KINETOSTATIC ANALYSIS OF RATCHET MECHANISM THAT ACTS ON CONVEYOR OF MANURE SPREADING MACHINE

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Abstract. The paper presents the kinetostatic analysis of the ratchet mechanism that acts on the conveyor, which is a part of solid organic fertilizer spreader, MGL-3, namely the reactions in the kinematic couplings as well as the driving torque in the mechanism drive coupling have been determined. The kinetostatic analysis of a mechanism is based on its kinematic analysis. In this respect, both the structural and kinematic analysis of the mechanism are necessary. Structural analysis allows determining the components that the mechanism is made of. Kinematic analysis provides us with the positions, velocities and accelerations of the application points of forces acting on the mechanism elements. The paper presents the hodograph of reactions in the mechanism drive coupling as well as the driving torque variation diagram. The research can improve the construction of the ratchet mechanism.

Keywords: ratchet system, torque variation, manure spreading machine.

Introduction

According to the research conducted by the researchers involved in the project, it is considered that the ratchet mechanism influences, by its characteristics, the manure quantity spreading in the orchards, providing the necessary rate of the organic fertilizers [1-5]. A good dimensioning of this mechanism can influence the drag chain conveyor functionality and can assure good working of the machine. The ratchet and pawl mechanism plays a crucial role in providing one way transmission and safety against heavy loading conditions [6; 7].

In the structural and kinematic analysis of the ratchet mechanism that acts on the chain conveyor of a manure spreader, methods for solving mechanisms and also numerical methods have been used [8-20]. In order to write the equations necessary for the mechanism analysis, the independent contours method was used [6; 8-10]. To solve the systems of positions' non-linear equations, Newton-Raphson and gradient methods were used, while to solve the systems of linear equations of velocities and accelerations the Gaussian elimination method was used [9; 11; 12].

Materials and methods

1. Mechanism kinematic analysis

Kinematic analysis of the ratchet mechanism consists in determining the parameters of positions, velocities and accelerations, corresponding to all elements. For this, the initial position of the mechanism is defined (Fig.1), **DiadaRRR.m** and **GrupaMotR.m** functions applied, drawn up by the authors in the MATLAB syntax, to determine the kinematic parameters and the diagrams of the determined parameters are drawn. Using the methodology described above, an analysis of the mechanism that is a part of the manure spreader, MGL-3, was made in order to optimize it.

In Fig. 1 the kinematic scheme of the entire mechanism is presented, highlighting the position parameters.

For the analysis of the mechanism the following are known:

1. kinematic scheme of the mechanism;
2. elements' dimension and positions of couplings adjacent to the base, as follows: AB = 0.060 m, BC = 0.300 m, CD = 0.182 m, XA = 0.0 m, YA = 0.0 m.
3. mechanism initial position: $\varphi_1 = \varphi_{10}$ = 0.4617 rad (Fig. 1);
4. angular velocity of the element 1: ω_1 = 3.58979 s^{-1};
5. angular acceleration of the element 1: ε_1 = 0.0 s^{-2}.

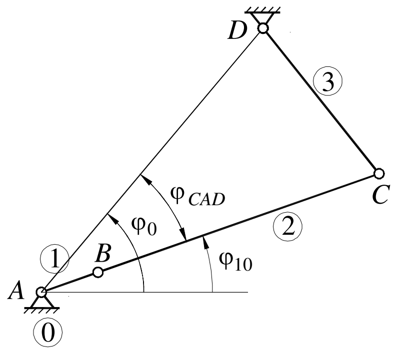

Fig. 1. **Defining initial position of mechanism**

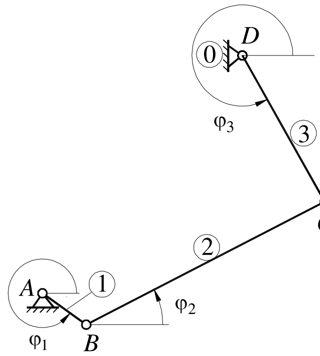

Fig. 2. **Defining position kinematic parameters of mechanism**

The calculations were made for 36 equidistant positions of element 1, starting from the initial position: $\varphi_1 = \varphi_{10} = 0.461696$ rad and for 2 positions of couple A: *XD*(1) = 0.170 m, *XD*(2) = 0.145 m *YD*(1) = 0.260 m; *YD*(2)=0.236.

