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A Vector Equation Method for Analyzing Kinematics and Kinetostatics of Toggle-Type Transmission Mechanism

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Abstract: With the help of vector equations and MATLAB software, this paper studied the kinematics and kinetostatics of toggle-type transmission mechanism (hereinafter referred to as "toggle mechanism" for short) and attained the analytical expressions of displacement, speed, and acceleration of slider punch, and the force and moment balance equations of each component in the toggle mechanism with their inertia force taken into consideration. Then, the toggle mechanism was compared with conventional crank-link mechanism and their kinematic characteristics were comparatively analyzed. The proposed kinematics analysis method of toggle mechanism could figure out the kinematic characteristics of the target mechanism and reveal its operating advantages on the basis that its functional requirements are met, in this way, the research purpose of optimizing the design of the mechanism could be realized, and the attained conclusions could provide useful evidence for the design of other types of transmission mechanisms.

Keywords: Toggle-type; Transmission mechanism; Kinematics; Kinetostatics; Vector equation

1. Introduction

Mechanical press is a widely used stamping equipment in machining process, and the characteristics of mechanical press can be divided into kinematic characteristics and dynamic characteristics [1]. In recent years, toggle-type press has become a favor in the market due to its structural characteristics. Compared with conventional crank-link mechanism, toggle mechanism has a series of merits: it can run at a low and constant speed during the operating stage and ensure the quality of workpieces; it can increase idle stroke and return stroke speed of slider, thereby improving production efficiency; it can reduce the contact speed between mold and workpiece, which is conductive to prolong the service life of the mold; it can reduce the torque borne by crank, thereby reducing the size of parts in the transmission mechanism and decreasing the inertia of rotating parts; it can get a large slider stroke with a small crank radius, and increase the nominal pressure stroke. Thanks to its significant advantage in increasing pressure, the toggle-style servo press is still a hot spot in the research of this field.

Many scholars in the world have made efforts to explore the toggle mechanism, such as Chen et al. [2] proposed a kinetostatics and dynamics analysis method based on the mechanism of a single-DoF (degree of freedom) six-link mechanical press, and their methodology has received wide attention from the academic circle. Behzadipour and Khajepour [3] proposed a method of using vector bond graphs in dynamic analysis of multi-body dynamic systems, and the work could serve as a useful instruction for optimizing the design of toggle mechanism. Seok et al. [4] proposed a gradient-based plane toggle mechanism and attained multiple candidate mechanisms in the design space. Park et al. [5] applied a toggle mechanism analysis method to solve the kinematics of three design schemes, in this paper, an angular speed ratio was created based on the principle of virtual work, with attaining the maximum output torque in case that the output joint angle is less than 1° as the optimization goal, the size of each component in the mechanism was analyzed and optimized. Zhao et al. [6] designed three types of triangular toggle mechanisms based on a single toggle mechanism, analyzed their kinematics and pointed out their respective pros and cons. Jia et al. [7] proposed a design scheme of a novel 200kN direct drive rotary head press with double motor screw pair and studied its kinematics and dynamics. Some scholars such as Hu et al. [8], Bai and Wen [9], Qiu et al. [10], and Fang et al. [11] took the mechanical benefit of mechanism as optimization goal, and performed design optimization based on ADAMS after qualitative analysis, and the optimized mechanism showed better kinematic characteristics and power increase effects. Fan et al. [12] optimized an eight-link mechanism and improved the uniformity of moving speed in the operating area. After reviewing these studies on toggle mechanism press, it's found that there are few innovative theoretical analysis methods for the kinematics and dynamics of toggle mechanism.

The research of kinematics can be divided into two categories: mechanism analysis and mechanism synthesis, both are very important in mechanical design. The so-called kinematics analysis of mechanism is to analyze the displacement, speed, and acceleration of the mechanism [13]. Under the condition that the motion laws of driving link is known, it analyzes the displacement, trajectory, velocity, and acceleration of various points on rest components in the mechanism, as well as their angular displacement, angular velocity, and angular acceleration. With these motion parameters, we can analyze and evaluate the performance of machines, and they are the basis for the optimization of new machinery. Furthermore, we can say that kinematics analysis is the foundation of dynamics analysis [14]. With the help of vector equations and MATLAB software, this paper analyzed the kinematics and dynamics of a designed toggle mechanism, attained the curves of displacement, speed and acceleration of all components in the toggle mechanism under the condition of different rod lengths, thereby summarizing the kinematic characteristics of each rod in the toggle mechanism, which were then compared with conventional crack-link mechanism to reveal the motion law of toggle mechanism during working process and lay a theoretical foundation for deeper-level and more detailed research on toggle mechanism in the future. Through the kinematics analysis of toggle mechanism, this paper proposed a methodology for analyzing the kinematic characteristics of transmission mechanisms using vector equations, which provides an evidence for the kinematics analysis of other transmission mechanisms.

