Gear-Lever Differential Transmission Mechanism of Different Diameters

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Abstract: - In all spheres of society and country, industries are rapidly developing, requiring the implementation of reforms based on modern innovative ideas, developments, and technologies that ensure rapid and high-quality progress of each country on the path to becoming one of the leaders of world civilization. At the same time, the analysis showed that activities to modernize and diversify production, increase its output, and expand the types of competitive products in the domestic and foreign markets are not conducted properly. To update the fleet of equipment processing local raw materials, constant improvement of technologies and designs of technological machines is necessary. An improved design of the gear-lever mechanism is proposed for the working parts of various technological machines that process fibrous raw materials. This mechanism ensures equal linear velocity at the contact points of the lower and upper rollers. Consequently, high-quality indices of processing and the resulting material are ensured.

Key-Words: - gear-lever mechanism, technological machines, fibrous raw materials, working rollers, mechanical design, working bodies, different diameters.

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1 Introduction

All spheres of society and state life are rapidly developing, requiring the implementation of reforms based on modern innovative ideas, developments, and technologies that ensure rapid and high-quality progress of our country on the way to becoming one of the leaders of world civilization. At the same time, the conducted analysis showed that work on modernization, diversification of production, an increase in its size, and expansion of types of competitive products in domestic and foreign markets are not carried out properly. Consider current trends, innovations, and methods in the field of design and modeling of drive systems used in technology, in various machines and industries.

In [1], the authors studied the parameters of the lever device and developed a model that describes the force load exerted on the elements of the lever device in mining hauling units.

The authors of [2], developed a new method for synthesizing planetary mechanisms with one and several degrees of freedom. This method of mechanism synthesis is effective for designing automatic transmission mechanisms.

A new method of assembly accuracy prediction for a planetary gear was developed in [3]. This method considers the assembly accuracy when the bearing gear shaft and bolt engage. Bolt assembly models developed by finite element and analytical methods are used to account for the interaction between the toothed shaft of the bearing and the bolt.

In [4], a new model of the gear system was developed that accounts for the meshing of gears and their reactions to torsion. The characteristics of friction between gears were investigated.

In [5], an improved method for spring balancing of articulated manipulators was developed. The ideal energy balance of articulated manipulators is achieved through the arrangement of springs without the use of auxiliary links.

The study in [6], is devoted to developing a new compact drive with adjustable stiffness. The transmission precision of this drive is based on a planetary gear train with a rocker link. A prototype of the drive was built, and its performance and the algorithm for identifying stiffness were experimentally substantiated.

In [7], a new type of drive was developed that increases the energy efficiency of a technical system due to torque variability and gravity compensation. This drive is effective when used in the design of robotic systems.

In [8], the authors of the article solved the problem of the permissible six-dimensional position of the working body of serial mechanisms with low mobility using the inverse kinematic modeling method.

The authors of [9], developed an experimental method for analyzing the vibration of systems with straight bevel gears. Using the developed analytical method, cracks in the teeth of bevel gears were studied.

In [10], the durability of plastic gears of vehicles was investigated and analyzed based on the ISO standards. To determine the resonant frequency, plastic gears were vibration tested. Based on the research results, specific recommendations were issued to improve the durability of vehicle drives.

In [11], the authors studied the dynamics of a planetary bearing of the separator based on a complex model that considers gearing and gear contacts. It was determined that the centrifugal force of roller rotation, which causes impact force, has a greater influence on the separator.

In [12], the balancing of the gear mechanism of a cardan drive was considered and studied. Theoretically, it was determined that the mechanism could be balanced using springs. Robotic manipulators were experimentally tested on a prototype of the developed balancer with one degree of freedom.

The study, [13], examined gear contacts considering their transmission efficiency. For this purpose, a method for treating the surface of gears was developed. The research results showed improved gear performance achieved through flow machining of gear surfaces.

In [14], the dynamic performance of planetary gears was studied. For this purpose, a matrix method for searching for planetary gear configurations was developed. As a result of applying this method, it became possible to ensure the required gear ratios of the planetary mechanism.

