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ORIGINAL ARTICLE

Small mechanical press with double-axis servo system for forming of small metal products

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Abstract Multi-axis servo systems have been applied to many machine tools in recent decades, but they are not commonly used for press machines. They are particularly uncommon for small presses because it is still believed that the precision of a small press is dependent on the precision of its components and parts. However, high-precision components and parts are difficult to manufacture and assemble, so the costs of small precision presses are roughly equal to or higher than those of large press machines. Moreover, the accuracy of conventional small presses worsens over long operation times, particularly when dealing with off-center loading frequency. Therefore, we propose a small mechanical press with an adjustable driving system, using a slide and double-axis servo system, for manufacturing small products. Through this proposed design, we expect higher-quality products to be produced for lesser cost than conventional small presses.

Keywords Metal forming · Servo press · B-spline curve

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1 Introduction

At present, mechanical presses are widely used for punching, forging, drawing, and other plastic-forming processes. In the last decade, with rapid developments in computer technology, cell phones, and other electronic products, light alloys are being increasingly applied to many complex parts to reduce their weight and enhance their strength. Because of the consideration of cost, manufacturers tend to use press machines for the mass production of these parts with increasingly complex shapes, greater precision, and fewer processes.

To achieve these aspirations, users of press machines often require variable driving curves for the slide so as to deal with different forming applications. Manufacturers of press machines, mostly in Japan [1–8] and Germany [9–11], have developed several models of mechanical presses with servo drive technology. The servo mechanical press offers the flexibility and accuracy of position control to raise the quality and reliability of mass production [12].

For large mechanical presses (usually over 100 tons), precision is commonly subpar because of tolerance, clearance, and off-center load (Fig. 1). Therefore, the multiple-axis servo mechanical press was developed to adjust the parallelism of the slide and the precision of bottom dead center [13–19]. However, because of the limitations of large mechanisms and servo motor technologies, it remains difficult to punch small parts with high accuracy and high speed, even with the multiple-axis servo system.

Generally, small products are formed by smaller presses, and their qualities depend on the accuracy of the mold and press, which needs to be assembled with high-precision components and parts (tolerance < 10 μm). However, high-precision components and parts are difficult to manufacture and assemble, and are therefore usually very expensive. Moreover, when an off-center load occurs, such as with a progressive die (Fig. 1b) or a multi-point die [20, 21], the wear between slide and guide will deteriorate the accuracy

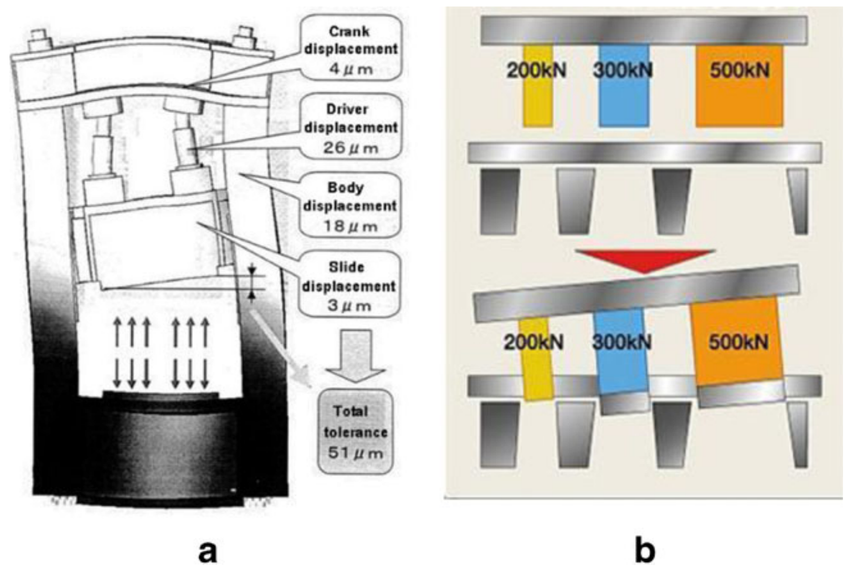

Fig. 1 Causes of unbalance of slide: **a** tolerance and clearance of parts and assemblies, **b** off-center load of progressive dies [17]

of the press, and the press operators must distribute the heavy maintenance costs. As a solution, we propose a small mechanical press with an adjustable driving system using a slide and double-axis servo system (Fig. 2). These characteristics allow the proposed press machine to form small products that require high-precision and a stable quality. Moreover, because of the use of components and parts of average precision only (tolerance≤20 μm), the manufacturing costs of this proposed press machine may be lower than those of conventional precision press machines.

2 Design of proposed press

2.1 Mechanism

The dimensions of the small mechanical press proposed in this paper are 460 mm (width), 300 mm (depth), and 790 mm (height). It weighs approximately 350 kg. Thus, it is a desktop-type machine. Other specifications are listed in Table 1. The proposed press is shown in Fig. 3.