Research content of this paper covers three aspects: kinematics analysis of toggle mechanism, dynamics analysis of toggle mechanism, and comparative analysis of kinematic characteristics. Research objectives include to analyze the kinematic characteristics in three aspects of displacement, speed, and acceleration, build force and moment balance equations of each component, and attain the corresponding curves, so as to indicate the superiority of toggle mechanism and provide theoretical evidence for determining parameters of the mechanism under actual operating conditions.

2. Kinematic Analysis

Forward kinematics analysis is to solve and output the motion law of output component (slider) based on the known motion law of input component (servo variable/constant speed input) [15]; and inverse kinematics analysis is to solve the motion law of input component (servo drive) based on the known motion law of output component (slider) with other geometric conditions taken into consideration.

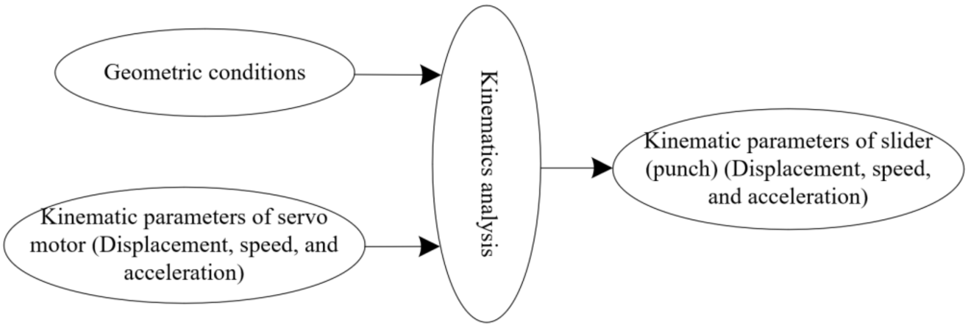

Figure 1. Principle of kinematics analysis

Once the mathematical model and design parameters of a system are determined, the mechanism can be subject to kinematics analysis, and the following kinematics analysis (forward kinematic analysis) was conducted based on the same mathematical model. Principle of kinematics analysis is shown in Figure 1. That is, the geometric relationship of the system and the kinematic parameters of the driving motor are known, and the aim is to solve the kinematic parameters of the slider.

As shown in Figure 2, the toggle mechanism is a new type of flexible mechanical transmission mechanism formed by connecting a crank slider P_5 in series with toggle components P_2, P_3, and P_4. Cranks P_1 and P_2 are directly driven by a servo motor; P_5 is the output point on slider; P_2, P_3, and P_4 are hinge points between connecting rods; together they form a triangular toggle rod. The included angle between rods P_2P_3 and P_4P_3 is α, wherein O and P_1 are fixed hinges on the frame; θ represents the angle between crank and x-axis, and the punch stroke is S at this time; when P_5 is at the lowest point (the lower dead point of press), its ordinate is zero, at this time, 0, P_3, P_4 and

P_5 are on the y-axis, and the three points P_1, P_2 and P_3 are collinear.

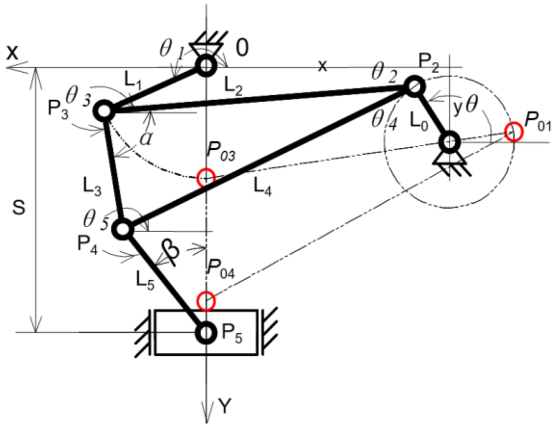

Figure 2. Simplified toggle mechanism of servo press

Size parameters of each component in Figure 2 are: L_0=77mm, L_1=300mm, L_2=490mm, L_3=490mm, L_4=760mm, L_5=300mm, P_1= (404.709 mm, 872.339 mm), the coordinate of point O is the origin; then vector equations could be adopted to solve the position, speed, and acceleration of slider P_5.

2.1 Position Analysis

With point O as the origin, a rectangular coordinate system was established, and the vector equations of the target mechanism are [16]:

$$\overrightarrow{L_0} + \overrightarrow{L_2} = \overrightarrow{L_1} + \vec{x} + \vec{y} \tag{1}$$

$$\overrightarrow{L_2} + \overrightarrow{L_3} = \overrightarrow{L_4} \tag{2}$$

$$\overrightarrow{L_1} + \overrightarrow{L_3} + \overrightarrow{L_5} = \vec{S} \tag{3}$$

