamic analysis of a tower crane was paid to the influence of vibration values during the operation of the slewing mechanism. Several simplified dynamic models were built in this study. Dissipative properties were not taken into account in the main parameters.

In articles [2-3], the authors conducted a dynamic analysis of the jib crane slewing mechanism. The level of dynamic loading of the mechanism was determined and relevant conclusions were drawn. However, in these works, during the mathematical modeling, the effect of the dynamic mechanical characteristics and the corresponding equations of motion of the asynchronous electric drive were not taken into account.

In the article [4], it is proposed to carry out fast positioning of the load of a jib crane fixed on a flexible suspension with the reduction of energy losses in the drive using linear algebra methods.

In the work [5], the main attention was paid to the amount of electrical energy losses in the electric drive of the load lifting winch of the jib crane. A mathematical model with the possibility of electricity recovery was developed. Theoretical calculations were confirmed by experimental studies.

In articles [6-7], the authors developed a system of non-linear control of a four degree of freedom feedback controller for a marine jib crane. However, the proposed optimal control does not take into account wind gusts that may act on the load.

In the article [8], the optimal control of the jib crane slewing mechanism was studied according to the speed of action. A nonlinear mathematical model was used for calculations. The variable parameter is the length of the flexible suspension.

In work [9] dynamic and kinematic loads of a jib crane when lifting a fixed load from below to a mountain were

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investigated. Lagrange's equation of the second kind was used to construct the mathematical model. However, in this study of dynamic loads, the corresponding dissipative forces were not taken into account.

In the article [10], the dynamics of the "trolley–load on a flexible suspension" system was modeled with different approaches to the formulation of the problem. The disadvantage of this study is that only the interaction of the trolley and the load is considered, without taking into account dynamic phenomena in the metal construction elements of the load-bearing elements of the crane.

An existing model is known, which takes into account the geometric features of the load suspension point and the trajectory of its movement. The model presented in the paper [11] for the transportation of long load by two bridge cranes at different levels deserves attention. The given technological process is carried out jointly.

Based on the theory of longitudinal and transverse bending, the authors of the article [12] established the differential equations of the deflection of the n-stage model of the column with double cables and established the combined equations of the lifting of the telescopic jib.

The article [13] presents the modeling of a crane with a flexible rotary jib, which is actuated by hydraulic cylinders and is modeled as a planar multi-purpose system. The simulation results are confirmed by ANSYS finite element analysis. The derived model is the basis for the design of the slewing cylinders and can potentially be used to study the performance of the crane control system.

In the article [14], it is proposed to eliminate the fluctuations of the load of the ship jib crane by integrating two methods of optimal control. This is a particle swarm optimization (PSO) method combined with the use of a PID controller. Simulations with different weather conditions were performed and analysed to verify the effectiveness of the proposed integrated optimal control.

In the article [15], the authors performed the minimization of energy losses in the asynchronous frequency-controlled electric drive of the truck trolley drive mechanism of the tower crane. The magnitude of energy losses was compared with direct start and start with standard frequency converter parameters.

The article [16] presents a control scheme for a marine crane, which takes into account the influence of the dynamic location of the vessel on the physical model of the load. This makes it possible to control and stabilize a heavy load fixed on a flexible suspension. At the same time, the crane is located near the shore.

The article [17] proposed a control strategy for designing the optimal trajectory of a slewing crane with a flexible jib. The strategy allows to minimizing vibration in the studied system. To evaluate the effectiveness of the proposed model, numerical simulations and an experiment were conducted on a laboratory crane, where the movement of the jib end and the lifted load were measured.

In the paper [18], the authors propose to perform dynamic and static calculations during the loading and unloading operations of a marine jib crane. A spatial model of the crane under study was used, the jib system of which consists of two sections. The crane model takes into account the flexibility of the jib, cylinders and rope-block system. The finite element method is used to discretize the jib system.

From the review of recent literary sources, it is clear that the dynamics of cranes is studied in different settings (taking into account different elements of cranes) and with different purposes (study of dynamic processes, optimal control, design needs). However, the assessment of resource increase and energy savings is insufficiently covered. This is the main subject of this article.

