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SIMULATION OF EQUATION OF MOTION OF THE NEW CONSTRUCTION GIN MACHINE

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Abstract: The article analyzes the moving mechanisms of the new gin design, creates kinematic diagrams of the feed rollers, peg drum, saw cylinder shafts and auger shaft for removing dead fiber. Based on the kinematic diagrams, dynamic models and equations of motion of the feed rollers, peg drum, saw cylinder shafts and auger shaft are created.

Keywords: saw cylinder, auger, peg drum, feeder, kinematics, dynamic model, equation of motion.

Introduction. Energy-efficient, high-efficiency technologies are widely used worldwide. Based on the results of the analysis of the energy efficiency of the republic's network enterprises, it requires increasing the energy efficiency of enterprises and introducing innovations into practice. Taking into account this aspect, cotton primary processing enterprises are among the enterprises that consume the most electricity and natural resources in our Republic. Regeneration of advanced and up-to-date technologies, as well as the increase of energy-efficient, high-security technologies and the use of these devices are of great importance in them. A new energy-efficient high-performance full-size engine for the introduction of high-efficiency technology. For the implementation of the experimental method, the theoretically defined kinematic scheme of the dynamical model and the structural equation of the mechanism were determined. Uneven rotation of the saw cylinder can lead to deterioration of the ginning process and fiber damage. Using Lagrange's second equation, we construct the motion equations of the machine block of the saw cylinder to determine the non-uniformity of rotation of the saw cylinder and reduce power consumption [1]. The saw gin of the new design is designed to separate the fiber from the seed of cotton raw materials with a medium moisture content of 7-9% in the technological process. The kinematic diagrams of the ginning machine of the new construction are shown figure-1[2].

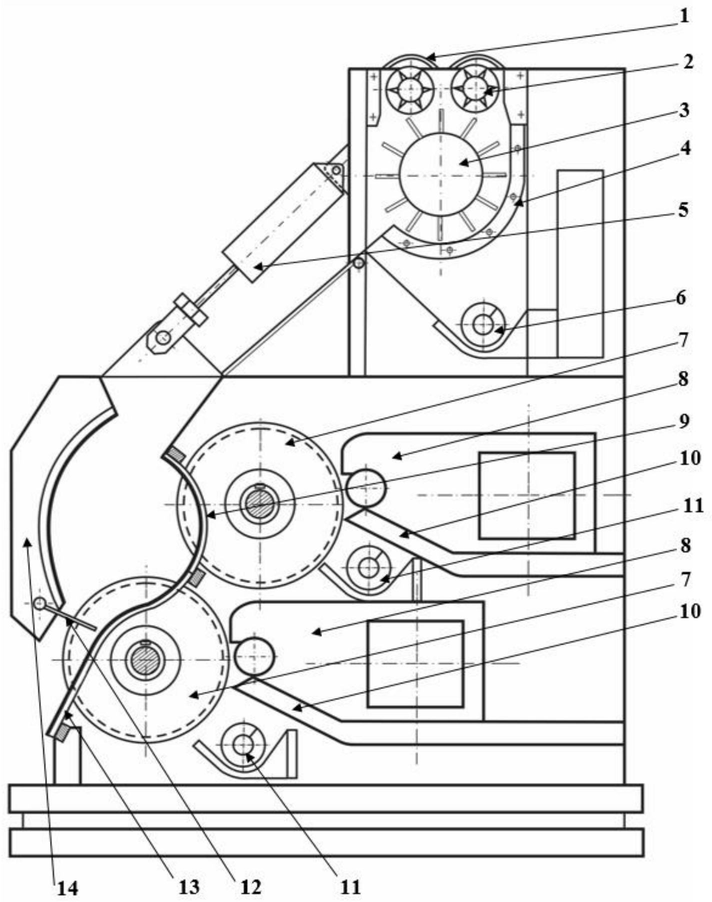

Figure 1. Schematic of the new construction gin machine
1-gear, 2-feeder, 3-pile drum, 4-sieve, 5-pneumo cylinder, 6-screw for dirty mixture, 7-saw cylinder, 8-air blowing system, 9- upper column, 10- fiber pulling system, 11- auger for fiber, 12- comb, 13 - bottom column, 14- working chamber

Research materials and methods. We model the kinematics of the rollers and pile drum that provide the gin machine of the new construction (Fig. 2).

