OPTIMIZATION DESIGN OF CRANK DOUBLE TOGGLE LINKAGE MECHANISM FOR SERVO MECHANICAL PRESSES

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According to the special process requirements of the servo mechanical presses, a virtual prototype model of the crank double toggle linkage mechanism was established in ADAMS (Automatic Dynamic Analysis of Mechanical Systems) software. Firstly, the working characteristics of the crank double toggle linkage mechanism is analyzed by comparing with the crank triangle toggle linkage mechanism. Secondly, kinematics of the crank double toggle linkage mechanism is analyzed by complex vector method. Thirdly, according to the constraint conditions and optimization objectives of the crank double toggle linkage mechanism, a mathematical model for optimization design is constructed. Using ADAMS to analyze the sensitivity of various dimensional variables of the transmission mechanism and optimize the design by reducing design variables. Finally, the model is optimized using the generalized reduced gradient method. The research results show that the performance of the optimized crank double toggle linkage mechanism has been greatly improved. The maximum speed of the slider is reduced by 37.5%, and the maximum acceleration of the slider is reduced by 23.7%.At the same time, the mechanism has great ability to maintain stress and high return speed.

Keywords: Servo mechanical presses, Crank double toggle linkage mechanism, Generalized reduced gradient method, ADAMS.

1. Introduction

With the development of power, motor, and computer control technology, AC (Alternating Current) servo drive technology has developed rapidly. The servo presses are gradually replacing mechanical presses due to their excellent performance and energy-saving and environmental protection characteristics [1]. Due to the significant advantage of toggle linkage mechanism in amplifying

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pressure, the research on toggle linkage servo presses is still a hot topic in this field [2].

At present, some researchers have done a lot of research work on the optimization design of crank transmission mechanisms by using various methods. Reza and Moslehi [3] proposed a kind of mechanism has been designed by the inspiration of two common linkages, crank-rocker and slider-crank mechanisms. In this research, an analytical approach is utilized to evaluate the position of members of the adjustable six-bar mechanism, and dimensional synthesis is carried out using a COA method(Cuckoo Optimization Algorithm).The results indicate that a more accurate path generated could be obtained for the adjustable six-bar mechanism compared with the four-bar linkage. Orvañanos-Guerrero et al. [4] presented the complete balancing optimization of a six-bar mechanism with the use of counterweights. A novel method based on fully Cartesian coordinates (FCC) is proposed to represent a balanced mechanism. A multi-objective optimization problem was solved using the Differential Evolution (DE) algorithm to minimize the shaking force (ShF) and the shaking moment (ShM) and thus balance the system. Acharya et al. [5] created a typical eight-bar mechanism based on detailed literature survey of various flap mechanisms. Flap positions for cruise, take-off and landing are configured. A novel synthesis procedure based on Optimization Technique is developed to calculate the linkage dimensions of the mechanism. The synthesis procedure is reduced as an error minimizing function and solved using Genetic Algorithm based optimization technique in MATLAB optimization toolbox. The results are verified using 2D and 3D kinematic models modelled in SOLIDWORKS. Kim and Shim [6] dealt with the dimensional synthesis of a Stephenson III six-bar slider-crank function generator, which is usually modified into an adjustable mechanism by adding one link and one joint. For this, displacement analysis equations were derived for the Stephenson III six-bar slider-crank mechanism, and a genetic algorithm was used to determine the dimensions of the mechanism that approximately satisfies a set of prescribed piston joint positions. Tuleshov et al. [7] proposed a new analytical method for studying the functionality and synthesis of the Stephenson II crank-slider mechanism according to a given slider change coefficient in average speed and the optimization pressure angle to ensure the highest force which was transfer from the input link to the functional body. Song et al. [8] used ADAMS to simulate and optimize the design of an eccentric horizontal crank toggle linkage press. The minimum average value of the pressure angle of the slider working stroke is used as the objective function to reduce the driving torque and improve the mechanical efficiency of the mechanism. Yang and Shao [9] applied the vector equation analytical method to establish a nonlinear kinematic equation for the crank toggle linkage mechanism, numerically solved it based on MATLAB software, obtained the kinematic solution of the mechanism, and graphically processed the calculation results.

Some of the above methods require tedious programming, while others can only obtain local optimization solutions, which are not practical in the actual design process. This paper selects a crank double toggle linkage mechanism for the special process requirements of servo mechanical presses, utilizes the built-in functions of ADAMS software to reduce the number of design variables and writes simple programs for optimization design. Practice has proven that this method improves the speed of optimization design and shortens the design process.

