Calculation of the chain drum with elastic fingers of potato harvesting machines

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Abstract. In agricultural machines, chain transmission is widely used, the main advantage of which is a high load capacity. Therefore, in this article, the methodology for calculating chain drives in mechanical drives of agricultural machinery, namely in mechanical drives of potato harvesters, is presented. As a result of experimental and theoretical studies, the availability and safety margin of the selected chain drive of a loosening drum with elastic fingers of a potato harvesting machine was confirmed. In potato-harvesting machines, the movement of the executive working bodies is carried out. Analysis of the design of domestic and foreign potato harvesters showed that it is possible to obtain a good quality of potato harvesting in difficult soil and climatic conditions due to the use of machines of a simpler design with a mesh number of working bodies, but at the same time with their rational layout, the correct choice of transmission drives and ensured optimal modes work. The advantage of these transmissions is their large load capacity and the ability to work under dynamic loads. Therefore, in these investigations, the works are analyzed and the theoretical foundations and methods for calculating the working capacity of the wearable chain and the margin of strength of the chain transmission for dynamic loads are presented. As a result of the experimental theoretical research, the reliability of the working capacity and the strength reserve of the selected chain drive of the drum with elastic pulleys of the potato harvester is supported.

1. Introduction

Currently, the following types of transmissions are used in agricultural engineering: mechanical, electrical, hydraulic and pneumatic. Among them, mechanical transmissions are the most widespread.

Mechanical transmissions can be divided into the following types:
- gear drives (cylindrical, bevel);
- screw drives (screw, worm, hypoid);
- flexible transmissions (belt, chain);
- frictional transmission (due to frictional forces).

By the method of transferring the motion from the engine to the executive transmission, the authority can be divided into:
- Transmission of motion from the shaft to the shaft, which is carried out due to friction forces (belt, friction, worm gear).
- Propulsion transmissions from the shaft to the shaft which are carried out by means of engagement (chain, toothed, screw, worm, with toothed belts).
Chain gears are widely used in agricultural machines, the main advantage of which is the high load capacity and ability to operate under overloads, i.e. dynamic loads. Therefore, in this work, there is the analysis of the work and the calculation of chain gears in the power drives of agricultural machines.

2. Materials and methods

An analysis of the operation of open chain drives in mechanical drives of agricultural machines shows that transmissions by the roller and bushing chains have been studied quite a lot [1–5] and experimental data on the wear of their working elements are presented in [1–3].

In work [3, 4] it is noted that the main disadvantages of chain drives are uneven wear and entrainment of chain steps is along the links. In addition, it is argued that choosing an even number of teeth will result in increased chain wear through one tooth. In the work [4, 5] a chain transfer star is represented as a polygon whose number of sides equals the number of teeth of a sprocket. On this basis it can be assumed that for each rotation of a sprocket the roller chain moves by the length of the perimeter of the polygon. Therefore, the average speed of a \( V_{av} \), m/s chain to an asterisk can be determined by the formula [9]:

\[
V_{av} = \frac{z \cdot c \cdot n}{60},
\]  

where: \( V_{av} \) - average running speed, m/s; \( z \) - number of sprocket teeth; \( c \) - chain pitch, m; \( n \) - chain speed per minute.

The actual speed of the chain is not constant and changes periodically. The duration of the period from the engagement of the chain drive is determined by the formula [9]:

\[
T = \frac{60}{z \cdot n}.
\]  

Figure 1 shows the elements of the chain transmission, including the chain and the drive sprocket, the speed of lifting the chain up \( V_1 \) and the speed of running \( V_2 \) of the chain onto the sprocket [4].

![Figure 1. Layout of the chain on the drive sprocket when engaging](image)

The values of the component speeds are determined by the expressions: the speed of lifting the chain up:

\[
V_1 = \frac{z \cdot c \cdot n}{60} \cdot \sin \beta.
\]  

The chain speed on the sprocket:

\[
V_2 = \frac{z \cdot c \cdot n}{60} \cdot \cos \beta.
\]

Accordingly, the average speed of the hinge \( A \) is determined by the expression:

\[
V_A = \sqrt{V_1^2 + V_2^2}.
\]  

The greatest influence on the reliable operation of the chain is the change in the rate of the chain rise, which is characterized by the uneven movement of the chain.

