

DESIGN OF MULTI-LINK TRANSMISSION FOR BOTTOM-DRIVE FAST PRESS

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In this paper, a multi-link transmission mechanism is designed for the bottom-drive fast press, which adopts the combination of gears and connecting rods, with a more compact structure and a wider range of velocity. In order to facilitate the analysis, it is simplified into a double crank mechanism for kinematics calculation and analysis. At the same time, MATLAB/SimMechanics module is used to conduct modeling and simulation analysis. In addition, the multi-link transmission mechanism is further optimized through variable control, which provides a reference for the design of the bottom-drive fast press.

Keywords: Transmission mechanism; Bottom-drive press; MATLAB

1.Introduction

In recent years, with the rapid development of electronics, home appliances, and the automotive industry, the demand for stamping parts has also increased[1]. High-speed precision presses can produce all kinds of stamped parts accurately,

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efficiently, and safely [2-3]. The famous foreign forging manufacturers such as BRUDERER in Switzerland, Schuler in Germany, MINSTER in the United States, KYORI in Japan, etc. have achieved speed breakthroughs in high-speed presses as far back as the 1970s. In the 80's, they entered the competition of precision control and formed a mature precision control technology [4]. However, due to the late start in China, the development of press machine is not ideal.

High-speed precision presses often use the crank-slider mechanism as a transmission mechanism, which converts the rotary motion of the motor into a reciprocating linear motion of the slider. The speed of the slider changes greatly around the bottom dead center, the working stroke is not uniform, and the speed of stamping the workpiece is high, which all easily affect the punching precision of the press. Therefore, the working condition of the press transmission mechanism directly determines the working performance of the press [5-6].

With the emergence of new technologies and new requirements for stamping processes, the development of high-speed presses has experienced servo motor drives, multi-link mechanisms, elliptic gear mechanisms, and planetary gear mechanisms [7].

In order to better realize the speed control of stamping, Tingting Zhao et al[8] used the full speed reduction function of the switched reluctance, canceled the flywheel and the clutch, and designed a new type of switch servo press transmission system; Yujian Xiang et al[9] used genetic algorithms to optimize the design of the linkage parameters, and a double-crank main transmission system for a two-point servo press was designed; H.M. Daniali et al[10] proposed a novel algorithm for kinematic and dynamic optimal synthesis of planar four-bar mechanisms with joint clearance.

In order to better adapt to the stretching process characteristics, Yanzhong He et al[11] designed a new type of transmission mechanism of mechanical stretching press with low speed and long full load working time around the bottom dead center, with the help of simulation analysis software ADAMS. In addition, the optimization algorithm combining penalty function with compound form[12], the analysis method of probabilistic modeling and reliability analysis combined[13], analysis on dynamic precision reliability based on Monte Carlo method[14] etc. are all applied into the design of the transmission mechanism of the press.

From the above analysis, it can be seen that in order to meet different requirements, different methods can be used to design the transmission

mechanism of the press. This paper designs a multi-link transmission mechanism for bottom drive presses for cap manufacturing. Firstly, the structure design of the transmission mechanism is carried out. Then the mechanism is optimized by kinematics calculation and modeling simulation analysis. Finally, the multi-link transmission mechanism is obtained.

2. The design of the transmission mechanism

This paper designs the transmission mechanism of bottom drive press based on the JD-100 press. The problems existing in the original press, such as high fuselage size, high center of gravity of press and pollution of stamping parts caused by oil leakage, have been solved. The press adopts the method of bottom drive, and the linkage system is connected with the crank-slider mechanism, which forms the multi-link transmission mechanism designed in this paper. In order to make the slider achieve the effect of non-uniform motion (especially the characteristic of "slow-forward and quick-return"), the multi-link mechanism drives the eccentric shaft in a form similar to the motion of the double crank mechanism. Take the multi-link mechanism shown in Fig. 1 as an example, the large gear 15 is mounted in the housing 9 and is meshed with the pinion 11A and can be rotated freely. In the hollow cylinder portion of the large gear 15, the connecting rod 20 is connected to the eccentric 18A. The pinion 11A drives the large gear 15 to rotate at a certain speed. At this time, the side crankshaft 18 is driven to rotate at unequal speeds.