2. Material and research method

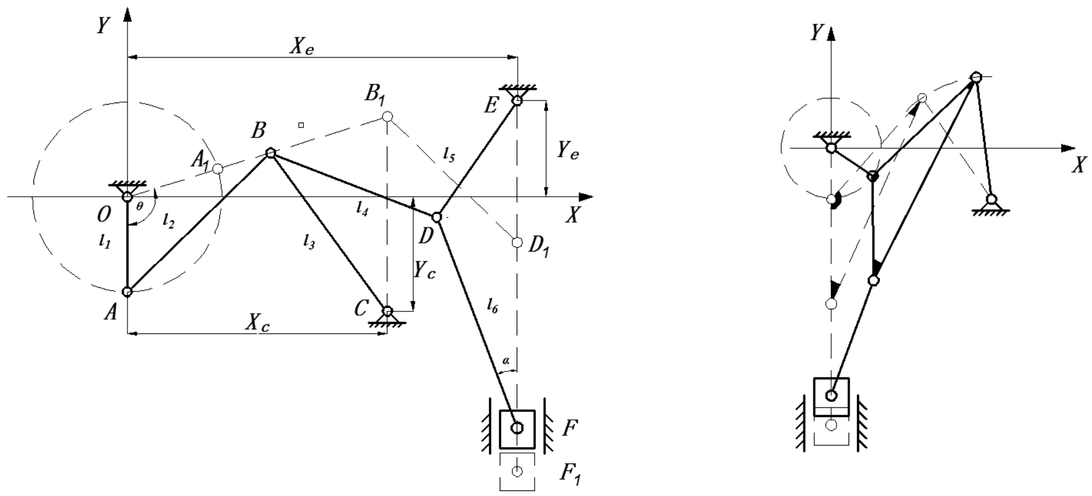
2.1. Selection of transmission mechanism

2.1.1. Two kinds of mechanisms

There are various forms of crank toggle linkage mechanisms. The superiority of the triangle toggle linkage mechanism is reflected in, large nominal pressure angle, lower contact speed, long time to maintain pressure, high return speed, large allowable force on the slider, etc. In addition, it can also improve the accuracy of the extruded parts and extend the lifespan of the mold. However, for processing lightweight plates with high strength and large rebound, the holding time of the crank triangle toggle linkage mechanism is not long enough, and the maximum speed within the working stroke is still relatively high. The crank double toggle linkage mechanism in this paper has a longer holding time and is more suitable for the above requirements.

Figure 1(a) shows a crank double toggle linkage mechanism, and Figure 1(b) shows a crank triangle toggle linkage mechanism. In Figure 1(a), the solid line indicates the position of each rod when the slider is running on the way, and the dotted line indicates the position of each rod when the slider is running at the bottom dead center. In this situation, $l_1(OA)$ is a crank, $l_2(AB)$ is a connecting rod, $l_3(BC)$, $l_4(BD)$, $l_5(DE)$ and $l_6(DF)$ are toggle linkages. Let the center of the crank shaft is O, A is the hinge point of the rods l_1 and l_2, B is the hinge point of the rods l_2 and l_3, C is the hinge point of the l_3 and the frame, D is the hinge point of the l_4, l_5 and l_6, E is the hinge point of the l_5 and the frame, and F is the hinge point of the l_6 and the slider. According to the formula $F = 3n - 2p_r - p_a$ for the degree of freedom of the mechanism, there are 7 moving members, 9 rotating pairs, and 1 moving pair in the mechanism, so $F = 1$,

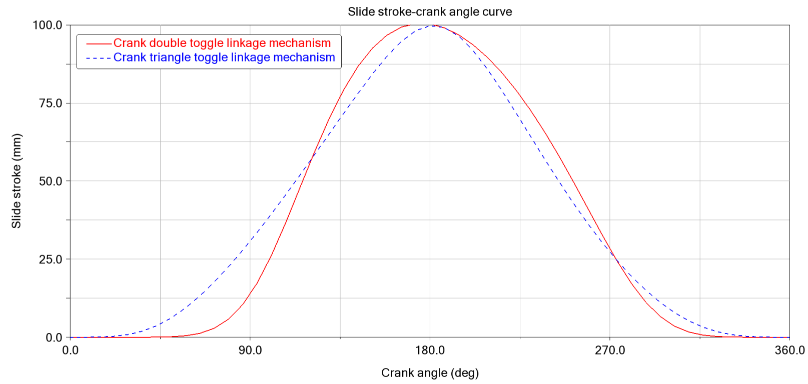
l_1 is the prime mover. When the slider moves to the bottom dead point, A moves to A_1, B moves to B_1, D moves to D_1, and E moves to E_1.In Figure 1(b), the solid line indicates the state of each rod during the slider movement, and the dotted line indicates the position of each rod when the slider is at the bottom dead point.