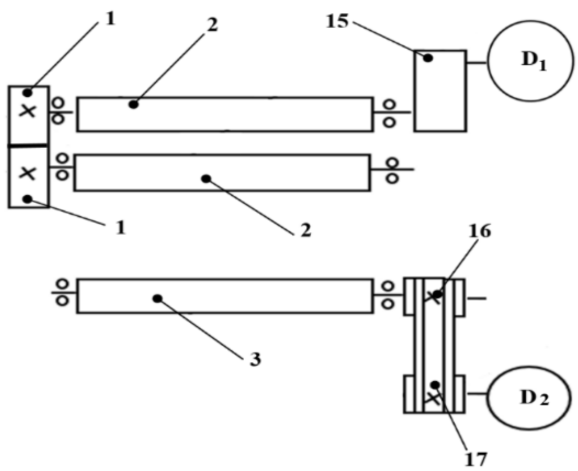

Figure 2. Rollers and pile drum kinematics that provide a new construction gin machine

D1 - supply roller electric motor, D2 - pile drum electric motor, 1- gear, 2-supply, 3-pile drum, 15- reducer, 16- belt transmission drum pulley, 17- belt transmission electric pulley

We model the kinematics of saw cylinder shafts of the new construction gin machine (Fig. 3).

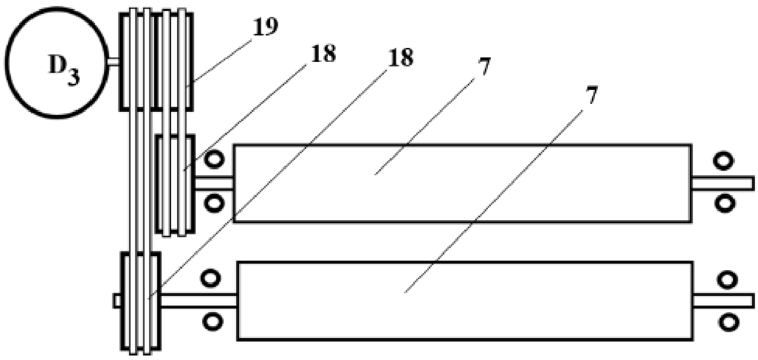

Figure 3. Kinematics of saw cylinder shafts of a new construction gin machine
D3 - saw cylinder electric motor, 7-saw cylinder, 18-belt transmission saw cylinder pulley, 19-belt transmission electric motor pulley

Let's model the kinematics of the gin machine of the new construction of the dead fiber removal snack (Fig. 4).

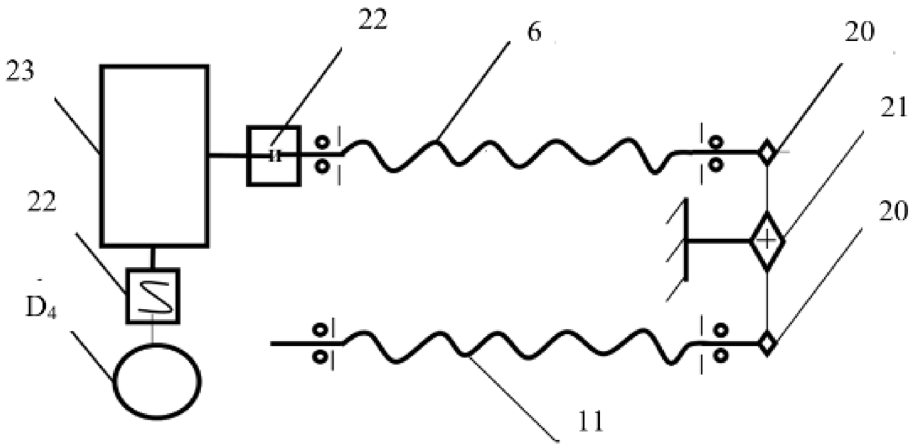

Figure 4. Sneak cinematics of the new build of the demon machine
D4 - auger electric motor 6- auger for dirty mixture, 11- auger for fiber, 20- chain transmission, 21 - chain tensioning star, 22- gear, 23- reducer.