The factors that affect the durability and thus the reliability of the chain are very diverse and numerous. Let us consider the effect of each factor on the wear of chain drive components. The main design characteristic determining the carrying capacity of the chain is the projection of the bearing surface of the hinge [5], which is determined according to the equation:
\[ F_{sh} = d_r \cdot l_h, \]  

(6)

where \( d_r \) – diameter of rollers, mm; \( l_h \) – chain hinge length, mm.

Another important design parameter of the chain, which significantly affects its wear, is the pressure arising in the hinge and determined by the formula [4]:

\[ p = \frac{s}{F_{sh}} \leq [p]. \]  

(7)

From formula (7) it follows that the pressure arising in the hinge depends on the load transmitted by the chain (tension of the leading branch) and the projection of the bearing surface of the hinge [5, 6]. Obviously, in order to decrease the pressure, it is necessary to increase the denominator of formula seven. Since the dimensions of the transmission limit the length of the chain hinge \( l_h \), the diameter of the chain roller \( d_r \) should be increased. Increasing the diameter of the chain pins without changing the pitch group will increase the carrying capacity and durability of the chain transmission. Another factor affecting the durability of chain drives is the accuracy of their installation. However, studies [1, 6] have established that during the operation of the chain transmission, the pressure can exceed the allowable one and subject to the accuracy standards for installation, in connection with the real operating conditions. The analysis of all the listed factors affecting the durability of chain drives shows that the safety margin of the chain drive is the most significant.

The calculation of the chain drive of a drum with elastic fingers for working capacity and safety factor is performed according to the results of experimental studies obtained in [7] according to the method of G.B. Iosilevich [8], taking these calculations as verification. Let us take the following as initial data:

1. The angle of the inclination of the chain transmission branch to the horizon is up to 40º.
2. Power transmitted by the chain:

\[ P = \frac{\tau_1}{\omega_1} = \frac{16.8 \cdot 29.3}{1000} \approx 0.492 \text{ kwt} \]

where:

\( \tau_1 \) – average value of the torque on the drum shaft with elastic pins; \( \omega_1 = 29.3 \text{ s}^{-1} \) - frequency of rotation of the first shaft of the elevator according to the kinematic diagrams [7].

3. The frequency of rotation of the driving gear is equal to \( n = 120 \text{ rpm} \) [7].
4. The relative increase in the length of the chain pitch is 0.003 mm.
5. The roller chain transmission with the following parameters:

- pitch of the chain is equal to \( t = 19.05 \text{ mm} \);
- area of the bearing surface of the hinge is equal to \( F_{sh} = 71 \text{ mm}^2 \);
- minimum breaking load is \( Q_{\text{min}} = 3180 \text{ N} \);
- linear mass of the chain is \( q = 1.9 \text{ kg/m} \).

The basis for calculating the performance of the wearing chain according to the recommendation [8] is based on the assumption that the chain has sufficient wear resistance if the \( P_A \) pressure in the hinge does not exceed the permissible value \([P_{all}]\):

\[ P_A \cdot K_0 \cdot F_{sh} \cdot K_{\text{row}} < [P_{all}] \]

where: \( P \) - useful circumferential load transmitted by the chain, N; \( K_0 \) - operating factor, taking into account the operating conditions of the transmission and its design, \( F_{sh} \) - surface of the hinge; \( K_{\text{row}} \) - coefficient taking into account the number of rows of the chain; \([P_A] = 2.83 \text{ kg/mm}^2 \) - admissible value of pressure in the joint according to tab. 5 [8].

3. Results

The theoretical calculations of the chain drive are carried out in the following order.

1. Let us determine the chain speed by the formula:

\[ v_{ch} = \frac{z_1 \cdot n_1 \cdot t}{60 \cdot 1000} = \frac{40 \cdot 120 \cdot 19.5}{60 \cdot 1000} = 1.5 \text{ m/s}. \]

2. Let us determine the useful circumferential load by the formula:

\[ P_{\text{max}} = 102 \cdot \frac{W}{v_{ch}} = 102 \cdot \frac{0.22}{1.5} = 14.96 \text{ kg} \cdot \text{s} \]
In chain drives operating with shock loads, typical of a drum chain drive with elastic pins, it is recommended to take the minimum number of teeth of the drive gear \( z_{\text{min}} = 40 \) [7]. Therefore, in the soil digger, the number of teeth of the sprocket of the drive of the first shaft of the elevator is recommended to be made \( Z_1 = 40 \) in place \( Z_1 = 22 \).