## 3. Research Methodology

Studies of the tense strain state of the metal structure of the tower crane were carried out using the 3D model of fig. 1. The technical characteristics of the 3D model correspond to the basic parameters of the KB-403A crane under study. The finite element method in the Ansys application was used to determine the equivalent tension. An example of the results of the tension state calculation is presented in Figure 2. From the results of the study, it can be seen that the difference between the maximum stress values in the welded nodes under optimal and suboptimal control modes is at least 5%. Since the purpose of this study is to quantify the increase in the resource, we will focus on explaining the patterns that allow it to be carried out.

To estimate the increase in service life and the period between repairs of the crane's metal structure, we will use the fatigue curve. The fatigue curve reflects the dependence of the durability of a part or structural unit on the level of operating state under stationary loading and is the result of a step test of a sample with a symmetrical or non-zero variable load.

The inclined section of the fatigue curve, which characterizes the area of operational endurance from the point of transition from short cycle failure C to the point of transition to the zone of unlimited endurance D, is described by the equation:

$$\sigma^m \cdot N = \text{const}\,, \qquad (1)$$

where $m$ – indicator of the degree of the slope of the fatigue curve, $m = \text{ctg}\,\varphi$ .

The dependence is used to determine the coordinates of a point on the fatigue curve:

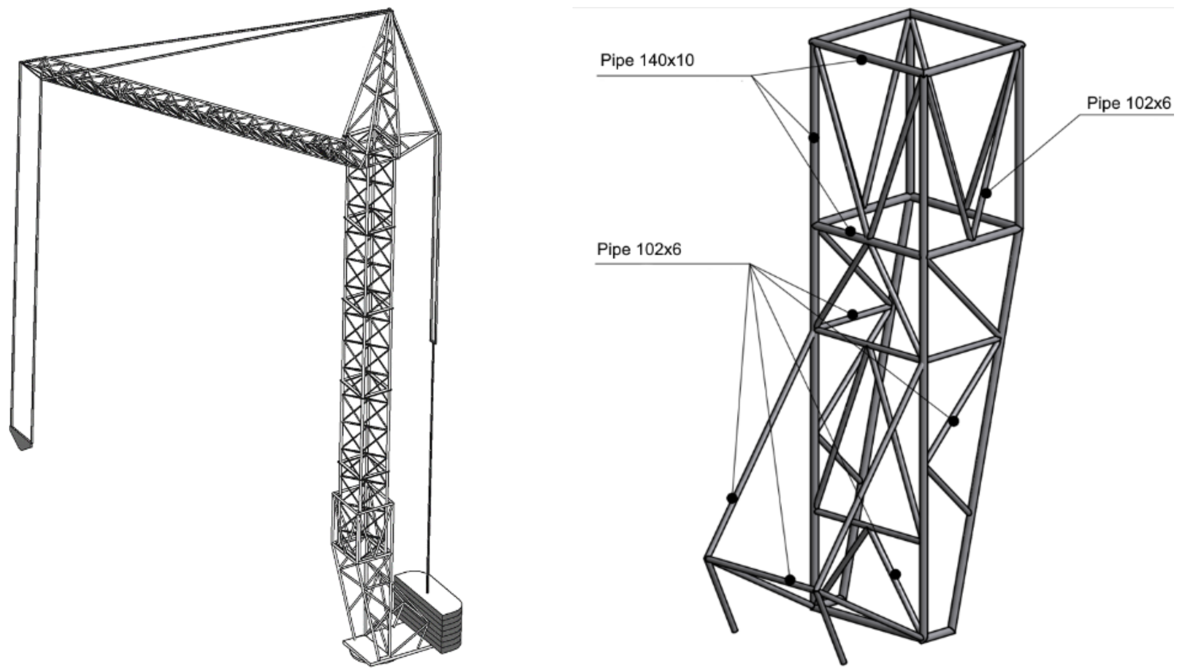
$$\sigma^m \cdot N = \sigma_{RK}^m \cdot N_0\,, \qquad (2)$$

where $\sigma$ and $N$ – coordinates of an arbitrary point on the inclined part of the fatigue curve.

