KINEMATIC STUDY OF A SIX-LINK MECHANISM FOR THE PURPOSE OF ITS USE IN THE DESIGN OF NEW MACHINES AND MECHANISMS

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Annotation. This article presents a kinematic analysis of a six-link mechanism aimed at its application in the design of new machines and mechanisms. Kinematic calculation, especially for multi-link mechanisms, is an essential stage in the development of modern mechanical systems. The study determines linear velocities, accelerations, angular velocities, and angular accelerations of the mechanism’s points and links. The relationships between motion parameters and kinematic characteristics are also analyzed. The obtained results provide a theoretical basis for designing efficient and reliable new machines and mechanisms.

Keywords: mechanism, kinematics, velocity, acceleration, rotation, linkage, calculation, parameter, design, dynamics

Annotatsiya. Ushbu maqolada olti zvenoli mexanizmning kinematik tadqiqi va uning yangi mashina hamda mexanizmlarni loyihalashdagi ahamiyati ko‘rib chiqiladi. Kinematik hisoblash, ayniqsa ko‘p zvenoli mexanizmlar uchun, yangi texnik qurilmalarni yaratishda zarur bosqich hisoblanadi. Tadqiqot jarayonida mexanizm nuqtalari va zvenolarining tezligi, tezlanishi, burchak tezligi hamda burchak tezlanishi

aniqlanadi. Shuningdek, harakat qonuniyatlari va parametrlarning o‘zaro bog‘liqligi tahlil qilinadi. Olingan natijalar yangi mashina va mexanizmlarni samarali hamda ishonchli loyihalash uchun nazariy asos bo‘lib xizmat qiladi.

Tayanch so‘zlar: mexanizm, kinematika, tezlik, tezlanish, aylanish, dinamika, zveno, parametr, loyiha, hisob

Аннотация. В статье рассматривается кинематическое исследование шестизвенного механизма с целью его использования при проектировании новых машин и механизмов. Кинематический расчет, особенно многозвенных механизмов, является необходимым этапом при создании современной техники. В процессе анализа определяются скорости, ускорения, угловые скорости и угловые ускорения точек и звеньев механизма. Также исследуются взаимосвязи кинематических параметров и закономерности движения системы. Полученные результаты могут быть использованы при разработке и совершенствовании новых машин и механизмов.

Ключевые слова: механизм, кинематика, скорость, ускорение, вращение, звено, расчет, параметр, проектирование, динамика

Kinematic calculations, especially of multi-link mechanisms, are essential when designing and creating new machines and mechanisms. Therefore, it is necessary to determine the speed, acceleration, angular velocity, angular acceleration, and other parameters of points, links, etc.

For this purpose, as an example, we perform a kinematic analysis of the planing machine mechanism shown in figure 1, a. The planing machine mechanism is designed in such a way that the radius of the driving link can be changed in order to change the stroke of the actuator link depending on the length of the workpiece. The following parameters are given to determine the kinematic parameters: $\varphi_1 = 300^o$, $l_{AB} = 0{,}06\, m$, $l_{AC} =$

$0{,}15\ m,\ l_{CD} = 0{,}22\ m,\ l_{DE} = 0{,}08\ m,\ H = 0{,}11\ m$. The angular velocity of the crank AB is constant and equal to $\omega_1 = 12\ c^{-1}$ [1, 2]

