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Research Article

The Development of the Dynamic Method for Control of the Mechanical Efficiency of a Chain Transmission and its Application for the Study of Factors Influencing the Efficiency of a Chain Transmission with a Double-strand Sleeve-type Chain

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Abstract: The purpose of this study is to develop a dynamic method for control of the mechanical efficiency of chain transmissions, as well as to substantiate it theoretically and experimentally. The proposed method, in contrast to currently used methods for control of the mechanical efficiency of chain transmissions, makes it possible to determine the level of mechanical losses in their parts mating without the use of strain measurement. It greatly simplifies the measurement procedure, eliminates errors associated with a need to calibrate the strain sensors, as well as being more cost-effective. On the basis of the method proposed in the present research factors influencing the mechanical efficiency of a double-strand sleeve-type chain transmission in a wide range of speed operating modes were studied, which is quite time-consuming task in the application of existing methods. The obtained regression equations give a clear picture of the influence rate of main factors on the chain transmission performance. Based on these results we can conclude the applicability of the method developed for the study of the research object and its control during production process.

Keywords: Chain gear, chain transmission performance, efficiency measurement, mechanical losses, moment of inertia

INTRODUCTION

The mechanical efficiency of chain transmissions is an indication of their energy perfection. Inertia characteristics of chain transmissions and friction conditions in their parts mating have a significant effect on the rate of degradation processes, such as wear and back-to-back endurance, which are the criteria of operability and consequently, have a significant impact on the life of chain transmissions. Thus, the mechanical efficiency of a chain transmission is one of the main indicators of the quality of their production (Vorobyev, 1968).

Therefore, the development of methods and instruments for control of the level of losses in chain transmissions, as well as research of the causes that affect their value, are important objectives of modern engineering.

Unfortunately, the majority of publications devoted to the study of chain transmissions (Conwell and Johnson, 1996a, 1996b; Huo *et al*., 2013; Liu *et al*., 1990; Srivastava and Haque, 2009; Troedsson and Vedmar, 1999; Usova, 2007; Zhang *et al*., 2012) pay little attention to the matter of measuring their energy performance, as it is known, that chain transmissions have comparatively high values of their efficiency (92... 97%) (Vorobyev, 1968) and the existing methods for the mechanical efficiency control do not meet necessary requirements of measurement accuracy. To register a change of friction losses rate in chain transmissions it is most often applied the strain method that requires high accuracy of signal measurement and a calibration of strain sensors. At that, the torque is registered with a low measurement rate, which is caused by the time required to recover elastodeformed state of a strain element. In addition, there are difficulties associated with the transmission of a signal from a strain element to a measurement system, as well as the high cost of instrumentation (Doppelbauer, 2011; Irimescu *et al*., 2011; Spicer, 2013; Spicer *et al*., 1999; Spicer *et al*., 2000; Levintov, 1984; Rybalchenko, 1981). Other control methods also have a number of disadvantages associated with significant methodological errors and the complexity of the measurement process (the method of reactionary torque) and the inability to create a rated force in a chain during the study (the run-down method) (Katsman, 2004).

Therefore, the research aimed at developing accurate, reliable and cost-effective method for control of the chain transmissions mechanical efficiency that extends the capabilities of existing methods is topical.

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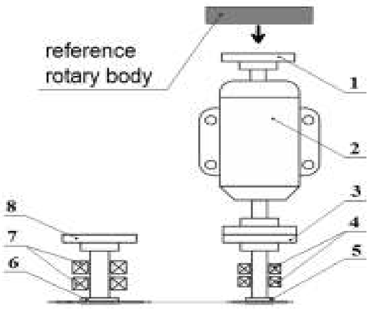

Fig. 1: Diagram of an asynchronous electric Chain drive: 1: Half-coupling for mounting a rotary body; 2: Induction motor; 3: Safety clutch; 4: Driving shaft support bearings, 5: Driving shaft with a driving chain-wheel; 6: Driven shaft with a driven chain-wheel; 7: Driven shaft support bearings; 8: Driven shaft half-coupling; 9: Chain

The purpose of this study is to develop the dynamic method for control of the mechanical efficiency of chain transmissions, which does not have disadvantages that other methods have, as well as to apply the developed method to find the influence rate of parameters of a chain transmission with a double-strand sleeve-type chain on its mechanical efficiency.

MATERIALS AND METHODS

The method for determining the efficiency of an asynchronous electric chain drive: The dynamic method for control of the mechanical efficiency of chain transmissions is based on the method for determining the asynchronous electric chain drive efficiency characterizing mechanical and additional losses in it. Let us consider the implementation of the method in more detail. Figure 1 shows a diagram of a chain transmission driven by an induction motor, which explains the principle of its efficiency determination by using the proposed method.

First, using an encoder we measure the resultant angular acceleration of the asynchronous electric chain drive without a reference rotary body:

$$\varepsilon_1 = \frac{M - M_R}{J_{ChD}} = \varepsilon_1^{/} - \frac{M_R}{J_{ChD}} \tag{1}$$

where, M is the torque developed by the induction motor, N·m; $\varepsilon_1^{/} = M/J_{ChD}$ is the angular acceleration of the asynchronous electric chain drive if there would be no mechanical and additional losses in the induction motor, rad/s^2; M_R is the antitorque moment caused by parasitic forces in the asynchronous electric chain drive, N·m; J_{ChD} is the moment of inertia of the asynchronous electric chain drive, normalized with respect to the driving shaft rotation axis, kg/m^2.