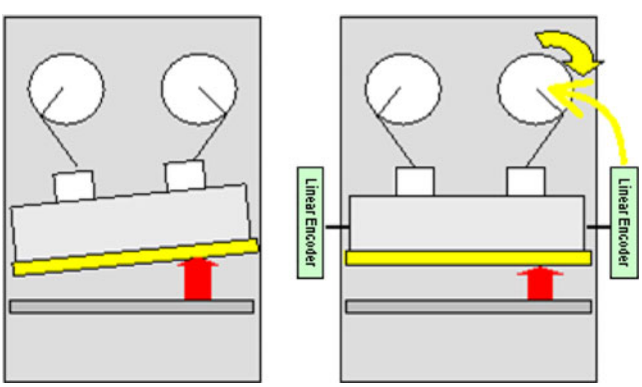

Fig. 2 Method of adjusting the slide with multi-axis servo system [6]

The design of the slide driving system is shown in Fig. 4. There are two AC servo motors, two shafts, four cranks, four ball joints, four linear guides, and one slide. Both servo motors are DELTA ASDA-A2 400 W, which have rated torque of 1.27 N·m, maximum torque of 3.82 N·m, moment of inertia of 0.278×10^{-4} kg·m^2, and rated speed of 3,000 rpm [22]. The shaft of the servo motor is directly assembled to a gear reducer (1:10). Each shaft is connected to a servo motor by a timing belt and two timing pulleys (ratio of teeth=1:2.4). Moreover, each shaft has two cranks on it. The maximum allowable tolerance of all parts and components is 20 μm. Therefore, all of these parts and components are easier to manufacture and are relatively inexpensive. To improve the tolerance of the mechanism, we designed four ball joints with adjustable screws (see Fig. 5) to connect the four cranks and slide. The four linear guides have adjustable screws to adjust the clearance between the guides and the machine body. Through this design, manufacturers can accurately adjust the parallelism of the slide relative to the bolster (see Fig. 6) before the press machines leave the factory. Furthermore, with the double-axis servo system, a controller can automatically correct the

Table 1 Specifications of experimental press

Item	Specification	Unit
Dimension	460 (W)×300 (D)×790 (H)	mm
Weight	350	kg
Max. capacity	5	kN
Stroke	18	mm
Max. speed	125	SPM
Tonnage rating point	1.0	mm
Die height	150	mm
Area of slide	230×130	mm

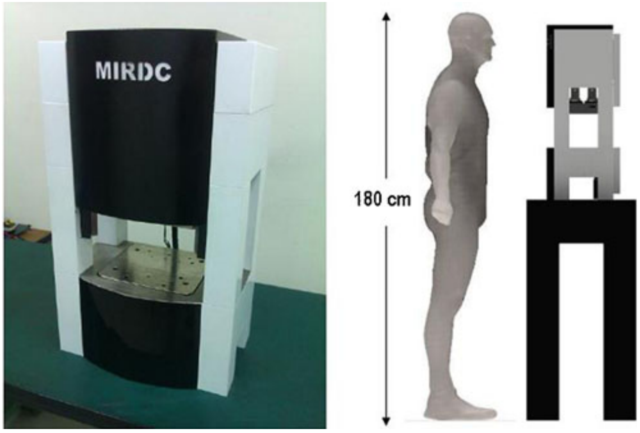

Fig. 3 Proposed press

balance of the slide when an off-center load occurs (see Fig. 2).

2.2 Control system

Figure 7 shows the control system layout for the proposed press. The controller is a PC-based system with an embedded Microsoft Windows XP operating system. In addition, we used Ardence RTX software (see Fig. 8), which provides the ability to control motors and I/O quickly and accurately under the Microsoft Windows system. The RTX system's interval time was set to 1 ms. Furthermore, we used V command to control the drivers of the servo motors so that they avoid speed and torque drops that could result from sudden impacts and servo-tuning of the position control. To synchronize the two running motors, we applied the proportional–integral–derivative (PID) control [23, 24], through which we could adjust the speeds of the two motors in real time. The feedback error of PID control is the difference between the encoder values of the two motors. When the position of one motor (which is measured by an encoder) is behind another motor, the rotary velocity of the slower motor will accelerate, and the

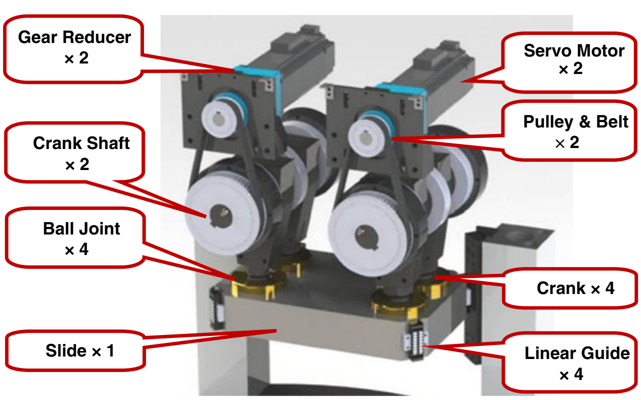

Fig. 4 Slide driving system

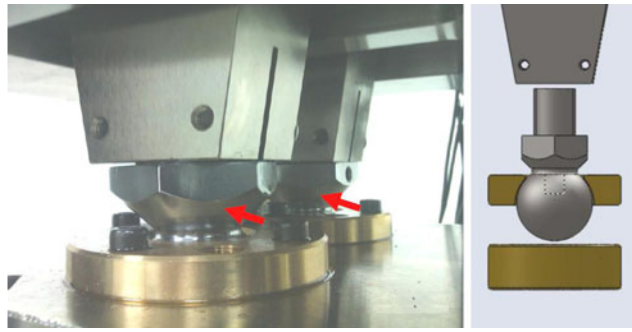

Fig. 5 Ball joints with screw to adjust the parallelism of slide

faster motor will decelerate oppositely. Hence, both sides of the slide can be maintained at a horizontal level when the press is running. A speed control chart of the motor motion is shown in Fig. 9. The formulas for the speed control of the two motors are described in (1) and (2).

