DESIGN AND ANALYSIS OF PRESS MECHANISM DRIVEN BY ECCENTRIC GEAR

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In order to improve the comprehensive performance of mechanical presses, this paper proposes a press driven by a modified eccentric gear with virtual constraints. To analyze the motion characteristics of this type of press, a kinematic mathematical model of the press was established by using geometric - mathematical relationships. A design - analysis software for analyzing the motion characteristics of the press was developed using MATLAB. Taking the quick - return characteristic as the objective function, the design parameters of the modified eccentric gear as the optimization parameters, and the convexity preservation of the non - circular gear pitch curve as the constraint condition, the genetic algorithm GA was used to optimize the press, and the motion characteristics of the press under the optimized parameters were analyzed and a virtual prototype was assembled. The results show that the designed press has the quick - return characteristic on the basis of the low - speed and smooth forging - punching process characteristics. Compared with the double - center crank - slider press, the advantages of the press proposed in this paper are highlighted.

Keywords: press mechanism; non - circular gear; quick - return characteristic; virtual constraint; genetic algorithm

1. Introduction

Mechanical presses [1]are one of the most important equipment in the plastic forming process and are widely used in various industrial sites. At present, the commonly used driving structures of presses include the double - center crank - slider structure [2], the offset crank - slider structure, the multi - link structure [3], and the cam - driven structure [4], etc.

Minhai, M.A.[5] proposed an optimization design method for mechanical presses integrating the analytic hierarchy process, the penalty function method, and the genetic algorithm. Compared with the existing presses, the press parameters

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optimized by this optimization design method effectively reduce the slider speed, increase the stroke - speed ratio, and reduce the press height; X. Dang [6] investigated the kinematics and dynamics of the hybrid-driven seven-bar mechanical press, the numerical simulation implemented based on MATLAB, and illustrated the dynamic performance of the mechanical press better than before. Xun Huang [7] proposed a new press driven by three servo motors in parallel, and the kinematics analyses of the press were conducted based on fuzzy theory, and numerical simulation was conducted using a typical-drawing process, the accuracy of kinematics model of the new press was proved by ADAMS. F. Jia [8] proposed a 3-degree-of-freedom dynamic model of mass-spring damping for forecasting the dynamic responses of press mechanism, the accuracy of dynamic model of press is proven by simulation and test. Guo Weizhong [9] proposed a new servo mechanical press driven by small servomotor, and a set of design parameters were obtained by a visual based global optimization method, and provided a feasible solution for developing the performance of heavy-duty servo mechanical press. Wang Zhongshuang [10] used the phasor bond graph method to design and analyze the press, realizing the automatic modeling and simulation analysis of the press mechanism dynamics, which improved the efficiency of press dynamics modeling and simulation to a certain extent; Zhao Shengdun [11] constructed a mathematical model of the mechanical press using the geometric analysis method and optimized the press structure parameters using the genetic algorithm. The obtained press has better speed characteristics than cold extrusion and drawing presses.

To improve the performance of the press, based on the study of the motion characteristics of the press, a press structure driven by a modified eccentric gear [12] with virtual constraints [13] was proposed. The mathematical model of this type of press was constructed using geometric - mathematical relationships, and the design - analysis software for the motion characteristics of this type of press was developed using MATLAB; the genetic algorithm was used to optimize the design of this type of press, the motion characteristics of the optimized press were analyzed, and a virtual prototype assembly was established. Compared with the crank - slider press, it shows that the designed press has motion characteristics such as low - speed forging - punching and quick - return characteristics and can adapt to the plastic forming processing conditions.

2. Link - Driven Press Mechanism

The press with quick - return characteristics is shown in Fig.1. Its working principle is as follows: the power source drives the active modified eccentric gear

1 to rotate at a constant speed. The ratio of the pitch curve radii of the driving and driven gear changes with time [14]. Therefore, through the meshing transmission of the tooth profiles, the driven non - circular gear 2 rotates at a variable speed. The crank 3 and the driven non - circular gear are fixedly connected to the same drive shaft, so their motion characteristics are the same. Through the connecting rod 4, the power is transmitted to the slider 5 to make a variable - speed reciprocating linear motion; the connecting rods 7, 6 and the sliders 5, 8 form a connecting frame rod with a virtual constraint and a degree of freedom of 0, which enhances the comprehensive performance of the press; the rotation center O_2 of the driven non - circular gear, the hinge point C and the center hinge point B of the slider 5 are on the same vertical line, the hinge point C and the center hinge point E of the slider 8 are on the same horizontal line, and the length of the connecting rod CD l_3 is equal to half of the length of the connecting rod BE (that is, the lengths of the ED, DB, and CD segments are equal).