The torque developed by the asynchronous electric chain drive during its speeding up can be determined as follows:

$$M = M_R + J_{ChD} \cdot \varepsilon_1 \tag{2}$$

where, J_{ChD}. ε_1 is the torque to change kinetic energy of the rotating masses of the asynchronous electric chain drive (effective torque), N·m.

On the other hand, in the absence of parasitic forces in the asynchronous electric chain drive an expression for the torque developed by the asynchronous electric chain drive would be calculated as follows:

$$M = J_{ChD} \cdot \varepsilon_1^{/} = J_{ChD} \cdot \frac{M}{J_{ChD}} \tag{3}$$

From expressions (1) and (3) we obtain:

$$\begin{aligned} M &= J_{ChD} \cdot (\varepsilon_1 + \frac{M_R}{J_{ChD}}) = J_{ChD} \cdot (1 + \frac{M_R}{J_{ChD} \cdot \varepsilon_1}) \cdot \varepsilon_1 = \\ &= J_{ChD} \cdot (1 + \frac{M_R \cdot J_{ChD}}{J_{ChD} \cdot (M - M_R)}) \cdot \varepsilon_1 = J_{ChD} \cdot \left(\frac{M}{M - M_R} \right) \\ &\cdot \varepsilon_1 = J_{ChD} \cdot \left(\frac{1}{\eta_{ChD}} \right) \cdot \varepsilon_1 \end{aligned} \tag{4}$$

where, η_{ChD} is the asynchronous electric chain drive efficiency characterizing mechanical and additional losses (depends on mechanical losses in the asynchronous electric chain drive parts mating and additional losses in the induction motor (all kinds of difficult calculated losses caused by the influence of the higher harmonics of magneto motive forces, magnetic induction pulsation and other causes)).

In this study, at the mention of the term of "the moment of inertia of a system of rotating masses taking into account losses" we mean a product of the moment of inertia of that system of rotating masses, J_{SRM} and the inverse efficiency of that system of rotating masses, (1/ η_{SRM}), which characterizes mechanical and additional losses in it.

At the next stage, a rotary body with the reference moment of inertia, J_{ref} is mounted to the half-coupling of the induction motor shaft. At this, the moment of inertia of moving parts increases and therefore, the resulting angular acceleration of the induction motor decreases. The induction motor starts and the angular velocity of the system of rotating masses is brought to rated. Thus, the resultant angular acceleration of the asynchronous electric chain drive with the reference body ε_2 is determined.

In view of the expression (4), an expression for the torque developed by the asynchronous electric chain drive can be represented as follows:

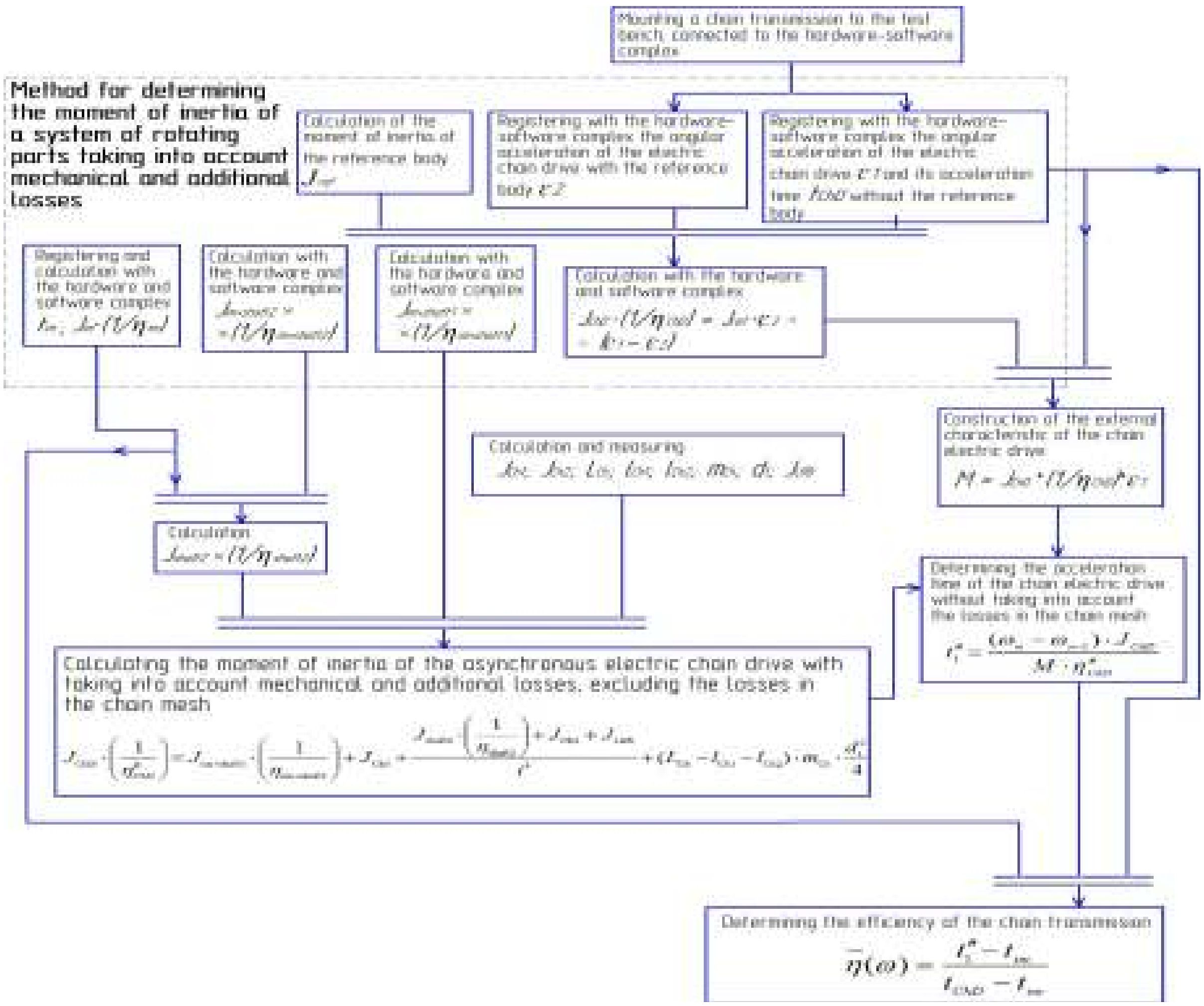

Fig. 2: Block diagram of the method for determination of the mechanical efficiency of the chain transmission

$$M = (J_{ChD} \cdot \left(\frac{1}{\eta_{ChD}}\right) + J_{ref}) \cdot \varepsilon_2 \tag{5}$$

where, J_{ref} is the moment of inertia of the reference body, normalized with respect to the driving shaft rotation axis of the asynchronous electric chain drive, kg/m^2.

Since the speed-torque curve of an induction motor at constant input voltage, power frequency and constant value of winding resistance is always constant, we can equate the right-hand parts of the expressions (4) and (5) to determine the moment of inertia of the asynchronous electric chain drive taking into account losses:

$$J_{ChD} \cdot \left(\frac{1}{\eta_{ChD}}\right) = J_{ref} \cdot \frac{\varepsilon_2}{\varepsilon_1 - \varepsilon_2} \tag{6}$$

Further, having determined the moment of inertia of the asynchronous electric chain drive we can determine its efficiency, taking into account mechanical and additional losses:

$$\eta_{ChD} = \frac{J_{ChD} \cdot (\varepsilon_1 - \varepsilon_2)}{J_{ref} \cdot \varepsilon_2} \tag{7}$$

Similarly, one can find the efficiency of any system of rotating parts, which includes an induction motor, taking into account losses.

At this, the torque developed by the induction motor can be calculated as follows:

$$M = \varepsilon_1 \cdot J_{ChD} \cdot \left(\frac{1}{\eta_{ChD}}\right) \tag{8}$$

The method for determining the mechanical efficiency of a chain transmission: In general, the method for determining the mechanical efficiency of a chain transmission can be represented as a block diagram (Fig. 2), where, $J_{im}\cdot(1/\eta_{im})$ is the moment of inertia of the induction motor with taking into account losses, kg/m^2; $J_{im+shaft2}\cdot(1/\eta_{im+shaft2})$ is the moment of inertia of a system of rotating parts (a rotor of the induction motor, induction motor bearings, a safety coupling, a driven shaft with a driven chain wheel,