The authors of [15], proposed a new graphical method for analyzing gear transmission to construct a geometric model and determine the rotation velocity of planetary drive elements. This graphical method simplifies the design stages of planetary gears.

Based on the analysis of the reviewed recent publications, [1], [7], [15], it was determined that when designing and calculating a differential transmission mechanism for a roller technological machine, it is necessary to consider gear ratios, torque of the working parts, balance of the generated power, losses in gear engagement, and transmission efficiency.

These values will contribute to the design and the calculation of the strength characteristics of the gear-lever differential transmission mechanism without loss of generated and transmitted power.

The mechanism that we have developed can be used in mechanical engineering and light industries with roller technological machines consisting of two working shafts of different diameters with a variable distance between axes and a center of rotation moving in a straight line, forward-reciprocating.

In the tanning, textile, mining, metallurgical, and other sectors of the global industry, various transmission mechanisms are used in shaft machines. These mechanisms are used to transmit rotational motion from the first shaft to the second shaft.

In particular, the design of the VOPM-1800-K roller squeezing machine (Russia) uses a squeezing roller drive mechanism. Gears with elongated teeth are fixed at the output ends of the squeezing roller axes. The elongated teeth of the gears of the drive mechanism make it possible to transmit rotational motion from the drive roller to the driven roller, considering changes in the thickness and uneven surface of the processed raw material.

Technological machines with different diameters of working bodies are also used in production. The designs of these machines usually use chain drive mechanisms. However, in this case, a violation of the requirements of the technological process is observed; in particular, when processing raw materials, geometric sliding occurs between the rollers and the raw material. Known drive mechanisms with a system of gears and levers have a rather complex design. This leads to significant difficulties in achieving the goal of transmitting rotational motion to the working organs of technological machines, [16], [17], [18].

The disadvantage of the existing transmission mechanisms is that they do not meet the technological requirements imposed on the shaft machine when the diameters of the working shafts are different.

2 Materials and Methods

Considering the elimination of existing shortcomings in known drive mechanisms and the simplification of their designs, a group of researchers has proposed an improved design of the drive mechanism for the working parts of technological machines. The drive mechanism is adapted and suitable specifically for use with working bodies of different diameters of technological machines, widely used in various manufacturing industries (Figure 1 and Figure 2).

The task of the drive mechanism is to provide equal linear velocities for each working element, in this case, to the upper one with a small diameter and the lower working element with a large diameter.

Thus, considering the diameters of the working bodies and by selecting rational parameters of the gear and lever system, technological conditions are provided for uniform and high-quality processing of raw materials. The drive mechanism under consideration is arranged in the following order:

Lower gear 6 of the drive mechanism is intended to rotate working body 2. Through gear 6, rotation is transmitted to gear 11, and through it to gear 12, through which gear 5 receives rotational motion. Gear 5 is fixed on the axis of working body 2. Each gear in sequence from bottom to top is equipped with levers 7, 13, 14, 8, 3 (Figure 1).

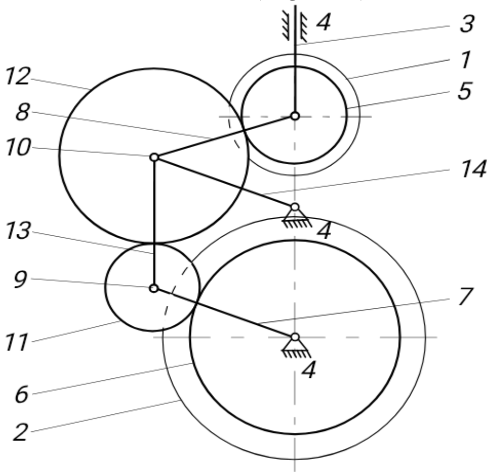


Fig. 1: Design of the drive mechanism for rotating working bodies
Source: created by the authors

Gear wheels 5 and 6 are fixed to the output ends of working shafts 1 and 2, and levers 7 and 8 are hinged; levers 7 and 8, 9 and 10 are axle supports. Intermediate gears on axles 9 and 10 are freely mounted on axles 11 and 12. The axles 11 and 12 of the intermediate gears are kinematically connected using the lever 13. The axis 10, which is installed close to the upper working shaft 1, is kinematically connected to the support by means of a lever 14. Wings 7, 13, and 14 wings together form a parallelogram with wings (Figure 1 and Figure 2).