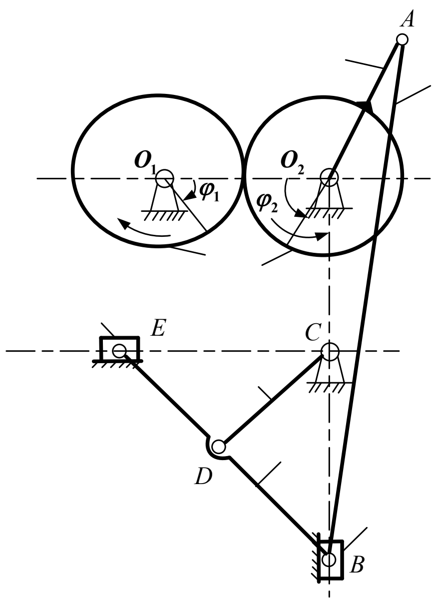

Fig.1 Schematic Diagram of Press Mechanism

3. Mathematical Model

To analyze the motion characteristics of the press, on the basis of analyzing the transmission path of the press, a kinematic mathematical model of the press was established using geometric - mathematical knowledge. To facilitate understanding, the parameters involved and their meanings are listed in Table1.

Table 1

Related Parameters

R	Radius of eccentric circle
e	Eccentricity
r_1	Radius of pitch curve of active modified eccentric circular gear
r_2	Radius of pitch curve of driven non - circular gear
m_{11}	First - stage modification coefficient
m_{12}	Second - stage modification coefficient
φ_1	Rotation angle of driving gear
φ_2	Rotation angle of driven gear
$\dot{\varphi}_2$	Angular velocity of driven gear
$\ddot{\varphi}_2$	Angular acceleration of driven gear
a	Center distance of gear pair
i_{12}	Transmission ratio of gear pair
l_1	Length of crank
l_2	Length of connecting rod 4
l_3	Length of connecting rod 7
l_4	Length of connecting rod 6
(x_A, y_A)	Coordinates of hinge A
(x_B, y_B)	Coordinates of hinge B
s	Displacement of slider
$\dot{s}$	Velocity of slider
$\ddot{s}$	Acceleration of slider

3.1 Modified Eccentric Gear

To establish the mathematical model of the modified eccentric gear, a coordinate system xO_1y was established with the rotation center O_1 of the active modified eccentric gear as the origin, as shown in Fig.2.

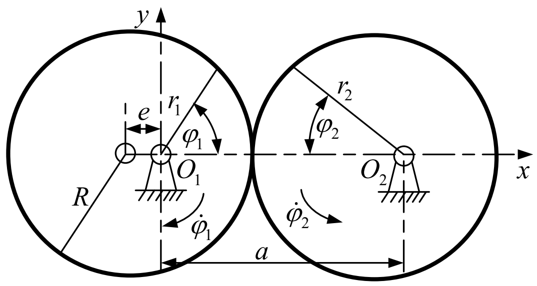

Fig.2 Modified Eccentric Gear

According to the geometric relationship, the mathematical model of the pitch curve of the active modified eccentric gear can be expressed as

$$r_1(\varphi_1)=\begin{cases}\sqrt{R^2-e^2\sin^2(m_{11}\varphi_1)}-e\cos(m_{11}\varphi_1) & \left(0\le\varphi_1\le\dfrac{\pi}{m_{11}}\right)\\ \sqrt{R^2-e^2\sin^2\left(m_{12}\left(2\pi-\varphi_1\right)\right)}-e\cos\left(m_{12}\left(2\pi-\varphi_1\right)\right) & \left(\dfrac{\pi}{m_{11}}\le\varphi_1\le 2\pi\right)\end{cases}$$

(1)

To ensure that the pitch curve of the active modified eccentric gear is closed [15], the two introduced modification coefficients need to satisfy the equation (2).