Solution and results. We will create a diagram of the dynamic movement model for the rollers and pile drum that provide the feeder module of the new construction gin machine (Fig. 5-6).

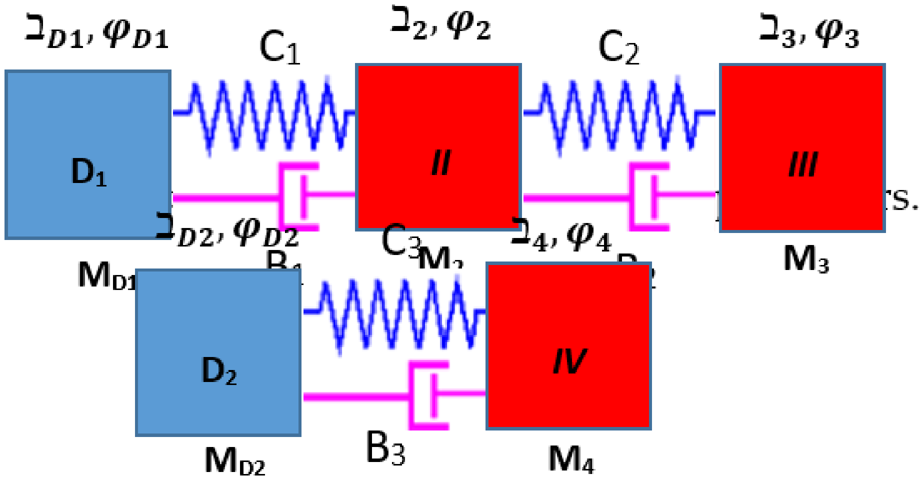

Figure 6. Pile drum speaker model

We construct the motion equations using Lagrange's second equation depending on the provider and the dynamic dynamic motion model of the supply module.

$$\begin{cases} J_{D1} * \ddot{\varphi_{D1}} = M_{D1} - c_1 * (\varphi_{D1} - i_{D12} * \varphi_2) - b_1 * (\ddot{\varphi_{D1}} - i_{D12} * \dot{\varphi}_2): \\ J_2 * \ddot{\varphi}_2 = c_1 * i_{D12} * (\varphi_{D1} - i_{D12} * \varphi_2) + b_1 * i_{D12} * (\ddot{\varphi_{D1}} - i_{D12} * \dot{\varphi}_2) \\ \quad -c_2 * (\varphi_2 - i_{23} * \varphi_3) - b_2 * (\varphi_2 - i_{23} * \dot{\varphi}_3) - M_2: \\ J_3 * \ddot{\varphi}_3 = c_2 * i_{D23} * (\varphi_{D2} - i_{D23} * \varphi_3) + b_2 * i_{D23} * (\ddot{\varphi_{D2}} - i_{D23} * \dot{\varphi}_3) \\ \quad -c_3 * (\varphi_3 - i_{34} * \varphi_4) - b_3 * (\varphi_3 - i_{34} * \dot{\varphi}_4) - M_3: \\ J_{D2} * \ddot{\varphi_{D2}} = M_{D2} - c_4(\varphi_{D3} - i_{D34} * \varphi_4) + b_4 * (\ddot{\varphi_{D3}} - i_{D34} * \dot{\varphi}_4) \\ J_4 * \ddot{\varphi}_4 = c_4 * i_{D34} \cdot (\varphi_4 - i_{45} * \varphi_5) - b_4 * i_{D34}(\varphi_4 - i_{45} * \dot{\varphi}_5) - M_4: \end{cases} \quad (1)$$

here

$J_{D1}, J_2, J_3, J_{D2}, J_4$ – moment of inertia of rotating masses of the supplier module kg m²;

$\ddot{\varphi_{D1}}, \ddot{\varphi}_2, \ddot{\varphi}_3, \ddot{\varphi}_4$ angular velocities of rotating masses of the supplier module system, $M_{D1}, M_{D2}, M_2, M_3, M_4$ -torques of loads acting on the rotating shafts of the supplier module, c_1, c_2, c_3 - - Strength of worm belt transmissions b_1, b_2, b_3, b_4 - are dissipation coefficients in elastic transmissions

We design a dynamic model of the cylinder shafts of the saw machine (Fig. 7).

