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Mathematical Model of Energy Processes in Industrial Electric Screwdriver

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Abstract. An electric screwdriver has been designed to work in an automatic cycle on assembly lines. A characteristic feature of the screwdriver is the use of a low power motor in comparison with commonly used devices. The work for tightening a threaded joint takes place at the expense of kinetic energy in components of a drive system and the working screwdriver. It has been proved that an ability of the screwdriver to perform tightening works is determined by the sum of a mass moment of inertia in the system of working components and a mass moment of inertia in a motor rotor, reduced to the axis of the screwdriver bit. The process of tightening is characterized by a number of screwing pulses, screwing torque values at the end of each pulse and time of screwing process. Limitation for the value of transmitted tightening torque takes place through an applied overload coupling. The construction work for the screwdriver has been preceded by development of a mathematical model for screwing process and balance of torque moments acting during each phase of the work. Such approach has made it possible to calculate a value of kinetic energy in components of a drive and an operating system converted into tightening work. Kinematic conditions which are to be carried out by the screwdriver at the end of a single tightening pulse have been determined in order to accumulate kinetic energy and start another tightening pulse.

Keywords: screwdriver, tightening process, tightening torque, mass moment of inertia

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Математическая модель энергетических процессов в промышленной электрической отвертке

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Реферат. Электрическая отвертка была разработана для автоматической работы на сборочных линиях. Характерной особенностью отвертки является использование двигателя малой мощности по сравнению с обычно применяемыми устройствами. Работа по натяжению резьбового соединения происходит за счет кинетической энергии компонентов системы привода и рабочей отвертки. Доказано, что способность отвертки выполнять затяжные работы определяется суммой массового момента инерции системы рабочих компонентов и массового момента инерции ротора двигателя, приведенного к оси наконечника отвертки. Процесс затягивания характеризуется количеством импульсов завинчивания, значениями момента завинчивания в конце каждого импульса и временем процесса завинчивания. Ограничение величины передаваемого момента затяжки происходит через применяемую перегрузочную муфту. Строительным работам по отвертке предшествовала разработка математической модели процесса завинчивания и балансировки крутящих моментов, действующих на каждом этапе работы. Это позволило рассчитать значение кинетической энергии компонентов привода и операционной системы в переводе на затяжные работы. Кинематические условия, которые должны быть выполнены отверткой в конце одного импульса затяжки, были определены для накопления кинетической энергии и запуска другого импульса затяжки.

Ключевые слова: отвертка, процесс затяжки, момент затяжки, момент инерции массы

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Introduction

The power screwdriver is an assembly tool used for driving in threaded fasteners. In automated assembly processes, special importance is given to the problem of microprocessor control of the device's operation and the possibility of smooth regulation of the tightening torque while maintaining its value within the specified tolerance range [1–4]. These requirements mean that screwdrivers must be designed in a way that will allow them to be used in integrated control and operation-parameter-monitoring systems. In [5], a control method based on artificial networks has been presented. The authors in article [6, 7] presents the results of research on the original screwdriver for industrial applications in the event of the need to obtain the value of the tightening torque in narrower tolerances. The operating of the screwdriver is supervised by the control program, which has implemented algorithms for selecting the operating mode. The program also allows archiving operating parameters and visualization of on-line characteristics of screwing.

A new criterion that has recently been proposed for electric screwdrivers requires that the tools be optimized toward reduced energy consumption by the drive motor. Studies [8–10] report the results of analyses of the impact of the electric motor's load mode and power consumption on the functional

parameters of electric screwdrivers. These tests allowed the authors to analyze changes in the design of working components necessitated by an increase in the energy efficiency coefficient of the electric motor. Paper [11] reports the results of experimental studies aimed at calculating differences in electric energy consumption of a screwdriver with a set tightening torque. Two variables of the screwing process were manipulated: the angular velocity of the screwdriver bit and the number of impacts. Tightening torque and energy consumption were studied as a function of angular velocity of the screwdriver bit and the operating time of the impact mechanism. It was concluded, with regard to the possibilities of controlling the operation of the impact screwdriver, that it was impossible to obtain a desired tightening torque by setting the working time of the impact mechanism and the initial angular velocity (turning rate) at which a threaded connector was driven in.