$\sigma_{RK}$ – state of limited durability at the number of cycles to failure

$$N_0 = 2 \cdot 10^6\,. \qquad (3)$$

Each endurance curve is constructed in such a way that all state cycles can be fully determined both experimentally and through simulation of possible load cycles.




a B

a – general view of the metal structure of the crane; b – detailed view of a complex element - assembly section

**Fig. 1.** 3D model of the metal structure of the tower crane

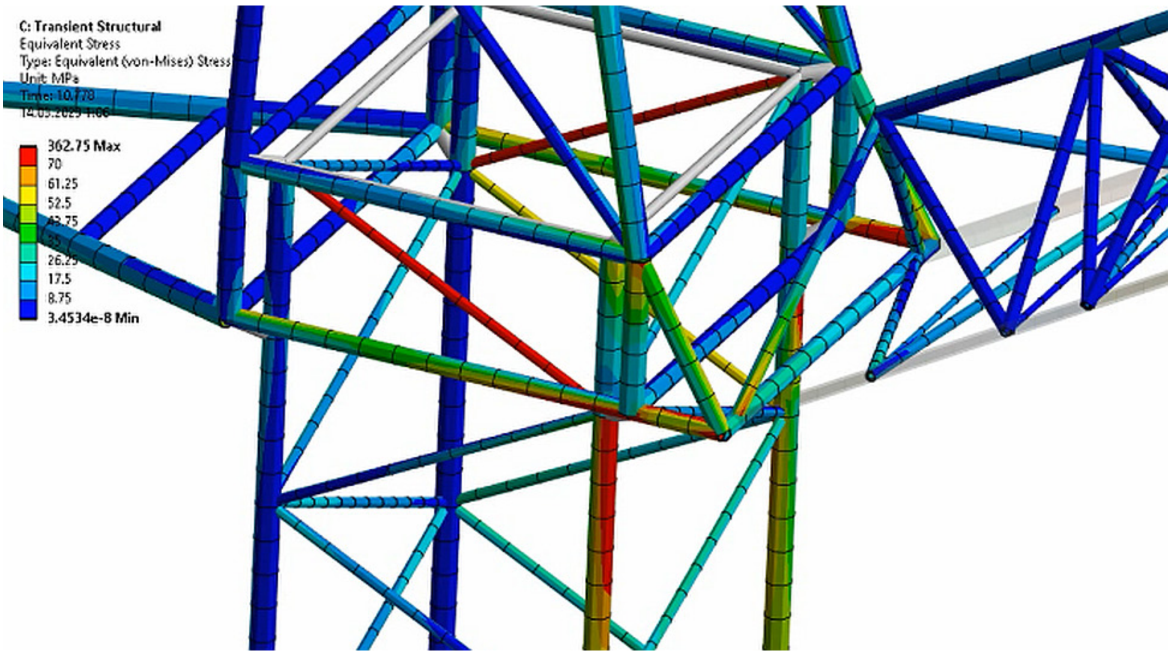



**Fig. 2.** The tense state of the upper section of the crane tower

To assess the achievable durability of a structural element, the number and magnitude of the individual loads that make up the set of loads are important. According to the Palmgren-Miner hypothesis, the total damage from a certain number of cycles with different parameters is defined as the sum of damages.

To use the fatigue curve, it is necessary to bring the stresses of asymmetric load cycles that occur with the optimal control law $\sigma_{1opt}$ and suboptimal $\sigma_2$ to equivalent symmetrical ones $\sigma_{E1opt}$ and $\sigma_{E2}$:

$$\sigma_E = \begin{cases} \sigma_{max} + \psi \cdot \sigma_m & \text{if } \sigma_m > 0 \\ \sigma_{max} & \text{if } \sigma_m \le 0 \end{cases}, \qquad (4)$$

where $\sigma_{max}$ – maximum cycle state,

$\sigma_m$ – average cycle state,

$\psi$ – coefficient of sensitivity to cycle asymmetry.

Given that for welded joints $\psi \approx 0 \div 0{,}1$, in the first approximation, we assume that $\sigma_{E1opt} = \sigma_{max1opt}$, and

$\sigma_{E2} = \sigma_{\max 2}$, that is $\psi = 0$ for formula (4). Such an assumption will make it possible to draw general conclusions without considering a specific load case.