$$V_1 = V_{\mathrm{commamd}} + \mathrm{Kp}_1 \times \mathrm{pError} + \mathrm{Ki}_1 \times \mathrm{iError} + \mathrm{Kd}_1 \times \mathrm{dError} \quad (1)$$

$$V_2 = V_{\mathrm{command}} + \mathrm{Kp}_2 \times \mathrm{pError} + \mathrm{Ki}_2 \times \mathrm{iError} + \mathrm{Kd}_2 \times \mathrm{dError} \quad (2)$$

where V_1 is the new speed of motor 1, V_2 is the new speed of motor 2, V_{command} is the speed of command for the two motors, Kp is the gain value of proportional control, Ki is the gain value of integral control, and *Kd* is the gain value of derivative control; pError, iError, and dError are the proportional, integral, and derivative errors, respectively. The

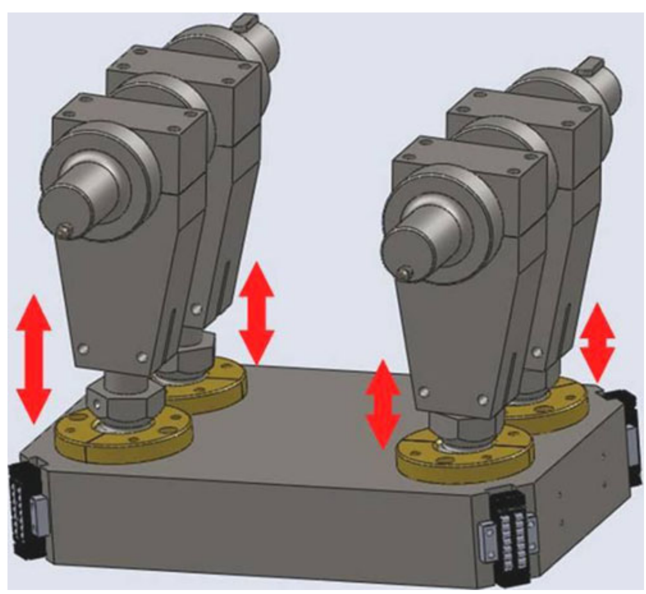

Fig. 6 Adjustable driving system of slide

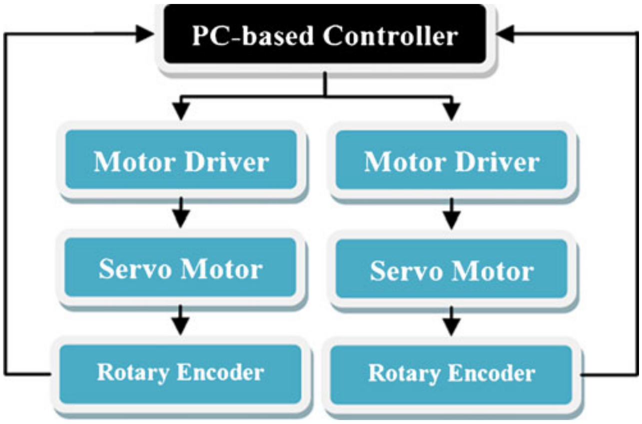

Fig. 7 Organization of control system

formulas for calculating pError, iError, and dError are given in (3)–(5):

$$\text{pError} = \text{Encoder1} - \text{Encoder2} \tag{3}$$

$$\text{iError} = \text{iError} + \text{pError} \tag{4}$$

$$\text{dError} = \text{pError} - \text{Last\_pError} \tag{5}$$

where Encoder1 is the encoder value of motor 1, Encoder2 is the encoder value of motor 2, and Last_pError is the pError of the last cycle time.

3 Free-motion

The biggest advantage of the servo press over a conventional mechanical press is the freedom to define the driving curve of the slide. Figure 10 shows driving curves designed to represent the different requirements in the manufacturing process. To allow users to easily define the driving curve of the slide by themselves, we designed a user-friendly human machine interface (HMI) (see Fig. 11). Users can freely adjust the given control points to define the tendency of the driving curve.

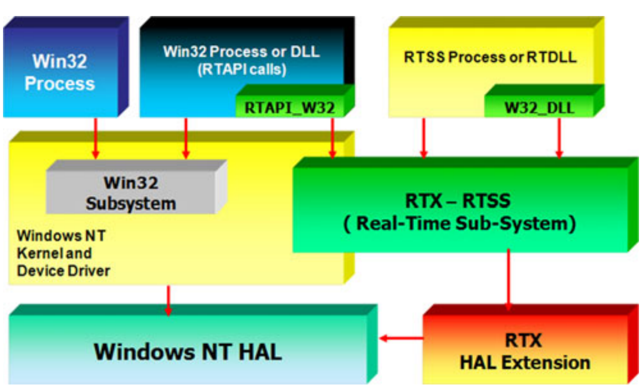

Fig. 8 Organization of Microsoft Windows with Ardence RTX system

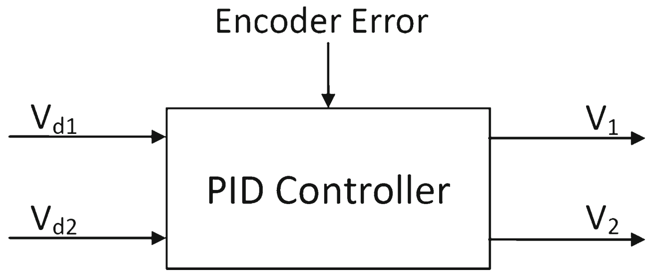

Fig. 9 Speed control chart of proposed press machine

Therefore, to make the slide move according to user expectations by applying the V command of speed control, the controller of the press machine must determine the speed of the servo motor for each instance within one stroke based on the information of the user-set control points.