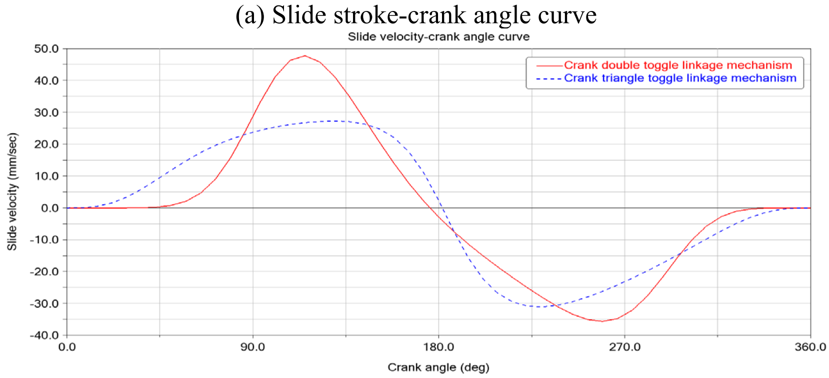
(a) Crank double toggle linkage mechanism (b) Crank triangle toggle linkage mechanism
Fig. 1. Diagram of two transmission mechanisms

2.1.2. Analysis of kinematic characteristics of two mechanisms

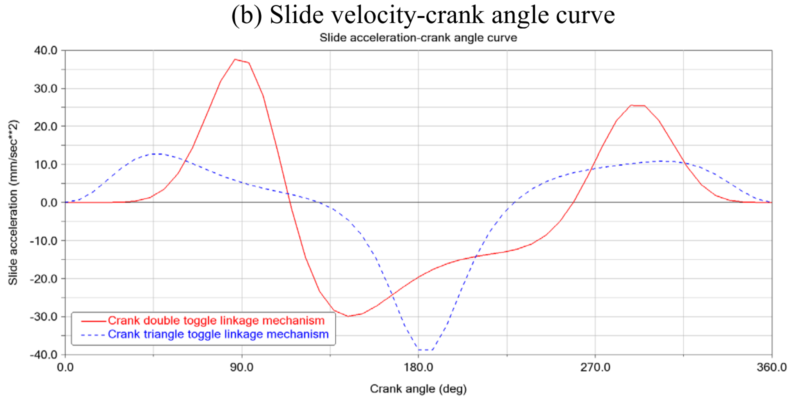
Under the same working parameter with a nominal force of 800 kN and a stroke of 100 mm, using ADAMS software, the kinematic characteristics of the crank double toggle linkage mechanism and the crank triangle toggle linkage mechanism are analyzed under the condition of idle operation, with the bottom dead center of the slider as the origin. The curves of the relationship between slider stroke, speed, acceleration, and crank angle is obtained, as shown in Fig. 2.



(a) Slide stroke-crank angle curve



(b) Slide velocity-crank angle curve



(c) Slide acceleration-crank angle curve

Fig. 2. Kinematic characteristics of two kinds of mechanisms

As can be seen from Fig. 2(a), the crank double toggle linkage mechanism has long time to maintain pressure near the bottom dead point compared to the crank triangle toggle linkage mechanism. As can be seen from Fig. 2(b), the crank double toggle linkage mechanism has higher speed and better returning characteristics. As can be seen from Fig. 2(c), the acceleration of the crank double toggle linkage mechanism is large, resulting in an increase in the inertial force of the slider [10]. In summary, it is necessary to optimize the design of the crank double toggle linkage mechanism to improve its performance.

2.2. Kinematics analysis of crank double toggle linkage mechanism

The working quality of the mechanism is determined by the kinematics characteristics of the mechanism, and the kinematic parameters of the mechanism are the dynamic analysis of the mechanism. The kinematic analysis of the mechanism can not only analyze the performance of the existing mechanism, but also test the comprehensive performance of the mechanism [11]. Therefore, the kinematics analysis of the mechanism is an essential link in the process of mechanical design.

2.2.1. Three methods of kinematics analysis

The kinematics analysis of mechanism mainly includes the analyses of position, velocity and acceleration. The main purpose of position analysis is to: (a) Determine the position of the mechanism and draw a position map of the mechanism; (b) Determine whether interference is found in the motion space of the component; (c) Determine the travel of components and identify the upper and lower limit positions; (d) Determine the trajectory of the point. The main purpose of speed analysis is to know whether the speed variation pattern at the output end of the mechanism is full meet working requirements and prepare for the acceleration analysis. The main purpose of acceleration analysis is to determine the inertial force of the mechanism. At present, there are three main methods of kinematics analysis: the graphical method, the experimental method and the analytical method. With the development of computer technology, the graphical method is gradually being phased out. The experimental method can be used to verify complex mechanisms and solve the problem of achieving predetermined trajectories through the use of a linkage curve atlas, but this method has a higher cost. The analytical methods mainly include the complex vector method, the matrix method and the rod group method [12]. The design equation is established according to the kinematics principle, and the unknown motion parameters of the mechanism are expressed by the known motion parameters and size parameters with functional relations. The analytical steps are shown in Fig. 3. The analytical method can not only accurately determine the motion characteristics of the mechanism throughout the entire motion process, but also connect the mechanism analysis with the mechanism synthesis problems to obtain the optimization solution. However, the process of establishing a mathematical model is complex and the computational workload is large. However, with the improvement of computer technology and mathematical tools, analytical methods are currently widely used.

