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Research on the drive of a plant for the preparation and transportation of building mortars

Abstract. Modern mixing plants for the preparation of mortar and concrete mixtures are an important component of construction technologies, which leads to increased requirements for their energy efficiency, operational reliability and stability of operation under conditions of variable loads. At the stage of designing the drive of the mixing plant, a well-founded determination of the power of the electric motor, gear ratios and kinematic parameters of the drive system requires special attention. The article examines the theoretical study of the drive of a compact installation for the preparation and pipeline transportation of construction mortars, the key design feature of which is the use of a single electric motor for driving a forced-action blade mixer and a piston solution pump through a common gear reducer and chain transmission. The kinematic scheme of the drive was analyzed taking into account the moments of inertia of the main elements of the system, dynamic models for estimating torsional oscillations were presented. The nature of variable moments of resistance caused by cyclic load from working bodies is considered. It was established that the dynamic loads that occur during the engagement of gears, starting and braking modes, as well as due to the unevenness of the resistance forces, significantly affect the fatigue strength and durability of the gearbox. The obtained results make it possible to substantiate the rational parameters of the drive, to propose measures to reduce peak loads and to increase the energy efficiency and reliability of small-sized installations for construction sites.

Keywords: mixing and transport installation, drive from one electric motor, gear reducer, torque, dynamic model, mortar mixer, mortar pump.

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Introduction

Modern mixing plants for preparing mortar and concrete mixtures are an important component of construction technologies, which determines increased requirements for their energy efficiency, operational reliability and stability of operation under variable loads. The decisive role in ensuring these characteristics is played by the drive system, the parameters of which directly affect the productivity of the mixing process, the level of energy consumption and the durability of the equipment as a whole [1].

When constructing small facilities, the use of powerful mixing stations is not economically feasible. It is more appropriate to use compact mortar and mixing plants, which ensure the preparation of solutions of various compositions and purposes, directly on the construction site from dry components and water, with subsequent transportation of the finished mixture through pipelines. However, such installations are currently of limited distribution in Ukraine, which is mainly due to their import origin and high cost. In this regard, the development and design of small-sized mortar and mixing plants using one electric motor is relevant, which allows to reduce the labor intensity of technological operations and reduce the cost of mortar preparation and transportation processes.

At the stage of designing the drive of the mixing plant, a well-founded determination of the power of the electric motor, gear ratios and kinematic parameters of the drive system requires special attention.

In this context, theoretical studies of the drive of the mixing plant acquire significant scientific and practical significance. They make it possible to establish the regularities of changes in torque and power consumption during the mixing process, as well as to justify rational design solutions and modes of operation of the drive aimed at increasing the efficiency and reliability of the mixing equipment.

Review of Research Resources and Publications

The most common equipment for preparing mortars is mechanical mixers, where the homogeneity of the mixture is achieved by the mechanical action of a special working body. Both domestic and foreign manufacturers are engaged in the design of such installations [2 – 4]. A significant number of works are devoted to the study of the properties of mortars [5 – 7], and the features of their transportation through various pipelines are described in detail in studies [8, 9].

In the work [10] theoretical studies of the movement of the solution in a vertical mixer whose screw tapes have a variable angle of inclination depending on the height of the placement are carried out. The authors obtained trajectories of solution particles, as well as expressions for their movements, velocities and accelerations, which can be useful for determining equipment parameters. At the same time, energy consumption issues were not considered in this work.

The study of rational parameters of the working bodies of mortar mixers is the subject of the work of Ukrainian and foreign scientists [11, 12], in particular in the context of reducing energy consumption without deteriorating the quality of the preparation of mixtures. As for the means of transportation, the analysis of various types of mortar pumps [13] indicates that piston mortar pumps meet the requirements of modern construction and have significant application prospects.

In general, the available publications focus on certain aspects of the preparation and transportation of construction mortars, however, a comprehensive study of the drives of installations using one electric motor for all working bodies requires further research, which determines the relevance of the proposed topic.

