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METHOD OF CALCULATION OF THE SERVICE LIFE OF A DRIVE ROLLER CHAIN DRIVE WITH AN ELASTIC ELEMENT

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Abstract
The article provides a General scheme and principle of operation of a chain transmission with elastic elements. The article presents an analysis of the elements and reasons for reducing the service life of chain gears used in drives of various technological machines. The analysis of existing methods for calculating the service life of the drive roller chain is given. There given an improved method for calculating the service life of the transmission, taking into account the elastic elements of the chain. Also, formula for calculating the service life of a chain transmission with a composite roller, including a rubber sleeve of the transmission chain, is presented.

Keywords: Chain, sprocket, roller, rubber sleeve, noise, wear resistance, service life, correction factors.

Introduction

In the known methods of calculating the drive roller chain, in order to keep its service life within close limits, it is foreseen to reduce the transmitted load or pressure in the joints with an increase in the speed of the chain. Due to this, during the operation of a chain drive with average parameters z_1, At and u under normal lubrication conditions corresponding to the chain speed v ($k_c = 1$) and a quiet load ($k\gamma = 1$), the chain life is C≈10000-t-15000 h, which in the DIN8195 standard is taken as the basic one. However, not all chain drives have to run for 10,000 hours or more. In practice, there are often such cases when the chain transmission is in operation for 2000-5000 hours and, therefore, there is no need to count on the chain for a much greater durability when designing such a transmission [1].

For gears with a chain service life less than the baseline, it is possible for the same drive chain to increase the transmitted power by increasing the admissible pressure in the joint or reducing the safety factor, as well as provided that the transmitted

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power remains within the limits corresponding to the base life chain, carry out a wider choice of lubrication method, for example, instead of continuous lubrication using an oil bath, apply drip lubrication or inside hinge lubrication.

Theoretical and experimental studies are carried out mainly on the wear resistance of the hinge surfaces and, to a lesser extent, on the chain endurance. And although in general terms the foundations laid down in the known methods for determining the service 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 calculated service life of the chain is different, while the results obtained also have large discrepancies with indicators of durability, established experimentally for typical parameters and operating conditions, adopted, in particular, when developing the standard DIN8195 [2].

These discrepancies are due to the fact that when calculating the chain for durability, you have to take into account a large number of values that directly or indirectly affect the wear of the joints and the endurance of the chain. In this regard, it is believed that the calculation of the service life of the chain can only be approximate [3].

In addition, to increase the service life of the chain transmission, the structural elements of the transmission are being improved, in particular, elastic shock-absorbing elements are introduced in the structures of the chain and sprockets [4,5].

The effective chain design with shock-absorbing elements. The known design of the chain transmission contains a driving and driven sprocket and a flexible chain element that transmits movement from the driving sprocket to the driven one [6].

The disadvantage of this chain drive is, during operation, a decrease in the angle of the chain of sprockets, a significant sagging of the driven (idle) chain branch, leading to a decrease in efficiency, and in some cases, rupture or disengagement of the chain with sprockets [7]. In addition, when significant loads are transferred at high-speed modes of movement, noise occurs and due to the impact interactions of the teeth of the sprockets with the surfaces of the chain rollers, friction increases, thereby also the wear of both the chain rollers and the teeth of the sprockets.

The chains used in mechanical engineering, according to the nature of the work they perform, are divided into three main groups: drive, traction and cargo. Drive chains are the most common. They transmit motion from the energy source to the