3.1 Curve fitting

Because users can define and provide data on only a few control points for the driving curve of the slide (the scrollable bars in Fig. 11), we use the theory of B-spline curves to enable the controller to obtain a formula for the user-set driving curve. B-spline is a spline function proposed by Cox and de Boor in 1978 [25–27]. They suggested a new basis function, $N_{i,k}(u)$, to describe the formula of a curve. These curve types, which are composed of $N_{i,k}(u)$, are called B-spline curves. The recursive equations of the B-spline curve are shown in (6)–(9):

$$B(u) = \sum_{j=0}^{n} B_j N_{i,k}(u) \quad t_{k-1} \leq u \leq t_{n+1} \tag{6}$$

$$N_{i,k}(u) = \frac{(u - t_i)N_{i,k-1}(u)}{t_{i+k-1} - t_i} + \frac{(t_{i+k} - u)N_{i+1,k-1}(u)}{t_{i+k} - t_{i+1}} \tag{7}$$

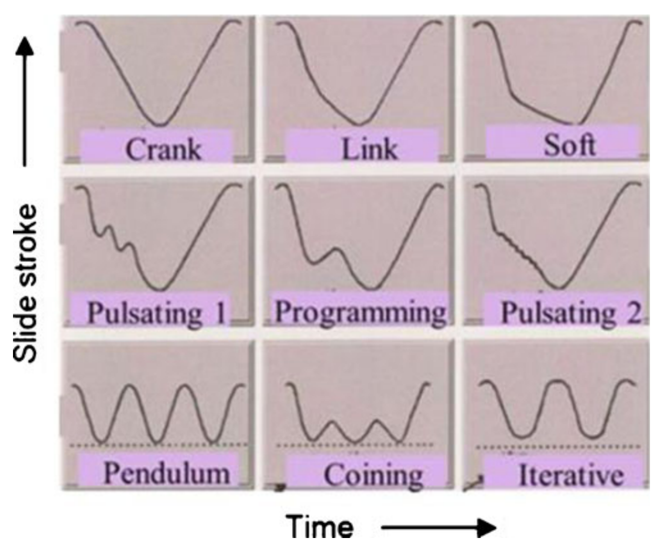

Fig. 10 Typical slide motions of servo press [3]

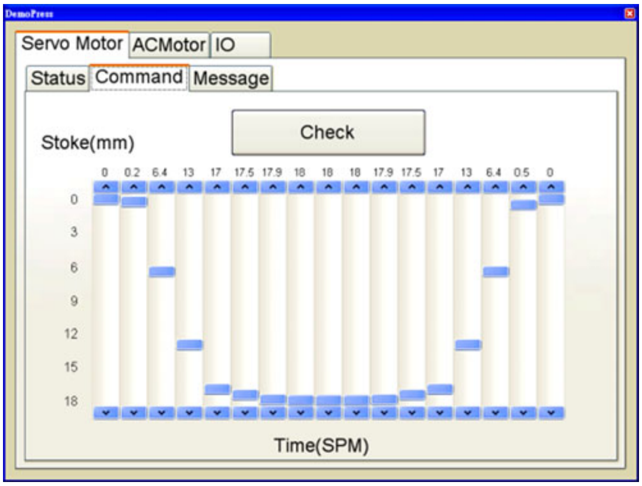

Fig. 11 Operation interface of free-motion function

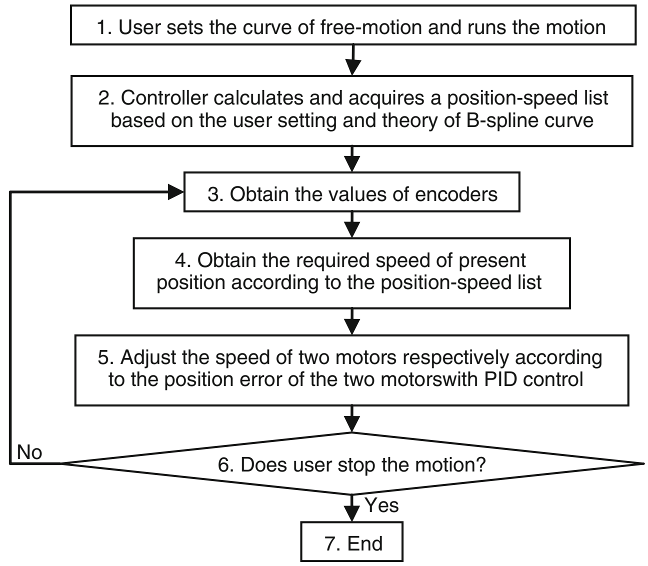

Fig. 13 Control workflow of proposed press

$$N_{i,1}(u) = \begin{cases} 1 & t_i \leq u \leq t_{i+1} \\ 0 & \text{otherwise} \end{cases} \quad (8)$$

where $n+1$ is the sum of control points; B_j are the polygon vectors of the control points; k is the order of the curve; u is the knot; and t_i are the knot values. The knot values can be decided through

$$t_i = \begin{cases} 0 & 0 \leq i < k \\ i-k+1 & k \leq i \leq n \\ n-k+2 & n < i \leq n+k \end{cases} \quad (9)$$

Then, we can plug the data on the control points that are determined by the user into B_j in (6), and we can obtain the corresponding points $B(u)$ of the driving curve by setting different values for u.