The operation procedure of the proposed transmission mechanism:

Torque is transmitted from the leading working shaft 1 to the leading working shaft 2 in the following way (Figure 1 and Figure 2). It is transmitted from the leading gear wheel 5 fixed to the output end of the leading working shaft 1 to the leading gear wheel 6 fixed to the output end of the driven working shaft 2 through intermediate gears 11, 12. Four sequentially contacting gear wheels 5, 6, 11, 12 are kinematically connected through the centers of rotation using mutual levers 7, 8, 13, 14 (Figure 3).

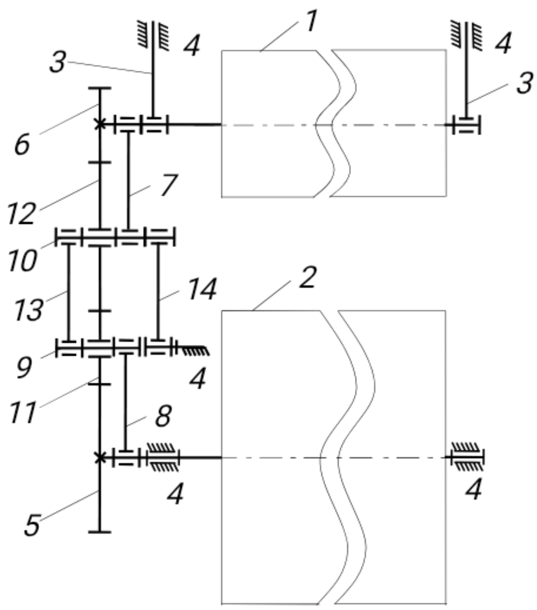
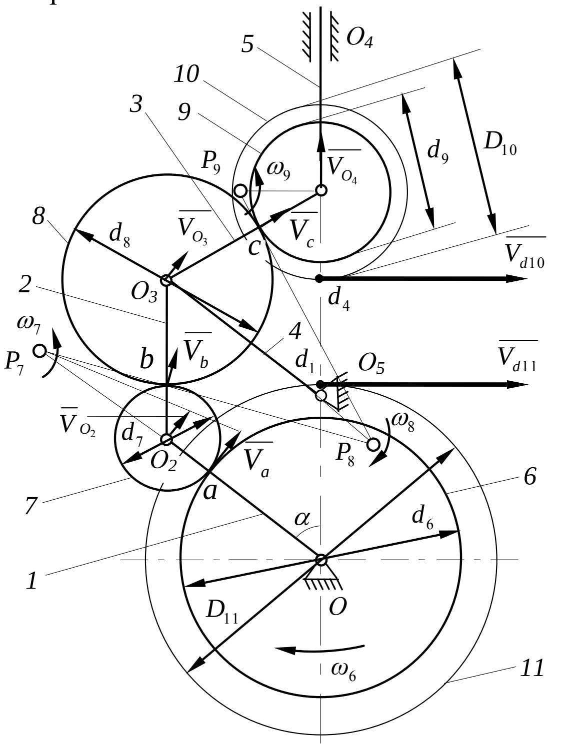


Fig. 2: Design of the lever system of the drive mechanism
Source: created by the authors

2.1 Model Assumptions

The proposed transmission mechanism is analyzed under the assumption of rigid-body motion of gears and levers. Elastic deformation of elements and friction losses in gear meshing are neglected in the first approximation. The kinematic constraints are defined strictly by the geometric relations shown in Figure 1, Figure 2 and Figure 3. The driven working shaft 1 is reciprocated along the vertical axis with the help of the lever 3.