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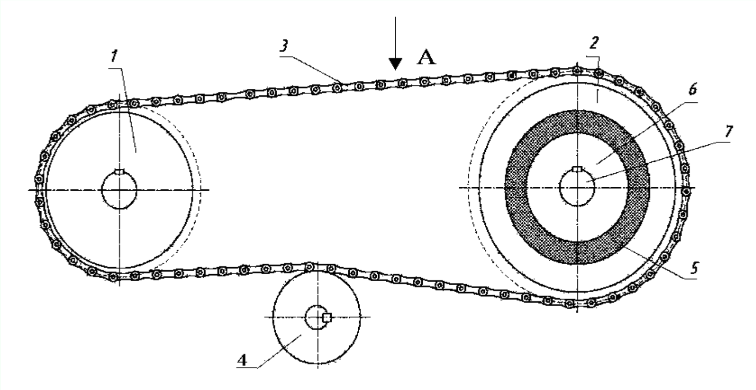
receiving body of the machine; work both at low and high speeds (up to 30-35 m/s) and at different distances between the axes (centers) of the sprockets. One chain can drive several at the same time. In almost all types of chain design, noise occurs in gears due to the impact interactions of the teeth of the sprockets with the surfaces of the chain rollers, and also increases the friction, thereby also the wear of both the chain rollers and the teeth of the sprockets [8]. To increase the reliability and resource of the chain transmission, the design of the transmission chain has been improved using elastic elements.



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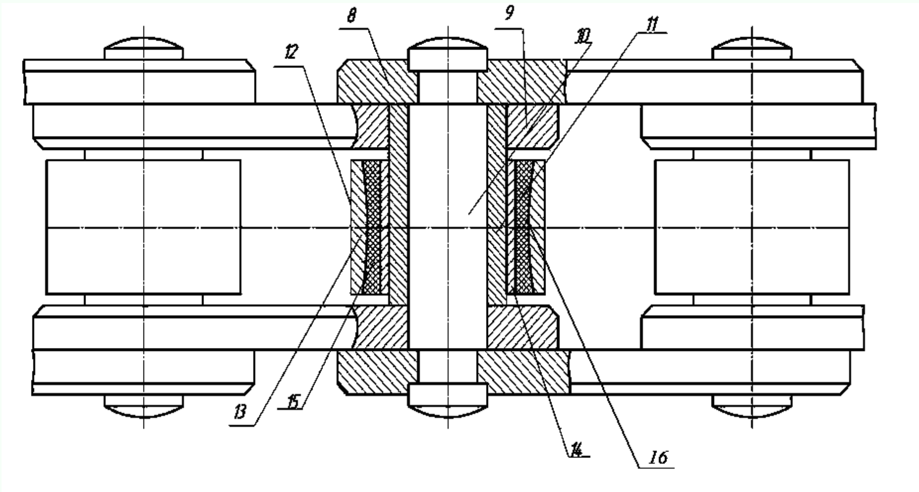

Fig.1. Chain drive

The design of the chain drive includes a driving 1 and driven 2 sprockets, a chain 3 enveloping them, a tension roller 4. A driven sprocket 2 is made of a composite consisting of an outer part 2 with teeth, a base 6 with an output shaft 7, an elastic annular sleeve 5 (Fig. 1). The chain 3 includes an outer 8 and an inner 9 links, a roller 10, a bushing 11 and a composite roller 12, consisting of an outer 13 and an

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inner 14 bushings, between which a rubber (elastic) bush 15 is installed. The outer surface 16 of the rubber bush 15 is made of a concave curved shape, accordingly, the inner surface of the outer sleeve 8 is made with a curved convex shape.
The chain transmission works as follows. Rotational motion from the driving sprocket 1 is transmitted to the driven sprocket 2 through the chain 3. Further, the movement from the sprocket 2 is transmitted to the base 6 with the output shaft 7 through an elastic annular sleeve 5. At the same time, the change in the angular displacements of the driven sprocket 2 arising from the gaps between the chain 3 and the teeth of the sprocket 2, as well as due to changes in friction and wear on the engagement, etc., are to some extent leveled (amortized, absorbed) by the elastic annular sleeve 5. In this case, the rotation of the base 6 with the output shaft 7 of the sprocket 2 becomes more uniform 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 sprockets 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 the durability and reliable operation of the chain transmission. In the process of operation, due to the execution of the outer surface 16 of the rubber bushing 15 when interacting with the teeth of the sprockets 1 and 2, the necessary deformations of the bushing 15 occur, especially along its edges, a kind of centering of the pressure on the roller 12 from the side of the teeth of the sprockets 1 and 2. This leads to a uniform distribution load along the entire length of the roller 12, which allows increasing reliability, thereby increasing the resource of the chain 3 transmission.