3.2 Speed control

By using the theory of B-spline, we can obtain information on any point of the driving curve. Furthermore, if a user has determined the traveling time of one stroke, we can calculate the speed according to displacement and time data. While the number of strokes per minute (SPM) (which is set by users) is N, the traveling time T (millisecond) of one stroke can be determined by

$$T = \frac{60,000}{N} \quad (10)$$

If the curve obtained using the theory of B-spline is divided into n segments (i.e., taking $n+1$ nodes), each time interval t (millisecond) can be determined by

$$t = \frac{T}{n} \quad (11)$$

Therefore, the speed V of the servo motor at each segment of the B-spline curve can be determined by (6) and (12).

$$V = \frac{\Delta B(u)}{t} \quad (12)$$

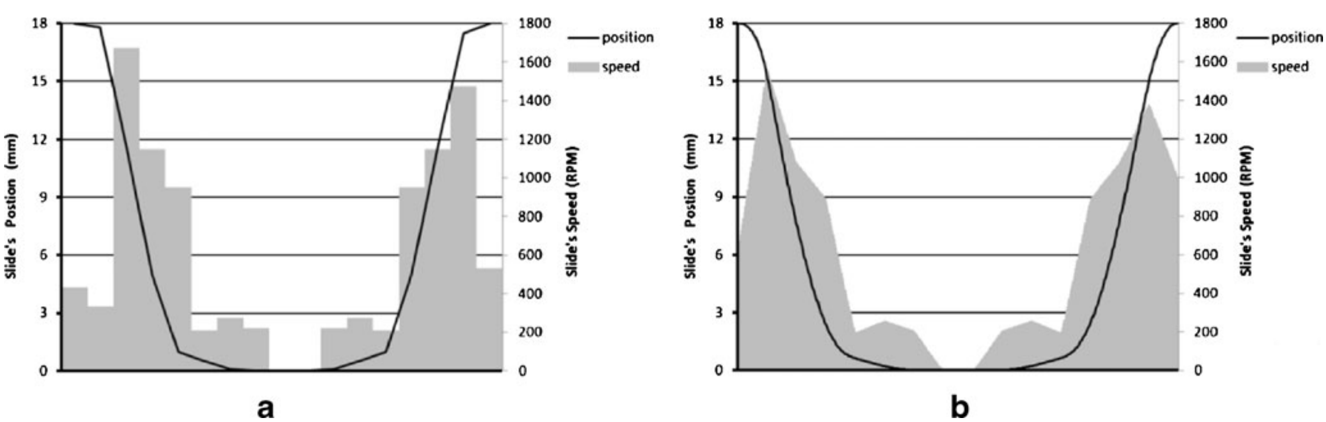

Fig. 12 Comparison of **a** user settings and **b** spline curve

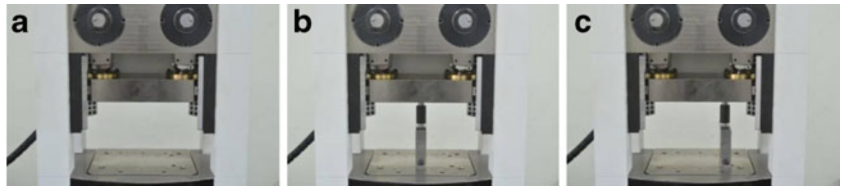

Fig. 14 Three cases of motor synchronization test: **a** slide only, **b** center load, **c** off-center load

When the value of n is large, the number of segments of the B-spline curve will also be large. Thus, the driving path of the slide obtained by means of the B-spline curve will be smooth, and so the curve of velocity of the slide driving calculated with (12) will also be smooth. In the case of Fig. 11, if the controller can transform the process path of the user settings (Fig. 12a) into a spline curve (Fig. 12b) with the theory of B-spline, this should enable the motors to run smoothly without any sudden rises or sudden drops in speed.

3.3 Workflow of control process

Figure 13 shows the control workflow of the proposed press. After the user sets the curve of free-motion by adjusting the scrollable bars in the free-motion window (Fig. 11), the controller divides the curve into hundreds of segments based on the theory of B-spline curves. Then, it also calculates the required speed of each segment by means of (12), so the controller acquires a position-speed list that records the speed at which the motor should run at every segment of the driving curve. Next, the controller obtains the values of the encoders every 1 ms to determine the segment that they are at; in this way, the controller obtains the required speed from the present position according to the position-speed list. Then, the controller adjusts and subsequently gives the speed command to the motors according to the position error of the two motors with PID control. Finally, if the user stops the motion, the controller stops the motion of the motors. Otherwise, the controller runs the process from steps 3 to 6 in Fig. 13 on a continual loop.

4 Results

In this section, we describe the results of a motor synchronization test, a free-motion test, and a processing test to verify the performance of the proposed press with a double-axis servo system.