2. Kinetostatic analysis of the mechanism

In the present paper, the kinetostatic analysis of the ratchet mechanism is made, the reactions of the kinematic couplings are determined as well as the actuation moment of the mechanism driving coupling. The kinetostatic analysis of a mechanism is based on its kinematic analysis. In this respect, the structural analysis of the mechanism is also required. The structural analysis allows determining the modules the mechanism is made of. The kinematic analysis provides the application points' positions, velocities and accelerations of the forces acting on the mechanism elements. The paper presents the hodograph of reactions in the mechanism driving coupling as well as the driving torque variation diagram.

The kinetostatic analysis of the mechanism is done by means of calculation procedures corresponding to each modular group of the mechanism. The calculation procedures were written in the MATLAB syntax [10; 12].

***Dyad RRR*(2, 3)**

On the dyad elements act (see Fig. 3.*a*):

- forces of weight: $\overline{G}_2 = -m_2\overline{g}$ and $\overline{G}_3 = -m_3\overline{g}$;
- resulting inertia forces: $\overline{F}_{i2} = -m_2\overline{a}_{G2}$ and $\overline{F}_{i3} = -m_3\overline{a}_{G3}$;
- resultant moments of inertia forces: $\overline{M}_{i2} = -I_{G_2}\cdot\overline{\varepsilon}_2$ and $\overline{M}_{i3} = -I_{G_3}\cdot\overline{\varepsilon}_3$;
- resistant torque of the coupling *D*: M_R.

Reduction points of the forces systems are considered in the gravity centres G_2 and G_3, of elements 2 and 3. Kinematic parameters of these points are calculated using the function **A1R.m**.

Acceleration of reduction points G_2 and G_3, being known by components on the coordinate axis, results that the inertia forces have the form

$$\overline{F}_{i2} = F_{i2X}\,\overline{i} + F_{i2Y}\,\overline{j}\ ,\ \overline{F}_{i3} = F_{i3X}\,\overline{i} + F_{i3Y}\,\overline{j}\,. \tag{1}$$

Resultants of the applied forces, of inertia and weight, are of the form

$$\overline{F}_{R2} = F_{2X} + F_{2Y}\ ,\ \overline{F}_{R3} = \overline{F}_{3X} + \overline{F}_{3Y} \tag{2}$$

or

$$\overline{F}_{R2} = F_{2X}\,\overline{i} + F_{2Y}\,\overline{j}\,,\ \overline{F}_{R3} = F_{3X}\,\overline{i} + F_{3Y}\,\overline{j}\,. \tag{3}$$

The projections on the axes of the applied inertia and weight forces resultants are:

$$F_{2X} = -m_2\cdot a_{G_2X}\,;\ F_{2Y} = -m_2\cdot(a_{G_2Y} + g)\,, \tag{4}$$

$$F_{3X} = -m_3 \cdot a_{G_3X}\ ;\ F_{3Y} = -m_3 \cdot (a_{G_3Y} + g)\ . \tag{5}$$

In report with reduction points, resultants moments are:

$$CM_2 = -I_{G_2} \cdot \ddot{\varphi}_2\ ;\ CM_3 = -I_{G_3} \cdot \ddot{\varphi}_3 + M_R\ .$$

The reactions from kinematic couplings *B*, *C* and *D* are forming the output data of the function **D1RC.m**.

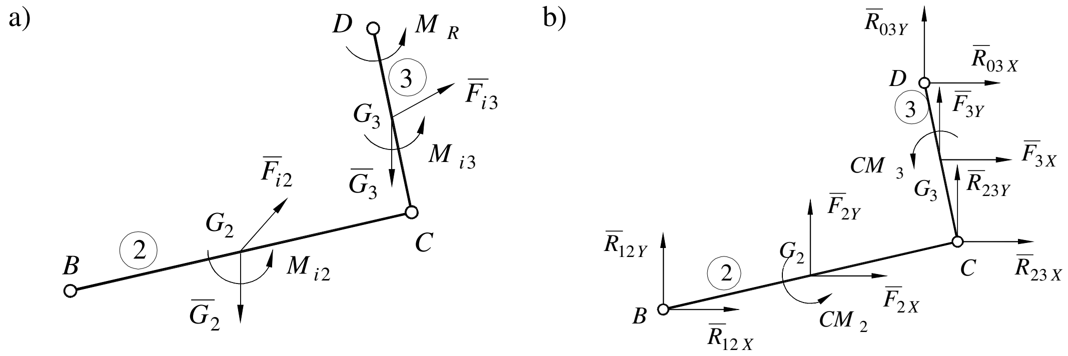

Fig. 3. **Dyad *RRR*(2, 3):** a – highlighting forces and moments acting on elements of dyad; *b* – kinetostatic scheme of dyad

