Dynamic analysis of a ten-link tooth-lever differential transmission

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Abstract. This article discusses the dynamics of a ten-link tooth-lever differential transmission mechanism. The force analysis of the transmission mechanism is given in order to find the dependence for determining the reaction in kinematic pairs and the balancing moment of the pair of forces and to show some features of the tooth-lever transmission mechanism. The force calculation was carried out taking into account the accelerated movement of links since their acceleration in modern high-speed machines is very significant. To obtain a more accurate concept of the external forces and moments loading the transmission mechanism in the accelerated movement of the links, the dynamics of the transient process of roller technological machines was considered. Cases of feeding the processed material were considered both from the side of the intermediate gears and from the side opposite to the parasitic gears. Dependencies were obtained to determine the force characteristics of this mechanism. Cases of pressure unloading and overloading on the processed material from the side of the free shaft, depending on the location of the transmission mechanism are shown. The dependence of the reaction force of intermediate gears on their own axes of rotation on the angle between the levers is shown. With an increase in the angle between the levers, the reaction of the intermediate gears on the axis of rotation increases.

1. Introduction

Currently, in the industry of our Republic and abroad, roller machines for various purposes are widely used. In metallurgical; mechanical engineering; agricultural; building; textile; rubber; chemical; pulp and paper; polygraphic; food; leather and light industries roller machines are among the main technological machines. The main advantages of roller machines are their fit into production lines, the possibility of continuous operation of the technological process, high productivity and simplicity of design. The main requirement for existing and newly created technological machines, including roller ones, and their mechanisms, is to ensure that technological requirements are met with maximum accuracy; this improves the quality of products, increases machine productivity and saves resources.

The main units of roller machines are the roller working body (rollers), conveyors transporting the processed material, mechanisms that create pressure between the working shafts, devices for feeding the processed material into the processing zone, transmission mechanisms between the working shafts, which transfers torque from one working shaft to another working shaft with set kinematic parameters and control mechanisms.
Modernization of existing and creation of new roller machines plays an extremely important role in increasing the productivity of the machine and improving the quality of products. Therefore, these issues attract the interest of scientists and engineers worldwide. In recent years, a number of new technical solutions have appeared [1-7] and a number of studies devoted to the problems of roller technological machines have been published.

So, the studies in [8-15] are devoted to the solution of contact interaction in two-roll modules. Mathematical models of roll contact curves, friction stresses and contact stress distribution patterns were obtained. The main aspect in improving mechanical processing in roller machines is the study and solution of the problem of contact interaction in two-roll modules. The theory of contact interaction should predict the shape of the contact area and the patterns of its growth with increasing load, as well as the magnitude and distribution of surface normal and tangential forces transmitted through the contact surfaces. In [16-18, 23] the results of the study of working bodies of various roller technological machines used in the leather industry are presented. The results of the research are presented in [19-21]. Optimization of energy-saving technologies in mechanical engineering using modern methods of simulation is becoming an urgent area of informatization of engineering in the process of cold asymmetric rolling of metal sheets in roll machines, accompanied by a non-homogeneous stress-strain state. A number of scientific research publications [22, 24-26] are devoted to the study of transporting devices used in roller technological machines.

In various technological roller machines, due to the features of the technological process run, the center distance of the working shafts changes. In addition to other requirements, inter-roll transmission mechanisms of such technological machines have a special requirement – to ensure synchronous rotation of the working shafts at the time of changing their center distance. The degree of accuracy of technological, agrotechnical and other requirements of the presented roller machine depends on the properties of the applied transmission mechanism. A variety of transmission mechanisms are used in roller machines: chain, belt, cardan, worm gear, gear, tooth-lever ones and others.

The requirements for ensuring the synchronous rotation of the working shafts at the time of changing their center distance can be met by tooth-lever differential transmission mechanisms [27-29]. Tooth-lever mechanisms are used in engineering for a very long time. However, it is only in recent years that some remarkable properties of these mechanisms have come to light, which make them one of the most promising for the creation of modern machines and devices. The tasks of designing tooth-lever mechanisms in the synthesis of self-adjusting mechanisms are the most modern and promising. Tooth-lever mechanisms are widely used nowadays and have good prospects in the field of converting the uniform motion of the input link, usually rotational, into reciprocating, rocking or rotational periodic motion of the input link. The periodic motion of the output link can be of three types:
1) unidirectional rotary with variable speed;
2) reciprocating, the so-called "pilgrim motion";
3) with a delay time of a certain duration.