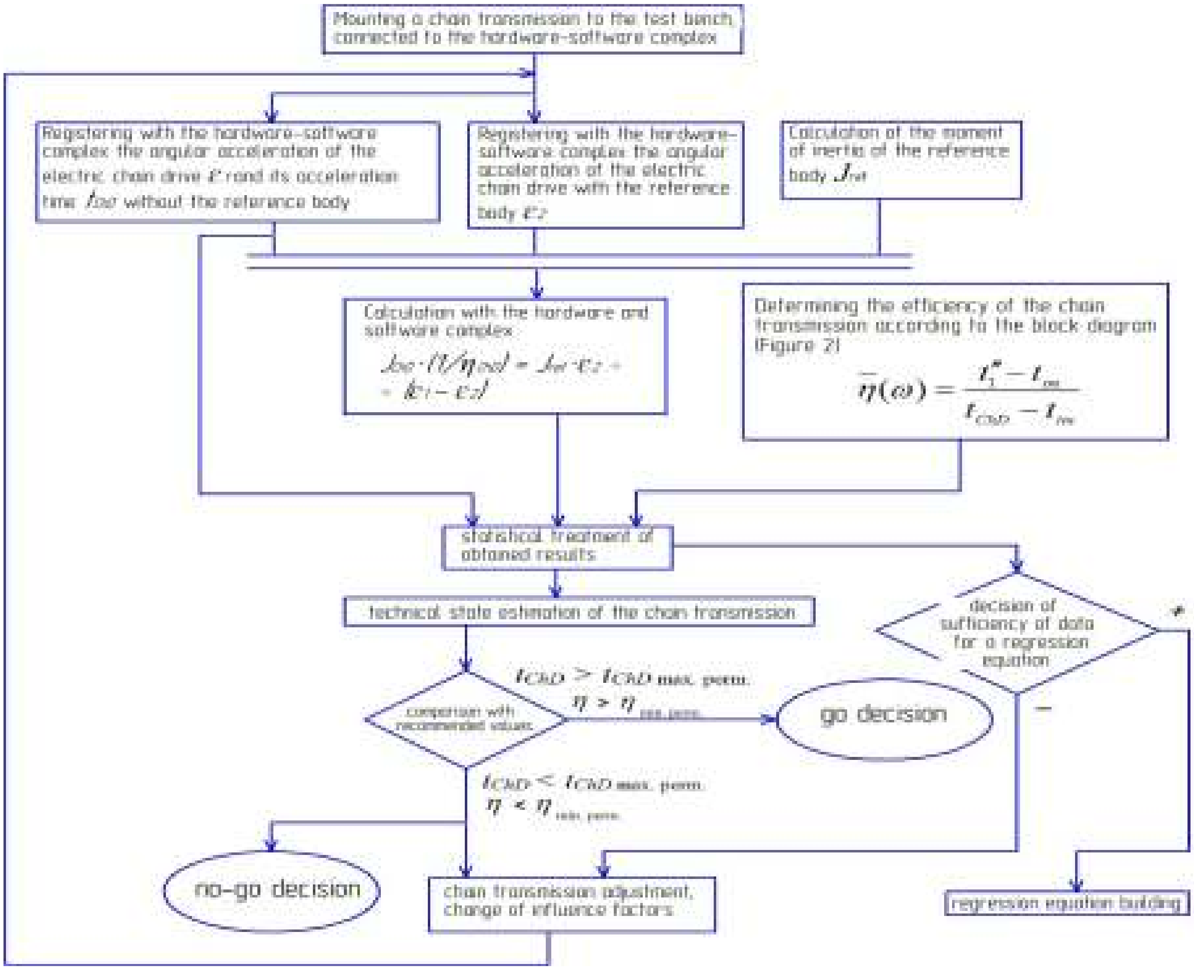

Fig. 3: Block diagram of the dynamic method for control of the mechanical efficiency of the chain transmission

driven shaft bearings) taking into account losses, kg/m^2; $J_{im+shaft1}$·(1 /$\eta_{im+shaft1}$) is the moment of inertia of a system of rotating parts (the rotor of the induction motor, the induction motor beatings, a driving shaft with a driving chain wheel, driving shaft bearings) taking into account losses, kg/m^2; J_{ChI} is the moment of inertia of the chain part wrapped around the driving chain-wheel about its rotation axis, kg/m^2 ; J_{Ch2} is the moment of inertia of the chain part wrapped around the driven chain-wheel about its rotation axis, kg/m^2; J_{shaft2}·(1/η_{shaft2}) is the moment of inertia of a system of rotating parts (the driven shaft, the driven shaft bearings) taking into account losses, kg/m^2; L_{Ch} is the total length of the chain, m; l_{Ch1}, l_{Ch2} is the driving and driven chain-wheel wrap perimeter, respectively, m; m_{Ch} - the linear specific weight of the chain, kg/m; d_1 is the diameter of the pitch circle of the driving chain-wheel, m; J_{LRB} is the moment of inertia of loading rotary bodies, kg/m^2; i is the gear ratio of the chain transmission; t_1'' - is the acceleration time of the asynchronous electric chain drive within selected speed range without taking into account losses, s; t_{im} is the acceleration time of the induction motor within selected speed range, s; t_{ChD} is the acceleration time of the asynchronous electric chain drive within selected speed range with taking into account losses, s; η''_{ChD} is the efficiency of the asynchronous electric chain drive characterizing its mechanical and additional losses without taking into account chain mesh losses; $\Delta\omega$ is the difference between the final and initial angular velocities within selected speed range, rad/s.

The dynamic method for control of the mechanical efficiency of a chain transmission: The method for control of the mechanical efficiency of the chain transmission is shown in the following block diagram (Fig. 3).

To control the mechanical parameters of the chain transmission the hardware-software complex is proposed. It consists of an encoder, transmitter (registrating unit) and Personal Computer (PC) with the installed software for recording and analysis of the digital signal (Fig. 4).

A test bench for the chain transmission study: Figure 5 shows a test bench for control of the mechanical efficiency of a chain transmission, which allows

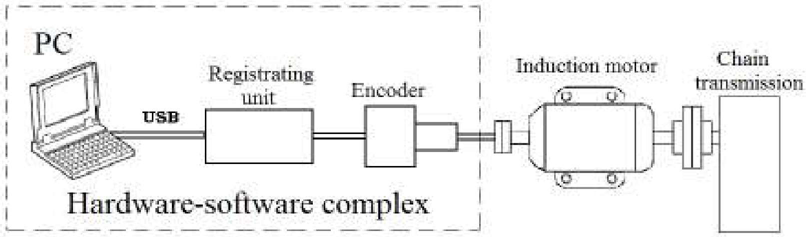

Fig. 4: Schematic diagram of the system for control of the mechanical efficiency of the asynchronous electric chain drive

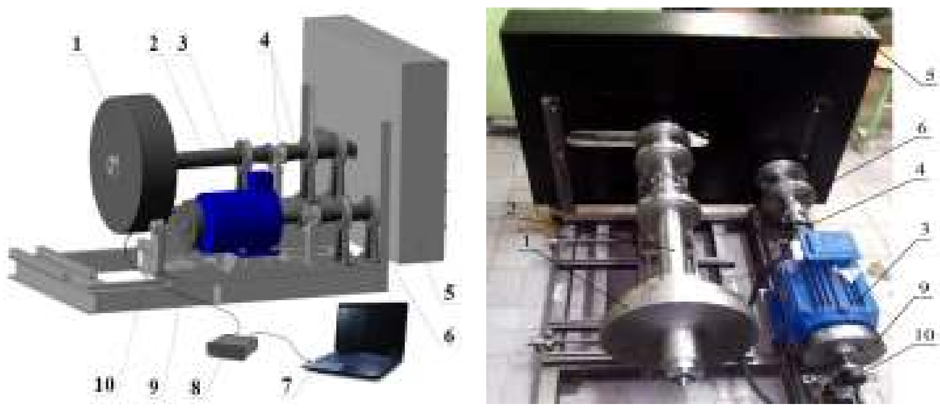

Fig. 5: The test bench for control of the mechanical efficiency of a chain transmission: 1: Loading rotary bodies; 2: Driven shaft; 3: Single-phase induction motor AIR 112MV6 (220/380 V), 1.1 kW, n = 940 rev/min; 4: Torque sensor M40-100; 5: Chain transmission casing; 6: Drive shaft; 7: PC with the installed software; 8: Registrating unit; 9: Reference body; 10: Encoder