***Motor group R*(1)**

On the motor group elements *R*(1) acts (Fig. 4.*a*):

- forces of weight $\overline{G}_1 = -m_1\overline{g}$;
- resulting inertia forces: $\overline{F}_{i1} = -m_1\overline{a}_{G1}$;
- resultant moments of inertia forces: $\overline{M}_{i1} = \overline{0}$ (was considered $\varepsilon_1 = 0$);
- reactions of element 2, of dyad *RRR*(2,3), on the element 1 of motor group *R*(1), namely: $\overline{R}_{21X} = -\overline{R}_{12X}$, $\overline{R}_{21Y} = -\overline{R}_{12Y}$.

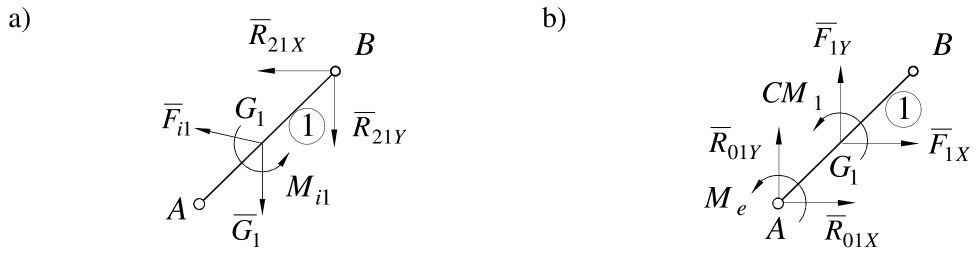

Fig. 4. **Motor group *R*(1):** a – highlighting forces and moments acting on elements of group; b – kinetostatic scheme of motor group

Weight and reactions forces resultant is by the form

$$\bar{F}_{R1} = \overline{F}_{1X} + \overline{F}_{1Y} \tag{6}$$

or

$$\bar{F}_{R1} = F_{1X}\ \bar{i} + F_{1Y}\ \bar{j}\ . \tag{7}$$

The projections on the axes of the resultant forces are:

$$F_{1X} = -R_{12\mathrm{X}}\ ;\ F_{1Y} = -m_1 \cdot g - R_{12\mathrm{Y}}\ . \tag{8}$$

In report with the reduction point *A*, the resultant moment is:

$$CM_1 = R_{12X}\,(YB - YA) - R_{12Y}\,(XB - XA)\ . \tag{9}$$

Reactions from active coupling *A*, as well as equilibration moment M_e are forming the output data of the procedure **A1RRC.m**.

Equilibration moment (motor moment) in active coupling *A* can be calculated, also, with the equation of virtual power, namely:

$$\sum \overline{P}\cdot\overline{v} = \overline{M}_{e1}\cdot\overline{\omega}_1 + (\overline{F}_{i1}+\overline{G}_1)\cdot\overline{v}_{G1} + (\overline{F}_{i2}+\overline{G}_2)\cdot\overline{v}_{G2} + (\overline{F}_{i3}+\overline{G}_3)\cdot\overline{v}_{G3} + \\ + \overline{M}_{i1}\cdot\overline{\omega}_1 + \overline{M}_{i2}\cdot\overline{\omega}_2 + \overline{M}_{i3}\cdot\overline{\omega}_3 + \overline{M}_R\cdot\overline{\omega}_3 = \overline{0} \quad . \tag{10}$$