1, 2, 3, 4, 5 – gears, 6, 7, 8, 9 – gear wheel
Fig. 3: Gear differential transmission mechanism
Source: created by the authors

Let us determine the values of the linear velocities of the contact points of working bodies 1 and 2 corresponding to the lower and front surfaces of the proposed processed raw material. The following designation is introduced:

$$U = \frac{D_{11}}{D_{10}}; \tag{1}$$

$$\frac{D_{11}}{D_{10}} = \frac{d_6}{d_9}; \tag{2}$$

$$\frac{d_6}{d_9} = \frac{d_8}{d_7}; \tag{3}$$

where: D_{11}, D_{10} – diameter of working shafts, selected parameters.
d_6, d_7, d_8, d_9 – diameter of the pitching circle of gear wheels, selected parameters.
U – transmission ratio.

In accordance with conditions (1), (2), (3), we check that the following condition (4) is fulfilled

$$V_{D_{10}} = V_{D_{11}}; \tag{4}$$

where: $V_{D_{11}}$, $V_{D_{10}}$ – linear speeds of contact points of working shafts.

We can write formulas (5), (6) from formulas (1), (2), and (3).

$$U = \frac{r_6}{r_9} = \frac{\omega_9}{\omega_6} \tag{5}$$

$$U = \frac{r_8}{r_7} = \frac{\omega_7}{\omega_8} \tag{6}$$

Then:

$$U = \frac{D_{11}}{D_{10}} = \frac{d_6}{d_9} = \frac{r_6}{r_9} = \frac{d_8}{d_7} = \frac{r_8}{r_7} = \frac{\omega_9}{\omega_6} = \frac{\omega_7}{\omega_8}; \tag{7}$$

We determine V_a, ω_6, ω_7, aP_7, V_b, bP_7, ω_8, bP_8, V_c, cP_8, ω_9, cP_9, V_{O_4}, O_4P_9, $V_{D_{11}}$, $V_{D_{10}}$ the characteristic points of the transmission mechanism.

$$V_{D_{10}} = \frac{V_c}{cP_9} \cdot \frac{R_{11}}{U}; \tag{8}$$

$$V_{D_{10}} = \frac{\omega_8 cP_8}{cP_9} \cdot \frac{R_{11}}{U}; \tag{9}$$

$$V_{D_{10}} = \frac{V_b}{bP_8} \cdot \frac{cP_8}{cP_9} \cdot \frac{R_{11}}{U}; \tag{10}$$

$$V_{D_{10}} = \frac{\omega_7 bP_7}{bP_8} \cdot \frac{cP_8}{cP_9} \cdot \frac{R_{11}}{U}; \tag{11}$$

$$V_{D_{10}} = \frac{V_a}{aP_7} \cdot \frac{bP_7}{bP_8} \cdot \frac{cP_8}{cP_9} \cdot \frac{R_{11}}{U}; \tag{12}$$

$$V_{D_{10}} = \frac{\omega_6 r_6}{aP_7} \cdot \frac{bP_7}{bP_8} \cdot \frac{cP_8}{cP_9} \cdot \frac{R_{11}}{U}; \tag{13}$$

$$V_{D_{10}} = \omega_6 R_{11} \left(\frac{r_6}{aP_7} \cdot \frac{bP_7}{bP_8} \cdot \frac{cP_8}{cP_9} \cdot \frac{1}{U} \right); \tag{14}$$

$$V_{D_{10}} = \omega_6 R_{11} \left(\frac{r_6}{aP_7} \cdot \frac{bP_7}{bP_8} \cdot \frac{cP_8}{cP_9} \cdot \frac{r_9}{r_6} \right); \tag{15}$$

$$V_{D_{10}} = \omega_6 R_{11} \left(\frac{r_6 \omega_7}{V_a} \cdot \frac{V_b}{\omega_7} \cdot \frac{\omega_8}{V_b} \cdot \frac{V_c}{\omega_8} \cdot \frac{\omega_9}{V_c} \cdot \frac{r_9}{r_6} \right); \tag{16}$$

$$V_{D_{10}} = \omega_6 R_{11} \left(\frac{\omega_9 r_9}{V_a} \right); \tag{17}$$

$$V_{D_{10}} = \omega_6 R_{11} \left(\frac{\omega_9 r_9}{\omega_6 r_6} \right); \tag{18}$$

$$V_{D_{10}} = \omega_6 R_{11} \left(U \frac{1}{U} \right); \tag{19}$$

$$V_{D_{10}} = \omega_6 R_{11}; \tag{20}$$

$$V_{D_{10}} = V_{D_{11}}. \tag{21}$$

2.2 Computational Procedure and Convergence Clarification

The kinematic relations of the proposed mechanism were obtained analytically (Eqs. 1–21). Numerical values presented in Section 3 were calculated by direct substitution of parameters into the derived analytical expressions. Since no iterative numerical solver or mesh-based discretization was employed, convergence tolerance parameters are not applicable in this study.

