Kinematics of the transmission gear mechanism for roller machines with different diameters of drive shaft

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Abstract: The article describes the arrangement, principle of operation and kinematic analysis of a ten-link gear-lever differential transmission mechanism with a parallelogram lever contour for roller machines with different diameters of the working shafts. The purpose of the kinematic analysis of the transmission mechanism is to determine the velocities and accelerations of the mechanism links, the gear ratio of the mechanism, and to show the constancy of the gear ratio of the transmission mechanism when the center distance of the driving and driven tooth gears changes. An important condition for the normal operation of roller machines with different diameters of the working shafts is the coincidence of the gear ratio of the working shafts and the gear ratio of the transmission mechanism between these shafts. Initially, the kinematics of the mechanism was performed using the analytical method. In the analytical study of the mechanism, the centroid method was used. Dependencies for determining the kinematic parameters of the ten-link gear-lever transmission mechanism are derived. The advantages and disadvantages of this method are shown. Further, to compare the results of the study, the kinematics of the mechanism is performed by the graphic-analytical method. The method developed by the authors was used in the graphic-analytical study.

1. Introduction
In roller machines (squeezing-washing, staking, corrugated crimping machines - for primary processing of bast crops, and squeezing, staking-softening machines - for processing leather and leather semi-finished products) the center distance of the working shafts changes during operation [1-3]. Normal operation of corrugated shafts in pairs takes place when the groove of one shaft is located in the center of the cavity of the other, which is possible only with the synchronous rotation of the corrugated shafts regardless of the change in their center distance [1]. Drives and transmission mechanisms between the working shafts, currently used in some roller machines, while changing their center-to-center distances, do not ensure the equality of the linear velocities of the surfaces of the working shafts [4]. In order to eliminate the above disadvantage in the transmission mechanisms of existing two-roll machines with unequal diameters of the working shafts, we have developed a four-wheel gear-lever differential transmission mechanism (GLDTM), which provides an equal linear velocity of the surfaces of these working shafts, regardless of the changes in their center distance, i.e. e., providing the synchronous rotation of the working shafts regardless of the changes in their center distance and the difference in the diameters of the working shafts [5]. One of the most promising mechanisms for creating modern machines and devices is a gear-lever mechanism [6-8]. Therefore, we have invented and investigated one of the types of differential gear-lever transmission mechanisms.
2. The arrangement and principle of operation of the roller machine with GLDTM

Figure 1 shows a structural-kinematic diagram of a roller machine with such transmission mechanism. The machine consists of two working shafts 1 and 2.

![Figure 1. Structural and kinematic diagram of a roller machine with a gear-lever differential transmission mechanism. 1, 2-working shafts; 3-rack; 4-drive gear; 5-driven gear; 6, 7, 12, 13-levers; 8, 9-axles of intermediary gears; 10, 11-intermediary gears.](image)

The upper working shaft has the ability to rotate around its own axis and move vertically along a line passing along the axis of two working shafts, and the lower working shaft is mounted on rack 3 and has the ability to rotate around its own axis. At the output ends of working shafts 1 and 2, gears 4 and 5 are rigidly fixed and levers 6 and 7 are hinged, which are the supports for axles 8 and 9. Intermediary gears 10 and 11 are freely installed on axles 8 and 9. Axles 8 and 9 are pivotally connected to lever 12. Axle (9) of the intermediary wheel (11), the nearest to the movable working shaft (2), is kinematically connected to the rack (3) by means of lever 13. Levers 6, 12 and 13 form a lever parallelogram \( O_1O_2O_3O_4 \).

3. Analytical kinematics of the GLDTM

The kinematic analysis of the mechanism was performed to determine the angular and linear velocities and accelerations of the mechanism links depending on the angular velocity of the driving link (gear 1) and the linear velocity and acceleration of the center of rotation of the free working shaft (2), and to prove the equality of linear velocities of the surfaces of the working shafts at the point of contact of these shafts with the processed material. Kinematic analysis was performed according to the centroid theory by the method of the instantaneous center of rotation of the links. Let the diameters of the working shafts be \( D_1 \) and \( D_2 \) (figure 2), the pitch diameters of the gears - \( d_1,d_2,d_3,d_4 \), and the working shaft 1 rotates at angular velocity \( \omega_1 \). The free working shaft moves at velocity \( v_{O_2} \).

Considering that in most roller machines the diameters of the working shafts are unequal to each other, as a special case, the transmission mechanism for such a machine. Consequently,
The linear velocities of the surface of the roller $V_{b_1}$ and $V_{b_4}$ to be equal to each other at $D_1 \neq D_4$:

$$D_1 \neq D_4$$

or

$$d_1 \neq d_2 \neq d_3 \neq d_4$$

or

$$\frac{D_1}{D_4} = \frac{d_1}{d_4} = \frac{d_3}{d_2}$$

where $d_1, d_2, d_3, d_4$ are the pitch diameters of gear wheels.

