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DESIGN OF A RESOURCE-EFFICIENT CHAIN DRIVE STRUCTURE FOR THE DEVICE DRIVE THAT DISTRIBUTES THE SEED IN THE BUNKER TO THE LINTERS

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Annotation. In the article, a new chain drive structure with elastic elements is designed for the seed distribution device drive. At the same time, the parameters of the proposed chain drive are defined.

Keywords. Chain, driver sprocket, driven sprocket, roller, bushing, elastic element, number of teeth, gear ratio, step, coefficient, drive.

Introduction. The creation of new types of resource-saving, high-efficiency transmissions and their continuous improvement are gaining urgent importance in the world for technological machinery drives. One of the important tasks is the development of chain transmissions that can work effectively in various technological processes, suppress noise and vibrations, and create methods for their calculation. Therefore, a lot of attention is being paid to the widespread use of chain mechanisms with elastic elements with low energy consumption and high productivity in the drives of various technological machines. The rapid development of mechanical engineering requires conducting scientific research aimed at creating resource-efficient types of chain drives with wide kinematic and dynamic capabilities. In this direction, especially in order to reduce friction, wear, noise and vibration in chain mechanisms, including the creation of constructions of chain drives with belt elements that adapt to the technological process used in the operation of cotton ginning machines, the development of substantiation and calculation methods of their technological work process. Conducting research is urgent. In our republic, special attention is paid to the rapid development of mechanical engineering, the creation of new types of resource-efficient machine and mechanism structures.

The main part. It is known from mechanical sciences that the set of mechanisms that adjust the necessary power and speed of movement from the source of mechanical energy to the working part of the machine is called the drive of the machine. However, if the shaft of the working part of the machine is directly connected with the shaft of the electric motor (for example, a fan, compressor, etc.), then the electric motor itself is considered as the driving mechanism of the machine. However, in addition to electric motors, car drives include various mechanical drives with several steps. The simplest such driving shafts are composed of a reducer and an electric motor connected by a mutual coupling. However, in most cases, there are various types of transmissions, except for the nodes indicated in the structure of the procedures used. It depends on the type of machines, the required movement speed and power of its working part. Today, every person with an engineering profession should have the ability to design transmissions in technological machinery drives. In such cases, the most important issue is the correct selection, placement and calculation of sprockets in the structure. A structural design was carried out for the drum drive that distributes the compound roller chain drive to the linters in the hopper. The kinematic scheme of the drum that distributes the seed in the bunker to the linters is presented in Fig. 1.

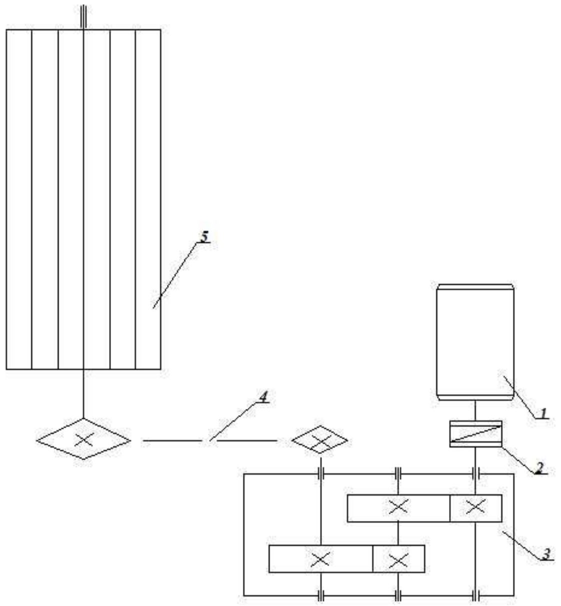

Figure 1. Kinematic scheme of the device that distributes the seed in the bunker to the linters

1-electric motor, 2-coupling, 3- two-stage cylindrical gear reducer, 4-chain drive, 5-seed distribution device