Definition of unsolved aspects of the problem Modern scientific research in the field of machines for the preparation and transportation of construction mortars is quite numerous, but in most of them attention is focused separately either on the drive of the working body of the mixer, or on the drive of the pumping part of the unit. Comprehensive consideration of the drive of the installation, which combines the functions of mixing and subsequent pipeline transportation of the finished solution, is practically absent. It became necessary to develop such a combined installation and conduct a detailed study of its power drive.

One of the key criteria for reliability and efficiency of the mixing and transport installation is the stable operation of the gearbox drive. The durability and contact strength of these gears are significantly affected by dynamic loads that occur during transient modes — , in particular, when engaging pairs of teeth during rotation, when starting the gearbox under full load, as well as when braking or stopping the system.

The study of such dynamic processes allows not only to justify the choice of gear parameters and gearbox design, but also to propose measures to reduce peak loads – for example, by optimizing the profile of the teeth, using damping elements or improving algorithms for controlling the starting and braking modes of the electric drive.

Problem statement

The main goal of this theoretical study is the analysis of the drive of the developed installation for the preparation and transportation of construction mortars, in which all working bodies – blade mixer of forced action and piston solution pump – are driven by a single electric motor through a common gearbox.

Basic material and results

Screw and blade mixers are the most common for preparing and transporting mortars in practice. At the same time, difficulties arise on small-sized construction sites related to the transportation of the finished mortar mixture directly to the working area. In order to reduce the overall dimensions of the equipment, its unification, as well as to increase the intensity and quality of the mixing process, this work focuses on the study of a universal installation, the design of which combines the functions of preparation, transportation and feeding of the solution mixture.

The designed installation consists of an electric motor 1, which is connected to a coupling 2 connecting the electric motor shaft to the input shaft of the reducer 3, a rigid coupling 4 connecting the output shaft of the reducer to the shaft of the solution pump 6 and a lever 5 for opening the shaft of the reducer to the shaft of the solution pump. Attached to the other end of the reduction gear is a coupling 7 connecting the output shaft of the reduction gear to the shaft of the chain gear 8 and the mixing hopper 9.

To determine the dynamic loads acting in the gearbox drive of the mixing plant during the operating mode of the technological process, consider its kinematic scheme, which is presented in Fig. 1.

The moment of the resistance forces M_s can be represented as the sum of the constant and variable moments of the resistance forces (Fig. 2).

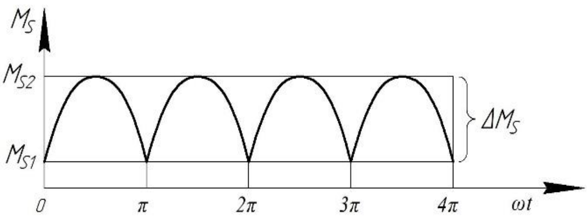

Figure 2 − Curve of change of moments of resistance forces on the drive shaft of the working body