4.1 Motor synchronization test

First, we verified the performance of the synchronization of motors. We set the speed of 25 SPM for the press machine, and ran three cases with different load statuses (as shown in Fig. 14): slide driving only, center load, and off-center load. We used a gas spring (Fig. 15) to simulate a load when the press machine was operating with a die. The gas spring had a stroke of 10 mm, and the required compressive force ranged from 3.6 to 6 kN based on the compressive length. We set the slide to compress the gas spring with a length of 1 mm. According to the information shown in Fig. 15b, the press machine needed to generate a compressive force of 3.6–4 kN, which is equal to 70–80 % of the rated load. We produced the charts shown in Figs. 16 and 17 according to the test results from the three situations of Fig. 14, which were measured by means of the rotary encoders and laser sensors individually. We compiled and analyzed the data as listed in Table 2.

The results shown in Figs. 16, 17, and Table 2 indicate that in the cases of slide driving only or center load (Fig. 14a, b), the angular error of the crank shafts and the position error of the slide were controlled within a small range. The maximum errors were approximately 0.00025°

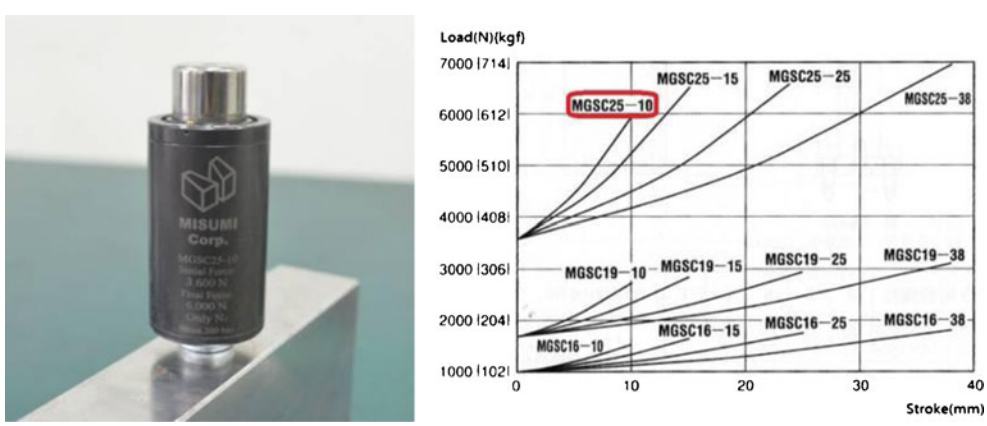

Fig. 15 Gas spring in the motor synchronization test

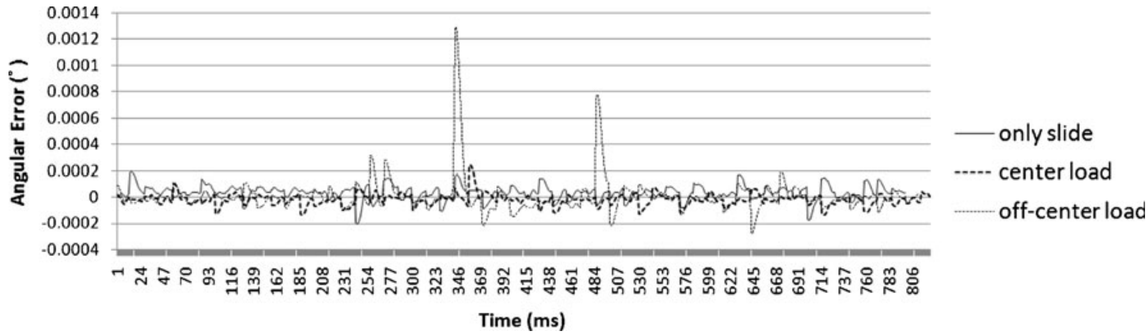

Fig. 16 Angular errors of crank shafts in one stroke

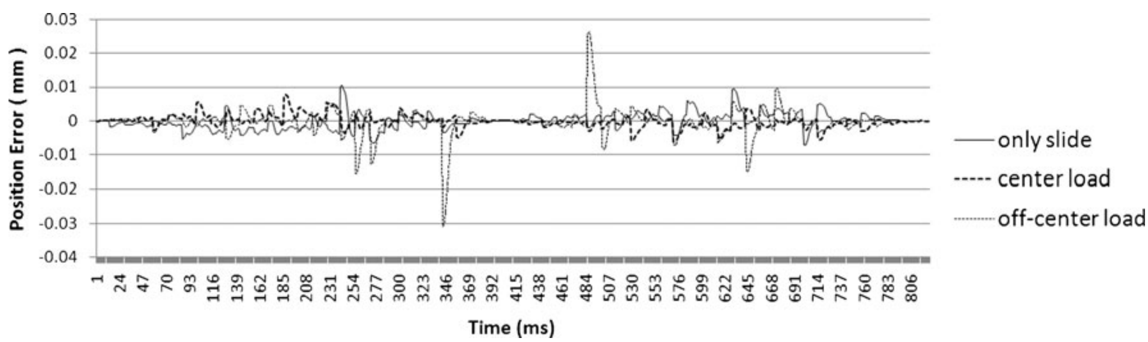

Fig. 17 Position errors of two sides of the slide in one stroke

Table 2 Results of motor synchronization test

	Slide only		Center load		Off-center load	
	Angular (°)	Position (mm)	Angular (°)	Position (mm)	Angular (°)	Position (mm)
Min. error	0.00000021	0.00000232	0.00000021	0.00000015	0.00000000	0.00000000
Max. error	0.00020297	0.01032154	0.00024257	0.00773124	0.00129105	0.03082938
Average error	0.00004999	0.00178383	0.00003507	0.00128272	0.00006343	0.00215117

