Application of the double crank mechanism in designing complex-stroke press

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ABSTRACT

Although having a development history of over 200 years, researches and developments about press machines are still draw interest of modern mechanical engineering. This paper introduces a new press structure that utilizes a double crank mechanism to enhance productivity and efficiency for the stamping process. By adjusting the positions between the component links, the dynamic characteristic of the ram can be modified, optimizing the performance and improving operations of the system. The proposed design also allows for easy adjustments to adapt with different production requirements, such as changing forging and stamping speeds (fast and slow) to avoid the impact of springback properties in sheet metal. The kinematics of the structure have also been determined by mathematic equations, demonstrating significant improvements in the operation of the ram. Results from the research offers a promising solution to enhance production efficiency and enable the creation of simple yet capable mechanisms to perform complex operation.

Keywords: Press machine; Linkage; Theoretical design; Complex stroke movement; Kinematic analysis.

1. INTRODUCTION

Metal forming is a traditional process that continues to be widely used in various industries. Among the machines used in metal forming, presses are a classic type that enables the shaping of workpieces by applying a continuous pressure. In addition to considering the design of presses, it is essential to take into account the mechanical properties of the materials being processed, as they have a significant impact on mold manufacturing and the overall capacity of the press machines. One crucial aspect related to the properties of the material is its stress relaxation characteristic, which involves the gradual decrease in stress under constant strain, allowing the material to deform easily without developing wrinkles or even cracks during the forming process [1, 2]. Understanding this behavior is crucial for ensuring successful and defect-free metal forming operations. By considering the springback behavior, manufacturers can optimize their metal forming processes to achieve the desired shapes with greater precision and quality. For example, by combining the process of holding pressure and ultrasonic vibration to a bending machine, designers can change the stress distribution state of the bending region and even improve the roughness for the bending surface [3]. Another simulation analysis for the punching operation of U-shaped sheet metal also indicates that just by employing consecutive load-unload operations can reduce the required load and springback of the resulting profiles, hence improving the forming precision and formability of products [4].

With traditional mechanical presses, the speed and force of pressing are often fixed parameters that cannot be changed, sometimes leading to wrinkles or tears when stamping complex-shaped thin metal sheets (such as laptop casings, household appliances, or even car frames, etc.). If the stamping speed is reduced, the entire processing speed will be affected, resulting in reduced processing efficiency.

However, if only the speed of a local stage of the process is changed, a flexible processing system is required. This necessitates a complete change in the structure of the press to integrate sensors and electronic feedback components [5-8]. Because these systems utilize servo motors to program the motion of the ram according to time, constructing complex ram movement using this system is very flexible. However, since controlling the movement of the machines in these devices requires the existence of feedback circuits with precise position checking, the use of a constant-speed motor and flywheels to increase torque and reduce motor power for the system is impossible, increasing both the cost and equipment investment.

To reduce the use of a servo motor, various methods have also been suggested, focusing on designing mechanisms that allow for adjustable dimensions to alter the dynamic parameters of the ram [9] or employing multi-link structures to execute the complex motion of the ram [10, 11]. These innovations aim to enhance the efficiency and flexibility of the stamping process, making it more adaptable to different production requirements and improving the overall performance of the machines.

This paper proposes using a single-degree-of-freedom mechanical press with a displaceable link to control the complex movement of the ram. Kinematic analysis was conducted to inspect the operation of every link in the mechanism. The new system provides a viable option for constructing single-stroke presses for elastic products while taking advantage of material relaxation characteristics. Moreover, since the new system allows the use of a flywheel to perform complex ram movement using constant-speed motors, reducing the required power, manufacturing complexity, and product cost.

2. KINEMATIC ANALYSIS

The structure proposed press consists of 6 links, as presented in figure 1.