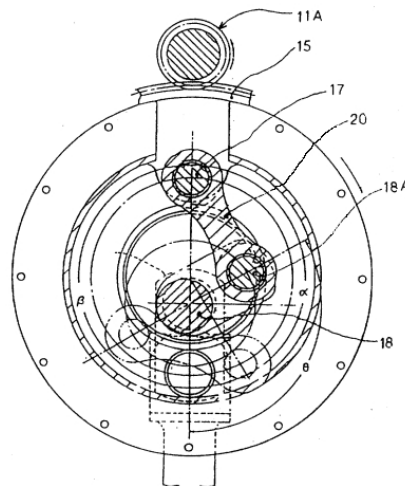


Fig. 1 Schematic diagram of multi-link mechanism

In order to facilitate the analysis, this mechanism can be simplified as a double-crank mechanism. The schematic diagram of the mechanism is shown in Fig. 2:

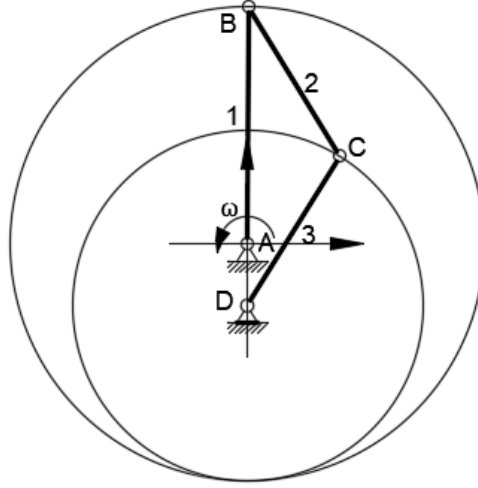


Fig. 2 schematic diagram of double-crank mechanism

Regarding the design of the double-crank mechanism, there is no unified solution. This paper designs the double-crank mechanism with minimum transmission angle. First, set the length of the mechanism to be a , b , c , d respectively. The condition that the four-bar linkage mechanism realizes double crank is that the length of the shortest bar plus the longest bar is less than or equal to the length of the remaining two bars, and the shortest bar is used as the frame. Suppose a is the longest bar and is the active rod, d is the frame; let $d=1$, and use the relative size to denote the mechanism:

$$\frac{a}{d} : \frac{b}{d} : \frac{c}{d} = a : b : c$$

After consulting the "Mechanical Design Handbook"[15] and performing related modeling and simulation analysis, the following results are obtained:

$$a : b : c : d = 200 : 175 : 150 : 50$$

At this point, the active rod reaches the corresponding size and the eccentricity is also within the appropriate range.

3. Kinematic analysis of multi-link transmission mechanism

For the general analysis of multi-link mechanism, the relevant mathematical model is firstly built, as shown in Fig.3. The number of original motive parts is

equal to the degree of freedom of the mechanism, so the mechanism has determined movement. The planar four-bar mechanism is a closed loop mechanism, which is solved by the closed vector polygon method.

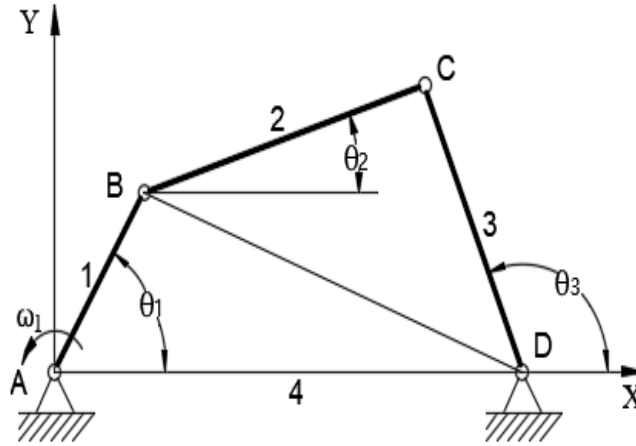


Fig. 3 Kinematic sketch of Plane four-bar mechanism

The closed graph ABCDA can be used to write closed vector equations, surrounded by the rods of the mechanism:

$$\vec{l}_1 + \vec{l}_2 = \vec{l}_3 + \vec{l}_4$$

Decompose each vector into X and Y axis and get the following equations:

$$\begin{cases} l_1 \cos \theta_1 + l_2 \cos \theta_2 = l_3 \cos \theta_3 + l_4 \\ l_1 \sin \theta_1 + l_2 \sin \theta_2 = l_3 \sin \theta_3 \end{cases} \quad (2)$$

Take the equation (1) to take the first derivative for time t, and get the velocity relation group:

$$\begin{cases} l_1 \omega_1 \sin \theta_1 + l_2 \omega_2 \sin \theta_2 = l_3 \omega_3 \sin \theta_3 \\ l_1 \omega_1 \cos \theta_1 + l_2 \omega_2 \cos \theta_2 = l_3 \omega_3 \cos \theta_3 \end{cases} \quad (3)$$

Among them, ω_1 , ω_2 and ω_3 are the angular velocity of rod 1, rod 2 and bar 3 respectively.

If expressed in matrix form, the above formula can be written as:

$$\begin{bmatrix} -l_2 \sin \theta_2 & l_3 \sin \theta_3 \\ l_2 \cos \theta_2 & -l_3 \cos \theta_3 \end{bmatrix} \begin{bmatrix} \omega_2 \\ \omega_3 \end{bmatrix} = \omega_1 \begin{bmatrix} l_1 \sin \theta_1 \\ -l_1 \cos \theta_1 \end{bmatrix} \quad (4)$$

Take the equation (1) to take the second derivative for time t , and get the acceleration relation group:

$$\begin{bmatrix} -l_2 \sin \theta_2 & l_3 \sin \theta_3 \\ l_2 \cos \theta_2 & -l_3 \cos \theta_3 \end{bmatrix} \begin{bmatrix} \alpha_2 \\ \alpha_3 \end{bmatrix} + \begin{bmatrix} -l_2 \omega_2 \cos \theta_2 & l_3 \omega_3 \cos \theta_3 \\ -l_2 \omega_2 \sin \theta_2 & l_3 \omega_3 \sin \theta_3 \end{bmatrix} = \omega_1 \begin{bmatrix} l_1 \omega_1 \cos \theta_1 \\ l_1 \omega_1 \sin \theta_1 \end{bmatrix} \quad (5)$$

Among them, α_2 and α_3 are the angular acceleration of rod 2 and rod 3 respectively.

Since the geometric parameters of each rod are determined, the equation with variable can be obtained by using geometric operation, which represents the variation of positions of certain points at each time. Therefore, it can preliminarily analyze whether the mechanism has met the design requirements and make reference for the subsequent optimization.

4. Modeling and simulation analysis based on MATLAB/SimMechanics

According to the above equations of motion, we compile the programs. Since the value range of the trigonometric function does not satisfy the entire motion process, only piecewise functions can be used to compile programs. Import the edited program into command window for simulation analysis, and get the curve shown in Fig.4:

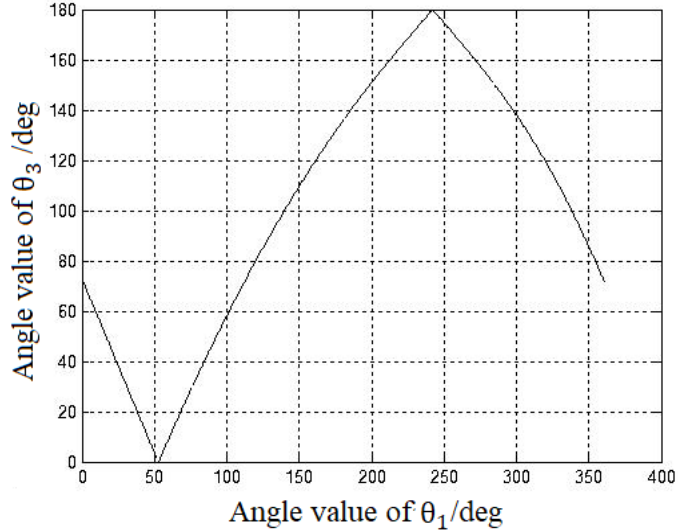


Fig. 4 Angle change curve of θ_3

Through the programming analysis of the double crank position, it is found that the function of the inverse trigonometric function is not very applicable enough for the double-crank mechanism, and it is necessary to compile the piecewise function according to the position of follower rod. Therefore, the SimMechanics module in Simulink will be used for modeling and simulation analysis of the double crank mechanism.

In order to make the modeling process simple, the connecting rods are placed in a special position, as shown in the Fig.5:

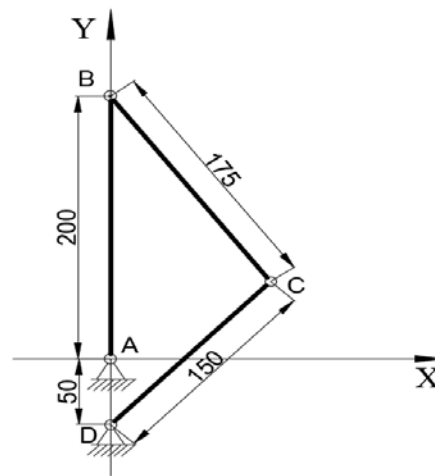


Fig. 5 Coordinate schematic diagram of a double crank mechanism

Next, set up a simulation model, the results are as follows:

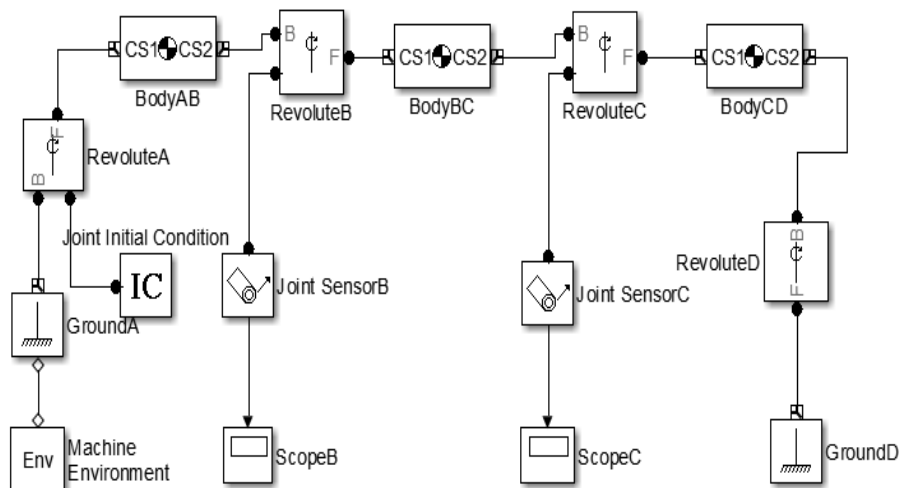


Fig.6 Simulation model of a double crank mechanism

Then, set the module parameters of the established simulation model, simulate and get the change curve of the angular velocity and angular acceleration of the rod CD.