In order to obtain the parameter equations of the stroke, speed and acceleration of the sliding block of the transmission mechanism, so as to optimize the transmission mechanism, this paper selects the complex vector method to carry out kinematics analysis of the transmission mechanism.

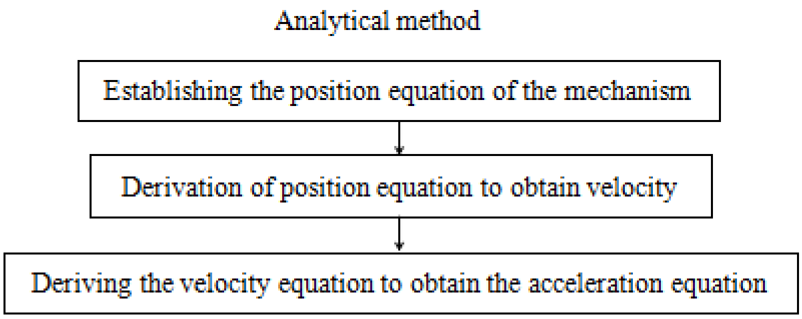


Fig. 3. Analytical steps

2.2.2. Kinematics analysis of crank double toggle linkage mechanism

Firstly, the mathematical model of the mechanism is established, then the complex vector method is used to analyze the kinematics of the mechanism, and the displacement, velocity and acceleration equations of the slider are obtained. As shown in Figure 4,it is the geometric modeling of the crank double toggle linkage mechanism, where l_8 is the line from the center of O to the center of E;The l_7 is the line connecting the O center to the center of C.The θ_1 is the angle between crank l_1 and the counterclockwise direction of the X-axis. The θ_2 is the angle between l_2 and the counterclockwise direction of the X-axis. The θ_3 is the angle between l_3 and the counterclockwise direction of the X-axis. The θ_4 is the angle between l_4 and the counterclockwise direction of the X-axis. The θ_5 is the angle between l_5 and the counterclockwise direction of the X-axis. The θ_6 is the angle between l_6 and the counterclockwise direction of the X-axis. The θ_7 is the angle between l_7 and the counterclockwise direction of the X-axis. The θ_8 is the angle between l_8 and the counterclockwise direction of the X-axis. The β and is the angle between l_6 and the vertical direction. The γ is the angle between l_5 and the vertical direction. The servo motor drives the rotation of crank l_1, which drives the double toggle linkage mechanism through a connecting rod to cause the slider to move up and down in reciprocating motion.

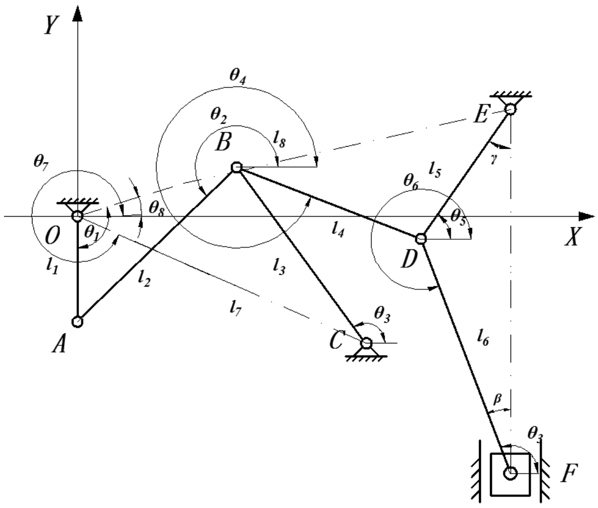


Fig. 4. Geometric modeling of crank double toggle linkage mechanism

The kinematics analysis of the crank double toggle linkage mechanism is carried out, and its vector equations are as follows:

$$\left\{ \begin{array}{l} \vec{l_1}+\vec{l_2}=\vec{l_3}+\vec{l_7} \\ \vec{l_1}+\vec{l_2}=\vec{l_4}+\vec{l_5}+\vec{l_8} \\ \vec{l_5}+\vec{l_6}=\overrightarrow{EF} \end{array} \right\} \tag{1}$$

In equation (1) above, $l_7 = \sqrt{\left(l_1 + l_2\right)^2 + l_3^2}$, $l_8 = \sqrt{\left(l_1 + l_2 - l_5\right)^2 + l_4^2}$.

Then we write each vector of equation (3-1) in complex exponential form, and there is the following equation system:

$$l_1 e^{i\theta_1} + l_2 e^{i\theta_2} = l_3 e^{i\theta_3} + l_7 e^{i\theta_7} \tag{2}$$

$$l_1 e^{i\theta_1} + l_2 e^{i\theta_2} = l_4 e^{i\theta_4} + l_5 e^{i\theta_5} + l_8 e^{i\theta_8} \tag{3}$$

$$l_5 e^{i\theta_5} + l_6 e^{i\theta_6} = l_{EF} e^{i3\pi/2} \tag{4}$$

In the formula: i is an imaginary number; l_{EF} is the distance between point E and point F .