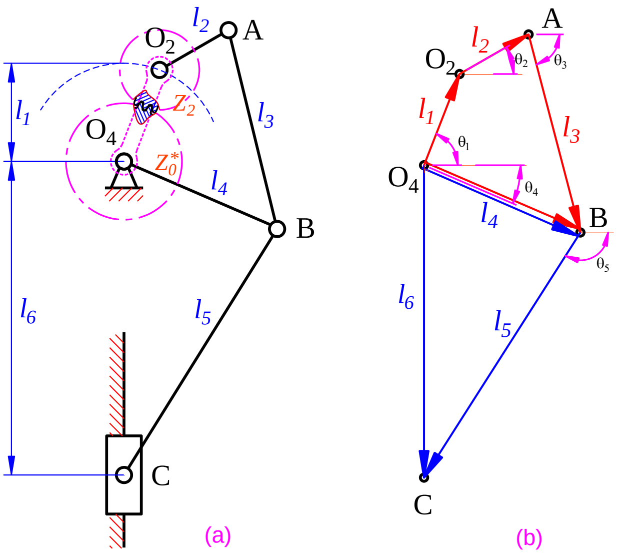

Figure 1. *Structure of the proposed system: (a) Theoretical model; (b) Closed loop shaped.*

According to figure 1a, point O_2 is allowed to be adjusted around point O_4 with a constant distance l_1 (the length of frame O_2O_4). The gear set Z_0*-Z_2 is used for transmits the rotation for driving link O_2A with a constant speed from the motor in O_4. When the machine works, framed O_2O_4 is fixed to the frame, makes points O_2 and O_4 becomes grounded rotating joints. As these links are directly connected to Links 3 and 4, the position of joint B can be determined from the four-bar linkage O_2O_4AB, and the movement of the ram head C can be calculated from the crank-slider linkage O_4BC.

In this system, the dimensions of the component links are chosen so dimension O_2O_4 of the link 1 is the shortest link to ensure the four-bar linkage O_2O_4AB becomes a double crank mechanism. Since link 4 can rotate with a full circle, the stroke of slider C (the press's ram) is also constrained. However, since the crank link 4 receives rotation from link 2, the rotation velocity of this link is not constant but varies. This results in the ram moving with a quick-return operation, even though the eccentric distance between the ground sliding frame and point O_4 is zero. To determine position of the component frames, vector loops are assigned to the machines, which can be divided into two parts separated by colors, as presented in figure 1b. By solving the position equation for each loop, the movement of the ram can be easily determined as follow:

For the first loop: $\vec{l_1} + \vec{l_2} + \vec{l_3} = \vec{l_4}$

$$\begin{cases} l_1 \cos\theta_1 + l_2 \cos\theta_2 - l_4 \cos\theta_4 = -l_3 \cos\theta_3 \\ l_1 \sin\theta_1 + l_2 \sin\theta_2 - l_4 \sin\theta_4 = -l_3 \sin\theta_3 \end{cases} \tag{1}$$

Based on this system, the position of links 4 can be determined as:

$$\theta_4 = \arcsin\left[\frac{l_1^2 + l_2^2 + l_4^2 - l_3^2 + 2l_1l_2 \cdot \cos(\theta_1 - \theta_2)}{2l_4 \cdot \sqrt{l_1^2 + l_2^2 + 2l_1l_2\cos(\theta_1 - \theta_2)}}\right] - \arcsin\left[\frac{l_1\cos\theta_1 + l_2\cos\theta_2}{\sqrt{l_1^2 + l_2^2 + 2l_1l_2\cos(\theta_1 - \theta_2)}}\right] \tag{2}$$

For the second loop: $\vec{l_4} + \vec{l_5} = \vec{l_6}$

$$\begin{cases} l_4 \cos\theta_4 + l_5 \cos\theta_5 = l_6 \cos\theta_6 \\ l_4 \sin\theta_4 + l_5 \sin\theta_5 = l_6 \sin\theta_6 \end{cases} \tag{3}$$

Since $\theta_6 = -90°$, the position of the ram can be drawn from the system as:

$$l_6 = l_4 \sin\theta_4 + \sqrt{l_5^2 - l_4^2 \cos^2\theta_4} \tag{4}$$

The velocities of the system can also be acquired by deriving the closed loops equations of the position. Since the positions of every link were obtained from the equations above, the velocities of the component links can be determined as follow:

Transform the first loop into Euler's formula and derivate it to obtain the velocity:

$$l_2 j\omega_2 e^{j\theta_2} + l_3 j\omega_3 e^{j\theta_3} = l_4 j\omega_4 e^{j\theta_4} \tag{5}$$

Draw the rotation velocity of link 4 from the equation:

$$\omega_4 = \frac{l_2\omega_2 \sin(\theta_2 - \theta_3)}{l_4 \sin(\theta_4 - \theta_3)} \tag{6}$$

Transform the second loop into Euler's formula and derivate this equation to obtain the velocity:

$$l_4 j\omega_4 e^{j\theta_4} + l_5 j\omega_5 e^{j\theta_5} = \dot{l}_6 e^{j\theta_6} \tag{7}$$

Draw the ram velocity from the equation:

$$v_6 = \dot{l}_6 = -\frac{l_4\omega_4 \sin(\theta_4 - \theta_5)}{\sin\theta_5} \tag{8}$$

3. RESULT AND DISCUSSION

In the proposed configuration, the gear sets Z_0-Z_2 are used for manual positioning of point O_2 and for transmitting rotation to driving link 2 from fixed point O_4. Based on the system's movement concept, a model of the system with lengths is proposed, respectively as: $l_1 = 50$, $l_2 = 87.5$, $l_3 = 137.5$, $l_4 = 150$ and $l_5 = 300$ mm. To analyze the influence of angular θ_1 on the movement of the ram, different values of θ_1 were assigned to the system for evaluation. Figure 2 present the operation of ram according to different value of point O_2, using equation (4).

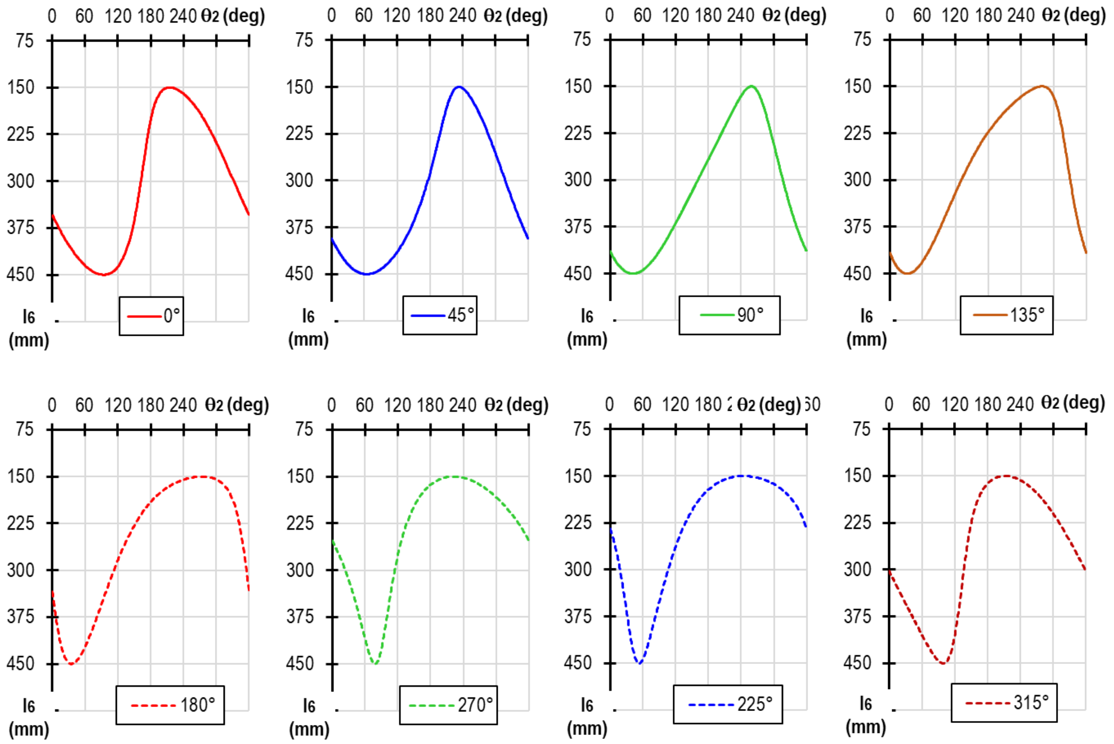

Figure 2. Movement of the ram according to different position of link 1.

