GRAPHIC-ANALYTICAL STUDY OF A TOOTH-LEVER DIFFERENTIAL TRANSMISSION MECHANISM OF DIFFERENT DIAMETERS

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Abstract. *This article provides an overview and analysis of modern scientific research in the field of technological machines. The results of the graphic-analytical study of the tooth-lever differential transmission mechanism for roller technological machines with variable Inter-axis distance and different diameters of working shafts are presented. Geometric conditions have been developed that prevent geometric displacement between the working shafts and the material to be processed. Tables of calculations performed and graphs of variation of the linear velocity of the gear-to-wheel joints of the mechanism are given. When the specified geometric condition is met, it is shown that the gear ratio of the mechanism does not change regardless of the change in the inter-axis distance of the working shafts.*

Keywords: *roller machines, transmission mechanism, differential, tooth wheel, lever, line speed, corner speed.*

Introduction

Roller technology machines for various purposes are widely used in the industries of many countries. Roller machines are among the main technological machines in Mechanical Engineering, Agriculture, metallurgy, construction, textile, chemical, rubber, cellulose-paper, polygraph, food, light and leather industries. The main advantages of roller technological machines are simplicity of design, their compatibility with production lines, the possibility of continuing the technological process and high productivity. The technological process of processing materials on roller machines consists mainly in transferring the material to be processed between the working rollers. Roller technology machines include control system, transporter, transmission mechanism between working shafts, pressure-generating mechanism on recycled material, feed mechanism of recycled material, basic block control systems and other processes [1]. The main requirement for existing and newly created technological machines, including roller machines and their mechanisms, is to ensure that technological requirements are met with maximum accuracy, which improves product quality, increases machine productivity and saves resources [11, 12]. Technological, agrotechnical and other requirements for roller technological machines are more accurately met if all units of the roller technological machine work in a coordinated manner and provide the necessary kinematic, dynamic and other indicators. The creation of new types of roller machines and the modernization of existing ones are important in improving the quality of the product, as well as increasing the productivity of the machine. Therefore, these problems have been studied by many scientists all over the world and are still studied [1-2, 13, 20-22].

The main aspect of improving mechanical processing in roller machines is the study and solution of the problem of the interaction of contact points in two-roller modules [14, 16]. The articles give the results of a study on the working organs of various technological roller machines used in the leather industry [17-19]. The scientific research work presented in the articles is devoted to the study of transport devices used in roller technological machines [20-22].

As roller technology machines are widely used [1, 22-23], various technological requirements are also imposed on these machines [24]. Compliance of roller machines with one or another technological requirement involves the use of various structures of working bodies (working shafts) and various structures of auxiliary mechanisms. If a whole complex of sometimes conflicting technological requirements is imposed on roller machines with fixed working shafts of inter-axis distance, then for roller machines with variable working shafts of inter-axis distance, these technological requirements are not suitable. The complexity of technological requirements (to satisfy these requirements) on roller machines with variable working shafts of inter-axis distance implies the use of mechanisms with complex kinematic properties [15].

In most roller-driven technology machines, the inter-axial distance of the working shafts varies depending on the characteristics of the technological process. In addition to other requirements, the transmission mechanisms of such technological machines between the working shafts have a special requirement to ensure that the working shafts rotate synchronously at the time of changing their inter-axis distance. The degree of accuracy of the technological, agrotechnical and other requirements of the presented roller machine depends on the characteristics of the transmission mechanism used. In roller machines, various transmission mechanisms are used: chain, belt, cardan, worm gear, tooth, tooth-lever and others.

The requirements to ensure synchronous rotation of working shafts at the time of changing the inter-axis distance can be met by tooth-lever differential transmission mechanisms [1, 22-23]. Gear-lever mechanisms have been known in engineering for a very long time. However, in recent years, some amazing features of these mechanisms have appeared, which makes them one of the most promising for the creation of modern machines and devices. In connection with the tasks of synthesizing self-adjusting mechanisms, the design of gear-lever mechanisms is one of the most modern and promising.

The movement of gear-lever mechanisms has good dynamic characteristics, comparable to the characteristics of other types of mechanisms. We have developed and examined a number of tooth-lever differential transmission mechanisms in relation to technology machines with variable Inter-axis distances [1-10]. The article presents a kinematic analysis of the tooth-lever differential transmission mechanism developed by the authors, which eliminates the geometric shift between the working shafts and the material to be processed.