By solving the equations, various technical parameters can be determined by using the crank angle θ_1 and the length of each rod.

Slide displacement equation:

$$s = l_5 + l_6 + l_5 \sin\theta_5 + l_6 \sin\theta_6 \tag{5}$$

Take the derivative of time on both sides of the displacement equation to obtain the slider velocity equation:

$$v = \frac{ds}{dt} = \omega_5 l_5 \cos\theta_5 + \omega_6 l_6 \cos\theta_6 \tag{6}$$

Take the derivative of time on both sides of the velocity equation to obtain the slider acceleration equation:

$$a = \frac{dv}{dt} = a_5 l_5 \cos\theta_5 - \omega_5^2 l_5 \sin\theta_5 + a_6 l_6 \cos\theta_6 - \omega_6^2 l_6 \sin\theta_6 \tag{7}$$

In the formula: s is the stroke of the slider; v is the slider speed; a is the acceleration of the slider; ω_5 and ω_6 is the angular velocity; l_5 and l_6 are rod lengths; a_5 and a_6 are angular acceleration.

2.3. Multi-objective optimization analysis of crank double toggle mechanism

The mechanical design process generally involves many design parameters. Some design parameters have a greater impact on mechanical performance, while others have a smaller impact. Therefore, design parameters that have a significant impact on mechanical performance should be selected as design variables.

There are numbers of design parameters in the crank double toggle mechanism. If all design parameters are taken as design variables, it will increase the complexity of the optimization process, leading to an expansion of the search scope, making it difficult to obtain the optimization solution, or even failing to solve it. Therefore, by using some methods to treat design parameters that have a significant impact on mechanical performance as variables and design parameters that have a small impact on mechanical performance as constants, the number of design variables can be significantly reduced, thereby improving the optimization process [13].

The ADAMS/Insight module [14] provides a very powerful parameterized modeling function, which allows for easy modeling by setting relevant design parameters as design variables that can be changed. The design research module can obtain the degree of impact of changes in various design variables on the model, which is called the sensitivity of design variables. The sensitivity of design variables can be represented as follow.

$$S = \frac{O_{i+1}}{V_{i+1}} \times \frac{O_i}{V_i} + \frac{O_i - O_{i+1}}{V_i - V_{i+1}} \tag{8}$$

In the equation, S represents the sensitivity of design variables, O represents the target value, V represents the design parameter value, i represents the number of iterations.

Before optimization, the dimensions of each member of the crank double toggle linkage mechanism are shown in Table 1. The optimization design of a crank double toggle linkage mechanism is to optimize its performance by changing the length of each rod.

Table 1

Length of each rod of crank double toggle linkage Mechanism (mm)

Rod	l_1	l_2	l_3	l_4	l_5	l_6
Length	80	230	250	120	180	230

2.3.1. Selection of design variables

According to Figure 1(a), there are 6 design parameters for the mechanism, namely, $X^{'}=[x_1,x_2,x_3,x_4,x_5,x_6]=[l_1,l_2,l_3,l_4,l_5,l_6]$. Set the higher sensitivity as a variable. In order to find out the mechanism design parameters with high sensitivity to each optimization goal, the ADAMS optimization design module is used to analyze each design parameter individually, and obtain the sensitivity of each design parameter to slider speed and acceleration [15], as shown in Table 2. As can be seen from Table 2, the design parameters l_1, l_2, and l_3 are highly sensitive to speed and acceleration. Thereby, The design variables are l_1, l_2, and l_3, namely, $X=[x_1,x_2,x_3]=[l_1,l_2,l_3]$.

Table 2

Sensitivity of each design parameter to slider speed and acceleration

Variables	l_1	l_2	l_3	l_4	l_5	l_6
Sensitivity to speed	26.412	-1.652	-3.2568	-0. 4287	0. 3810	-1. 5889
Sensitivity to acceleration	93.621	-7.4985	-12.3622	-4. 7263	2. 4931	-4. 2297

2.3.2. Optimization design

When using ADAMS for parametric modeling, the size of the rod is adjusted by changing the coordinates of the hinge points, and its kinematic characteristics are calculated. The design variables corresponding to each hinge point are shown in Table 3, and $\pm 10\%$ of each coordinate value is taken as range.

Table 3

Parametric coordinate points

Coordinate points	X coordinate	Y coordinate
O	0	0
A	DV_1	DV_2
B	DV_3	DV_4
C	DV_5	DV_6
D	DV_7	DV_8
E	DV_9	DV_10

2.3.3. Constraint condition

(1) The conditions for the existence of OABC crank in hinged four-bar mechanism.