recording the acceleration time of an asynchronous electric chain drive. The test bench is designed and built for testing of the developed dynamic method, its experimental substantiation and a study of factors influencing the mechanical efficiency of a chain transmission. The test bench allows changing the qualitative and quantitative values of factors influencing independently of each other to obtain an objective regression equation.

As a test chain a double-strand sleeve-type chain has been selected with the pitch of 9.525 mm (2PV-9, 525-17), which is used in the valve gears (VAZ 2101, 21011, 21013, 2103, 21061, 2107), the engines (GAS 3110, 3105, 3102) and other mechanisms. The number of teeth of the drive chain-wheel is z = 23, the number of teeth of the driven chain-wheel is z = 38.

Presented test bench allows simultaneously varying several parameters of the chain drive, which, according to preliminary theoretical analysis, can influence the chain transmission mechanical efficiency.

Preload chain tension is adjusted by moving a support of the driven shaft along the tracks parallel to the chain-wheels plane.

A necessary condition for improving the accuracy of determining the moment of inertia and the acceleration time of the studied system of rotating masses is the stability of the input voltage for both motor starts (with a reference body and without it), since the torque is quadratic in relation to the voltage according to an expression (Arkhiptsev, 1986):

$$M = \frac{3 \cdot \frac{r_2^I}{s} \cdot p \cdot U_1^2}{6,28 \cdot f \cdot [(r_1 + r_2^I / s)^2 + (x_1 + x_2^I)^2]} = \frac{k \cdot U_1^2}{f} \qquad (9)$$

where, r_2^I and x_2^I is the active resistance and the reactance of the rotor winding, respectively; r_1 and x_1 is the active resistance and the reactance of the stator winding, respectively; p is the number of poles; s is the slip; k is the quantity that can be taken as constant at both starts of the asynchronous electric motor (with a reference body and without it); f is the power frequency, Hz.

Therefore, in experiments we use a voltage stabilizer Saturn SNE-O-10 (11 kVA, 50 A) with a relative error of stabilizing of 0.5%.

RESULTS

Experimental substantiation of data obtained with the dynamic method:

Construction of the speed-torque curve of the asynchronous electric chain drive with different moments of inertia of the loading rotary bodies: According to the theory of the developed dynamic method, the speed-torque curve of an asynchronous

Table 1: The average values of the torque developed by the asynchronous electric chain drive with different loading rotary bodies, N · m

	Number of revolutions, rev/min						
Variable	200-400	400-500	500-600	600-700	700-800	800-850	850-950
$\overline{M_{im1}}$, N·m	6.2803	8.8749	11.2361	13.8035	16.6357	17.2418	16.2469
$\overline{M_{im2}}$, N·m	6.2267	8.7999	11.2734	13.7526	16.6227	17.2794	16.0958
$\overline{M_{im3}}$, N·m	6.2304	8.8159	11.2968	13.7966	16.4589	17.2546	16.1548
$\overline{M_{im4}}$, N·m	6.2765	8.8426	11.1956	13.7195	16.4786	17.2965	16.1453

Table 2: The average values of the forces developed on the driving chain-wheel, N

Moment of inertia of the loading rotary bodies, kg/m^2	Number of revolutions, rev/min						
	250-400	400-500	500-600	600-700	700-800	800-850	850-950
0.26166	140.1749	198.0881	250.7884	308.0929	370.3694	387.0671	362.6291
0.43939	148.1077	206.9058	270.2871	329.7271	392.5451	411.6934	381.8294
0.61712	156.6213	221.3934	288.8346	346.4463	422.1020	434.4471	402.2354
0.87878	160.8291	227.8979	291.4409	358.2629	425.9306	452.5340	410.4340

motor remains constant regardless of the load at constant input voltage, power frequency and winding resistance. To confirm this hypothesis, we construct the speed-torque curve of the asynchronous electric chain drive with different moments of inertia of the loading rotary bodies.

In addition, we check the change of the force in the chain when changing the moments of inertia of the loading rotary bodies. In order to determine the average value of the effective force in the chain within selected speed range, $\overline{P_{1Ch}}$, it is necessary to use the following expression:

$$\overline{M_{1Ch}} = \overline{M_{im}} - J_{im+shaft1} \cdot \left(\frac{1}{\overline{\overline{\eta_{im+shaft1}}}} \right) \cdot \overline{\varepsilon_{ChD}}, \qquad (10)$$

where, $\overline{M_{im}}$ is the average value of the torque developed by the induction motor within selected speed range, N·m; $J_{im+shaft1} \cdot \left(1/\overline{\eta_{im+shaft1}}\right)$ is the average value of the moment of inertia of a system of rotating masses (the induction motor rotor; the induction motor bearings; the chain transmission drive shaft; the drive shaft bearings; the drive chain-wheel) normalized with respect to the driving shaft rotation axis taking into account losses within selected speed range, kg/m^2; $\overline{\varepsilon_{ChD}}$ is the average value of the acceleration of the asynchronous electric chain drive with the load within selected speed range, rad/s^2.