Method

The formulas obtained analytically to calculate the parameters of gear-rich differential transmission mechanisms are very complex and require a large amount of calculations, which, as a number of researchers have shown, do not have accuracy [25]. Graphic-analytical and graphic methods of studying differential mechanisms do not have such disadvantages. The main disadvantage of these methods is the comparative inaccuracy of the results obtained. The creation of a number of programs of computer graphics (AutoCAD, Solid Works, Compass-Graphic, Mathcad, etc.) eliminated this fundamental disadvantage of graphic and graphic-analytical research methods, while retaining all its advantages and giving impetus to its development [25].

Previously, we studied the tooth-lever differential transmission mechanism for a roller pair with working shafts diameters equal to each other and proved the possibility of preventing geometric displacement between the working shafts and the material to be processed. [15]

In this study, we studied the tooth-lever differential transmission mechanism for Val pairs with different diameters of working shafts. *(fig.1.)*

Let us prove geometric conditions that prevent geometric displacement between working shafts with different diameters and the material to be processed. In this case, the ratio of the diameters of the intermediate, leading and leading gear wheels of the mechanism in proportion to the diameters of the working shafts is expressed in the formula below.

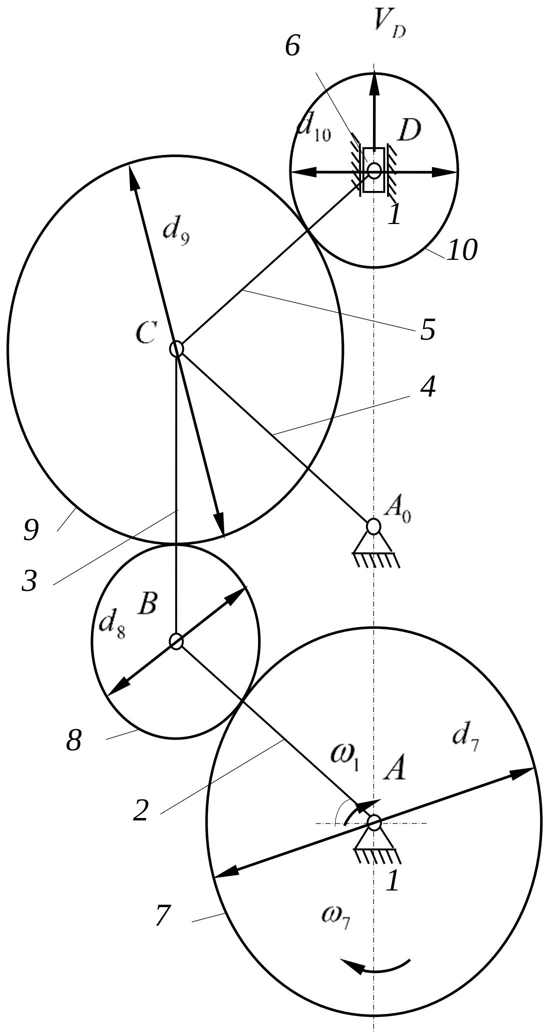

Figure.1. Tooth-lever differential transmission mechanism. 1- bed frame; 2, 3, 4, 5- lever 6- slider; 7, 8, 9, 10- tooth wheels.

$$U = \frac{D_1}{D_2} = \frac{d_{10}}{d_7} = \frac{d_8}{d_9}$$

where U - gear ratio of the mechanism, D_1 and D_2 - working shafts diameters, d_7, d_8, d_9, d_{10} - diameter of tooth wheels.

Graph-analytical studies of the tooth-lever differential transmission mechanism are conducted using the method developed by the authors. Linear and angular velocities of the characteristic points of the mechanism links are determined. Since for tooth-lever differential transmission mechanisms of this type, an important criterion is the gear ratio of the mechanism and its change pattern, the article shows graphs of the change in the gear ratio of these mechanisms depending on the change in the center distance of the driving and driven links.

Figure 2 shows a graph of the kinematics of the gear-wheel differential transmission mechanism.