Using the upper relation, results:

$$M_{e1} = (m_1(a_{G1X}\cdot v_{G1X} + (a_{G1Y}+g)\cdot v_{G1Y} + I_{G_1}\cdot\omega_1\cdot\varepsilon_1 + \\ + m_2(a_{AG2X}\cdot v_{VG2X} + (a_{AG2Y}+g)\cdot v_{BY} + I_{G_2}\cdot\omega_2\cdot\varepsilon_2 + \\ + m_3(a_{G3X}\cdot v_{G3X} + (a_{G3Y}+g)\cdot v_{G3Y} + I_{G_3}\cdot\omega_3\cdot\varepsilon_3 - M_R)/\omega_1. \tag{11}$$

Besides the kinematic scheme, for the kinetostatic analysis of mechanism, are also known:

- masses of mechanism elements: m_1 = 2.81 kg, m_2 = 2.0 kg, m_3 = 2.5 kg;
- inertia moment of elements, in relation to an axis perpendicular to the plane of movement and passing through the centre of mass: I_{G_1} =0.005 kgm^2, I_{G_2} = 0.013 kgm^2, I_{G_3} = 0.007 kgm^2;
- resistant moment in coupling *D*: $M_R = 2600$ Nm.

Results

Based on the above data a calculation program was drawn up for determining the reactions from kinematic couplings of the mechanism, as well as of the actuation moment of it. In the main program the calculation procedures mentioned above were called.

Table 1 shows the components of the reactions in coupling *A*, as well as the driving torque calculated by the kinetostatic method and by the method of virtual power.

Table 1

Reactions and the driving torque in coupling *A*

Pos.	φ_1, rad	***R01X***, N	***R01Y***, N	***Me_CT***, Nm	***Me_PV***, Nm	***R01X***, N	***R01Y***, N	***Me_CT***, Nm	***Me_PV***, Nm
1	0.64	-15342.98	-6903.16	213.16	213.16	-17377.49	-8641.07	246.91	246.91
2	0.81	-15796.71	-6359.09	423.55	423.55	-18049.47	-8017.63	495.80	495.80
5	1.33	-16275.66	-4357.40	887.94	887.94	-18655.45	-5279.45	1035.52	1035.52
6	1.51	-16124.47	-3750.12	951.64	951.64	-18355.03	-4378.29	**1095.11**	**1095.11**
7	**1.68**	-15866.85	-3258.08	**968.04**	**968.04**	-17891.14	-3627.99	1094.18	1094.18
8	1.86	-15547.82	-2907.89	944.32	944.32	-17340.23	-3070.06	1044.78	1044.78
13	2.73	-14081.76	-3189.32	513.72	513.72	-14875.50	-2896.56	486.72	486.72
14	2.91	-13887.56	-3543.98	402.71	402.71	-14537.19	-3214.08	360.63	360.63
16	3.25	-13566.69	-4407.86	172.13	172.13	-13976.70	-4015.43	111.92	111.92
17	3.43	-13420.57	-4881.22	53.61	53.60	12.57	41.55	-1.34	-1.34
23	4.48	9.08	44.08	0.10	0.10	12.35	45.73	0.35	0.35
25	4.83	9.54	45.23	0.78	0.78	12.75	47.09	1.06	1.06
35	6.57	14.97	45.63	1.58	1.58	19.19	48.79	1.62	1.62

In Fig. 5 the hodograph of the reaction forces in coupling *A* is presented.