Figure 6 shows the speed-torque curves depending on the moment of inertia of the loading rotary bodies attached to the driven shaft.

Figure 6 and Table 1 show that the average value of the torque is changed depending on the value of the moment of inertia of the loading rotary bodies with the maximum relative deviation of 0.95%, what can be a validity of the expression (6). With an increase of the moment of inertia of the loading rotary bodies (the angular acceleration of the drive decreases), the curves of the torques developed on the driving chain-wheel tends to the speed-torque curve of the induction motor that can be an experimental confirmation of the validity of the expression (10) and hence the reliability of the results obtained with the developed method.

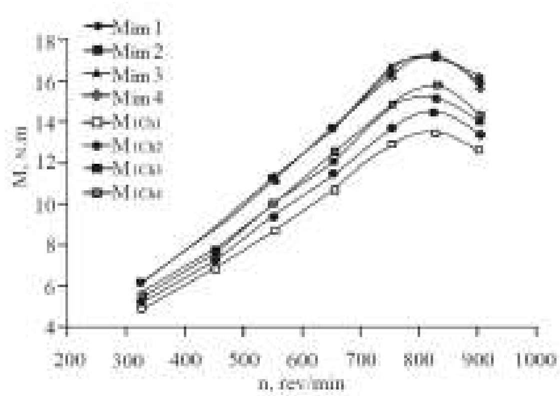

Fig. 6: The speed-torque curves of the single-phase induction motor and the curves of the torques developed on the driving chain-wheel: M_{im1}, M_{im2}, M_{im3}, M_{im4} are the torques of the asynchronous electric chain drive with the moment of inertia of the loading rotary body of 0.261 kg/m^2; 0.439 kg/m^2; 0.617 kg/m^2; 0.878 kg/m^2, respectively; M_{1Ch1}, M_{1Ch2}, M_{1Ch3}, M_{1Ch4} are the torques developed on the driving chain-wheel with the moment of inertia of the loading rotary body of 0.261 kg/m^2; 0.439 kg/m^2; 0.617 kg/m^2; 0.878 kg/m^2, respectively

Table 2 shows the average values of the forces developed on the driving chain-wheel when changing the moment of inertia of the loading rotary bodies within selected speed range.

The comparison of the values of the torque obtained with the developed method and with the torque sensor M40-100 during the induction motor speeding-up: The experiment was carried out according to the diagram shown in Fig. 7.
Table 3 shows the values obtained.

The comparison of the moment of inertia values of the rotary body obtained with the calculation

Table 3: The comparison of the average torque values within selected speed range obtained with the Dynamic Method (DM) and using the Torque Sensor (TS)

Variable	Number of revolutions, rev/min						
	250-400	400-500	500-600	600-700	700-800	800-850	850-950
M_{DM}, N·m	5.262	7.438	9.420	11.555	13.886	14.515	13.582
M_{TS}, N·m	5.320	7.356	9.498	11.620	14.034	14.686	13.692
δ, %	1.102	-1.120	0.824	0.560	1.055	1.178	0.803

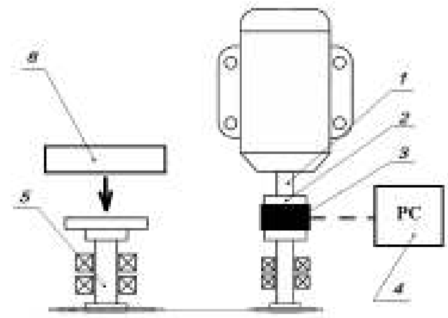

Fig. 7: Diagram for control of the asynchronous electric chain drive torque with torque sensors M40-100: 1: Motor shaft; 2: Safety coupling; 3: Torque sensor M40-100; 4: PC with installed software; 5: Driven shaft with a coupling; 6: Loading rotary body

Table 4: The error of determining the moment of inertia of the rotary body with a known value of the moment of inertia

Variable	Number of revolutions, rev/min		
	200-400	400-600	600-800
J_{true}, kg/m^2	0,016000	0,016000	0,016000
J_{add}, kg/m^2	0,015855	0,015898	0,015911
Δ, kg/m^2	0,000145	0,000102	0,000089
δ, %	0,906250	0,637500	0,556250

(measuring its geometry and weight) and using the developed dynamic method: To substantiate the reliability of the results obtained with the dynamic method according to the present experiments we carried out two series of measurements. While using average values of variables within selected speed range.

First, we determine the moment of inertia of the asynchronous electric chain drive within selected speed range taking into account losses, $1//\eta_{ChD}$. J_{ChD}. Then determine the moment of inertia of the asynchronous electric chain drive with the additional rotary body (its moment of inertia, J_{add}, is 0,016 kg/m^2) within selected speed range taking into account losses:

$$\frac{1}{\eta_{ChD+}}\cdot J_{ChD+} = \frac{1}{\eta_{ChD}}\cdot J_{ChD} + J_{add}$$

Knowing the average values of the moments of inertia of both systems of rotating masses, one can easily calculate the average value of the moment of inertia of the additional rotary body and compare it with the calculated one.