Comparing \( P_{\text{max}} \) with the minimum breaking load for this type of chain, we see that \( P_{\text{max}} < Q_{\text{min}} \). Therefore, it can be noted that the static load capacity of the chain is ensured. Since the speed of the chain transmission is less than 2 m/s i.e. \( v_{ch} < 2 \text{ m/s} \), then the chain transmission of the drum shaft with elastic fingers can be classified as a low-speed transmission.

2. According to table 4 [9], let us take the following values of the operating factors. The coefficient of dynamism of loading with jerks is \( K_d = 1.5 \). The coefficient of the influence of the chain length on wear is \( K = 1.25 \), since the center-to-center distance is \( d = 340 \text{ mm} < 25t \). The gear arrangement ratio is \( K_a = 1.1 \), since the inclination of the sprocket center line to the horizontal is \( < 40^\circ \). The gear mounting ratio is \( K_{\text{rat}} = 1.15 \) since there is a tensioning sprocket. The lubrication coefficient is \( K_{\text{lub}} = 1.5 \) since the chain drive is periodically lubricated. The coefficient of the operating mode is \( K_{\text{dig}} = 1 \), since the digger works during the day.

Under these conditions, the operating factor is:
\[
K_9 = K_d \cdot K_a \cdot K_{\text{rat}} \cdot K_{\text{lub}} \cdot K_{\text{dig}} = 1.5 \cdot 1.125 \cdot 1.1 \cdot 1.15 \cdot 1.5 \cdot 1 = 3.23.
\]

4. Let us calculate the pressure in the chain joint:
\[
P_{\text{A}} = \frac{P_{\text{A}} \cdot K_{\sigma}}{F_{\text{ch}} \cdot K_{\text{m}}} = \frac{1.96 \cdot 3.23}{71.1} = 0.068 \text{ kgs/(mm)}^2.
\]

The resulting design pressure in the chain joint is less than the allowable \( 0.068 < 1.5 \text{ kgs/(mm)}^2 \). Therefore, the selected chain satisfies the wear resistance condition - this means that the chain is operational.

To calculate the safety margin of the chain drive, we first determine the amplitude of the stress variables according to the well-known formula:
\[
\sigma = \frac{P_{\text{max}}}{4 \cdot S \cdot (h - d)} = \frac{31.8}{4 \cdot 2 \cdot (14.8 - 5.08)} = 0.41 \text{ kgs/(mm)}^2
\]

where: \( S = 2 \text{ mm} \) - plate thickness according to table 2 [9]; \( H = 14.8 \text{ mm} \) - plate width; \( d = 5.08 \text{ mm} \) - bushing diameter.

Then, based on this, we find the calculated value of the safety factor for alternating stresses:
\[
n_{\sigma}^c = \frac{\sigma_1}{\sigma_p \cdot K_{\sigma}} = \frac{20}{0.82} = 24.4
\]

where: \( \sigma_1 \) - endurance limit of the chain material equal to 20 kgs/(mm)\(^2\); \( K_{\sigma} \) - effective stress concentration factor, which is determined by the formula:
\[
K_{\sigma} = 1 + q(a_0 - 1).
\]

Here: \( q = 0.4 \ldots 0.6 \) - coefficient characterizing the sensitivity of a material to stress concentration; \( a_0 = 2.8 \ldots 3 \) - theoretical stress concentration factor, taking into account the fit of the plates on the bushings and the bending of the plates. Then:
\[
K_{\sigma} = 1 + q(a_0 - 1) = 1 + 0.5 (3 - 1) = 2.
\]

The recommended safety factor for the chain should be \( [n_\sigma] > 3 \). Therefore, comparing the calculation results, we make sure that \( [n_{\sigma}^c] > [n_\sigma] \). This confirms that the selected chain is able to transmit torque on the drum shaft with elastic pins.

4. Conclusions
The results of the calculations made it possible to note the following. The performance of the wearing chain (IP1-19.05-31.8) and the safety margin of the chain transmission according to the calculations are provided. This is because the design pressure in the chain hinge \( [P_{\text{A}}]_{\text{all}} = 1.4 \text{ kgs/(mm)}^2 \) is less than the allowable \( [P_{\text{A}}]_{\text{all}} = 2850 \text{ N} \) and the calculated value of the safety factor of the chain for alternating stresses \( n_{\sigma}^c = 25 \) is
more than 8 times higher than its recommended value $[n_b]_{all}>3$.

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