The significant changes in the motion trajectory chart of the ram in figure 2 present different types of press operation of the system. During the stamping motion of the ram (the downward stroke with l_6 increasing from 150 to 450 mm), when the guiding angle $\theta_1 = 0°$, this motion occupies 1/3 of the driving phase (from 94° to 214°), corresponding to the quick stamping motion. On the other hand, when the guiding angle $\theta_1 = 270°$, this motion occupies 3/5 of the driving phase (from 222° to 438°), corresponding to the slow stamping characteristic of the machine.

For a detailed analysis, the productivity of the rams' movement from figure 2 is extracted, as presented in figure 3. This parameter describes the rate at which the ram moves during the "return" stroke compared to the "downward" stroke. The higher coefficient values indicate operations where the return strokes are faster than the forward strokes, while lower coefficient values indicate operations where the return stroke is slower.

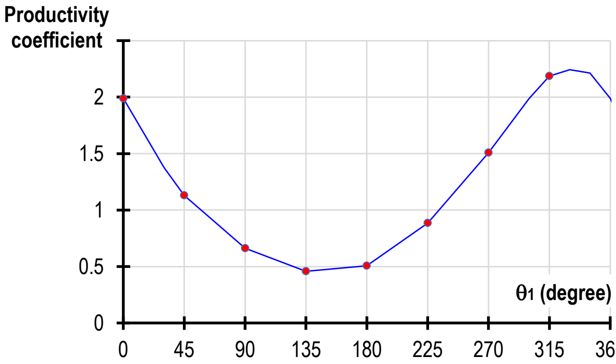

Figure 3. Productivity of the system according to position of driving link 1.

For the rotation of the drive link at 10 rad/s (approximately 95 rpm), the corresponding velocity of the ram can be calculated using Equation (8), and summarized in figure 4.