Gear-link mechanisms have good dynamic characteristics, comparable to the characteristics of other types of mechanisms of periodic motion. We have developed and investigated a number of tooth-lever differential mechanisms in relation to roller technological machines with variable center distances [2-4].

2. Methods and Results
2.1. Dynamics of tooth-lever differential transmission mechanisms of roll machines with a variable center distance of the working shafts

The problems of the dynamics of tooth-lever differential transmission mechanisms of roll machines with a variable center distance of the working shafts have not yet received wide coverage in the special literature.

Researchers investigated the tooth-link mechanisms kinematics and some issues of the synthesis of tooth-lever mechanisms. The force analysis of the ten-link tooth-lever differential transmission mechanism of roller machines with a variable center distance of the working shafts is given in order to
find dependencies for determining the reaction in kinematic pairs and the balancing moment of the pair of forces, as well as to show some features of the ten-link tooth-lever transmission mechanism. When solving the problems of kineto-statics of mechanisms, the law of motion of the driving link, as well as the masses and moments of inertia of the mechanism links are assumed to be given. External forces and moments of forces are also considered known in each position of the mechanism. The force calculation of the ten-link tooth-lever differential transmission mechanism of roller machines with a variable center distance of the working shafts is performed on the assumption that there is no friction in the kinematic pairs and all the forces acting on the links of the mechanism are located in the same plane. In the absence of friction forces, the force of interaction between two links is always directed along the normal to the surface of their contact. In a translational pair, all the elementary forces of interaction and their resultant force, are located perpendicular to the direction of the translational pair. The force calculation should be performed taking into account the accelerated motion of the links since their acceleration in modern high-speed machines is very significant. Ignoring the accelerated motion of the links will cause an underestimation of the load forces, which can lead to errors in further engineering calculations. We take into account the accelerated motion of the links by the kineto-statics method, conditionally applying the main vector $\overrightarrow{\Phi}_i$ and the main moment $\overrightarrow{M}_{\Phi_i}$ of inertia forces to each moving link of the mechanism. Then, for each link, three kinetostatic equations can be written.

2.2. Force analysis of the transient process of roller technological machines with a variable center distance of the working shafts

Let us consider the possible modes of operation of roller technological machines with a variable center distance of the working shafts.
1. Starting the machine. The machine is started without processed material in the nib of the working shafts. In this case, the transient process is characterized by a short time, the toothed wheels of the transmission mechanism and the working shafts are loaded with a moment from the forces of inertia of rest relative to the centers of rotation. The speed of shaft rotation and therefore the acceleration during start-up, are not high. Therefore, the moment from the forces of inertia can be ignored in the calculations. When starting the machine without the material being processed in the nib of the working shafts, the center distance of the working shafts does not change.

2. Steady motion. Figure 1 shows the design diagram of the dynamics of roller machines in a steady motion. The center distance of the working shafts does not change, the layer passed between the working shafts is constant in thickness. The shafts are loaded by the force of the pressing device on the bearings (reaction forces) $R$, the force of the roll pressure on the layer of the processed material $Q$, the gravity force of the upper roll $G$, the force driving the working shaft $P$, the resultant of the friction forces $F$ (Figure 1). Until now, this scheme is taken as the basis for calculating the torque and forces in a steady process. Moreover, if we expand the forces of normal pressure and friction into horizontal and vertical components, then the force $Q$ and the sum of the vertical components of the forces $N$ and $F$ form a pair of resistance forces

$$M = h \cdot Q,$$

where $h$ is the arm of the pair, which is the coefficient of friction of the roller over the layer, determined from the pull-in condition of the processed material into the rollers.

3. Transient process caused by the roughness of the processed material. In this case, the center distance of the rollers changes. However, this change is not substantial. In addition to the forces specified in item 2, the upper shaft is acted upon by the inertial force caused by the acceleration of the center of rotation of the upper roller.