Each rod vector can be written in the form of a complex number of exponents, namely $\vec{L} = \mathrm{l}e^{i\theta}$, wherein l is the length of rod, θ is the azimuth angle of rod vector, then the complex vector equation of the mechanism could be attained, and the complex number form of Formula 1 is:

$$l_0 e^{i\theta} + l_2 e^{i\theta_2} = l_1 e^{i\theta_1} - x + iy \tag{4}$$

By applying the Euler's formula $e^{i\theta} = cos\theta + isin\theta$, the above formula can be written as:

$$l_0 cos\theta + il_0 sin\theta + l_2 cos\theta_2 = l_1 cos\theta_1 + il_1 sin\theta_1 - x + iy \tag{5}$$

The real part and imaginary part of above formula are separated to get:

$$\begin{cases} l_0 cos\theta + l_2 cos\theta_2 = l_0 cos\theta_1 - x \\ l_0 sin\theta + l_2 sin\theta_2 = l_0 sin\theta_1 + y \end{cases} \tag{6}$$

Formula 5 can be transformed to:

$$\begin{cases} l_1 cos\theta_1 - l_2 cos\theta_2 = l_0 cos\theta_1 + x \\ l_1 sin\theta_1 - l_2 sin\theta_2 = l_0 sin\theta_1 - y \end{cases} \tag{7}$$

Collate the formula and eliminate θ_2 to get:

$$\theta_1 = 2atan\frac{B_1 + \sqrt{A_1^2 + B_1^2 - C_1^2}}{A_1 - C_1} \tag{8}$$

where, $A_{1=}2l_1(l_0 cos\theta + x)$;$B_1 = 2l_1(l_0 sin\theta - y)$.

$$C_1 = l_2^2 - l_1^2 - (l_0 sin\theta - y)^2 - (l_0 cons\theta + x)^2 \tag{9}$$

By the same token, we also have:

$$\theta_2 = 2atan\frac{B_2 + \sqrt{A_2^2 + B_2^2 - C_2^2}}{A_2 - C_2} \tag{10}$$

where, $A_2 = 2l_2(l_0 cos\theta + x)$;$B_2 = 2l_2(l_0 sin\theta - y)$.

$$C_2 = l_1^2 - l_2^2 - (l_0 sin\theta - y)^2 - (l_0 cons\theta + x)^2 \tag{11}$$

Similarly, the complex form of Formula 2 is:

$$l_2 e^{i\theta_2} + l_3 e^{i\theta_3} = l_4 e^{i\theta_4} \tag{12}$$

Again, by applying the Euler's formula $e^{i\theta} = cos\theta + isin\theta$ to separate the real part and imaginary part of above formula, we can get:

$$\begin{cases} l_2 cos\theta_2 + l_3 cos\theta_3 = l_4 cos\theta_4 \\ l_2 sin\theta_2 + l_3 sin\theta_3 = l_4 sin\theta_4 \end{cases} \tag{13}$$

Collate the formula and eliminate θ_4 to get:

$$\theta_3 = 2atan\frac{E_1 + \sqrt{D_1^2 + E_1^2 - F_1^2}}{D_1 - F_1} \tag{14}$$

where, $D_1 = -2l_2 l_3 cos\theta_2$; $E_1 = 2l_4 l_2 sin\theta_2$.

$$F_1 = l_4^2 - l_3^2 - (l_2 cos\theta_2)^2 - (l_2 sin\theta_2)^2 \tag{15}$$

Similarly, the complex form of Formula 3 is:

$$l_1 e^{i\theta_1} + l_3 e^{i\theta_3} + l_5 e^{i\theta_5} = iS \tag{16}$$

Once again, by applying the Euler's formula $e^{i\theta} = cos\theta + isin\theta$ to separate the real part and imaginary part of above formula, we can get:

$$\begin{cases} l_1 cos\theta_1 + l_3 cos\theta_3 + l_5 cos\theta_5 = 0 \\ l_1 sin\theta_1 + l_3 sin\theta_3 + l_5 sin\theta_5 = 0 \end{cases} \tag{17}$$

where,

$$\theta_5 = arcos\frac{-l_1 cos\theta_1 - l_3 cos\theta_3}{l_5} \tag{18}$$

By combining Formulas 8, 11, 14, and 18, the displacement equation of the slider is:

$$\mathrm{S} = l_1 sin\theta_1 + l_3 sin\theta_3 + l_5 sin\theta_5 \tag{19}$$

This equation is nonlinear and is hard to solve. In this paper, the numerical solution was obtained by using the symbolic calculation function of MATLAB, after collating, the attained results are shown in Figure 3.