Another method of reducing the energy consumption of an electric screwdriver is to analyze changes in the kinetic energy of the components of the power transmission (drive) mechanism and the working components during the screwdriving process and then use these data to select an appropriate motor. The original screwdriver described in this present article is a tool [4] which uses a shunt-wound DC motor with a voltage of 30 V and a torque-limiting overload clutch (fig. 1).

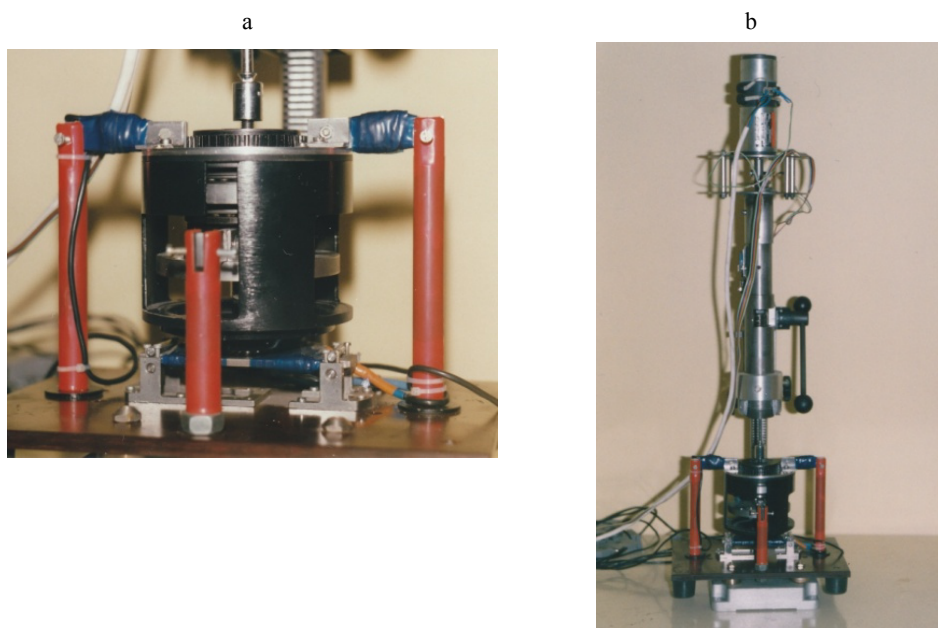


Fig. 1. Testing bench for screwing parameters: a – tool to measure course of force and torques in the process of screwing [12]; b – screwdriver on testing bench

The operating principle of the screwdriver

In designing the screwdriver, we performed design synthesis to make sure that the structure and functional features of the drive system and the working components system would allow to exploit the kinetic energy of the device for performing external work.

The kinematic forces of the individual components of the screwdriver are generated under the influence of the operation of the motor coupled to a planetary gear transmission (fig. 2). They are a function of the motor's angular velocity ω_{zs} , the transmission ratio i_p , and the torque setting of the overload clutch. The rotational speed of the motor changes during a work cycle as a result of increasing tightening torque of the threaded fastener and a lack of a motor power reserve. The kinematic system is switched on and off depending on the relationship between the value of the tightening torque of the threaded fastener and the torque set on the overload clutch. The system is disengaged when the tightening torque of the threaded connector exceeds the torque setting in the clutch. The sum of the values of the mass moment of inertia of the working components system and the mass moment of inertia of the motor's rotor, reduced to the axis of the screwdriver bit I_z , determines the ability of the screwdriving head to drive in a threaded fastener of a given diameter. The movement of the screwdriver bit driving in a threaded connector is related to the movement of the spindle charac-

terized by diameter D , thread pitch H , and length of the threaded part L .