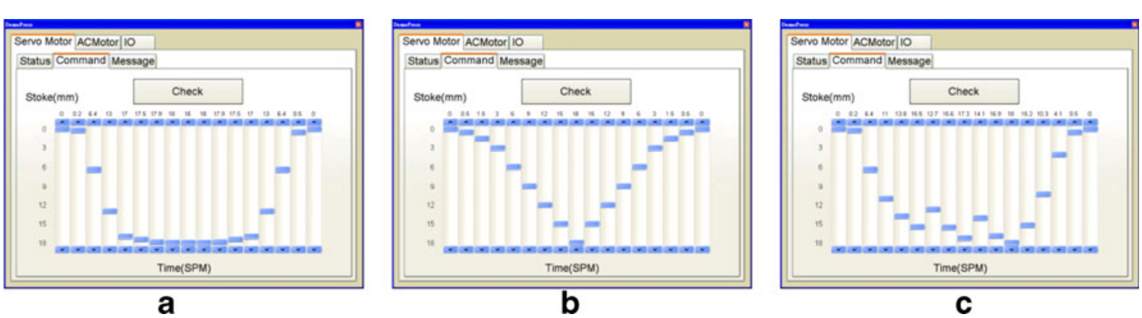

Fig. 18 Three processing curves on HMI **a** coining, **b** cutting, **c** pulsating

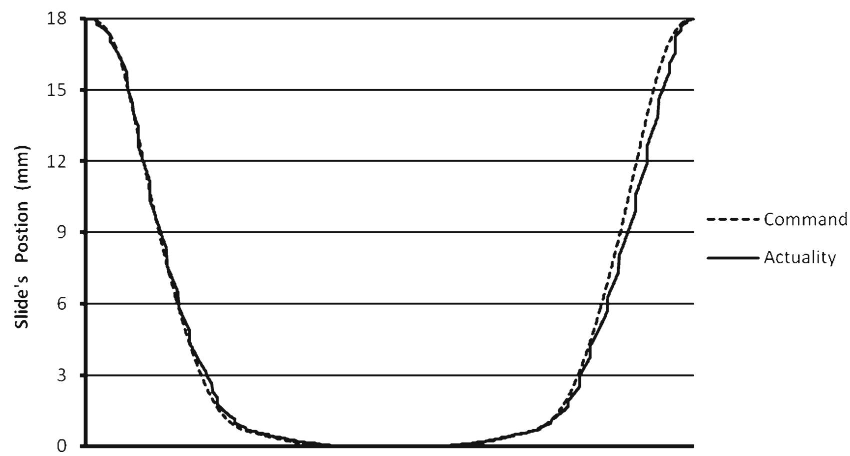

Fig. 19 Results of coining process

Table 3 Results of free-motion test

	Coining	Cutting	Pulsating
Min. error (mm)	0.0000000051	0.0000217195	0.0000262731
Max. error (mm)	1.9748875323	0.5026560260	1.1344073228
Average error (mm)	0.2637459612	0.1145871043	0.3067714133