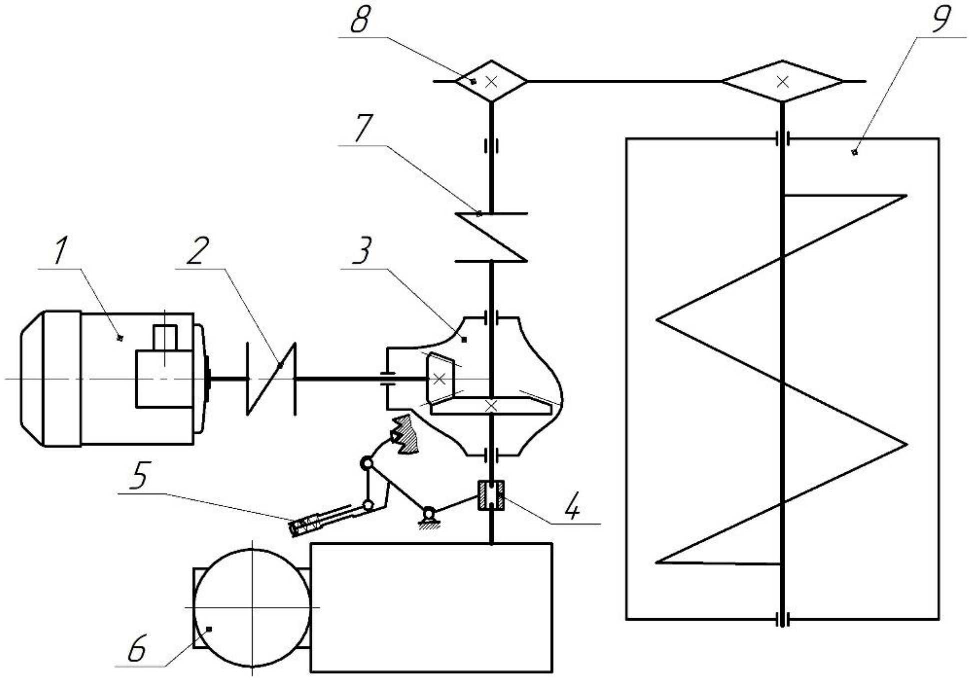

Figure 1 − Kinematic drive diagram
1 − electric motor; 2 − clutch connecting the electric motor shaft to the gearbox input shaft; 3 – gearbox; 4 − rigid clutch connecting the gearbox output shaft to the pump shaft; 5 – gear shaft opening lever with the solution pump shaft; 6 – solution pump; 7 − clutch connecting the gearbox output shaft to the chain transmission shaft; 8 – chain transmission; 9 – mixing hopper.

The moment of forces at the support can be represented in terms of the following function:

$$M_S = M_{S1} + \Delta M_S \left| \sin \omega t \right| . \qquad (1)$$

Let's decompose the variable moment of the resistance forces into a Fourier series and do some simplifications and we get:

$$M_{S1} + \Delta M_S \left| \sin \omega t \right| = \frac{2\Delta M_S}{\pi} - $$
$$- \frac{4\Delta M_S}{\pi} \left[\frac{\cos 2\omega t}{1 \cdot 3} + \frac{\cos 4\omega t}{3 \cdot 5} + ... \right] = \qquad (2)$$
$$\frac{2\Delta M_S}{\pi} - - \frac{4\Delta M_S}{\pi} \sum_{n=1}^{\infty} \frac{\cos(2n\omega t)}{(2n-1)(2n+1)}$$

Substituting equation (2) into equation (1) and performing the transformation, we will find the common moment of the resistance forces:

$$M_S = M_{S1} + \frac{2\Delta M_S}{\pi} - \frac{4\Delta M_S}{3\pi} \cos 2\omega t \qquad (3)$$

In order to simplify the analysis of the obtained complex dynamic system, it is advisable to present it in the form of two simpler equivalent dynamic systems, adjusted according to the input and output shafts of the gear reducer.

Fig. 3 illustrates a dynamic model represented by an equivalent two-mass system. The subsequent analysis is restricted to the steady-state operating regime, in which the dynamic behavior of the system can be sufficiently described by the corresponding system of equations of motion.

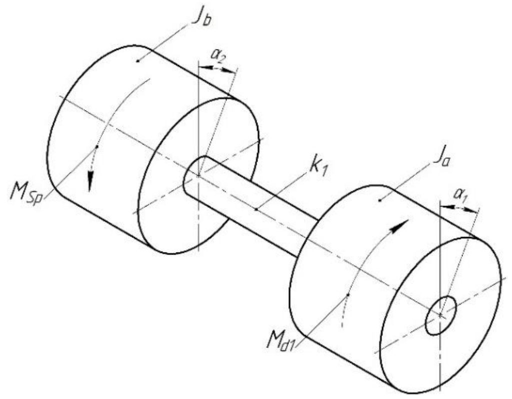

Figure 3 − Two-mass dynamic model formed as a result of bringing the inertial masses of the entire system to the input shaft of the gearbox

α_1 − the angle of rotation of the mass with the induced moment of inertia; α_2 − the angle of rotation of the mass with the induced moment of inertia J_b ; k_1 − the coefficient of torsional stiffness on the input shaft of the gearbox; M_{Sp} − given moment of resistance forces; M_{d1} − moment of driving forces.

$$J_a \frac{d^2\alpha_1}{dt^2} + k_1(\alpha_1 - \alpha_2) = M_{d1}, \qquad (4)$$

$$J_b \frac{d^2\alpha_2}{dt^2} - k_1(\alpha_1 - \alpha_2) = -M_{sp}, \qquad (5)$$

where J_a − is the moment of mass inertia to the end of the input shaft of the gearbox;

$$J_a = J_1 + J_2 + J_3, \qquad (6)$$

where J_1 − the moment of inertia of the rotor of the electric motor;

