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Improvement of the Term of Service Life of the Drive Roller Chain of Transmission

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ABSTRACT: In the article are resulted a general scheme and principle of operation of the chain transmission with elastic elements. The article presents an analysis of the elements and reasons for reducing the service life of the chain drives used in drives of various technological machines. An analysis of existing methods for calculating the service life of a drive roller chain of gears is given. An improved method for calculating the transmission with the elastic elements of the chain is presented. A formula is presented for calculating the service life of a belt drive with a composite roller, including a rubber sleeve in a transmission chain.

KEYWORDS: Chain, sprocket, roller, rubber bushing, noise, wear resistance, service life, correction factors.

I. INTRODUCTION

In the known methods of calculating the drive roller chain in order to maintain in the close limits of its service life, a reduction in the transmitted load or pressure in the hinges with an increase in the velocity of the chain is foreseen. Due to this, when the chain drive operates, having average parameters z_1, At and u, under conditions of normal lubrication, corresponding to the velocity of movement of the chain v (k_c = 1) and quiet load (k_γ = 1), the durability of the chain is C ≈10000-т-15000 h, which in DIN8195 is adopted as the base. However, not all chain drives must operate for 10,000 hours or more. In practice, there are often such cases when a chain drive is in operation for 2000–5000 hours and, therefore, there is no need to rely on a chain for much greater durability when designing such a transfer [1].

For transmissions with a chain service life less than the base, it is possible for the same drive chain to increase the transmitted power by increasing the permissible pressure in the hinge or reducing the safety factor, as well as maintaining the transmitted power within the limits corresponding to the base service life chains, to carry out a wider choice of lubrication method, for example, instead of continuous lubrication with the help of an oil bath, apply drip or intra hinged lubricant.

Theoretical and experimental studies are carried out mainly on the wear resistance of the hinged surfaces and to a lesser extent on the endurance of the chain. And although, in general, the fundamentals laid down in the known methods for determining the life of a chain are recognized as correct, however, a comparison of the calculation results for each method shows that, with the same transmission parameters, the estimated service lives of the chain are different, and the results obtained also have large discrepancies with the durability indicators established experimentally for typical parameters and working conditions adopted, in particular, in the development of standard DIN8195 [2,3].

These discrepancies are due to the fact that when calculating the chain for durability, one has to take into account a large number of quantities that directly or indirectly affect the wear of the hinges and chain endurance. In this regard, it is believed that the calculation of the service life of the chain can only be approximate [4].

In addition, in order to increase the service life of the chain transmission, the structural elements of the transmission are improved, in particular, elastic damping elements are introduced in the chain and sprocket designs [5,6,7].

II. EFFECTIVE CIRCUIT DIAGRAM WITH SHOCK-ABSORBING ELEMENTS.

The known construction of the chain transmission contains the leading and the driven stars and the flexible element-chain transmitting the movement from the driving star to the driven [8]. The disadvantage of this chain drive is, in the process of reducing the angle of the chain of asterisks, a significant sag of the driven (idle) chain branch,

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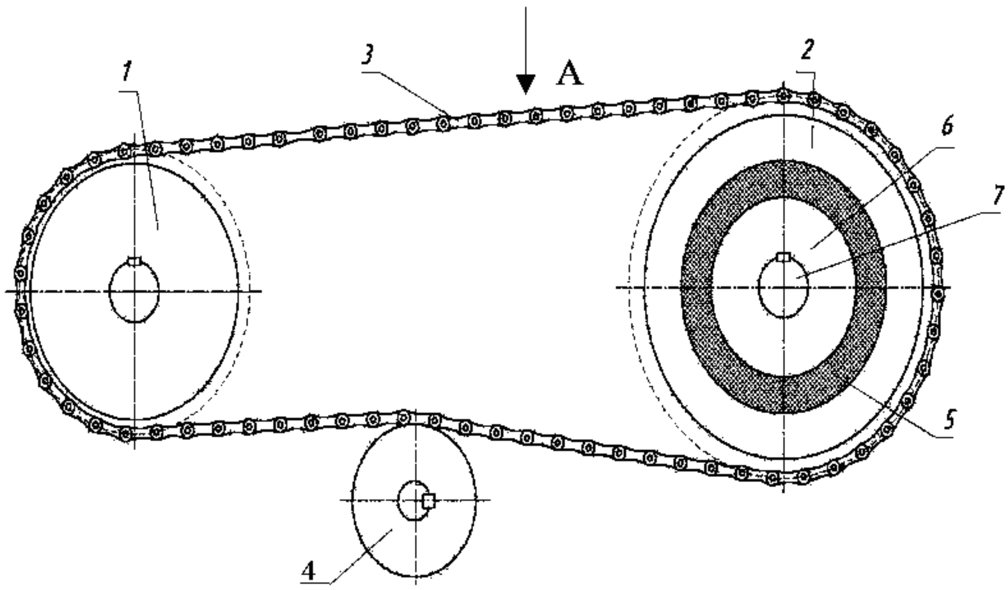
leading to a decrease in efficiency, and in some cases breaking or disengaging the chain with asterisks [9]. In addition, when transmitting significant loads at high speed modes of movement, noise also occurs due to the shock interactions of the teeth of the sprockets with the surfaces of the chain rollers, friction increases, and thus the wear of both the chain rollers and the teeth of the sprockets.