The value of the equivalent stress is equal to the maximum stress of the cycle, which occurs under the action of the largest loads in the start-braking modes. Simulation results show that the largest loads under optimal control are less than 5%. Thus, it is acceptable: $\sigma_{E2} / \sigma_{E1opt} = 1{,}05$.

To estimate the increase in durability, we will use the value

$$e_1 = \frac{N_{1opt}}{N_2}. \qquad (5)$$

This variable reflects the ratio of the maximum number of cycles to failure when applying optimal control $N_{1opt}$ up to the maximum number of cycles under ordinary control $N_2$. That is, the entered value $e_1$ shows how many times the service life of the metal structure node will increase. At the same time, we will consider the number of load cycles to failure as the service life of the metal structure a unit $N$, which for cranes is proportional to the number of crane operation cycles.

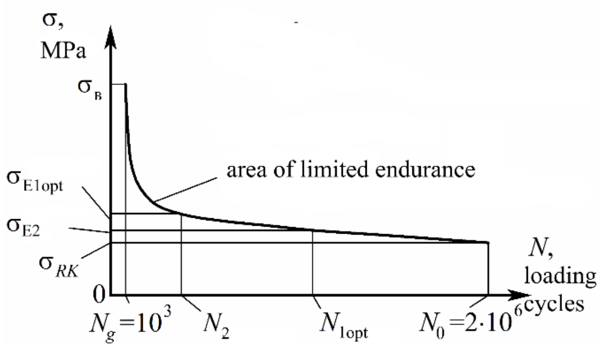



**Fig. 3.** Scheme for evaluating the increase in the number of cycles to failure

In order to estimate how many times the allowable number of load cycles of the metal structure of the crane will increase, consider the groups of welded nodes according to the degree of state concentration 4 – 10. Such nodes are typical for the vast majority of metal structures of tower cranes. We will accept the material Steel20 DSTU 7809:2015 ($\sigma_{ts} < 400$ МПа), which is also quite widespread.

To determine the durability of a node $N$ it is necessary to have a description of the fatigue curve. We will use the method of determining the limit of endurance on an arbitrary basis and the following assumptions:

– the sloping section of the fatigue curve is described by the equation (2)

– all fatigue curves intersect at a point with coordinates [ $N_g = 1000$ циклів; $\sigma_g = \sigma_в$ ] (fig. 3)

– the base of all fatigue curves for welded joints has the value $N_* = 5 \cdot 10^6$ cycles.

Based on these assumptions, the endurance limit on an arbitrary basis is $N$:

$$\sigma_{RKN} = \begin{cases} \sigma_{RK} \sqrt[m]{N_0 / N}, & \text{if } N \le N_* \\ \sigma_{RK} \sqrt[m]{N_0 / N_*} = \sigma_{RK} \sqrt[m]{0{,}4}, & \text{if } N > N_* \end{cases}, \qquad (6)$$

where $N_0 = 2 \cdot 10^6$ cycles,

$m$ – indicator of the degree of the fatigue curve, which is calculated according to the following dependence:

$$m = \frac{3{,}3}{\lg \sigma_в - \lg \sigma_{RK}}. \qquad (7)$$

Thus, the durability of the unit $N$ with an arbitrary level of the maximum cycle stresses $\sigma$ is

$$N = \begin{cases} N_0 \left(\sigma_{RK} / \sigma\right)^m, & \text{if } \sigma_m \ge \sigma \ge \sigma_{RK} \sqrt[m]{0{,}4} \\ \infty, & \text{if } \sigma < \sigma_{RK} \sqrt[m]{0{,}4} \end{cases}. \qquad (8)$$

According to the described rule, the coordinates of the characteristic interpretations of fatigue curves for looking at groups of nodes and the methods of regression analysis know how to describe them.

Figure 4 shows the scheme of energy consumption during the operating cycle of the turning mechanism.