2.2 Speed Analysis

By taking the derivative of Formulas 7, 13, and 17 respectively, we can get:

$$\begin{cases} -l_0 \omega sin\theta - l_2 \omega_2 sin\theta_2 = -l_1 \omega_1 sin\theta_1 \\ l_0 \omega cos\theta - l_2 \omega_2 cos\theta_2 = l_1 \omega_1 cos\theta_1 \end{cases} \tag{20}$$

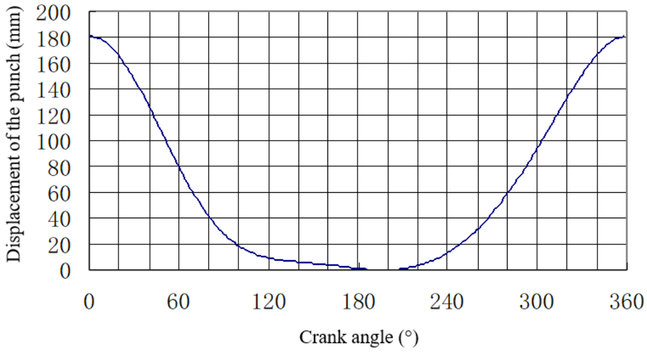

Figure 3. Displacement curve of the slider

$$\begin{cases} -l_2\omega_2 sin\theta_2 - l_3\omega_3 sin\theta_3 = -l_4\omega_4 sin\theta_4 \\ l_2\omega_2 cos\theta_2 + l_3\omega_3 cos\theta_3 = l_4\omega_4 cos\theta_4 \end{cases} \tag{21}$$

$$\begin{cases} -l_1\omega_1 sin\theta_1 - l_3\omega_3 sin\theta_3 - l_5\omega_5 sin\theta_5 = 0 \\ l_1\omega_1 cos\theta_1 + l_3\omega_3 cos\theta_3 + l_5\omega_5 cos\theta_5 = v \end{cases} \tag{22}$$

By solving the above three equations, ω_1,ω_3, and ω_5 could be attained, after collating, the speed equation of the slider is:

$$v = l_1\omega_1 cos\theta_1 + l_3\omega_3 cos\theta_3 + l_5\omega_5 cos\theta_5 \tag{23}$$

Similar to the solution process of speed equation, this equation is also a nonlinear one and it's hard to solve it. In this paper, its numerical solution was also obtained by using the symbolic calculation function of MATLAB, after collating, the attained results are shown in Figure 4.

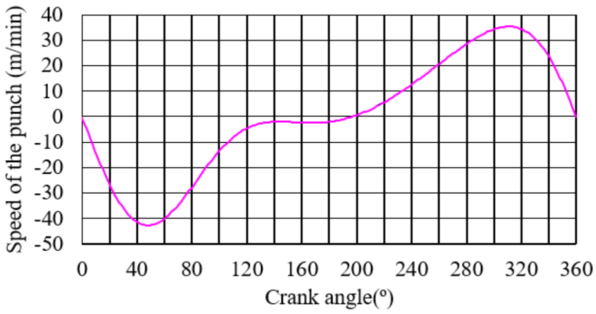

Figure 4. Speed curve of the slider

2.3 Acceleration Analysis

By taking the second derivative of Formulas 7, 13, and 17 respectively, we can get:

$$\begin{cases} l_0\alpha sin\theta + l_0\omega^2 cos\theta + l_2\alpha_2 sin\theta_2 + l_2\omega_2^2 cos\theta_2 = l_1\alpha_1 sin\theta_1 + l_1\omega_1^2 cos\theta_1 \\ l_0\alpha cos\theta - l_0\omega^2 sin\theta + l_2\alpha_2 cos\theta_2 - l_2\omega_2^2 sin\theta_2 = l_1\alpha_1 cos\theta_1 + l_1\omega_1^2 sin\theta_1 \end{cases} \tag{24}$$

$$\begin{cases} l_2\alpha_2 sin\theta_2 + l_2\omega_2^2 cos\theta_2 + l_3\alpha_3 sin\theta_3 + l_3\omega_3^2 cos\theta_3 = l_4\alpha_4 sin\theta_4 + l_4\omega_4^2 cos\theta_4 \\ l_2\alpha_2 cos\theta_2 - l_2\omega_2^2 sin\theta_2 + l_3\alpha_3 cos\theta_3 - l_3\omega_3^2 sin\theta_3 = l_4\alpha_4 cos\theta_4 - l_4\omega_4^2 sin\theta_4 \end{cases} \tag{25}$$

$$\begin{cases} l_1\alpha_1 sin\theta_1 + l_1\omega_1^2 cos\theta_1 + l_3\alpha_3 sin\theta_3 + l_3\omega_3^2 cos\theta_3 + l_5\alpha_5 sin\theta_5 + l_5\omega_5^2 cos\theta_5 = 0 \\ l_2\alpha_2 cos\theta_2 - l_2\omega_2^2 sin\theta_2 + l_3\alpha_3 cos\theta_3 - l_3\omega_3^2 sin\theta_3 + l_5\alpha_5 cos\theta_5 - l_5\omega_5^2 sin\theta_5 = \alpha_s \end{cases} \tag{26}$$

By solving and collating above three equations, the acceleration equation of the slider could be attained as:

$$\alpha_s = l_1\alpha_1^2 cos\theta_1 - l_1\omega_1^2 sin\theta_1 + l_3\alpha_3 cos\theta_3 - l_3\omega_3^2 sin\theta_3 + l_5\alpha_5 cos\theta_5 - l_5\omega_5^2 sin\theta_5 \tag{27}$$

3. Dynamics Analysis

The matrix method was adopted to perform kinetostatics analysis on the transmission mechanism of servo press (the gravity of each component and the friction between components were ignored). A coordinate system shown in Figure 5 was established, and the Y direction is the direction of gravity.