The tightening operation takes place at the expense of the kinetic energy of the components of the drive and the working components. The course of the screwdriving process shows that the demand for kinetic energy for the tightening operation is short-lasting and the maximum kinetic energy necessary for fastening occurs only in the phase of tightening the threaded fastener. The basic conclusion that follows from the assumption that the cycles of installing in threaded fasteners are executed automatically is that none of the components of the drive system and working components can reach a zero angular velocity. The above also implies that these systems (power transmission and working components) can only transmit a limited portion of the kinetic energy necessary for tightening the threaded connection. For the screwdriver to be able to operate in a continuous fashion, after performing the external work, the whole system should accumulate energy to reach the same angular velocity before the next cycle. This requires a comprehensive analysis of functional relationships between the drive system and the working components system, which will allow proper selection of a motor, a gear transmission ratio, and the weight of the working components which perform rotational motion.

The parameters and variables which describe the operation of the screwdriver reflect the course of the screwdriving process shown in fig. 3, 4.

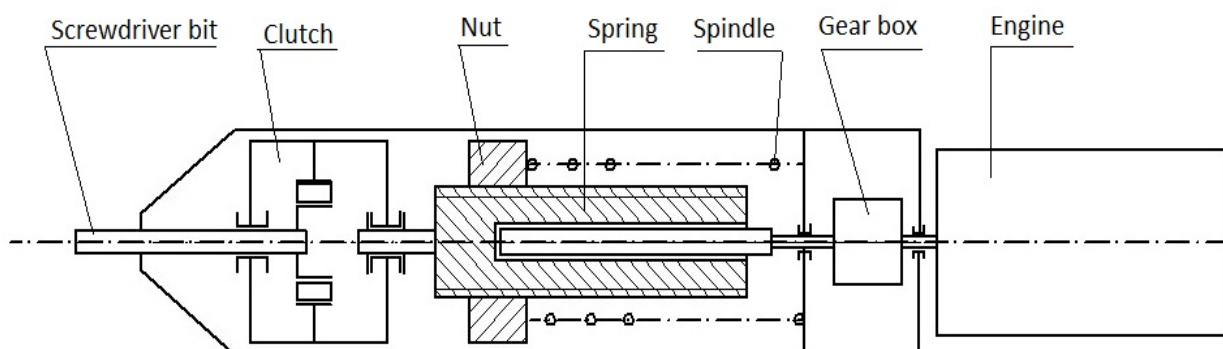


Fig. 2. Diagram of screwdriver construction

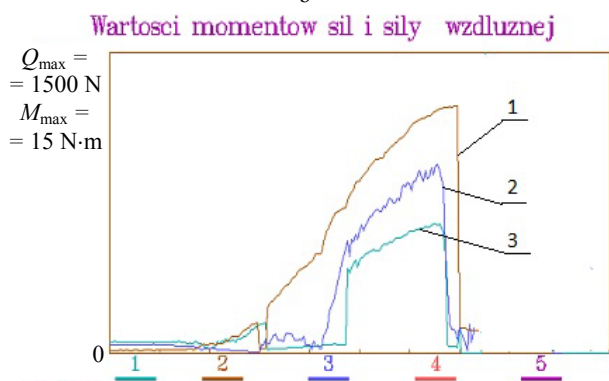
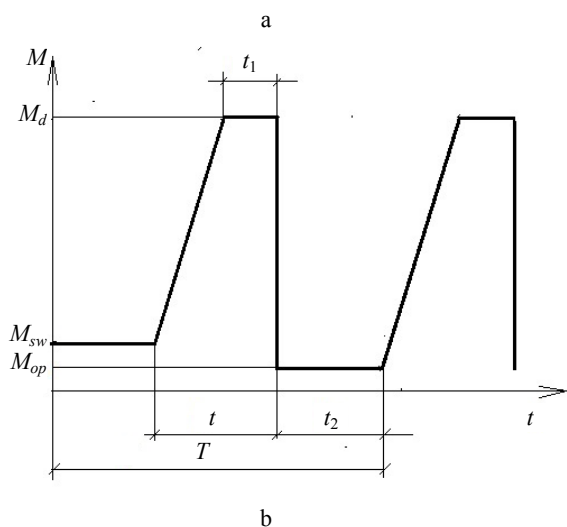