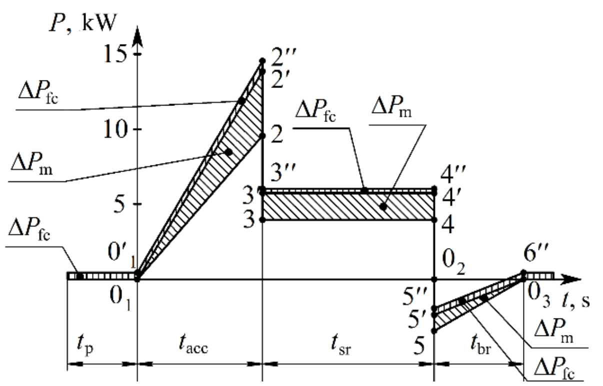



**Fig. 4.** Scheme of saving and recuperating energy during linear acceleration

Significance in Figure 4: $\Delta P_{fc}$ – put pressure on the frequency converter; $\Delta P_m$ – power loss in the motor, taking into account $\eta_m$; $t_p$ – the duration of the pause between the working cycles of the mechanism, $t_{acc}$ – duration of acceleration, $t_{sr}$ – the duration of steady run, $t_{br}$ – duration of braking.

The power consumed by the drive is proportional to the speed on the motor shaft, taking into account the efficiency $\eta_m$ and the frequency converter $\eta_{fc}$, and corresponds to the broken line $0_1$-2"-3"-4"-5"-$0_2$ (fig. 4). Also, energy is consumed when the mechanism is at rest. It is spent on the operation of the frequency converter and is approximately 1.5% of the engine power. These losses can be approximately neglected.

Therefore, the assessment of the energy efficiency of various control laws should be carried out taking into account the entire working cycle of the mechanism with the stages of acceleration, smooth running and braking.

Such an assessment can be made using the equivalent power calculation method $P_e$

$$P_e = \sqrt{\frac{\int_0^t P^2 dt}{t}}, \quad (9)$$

where $P$ - the current instantaneous power consumed by the electric motor; $t$ - time.

Equivalent power reflects the amount of average power consumed by the drive during the work cycle. A comparison of its value for two control methods for the same cycle will show in which case the energy consumption per work cycle is greater.

When applied to electric crane drives, equation (9) can be represented as:

$$P_e = \sqrt{\frac{\Sigma P_{acc}^2 \cdot t_{acc} + \Sigma P_{sr}^2 \cdot t_{sr} + \Sigma P_{br}^2 t_{br}}{t_{acc} + t_{sr} + t_{br}}}, \quad (10)$$

where: $P_{acc}$, $t_{acc}$ – the power consumed during overclocking and the duration of overclocking, respectively;

$P_{sr}$, $t_{sr}$ – the power consumed during a steady stroke and the duration of a steady movement, respectively;

$P_{br}$, $t_{br}$ – the power consumed during braking and the duration of a uniform movement, respectively;

We compare the ratio of energy consumption during normal $P_2$ and optimal $P_{1opt}$ control depending on the duration of the stage of uniform movement. For each case, we will calculate the average power for different values of the time of uniform movement. The amount of consumed energy per cycle is the product of the equivalent power by the duration of the cycle. To compare energy consumption in both drives, we enter the value:

$$e_2 = \frac{P_{E2} \cdot t_c}{P_{E1opt} \cdot t_c} = \frac{P_2}{P_{1opt}}. \quad (11)$$

This value shows the ratio of the amount of consumed energy with optimal control in relation to ordinary.

## 4. Results

In fig. 5 shows graphs of fatigue curves calculated according to the described method. To visualize the results of the calculations, $1\sigma_{E1opt\ max} = 165$ MPa et's plot the graph of the function $e_1 = f\left(\sigma_{1opt}\right)$. It is possible to estimate the increase in the service life of the node only in the zone of limited endurance. Therefore, we will accept the minimum value $\sigma_{E1opt\ min} = 75$ MПa, corresponding to the point of transition to the zone of unlimited endurance for the 4th group. Groups of welded joints 1-3 are not considered in this study, since they are rarely the most dangerous place in the structure and are least susceptible to fatigue phenomena. We take the maximum value – as the largest practically possible. Calculated values $e_1 = f\left(\sigma_{E1opt}\right)$ for different groups of nodes are presented in fig. 6.