It is recommended to have a roller chain transmission with the content shown in Figure 2 as a chain drive of the machine shown in Figure 1, taking into account the loading characteristics. The proposed chain transmission works in the following order [1]: the drive from the leading sprocket 1 to the driven sprocket 2 is transmitted through the chain 3. Then, the drive sprocket 2 is transmitted to the output shaft 7 through the elastic bushing 5 and base 6. It is observed that the frictional force and other harmful forces generated between the chain 3 and the sprocket 2 are somewhat reduced when passing through the elastic element 5. When the chain 3 interacts with the driver 1 and the driven 2 sprokets, the feed of the bushing 13 and the sprocket is reduced due to the deformation of the elastic element 15 in the chain roller 12.

Due to the deformation of the elastic element 15, the friction between the bushing 11 and the roller 10 also decreases. This leads to an increase in the durability of the chain drive elements. During operation, it is recommended that the surface 16 of the rubber bushing 15 be concave in order to center the pressure forces acting on the roller 12 from the driver 1 and driven 2 sprockets. The elastic element in the roller 12 ensures even distribution of external pressure forces. This causes the service life of the chain to increase [2-7].

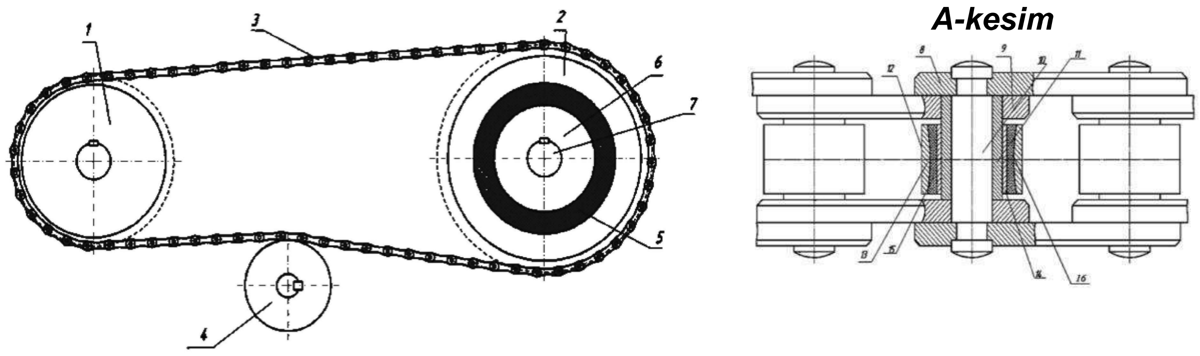

Figure 2. Containing roller chain drive

The values obtained from the calculation of the kinematic and dynamic parameters of the drive to calculate the chain drive are given below:

electric motor power: $N = 11kW$*; number of revolution per minute* $n_e = 1458\ rpm$*;*
the number of revolutions of the drive shaft of the chain drive: $n_1 = 72\ rpm$*;*
gear ratio of the chain drive: $u_z = 4$*;*
gear ratio of two-stage cylindrical gear reducer $u_r = 20{,}1$.

Based on the available parameters, we calculate the structural parameters of the chain drive of the device drive that distributes the seed in the bunker to the linters:

1. The number of teeth of the driver sprocket:

$$Z_1 = 31 - 2u_z \tag{1.1}$$

where, u_z – is the gear ratio of the chain drive.
Then, according to expression (1.1), the number of teeth of the driver sproket of the chain drive:

$$Z_1 = 31 - 2 \cdot 4 = 23.$$

2. The number of teeth of the driven sprocket is determined by the following expression:

$$Z_2 = Z_1 \cdot u_Z, \tag{1.2}$$

then the number of teeth of the chain drive sprocket is as follows:

$$Z_2 = 23 \cdot 4 = 92.$$

3. The step of the chain is determined by the following expression;

$$t \geq 2{,}8 \cdot \sqrt[3]{\frac{T_1 \cdot k_e}{Z_1 \cdot [P] \cdot m}} \tag{1.3}$$