Computational consistency was verified by repeated evaluation under identical parameter inputs. Therefore, numerical stability depends solely on the correctness of the analytical formulation. The resulting formula (21) shows the provision of the condition of constancy of linear velocities in the contacting surfaces of the working bodies with the raw material being processed.

Non-slippage between the contact points of the semi-finished product and the working shaft prevents the raw material from wrinkling and stretching, [16], [17], [18], [19], [20], [21], [22]. As a result, deterioration of the quality of mining semi-products (tearing, perforation) is prevente, [2], [23], [24].

The developed mechanism for driving working bodies is ideal and can be used in the design of devices and roller technological machines to improve the quality of mechanical processing of elastic-viscous fibrous materials, [25], [26], [27], [28], [29], [30].

3 Results

The theoretical analysis based on Eqs. (1–21) was used to obtain numerical values of angular and linear velocities for the characteristic points of the gear-lever differential transmission mechanism. The results were calculated for different gear diameters and plotted graphically.

For the drive mechanism under consideration, we set the following technical and technological parameters: d_6=80, d_7=40, d_8=80, d_9=40 mm, ω_1=1 s^{-1}, V_{O4}=20 m/s, and φ_1=45÷65°. Using the received values, we can determine the values of the angular and linear velocities of the characteristic points of the mechanism using the Microsoft Excel program. Based on the determined values, we express the speed change of the mechanism links through graphs.

The Table 1 (Appendix) contains indices of the centers of rotation of the small (driven) and large (driven) gears of the drive mechanism. Based on certain values of the drive mechanism from the table, graphs of the dependences of the angular and linear velocities of the contacting points of the rotating working bodies of the technical system were constructed (Figure 4 and Figure 5 in Appendix). Thus, a drive mechanism has been developed that makes it possible to select rational technical parameters depending on the technological requirements of the processing process and the parameters of the processed raw materials. This drive mechanism can be used in the designs of various technological machines that have working bodies with different diameters. The drive mechanism is ideal for processes where constant rotational movement of the processing elements is required.

3.1 Error Analysis and Numerical Stability

To assess the robustness of the proposed mechanism, the main geometric parameters (d6, d7, d8, d9) and angular velocity ω1 were varied within ±5%. The recalculated angular and linear velocities showed proportional behavior, and the equality condition of linear velocities at the contact points (eq. 4) was preserved. The maximum deviation of calculated linear velocities did not exceed 6%, which confirms the stability of the analytical model under moderate parameter perturbations.

3.2 Sensitivity Analysis

A sensitivity study was conducted by varying the diameters of the gear wheels and lever geometry within ±10% of their nominal values. The results demonstrate that the proposed mechanism maintains the no-slip condition and preserves the constancy of linear velocities at the contact surfaces. This indicates that the kinematic performance of the gear-lever differential transmission mechanism is not critically sensitive to moderate geometric deviations.

3.3 Benchmark Consistency

The obtained kinematic relations are consistent with classical gear transmission theory, [29], [30], which states that equal linear velocities at contact points eliminate geometric slipping. Compared to traditional chain-driven systems used for rollers of different diameters, the proposed gear-lever differential transmission mechanism ensures improved kinematic compatibility.

4 Conclusion

The drive mechanism we considered solves one of the main problems of the technology for processing raw materials with rotating working bodies, namely, the constancy of their rotational motion. The conditions for the constancy of the rotational movement of the working bodies are ensured even with a constant change in the thickness over the area of the processed raw material. The obtained analytical formulas make it possible to conduct further power calculations of the considered drive mechanism of the working parts of technological machines.