The chains used in mechanical engineering are divided into three main groups according to the nature of the work they perform: drive, traction and cargo. Drive chains are most prevalent. They transmit motion from the energy source to the receiving body of the machine; They work both at small and at high speeds (up to 30-35m/s) and at different distances between the axes (centers) of stars. One chain can be set in motion simultaneously. Practical in all types of chain construction in gears, noise arises due to the shock interactions of the teeth of the sprockets with the surfaces of the chain rollers, and friction also increases, thereby also the wear of the chain rollers and the teeth of the sprockets [10,11]. In order to increase the reliability of operation and the life of the chain transmission, the design of the transmission chain using elastic elements has been improved.

The chain drive design includes a drive 1 and a driven 2 sprocket, a chain 3 spanning them, a tension roller 4. The driven sprocket 2 is made of a composite consisting of an outer part 2 with teeth, a base 6 with an output shaft 7, and an elastic ring sleeve 5 (fig.). 3 includes an outer 8 and inner 9 links, a roller 10, a sleeve 11 and a composite roller 12 consisting of an outer 13 and inner 14 sleeves, between which a rubber (elastic) sleeve 15 is installed. The outer surface 16 of the rubber sleeve 15 is made concave curved shape, respectively internal surface of the outer sleeve 8 is made of a curved convex shape.

Chain transmission works as follows. The rotational movement from the drive sprocket 1 is transmitted to the driven sprocket 2 through the chain 3. Next, the movement from the asterisk 2 is transmitted to the base 6 with the output shaft 7 through the elastic annular sleeve 5. while changing the angular displacements of the driven sprocket 2, resulting from gaps between chain 3 and the teeth of the sprocket 2, as well as due to changes in friction and wear and the gearing, etc., are to some extent aligned (absorbed, absorbed) by an elastic ring sleeve 5. In this case, the rotation of the base 6 with the output shaft 7 of the sprocket 2 becomes more avnomernym and smooth. When the teeth of the sprockets 1 and 2 interact with the roller 12 due to the deformation of the rubber sleeve 15, the wear of the sleeve 13 and the teeth of the stars 1 and 2 is significantly reduced. This also reduces the friction between the sleeve 11 and the roller 10. This leads to an increase in durability and reliable operation of the chain transmission.In the process, due to the outer surface 16 of the rubber bushing 15, when interacting with the teeth of sprockets 1 and 2, the necessary deformations of the bushing 15 occur, especially along its edges, a kind of centering of pressure on the roller 12 from the teeth of sprockets 1 and 2 occurs.This leads to uniform distribution load over the entire length of the roller 12, allowing increased reliability, thereby increasing the resource chain 3 transmissions.