At a constant center distance of the working shafts $(O_1O_2 = \text{const})$, i.e. in the absence of a linear velocity of the axles of the free working shaft $(\overline{V}_{O_1} = 0)$, the velocity of the centers of rotation of the intermediary gears is also zero $(\overline{V}_{O_1} = 0, \overline{V}_{O_2} = 0)$; the compliance with conditions 2, 3 or 4 ensures the equality of the linear velocities of the surfaces of the working shafts $(\overline{V}_{b_1} = \overline{V}_{b_4})$.

It is necessary to prove that $\overline{V}_{b_1} = \overline{V}_{b_4}$ when the center distance of the working shafts changes, that is, in the presence of velocity $\overline{V}_{O_2}$.

So,

$$\overline{V}_{b_1} = \omega_1 \cdot R_1,$$

$$\overline{V}_{b_4} = \omega_4 \cdot R_4,$$

then, taking into account equality (1), and $D = 2R$, it suffices to prove that $\omega_1 = \omega_4$.

Let consider the lever contours of the gear-lever mechanism. From the rocker-slider contour, the velocity of point $O_2$ $(\overline{V}_{O_2})$ can be determined depending on the velocity of point $O_1$ $(\overline{V}_{O_1})$:

$$\overline{V}_{O_2} = \overline{V}_{O_1} + \overline{V}_{O_2O_1},$$

$$\overline{V}_{O_2} = \frac{\overline{V}_{O_1}}{2 \sin \alpha}$$

Given that the lever contour $O_1O_2O_3$ is a parallelogram, we can write

$$\overline{V}_{O_1} = \overline{V}_{O_3}.$$

Velocity $(\overline{V}_a)$ of the contact point $(a)$ of gears 1 and 2 can be determined by the following equation

$$\overline{V}_a = \omega_1 \cdot R_1.$$
where $r_i$ is the pitch radius of gear wheel $i$.

Knowing the velocities of two points of a rigid body, we can determine its instantaneous center of rotation [6]. Knowing the velocities of two points $O_1$ and $a$ of gear wheel 2, we determine the instantaneous center of rotation $(P_1)$ of this gear wheel; and the angular velocity of this wheel is determined by the following equation:

$$\omega_1 = \frac{\overrightarrow{O_1 \alpha}}{P_1 O_1}.$$  \hspace{1cm} (11)

Connecting the point of contact $(b)$ of gears 2 and 3 with point $P_1$, we can determine the velocity of point $b$ by the following equation

$$\overrightarrow{V_b} = \omega_1 \cdot bP_1.$$  \hspace{1cm} (12)

Knowing the velocities of two points of wheel 3 $(\overrightarrow{V_b}$ and $\overrightarrow{O_1})$, we determine the instantaneous center of rotation $(P_2)$ of this wheel; its angular velocity $(\omega_2)$ is determined by the equation

$$\omega_2 = \frac{\overrightarrow{V_b}}{P_2}.$$  \hspace{1cm} (13)

Connecting the point of contact $(c)$ of gears 3 and 4 with point $P_2$, we determine the velocity of point $(c)$ by

$$\overrightarrow{V_c} = \omega_2 \cdot cP_2.$$  \hspace{1cm} (14)

Knowing the velocities of two points of wheel 4 $(\overrightarrow{V_c}$ and $\overrightarrow{O_1})$, we determine the instantaneous center of rotation of this wheel $(P_3)$; and its angular velocity $(\omega_4)$ is determined by the following equation

$$\omega_4 = \frac{\overrightarrow{V_c}}{P_3}.$$  \hspace{1cm} (15)

Figure 2. Design diagram of the kinematics of a roller machine with a gear-lever differential transmission mechanism. 1-drive gear; 2, 3-parasitic gears; 4-driven gear.
\[ \omega_e = \frac{V_c}{cP_j}. \] (15)

Taking into account equations (11), (12), (13) and (14), equation (15) takes the form

\[ \omega_e = \frac{V_{o_f} \cdot bP_j \cdot cP_j}{P_jO_j \cdot bP_j \cdot cP_j}. \] (16)

The following relation can be written for wheel 2

\[ \omega_2 = \frac{V_a}{P_jO_j + r_2} = \frac{V_{o_f}}{P_jO_j}. \] (17)

where \( r_2 \) is the pitch radius of the gear wheel 2.

From relation (17) we determine \( V_{o_f} \)

\[ V_{o_f} = \frac{V_a \cdot P_jO_j}{P_jO_j + r_2}. \] (18)

Using (10), equation (18) can be written in the following form

\[ V_{o_f} = \omega_2 \cdot \frac{r_1 \cdot P_jO_j}{P_jO_j + r_2}. \] (19)

Using (19), equation (16) takes the following form

\[ \omega_e = \omega_2 \cdot \frac{r_1 \cdot bP_j \cdot cP_j}{(P_jO_j + r_2) \cdot bP_2 \cdot cP_j}. \] (20)

In equation (20), the values of \( P_jO_j, bP_j, bP_2, cP_2, cP_j \) are determined by solving the triangles \( cO_jP_2, O_jP_j, b, P_jO_j,b \). Using the derived equations ((8) - (20)) to determine the angular and linear velocities of the characteristic points of all links, and taking into account the relation of \( V_{o_f} \) with angle \( \alpha \), we determine the angular and linear accelerations of the characteristic points of all links.