As can be seen from Figure 1(a), the hinge four bar mechanism OABC is a crank rocker mechanism, OA is a crank, and the rotation pair O is an integral rotation pair. According to the conditions for the existence of the crank, there are three constraints [16] as follows:

$$OA + AB < BC + OC$$
$$OA + BC < AB + OC$$
$$OA + OC < AB + BC$$

These can be represented by design variables:

$$g_1(X) = l_1 + l_2 - l_3 + \sqrt{X_c^{\ 2} + Y_c^2} < 0 \tag{9}$$

$$g_2(X) = l_1 + l_3 - l_2 + \sqrt{X_c^{\ 2} + Y_c^2} < 0 \tag{10}$$

$$g_3(X) = l_1 + \sqrt{X_c^{\ 2} + Y_c^2 - l_2 + l_3} < 0 \tag{11}$$

(2) The conditions for the existence of OABDE crank in the hinged five-bar mechanism [17] as follows:

$$g_4(X) = l_1 + l_4 + l_2 - l_5 + \sqrt{X_e^{\ 2} + Y_e^2} \le 0 \tag{12}$$

2.3.4. Optimization objectives

The optimization design aims at reducing the maximum speed and acceleration [18] during the working stroke of the mechanism. In summary, the mathematical model for the optimization design of the crank double toggle linkage mechanism is as follows:

$$f_1(X) = |v|_{\max}$$
$$f_2(X) = |a|_{\max}$$
$$g_u(X) \le 0 (u = 1,2,,3,4)$$
$$X = \begin{bmatrix} DV\_1, DV\_2, DV\_3, DV\_4, DV\_5, \\ DV\_6, DV\_7, DV\_8, DV\_9, DV\_10 \end{bmatrix}$$

2.3.5. Optimization calculation

The optimization calculation is performed using the GRG (Generalized Reduced Gradient Method) optimization algorithm, which first converts inequality constraints into equality constraints [19]. For the above inequality constraints, we can introduce six relaxation variables as follow:

$$M = [m_1, m_2, m_3, m_4, m_5, m_6]^T$$

In order to make $g_u(x) + M = 0$, among which, m_4, m_5, m_6 can be represented linearly by m_1, m_2, m_3, so only three relaxation variables $M = [m_1, m_2, m_3]^T$ need to be introduced. The design variables are as follows:

$$X = \begin{bmatrix} DV\_1, DV\_2, DV\_3, DV\_4, DV\_5, DV\_ \\ 6, DV\_7, DV\_8, DV\_9, DV\_10, m_1, m_2, m_3 \end{bmatrix}^T$$

The base variable is $X_B = [DV\_1, DV\_2, DV\_3]^T$ and the non-base variable is $X_N = [DV\_4, DV\_5, DV\_6, DV\_7, DV\_8, DV\_9, DV\_10, m_1, m_2, m_3]^T$. The base variables can be expressed linearly by non-base variables, so the objective function becomes an expression for non-base variables as $f(X) = f(X_B, X_N) = F(X_N)$.

To sum up, iterative calculation can be performed according to the general steps of the generalized reduced gradient method [20].

3. Results

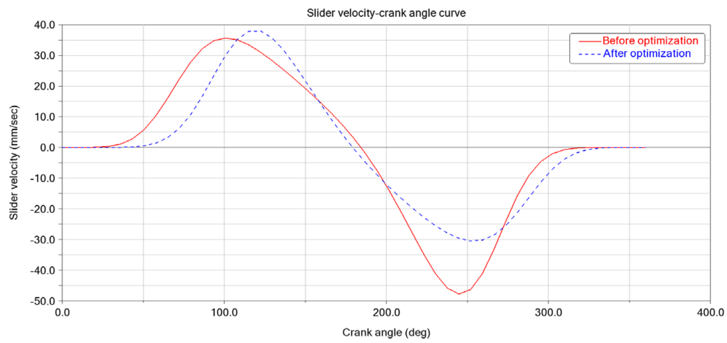
After writing a program and performing calculations in ADAMS, the calculation results are shown in Table 4.

Table 4

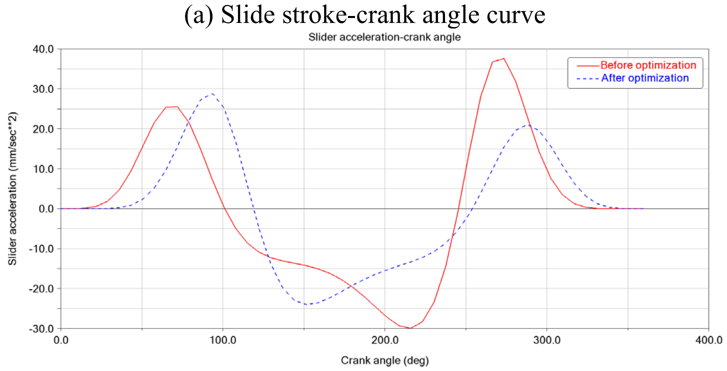
Calculation results

Design variables	Initial value	Optimization value
DV_1	76.549	80.254
DV_2	25.458	23.184
DV_3	314.315	323.412
DV_4	100.581	104.127
DV_5	316.325	305.275
DV_6	-147.825	-154.362
DV_7	430.244	462.151
DV_8	24.235	23.156
DV_9	433.117	438.235
DV_10	204.221	206.985