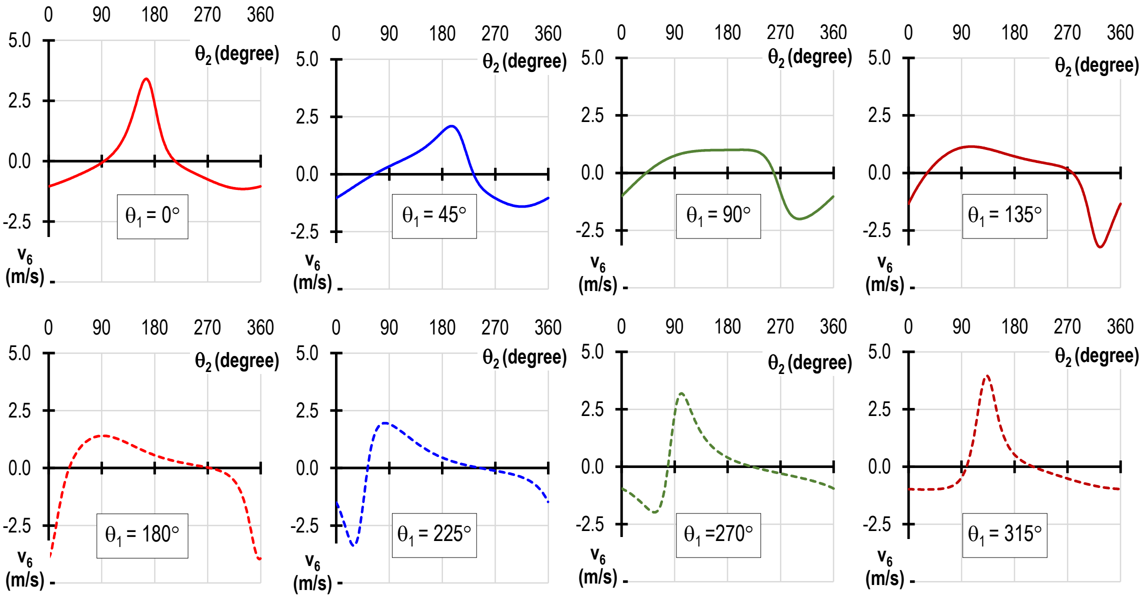

Figure 4. *Velocities of the ram according to different position of link 1.*

In the diagrams, positive velocities represent downward movements of the ram, and negative velocities represent the return movements. It can be seen that the maximum velocity of these diagrams occurs in the case where $\theta_1 = 180°$ ($|v_{max}| = 3.96$ m/s), and the minimum is at the case where $0_1 = 90°$ (corresponding to $|v_{max}| = 2.3$ m/s). It is noticed that the velocity diagrams are calculated for the entire cycle of the system and do not specifically represent the position where the press contacts the anvil. Therefore, they cannot provide detailed information about the press operation during specific stages or interactions with the workpiece. For further understanding the behavior of the press, velocities at specific positions, such as 50 millimeters before and after the lowest position of the ram were analysis, as presented in figure 5.

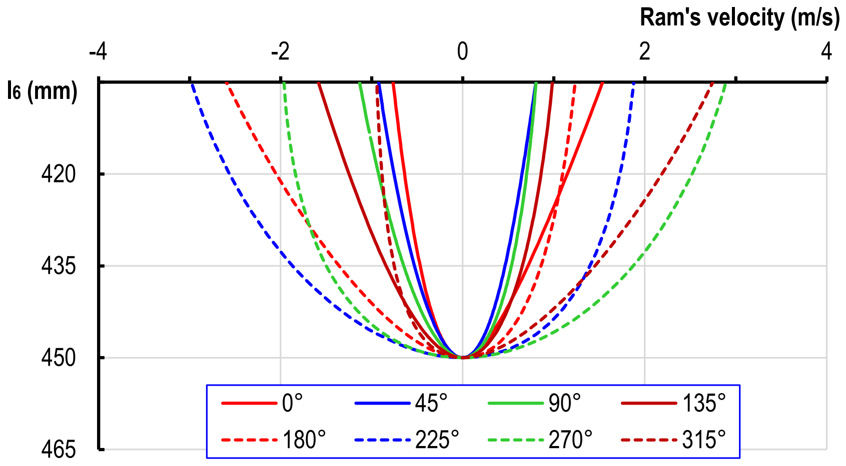

Figure 5. *Ram's speed at position before going to the lowest positions (450 mms).*

The results in the figure show that with different positions of point O_2 (corresponding to different angular θ_1), the velocity of the ram head is not uniform, varying within the range of 0 to 3 m/s. The smaller the angle θ_1, the smaller the velocity of the ram, and vice versa. If angle θ_1 is larger than 180°, the difference in velocities between the downward and upward strokes also increases, providing greater kinetic energy and higher load for the stamping process.