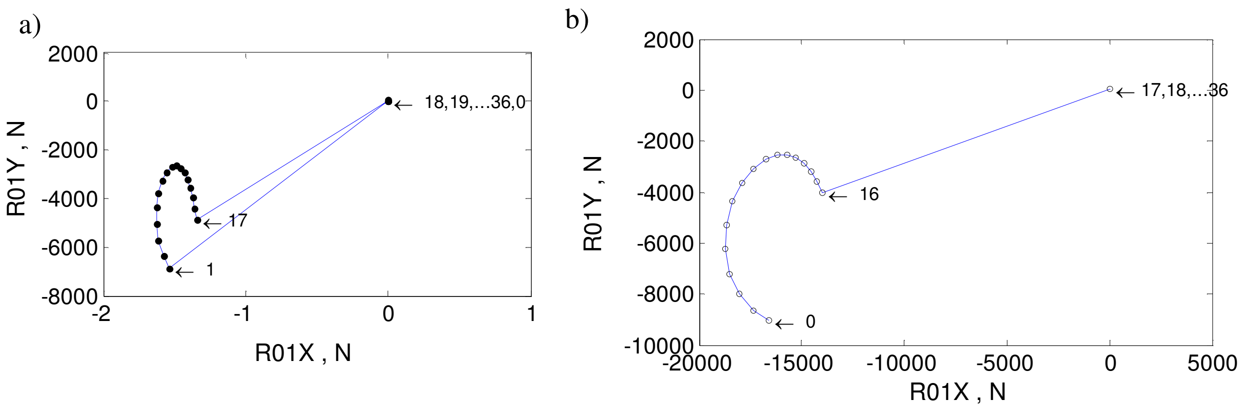

Fig. 5. **Hodograph reaction forces in coupling *A***

In figure 6 the torque variation diagram in coupling *A* is presented.

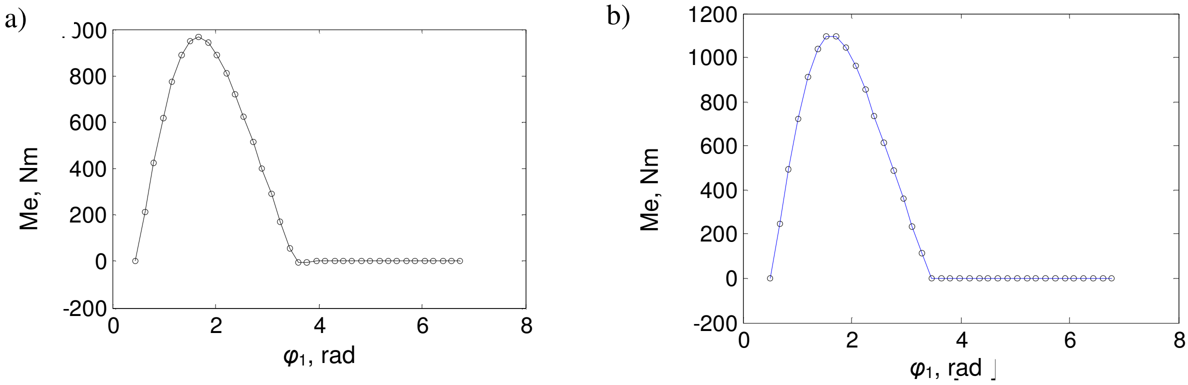

Fig. 6. **Torque variation diagram in coupling *A***

Discussions

- In order to make the kinetostatic analysis of the ratchet mechanism, it was necessary to carry out, in the first phase, the structural and kinematic analysis. In the structural analysis, the modular groups the mechanism is made of were determined.
- Within the kinematic analysis, the kinematic parameters of the mechanism elements, as well as of the elements mass centres, were determined.
- Considering the resistant moment of the working beam, as well as the masses and inertia moments of the mechanism components, the reactions from the kinematic couplings and the driving torque in the driving coupling *A*, were determined. The driving torque was calculated by the kinetostatic method and by the method of virtual power. The results of the calculations were presented in tabular and graphic form.
- In Figure 6 it is shown that the driving torque reaches 100 daNm at a certain position of element **3**, when element **1** is at an angle of 2 rad, for a load of the scraper conveyor with a mass of 3000 kg of manure. This allows carrying out some calculations to verify the necessary power from the PTO shaft of the tractor to determine what tractor must be used in the working aggregate.

Conclusions

Table 1 shows that the values of the driving torque, calculated by the two methods, are the same, which means that the whole dimensioning calculation of the mechanism has been well realized.

For the torque to be minimal, the position and length of the mechanism elements must be optimized (the mechanism must operate at minimum pressure angles).

Analysing the torque value in the driving coupling *A*, for two positions of the driven coupling *D* (*XD*(1), *YD*(1) respectively *XD*(2), *YD*(2)), we can see that the values are different (see Table 1 and

Fig.6), which confirms the hypothesis that the relative position of the *A* and *D* couplings influences the torques and reactions in the driving coupling *A*.

Identification of optimal positions of the coupling *A* and *D* allows efficient choice of the engine torque. In the first position of the coupling *D*, the torque reaches values less than 100 daNm, and in the second position, the torque reaches values over 100 daNm.

Also, the position between the couplings *A* and *D* can influence the reactions in these couplings.

Simulating the process in Mathlab soft, offers the possibility to reduce the designing process and testing researches can be improved.

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