J_2 − moment of inertia of the coupling;

J_3 − moment of inertia of the input shaft of the gearbox;

where J_b − the moment of mass inertia is applied to the gear on the input shaft of the gearbox;

$$J_b = J_4 + (J_5 + J_6 + J_7 + J_8 + J_9 + J_{10} + J_{11})\frac{\omega_2^2}{\omega_1^2}, \qquad (7)$$

where J_4 and J_5 − are moments of inertia of the drive and driven gear wheels of the gearbox, respectively;

J_6 − moment of inertia of the output shaft of the gearbox;

J_7 − moment of inertia of the rigid coupling;

J_8 – the moment of inertia of the coupling connecting the output shaft of the reduction gear to the shaft of the chain drive;

J_9 and J_{10} – moments of inertia of the leading and leading chain transmission sprocket, respectively;

J_{11} − is the given moment of inertia of the blade shaft of the mixing hopper.

Let's bring out the given moment of resistance forces,

$$M_{Sp} = \frac{[2M_{S2} + (\pi - 2)M_{S1}]\omega_2}{\pi\omega_1} - \frac{4\Delta M_S \omega_2}{3\pi\omega_1}\cos 2\omega t. \qquad (8)$$

With steady motion, the moment of the driving forces M_{d1} is equal to the reduced moment of the resistance forces M_{Sp},

$$M_{d1} = M_{Sp}. \qquad (9)$$

Using expressions (8) and (9), we transform equations (4) and (5) describing the torsional oscillations of the considered dynamic system to the following form:

$$J_a\frac{d^2\alpha_1}{dt^2} + k_1(\alpha_1 - \alpha_2) = \frac{[2M_{S2} + (\pi - 2)M_{S1}]\omega_2}{\pi\omega_1} - \frac{4\Delta M_S \omega_2}{3\pi\omega_1}\cos 2\omega t; \qquad (10)$$

$$J_b\frac{d^2\alpha_2}{dt^2} - k_1(\alpha_1 - \alpha_2) = -\frac{[2M_{S2} + (\pi - 2)M_{S1}]\omega_2}{\pi\omega_1} - \frac{4\Delta M_S \omega_2}{3\pi\omega_1}\cos 2\omega t. \qquad (11)$$

Fig. 4 shows a dynamic model in the form of a two-mass system, formed as a result of bringing the inertial masses of the entire system to the output shaft of the gearbox.

$$J_c\frac{d^2\alpha_3}{dt^2} + k_2(\alpha_3 - \alpha_4) = M_{d2}, \qquad (12)$$

$$J_d\frac{d^2\alpha_4}{dt^2} - k_2(\alpha_3 - \alpha_4) = -M_S, \qquad (13)$$

where − J_C is the moment of mass inertia to the driven gear wheel on the output shaft of the gearbox,

$$J_C = (J_1 + J_2 + J_3 + J_4)\frac{\omega_1^2}{\omega_2^2} + J_5, \qquad (14)$$

J_d − the induced moment of mass inertia to the end of the output shaft of the gearbox,

$$J_d = J_6 + J_7 + J_8 + J_9 + J_{10} + J_{11}, \qquad (15)$$

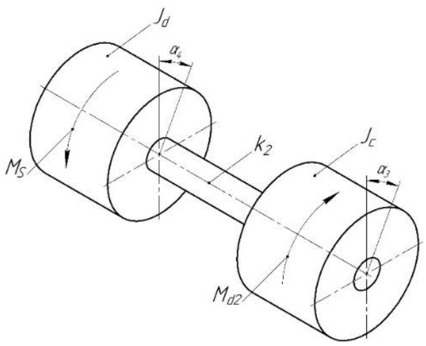

Figure 4 − Two-mass dynamic model formed as a result of bringing the inertial masses of the entire system to the output shaft of the gearbox

α_3 − angle of rotation of the mass with the induced moment of inertia J_C; α_4 − angle of rotation of the mass with the induced moment of inertia J_d; k_2 − stiffness coefficient on the output shaft of the gearbox; M_S − given moment of resistance forces; M_{d2} − moment of driving forces.