$$\frac{\pi}{m_{11}}+\frac{\pi}{m_{12}}=2\pi \tag{2}$$

To ensure that the modified eccentric gear pair can transmit continuously, that is, when the driving gear rotates one circle, the driven gear meshing and transmitting with it also rotates exactly one circle. According to the meshing principle of non - circular gears [17], the relationship between the rotation angles of the driving and driven gears needs to satisfy the following equation:

$$2\pi=\int_0^{2\pi}\frac{r_1(\varphi_1)}{a-r_1(\varphi_1)}\mathrm{d}\varphi_1=\int_0^{\frac{\pi}{m_{11}}}\frac{\sqrt{R^2-e^2\sin^2\left(m_{11}\varphi_1\right)}-e\cos\left(m_{11}\varphi_1\right)}{a-\left(\sqrt{R^2-e^2\sin^2\left(m_{11}\varphi_1\right)}-e\cos\left(m_{11}\varphi_1\right)\right)}d\varphi_1$$
$$+\int_{\frac{\pi}{m_{11}}}^{2\pi}\frac{\sqrt{R^2-e^2\sin^2\left(m_{12}\left(2\pi-\varphi_1\right)\right)}-e\cos\left(m_{12}\left(2\pi-\varphi_1\right)\right)}{a-\left(\sqrt{R^2-e^2\sin^2\left(m_{12}\left(2\pi-\varphi_1\right)\right)}-e\cos\left(m_{12}\left(2\pi-\varphi_1\right)\right)\right)}d\varphi_1$$

(3)

The numerical analysis was used to solve equation (3) to obtain the center distance of the gear pair that meets the requirements of continuous transmission of the driving and driven gears.

The specific solution process is as follows:

① Use the advancing - retreating method to obtain the interval range of the numerical value of the gear pair center distance;

② After obtaining the interval range, use the golden section method to obtain the accurate numerical value of the gear pair center distance.

After knowing the mathematical model $r_1(\varphi_1)$ of the pitch curve of the active modified eccentric gear and the center distance *a* of the gear pair, the equation of the pitch curve of the driven non - circular gear and the rotation angle mathematical model can be expressed as

$$\begin{cases} r_2(\varphi_1) = a - r_1(\varphi_1) \\ \varphi_2 = -\int_0^{\varphi_1} \frac{1}{i_{12}} d\varphi_1 = -\int_0^{\varphi_1} \frac{r_1(\varphi_1)}{a - r_1(\varphi_1)} d\varphi_1 \end{cases} \tag{4}$$

where the rotation angle φ_2 of the driven gear can be obtained by solving equation (4).

3.2 Crank - Slider Mechanism

To establish the mathematical model of the crank - slider, a coordinate system $x'O_2y'$ was established with the rotation center O_2 of the crank as the origin, as shown in Fig.3.

In the coordinate system $x'O_2y'$, the coordinate value of the hinge A is

$$\begin{cases} x_A = l_1 \cos\varphi_2 \\ y_A = l_1 \sin\varphi_2 \end{cases} \tag{5}$$

The coordinate value of the center hinge B of the slider 5 is

$$\begin{cases} x_B = x_A + l_2 \cos\alpha \\ y_B = y_A + l_2 \sin\alpha \end{cases} \tag{6}$$

where $\cos\alpha = (x_B - x_A) / l_2$. Since the press mechanism adopts the double - center crank - slider [18], at this time $x_B = 0$, so $\cos\alpha = -x_A / l_2$. The displacement mathematical model of the slider 5 can be expressed as

$$s = -y_B - (l_2 - l_1) \tag{7}$$

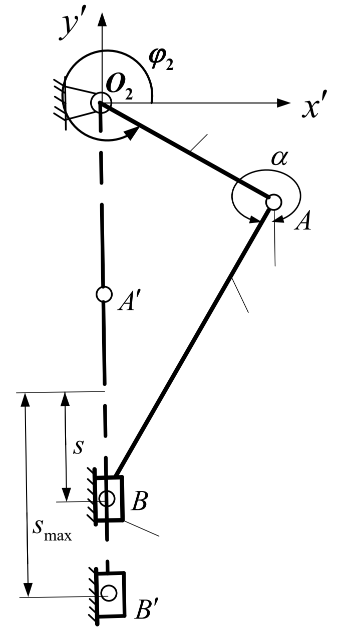

Fig.3 Crank - Slider Mechanism

From the above equations (5), (6) and (7), we can get

$$s = -(\sqrt{l_2^{\,2} - l_1^{\,2}\cos^2\varphi_2} + l_1\cos\varphi_2) - (l_2 - l_1) \tag{8}$$