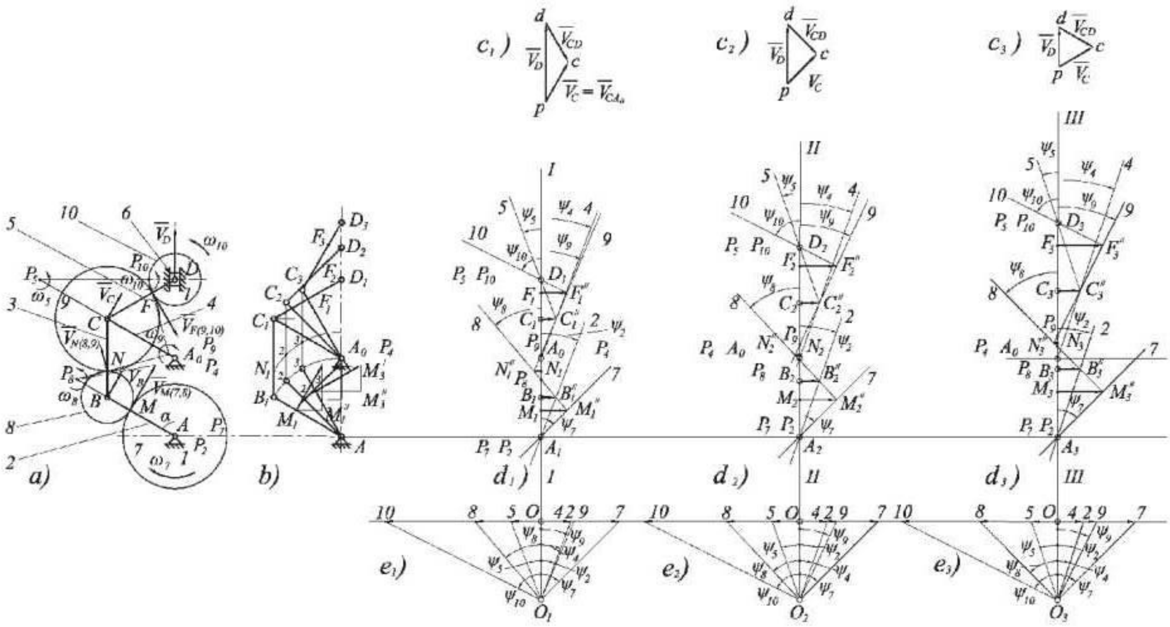

Figure 2. A pattern of velocities of a tooth-lever differential transmission mechanism with a parallelogram lever contour: a) a plan of the mechanism; *b*) a plan of the lever contour in various positions; *c_1), c_2), c_3)* a plan of the lever contour velocities; *d_1), d_2), d_3)* - a pattern of velocities in *I, II, III*- positions of the mechanism, respectively; *e_1), e_2), e_3)* - a pattern of angular velocities in *I, II, III* positions of the mechanism, respectively.

Figure 2, *a*) shows the plan of the tooth-lever differential mechanism; where 1 is the bed frame; 2, 3, 4, 5 are the levers; 6 is the slider; 7, 8, 9, 10 is the tooth wheels. Fig. 1 also shows the patterns of the velocities of a tooth-lever differential transmission mechanism with a parallelogram lever contour: *a*) a plan of the mechanism; *b*) a plan of the lever contour in various positions; c_1), c_2), c_3) - plans of the lever contour velocities; d_1), d_2), d_3) - patterns of velocities in *I, II, III*- positions of the mechanism, respectively; e_1), e_2), e_3) - patterns of angular velocities in *I, II, III* - positions of the mechanism, respectively.

From the condition of the matter, we consider that the state of this mechanism (φ) and geometric parameters (. l_{AB}, l_{BC}, l_{CD}, d_5, d_6, d_7, d_8), the linear velocity of the center of rotation, the linear velocity of the moving gear wheel ($\overline{V}_D$) and angular velocity (ω_5) in size and direction we know.

Adopting the scale of the mechanism plan (μ_l), we draw the mechanism plan in several positions, where (μ_l) is defined by the following expression

$$\mu_l = \frac{AB}{l_{AB}} \left[\frac{mm}{m}\right], \quad (1)$$

where *AB* - is the length of the link of the mechanism in plan *[mm]*,
l_{AB} - is the true length of the described link in meters *[m]*.