Considering the actual situation, the length dimensions of each rod converted from the optimization results are adjusted to the nearest integer value, and the adjustment results are as follows: $l_1 = 77mm$, $l_2 = 229mm$, $l_3 = 258mm$, $l_4 = 123mm$, $l_5 = 180mm$, $l_6 = 248mm$. Comparative analysis of characteristics of the mechanisms before and after optimization is shown in Figure 5. As shown in Figure 5(a), the maximum speed of the slider decreases by 37.5% from $48mm/s$ to $30mm/s$. As can be seen from Figure 5(b), the maximum acceleration of the slider has decreased by 23.7% from $38mm/s^2$ to $29mm/s^2$.



(a) Slide stroke-crank angle curve



(b) Slide velocity-crank angle curve

Fig. 5. Comparison of operating characteristics of crank double toggle mechanism before and after optimization

4. Conclusion

(1) It is important to choose the type of transmission mechanism according to the special working requirements of servo mechanical presses. Compared with the crank triangle toggle mechanism, the crank double toggle mechanism has better pressure retaining performance and quick return characteristics, and is more suitable for processing workpieces with large forming rebound. However, due to the high speed and acceleration of the crank double toggle linkage mechanism, the optimization design is needed.

(2) Using the ADAMS/Insight module, the impact of changes in design variables on changes in the objective function can be obtained. The absolute value of the degree of influence is large, indicating that the design variable is highly sensitive to optimization objectives. Conversely, it is less sensitive to optimization objectives. Setting design variables that are less sensitive to optimization goals as constants can reduce the number of design variables, thereby reducing the difficulty of optimization work.

(3) The optimization calculation is carried out using the GRG (Generalized Reduced Gradient Method) optimization algorithm. The advantages of the GRG method mainly lie in its ability to eliminate variables to achieve the goal of reducing the dimensionality of the problem, thereby accelerating the convergence speed of the algorithm, which means accelerating the speed of optimization design.

(4) The analysis of the optimization results in ADAMS shows that the performance of the crank double toggle mechanism has been greatly improved.

There are still some shortcomings in this paper, which is not systematic and comprehensive enough. The future research can be carried out from the following aspects:

(1)This paper was conducted in the virtual prototype analysis software ADAMS and did not consider the impact on the transmission mechanism in actual situations. Therefore, the next step is to conduct experimental analysis on the transmission mechanism under the condition of having a physical prototype.

(2)This paper did not address the structural dimensions of the transmission mechanism. Therefore, the next step is to design the structure of the transmission mechanism and conduct finite element analysis on each member to study the force changes of each member.

(3)This paper used optimization methods such as ADAMS and GRG optimization algorithm to optimize the design of transmission mechanisms. There are many methods for optimizing transmission mechanisms, and the selection of optimization objectives has a significant impact on the optimization results. Therefore, different optimization methods can be used to optimize the transmission mechanism and compare and analyze the optimization results.

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REFERENCES

[1] *A. Del Prete and T. Primo*, "Sheet Metal Forming Optimization Methodology for Servo Press Process Control Improvement", Metals, **vol. 10**, no. 2, 2020, pp. 271. DOI:10.3390/met10020271

[2] *S. Kitayama, T. Higuchi, M. Takano and A. Kobayashi*, "Determination of back-pressure profile and slide motion of servo press in cold forging using sequential approximate optimization", Journal of Advanced Mechanical Design, Systems, and Manufacturing, **vol. 14**, no. 4, 2020, pp. JAMDSM0046. DOI:10.1299/jamdsm.2020jamdsm0046

[3] *A. Reza and M. Moslehi*, "Optimization synthesis of an adjustable six-bar path generator mechanism using a cuckoo optimization algorithm", Journal of Mechanics of Materials and Structures, **vol. 17**, no. 2, 2022, pp. 149-167. DOI:10.2140/JOMMS.2022.17.149

[4] *M. T. Orvañanos-Guerrero, M. Acevedo, C. N. Sánchez, D. U. Campos-Delgado, A. A. Ghavifekr, P. Visconti and R. Velázquez*, "Complete balancing of the six-bar mechanism using fully cartesian coordinates and multiobjective differential evolution optimization", Mathematics, **vol. 10**, no. 11, 2022, pp. 1830. DOI: 10.3390/MATH10111830

[5] *V. Acharya, K. V. Kumar, B. Gopalsamy and B. S. Suresh*, "Genetic Algorithm based Kinematic Synthesis of an Eight Bar Flap Deployment Mechanism in a Typical Transport Aircraft", Materials Today: Proceedings, **vol. 5**, no. 11, 2018, pp. 24887-24897. DOI: 10.1016/j.matpr.2018.10.288