Fig. 3. Screwing process realized by screwdriver: a – theoretical course of screwing process: t – tightening time; t_1 – time of clutch-off; t_2 – time of energy accumulation; T – period of one screwing impulse; M_{sw} – frictional torque during normal screw-driving phase; M_{op} – sum of frictional torques in drive and working component systems; M_d – tightening torque; b – curve illustrating screwing for M6 B 6.8 bolt by means of screwdriver [4], in-screwing parameter measurement on device [12]; 1 – moment on thread; 2 – axial force in bolt; 3 – friction moment under bolt head

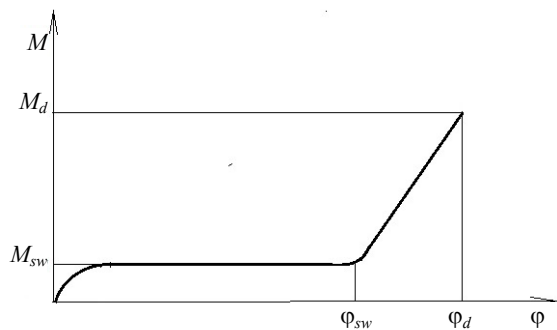


Fig. 4. Theoretical course of screwing process: M_{sw} – frictional torque during normal screw-driving phase; M_d – tightening torque; φ_{sw} – rotation angle of threaded joint at the end of normal screw-driving phase; φ_d – rotation angle of threaded joint at the end of tightening phase

Equation of motion for a system in the tightening and clutch-off phases

During the tightening phase, the tightening torque transmitted by the screwdriver bit to the threaded fastener increases to M_d . Once this value has been reached, the overload clutch disconnects. Kinetic energy of the drive and operating system of the screwdriver

$$E_k = \frac{1}{2} I_s \dot{\varphi}_s^2 + \frac{1}{2} I_{UR} \left(\frac{\dot{\varphi}_s}{i_p} \right)^2 + \frac{1}{2} m_n \left(\frac{\dot{\varphi}_s H}{2\pi i_p} \right)^2 + \frac{1}{2} m_1 \left(\frac{\dot{\varphi}_s H}{2\pi i_p} \right)^2 \quad (1)$$

Energy dissipation function

$$D = M_s (1 - \eta_s) \dot{\varphi}_s + M_s (1 - \eta) + F_{op} \left(\frac{\dot{\varphi}_s H}{2\pi i_p} \right) \quad (2)$$

Potential energy

$$E_p = \frac{1}{2} k \frac{\varphi_s H}{2\pi i_p} + m_n g \frac{H \varphi_s}{2\pi i_p} + m_1 g \frac{H \varphi_s}{2\pi i_p} \quad (3)$$

Components of the Lagrange equation of the second degree

$$\frac{\partial E_k}{\partial \dot{\varphi}_s} = I_s \dot{\varphi}_s + I_{UR} \frac{\dot{\varphi}_s}{i_p} + m_n \left(\frac{H}{2\pi i_p} \right)^2 \dot{\varphi}_s + m_1 \left(\frac{H}{2\pi i_p} \right)^2 \dot{\varphi}_s \quad (4)$$

$$\frac{d}{dt} \left(\frac{\partial E_k}{\partial \dot{\varphi}_s} \right) = \left(I_s + I_{UR} \frac{1}{i_p^2} \right) \ddot{\varphi}_s + m_n \left(\frac{H}{2\pi i_p} \right)^2 \ddot{\varphi}_s + m_1 \left(\frac{H}{2\pi i_p} \right)^2 \ddot{\varphi}_s \quad (5)$$

By deleting the first part

$$\left(I_s + I_{UR} \frac{1}{i_p^2} \right) = \left(I_s i_p^2 + I_{UR} \right) \frac{1}{i_p^2} = I_z \frac{1}{i_p^2} \quad (6)$$