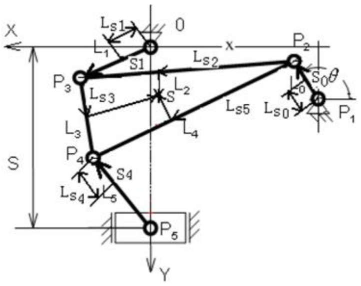

Figure 5. Kinetostatics analysis of servo press

Known geometric parameters of components are: L_0, L_1, L_2, L_3, L_4, L_5.
Distances from mass centers of rods to front hinges are: L_{s0}, L_{s1}, L_{s3},L_{s5}.
Displacement angles of components are:θ, θ_1, θ_2, θ_3, θ_4, θ_5.

3.1 Torque Parameters

According to the D'Alembert's principle, if the inertial force and inertial moment are regarded as general external forces applied to the component that produces the inertial force, then the machine can be regarded as being in an equilibrium state, and static force can be analyzed by statics analysis method, namely to adopt kinetostatics analysis method [17].

For component performing plane complex movement and having a symmetry plane parallel to the plane of motion, its inertial force can be simplified as an inertial force $\overrightarrow{F_\iota}$ applied to mass center S_i and an inertial torque $\overrightarrow{M_\iota}$ [18], therefore, its torque parameters can be expressed as:

The sum of inertial forces applied at mass center S_i of component *i*: $\overrightarrow{F_\iota}$.

The sum of the external force torques applied at mass center S_i of component *i*: $\overrightarrow{M_\iota}$.

3.2 Physical Parameters

Mass of component *i*: m_i.
Rotational inertia of component *i* about its mass center: J_i(i=1, 2, 3, 4, 5).

3.3 Kinematic Parameters

At a certain moment, the initial phase angles of two driving links are: θ_1 and θ_3.
Angular speeds:ω_1, ω_3.
Angular accelerations: α_1, α_3.
Component mass: m_i(i=1, 2, 3, 4, 5).
Force on the component: F_{RPi}(i=1, 2, 3, 4, 5).
Force on the component in x direction: F_{RPix}(i=1, 2, 3, 4, 5).
Force on the component in y direction: F_{RPiy}(i=1, 2, 3, 4, 5).
Determine the equilibrant moment M_d applied to the driving link at this position.

In this paper, the servo press transmission mechanism shown in Figure 5 was split into several single components to perform kinetostatics analysis.

In Figure 6, assuming Q represents the resistance force the slider is subjected to, to make the mechanism be in a balanced state, it needs to apply an equilibrant moment M_d on driving link L_0, according to force analysis shown in Figure 6, the force and moment balance equations are listed as follows:

Component 1:

$$\begin{cases} -F_{RPx} + F_{RP2x} = -m\ddot{S}_x \\ F_{RPy} - F_{RP2y} = -m\ddot{S}_y \\ -F_{RPx}L_s - F_{RPy}L_s cos\theta + F_{RP2x}(L_0 - L_s cos\theta) + M_d = -J_0\alpha \end{cases} \tag{28}$$

Component 2:

$$\begin{cases} -F_{ROx} + F_{RP3x} = -m_1\ddot{S}_{x1} \\ F_{ROy} - F_{RP3y} = -m_1\ddot{S}_{1y} \\ -F_{ROx}L_{s1}sin\theta_1 - F_{ROy}L_{s1}cos\theta_1 + F_{RP3x}(L_1 - L_{s1}sin\theta_1) + F_{RP3x}(L_1 - L_{s1}cos\theta_1) = -J_1\alpha_1 \end{cases} \quad (29)$$

Toggle rod:

$$\begin{cases} -F_{RP2x} - F_{RP3x} + F_{RP4x} = -m_2\ddot{S}_{2x} \\ F_{RP2y} - F_{RP3y} + F_{RP4Y} = -m_2\ddot{S}_{2y} \\ -F_{RP2x}L_{s2}sin(\theta_2 + \emptyset) - F_{RP2y}L_{s2}cos(\theta_2 + \emptyset) - F_{RP3x}L_{s2}sin(\theta_3 + \emptyset) - F_{RP3y}L_{s2}sin(\theta_3 + \emptyset) \\ +F_{RP4x}L_{s2}sin(\theta_4 + \emptyset) - F_{RP4y}L_{s2}sin(\theta_4 + \emptyset) = -J_2\alpha_2 \end{cases} \quad (30)$$

Component 4:

$$\begin{cases} -F_{RP4x} + F_{RP5x} = -m_4\ddot{S}_{4x} \\ F_{RP4y} - F_{RP5y} = -m_4\ddot{S}_{4y} \\ -F_{RP4x}L_{s4}sin\theta_5 - F_{RP4y}L_{s4}cos\theta_5 + F_{RP5x}(L_4 - L_{s4}sin\theta_5) + F_{RP5y}(L_4 - L_{s4}cos\theta_5) = -J_4\alpha_4 \end{cases} \quad (31)$$