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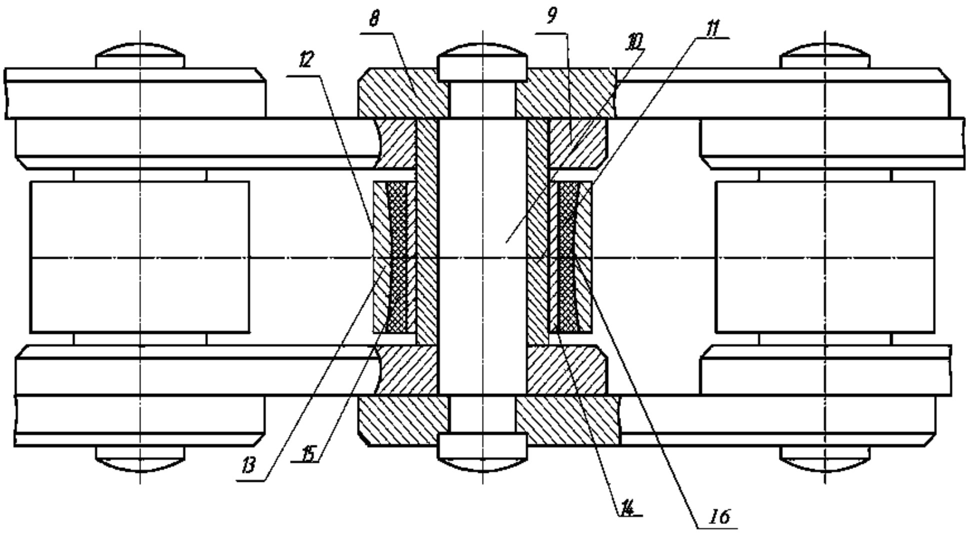

Fig 1. Chain Gear

III.ANALYSIS OF METHODS FOR CALCULATING THE LIFE OF CHAIN TRANSMISSIONS.

In this case, we use the materials presented in [12,13]. Determination of the service life of the circuit according to DIN8195. Based on experimental studies, it was found that the wear life of the hinges of normal quality roller chains is 10,000 h with $\Delta t = 2\%$ or 15,000 h with $\Delta t = 3\%$, while the initial parameters and operating conditions of a two-star chain transfer the following: the number of teeth of a smaller asterisk $z_1 = 19$; gear ratio and - 3; center distance $A_t = 40$ steps; allowable increase in the average step $\delta_t = 2\%$; the number of links in the chain loop L_t – is even; grease is abundant ($k_0 = 1$).

The initial parameter and conditions correspond to the basic pressure values p0 given in DIN, depending on the given v and z_1. For parameters and conditions different from the original, the pressure must be corrected by means of correction factors λ_1, K_N, and k_γ values of which are given in DIN. Thus, the specific wear rate (friction path) λ_1 takes into account changes in the center-to-center distance in the range $A_t = 20 \div 160$ depending on $u = 1 \div 7$ and the nature (stress) of the load, the value of the power factor K_N depends on and, u z_1. by increasing the values of the parameters z_1, A_t, and u the permissible pressure [p] increases in power law.

Allowable pressure to ensure the durability of the chain, regulated DIN8195, can be determined by dependencies [12].

$$[p] = p_0 k_c \frac{\lambda_1 K_N}{k_\gamma} \tag{1}$$

where k_c – is the coefficient of lubrication, taken in accordance with the speed of movement of the circuit v.

The permissible pressure must be reduced in the following cases of chain operation: with v < 4 m/s in conditions of insufficient lubrication – 1,7 times, and in the absence of lubrication – 6,7 times; when v = 4 ÷ 7 m/s and insufficient lubrication – 3,3 times.

Using the tables [12,13] and DIN8195 plots, you can choose a chain that provides a regulated transmission time, but it's almost impossible to calculate the life of the chain but given parameters due to the absence of formulas.

In the method of calculation, based on the specific work of friction and wear criteria. At the same time, the accuracy of the calculation depends on the presence of a number of coefficients, the value of which can be obtained as a result of testing chain transmissions with parameters that are close to the parameters to be calculated and operating conditions. A detailed description and recommendations on the application of this method are given in. This method correctly and justifiably provides for determining the life of a chain, based on the maximum allowable increase in the chain pitch, according to two criteria: to engage the chain with a larger sprocket at $z_2 > 50$ and to lose the strength of the hinge at $g_2 < 50$. More accurate method, should be used primarily in the calculation of chain transmissions intended for machines and mechanisms produced in large series, conducting experimental studies and tests to establish and refine

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the correction factors for the actual pair etram and conditions of the transfer operation. The calculation is divided into two parts: 1) the calculation of the factor of safety; 2) determining the service life of the hinge wear.