where: T_1 – is the torque [N·mm] on the driver sprocket shaft, we determine it by the following expression

$$T_1 = T_d \cdot u_r \cdot \eta_r, \tag{1.4}$$

where: T_d – is the value of torque on the motor shaft, it is determined as follows:

$$T_d = \frac{N_{el}}{\omega_d} = \frac{11 \cdot 10^3 Vt}{\frac{\pi \cdot n_{el}}{30}} = \frac{11\ 000\ Vt}{152{,}6\ rad/sek} = 72\ N \cdot m = 72 \cdot 10^3 N \cdot mm,,$$

given in this expression ω_d – represents the angular velocity value of the motor shaft.
η_r –is the useful work coefficient of the two−stage cylindrical reducer, its value according to available data is $\eta_r = 0{,}84$.

Then the value of the torque on the drive shaft of the chain drive according to the expression (1.4) is equal to the value of the torque on the drive shaft of the chain drive:

$$T_1 = T_d \cdot u_r \cdot \eta_r = 72 \cdot 10^3 N \cdot mm \cdot 0{,}84 \cdot 20{,}1 = 1215{,}648 \cdot 10^3\ N \cdot mm.$$

K_e– is the coefficient of operation, [P] is the permissible value of the pressure acting on the chain joint, which is obtained based on the available tables, for the chain transmission joint of the technological machine that we are considering, $n_1 = 72$ rpm when [P]=20MPa is appropriate. In the case of m-chain row, it is single-row, but taking into account that it breaks quickly during the work process, we get the value of $m = 2$, i.e. double-row chain.

The coefficient of operation $k_e -$ is a coefficient that takes into account all external factors in the introduction of chain transmission, its value is determined by the following empirical expression:

$$k_e = k_1 \cdot k_2 \cdot k_3 \cdot k_4 \cdot k_5 \cdot k_6 \cdot k_7 \quad (1.5)$$

$k_1 -$ is a coefficient that takes into account the nature of the change in the value of the load on the drive shaft of the chain drive, taking into account the normality of the load of the working bodies of the drum that distributes the seed in the bunker to the linters, it is appropriate to take $k_1 = 1$;

$k_2 -$ is a coefficient that takes into account the distance between the shafts of the driver and driven sprockets of the chain drive (interaxial distance), and it is advisable to take its value $k_2 = 1{,}15$ in accordance with the existing transmission location scheme;

$k_3 -$ is the coefficient that takes into account the value of the angle θ between the direction of the line connecting the centers of the stars of the chain drive and the horizontal plane, taking into account the value of $\theta < 60^0$ for the case we are looking at and taking into account the presence of a tensioning device from time to time, the valuee $k_3 = 1{,}1$, it is desirable;

$k_4 -$ is a coefficient that takes into account the method of adjusting the tension of the chain drive chain, $k_4 = 1.23$ was adopted, taking into account that the tension of the chain transmission is adjusted from time to time;

$k_5 -$ is a coefficient that takes into account the lubrication method, and since the details of the chain drive are periodically lubricated in the dusty environment of vegetable oil production enterprises, we can get the value of $k_5 = 1{,}35$;

$k_6 -$ is the coefficient taking into account the working mode, so $k_6 = 1{,}26$ was accepted because the work is organized in two shifts on average in accordance with the seasonal production values of the researched enterprise.

$k_7 -$ it is a coefficient that takes into account the presence of a elastic element in the chain transmission roller, and it is desirable to take, $k_7 = 0.92$. It should be noted that the element of the roller elastic with the content significantly reduces the value of the external load and leads to a decrease in the value of the negative impact on other parts of the transmission.