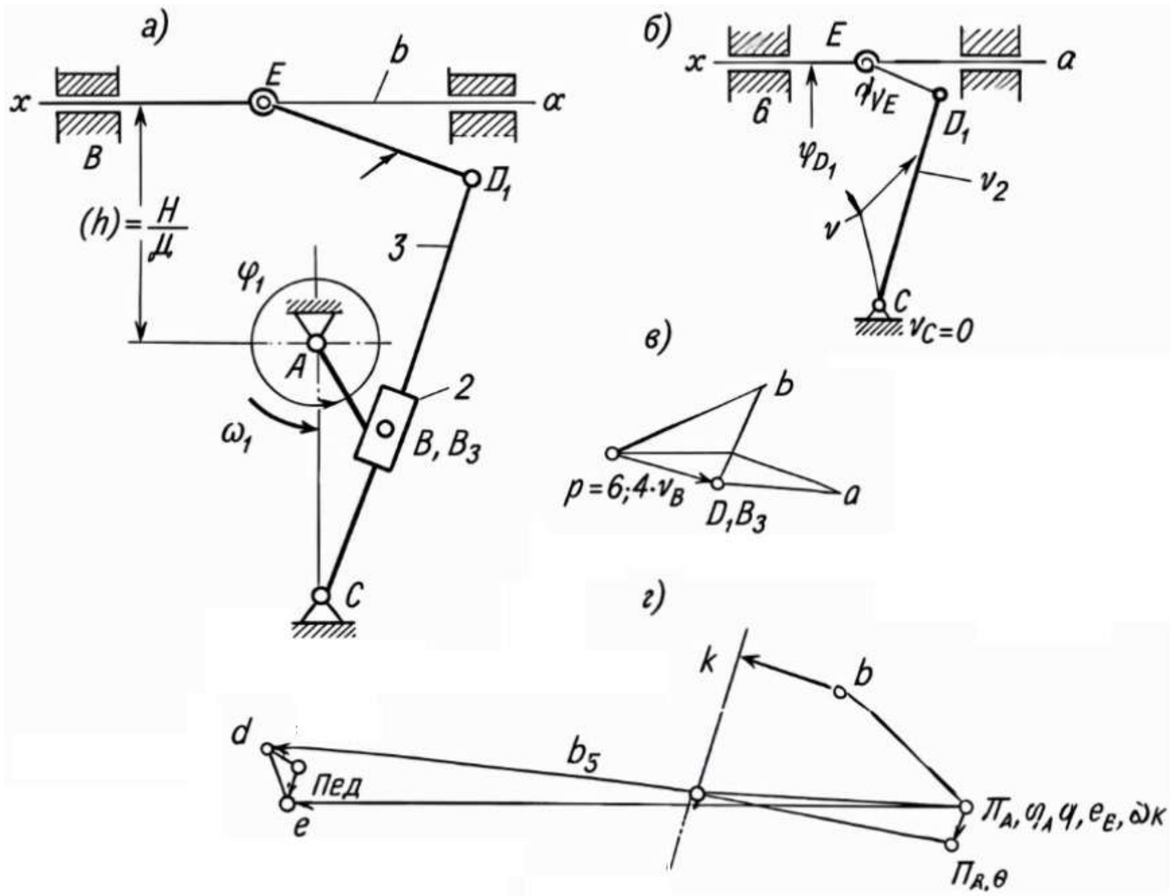

Figure 1. Kinematic analysis of the planing machine mechanism;

a) mechanism diagram, b) position plan, c) speed plan, d) acceleration plan

Solution. 1) We draw a plan of the mechanism's position. We choose the length of the segment (*AB*) equal to 30 *mm*, so the scale of the diagram will be [3]

$$\mu_l = \frac{l_{AB}}{(AB)} = \frac{0{,}06}{30} = 0{,}002\ \ \frac{m}{mm}$$

The lengths of the remaining segments in the drawing:

$$(AC) = \frac{l_{AC}}{\mu_l} = \frac{0{,}15}{0{,}002} = 75\ \ mm,$$

$$(CD) = \frac{l_{CD}}{\mu_l} = \frac{0{,}22}{0{,}002} = 110\ \ mm,$$

$$h = \frac{H}{\mu_l} = \frac{0{,}11}{0{,}002} = 55\ mm, \qquad (DE) = \frac{l_{DE}}{\mu_l} = \frac{0{,}08}{0{,}002} = 40\ mm$$

Using the obtained dimensions, we construct a plan of the mechanism's position (Fig. 1, b).

2) We construct a plan of the mechanism's speeds. We begin with the group consisting of links 2 and 3, since it is directly connected to the drive link and the rack. We construct the plan using the following vector equations [4, 5]:

$$v_{B_3} = v_B + v_{B_3B}, \qquad v_{B_3} = v_C + v_{B_3C}$$

where $v_{B_3} -$ is the velocity of point B_3 of link 3, which lies above point *B*; v_B is the velocity of point B, equal in magnitude to $v_B = \omega_1 \cdot l_{AB} = 12 \cdot 0{,}06 = 0{,}72\ m/s$ and directed perpendicular to *AB* in accordance with the direction of the angular velocity ω_1; $v_{B_3B} -$ is the velocity of point B_3 relative to point *B*, directed parallel to line *BC*; $v_C -$ is the velocity of point *C*, equal to zero; $v_{B_3C} -$ is the velocity of point *B* due to rotation of link 3 relative to point *C*, equal in magnitude to $v_{B_3C} = \omega_3 \cdot l_{B_3C}$ and directed perpendicular to *BC* (unknown to us yet) [6, 7].