The safety factor is taken in the range k = 13 ÷ 40 and [p] = 8 ÷ 30 MPa, depending on v. With this principle of choice of the chain, according to the author, the expected service life at constant operation *(k_c = 1; λ = 1; k_m = 1)* will be C = 5000 ÷ 30000 h. The validity of this assumption corresponds to the calculation results according to his proposed formula

$$C = 873{,}4\frac{L_t t}{ud}\cdot\frac{z_1 u}{1+u}\left(\frac{10ck_n}{p_0\lambda k_v}\right)^3 \qquad (2)$$

where c – is the wear coefficient, which for v = 1 ÷ 12 m/s is in the range of 43,6 – 39,2; k_n – is the ratio of the number of sprocket teeth, corresponding to the power factor K_N according to DIN8195; p_0 – base pressure, MPa; λ – is the coefficient of friction path.

The most acceptable from a practical point of view is the method of calculating the service life of a transmission is the representations in [1,4]. The basis of the recommended method and the derivation of formulas for calculating the service life of roller chains for the wear resistance of hinges:

$$C = 4350\frac{\Delta_t k_u k_m k_c \sqrt[3]{z_1}}{k_v p}\sqrt[3]{\frac{uA_t}{v}} \qquad (3)$$

where, k_u – is the coefficient taking into account the type of chain, k_m – is the coefficient of row spacing of the chain, k_v – is the coefficient taking into account the nature of the load, k_c – is the lubrication coefficient (nature of friction), p – is the specified pressure.

Where in $p \leq \frac{50}{\sqrt[3]{v}} MPa$

v – is the chain speed, z – is the number of sprocket teeth, k – is the gear ratio, A_t – is the center-to-center distance, Δt – is the change (increase) in the chain pitch.

IV. THE CALCULATION OF THE SERVICE LIFE OF THE CHAIN, TAKING INTO ACCOUNT THE RUBBER SLEEVE OF THE TRANSFER ROLLER.

In the process of chain drive operation, due to the damping of the loads on the roller and, accordingly, on the teeth of the sprockets, the wear of the roller will significantly decrease, and the chain pitch will change poetically [14,15]. In this case, taking into account the deformation of the rubber sleeve of the chain, the coefficient of pitch change is determined from the expression.

$$\Delta_t = \frac{t - 2h_u + d/c_в}{t} \qquad (4)$$

where, t – is the chain pitch, hu is the amount of roller wear, $C_в$ – is the stiffness coefficient of the rubber bushing, d – is the load on the roller.

The coefficient taking into account lubrication, the nature of the friction roller is proposed to determine from the expression:

$$k_c = 1 + \frac{x_в}{d_в} \qquad (5)$$

where $X_в$ – is the deformation of the roller sleeve, $d_в$ – is the average diameter of the rubber sleeve.

The pressure coming on the roller chain with elastic elements is depreciated. But, at the same time, the linear speed of the chain will change cyclically due to this deformation [16,17]. Therefore, the value of p will be much smaller and is determined from the formula:

$$p \leq \frac{50kg}{\sqrt[3]{v \pm \Delta v}} \qquad (6)$$

kg = 0,3 ÷ 0,5 - coefficient taking into account the reduction of pressure on the roller due to the depreciation (deformation) of the rubber roller sleeve, Δv = (0,05 ÷ 0,1) bushing roller chain transmission.

In this case, the calculation of the transmission chain with the elastic rubber roller sleeve is proposed to be calculated using the following formula:

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$$C = \frac{87\, k_{\text{u}} k_m (d_{\text{в}} + x_{\text{в}})(t - 2h_u + d/c_{\text{в}})\sqrt[3]{z^2 u^2 A_t^2}}{k_v k_g d_v t} \tag{7}$$

A driven example of the calculation of the service life of a chain transmission, taking into account the rubber sleeve of the roller with the following initial values of the parameters at: P = 1,6 Kwt; n_1 = 300 rot/min; z_1 = 12; z_2= 38; u = 3,0; L_t = 108; k_v = 1,0; v = 1,15 m/s; A_t = 42; k_u = 1,25; k_m = 9 ÷ 10; h_u = 0,05 ÷ 0,08; $C_{\text{в}}$ = $1{,}4 \cdot 10^4$ N/m; d = (25 ÷ 40) H; $x_{\text{в}}$ = $(0{,}2 \div 0{,}35) \cdot 10^{-3}$ m; $d_{\text{в}}$ = $5{,}6 \cdot 10^{-3}$ m; k_y = (0,3 ÷ 0,5); Δv = (0,05 ÷ 0,1) v.