Let us proceed to the proof of \( V_{b_4} = V_{b_4} \), from the similarity of triangles \( O_4P_2 \) and \( O_2P_j \) we can write

\[ \frac{r_2}{r_3} = \frac{bP_j}{bP_2}. \] (21)

With equations (12) and (21), equation (13) can be written in the following form

\[ bP_2 = \frac{bP_j \cdot r_3}{r_2}. \] (22)
$P_jO_j$ is determined from equation (18)

$$P_jO_j = \frac{\overline{V}_{o_j} \cdot r_j}{\overline{V}_{o_j} - V_{o_j}}. \quad (23)$$

From triangle $cO_2P_2$, by the cosine theorem, we write

$$cP_2 = \sqrt{r_j^2 + (O_jP_j)^2 + 2 \cdot r_j (O_jP_j) \cos 2\alpha} \quad (24)$$

Consider similar triangles $P_jO_jc$ and $O_2cn$.

From the similarity of triangles, we can write the following relation

$$\frac{r_j}{r_4} = \frac{cn}{cP_3}, \quad (25)$$

Hence

$$cP_3 = \frac{r_4(cn)}{r_4}. \quad (26)$$

From triangles $O_jcP_2, nO_2P_2, nO_2c$ we determine $nO_2$

$$nO_2 = \sqrt{r_j \cdot O_jP_j \cdot (r_j + O_jP_j)^2 - (cP_3)^2} \over O_jP_j + r_j. \quad (27)$$

Applying the cosine theorem to triangle $O_jcn$ we determine $cn$

$$cn = \sqrt{r_j^2 + (nO_2)^2 - 2r_j(nO_2) \sin \alpha}, \quad (28)$$

With (27), equation (28) takes the following form

$$cn = \frac{r_j}{r_j + O_jP_j} cP_2. \quad (29)$$

Substituting equation (29) into equation (26), we obtain

$$cP_3 = \frac{r_4 cP_2}{r_4 + O_jP_j}. \quad (30)$$

With (1), (2), (21), (24) and (30), equation (20) takes the following form

$$\overline{V}_{b_i} = \omega_i \frac{r_i \cdot bP_i \cdot cP_2}{(P_jO_j + r_2) \cdot bP_2 \cdot cP_3}. \quad (31)$$
Consequently,

$$\bar{V}_{b_i} = \bar{V}_{b_j} \quad (32)$$

The gear-lever differential transmission mechanism considered makes it possible to provide similar linear velocities of the contact surfaces of the working shafts, regardless of the change in their center-to-center distance. The results of the analysis and the derived equations make it possible to investigate the dynamics and to perform the geometric and dynamic synthesis of this transmission mechanism.

4. Conclusions
The gear-lever differential gear mechanism considered in the article, with the correct choice of the scheme of the two-roll module and the geometric parameters of the intermediary driving and driven gears, allows us to provide the same linear velocities of the contact surfaces of the working shafts and the processed material, regardless of the change in their center distance and the difference in the diameters of the working shafts. The results of the analysis and the equations derived make it possible to study the dynamics and to perform the geometric and dynamic synthesis of this transmission mechanism.

5. References
[1] Smirnov B.I., Kuznetsov G.K. 1967 Design of machines for primary processing of bast fibers. M.: Mechanical Engineering 270.
[2] Burmistrov A.G. 2006 Machine and apparatus for the production of leather and fur. M.: Kolos.
[3] Bahadirov G. 2010 Mechanics of a squeezing roller pair. Tashkent: "Fan". 156 p.
[4] Amanov T.Yu., Abdukarimov A., Nabiev A.M. 2007 Kinematic study of the process of dehydration of wet materials J. Problems of mechanics 6 43-45.
[5] Invention certificate 1632047 A 1 USSR, MKI S14 B 1/00, 1/02. Transfer mechanism of roller machines / A. Abdukarimov, T.Yu. Amanov, G.A. Bahadirov. No. 4621476/12; Appl. 19.12.88. Reg. 01.02.90.
[6] Frolov K.V., Popov S.A., Musatov A.K. et al. 2001 Theory of mechanisms and mechanics of machines M.: Higher school 496.
[7] Volmer, J., 1969 Getriebetechnik. Lehrbuch. VEB-Verlag “Technic”. Berlin.
[8] Levitsky, N.I., et al. 1974 Theory and application of gear-lever mechanisms. M.: Nauka 135-136.