The velocity of point *D* ($\overline{V}_D$) in the velocity plan is denoted by segment $\overline{Pd}$ determined by the following formula

$$\overline{Pd} = \mu_v \cdot \overline{V}_D, \tag{2}$$

$$\mu_v = \frac{\overline{Pd}}{\overline{V}_D} \tag{3}$$

where μ_v is the scale velocity $\left[\frac{mm}{ms^{-1}}\right]$.

To build a velocity plan, we solve the system of equations

$$\begin{cases} \overline{V}_C = \overline{V}_D + \overline{V}_{CD} \\ \overline{V}_C = \overline{V}_B + \overline{V}_{CB} \end{cases} \tag{4}$$

So, equating the right sides of the equation, we get

$$\overline{V}_C + \overline{V}_{CD} = \overline{V}_B + \overline{V}_{CB} \tag{5}$$

V_{CD} and V_{CB} we determine the values of their speeds by solving the equation (5) graphically.

$$\overline{V}_C = \overline{V}_{CB} = \frac{\overline{P_C}}{\mu_v} \tag{6}$$

$$\overline{V}_{CD} = \frac{\overline{d_C}}{\mu_v} \tag{7}$$

Having solved the lever contour of the tooth-lever mechanism, we obtain the linear and angular velocities of the crank-slider four-link mechanism 12341.

We proceed to the solution of the gear contour.

According to the condition of the problem, we have the angular speed of the tooth wheel 7 ω_7 and its step diameter d_7 we know.

M the velocity of a point is determined by the formula

$$\overline{V}_M = \omega_5 \cdot r \tag{8}$$

$\overline{V}_M$ the direction of is known $\overline{V}_M \perp BA$. $\overline{V}_M$ the segment corresponding to the velocity is determined by the formula

$$\overline{M_1 M_{1^1}} = \overline{V}_M \cdot \mu_v \tag{9}$$

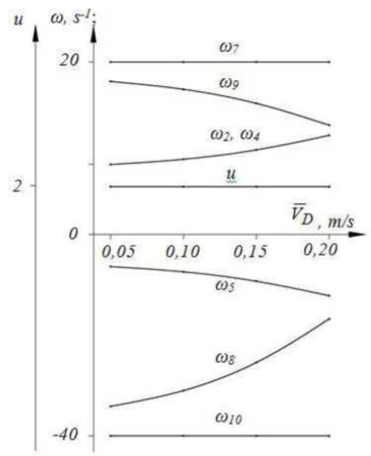

Figure 3. Graph for tooth-lever differential transmission mechanisms with different diameters of tooth wheels

Of the leading and leading tooth wheels (*AD*) drawing a vertical line I-I (for the first position of the mechanism) parallel to the line passing through the axes, we pass the points *A*, *M*, *D*, *B*, *N*, *C*, *F* to this line with horizontal Rays (Figure 1, d_1), the points obtained are respectively A_1, M_1, B_1, N_1, C_1, F_1, D_1 is defined as.

M_1 M_1 segment M_1 move to Point (Figure 1, d_1) and M_1 M_1 move it to a horizontal line to get the segment". M_1 from the point A_1 we draw an oblique beam passing through the point and mark it with 7. This beam will be a straight line of the speed distribution of wheel 7. The angle of the 7th beam with the i-i line is the angular velocity of the 7th wheel ω_7 gives.

Next, we determine the angular velocity of all joints and the transfer ratio of the mechanism.

In the same way, we make plans for the remaining positions of the mechanism (*fig.2*, c_2, c_3, d_2, d_3, e_2, e_3).

Based on the results of the initial data and the calculations performed, we build graphs of the angular velocities of the links and the transfer ratio of the mechanism (*fig. 3*).

Conclusions

Geometric conditions have been developed that prevent geometric displacement between the working shafts and the material to be processed for the tooth-lever differential transmission mechanism used in roller technology machines with variable Inter-axis distance and different diameters of the working shafts. This position is characteristic as a result of the proportional ratio of the diameter of the leading, leading and intermediate gears of the mechanism to the diameter of the working shafts.

In the studied gear-bearing differential transmission mechanism, there is no geometric slip between the working shafts and the material to be processed under structured geometric conditions, which leads to an improvement in the quality of the material to be processed.

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