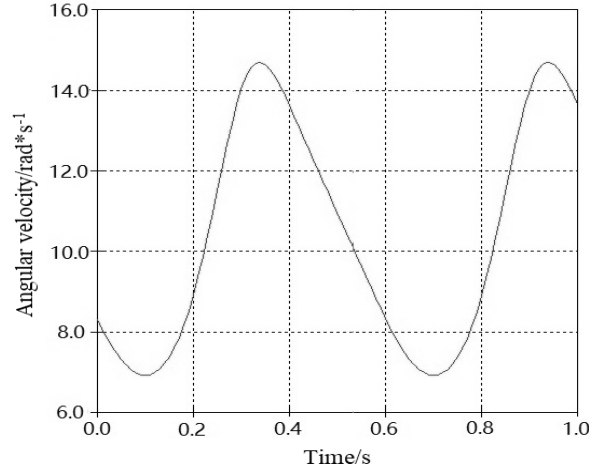


Fig.7 The angular velocity variation curve of rod CD

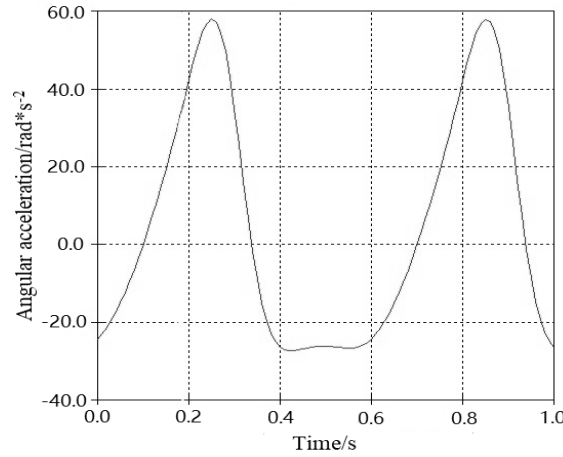
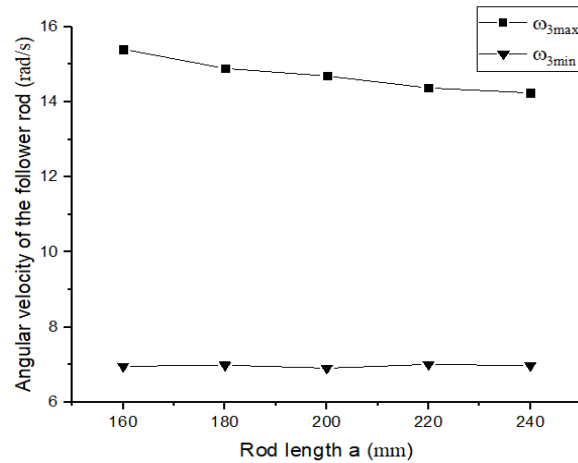


Fig.8 The angular acceleration variation curve of rod CD

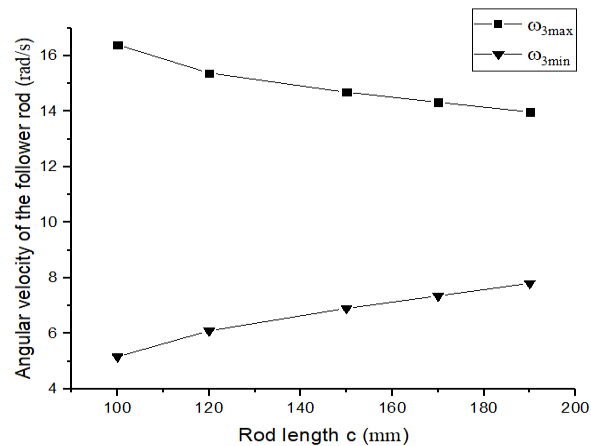
As can be seen from Fig.7, the angular velocity of rod CD fluctuates widely, $\omega_{3\max} = 14.69\text{rad/s}$, $\omega_{3\min} = 6.91\text{rad/s}$, so the rod CD has the characteristic of "slow-forward and quick-return". According to different design requirements, different output angular velocity can be obtained by adjusting the rod length. Combining the velocity variation of the connecting rod with the change of the sinusoidal velocity of the eccentric, ultimately, the velocity variation of the slider will be wider.