4. Transient process caused at the time of gripping. In this transient process, the upper shaft is loaded with the same forces as in item 3, but in terms of its kinematic and dynamic parameters; it is more dynamic and can be described mathematically. The transient parameters caused by the roughness of the processed material ($a$ and $t$), are less than the transient parameters during gripping ($a_3$ and $t_3$). I.e., $a_3 > a, t_3 < t$. In some roller machines with a variable center distance of the working shafts, for example, in vertical roller machines, these parameters are quite significant since the center distance of the working shafts changes in a short time over a rather large distance. Based on the above, we will consider in more detail the transient process during gripping, which is characterized by a high acceleration of the center of mass of the upper working shaft, as well as by its significant mass. The dynamic pressure of the upper shaft on the processed material at the transient process is significant; it can be observed even without devices as a shaking of the framework. Figure 2 shows the design diagram of the dynamics of the transient process at the time of gripping. The initial position of the transient process is shown in Figure 2. The upper roller of the machine is affected by:

- $Q$ – the pressure force of pressing devices;
- $G$ – the force of gravity of the upper roller;
- $P$ – the force driving the roller;
- $N$ – the resultant forces of normal pressure of the processed material acting on the working shaft;
- $F$ – the resultant of friction forces;
- $R$ – the horizontal reaction of the bearing;
- $U$ – the force of inertia of the upper working shaft;
- $\Phi$ – the moment of inertia of the upper shaft.
The moment of inertia of the working shaft can be ignored since during the transient process, the upper shaft does not receive an additional angular velocity; accordingly, its angular acceleration is zero. The forces of inertia ($U$) of the upper working shaft are determined by the formula

$$U = -ma,$$

where $m$ is the mass of the upper roller with a driven gear; $a$ is the acceleration of the center of rotation, determined by the formula (2.23) [30]

$$U = -2m\omega^2R\sin(\alpha - \varphi).$$

Forces $Q$, $G$ and $U$ are directed in one direction, therefore, to simplify the formulas, in the future we denote them by one symbol

$$Q^* = Q + G + U.$$

In this case, the moment of useful resistance is determined by the formula

$$M_{n.c} = h \cdot Q^* = h \cdot T.$$

Since at the beginning of the transient process the gripping condition is provided, $h$ can be determined from the parameters of the beginning of the transient process

$$h = R \cdot \sin \alpha,$$
where \( R \) is the radius of the upper working shaft; \( \alpha \) is the angle of the beginning of the transient process, determined by the formula (2.21) [30].

Thus, the final loading of the driven working shaft with a driven gear of a ten-link tooth-lever differential transmission mechanism of roller machines with a variable center distance of the working shafts, forces and moments was obtained. Figure 3 shows the forces acting on the driven working shaft with the driven gear. It should be noted since the acceleration of the center of rotation of the working shaft at the transient process related to the gripping does not change direction, the inertial forces also do not change direction, respectively, and formula (4) in the transient process related to the gripping is valid.

![Figure 3. To the calculation of the dynamics of the transient process](image)

2.3. *Force analysis of a ten-link tooth-lever differential transmission mechanism*

Figure 4 shows a diagram of the transmission mechanism at the gripping by all acting external forces and moments of the driven link.

![Figure 4. Design diagram of a ten-link tooth-lever differential transmission mechanism](image)
To satisfy the kinematic requirements, so that the angular velocities of the driven and driving gearwheels were equal, the diameters of the toothed wheels of the mechanism are taken pairwise identical:

\[ d_7 = d_{10}, \quad d_8 = d_q. \tag{7} \]

and the diameters of the shafts are equal to each other:

\[ D_7 = D_{10}. \tag{8} \]

**Figure 5.** Design diagram of the balance of the driven gearwheel with the driven working shaft

The feeding of the processed material for the case under consideration is done from the side of the intermediate gearwheels. Consider the balance of gearwheel 10 with the upper working shaft (Figure 5). The equation of kineto-statics is written as

\[ T \cdot h - P_{109} \cdot r_{10} \cdot \cos \psi = 0; \tag{9} \]

\[ R_{65} \cdot \cos \phi + P_{109} \cdot \cos(\phi_n - \psi) = 0; \tag{10} \]

\[ T - R_{65} \sin \phi - Q^* - P_{109} \sin(\phi_n - \phi) = 0; \tag{11} \]

where \( P_{109} \) is the driving force of link 10;

\( R_{65} \) – is the forces of reaction to link 5 from the side of link 6;

\( T \) - is the force of interaction of the processed material with the upper working shaft reduced to one vertical force;

\( Q^* \) – is the force determined by the formula (4);

\( h \) – is the arm of force \( T \) determined by the formula (6);
$r_{10}$ – reference radius of gearwheel 10;  
$\psi$ – the pressure angle of gearwheels.