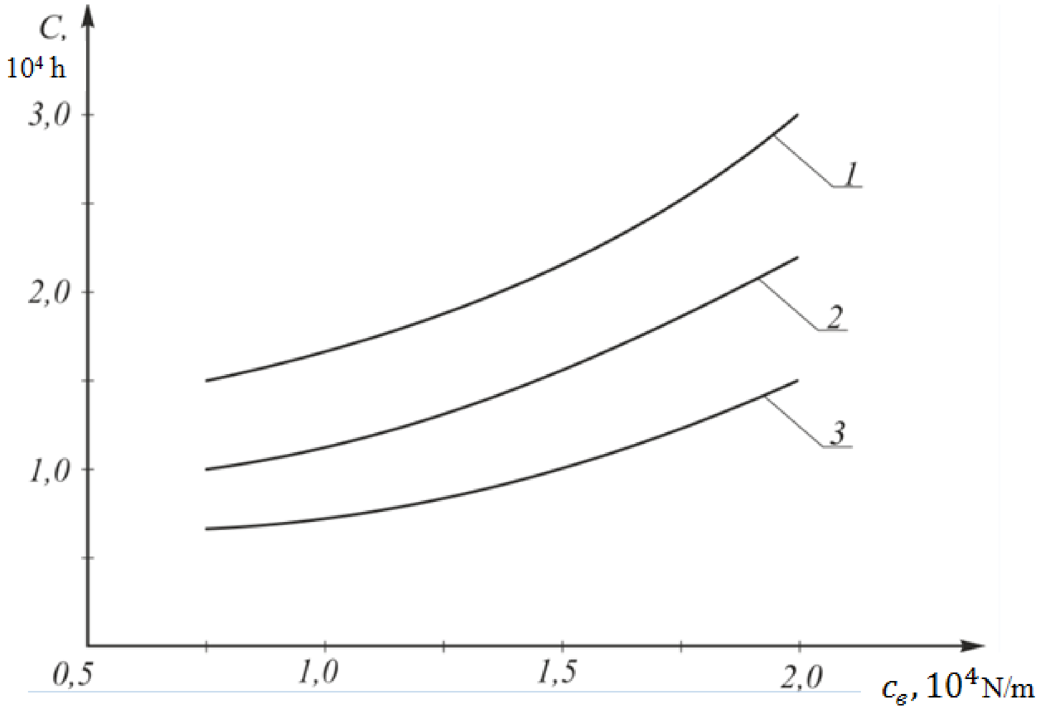

Fig. 2. (I) Dependences of the change in the estimated service life of the chain on the change in the stiffness coefficient of the rubber bush of the roller.

Where 1 – at d = 25 H; 2 – at d = 35 H; 3 – at d = 50 H;

V. ANALYSIS OF THE RESULTS.

Based on the numerical solution of the problem (7) for a chain drive with a composite roller of a chain with a rubber bushing, graphical dependencies of the change in the estimated service life of the chain from the change in the stiffness coefficient of the rubber roller bushing were constructed. The analysis of dependences shows that an increase in the stiffness coefficient of the rubber roller bush leads to an increase in the service life of the chain according to a nonlinear pattern. So when changing the stiffness of the rubber bushing from $0{,}78 \cdot 10^4$ N/m to $2{,}05 \cdot 10^4$ N/m, the service life of the chain increases from $1{,}05 \cdot 10^4$ h to $3{,}0 \cdot 10^4$ h with a load of 25 N on the roller. When driving the load on the roller, the life of the chain decreases. So, with d = 50 N, the service life with increasing $C_{\text{в}}$ to $2{,}05 \cdot 10^4$ N/m increases from $0{,}71 \cdot 10^4$ h to $1{,}51 \cdot 10^4$ h. Therefore, to increase the service life of the transmission chain with a composite chain roller with a rubber bushing up to $(2.5 \div 2.8) \cdot 10^4$ h. It is advisable to choose the following stiffness values of the rubber sleeve with = $(1.7 \div 2.1) \cdot 10^4$ N/m for d ≤ (25 ÷ 30) N.

It should be noted that to increase the service life of the transmission, it is important to study the values of the deformation of the rubber bush, which directly affects the wear of the roller, as well as the change in pitch and reduction of noise in the transmission.