Analysis of methods for calculating the service life of chain drives. In this case, we will use the materials presented in the works. Determination of chain life according to DIN8195. Based on experimental studies, it was found that the wear life of the joints of normal quality roller chains is 10,000 hours at $\Delta t = 2\%$ or 15,000 hours at Δt=3%, while the initial parameters and operating conditions of the two-star chain drive are as follows: number of teeth smaller asterisk z_1=19; gear ratio u-3; center-to-center distance At=40 steps; allowable increase in the average step δt=2%; the number of links in the chain circuit Lt is even; abundant lubrication (k_0=1).

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The reference values and conditions correspond to the reference pressure values p_0 given in DIN, depending on the set values for v and z_1. For parameters and conditions that differ from the original, the pressure must be corrected using the correction factors λ_1, K_N, u, $k\gamma$, the values of which are given in DIN. Thus, the specific wear factor (friction path) λ_1 takes into account changes in the center-to-center distance in the range At=20÷160 depending on u=1÷7 and the nature (impact) of the load, the value of the power factor K_N depends on and and z_1. In this case, with an increase in the values of the parameters z_1, A_t and u the allowable pressure [p] increases according to the power dependence.
The permissible pressure for chain durability as specified in DIN8195 can be determined from the relationship.

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

where k_c is the lubrication coefficient, taken in accordance with the speed of the chain v.
The permissible pressure must be reduced in the following cases of chain operation: at v<4 m/s under conditions of insufficient lubrication - by 1.7 times, and in the absence of lubrication - by 6.7 times; at v=4÷7 m/s and insufficient lubrication - 3.3 times.
Using tables and diagrams DIN8195, you can choose a chain that provides a regulated duration of the transmission, but it is almost impossible to calculate the service life of a chain but given parameters due to the lack of formulas.
In the calculation method based on specific work of friction and wear criteria. In this case, 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 drives with parameters and operating conditions close to the calculated ones. The work [4] provides a detailed description and recommendations for the application of this method. This method correctly and reasonably provides for determining the service life of the chain, based on the maximum permissible increase in the chain pitch, according to two criteria: by the engagement of the chain with a larger sprocket at z_2>50 and by the loss of strength of the hinge at r_2<50. The method is more accurate, should be applied in first of all, when calculating chain transmissions intended for machines and mechanisms produced in large series, conducting experimental research and testing to establish and refine correction factors in relation to the actual parameters and operating conditions of the transmission. The

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calculation is divided into two parts: 1) calculation of the safety factor; 2) determination of 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 circuit selection, the expected life of its service under constant operation (k_c=1; λ=1; k_m=1), according to the author, will be C=5000÷30000 hours. The validity of this assumption corresponds to the results of the calculation according to the formula proposed

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

where c is the wear factor, which for v=1÷12 m / s is in the range of 43.6–39.2; k_n is the factor of the number of teeth of the sprocket corresponding to the power factor K_N according to DIN8195; p_0 — base pressure, MPa; λ is the coefficient of the friction path.

The most acceptable from a practical point of view is the method for calculating the service life of the transmission is presented in the works [1,4]. The recommended method and the derivation of formulas for calculating the service life of roller chains based on the wear resistance of the joints are based on:

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

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

Wherein $p \le \frac{50}{\sqrt[3]{v}}$ МПа

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

Calculation of the service life of the chain taking into account the rubber bush of the transfer roller. During the operation of the chain transmission, due to the amortization of the loads on the roller and, accordingly, on the teeth of the sprockets, the wear roller is significantly reduced, the chain pitch is significantly changed [9]. In this case, taking into account the deformation of the rubber bush of the chain, the coefficient of change of the pitch is determined from the expression.