Taking the first - order derivative of both sides of equation (8) with respect to time t, the velocity mathematical model of the slider 5 can be expressed as

$$\dot{s} = -\left[l_1(\cos(\varphi_2) + \frac{l_1\sin(2\varphi_2)/l_2}{2\sqrt{1-((l_1/l_2)^2\cos^2\varphi_2)}}) \right]\dot{\varphi}_2 \tag{9}$$

Taking the first - order derivative of both sides of equation (9) with respect to time t, the acceleration mathematical model of the slider 5 can be expressed as

$$\ddot{s} = \left(l_1\sin\varphi_2 - \frac{l_1^2\cos(2\varphi_2)}{\sqrt{l_2^2 - l_1^2\cos^2\varphi_2}} + \frac{\left(l_1^2\sin(2\varphi_2)\right)^2}{4\left(l_2^2 - l_1^2\cos^2\varphi_2\right)^{\frac{3}{2}}} \right)\dot{\varphi}_2 - \left(l_1\cos\varphi_2 + \frac{l_1^2\sin(2\varphi_2)}{2\sqrt{l_2^2 - l_1^2\cos^2\varphi_2}} \right)\ddot{\varphi}_2 \tag{10}$$

3.3 Connecting Frame Rod with Virtual Constraint

The connecting rod mechanism composed of connecting rods 6 and 7, sliders 5 and 8 is shown in Fig.4. According to the calculation formula of the degree of freedom of the planar mechanism, the degree of freedom of this mechanism can be obtained as 0. However, since the moving directions of sliders 5 and 8 are vertically distributed, and the lengths of connecting rod CD, segment DE, and segment BD are equal, a virtual constraint is introduced. Therefore, this mechanism can move. [11]

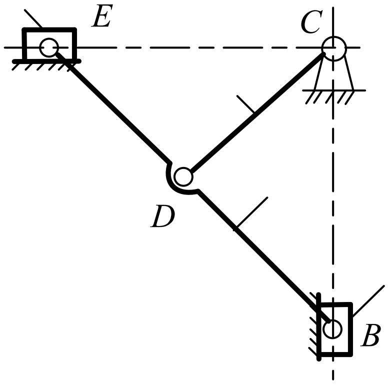

Fig.4 Connecting Rod with Virtual Constraints

4. Design - Analysis Software and Parameter Optimization

4.1 Design - Analysis Software

To facilitate the analysis of the influence of each design parameter on the transmission characteristics of the press, on the basis of the established mathematical model of the press, the analysis - design software of the press driven by the modified eccentric gear was developed using MATLAB, as shown in Fig. 5.

The designer inputs the design parameters in the software input parameter area. Through the button - controlled method, the software automatically calculates and displays the results in the corresponding areas of the software, and can also perform simulated motion to preliminarily verify the correctness of the established software and provide accurate design parameters for the subsequent structural design.