Component 5:

$$\begin{cases} -F_{RP5x} + Q = 0 \\ -F_{RP5x} - F = m_5\alpha_5 + m_5g \end{cases} \quad (32)$$

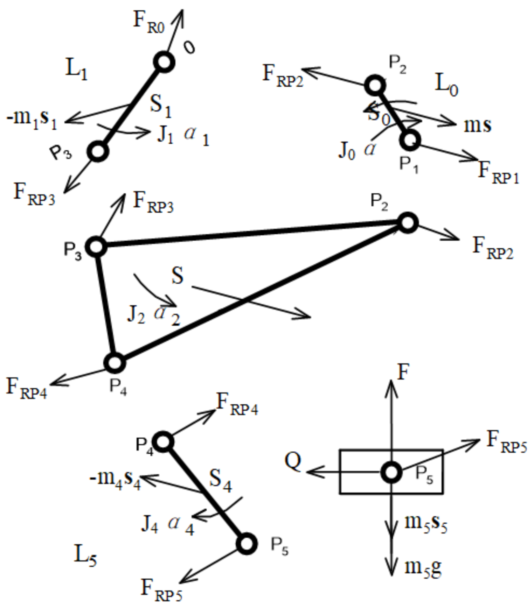

Figure 6. Force analysis of each rod component

3.4 Moving Torque Curve

Since the processing of press has the features of large load and strong impact, compared with the torque of conventional crank-link mechanism, the torque of toggle mechanism requires a greater output torque and it has a stronger anti-disturbance capability [19]. Therefore, based on the kinetostatics analysis of toggle mechanism, it's necessary to analyze the moving torque of toggle mechanism. Figure 7 shows a motion model of toggle mechanism under normal operating condition.

The dynamics analysis of toggle mechanism includes analyzing and calculating the force, inertia, and dynamic response, etc. They contribute a lot to the design and control of mechanism and are the basis for determining main structural parameters of the system [20]. When the slider performs stamping along a certain track, the force acted on the mechanism would vary according to different depths of the punch cutting into the stamping piece, and the kinetic energy of each component in the mechanism would change accordingly. Through dynamics analysis, the laws of the changes in force and kinetic energy of each component could be figured out [21]. According to the motion model of toggle mechanism shown in Figure 7, the force and moment equations were solved by the symbolic calculation function of MATLAB [22], and the results are shown in Figure 8.

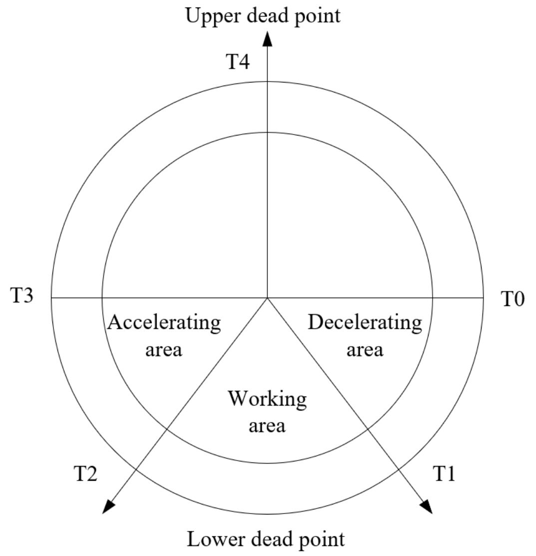

Figure 7. Motion model of toggle mechanism

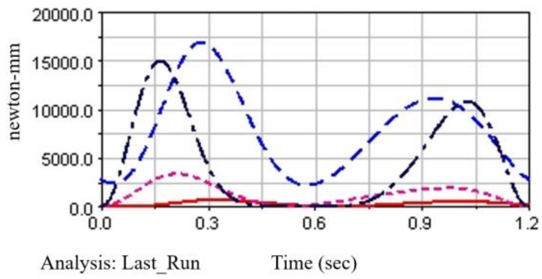

Figure 8. Torque curves of toggle mechanism components

Note: The solid line curve represents Component 1; the short dotted line curve represents Component 2; the long dotted line curve represents the toggle rod component; the chain dotted line represents Component 4

4. Comparative Analysis of Kinematic Characteristics

To verify the scientific nature of kinematics analysis, this paper took a 1600kN toggle press as the subject and used vector equations to analyze its kinematics and dynamics, and its features were compared with those of the crank-link mechanism of a crank press with the same tonnage [23].

Technical parameters of the 1600kN toggle press: nominal force Pg=1600kN; nominal force stroke Sg=6mm; slider stroke S=180mm; number of times of stroke n=40-60spm. Technical parameters of crank press with same tonnage: nominal force Pg=1600 kN; stroke is 180mm; length of connecting rod is 1390mm, nominal force stroke Sg=6mm, maximum rotation speed is 60spm. The comparison results are shown in Table 1.