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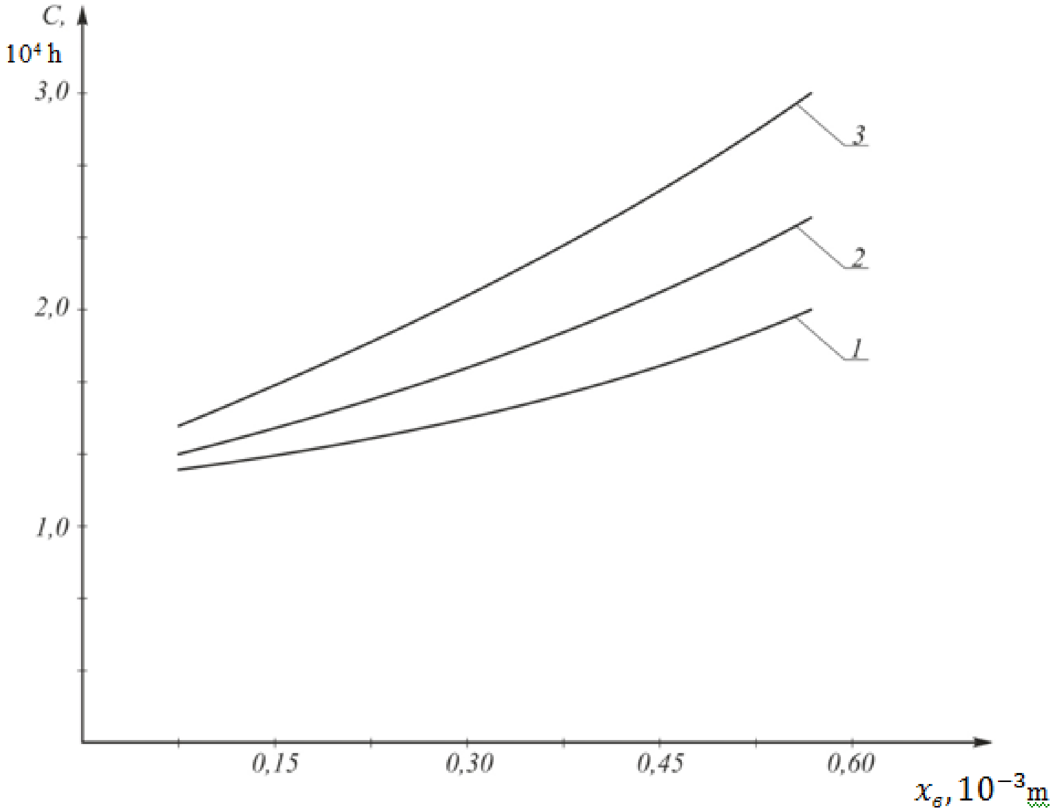

Fig. 3. Graphic dependences of the change in the service life of the chain on the change in the deformation of the elastic sleeve of the transmission chain roller
1– at p = 10 MPa; 2 – at p = 12 MPa; 3 – at p = 14 MPa

Figure 3 presents the dependences of the change in the service life of the chain on the magnitude of the deformation of the elastic sleeve of the transmission chain roller. With an increase in the deformation value of $0{,}085 \cdot 10^{-3}$ m to $0{,}54 \cdot 10^{-3}$ m, the service life of the transmission increases from $1{,}52 \cdot 10^4$ h to $3{,}05 \cdot 10^4$ with a value of p = 10 MPa. As p increases to 14 MPa, it increases from $1{,}26 \cdot 10^4$ h to $1{,}92 \cdot 10^4$ h. This is explained by the fact that when the teeth of the sprockets engage with the chain, the deformation of the rubber bushing of the roller absorbs loads.

This significantly reduces wear on the rollers and the teeth of the sprockets, and reduces the noise in the gear. To ensure the service life of the chain with a composite roller with a rubber bushing up to $(25 \div 30) \cdot 10^4$ h, the 1847 rubber grade is recommended for the manufacture of an elastic bushing, which allows deformations within $(0{,}3 \div 0{,}6) \cdot 10^{-3}$ m.

VI. FINDINGS

Recommended efficient resource-saving design of the chain transmission with composite rollers with an elastic element of the chain. Based on the analysis of existing methods for calculating the transmission chain service life, a formula is recommended for calculating the transmission service life with a composite roller with a rubber bushing. The transmission parameters are justified, allowing the transmission service life to be increased 1.5 times.

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