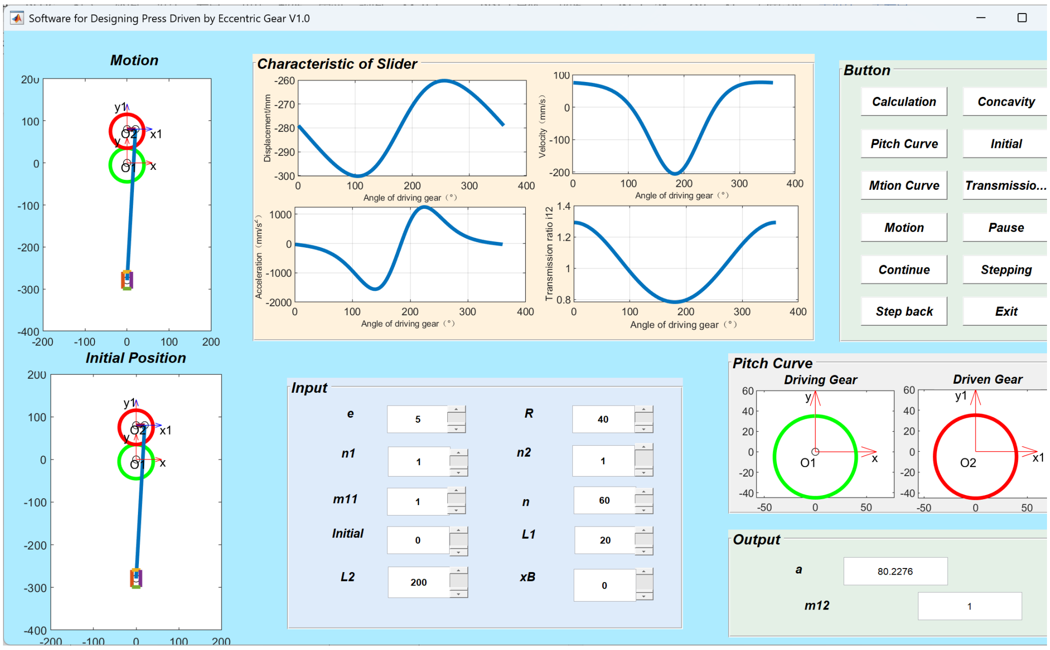

Fig.5 Analysis and Design Software

4.2 Parameter Optimization

To obtain a press that satisfies the quick - return characteristic function, when using the improved genetic algorithm[19] for optimization, the basic parameters are selected as shown in Tab.2. The specific optimization process is as follows:

Table 2

Related Parameters

Parameter	Value
Initial population number	20
Number of genetic generations	100
Number of binary digits of variables	20
Individual selection ratio between populations	0.9
Crossover probability	0.8
Mutation probability	0.1

Optimization parameters: Since the connecting rod structure only affects the length of the slider stroke and has no influence on the motion characteristics of the press; therefore, the motion characteristics of the press only depend on the design parameters of the modified eccentric gear. The connecting rod structure parameters are selected as $l_1 = 20mm$, $l_2 = 200mm$, and the connecting rods l_3 and l_4 do not affect any characteristics of the press. Therefore, the optimization parameter equation is

$$x_p = \left[R, e, m_{11}\right] \tag{11}$$

Optimization objective: The press has a quick - return characteristic. In essence, the ratio of the forward stroke time to the return stroke time is greater than 1, the larger the better, and conversely, less than 1, the smaller the better. Therefore, the optimization objective equation is

$$F = \frac{\overline{v}_h}{\overline{v}_t} \tag{12}$$

According to the set optimization objective, the smaller the F value, and the more obvious the quick - return characteristic of the press, and the higher the working efficiency of the press.

Constraint condition: To ensure the smooth transmission of the noncircular gear pair, the noncircular gear should meet the convexity preservation requirement [10]. Therefore, the established constraint equation is

$$\begin{cases} 1 + i_{12} + \ddot{i}_{12} \geq 0 \\ 1 + i_{12} - i_{12}\ddot{i}_{12} + \dot{i}_{12}^{2} \geq 0 \end{cases} \tag{13}$$

A set of design parameters was obtained by using the genetic algorithm for optimization. For the convenience of the subsequent prototype trial - production and considering factors such as the smoothness of the modified eccentric gear transmission, the optimized design parameters were rounded off to obtain

$$\left[R, e, m_{11}\right] = \left[40mm, 10mm, 0.8\right] \tag{14}$$

4.3 Virtual Prototype Assembly

The tooth profile of eccentric gear is enveloped by a shaper, the shaper's pitch circle rolls on the pitch curves of eccentric gear without sliding. In this paper, the mathematical model of tooth profiles of eccentric gear is based on the following equation:

$$\frac{dy}{d\theta} = \cos^2 \alpha_0 \left[R(\theta) - \frac{dR(\theta)}{d\theta} \tan \alpha_0 \right] \tag{15}$$

Where dy is the displacement of the contact point, as projected onto the normal direction to the eccentric and its conjugated gear crossing O_1 and O_2, respectively, $d\theta$ is the rotation of the eccentric gear, α_0 is the pressure angle of the shaper, the value is 20°. In order to calculate coordinates of the tooth profile of eccentric gear, a software for generating tooth profile is developed by a numerical interface based on MATLAB, as shown in Fig.6.