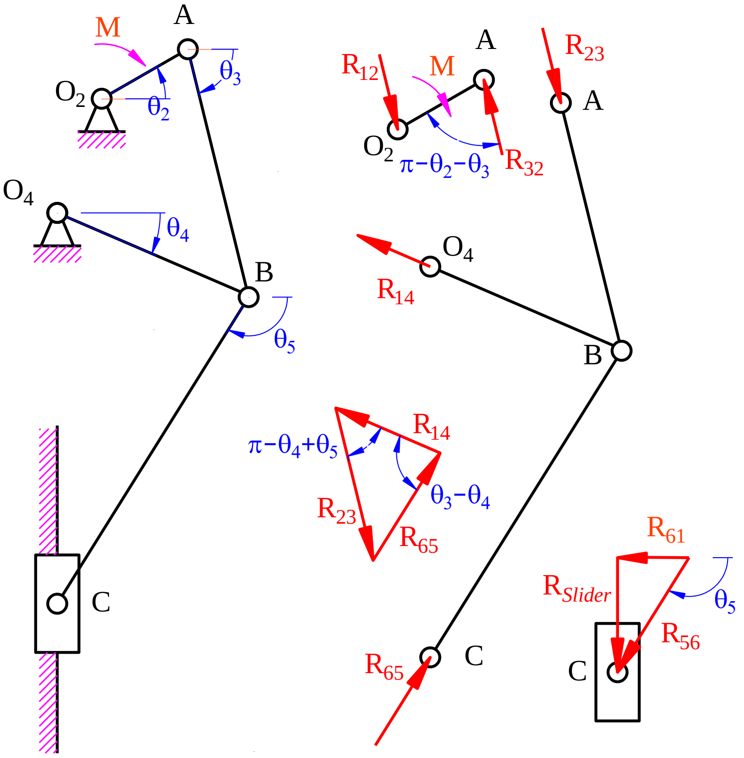

Figure 6. *Analysis the static force in the mechanism.*

To evaluate the influence of the proposed structure on the stamping force, a simple force model is introduced for investigation, as presented in figure 6. Assuming that link 2 is driven by a motor with a constant torque M, and the model is evaluated in a static state to neglect the effect of inertia and kinetic energy of links. In this case, the force generated on the slider is determined by an equation that depends on the position of the component links, as described in equation 9:

$$R_{Slider}=\frac{R_{56}}{\sin\theta_5}=\frac{R_{23}\cdot\sin\left(\pi-\theta_4+\theta_5\right)}{\sin\left(\theta_5-\theta_6\right)\cdot\sin\theta_5}=\frac{M\cdot\sin\left(\pi-\theta_4+\theta_5\right)}{l_2\cdot\sin\left(\pi-\theta_2-\theta_3\right)\cdot\sin\left(\theta_5-\theta_6\right)\cdot\sin\theta_5} \qquad (9)$$

Results of the calculation process allow to determine the stamping force from the slider for every position of the mechanism. Figure 8 presents the force diagram R_{Slider} on the slider corresponding to a torque of M = 10 N.m. Although the motions of slider in these cases are quite similar (the force generated increases when the slider moves towards the lowest positions), different models will produce different slider force diagrams. Specifically, when $\theta_1 = 0°$ or 45°, the forces generated within the stamping range (from 300 mm downward) are highest, and when $\theta_1 = 180°$ or 225°, the generated force is smallest compared to the other positions (see figure 7).

Comparing to simple presses that receive direct rotation from motors, it is observed that even with an isochronous motor, the proposed system can generate complex movements similar to those from servo stamping presses. Furthermore, since the proposed system does not require servo motors to create complex operations, flywheels can be employed to increase energy storage capacity for heavy-duty tasks. Since the component links are designed with fixed dimensions, the proposed structure more stable and rigid, resulting in reduced construction costs and machine size. Finally, the ability of link 4 to rotate with a full rotation allows for a constant stroke for every adjustment of the system, enabling not only stamping different types of products but also expanding punching, stamping, and even forging processes. This improvement enhances efficiency and enables the shaping of various objects without replacing dies.