Table 4 shows the values of errors of the moment of inertia measurement of the additional rotary body

Table 5: The determination of indirect errors of the moment of inertia of the rotating masses system measurement

Variable	Number of revolutions, rev/min		
	200-400	400-600	600-800
$\overline{\varepsilon_0}$,, rad/s^2	46.833	71.445	107.494
$\overline{\varepsilon_{ref}}$,, rad/s^2	20.145	31.576	47.64
$\bar{J}$. $1/\eta \times 10^{-4}$, kg/m^2	0.1341	0.1407	0.1414
J_{ref}, kg/m^2	0.177652	0.177652	0.177652
$\delta\varepsilon_0$, %	0.363	0.368	0.325
$\Delta\varepsilon_{ref}$, %	0.427	0.357	0.255
δJ_{ref}, %	0.5	0.5	0.5
δ(J. $1/\eta$), %	1.10329	1.04602	0.89466

with the dynamic method taking into account the fact that the true value of its moment of inertia is J_{add} = 0,016 kg/m^2.

According to Table 4, we obtained the convergence of the results obtained with the developed method and with the calculation with a maximum relative deviation of 0.90625%.

Statistical treatment of experimental data: According to the developed method for indication of the moment of inertia of a rotating masses system, it is necessary to measure the angular acceleration of a driving shaft with a reference body and without it. Thus, for the calculation of a systematic error of determination of the moment of inertia there were performed two series of experiments, consisting of 15 measurements of the angular acceleration each.

The fractional systematic error of the determined value of the moment of inertia is estimated as:

$$\delta(J\cdot\frac{1}{\eta}) = \pm\sqrt{\left(\frac{\varepsilon_0}{\varepsilon_0-\varepsilon_{ref}}\cdot\delta\varepsilon_{ref}\right)^2+\left(\frac{\varepsilon_0}{\varepsilon_{ref}-\varepsilon_0}\cdot\delta\varepsilon_0\right)^2+\left(\delta J_{ref}\right)^2} \quad (11)$$

where, ε_{ref} and ε_0 is the angular accelerations of the rotating masses with the reference body attached to the driving shaft and without it, respectively; $\delta\varepsilon$ is the fractional systematic error of the angular acceleration measurement; δJ_{ref} is the fractional systematic error of the measurement of the moment of inertia of the reference body.

The values of errors of the moment of inertia of a rotating masses system measurement with the dynamic method $\delta(J\cdot(1/\eta))$ are presented in Table 5.

Thus, the indirect error of the moment of inertia of the rotating masses system measurement with the dynamic method, based on systematic and random errors of the measurement of the angular acceleration, is about 1%.

Table 6: The average values of the quantities required for determining the mechanical efficiency of the chain transmission

Variable	Speed range, rad/s 200-400	400-600	600-800
The moment of inertia of the rotary masses system (the rotor of the single-phase induction motor P = 1.1 kW; the induction motor bearings, the safety coupling, the driving shaft with the driving chain-wheel; the driving shaft bearings), kg/m^2	0.02720	0.02890	0.02920
The moment of inertia of the rotary masses system (the safety coupling of the driven shaft; the driven shaft with the driven chain-wheel; the driven shaft bearings; the loading rotary body with the moment of inertia of 0.26166 kg/m^2), kg/m^2	0.28684	0.29154	0.29214
The moment of inertia of the chain part wrapped around the driving chain-wheel, kg/m^2	0.000058	0.000058	0.000058
The moment of inertia of the chain part wrapped around the driven chain-wheel, kg/m^2	0.000283	0.000283	0.000283
The gear ratio of the chain transmission	1.650000	1.650000	1.650000
The total length of the chain, m	0.992515	0.992515	0.992515
The driving chain-wheel wrap perimeter, m	0.105413	0.105413	0.105413
The driven chain-wheel wrap perimeter, m	0.188576	0.188576	0.188576
The linear specific weight of the chain, kg/m	0.457800	0.457800	0.457800
The diameter of the pitch circle of the driving chain-wheel, m	0.070000	0.070000	0.070000
The moment of inertia of the asynchronous electric chain drive without taking into account friction losses in the chain mesh, kg/m^2	0.133110	0.136532	0.137050
The moment of inertia of the rotary masses system (the rotor of the single-phase induction motor P = 1.1 kW; the induction motor beatings; the safety couplings), kg/m^2	0.022300	0.023050	0.023250
The torque of the asynchronous electric chain drive, N·m	6.280000	10.06000	15.20000
The angular acceleration of the asynchronous electric chain drive without taking into account losses, rad/s^2	47.18000	73.68000	110.9100
The calculated acceleration time of the asynchronous electric chain drive without taking into account losses, s	0.442980	0.283650	0.188450
The calculated acceleration time of the rotary masses system (the rotor of the single-phase induction motor P = 1.1kW; the induction motor bearings; the safety coupling), s	0.074210	0.047890	0.031970

Importantly, in the course of the study of factors influencing the mechanical efficiency of the chain transmission, the acceleration time of the asynchronous electric chain drive was used as an optimization criterion. Its error can be equated to a random error of measurement of the angular acceleration, the average value of which is less than 0.4% as shown in Table 5.

The determination of constants characterizing the test bench for the chain transmissions control: Let us use the method for determination of the chain transmission efficiency (Fig. 2) and determine the moment of inertia of the asynchronous electric chain drive taking into account friction losses in the chain mesh. To do this, we determine the average values of the moments of inertia of chain drive elements about the axis of rotation of the driving shaft and its reduced acceleration time within selected speed range. Table 6 shows the values obtained.