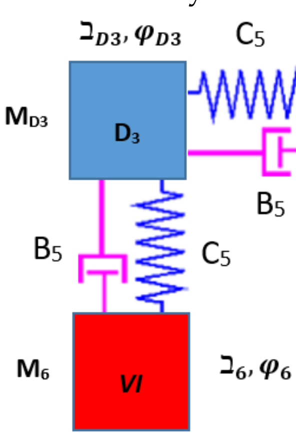

Figure 7. A dynamic model of the cylinder shafts of a saw machine

Based on the dynamic model of the cylinder shafts of the saw machine, we construct the equations of motion using the second Lagrange equation (2) [3].

$$\begin{cases} J_{D3} * \varphi_{D3} = M_{D3} - c_5 * (\varphi_{D3} - \iota_{D35} \ddot{*} \varphi_5) - b_5 * (\ddot{\varphi_{D3}} - \iota_{D35} * \dot{\varphi}_5): \\ J_5 * \ddot{\varphi}_5 = c_5 * i_{D35} * (\varphi_{D3} - i_{D35} * \varphi_5) + b_5 * i_{D35} * (\ddot{\varphi_{D3}} - i_{D35} * \dot{\varphi}_5) - M_5: \\ J_6 * \ddot{\varphi}_6 = c_5 * i_{D36} * (\varphi_{D3} - i_{D36} * \varphi_6) + b_5 * i_{D36} * (\ddot{\varphi_{D3}} - i_{D36} * \dot{\varphi}_6) - M_6 \end{cases} \quad (2)$$

Here

J_{D3}, J_5, J_6 – moment of inertia of the saw cylinder rotating masses kg m²;

$\ddot{\varphi_{D3}}, \ddot{\varphi}_5, \ddot{\varphi}_6$ – angular velocities of the rotating masses of the saw cylinder system, M_{D3}, M_5, M_6 -moments of loads acting on the rotary saw cylinder

b_5- dissipation coefficients in elastic transmissions

FIK = 0,905 - useful duty ratio of the transmission

$U^1 = 0{,}5$ - the number of transmissions of the belt drive

n1=1480 r/pm-engine rotation frequency

P1 =22 KVt - engine power in transmission

We calculate the torque on the engine pulley(3)

$$T_1 = \frac{30 \cdot P_1}{\pi \cdot n_1} = \frac{30 \cdot 22000}{3{,}14 \cdot 1480} = 144{,}886\ n \cdot m; \quad (3)$$

We open GOST1284.3-96, in accordance with clause 3.2 (table 1 and table 2) set and write the dynamic coefficient of the load and the operating mode as Cp=1.

We determine the calculated power of the transmission R v KW, we find the cross section of the tape using this calculator (4).

$$P = P_1 \cdot C_p = 22 \cdot 1 = 22\ kvt; \quad (4)$$

In accordance with clause 3.1 of GOST1284.3-96 (Fig. 1), we select and enter the belt cross-section and standard dimensions C(B) [4].

We open GOST20889-80, set the calculated diameter of the small pulley according to clause 2.2 and clause 2.3 with d1=200mm and write [5].

We calculate the linear speed of the belt drive v (5),

$$v = \frac{\pi \cdot d_1 \cdot n_1}{60000} = \frac{3{,}14 \cdot 200 \cdot 1480}{60000} = 15{,}2\ \ m/sek; \quad (5)$$

We calculate the diameter of the saw cylinder pulley (6)

$$D_2 = d_1 \cdot U^1 = 200 \cdot 0{,}5 = 400mm; \quad (6)$$

According to GOST 20889-80, in accordance with clause 2.2, we define the calculated diameter of the saw cylinder pulley as D_2=400mm

We calculate the number of transmissions (7)

$$U = \frac{D_2}{d_1} = \frac{400}{200} = 2; \quad (7)$$

We calculate the initial deviation Δ delta of the final transmission ratio in % and compare it with the permissible value given in note (8)

$$\Delta = \frac{U - U^1}{U^1} = \frac{2 - 0.5}{0.5} = 3\%; \quad (8)$$

We calculate the rotation frequency of the saw cylinder pulley using the following formula (9)

$$n_2 = \frac{n_1}{U} = \frac{1480}{2} = 724\,\frac{ayl}{min}; \qquad (9)$$

We determine the power of the saw cylinder shaft (10)

$$P_2 = P_1 \cdot FIK = 22 \cdot 0{,}905 = 19{,}910 \text{ kvt}; \qquad (10)$$

We calculate the torque of the saw cylinder for a large pulley (11)

$$T_2 = \frac{30 \cdot P_2}{3{,}14 \cdot n_2} = \frac{30 \cdot 19{,}910}{3{,}14 \cdot 724} = 262{,}243\ N \cdot m; \qquad (11)$$