We construct a solution to the first vector equation indicated above. From the pole p (Fig. 1, a) we set aside a segment (*pb*), depicting the velocity v_B of point *B*. The length of this segment is taken equal to (*pb*) = (*AB*) = 30 *mm*, *i.e*. the plan is constructed in the scale of the crank. Through point b we draw the direction of the velocity v_{B_3B} a line parallel to CB_3. We proceed to constructing a solution to the second vector equation indicated above. It is necessary to set aside the velocity vector of point *C*, but since its modulus is equal to zero, then we place its end c at the pole of the plan p and from point p we draw the direction of the velocity $v_{B_3C} -$ a line perpendicular to *CB*. Its intersection with the previously drawn line parallel to *CB* gives the end of the velocity vector $v_{B_3} -$ point b_3. Point *d* – the end of the velocity vector of point *D* – is found according to the similarity rule from the relation

$$\frac{(cd)}{(cb_3)} = \frac{(CD)}{(CB_3)},$$

where

$$(cd) = (cb_3)\frac{(CD)}{CB_3} = 18\frac{105}{42} = 45\ mm,$$

Let’s move on to constructing the speed plan for group 4, 5. We construct this plan using the equations [8]

$$v_E = v_D + v_{ED}, \qquad v_E = v_{E_6} + v_{EE_6},$$

where v_E is the velocity of point E; v_D is the velocity of point D (its vector is plotted on the velocity plane as a segment (pd)); v_{ED} is the velocity of point E during rotation of link 4 relative to point D, equal in magnitude to $v_{ED} = \omega_4 \cdot l_{DE}$ and directed perpendicular to line DE (which is unknown to us at this point); v_{E_6} – is the velocity of point E_6 of link 6, which is coincident with point E (its modulus is zero since link 6 is motionless); v_{EE_6} – is the velocity of point E relative to point E_6, directed parallel to line xx. The construction is reduced to drawing through point d (according to the first equation) a line perpendicular to DE, *i.e*. to the direction of velocity v_{ED}, and drawing through point p (according to the second equation) a line parallel to xx. The point e of intersection of these lines is the end of the velocity vector v_E of point E. We place at points c, e_6, a and at this point we complete the construction of the velocity plan of the mechanism [9, 10].

The scale of the speed plan is

$$\mu_v = \frac{v_B}{(pb)} = \frac{\omega_1 (AB)\mu_l}{(pb)} = \omega_1 \mu_l = 12 \cdot 0{,}002 = 0{,}024\ \frac{m/s}{mm}$$

The scale of the plan of speed analogues is

$$\mu_{\varphi_v} = \frac{\mu_v}{\omega_1} = \mu_l = 0{,}002\ \frac{m}{mm},$$

The required speed of the support (speed of point E) is equal to

$$v_E = (pe)\mu_v = 48 \cdot 0{,}024 = 1{,}152\ \frac{m}{s},$$

3) We construct the acceleration plan for group 2, 3. We construct it using the following two vector equations [11, 12]

$$a_{B_3} = a_B + a_{B_3B}^k + a_{B_3B}^r, \quad a_{B_3} = a_C + a_{CB_3}^n + a_{CB_3}^t,$$

where a_{B_3} − is the acceleration of point B_3, which belongs to link 3 and coincides with point B of link 1; a_B − is the normal (aka total) acceleration of point B, equal in magnitude to $a_B = \omega_1^2 \cdot l_{AB} = 12^2 \cdot 0{,}06 = 8{,}64\ m/s^2$ and directed parallel to AB from point B to point A; $a_{B_3B}^k$ − is the Coriolis acceleration in the motion of point B_3 relative to link 2, equal in magnitude to [13, 14]

$$a_{B_3B}^k = 2\omega_2 v_{B_3B} = 2\frac{v_{B_3C}}{l_{BC}} v_{B_3B}$$

(since $\omega_2 = \omega_3$ and $\omega_3 = \frac{v_{B_3C}}{l_{BC}}$) and having the direction of the relative velocity vector v_{B_3B}, rotated by 90° in the direction of the angular velocity ω_2 of the translational motion (motion of link 2); $a_{B_3B}^r$ − relative (relative) acceleration of point B_3 relative to point B, directed parallel to line CB; a_C − acceleration of point C (it is equal to zero); $a_{B_3C}^n$ − normal acceleration of point B_3 in the rotation of link 3 relative to point C, equal in magnitude [15, 16]

$$a_{B_3C}^n = \frac{v_{B_3^2C}}{l_{B_3C}}$$

directed parallel to line CB_3 from point B_3 to point C; $a_{B_3C}^t$ − is the tangential acceleration of point B_3 in the same motion of link 3, equal in magnitude to $a_{B_3C}^t = \varepsilon_3 l_{B_3C}$ (not yet known to us) and directed perpendicular to CB_3.

We construct the solution to the first vector equation indicated above (Fig. 1, d). We define the segment (πb) = (AB) = 30 *mm*, which depicts the acceleration a_B in the plan (since (πb) = (AB), the plan is constructed in the crank scale).