Table 1. Comparison of kinematic characteristics between the toggle mechanism and the crank-link mechanism

Category	Pressure angle (°)	Uniform speed range of the slider (°)	Linear speed of slider at the position of nominal force v (m/s)	Torque at the position of nominal force T(N·m)
Toggle mechanism	36.0	135-180	-112.6	1563
Crank-link mechanism	20.6	153-180	-228.2	3615

Through comparative analysis, the following conclusions have been drawn:

(1) The toggle mechanism has an ideal low-speed movement feature in the working area, the contact speed between the mold and the workpiece is low, which is conducive to increasing the service life of the mold.

(2) The displacement curve of slider in the toggle mechanism is asymmetric and has a quick return feature, it can perform quick return motions, thereby improving the production efficiency [24].

(3) The slider can stay near the lower dead point for a long time, which has a function of maintaining pressure, thereby ensuring the quality of cold-extruded workpieces [25].

(4) The toggle mechanism has a good effect of enhancing power, when the performance parameters of different type press machines are the same, the torque borne by the crank is smaller.

5. Conclusion

This paper took toggle mechanism as subject and performed kinematics analysis and dynamics analysis on it to figure out the impact of each component on its kinematic characteristics, as a result, the corresponding curves were attained, which had provided theoretical evidence for parameter setting of the mechanism under actual operating conditions. Moreover, these research findings could also offer references for the design of mechanism size, the calculation of strength of bearings and components, and the selection of power and torque of the driving motor. However, in actual applications, in view of the complex transmission systems of press machines, it is suggested to fully integrate the influence of inertia force, resistance and other factors during the movement process of press machines, comprehensively consider the influence of structural size, kinematic parameters and component mass parameters of the mechanism on the performance of the press, and apply to the comprehensive analysis of mechanism scale, in this way, we could avoid blind mechanism design and put forward a new idea for studies on the comprehensive performance of mechanisms considering power distribution.

Data Availability

The data supporting our research results are included within the article or supplementary material.

Conflicts of Interest

The authors declare no conflict of interest.

References

[1] Y. S. Sun and Z. R. Zhang, “Servo forming technology and its several developing trends,” *Forging Stamping Technol.*, vol. 47, no. 1, pp. 1-16, 2022. https://doi.org/10.13330/j.issn.1000-3940.2022.01.001.

[2] X. L. Chen, W. H. Gao, S. Jiang, H. Song, and Y. Y. Gou, “Static and dynamic analysis of a novel single-DOF six-bar mechanical press mechanism,” *J. Shandong Univ. Sci. Technol. (Nat. Sci.)*, vol. 36, no. 5, pp. 80-90, 2017. https://doi.org/10.16452/j.cnki.sdkjzk.2017.05.012.

[3] S. Behzadipour and A. Khajepour, “Causality in vector bond graphs and its application to modeling of multi-body dynamic systems,” *Simul. Model. Pract. Theory*, vol. 17, pp. 279-295, 2005. https://doi.org/10.1016/j.simpat.2005.06.001.

[4] W. K. Seok, I. K. Suh, and Y. K. Yoon, “Topology optimization of planar linkage systems involving general joint types,” *Mech. Mach. Theory*, vol. 104, pp. 130-160, 2016. https://doi.org/10.1016/j.mechmachtheory.2016.05.015.

[5] S. Park, J. Bae, Y. Jeon, K. Chu, J. Bak, T. W. Seo, and J. Kim, “Optimal design of toggle-linkage mechanism for clamping applications,” *Mech. Mach. Theory*, vol. 120, pp. 203-212, 2018. https://doi.org/10.1016/j.mechmachtheory.2017.08.013.

[6] S. T. Zhao, Z. K. Shao, and Z. H. Sheng, “Discussion on Rationality of Working Mechanism of Mechanical Press,” *Forging Equip. Manuf. Technol.*, vol. 3, pp. 14-19, 2013. https://doi.org/10.3969/j.issn.1672-0121.2013.03.003.

[7] X. Jia, S. T. Zhao, and S. Q. Fan, “Research on kinematics and dynamics of a new 200 kN direct drive rotary head press with double motor screw pair,” *Mech. Sci. Technol. Aerospace Eng.*, vol. 36, no. 8, pp. 1205-1211, 2017. https://doi.org/10.13433/j.cnki.1003-8728.2017.0810.

[8] J. G. Hu, Y. S. Sun, Z. R. Zhang, et al. “Optimization design of servo press triangular toggle working mechanism based on ADAMS,” *Forging & Stamping Technol.*, vol. 44, no. 2, pp. 126-131, 2019. https://doi.org/10.13330/j.issn.1000-3940.2019.02.021.

[9] Y. Q. Bai and X. Z. Wen, “Optimization design of eight-link stamping mechanism based on ADAMS,” *J. Yanbian Univ. (Nat. Sci.)*, vol. 2, no. 175-178, 2015. https://doi.org/10.3969/j.issn.1004-4353.2015.02.017.