From the formula (9) follows

$$P_{109} = \frac{T \cdot h}{r_{10} \cdot \cos \psi}.$$  \hspace{1cm} (12)

and (10)

$$R_{65} = \frac{P_{109} \cdot \cos(\varphi_n - \psi)}{\cos \varphi}.$$ \hspace{1cm} (13)

Consider the balance of gearwheel 9 (Figure 6). Let us write the equation of kinetostatics as

$$P_{98} \cdot r_9 \cos \psi - P_{910} \cdot r_9 \cos \psi = 0,$$  \hspace{1cm} (14)

$$R_{94} \cos(\theta - \varphi_n) - U_9 \cos \xi - P_{910} \cos(\varphi_n - \varphi) - P_{98} \cos \psi = 0.$$  \hspace{1cm} (15)

where $U_9$ is the inertia force of gearwheel 9

$$\overline{U} = -m_9 \overline{a}_9,$$  \hspace{1cm} (16)

$\overline{a}$ – is the acceleration of the center of mass.

From formula (14) and the equilibrium of wheel 8 (Figure 7) we have

$$P_{98} = P_{910},$$  \hspace{1cm} (17)

$$P_{910} = P_{109}.$$ \hspace{1cm} (18)

From formula (15), considering formulas (12), (18), we obtain

**Figure 6.** Design diagram of the balance of the intermediate gearwheel 9  
**Figure 7.** Design diagram of the balance of the intermediate gearwheel 8
From the equilibrium of wheel 8 (Figure 7) we have
\[ P_{98} \cdot r_8 \cos \varphi - P_{87} \cdot r_8 \cos \psi = 0 , \]  \hspace{1cm} (20)
\[ P_{89} \cos \psi + P_{89} \cos (\varphi_n + \psi) - U_8 \cos \xi - R_{83} \cos \left( 90 - \frac{\nu}{2} \right) = 0 . \]  \hspace{1cm} (21)

Therefore,
\[ P_{89} = P_{87} , \]  \hspace{1cm} (22)
\[ R_{83} = \frac{T \cdot h \cdot \left[ \cos (\varphi_n + \psi) + \cos \varphi \right]}{r_{10} \cdot \cos \psi \cdot \sin \left( \frac{\nu}{2} \right)} - \frac{U_8 \cdot \cos \xi}{\sin \left( \frac{\nu}{2} \right)} . \]  \hspace{1cm} (23)

From the balance of gearwheel 7 (Figure 8) we can write
\[ - P_{78} \cdot r_7 \cdot \cos \psi + M_{yp} = 0 , \]  \hspace{1cm} (24)
\[ R_{71} \cdot \cos (\varphi_n + \varphi) - P_{78} \cdot \cos (\varphi_n + \psi) - G_7 = 0 \]  \hspace{1cm} (25)

Therefore,
\[ R_{71} = R_{78} - \frac{G_7}{\cos (\varphi_n + \psi)} , \]  \hspace{1cm} (26)
\[ M_{yp} = P_{78} \cdot r_7 \cdot \cos \psi . \]  \hspace{1cm} (27)
Substituting the values of \( P_{78} \) after a series of transforms, we obtain

\[
M_{3y} = [(Q + G - 2m_o^2R\sin(\alpha - \varphi)]R\sin \alpha
\]

(28)

Considering the equilibrium of the links, it is possible to determine the remaining reaction forces

\[
R_{105} = R_{310} = \frac{T \cdot h \cdot \cos(\varphi - \psi)}{r_{10} \cdot \cos \psi};
\]

(29)

\[
R_{106} = R_{610} = R_{310};
\]

(30)

\[
R_{45} = R_{34} = \frac{U_g \cdot r_{10} \cdot \cos \psi \cdot \cos \xi \cdot T \cdot h \cdot (\cos \varphi - \psi) \cdot \cos \psi}{r_{10} \cdot \cos \psi \cdot \sin \varphi};
\]