[6] *H. K. Kim and J. K. Shim*, "Dimensional Synthesis of a Stephenson III Six-Bar Slider-crank Function Generator Using Genetic Algorithm", Transaction of the Korean Society of Automotive Engineers, **vol. 28**, no. 5, 2020, pp. 333-338.DOI:10.7467/ksae.2020.28.5.333

[7] *A. Tuleshov, B. Akhmetova and B. Merkibayeva*, "Analytical Synthesis and Study of the Functional Capabilities of the Stephenson II Mechanism for Crank Presses", International Journal of Mechanical Engineering and Robotics Research, **vol. 11**, no.11, 2022, pp. 850-857. DOI:10.18178/IJMERR.11.11.850-857

[8] *M. L. Song, Y. X. Fan and L. Zu*, "Imitation and Optimization Design of six-bar Linkage Punch Mechanism Based on ADAMS", Journal of Nanjing University of Science and Technology, **vol. 30**, no. 3, 2006, pp. 285-287. DOI:10.14177/j.cnki.32-1397n.2006.03.009

[9] *C. H. Yang and X. Shao*, "Kinematics modeling and analysis on toggle mechanism for mechanical press", Forging & Stamping Technology, **vol. 42**, no. 12, 2017, pp. 87-91. DOI:10.13330/j.issn.1000-3940.2017.12.016

[10] *V. Korendiy, O. Kachur, V. Gursky, V. Gurey, N. Maherus, O. Kotsiumbas and O. Havrylchenko*, "Kinematic and dynamic analysis of three-mass oscillatory system of vibro-impact plate compactor with crank excitation mechanism", Vibroengineering PROCEDIA, **vol. 40**, 2022, pp. 14-19. DOI: 10.21595/VP.2022.22393

[11] *D. Y. Sari, H. Nurdin, and A. Yufrizal*, "The Use of MATLAB in Learning the Velocity Analysis with Relative Velocity Method on Slider Crank Mechanism", Journal of Physics: Conference Series, **vol. 1940**, no. 1, 2021, pp. 012078. DOI: 10.1088/1742-6596/1940/1/012078

[12] *T. X. Yan and X. X. Li,* "The Kinematics Analytical Method and Simulation with ADAMS Based on Bias Slider-Crank Mechanism", Applied Mechanics and Materials, **vol. 602-605**, 2014, pp. 185-188. DOI: 10.4028/www.scientific.net/AMM.602-605.185

[13] *Y. F. Zhu, G. Wang and Y. W. Yao*, "Multiobject Optimum Design and Simulation of Planar Combination Linkage Mechanism", Journal of Gun Launch and Control, no. 4, 2008, pp. 94-96. DOI: 10.19323/j.issn.1673-6524.2008.04.024

[14] *Q. Y. Song, J. Li and W. Q. Yin*, "Mechanical press six-link mechanism design based on multi-objective", Transactions of the Chinese Society for Agricultural Machinery, **vol. 43**, no. 4, 2012, pp. 225-229+234. DOI: 10.6041/j.issn.1000-1298.2012.04.043

[15] *Y. X. Fan, B. Mou and C. C. Guan*, "Optimization Design of Eight-bar Press Based on ADAMS", Mechanical Building & Automation, **vol. 43**, no. 2, 2014, pp. 105-107. DOI: 10.19344/j.cnki.issn1671-5276.2014.02.032

[16] *K. Suresh, K. Shankar and C. Sujatha*, "A four bar mechanism as dynamic magnifier for improved performance of multi-modal piezoelectric harvester beams", The European Physical Journal Special Topics, **vol. 231**, no. 8, 2022, pp. 1373-1382.DOI: 10.1140/EPJS/S11734-022-00505-W

[17] *A. Parnianifard, A. Zemouche, R. Chancharoen, M. A. Imran and L. Wuttisittikulkij*, "Robust optimization design of FOPID controller for five bar linkage robot in a Cyber-Physical System: A new simulation-optimization approach", PLoS ONE, **vol. 15**, no. 11, 2020, pp. e0242613. DOI: 10.1371/JOURNAL.PONE.0242613

[18] *X. N. Chen*, Mechanical optimization design, Version 2, Hangzhou: Zhejiang University Press, pp. 119-120, 2010.

[19] *M. S. Toplu and D. S. Körpe*, "Development of an optimization solver with generalized reduced gradient method", Sakarya University Journal of Science, 2018. DOI: 10.16984/saufenbilder.298972

[20] *H. W. Li*, "Application of generalized reduced gradient method in mechanical design. Mechanical and Electrical Engineering Technology", **vol. 39**, no. 5, 2010, pp. 75-77+113. DOI: 10.13434/j.cnki.1007-4546.2010.02.005