According to the expression (1.5), the value of the exploitation coefficient k_e is as follows:

$$k_e = 1 \cdot 1{,}15 \cdot 1{,}1 \cdot 1{,}23 \cdot 1{,}35 \cdot 1{,}26 \cdot 0{,}92 = 2{,}435.$$

Based on the obtained results, it is possible to determine the pitch of the chain drive being designed by the expression (1.3):

$$t \geq 2{,}8 \cdot \sqrt[3]{\frac{1215{,}648 \cdot 10^3\ N \cdot mm \cdot 2{,}435}{23 \cdot 20 \cdot 2}} = 2{,}8 \cdot 14{,}76 = 41{,}33mm.$$

Based on the result, we choose the type of chain with the following parameters standardized by GOST:

- *Designation– 2R-44.45-344;*
- *28A-2 according to ISO 606;*
- *step t=44,45 mm;*
- *roller diameter $d_1 = 25{,}4\ mm$;*

- *internal plate spacing width* $b_{1min} = 25{,}4\ mm$;;
- *roller diameter* $d_2 = 12{,}7mm$;
- *roller length* $b_{7max} = 110\ mm$;
- *inner plate width* $h_{max} = 42{,}4\ mm$;
- *transverse step* $A = 48{,}87\ mm$;
- *roller support surface* $A_{tay} = 946\ mm^2$;
- *chain breaking load* $Q_{min} = 344{,}8\ kN$;
- *the weight of one meter of chain* $q = 14{,}4\ kg$.

According to the received chain pitch t, the number of revolution per minute of the driver sprocket is checked for compliance with the permissible number of revolution per minute. The number of revolution per minute allowed for the 2R-44.45-344 type chain is $[n_1] = 400\ rpm$ according to the available data. Considering $n_1 = 72rpm$, we can see that the condition $n_1 \leq [n_1]$ is fulfilled.

The linear speed of the chain is determined by the following expression:

$$v = \frac{Z_1 \cdot t \cdot n_1}{60000}, \tag{1.6}$$

based on the expression chain speed is equal to:

$$v = \frac{23 \cdot 44.5mm \cdot 72}{60000} = 1{,}227\frac{m}{sek}.$$

The value of the rotational force on the drive shaft of the chain drive was determined as follows:

$$F_t = \frac{N_2 \cdot 10^3}{v}, \tag{1.7}$$

where N_2 – is the power on the drive shaft of the chain drive, which was determined by the following expression:

$$N_2 = N_1 \cdot \eta_r, \tag{1.8}$$

we determine the calculated value of the power on the drive shaft of the chain drive from the expression:

$$N_2 = 11 \cdot 0{,}84 = 9{,}24\ kVt.$$

(1.7) we determine the value of the rotational force on the drive shaft of the chain drive, putting the values obtained in the expression:

$$F_t = \frac{9{,}24kVt \cdot 10^3}{1{,}227\ m/sek} = 7530{,}56N.$$

The value of the pressure applied to the chain hinge is checked according to the obtained chain parameters using the following expression:

$$P = \frac{F_t \cdot k_e}{A_{tay}} \leq [P], \tag{1.9}$$

according to the expression,

$$P = \frac{7530{,}56 \cdot 2{,}435}{946mm^2} = 19{,}4MPa \leq [32\ MPa] \text{ the condition is met .}$$

The number of chain links is determined as follows:

$$a_t = \frac{a}{t} = \frac{(30 \div 50)t}{t} \tag{1.10}$$

Taking into account the location of the device that distributes the seed in the bunker to the linters, it is desirable to take a value of $a_t = 50$.

We determine the sum of the number of sprocket teeth as follows:

$$z_{\sum} = z_1 + z_2 = 23 + 92 = 115.$$

To simplify calculations, we introduce the following notation:

$$\Delta = \frac{z_2 - z_1}{2\pi} = \frac{92 - 23}{2 \cdot 3{,}14} = 10{,}99,$$

We determine the number of links in the chain:

$$L_t = 2a_t + 0{,}5z_{\Sigma} \quad + \frac{\Delta^2}{a_t}, \tag{1.11}$$

According to the expression, $L_t = 2 \cdot 50 + 0.5 \cdot 115 + \frac{10{,}99^2}{50} \approx 160.$