The considered gear-lever differential transmission mechanism ensures rotation of the contact points of the working shafts at various angular speeds, regardless of the change in the distance between the axes of the working shafts. The results of the analysis and obtained formulas allow for carrying out dynamic studies, as well as dynamic and geometric synthesis of the transmission mechanism.

The kinematic parameters of the considered gear-lever differential transmission mechanism were determined by numerical calculations, and graphs were constructed. Analysis of the graphs shows that the correct selection of the considered gear-lever differential transmission mechanism allows working without geometric slippage between the working shafts and the processed material in roller machines. the geometric dimensions of the mechanism are d_6=80, d_7=40, d_8=80, d_9=40 mm and the kinematic parameters are ω_1 =1 s^{-1}, V_{O4} =20 m/s, the contact between the mechanism and the processed material It should be noted that when using the proposed drive mechanism, it is possible to achieve an increase in the share of high added value of the finished product by ensuring uniform and high-quality processing of raw materials.

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The authors equally contributed in the present research, at all stages from the formulation of the problem to the final findings and solution.

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Conflict of Interest

The authors have no conflicts of interest to declare that are relevant to the content of this article.

APPENDIX

Table 1. Technical characteristics of the drive mechanism of the working bodies

φ_1, (°)	45	48	51	54	57	60	63	66
φ_2, (°)	45	48	51	54	57	60	63	66
φ_3, (°)	45	48	51	54	57	60	63	66
φ_4, (°)	90	84	78	72	66	60	54	48
φ_5, (°)	45	42	39	36	33	30	27	24
φ_6, (°)	90	96	102	108	114	120	126	132
φ_7, (°)	45	42	39	36	33	30	27	24
φ_8, (°)	29	28	27	26	25	24	22	21
φ_9, (°)	29	28	27	26	25	24	22	21
φ_{10}, (°)	106	110	114	118	122	126	131	135
P_3O_4, mm	85	89	93	97	101	104	107	110
P_3O_3, mm	60	60	60	60	60	60	60	60
O_2P_7, mm	10.93836	10.1390	9.485038	8.944271	8.493714	8.116548	7.800232	7.535278
aP_7, mm	30.93836	30.1390	29.48503	28.94427	28.49371	28.11654	27.80023	27.53527
bP_7, mm	14.50048	15.2127	15.84931	16.42277	16.94410	17.42261	17.86601	18.28074
bP_8, mm	29.00097	30.4254	31.698632	32.84555	33.88821	34.84522	35.73202	36.56148
O_3P_8, mm	21.87672	20.2780	18.97007	17.88854	16.98742	16.23309	15.60046	15.07055
cP_8, mm	45.59156	42.9141	40.55041	38.44178	36.54887	34.84522	33.31361	31.94351
O_4P_9, mm	10	10	10	10	10	10	10	10
cP_9, mm	14.73625	14.2387	13.75287	13.28131	12.82699	12.39313	11.98321	11.60094
ω_3, 1/s	0.235702	0.22427	0.214459	0.206011	0.198727	0.192450	0.187054	0.182439
ω_4, 1/s	0.235702	0.22427	0.214459	0.206011	0.198727	0.192450	0.187054	0.182439
ω_1, 1/s	0.235702	0.22427	0.214459	0.206011	0.198727	0.192450	0.187054	0.182439
ω_6, 1/s	1	1	1	1	1	1	1	1
ω_7, 1/s	1.292893	1.32718	1.356620	1.381966	1.403818	1.422649	1.438836	1.452681
ω_8, 1/s	0.646446	0.66359	0.678310	0.690983	0.701909	0.711324	0.719418	0.726340
ω_9, 1/s	2.00000	2.0000	2.00000	2.00000	2.00000	2.00000	2.00000	2.00000
V_{O4}, m/s	20	20	20	20	20	20	20	20
V_{O3}, m/s	14.14213	13.4563	12.86759	12.36067	11.92363	11.54700	11.22326	10.94636
V_{O2}, m/s	14.14213	13.4563	12.86759	12.36067	11.92363	11.54700	11.22326	10.94636
Va, m/s	40	40	40	40	40	40	40	40
Vb, m/s	18.74758	20.1900	21.50150	22.69571	23.78645	24.78627	25.70627	26.55609
Vc, m/s	29.47251	28.4775	27.50575	26.56262	25.65398	24.78627	23.96642	23.20188
V_{d10}, m/s	100	100	100	100	100	100	100	100
V_{d11}, m/s	100	100	100	100	100	100	100	100