In the Table 6 the calculated average value of the acceleration time of the chain drive without taking into account losses shows the average time required for an acceleration of the asynchronous electric chain drive within selected speed range, assuming that friction and power losses, associated with vibration of the chain strands, are reduced to zero. That is the ideal acceleration time, at which the chain transmission efficiency would be the highest, taking into account friction losses in the bearing units.

Planning an experiment: In the study of the influence rate of parameters of the chain drive with a double-stranded sleeve-type chain on its mechanical efficiency, it was decided to apply the multifactorial experiment using the approach of mathematical planning.

Table 7: The determination of levels of the factors influencing the mechanical efficiency of the chain transmission with a double-stranded sleeve-type chain

Coded value	Parameters Δb, mm (x_1)	Δl, mm (x_2)	k_{LP} (x_3)	Δt_{Ch}, % (x_4)
+1	10	1,00	3	2,00
0	5	3,75	2	1,22
-1	0	6,50	1	0,44
$2x_i$	10	5,50	1	1,56

Based on the theoretical study of the mechanical efficiency of chain transmissions the following parameters were taken as influence factors: plane-parallel displacement of chain-wheels (x1), deflection of the chain (x2), lubrication process (x3), change of the chain pitch (x4).

Table 7 shows the influence factors, their levels and coded values for the experiment.

The range of variation of a factor $2xi$ is selected on the basis of a preliminary qualitative and quantitative analysis of its influence on the optimization criterion, which is the acceleration time of the asynchronous electric chain drive, t_{ChD}.

In setting the range of variation of the lubrication process, k_{LP}, reference levels were used: k_{LP} = 1-crankcase lubrication; k_{LP} = 2-intermittent lubrication (in the course of the experiment the chain transmission preliminary worked for 24 h at a rated load).

Table 8 shows regression equations obtained.

Statistical analysis of experimental data was carried out: homogeneity of variance was determined; model adequacy, that is, its validity with Fisher's test at

Table 8: Regression equations

Speed range, rev/min	Regression equations, η_{ChD}, %
200-400	$\eta_{ChD} = 88,23 + 2,37\,x_1 + 1,72\,x_2 + 5,33\,x_3 + 0,94\,x_4 - 0,19\,x_2x_4 - 0,86\,x_1x_2$
400-600	$\eta_{ChD} = 86,83 + 2,05\,x_1 + 1,18\,x_2 + 4,76\,x_3 + 0,90\,x_4 - 0,24\,x_2x_4 - 0,22\,x_1x_2$
600-800	$\eta_{ChD} = 86,46 + 1,96\,x_1 + 1,25\,x_2 + 4,97\,x_3 + 1,04\,x_4 - 0,31\,x_2x_4 - 0,19\,x_1x_2$

the significance point of five percent was verified; and the significance of the coefficients of the regression equations was verified as well. Based on these results we can conclude on the adequacy of the obtained model.

DISCUSSION

Based on the results obtained we can conclude that the developed method for control of the mechanical efficiency of a chain transmission makes it possible to control the change of mechanical losses in chain transmissions via the acceleration time of an asynchronous electric chain drive, that increases the control accuracy compared with existing methods due to the negligible quantity of the systematic error and the lack of a need for calibration of measurement elements. The proposed control method involves the use of loading rotary bodies attached to a driven shaft of a chain drive to create the rated force in a chain, eliminating a need for a loader. The developed dynamic control method solves the problem of determining the mechanical efficiency of chain transmissions without taking into account losses in their bearing units.

First designed hardware-software complex makes it possible to control the mechanical efficiency of chain transmissions and find dependences under study in a wide speed range and with higher frequency characteristics than using existing methods.

The speed-torque curves of the induction motor with various values of the moments of inertia of the loading rotary bodies agree in the whole speed range within 0.95% that confirms the reliability of the results obtained with the dynamic method. The experimental results show that the convergence of the results obtained with the dynamic method and with the calculation in determining the moment of inertia of the additional rotary body varies within 1%.

The obtained regression equation shows that the greatest impact on the efficiency of the chain transmission with a double-stranded sleeve-type chain has such factor as a lubrication process k_{LP}, the influence rate of which is almost 2.5 times higher than the influence rate of such factor as a plane-parallel displacement of chain-wheels at a speed higher than 600 rev/min.

Detection of changes in the chain pitch, Δt_{Ch}, using the developed control method when estimating the technical state of the chain transmission seems to be a problematic task, since the rate of influence on the mechanical efficiency of the chain transmission is slightly higher than the random error of an acceleration measurement.

Based on the data we can conclude that the regression equation provide a visual presentation of the rate of factors influence on the mechanical efficiency of chain transmissions. Thus, the application of the developed dynamic method allows the study of influence of parameters of the chain transmission on its mechanical efficiency with an accuracy sufficient to obtain a clear picture of the optimization criterion change due to varying of different influence factors in a wide range of speed modes.

The developed method can be used in companies that manufacture chain transmissions and other chain mechanisms. Determining acceptable levels of the acceleration time of an induction motor attached to a chain transmission under study, it is possible to control its mechanical efficiency, which is influenced by the build quality, lubrication and materials used. It is recognizing that the mechanical efficiency of chain transmissions characterizes degradation processes occurring in their elements and therefore may be a criterion of their lifetime.

Moreover, this method can be applied for companies that produce lubricants for chains. Study of factors influencing the chain gears performance at different conditions and types of lubricants helps to improve their quality and to determine their use limit more accurately.

Further development of the proposed method can take place in the study of other types of mechanical transmissions, such as belt, gear, worm and others; and the developed method can be used to control their mechanical efficiency as well.

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