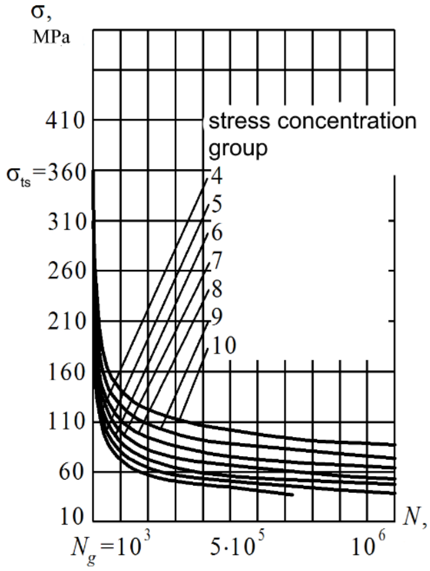



**Fig. 5.** Fatigue curves for groups of welded joints 4-10

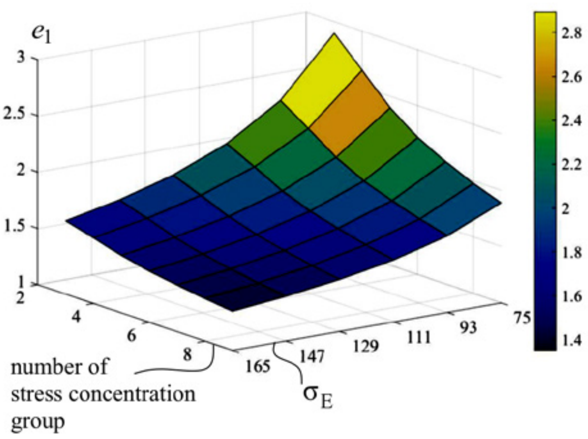



**Fig. 6.** The ratio of the maximum number of cycles with a frequency drive and a motor with a phase rotor $e_1 = N_{1opt}/N_2$ for groups of welded nodes by state concentration 4-10 and equivalent stress from 75 to 165 MPa

From the calculations, it can be seen that the use of optimal control is an effective way to increase the permissible number of load cycles of welded joints of metal structures of tower cranes.

The ratio $e_2$ calculated for the case of transporting a nominal load under different braking modes. The dependence of the ratio change $e_2$ on the time of uniform travel is presented in fig. 7. It can be seen from the graphs that $e_2 \le 1$ at such values $t_{\text{sr}}$, which are rarely found in the practice of operating cranes (50-60 s). For example, for the trolley under study, the duration of a uniform stroke when moving for the length of a characteristic work cycle is $t_{\text{sr}} = 6,5$ s .

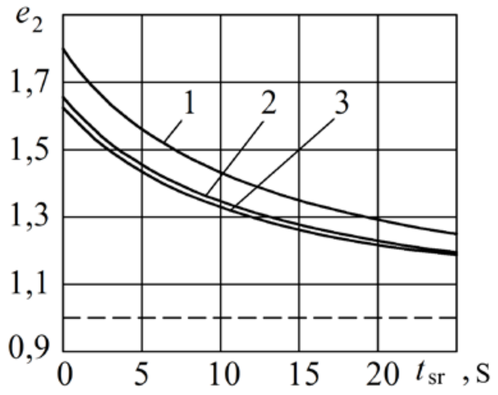



**Fig. 7.** Graphs of the dependence of the ratio of energy consumption $e_2 = P_2 / P_{1\text{opt}}$ on the time of uniform movement $t_{\text{sr}}$ with different methods of braking:
1 – engine braking with optimal control in generator mode;
2 – engine braking with normal mechanical brake control and optimal drive control in generator mode;
3 – braking of both drives with a mechanical brake

## 5. Conclusion

A reduction in the equivalent cycle state by 5% can increase the number of allowable load cycles by at least 1.4 times, for example, for a state concentration group of 10 at $\sigma_{\text{E1opt}} = 165$ MPa for the material Steel 20. Growth ratio $e_1$ and the corresponding number of cycles to failure, increases with a decrease in the equivalent cycle stress and reaches its maximum value in the region of the transition to the zone of unlimited endurance. The larger the number of the group of the welded node in terms of state concentration, the slower this growth occurs, due to the gradual decrease in the limit of unlimited endurance. Further research in this direction can be directed to the consideration of specific structures and load cases.

Studies of the change in the ratio of equivalent energy show that the greatest advantage is achieved when using a frequency-regulated drive in mechanisms that work for a long time in start-braking modes at various values of steady speeds below the nominal and a small time in steady modes at the nominal speed.

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