Source: created by the authors

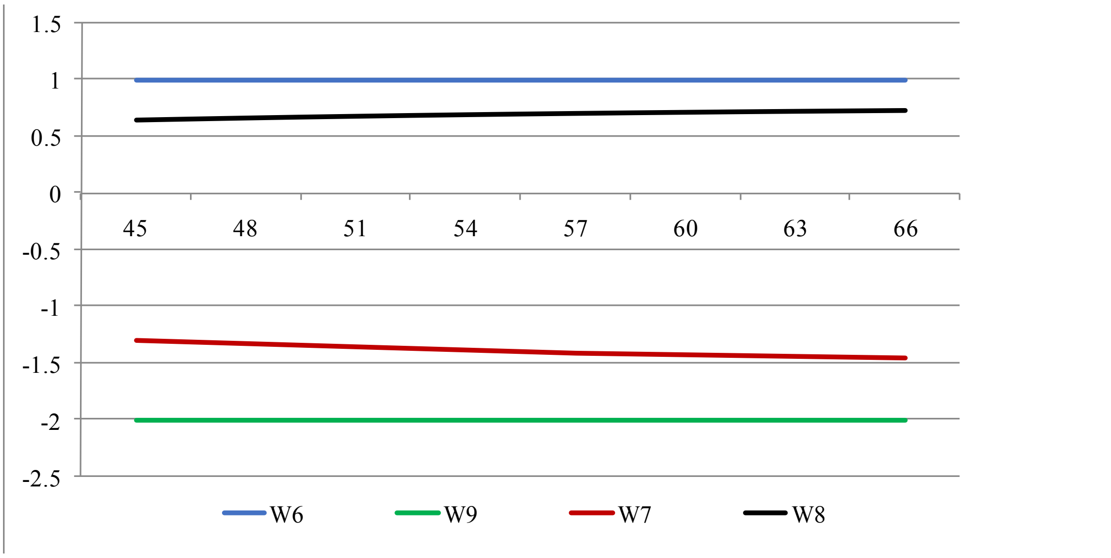

Fig. 4: Graph of changes in angular velocities at the points of contact of the working parts of the machine
Source: created by the authors

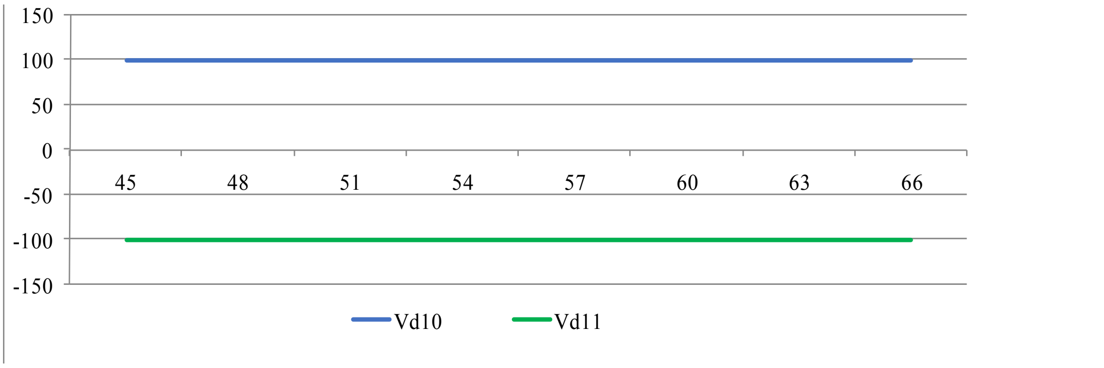

Fig. 5: Graph of changes in linear velocities at the points of contact of the working parts of the machine
Source: created by the authors