5.The influence of rod length variation on angular velocity of driven rod

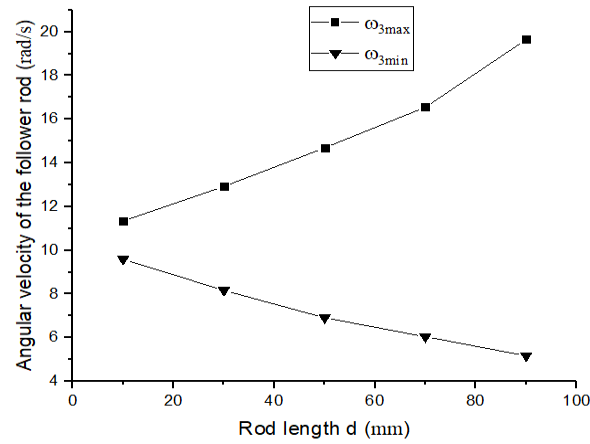
Changes in rod lengths a , b , c , d will affect the change in angular velocity and angular acceleration of the follower rod. However, the double-crank mechanism is limited by the hollow gear, so the four variables cannot be arbitrarily changed. The following we will use the variable control method to analyze the influence of the changes of the rod length a , c , and d on the angular velocity of the follower rod with a certain length, and draw its extreme value line chart, as shown in the Fig.9:



(1) Rod 1



(2) Rod 3



(3) Rod 4

Fig.9 Diagram of the extreme value change of the follower rod's angular velocity

For the above analysis, the rod length a and c is reduced, and the rod length d increases, which will widen the angular velocity curve of the follower rod. However, the change of angular velocity of follower rod caused by the change of the rod length a is not obvious, so when designing, it is mainly considered to design the length of the follower rod 3 and the distance between two rotation centers. The reasonable connecting rod length makes the range of angular velocity of the follower rod wider, the velocity near the bottom dead center is smaller, the precision of the stamping parts is higher, and the returning speed is also greater, which improves the working efficiency. In addition, the spacing of the two centers corresponds to the rod length d. In order to meet the multi-link installation in the large gear, the motion trajectory of the follower rod 3 must be located in the large gear.

Finally, the multi-link transmission mechanism designed is shown in the Fig.10:

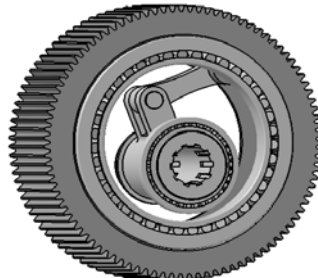


Fig.10 Assembly drawing of transmission mechanism

6. Conclusions

In this paper, a multi-link transmission mechanism of the bottom drive press is designed. On the basis that the transmission mechanism is simplified as a double-crank mechanism, kinematics analysis and MATLAB modeling simulation analysis are performed. Finally, an ideal design scheme is obtained.

(1) The stamping speed of the bottom dead center is the key data of the punching machine, which is directly related to the precision of the stamping parts. The current crank slider press can only provide sinusoidal motion, so the velocity curve is sinusoidal. The characteristic of “slow forward and quick return “can not be well represented. In this paper, the multi-link mechanism is combined with sine mechanism, so that the range of velocity variation of slider is larger, and it can improve the speed of returning while reducing the stamping speed, which can improve the punching precision and working efficiency.

(2) This article adopts the transmission mechanism with double-crank mechanism arranged in the hollow gear, not only embodies the advantage of double-crank mechanism velocity variation, but also makes the integrated layout of the transmission mechanism more compact, reducing the size of the fuselage and the difficulty of manufacture.

(3) In addition, the position of the rod, the output phase angle and other factors will affect the stamping speed of the slider, which can provide reference for the optimal design of the transmission mechanism of the bottom drive press.

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