We calculate the minimum distance between saw cylinder and engine axles (12)

$$a_{min} = 0{,}7 \cdot (d_1 + D_2) = 0{,}7 \cdot (200 + 400) = 420mm; \qquad (12)$$

We calculate the minimum distance between saw cylinder and engine axles (13)

$$a_{max} = 2 \cdot (d_1 + D_2) = 2 \cdot (200 + 400) = 1200mm; \qquad (13)$$

Based on the obtained results and the construction of the project, we accept the value of the distance between the axes of the saw cylinder and the engine as $a^1 = 700mm$[6].

Based on the obtained results, we calculate the length of the belt of the belt drive (14)

$$L_{p1} = 2 \cdot a^1 + \left(\frac{\pi}{2} \cdot (d_1 + D_2)\right) + \frac{(D_2 - d_1)^2}{4 \cdot a^1} = 2357mm; \qquad (14)$$

We open GOST1284.1-89 and according to paragraph 1.1 (table 2) select the approximate length of the tape in mm Lp=2500mm [7].

We recalculate the distance between the axles and find the value of mm (15)

$$\boldsymbol{a = 0{,}25 \cdot \left(L_p - \frac{\pi}{2} \cdot (d_1 + D_2) + \left(\left(L_p - \frac{\pi}{2} \cdot (d_1 + D_2)\right)^2 - 8 \cdot \left(\frac{(D_2 - d_1)}{2}\right)^2\right)^{0{,}5}\right) = 772mm}; \qquad (15)$$

We find the angle of coverage of the small pulley, i.e. the engine pulley A belt

$$(16) A = 2 \cdot arcos\left(\frac{D_2 - d_1}{2a}\right); \qquad (16)$$

According to GOST 1284.3-96 p.3.5.1 (tables 5-17), we define and write down the nominal transmitted power of one P0 strip in kW. P0=9.99kw

According to GOST 1284.3-96 p.3.5.1 (Table 18), we determine and enter the coverage angle coefficient CA. CA=0.982

According to GOST 1284.3-96 p.3.5.1 (table 19), we determine and write down the CL strip length coefficient. CL=0.920

According to GOST 1284.3-96 p.3.5.1 (table 20), we determine and write the coefficient of the number of belts in the transmission CK. When CK=0.760 [8]

We determine the approximate required number of K' strips (17).

K'=P/(P0*CA*CL*CK)=3.2 units (17)

Finally, we determine the number of belts K, rounding to the nearest whole number equal to K= 4 (K') We assume that the number of belts will be 4

Here

J_{D3}, J_5, J_6 – Moment of inertia of masses rotating screw shafts kg-m²;

$\ddot{\varphi}_{D4}, \ddot{\varphi}_7, \ddot{\varphi}_8, \ddot{\varphi}_9$ – Angular velocities of the rotating masses of the system of screw shafts,

M_{D4}, M_7, M_8, M_9 – Moments of loads acting on rotating screw shafts

c_6, c_7, c_8 - Durability of auger belt drives

b_6, b_7, b_8- dissipation coefficients in elastic transmissions

We will design a dynamic model for the snek mechanism for removing dead fiber from a new construction gin machine (Fig. 8)

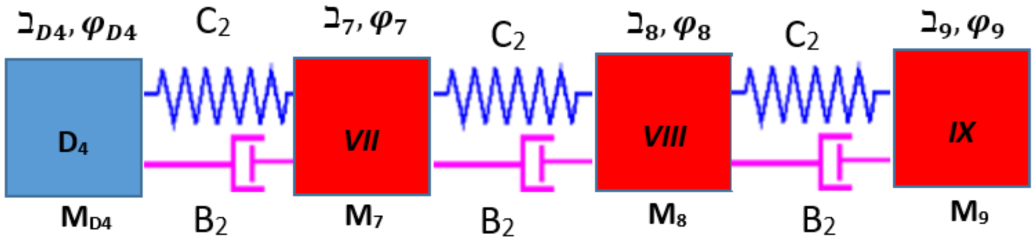

Figure 8. Dynamic model of the screw shafts of the gin machine

Based on the dynamic model of the screw shafts of the gin machine, we construct the equations of motion using the second Lagrange equation (18) [9].