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

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where, t is the chain pitch, h_u is the amount of roller wear, $c_в$ is the stiffness coefficient of the rubber sleeve, d is the load on the roller.
In this case, the coefficient taking into account lubrication, the nature of roller friction is proposed to be determined from the expression:

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

where, $x_в$ is the deformation of the roller sleeve, $d_в$ is the average diameter of the rubber sleeve.
The pressure applied to the chain roller with elastic elements is absorbed. But, in this case, the linear speed of the chain will change cyclically due to this deformation [10]. Therefore, the p value will be much less and is determined from the formula:

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

kg=0,3÷0,5 – coefficient taking into account the decrease in pressure on the roller due to amortization (deformation) of the rubber bush of the roller, Δv=(0.05÷0.1) v is the value of the change in the linear speed of the chain due to the cyclic deformation of the rubber bush of the roller of the chain drive.

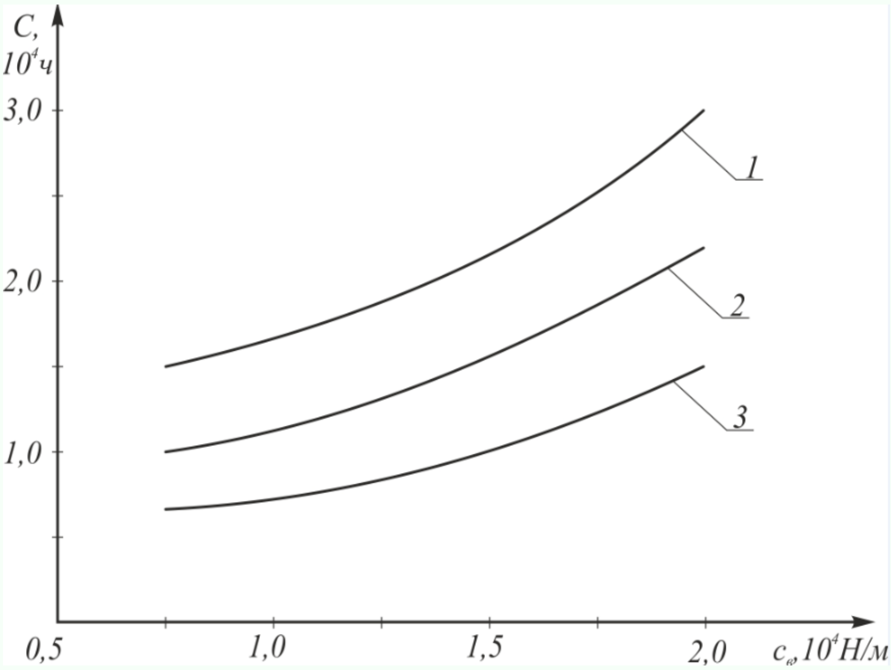

Fig.2. Dependences of the change in the estimated life of the chain on the change in the stiffness coefficient of the rubber roller sleeve

where, 1 - at d = 25N; 2 - at d = 35N; 3 - at d = 50N;

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

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

An example of calculating the service life of a chain drive, taking into account the rubber bush of a roller, with the following initial values of parameters for:

N=1,6 kBt; n_1=300 rev/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; $k_{ц}$=1,25; k_m=0,9÷0,1; h_u=0,05÷0,08; $c_{в}=1,4\cdot10^4$ N/m; d=(25÷40)N; $x_{в}=(0,2÷0,35)\cdot10^{-3}$ m; $d_{в}=5,6\cdot10^{-3}$ m; k_y=(0,3÷0,5); Δv=(0,05÷0,1)v.