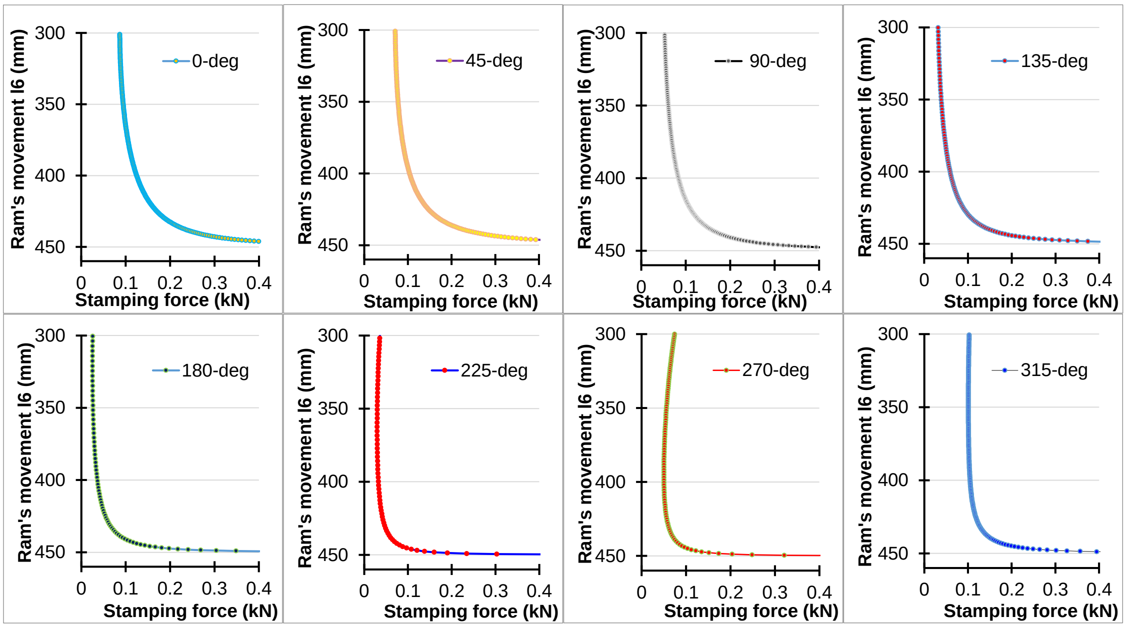

Figure 7. *Displacement-force diagram of the slider with different values of* θ_1.

4. CONCLUSIONS

This research proposes a new structure for press machines that can execute complex ram movements. By adjusting position of component links, complex movements of the rams are obtained, thus demonstrating the applicability of the proposed system in the deep drawing process. The main conclusions and achievements of this study are as follows:

By changing the position of driving link, the new mechanism can execute complex movements, which are beneficial for metal forming while utilizing the relaxation of material, avoid the crack or wrinkles in the deep drawing processes.

By solving simple kinematic equations for the lengths of component frames, the kinematic properties (position and velocity) of the ram can be synthesized and summarized. From the obtained results, the required combination of the structure can be determined according to the press specifications and forming requirements.

Since the proposed system only uses the isochronous motors, its construction is much simpler than systems that require servo motors to achieve complex stamping processes. Furthermore, eliminating the need for a feedback circuit to control its complex movement, the new system can incorporate flywheels to store energy and decrease the required motor power, leading to enhanced manufacturing efficiency and reduced product costs.

The flexible adjustment of the output speed and movement of the system contributes to automating the processing without direct intervention of labor, enhancing processing efficiency, and ensuring safety for operators. This model can be applied in various technological stamping processes in Vietnam, such as stamping car frames with long strokes applying force, deep drawing of complex structures with large strokes but variable speeds (sink basins, kitchenware, and even large-size bullet jackets).

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TÓM TẮT

Ứng dụng cơ cấu hai tay quay trong thiết kế máy dập có hành trình phức tạp

Mặc dù có lịch sử phát triển hơn 200 năm, hiện nay các nghiên cứu và phát triển về máy dập vẫn thu hút sự quan tâm đặc biệt trong lĩnh vực cơ khí. Bài báo này giới thiệu một cấu trúc máy dập mới sử dụng cơ cấu bốn khâu bản lề hai tay quay để nâng cao năng suất và hiệu quả cho quy trình dập. Bằng cách điều chỉnh vị trí giữa các khâu thành phần, các thông số động học của chày có thể được điều chỉnh giúp tối ưu hóa hiệu suất và cải thiện hiệu quả làm việc của hệ thống. Thiết kế máy cũng cho phép dễ dàng điều chỉnh cho phép phù hợp với các điều kiện sản xuất khác nhau, chẳng hạn như thay đổi tốc độ rèn, dập (nhanh, chậm) để giảm bớt ảnh hưởng đặc tính đàn hồi trong các tấm kim loại. Động học của cấu trúc cũng đã được xác định bằng các phương trình toán, giúp xác định chính xác quỹ đạo chuyển động của chày. Nghiên cứu này cũng hứa hẹn một giải pháp thiết kế cho việc nâng cao hiệu quả sản xuất và giúp thiết kế các cơ cấu máy có kết cấu đơn giản nhưng có thể thực hiện các chuyển động phức tạp.

Từ khóa: Máy dập; Cơ cấu máy; Thiết kế nguyên lý; Chuyển động có chu kỳ phức tạp; Phân tích động học.