With steady motion, the moment of the driving forces is equal to the reduced moment of the resistance forces,

$$M_{d2} = M_S. \qquad (16)$$

Using expressions (3) and (16), we transform equations (12) and (13) describing the torsional oscillations of a given dynamic system (Fig. 4) to the following form:

$$J_c\frac{d^2\alpha_3}{dt^2} + k_2(\alpha_3 - \alpha_4) = \frac{2M_{S2} + (\pi - 2)M_{S1}}{\pi} - \frac{4\Delta M_S}{3\pi}\cos 2\omega t; \qquad (17)$$

$$J_d \frac{d^2\alpha_4}{dt^2} - k_2(\alpha_3 - \alpha_4) = \\ = -\frac{2M_{S2} + (\pi - 2)M_{S1}}{\pi} + \\ + \frac{4\Delta M_S}{3\pi} cos 2\omega t. \quad (18)$$

The analysis of the obtained dependencies shows that the level of dynamic loads that occur in the gearbox drive of the technological machine is significantly determined by the nature of the variable moments of the driving forces and resistance forces, the values of the moments of inertia of the moving masses, the angular speeds of rotation of the gearbox shafts, as well as the ratio of the angular frequencies of forced oscillations and natural frequencies of the dynamic system.

Conclusions

In the work, a theoretical study of dynamic processes in the drive of the mixing plant was carried out, taking into account the variable nature of the moments of the driving forces and resistance forces, as well as the inertial characteristics of the elements of the mechanical system. Equivalent dynamic models were formed on the basis of bringing inertial masses to the input and output shafts of the gearbox, which made it possible to adequately describe the interaction of the drive elements in established operating modes.

The developed and analyzed dynamic model of the drive allows taking into account the peculiarities of the interaction of the working body with the soluble mixture and the variable nature of the loads characteristic of the mixing process. The use of equivalent two-axis models ensured the acquisition of analytical dependencies for the estimation of torque and dynamic loads in gear drive elements.

It was established that the level of dynamic loads is significantly determined by the ratio of angular velocities and rotation frequencies of the shafts, moments of inertia of moving masses, as well as the spectral composition of variable moments of driving forces and resistance forces, which must be taken into account during the design and selection of rational driving parameters of mixing plants.

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Дослідження приводу установки для приготування та транспортування будівельних розчинів

Аннотація. Сучасні змішувальні установки для приготування розчинових і бетонних сумішей є важливою складовою будівельних технологій, що зумовлює підвищені вимоги до їх енергоефективності, експлуатаційної надійності та стабільності роботи в умовах змінних навантажень. На етапі проектування приводу змішувальної установки особливої уваги потребує обґрунтоване визначення потужності електродвигуна, передавальних чисел редуктора та кінематичних параметрів приводної системи. У статті розглянуто теоретичне дослідження приводу компактної установки для приготування та трубопровідного транспортування будівельних розчинів, ключовою конструктивною особливістю якої є використання єдиного електродвигуна для приведення в рух лопатевого змішувача примусової дії та поршневого розчинонасоса через спільний зубчастий редуктор і ланцюгову передачу. Проаналізовано кінематичну схему приводу з урахуванням моментів інерції основних елементів системи, представлено динамічні моделі для оцінки крутильних коливань. Розглянуто характер змінних моментів опору, зумовлених циклічним навантаженням від робочих органів. Встановлено, що динамічні навантаження, які виникають під час зачеплення зубчастих передач, пускових і гальмівних режимів, а також від нерівномірності сил опору, суттєво впливають на втомну міцність і довговічність редуктора. Отримані результати дозволяють обґрунтувати раціональні параметри приводу, запропонувати заходи щодо зниження пікових навантажень і підвищити енергоефективність та надійність малогабаритних установок для будівельних майданчиків.

Ключові слова: змішувально-транспортувальна установка, привід від одного електродвигуна, зубчастий редуктор, крутний момент, динамічна модель, розчинозмішувач, розчинонасос

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