(31)

\[
R_{34} = R_{43} = \frac{T \cdot h \cdot [\cos(\varphi + \psi) + \cos \psi]}{r_{10} \cdot \cos \psi \cdot \sin \frac{\nu}{2}} - \frac{U_g \cos \xi}{\sin \frac{\nu}{2}};
\]

(32)

\[
R_{23} = R_{32} = R_{43};
\]

(33)

\[
R_{23} = R_{72} = \frac{T \cdot h}{r_{10} \cdot \cos \psi}.
\]

(34)

Figure 9 shows the loading of the driven shaft in the case when the processed material is pulled out from the side of the idler gearwheels.

In this case, from formula (11), we have

\[
T = Q^* + P_{109} \cdot \sin(\varphi - \psi).
\]

(35)

Let us consider the case when the material to be processed is pulled in from the side opposite to the idler gearwheels, while the direction of \( T \) and \( Q^* \) does not change (Figure 9.).

From the equilibrium of the upper shaft, we have

\[
T - Q^* + P_{109} \cdot \sin(\varphi + \psi) - R_{65} \sin \varphi = 0,
\]

(36)

hence

\[
T = Q^* - P_{109} \cdot \sin(\varphi + \psi) + R_{65} \sin \varphi.
\]

(37)

Thus, the load on the processed material is different in different cases of installation of the transmission mechanism.

When the material to be processed is fed from the intermediate gearwheels, the pressure on the material to be processed increases. At large angles \( \varphi_n \), these increases per layer are significant, and can lead to a substantial increase in pressure. Wedging of the processed material and an increase in \( \varphi_n \)
can occur. Hence, an increase in $T$ results in greater reactions on the bearings. Hence, there are large losses and a sharp increase in the balance moment.

![Design scheme for force analysis, the case when the processed material is pulled in from the side opposite to the idler gearwheels](image)

**Figure 9.** Design scheme for force analysis, the case when the processed material is pulled in from the side opposite to the idler gearwheels

When the processed material is fed from the back of the idler gearwheels, the pressure on the processed material decreases. Consequently, the reactions in the bearings are reduced and the balance moment is also reduced.

From formula (27) it can be seen, that since the pressure angle $\psi$ is a constant value $\psi = 20^\circ$, then at $\varphi_n = 70^\circ$

$$T = Q^* - P_{109}.$$  

With a further increase in the angle $\varphi_n$, $T$ begins to increase. From this point of view, it is advisable that angle $\varphi_n$ during operation changes as

$$\varphi_n \leq (90 - \psi) \pm \frac{\Delta \varphi}{2},$$  \hspace{1cm} (38)

where $\Delta \varphi$ is the maximum change of angle $\varphi_n$ at the time of the transient process.

Consider the reaction forces caused by the intermediate gears on the axis of rotation.

From formula (23), after some transforms, we obtain

$$R_{83} = 2P_{89} \cdot \sin \left( \frac{\nu}{2} - \psi \right) - U_8 \cos \left[ \xi + \left( 90 - \frac{\nu}{2} \right) \right].$$  \hspace{1cm} (39)

Since the masses of the intermediate gearwheels are not significant in comparison with the masses of the shafts and the pressure forces, the second term can be excluded from formula (39), then

$$P_{83} = 2P_{89} \cdot \sin \left( \frac{\nu}{2} - \psi \right).$$  \hspace{1cm} (40)
Formula (40) shows that with an increase in the angle between the levers \( \nu \), the reaction on the bearings increases.

In order to decrease angle \( \nu \), it is necessary to increase angle \( \varphi_n \) since

\[
\nu = 180 - \varphi_n \tag{41}
\]

3. Conclusions
Dependences were obtained for determining the force characteristics of a ten-link tooth-lever differential transmission mechanism. Formulas for determining the reaction force in bearings and links were derived. Dependencies for determining the balance moment were obtained. Cases of unloading and overloading of pressure on the processed material from the side of the free shaft are shown, depending on the location of the transmission mechanism relative to the motion of the processed material. The reaction of intermediate wheels on their own axes of rotation depending on the angle between the levers was shown. As the angle between the levers increases, the response on the axes of rotation increases.

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