[10] Y. Qiu, W. M. Li, and Z. Wei, “Performance research and simulation of servo press with triangle elbow,” *J. Mach. Des.*, vol. 31, no. 7, pp. 22-24, 2014. https://doi.org/10.7666/d.D279422.

[11] Y. Fang, Y. S. Sun, J. G. Hu, Y. Q. Cheng, P. Zhang, and W. P. Ruan, "Research on kinematics/dynamics simulation of servo press transmission mechanism based on MATLAB," *China Mech. Eng.*, vol. 23, no. 3, pp. 339-342, 2021. https://doi.org/10.3969/j.issn.1004-132X.2012.03.020.

[12] Y. X. Fan, B. Mou, and C. C. Guan, "Optimization design of eight-link press based on ADAMS," *Mach. Build. Autom.*, vol. 2, pp. 105-107, 2014. https://doi.org/10.3969/j.issn.1671-5276.2014.02.034.

[13] M. J. Duan and D. Zhou, "Finite-time composite guidance law with input constraint and dynamics compensation," *Chinese J. Aeronaut.*, vol. 8, pp. 664-671, 2017. https://doi.org/10.1016/j.mechmachtheory.2017.06.018.

[14] M. Neeraj and A. Vaz, "Bond graph modeling of a 3-joint string-tube actuated finger prosthesis," *Mech. Mach. Theory*, vol. 20, pp. 1-20, 2017. https://doi.org/10.1016/j.mechmachtheory.2017.06.018.

[15] C. Mishra, A. K. Samantaray, and G. Chakraborty, "Bond graph modeling and experimental verification of a novel scheme for fault diagnosis of rolling element bearings in special operating conditions," *J. Sound Vib.*, vol. 29, pp. 302-330, 2016. https://doi.org/10.1016/j.jsv.2016.05.021.

[16] A. Abboudi and F. Belmajdoub, "Hybrid diagnosis method applied to switched mechatronic systems," *J. Eur. Syst. Automat.*, vol. 54, no. 5, pp. 683-691, 2021. https://doi.org/10.18280/jesa.540503.

[17] B. Boudon, T. T. Dang, R. Margetts, W. Borutzky, and F. Malburet, "Simulation methods of rigid holonomic multibody systems with bond graphs," *Adv. Mech. Eng.*, vol. 67, no. 3, pp. 37-39, 2019. https://doi.org/10.1177/1687814019834153.

[18] B. Boudon, F. Malburet, and J. C. Carmona, "Simulation of a helicopter's main gearbox semiactive suspension with bond graphs," *Multibody Syst. Dyn.*, vol. 31, pp. 375-405, 2017. https://doi.org/10.1007/s11044-016-9536-5.

[19] S. M. He and C. H. Lee, "A publication of the American institute of aeronautics and astronautics devoted to the technology of dynamics and control," *J. Guid. Control Dyn.*, vol. 41, no. 1, pp. 171-183, 2018. https://doi.org/10.2514/1.G002949.

[20] W. Du, S. D. Zhao, and L. Y. Jin, "Optimization design of triangular elbow mechanism of mechanical press based on NSGA II multi-objective optimization algorithm," Forg. Stamp. Technol., vol. 43, no. 11, pp. 77-82, 2018. https://doi.org/10.13330/j.issn.1000-3940.2018.11.014.

[21] J. C. Xu, X. Y. Ren, F. L. Zhang, H. Y. Xu, and Y. S. Huang, "Design improvements of the clutch and brake of 35000kN mechanical press," *Automotive Eng.*, vol. 15, no. 2, pp. 122-128, 1993. https://doi.org/10.3969/j.issn.1004-2539.2010.10.021.

[22] Y. Y. Wang, X. X. Jin, and D. S. Zhang, "Research on optimal design method of steering mechanism of double front axle," *Automotive Eng.*, vol. 28, no. 6, pp. 574-577, 2006. https://doi.org/10.3321/j.issn:1000-680X.2006.06.016.

[23] C. Cheng, W. X. Ding, and Y. Sun, "Optimization design of transmission mechanism of eight-link press," *Forg. Stamp. Technol.*, vol. 42, no. 8, pp. 88-92, 2017. https://doi.org/10.13330/j.issn.1000-3940.2017.08.018.

[24] C. Zhang, J. H. Mo, H. Yan, Y. Zhu, and Y. R. Wang, "Simulation and optimization of triangle-link transmission mechanism of servo press," *Forg. Equip. Manuf. Technol.*, no. 3, pp. 28-32. https://doi.org/10.3969/j.issn.1672-0121.2013.03.006.

[25] C. H. Yang and X. Shao, "Kinematics modeling and analysis of toggle working mechanism of mechanical press," *Forg. Stamp. Technol.*, vol. 42, no. 12, pp. 87-91, 2017. https://doi.org/10.13330/j.issn.1000-3940.2017.12.016.