We find the exact value of the distance between the axes by the following expression:

$$a = 0{,}25t[L_t - 0{,}5z_{\Sigma} \quad + \sqrt{(L_t - 0{,}5z_{\Sigma} \quad)^2 - 8\Delta^2}, \tag{1.12}$$

The value of the distance between the axes was determined by putting the existing values into the expression:

$$a = 0{,}25 \cdot 50[160 - 57{,}5 + \sqrt{10506{,}25 - 966.2408} = 2502{,}162mm.$$

We determine the diameters of the dividing circle of the driver and driven sprockets:

driver,

$$d_{D_1} = \frac{t}{sin\frac{180^{\circ}}{z_1}} = \frac{44{,}45\ mm}{\sin{(7{,}826)}^{\circ}} = 326{,}36mm,$$

driven, $$d_{D_2} = \frac{t}{sin\frac{180^{\circ}}{z_2}} = \frac{44{,}45\ mm}{\sin{(1{,}957)}^{\circ}} = 1303{,}52mm.$$

We determine the outer diameters of the driver and driven stars:

driver,

$$D_1 = t\left(ctg\frac{180}{z_1} + 0.7\right) - 0.31d_1 = 346{,}643\ mm,$$

driven,

$$D_2 = t\left(ctg\frac{180}{z_2} + 0.7\right) - 0.31d_1 = 1324{,}413\ mm.$$

We determine the forces acting on the chain:

rotational force: $F_t = 7530{,}56N$;

centrifugal force: $F_v = q \cdot v^2 = 14{,}4kg \cdot 1{,}227^2 = 21{,}68\ N$;

the force due to chain tension:

$F_f = 9{,}81 \cdot k_f \cdot q \cdot a = 9{,}81 \cdot 1{,}4 \cdot 14{,}4 \cdot 2{,}502162 = 494{,}851\ N,$

here k_f – is the coefficient that takes into account the location of the chain, and we get the value $k_f = 1{,}4$ taking into account that the chain is located obliquely in the process we are considering.

We calculate the value of the load acting on the shaft:

$F_v = F_t + 2F_f = 7530{,}56N + 989{,}702N = 8520{,}262N$

The strength reserve coefficient of the selected chain drive is determined by the following expression:

$$s = \frac{Q}{F_t \cdot k_1 + F_v + F_f} = \frac{344{,}8 \cdot 10^3}{7530{,}56N \cdot 1 + 8520{,}262 + 494{,}851} = 20{,}84$$

We consider the condition that the determined strength reserve coefficient is greater than the determined reserve normative coefficient $s \geq [s]$ for the selected chain:

$$s = 20{,}84 \geq [s] = 8{,}1.$$

Summary. From the results of the above project, it can be concluded that the proposed chain drive structure can reduce the wear indicators of the chain elements and star teeth, and it can be seen that the value of the reserve coefficient $[s]$ has increased significantly.

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ANALYSIS OF EXISTING METHODS AND APPROACHES TO THE ASSESSMENT OF RESIDUAL RESOURCES OF TRACTION ROLLING STOCK

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Abstract:

Objective. The article presents an analysis of existing methods and approaches to assessing the residual life of elements of load-bearing structures of traction rolling stock. In this work, methods for assessing reliability indicators and reliability during the use of torsion action content are presented. The analysis of the characteristics characterizing the reliability of technical objects is presented. To confirm the validity of the forces acting on the motion, algebraic formulas are given.

Methods. Based on existing methods and approaches, statistical, quasi-static, dynamic, variable, multiple and low-cycle effects of loads were analyzed in detail. The Weibull equation, the Mazing principle.

Results. Structures of railway traction rolling stock during operation take on loads (static, quasi-static, dynamic, variable, multiple and low-cycle) that affect the design of the support at different levels and periods of use. At the same time, it causes both elastic and plastic deformations.

Conclusion. At the same time, it is necessary to achieve a uniform distribution of deformations and loads in the structure as a result of a gradual change and displacement of the mechanical properties of the

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