After substitution, we get

$$\frac{d}{dt} \left(\frac{\partial E_k}{\partial \dot{\varphi}_s} \right) = I_z \frac{1}{i_p^2} \ddot{\varphi}_s + (m_n + m_1) \left(\frac{H}{2\pi i_p} \right)^2 \ddot{\varphi}_s \quad (7)$$

Energy dissipation function

$$\frac{\partial D}{\partial \dot{\varphi}_s} = M_s (1 - \eta_s) + M_s \frac{1}{i_p} (1 - \eta) + F_{op} \left(\frac{H}{2\pi i_p} \right) \quad (8)$$

Equation of motion for a system in the tightening and clutch-off (disengagement) phases

$$I_z \frac{1}{i_p^2} \ddot{\varphi}_s + (m_n + m_1) \left(\frac{H}{2\pi i_p} \right)^2 \ddot{\varphi}_s + \left(\frac{1}{2}k + m_n + m_1 \right) \frac{H}{2\pi i_p} + M_s \left[(1 - \eta_s) + \frac{1 - \eta}{i_p} \right] + F_{op} \frac{H}{2\pi i_p} = M_d. \quad (9)$$

The value of the kinetic energy of translational motion and the value of potential energy are negligibly small in relation to the kinetic energy of rotary motion. It is then true that

$$I_z \frac{1}{i_p^2} \dot{\varphi}_s + M_s \left[(1 - \eta_s) + \frac{1 - \eta}{i_p} \right] + F_{op} \frac{H}{2\pi i_p} = M_d, \quad (10)$$

where I_s – mass moment of inertia of the engine's rotor; I_z – mass moment of inertia of the engine's rotor, reduced to the axis of the screwdriver bit; φ_s – rotation angle of the engine rotor; I_{UR} – mass moment of inertia of work system elements; i_p – transmission ratio; m_n – weight of the nut; m_1 – mass of components of the working system without a nut; F_{op} – strength of friction resistance in sliding combination of the elements of the screwdriver; k – spring stiffness; η – mechanical efficiency of the working system; η_s – efficiency of the engine.

Energy losses in the drive system of the screwdriver described by the energy dissipation function D mean that at time t_0 the system does not reach velocity $\dot{\varphi}_s$ but a certain angular velocity $\dot{\varphi}_1$. It is then true that

$$I_z \frac{1}{i_p^2} \dot{\varphi}_1 = M_d, \quad \dot{\varphi}_p \leq \dot{\varphi}_1 \leq \dot{\varphi}_k. \quad (11)$$

According to this assumption, the work required for tightening and disconnecting the clutch is done at the expense of the kinetic energy of the system. Quantitatively, total work can be expressed as

$$L_c = \int_0^{\varphi_d} K \varphi d \varphi, \quad (12)$$

where φ_d – tightening angle of the threaded fastener; K – threaded fastener constant.

Tightening torque M_d , tightening angle φ_d , and threaded fastener constant K as a function of diameter d of the screw being inserted [7]:

$$M_d = 0,0699 d^{2,6421}; \quad r^2 = 0,9978; \\ K = 2,3262 d^{0,7527}; \quad r^2 = 0,9674; \\ \varphi_d = 0,03 d^{1,8897}; \quad r^2 = 0,993.$$

If at the starting time of the screwdriving process t_0 , the system has velocity ω_p , and at the moment of disengagement t , it has angular velocity ω_k , the following condition determining the kinetic energy of a system being in rotational motion during the tightening and disengagement phase is satisfied

$$0,5 I_z \dot{\varphi}_p^2 - 0,5 I_z \dot{\varphi}_k^2 = 0,5 K (\varphi_d)^2. \quad (13)$$

From this equation, one can calculate the angular velocity of the working components system after the tightening operation

$$\dot{\varphi}_k = \sqrt{\dot{\varphi}_p^2 - \frac{K}{I_z} (\varphi_d)^2}. \quad (14)$$