The scale of the acceleration plan is [17]

$$\mu_a = \frac{a_B}{(\pi b)} = \frac{\omega_1^2 (AB)\mu_l}{(\pi b)} = \omega_1^2 \cdot \mu_l = 12^2 \cdot 0{,}002 = 0{,}288 \ \frac{m/s^2}{mm}$$

The scale of the acceleration plan is

$$\mu_{\varphi_a} = \frac{\mu_a}{\omega_1^2} = \mu_l = 0{,}002 \ \frac{m}{mm}$$

We set aside the selected segment (πb) from the pole of the plane (π), then add to it the segment (bk) – the Coriolis acceleration vector – we find its length using the formula

$$(bk) = \frac{a_{B_3B}^k \cdot v_{B_3B}}{v_{B_3C}\mu_a} = \frac{2(b_3c)\cdot(bb_3)\mu_v^2}{(B_3C)\mu_l \cdot \mu_a} = \frac{2\cdot 18\cdot 21}{42{,}5} = 17{,}79 \ mm$$

The segments (b_3 c) = 18 *mm* and (bb_3) = 21 *mm* are taken from the velocity plane, and the segment (B_3 C) = 42.5 *mm* is taken from the position plane. Through point *k*, we draw a line parallel to *CB*, in the direction of acceleration $a_{B_3B}^r$ [18, 19].

We proceed to construct the second vector equation. We align point c with point π, since $a_C = 0$. From point π, we plot a segment (πn_{B_3C}), representing the normal acceleration $a_{B_3C}^n$, its length is

$$\left(\pi n_{B_3C}\right) = \frac{v_{B_3C}^2}{l_{B_3C}\mu_a} = \frac{(\pi b_3)^2\mu_v^2}{(B_3C)\mu_l \cdot \mu_a} = \frac{18^2}{42{,}5} = 7{,}62 \, mm$$

then, through the point n_{B_3C}, we draw the direction of acceleration $a_{B_3C}^t -$ a line perpendicular to *CB*, until it intersects with the line previously drawn through point *k*, parallel to *CB*. The intersection point b_3 represents the end of the acceleration vector a_{B_3}. We find the end of the acceleration vector of the hinge center *D* (point *d*) using the similarity rule from the relation

$$(\pi d) = (\pi b_3)\frac{(CD)}{(B_3C)} = 37 \cdot \frac{104}{42{,}5} = 90{,}54 \ mm,$$

Let’s move on to constructing the acceleration plan for group 4, 5 using the equations

$$a_E = a_D + a_{ED}^n + a_{ED}^t, \qquad a_E = a_{E_6} + a_{EE_6}^k + a_{EE_6}^r$$

where a_E is the acceleration of point E; a_D is the acceleration of point D (it is determined by the previously constructed segment (πd)); $a_D = (\pi d)\mu_a = 90{,}54 \cdot 0{,}288 = 26{,}08$ м/с^2; $a_{ED}^n = \frac{v_{ED}^2}{l_{ED}}$ – is the normal acceleration of point E due to the rotation of link 4 relative to point D (it is directed parallel to the line ED from point E to point D); $a_{ED}^t = \varepsilon_4 l_{ED}$ – is the tangential acceleration of the same point in the same motion of link 4 (it is directed perpendicular to the line ED); a_{E_6} – is the acceleration of point E_6, which belongs to link 6 and is coincident with point E (it is zero); $a_{EE_6}^k$ – is the Coriolis acceleration of point E in its motion relative to the support (point E_6; it is zero); $a_{EE_6}^r$ – is the relative acceleration of point E relative to the support (point E_6; it is directed parallel to the line xx).

According to the first vector equation, we plot a segment (dn_{ED}) from point d, representing the normal acceleration a_{ED}^n. Its length is

$$dn_{ED} = \frac{v_{ED}^2}{l_{ED}\mu_a} = \frac{(ed)^2\mu_v^2}{(ED)\mu_l \cdot \mu_a} = \frac{15^2}{42{,}5} = 5{,}29 \ mm,$$

Next, through the point n_{ED}, we draw the direction of acceleration a_{ED}^t (a line perpendicular to ED) and proceed to the constructions corresponding to the second vector equation indicated above. At the point π, we place the points e_6 and k', since the acceleration moduli a_{E_6} and $a_{EE_6}^k$ are equal to zero. From the point π, we draw the direction of acceleration $a_{EE_6}^r$ (a line parallel to xx) until it intersects with the line previously drawn from the point n_{ED}. The point of intersection e is the end of the acceleration vector of point E, i.e., the acceleration a_E. We place the point a at the pole of the plan and this completes the construction of the acceleration plan of the mechanism [20, 21].

The required acceleration of the support (point E) will be equal to

$$a_E = (\pi e)\mu_a = 87 \cdot 0{,}24 = 20{,}88 \ m/s^2$$

The completed kinematic calculation will be used in the design of new mechanisms and machines, and will also be useful to students, masters, and scientific researchers in completing coursework and research projects.

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