Analysis of the Results

Based on the numerical solution of problem (7) for a chain drive with a composite roller of a chain with a rubber bushing, graphical 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 were plotted. An analysis of the dependencies shows that an increase in the stiffness coefficient of the rubber roller sleeve leads to an increase in the service life of the chain according to a nonlinear pattern. So, when the stiffness of the rubber bush changes from 0.78 · 104 N/m to 2.05·104 N/m, the service life of the chain increases from 1.05·104 h to 3.0·104 h with a load on the roller of 25 N. As the load on the roller increases, the life of the chain will decrease. So, with d = 50 N and an increase in c_w to 2.05·104N/m, the service life increases from 0.71·104 h to 1.51·104 h. Therefore, to increase the service life of the chain transmission with a composite roller of the chain with a rubber bushing to (2.5÷2.8)·104 h, it is considered advisable to choose the following values of the stiffness of the rubber sleeve c=(1.7÷2.1)·104 N/m at d≤(25÷30)N.

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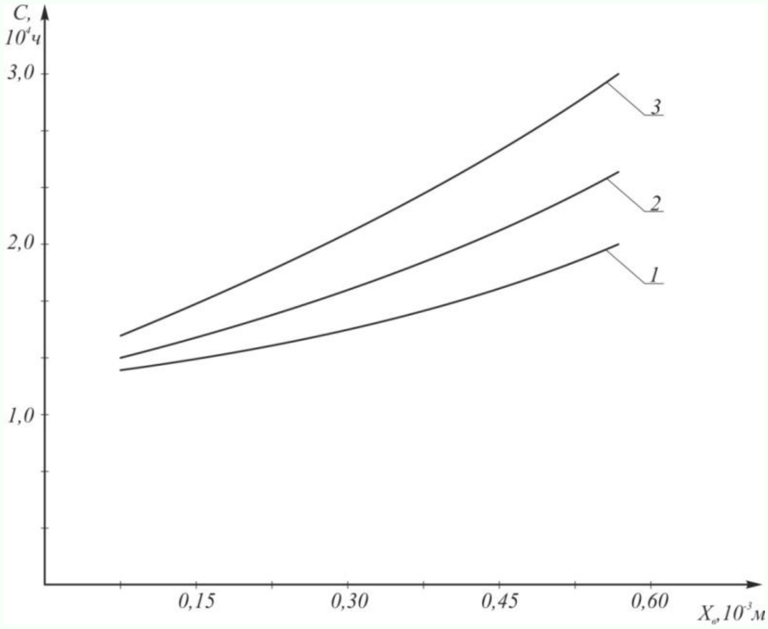

Fig.3. Graphical dependences of the change in the service life of the chain on the change in the deformation of the elastic sleeve of the roller of the transmission chain. 1 - at p = 10 MPa; 2 - at p = 12MPa; 3 - at p = 14 MPa.

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

Figure 3 shows the dependences of the change in the service life of the chain on the amount of deformation of the elastic sleeve of the chain roller. With an increase in the deformation value from $0.085 \cdot 10^{-3}$ m to $0.54 \cdot 10^{-3}$ m, the service life of the transmission increases from 1.52·104 h to $3.05 \cdot 10^{4}$ h with a value of p=10MPa. With an increase in *p* to 14 MPa leads to an increase in from $1.26 \cdot 10^{4}$ h to $1.92 \cdot 10^{4}$h. This is due to the fact that when the teeth of the sprockets mesh with the chain, the deformation of the rubber bush of the roller absorbs the loads. This significantly reduces the wear of rollers and sprocket teeth, and reduces transmission noise. To ensure the service life of the chain with a compound roller with a rubber sleeve up to $(2.5 \div 3.0) \cdot 10^{4}$ h, it is recommended not to use the rubber grade 1847, which allows its deformation within $(0.3 \div 0.6) \cdot 10^{-3}$ m.

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Conclusions

An efficient, resource-saving chain drive design with split rollers with an elastic chain element is recommended. Based on the analysis of existing methods for calculating the service life of a transmission chain, a formula is recommended for calculating the service life of a transmission with a compound roller having a rubber bushing. The parameters of the transmission are substantiated, allowing an increase in the service life of the transmission by 1.5 times.

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