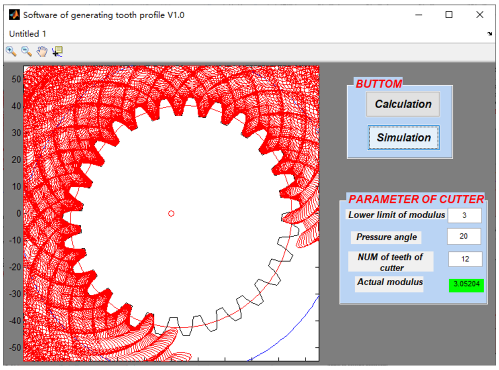

Fig.6 Software of generating tooth profile

The pitch curve of the modified eccentric gear obtained by the genetic algorithm optimization was generated by the generating method, and the corresponding gear tooth profile was obtained using the developed tooth profile generation software, as shown in Fig. 7.

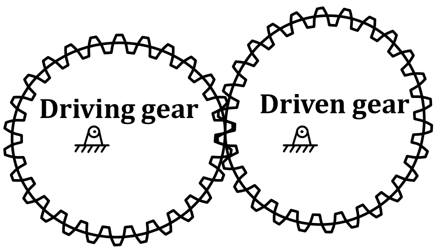

Fig.7 Pitch Curve and Tooth Profile

Fig.8 shows the motion characteristics of the optimized press, including the displacement, velocity, and acceleration curves of the slider. From the displacement curve in the figure, it can be seen that the ratio of the forward stroke time to the return stroke time of this mechanism is greater than 1, which has the quick - return characteristic, when the rotation angle of the driving gear is 232°, the slider is at the extreme position (with a displacement of 0mm), and at this time, the corresponding velocity of the slider is 0mm.s^{-1}, and the acceleration of the slider is -800 mm·s^{-2} at this moment; from the velocity and acceleration curves, it can be seen that this mechanism maintains a low - speed and stable forging - punching process in the forward stroke stage (the stage of forging - punching the workpiece).

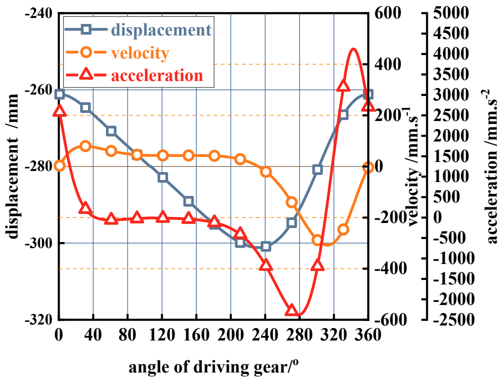

Fig.8 Transmission Characteristics of Slider

4.4 Comparative Analysis

To illustrate the advantages of the press structure proposed in this paper, a comparative analysis was carried out with the double - center crank - slider press. Fig.9 shows the displacement curves of the sliders of the two types of presses. From the displacement curves in the figure, it can be seen that: ① The ratio of the forward stroke time to the return stroke time of the double - center crank - slider press is 1:1, that is, this type of press has no quick - return characteristic; ② The ratio of the forward stroke time to the return stroke time of the press mechanism driven by the modified eccentric gear is greater than 1, that is, this type of press has the quick - return characteristic, which can improve the working efficiency of the press to a certain extent.

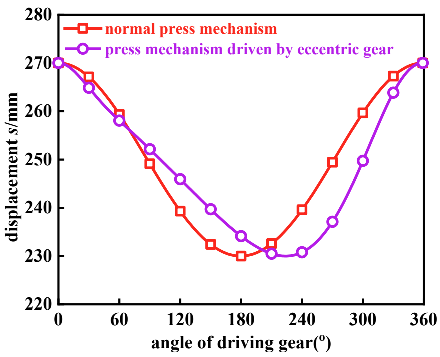

Fig.9 Comparison of Displacement Curves

5. Conclusion

The mathematical model of the press driven by the modified eccentric gear was established. The virtual prototype simulation test results are consistent with the theoretical calculation results, verifying the correctness of the established mathematical model.

By analyzing the motion characteristics of the press driven by the modified eccentric gear, it can be known that by adjusting the modification coefficients of the gears, the press can have the quick - return characteristic, which improves the working efficiency of the press to a certain extent.

By introducing the connecting frame rod with virtual constraints, the stability of the press is improved, and the product quality is improved to a certain extent.

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