$$\begin{cases} J_{D4} * \ddot{\varphi}_{D4} = M_{D4} - c_6 * (\varphi_{D4} - i_{D47} * \varphi_7) - b_6 * (\ddot{\varphi}_{D4} - i_{D47} * \dot{\varphi}_7): \\ J_7 * \ddot{\varphi}_7 = c_6 * i_{D47} * (\varphi_{D4} - i_{D47} * \varphi_7) + b_6 * i_{D47} * (\ddot{\varphi}_{D4} - i_{D47} * \dot{\varphi}_7) \\ \quad -c_7 * (\varphi_7 - i_{78} * \varphi_8) - b_7 * (\varphi_7 - i_{78} * \dot{\varphi}_8) - M_7 \\ J_8 * \ddot{\varphi}_8 = c_7 * i_{78} * (\varphi_7 - i_{78} * \varphi_8) + b_7 * i_{78} * (\varphi_7 - i_{78} * \dot{\varphi}_8) \\ \quad -c_8 * (\varphi_8 - i_{89} * \varphi_9) - b_8 * (\dot{\varphi}_{48} - i_{89} * \dot{\varphi}_9) - M_8: \\ J_9 * \ddot{\varphi}_9 = c_8 * i_{89}(\varphi_8 - i_{89} * \varphi_9) + b_8 * i_{89} * (\dot{\varphi}_8 - i_{89} * \dot{\varphi}_9) - M_9: \end{cases} \quad (18)$$

Conclusion. A general drawing scheme of the new construction gin machine was created. A scheme of kinematics of the rollers and pile drum providing the gin machine of the new design has been prepared. A dynamic motion model scheme was created for the rollers and pile drum that provide the feeder module of the gin machine. We constructed the motion equations using Lagrange's second equation depending on the model of the supply module and the dynamic dynamic motion. Since the main working part of our construction is saw cylinders, we covered this part in detail. In this case, a kinematics scheme of saw cylinder shafts of a new construction gin machine was created. Based on the model of the motion of the cylinder shafts of the saw machine, we created the equations of motion using the second Lagrange equation. We calculated the torque on the engine pulley. We determined the computational power of the extension. We determined the power of the saw cylinder shaft P_2=19.91 kW, we calculated the torque of the saw cylinder T_2 =262,243 N meters. We determined the length of the belt Lp1=2357mm. We calculated the number of strips to be converted into sawed cylinders and rounded to the nearest whole number equal to K= 4. A scheme of kinematics of the new design of the gin machine for removing dead fibers was created. According to the dynamic model of the screw shafts of the gin machine, the motion equations were constructed using the second Lagrange equation.

$J_{D1}, J_2, J_3, J_{D2}, J_4, J_{D3}, J_5, J_6$ — the moment of inertia of the masses rotating the shafts kg-m^2;

$\ddot{\varphi}_{D1}, \ddot{\varphi}_2, \ddot{\varphi}_3, \ddot{\varphi}_4, \ddot{\varphi}_{D3}, \ddot{\varphi}_5, \ddot{\varphi}_6, \varphi_{D4}, \ddot{\varphi}_7, \ddot{\varphi}_8, \ddot{\varphi}_9$ – masses of the system speeds;

$M_{D1}, M_{D2}, M_2, M_3, M_4, M_{D3}, M_5, M_6$ M_{D4}, M_7, M_8, M_9 – Moments of loads acting on rotating shafts; $c_1, c_2, c_3, c_5\, c_6, c_7, c_8$ - strength of belt drives; $b_1, b_2, b_3, b_4, b_6, b_7, b_8$ - dissipation coefficients and other values in elastic transmissions were taken from Gost and spravochnik konstruktor mashinostroitelya and cotton preliminary processing spravochnik books.

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C O N T E N T S

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