Equation of motion for a system in the phase of energy accumulation

The energy accumulation phase starts when the overload clutch is disconnected. The direction of rotation of the drive motor changes, the components of the drive system and the working components are unloaded and their angular velocity increases. The equation of motion of the drive system and the working components system of the screwdriver during the energy accumulation phase has the following form

$$I_s \ddot{\varphi}_s + I_{UR1} \frac{1}{i_p^2} \ddot{\varphi}_s + (m_n + m_2) \left(\frac{H}{2\pi i_p} \right)^2 \ddot{\varphi}_s + \left(\frac{1}{2}k + m_n g + m_2 g \right) \frac{H}{2\pi i_p} + M_s \left[(1 - \eta_s) + \frac{1 - \eta}{i_p} \right] \dot{\varphi}_s + F_{op} \frac{H}{2\pi i_p} \dot{\varphi}_s = -M_{opAE}. \quad (15)$$

The sum of the kinetic energy of translational motion and potential energy is negligibly small in relation to the kinetic energy of rotary motion. The following equation is then obtained

$$I_s \ddot{\varphi}_s + I_{UR1} \frac{1}{i_p^2} \ddot{\varphi}_s + M_s (1 - \eta_s) \dot{\varphi}_s \frac{1 - \eta}{i_p} + F_{op} \varphi_s \frac{H}{2\pi i_p} = -M_{opAE}. \quad (16)$$

The initial conditions for the system are defined by relationship (14). The system is loaded

with frictional torque M_{opAE} . This torque is equal to the frictional torque occurring during the normal screwdriving phase, which means that the frictional torque in the energy accumulation phase can be expressed as

$$M_{opAE} = M_{sw} + M_{op},$$

where M_{sw} – frictional torque during the normal screwdriving phase, $M_{sw} = 0.005M_d$; M_{op} – sum of frictional torques in the drive and working components systems; m_2 – mass of workpiece parts except for the nut, passive part of the coupling and screwdriver bit; I_{UR1} – mass moment of inertia of workpiece parts except for the nut, passive part of the coupling and screwdriver bit.

Energy losses described by the energy dissipation function mean that in the real system the motor does not reach velocity $\dot{\varphi}_s$ but a certain velocity $\dot{\varphi}_1$

$$I_s \ddot{\varphi}_1 + I_{UR1} \ddot{\varphi}_1 \frac{1}{i_p^2} = -M_{opAE}, \quad \ddot{\varphi}_k \leq \ddot{\varphi}_1 \leq \ddot{\varphi}_s. \quad (17)$$

The equation of motion in the energy accumulation phase is used to calculate the angular velocity of the end link $\dot{\varphi}_z$. Because

$$I_s + I_{UR} \frac{1}{i_p^2} = I_{SR}, \quad (18)$$

where I_{SR} – mass moment of inertia of the components of the drive system and the working components reduced to the rotor of the drive motor; M_{ops} – sum of frictional torques in the drive and working components systems reduced to the motor shaft; M_{sws} – frictional torque during the normal screwdriving phase reduced to the motor shaft

$$M_{sws} = \frac{0.005 M_d}{i_p}.$$

The frictional torque reduced the motor shaft has the following value

$$M_{opAE} = M_{sws} + M_{ops}. \quad (19)$$

After substitution, we get:

$$M_{sws} + M_{ops} = \frac{M_{zs}}{\omega_{zs}} (\omega_{zs} - \omega_{sws}); \quad (20)$$

$$I_{SR} \ddot{\varphi} = -M_{opAE}; \quad (21)$$

$$I_{SR} \frac{d\varphi}{dt} = \frac{M_{zs}}{\omega_{zs}} (\omega_{sws} - \omega). \quad (22)$$

The solution to the equation is:

$$\frac{d\omega}{dt} = k_1 (\omega_{sws} - \omega); \quad (23)$$

$$\omega = c e^{-k_1 t} + \omega_{sws}, \quad (24)$$

where ω_{zs} – rated speed of the engine; ω_{sws} – angular velocity of engine loaded M_{sws} torque; k_1 – constant

$$k_1 = \frac{M_{zs}}{\omega_{zs} I_{SR}}.$$

For $t = t_1 = 0$, $\omega = \omega_k i_p$, hence constant c

$$c = -\omega_{sws} + \omega_k i_p. \quad (25)$$

By integrating the equation we get

$$\varphi = \omega_{sws} t + \frac{1}{k_1} (\omega_{sws} - \omega_k i_p) e^{-k_1 t} + c_1. \quad (26)$$

For $t = 0$, $\varphi = 0$

$$c_1 = -\frac{1}{k_1} (\omega_{sws} - \omega_k i_p). \quad (27)$$

Finally, we get

$$\varphi = \omega_{sws} t + \frac{1}{k_1} (\omega_{sws} - \omega_k i_p) (e^{-k_1 t} - 1). \quad (28)$$

Angular speed of the screwdriver bit at the end of the energy accumulation process

$$\omega_z = \frac{1}{i_p} \omega_{sws} - \frac{1}{i_p} (\omega_{sws} - \omega_k i_p) e^{-k_1 t}. \quad (29)$$

During the energy accumulation phase, the system must return to its initial state, i. e. reach velocity $\omega_z = \omega_p$, or, in other words, to reach an angular velocity that will allow the screwdriver to operate again

$$\omega_p i_p = \omega_{sws} - (\omega_{sws} - \omega_k i_p) e^{-k_1 t}. \quad (30)$$

On the basis of the analyses of energy changes during the phases of the screwdriver's operation, design synthesis was performed to select the parameters that would allow the screwdriver to be used for driving in M6–M12 screws (tab. 1, 2). The exact range of screw diameters was determined by the maximum torque exerted by the screwdriver $M_d = 50.4 \text{ N}\cdot\text{m}$.

The tightening cycle may be implemented according to the two methods. The method of single pulse provides a short time of technological process while maintaining the condition that the screwing torque is equal to tightening torque. During the method of successive individual pulses the bolt is tightened with $i = 1..n$ pulses, of which the n th allows the development of the tightening torque. This method in practice is used when tightening the bolts with the dimensions of M8–M12.

Table 1

Technical data of screwdriver

Drive engine			Mass moment of inertia of		
Description	Symbol	Value	Description	Symbol	Value
Power	N	30 W	Engine rotor reduced to axis of screwdriver bit	I_z	0.0076 kg·m ²
Rated turning moment	M_{zs}	0.1 N·m	Engine rotor	I_s	0.00037 kg·m ²
Rated angular speed	ω_{zs}	314 rad/s	Work system elements	I_{UR}	0.000107 kg·m ²
Rated voltage	U	30 V	Components of drive system and working components reduced to rotor of drive motor	I_{SR}	0.000375 kg·m ²

Table 2

Values of screwing parameters obtained when screwing bolt M6

Description	Symbol	Value
Tightening torque	M_d	8.6 N·m
Frictional torque during the normal screwdriving phase	M_{sw}	0.043 N·m
Angular speed of the beginning of the screwing process	ω_p	62.72 rad/s
Angular speed of the end of the screwing process	ω_k	51 rad/s
Tightening angle	φ_d	0.86 rad

CONCLUSION

Electric screwdrivers are designed to meet two main criteria: they have to deliver a high tightening torque at a low power of the screwdriver motor and consume a minimum of energy. The available literature provides analyses related to the reduction of power consumption by the screwdriver's motor; at the same time, many articles pay attention to difficulties in controlling the screwdriver's operation. This article presents a different approach to the problem of energy consumption in screwdrivers. The analysis focused on changes in the kinetic energy of the components of the drive system and the working components of the device in the different phases of its operation. It was shown that the sum of the values of the mass moment of inertia of the working components and the mass moment of inertia of the motor's rotor, reduced to the axis of the screwdriver bit, determined the ability of the screwdriving head to drive in a threaded fastener of a given diameter. The parameters of the newly designed screwdriver for driving in fasteners in the diameter range M6–M12 were collated, and the parameters and variables of the screwdriving process were identified.

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