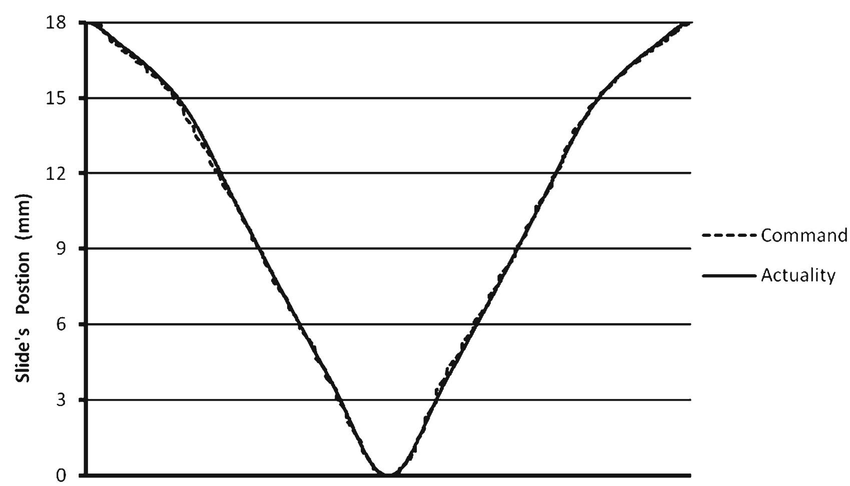

Fig. 20 Results of cutting process

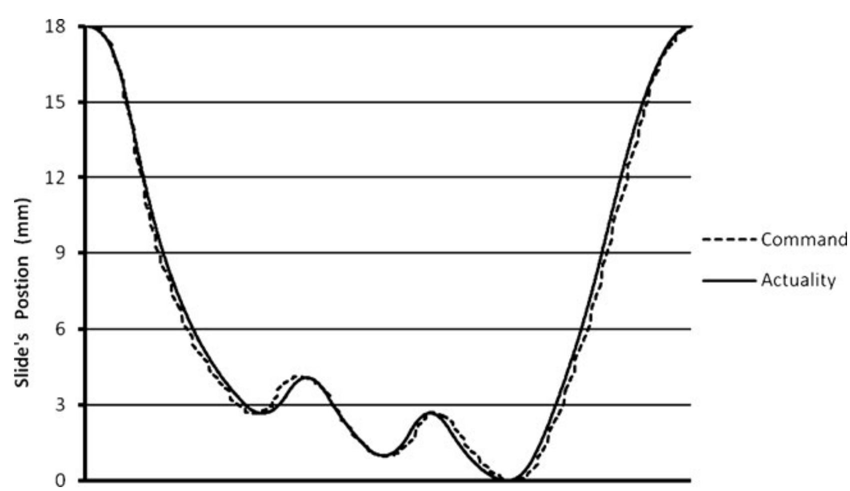

Fig. 21 Results of pulsating process

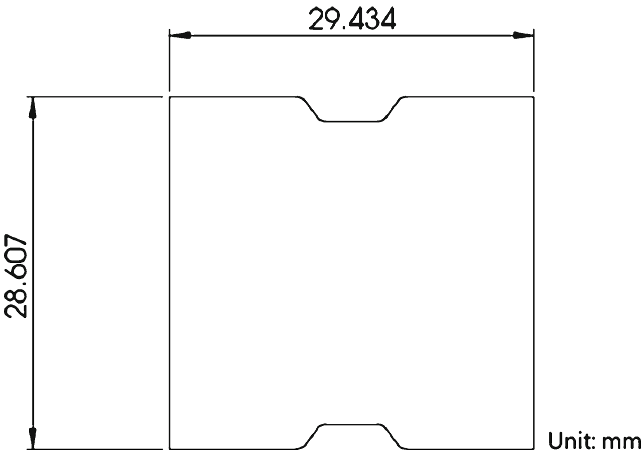

Fig. 22 Dimensions of the punch and expected product

and 10 μm, respectively, and the average errors were approximately 0.00005° and 1.8 μm, respectively. However, in the case of off-center load (Fig. 14c), the maximum error was approximately 0.0013° and 31 μm. This was caused by instantaneous imbalance loads, which resulted in larger angular errors for the two motors at the instant the slide touched and left the gas spring. By applying the synchronous servo control technology, the angular errors of the motors adjusted within 20 ms, so the average errors were only approximately 0.00006° and 2.2 μm in the case of an off-center load.

4.2 Free-motion test

Verification of performance was also implemented for the free-motion function. We designed three common processing curves for the press machine: coining, cutting, and pulsating processes (shown in Fig. 18). We also set the processing speed of 25 SPM for the press machine, and calculated the actual moving path of the slide according to the feedback signals of the linear encoders. Finally, we compared it with the expected path of the command, and we obtained the results shown in Figs. 19 to 21 for the respective processes. We also compiled and analyzed the data based on these three figures as listed in Table 3.

Fig. 23 Final product (on the *right*)

Table 4 Comparison of dimensions of punch and final product

	Length (mm)	Width (mm)
Punch	29.434	28.607
Final product	29.436	28.609

The results shown in Figs. 19, 20, and 21 indicate that we successfully used the speed command to control the slide to run various processing curves in the expected manner. The benefit of using the speed command rather than the position command is that the slide will not run the speed in the reverse direction, which may be caused by the servo-tuning of the position control. However, because we did not run the servo control to position the slide, the maximum error of the driving path between the command and the actuality was more than 1.9 mm as in the case of Fig. 19. However, this error is a type of phase error rather than a real position error. Therefore, the tolerance is acceptable for the press machine because the actual processing curve of the slide still closely resembles the processing curve that we designed. In fact, the final accuracy of the products still depends on the accuracy of the press machine and the die, rather than the accuracy of the driving curve of the slide. The advantage of the specific driving curve is the improvement of the formability of materials.

4.3 Processing test

Finally, a processing test was performed. In this test, we ran a cutting process for a sheet material of stainless steel (SUS304), the thickness of which was 0.07 mm. Figure 22 shows the dimensions of the punch used. These dimensions were the same as the final expected dimensions of the product. We set the processing speed to 35 SPM and implemented the processing curve as shown in Fig. 20. The final product is shown in Fig. 23. A comparison of the dimensions of the punch and

Table 5 Comparison of three types of presses

	Large press machine with multi-servo system	Conventional small press	Proposed press
Max. capacity	High	Low	Low
Working area	Large	Small	Small
Stroke	Long	Short	Short
Max. speed	Slow	Fast	Fast
Accuracy of parts	Bad	High	Normal
Accuracy of press	Normal	Good	Good
Resistance to off-center load	Good	Bad	Good
Stability of quality	Normal	Normal	Good
Cost	High	High	Low

the final product is given in Table 4. We can see that the tolerance of the dimensions between the punch and the final product is only approximately 2 μm. Therefore, the proposed press machine provides high accuracy and is practical.

5 Discussion

Compared with large press machines with multi-servo motors, our proposed press has a smaller working capacity and working area; yet, it provides the benefits of higher accuracy, higher working speeds, and lower costs. Therefore, our proposed press is more suitable than large press machines for the mass production of small parts. When compared with conventional small presses, the control system of our proposed press is more complicated, but it avoids the use of mechanical parts and components that necessitate high accuracy and high costs. Moreover, because our proposed press includes control compensation, it is better at reducing the effect of off-center loads and maintaining long-run accuracy. In summary, the proposed press may be less expensive and provide better accuracy than the other two previously mentioned presses. Table 5 presents a comparison of these three types of press machines: the large press machine with the multi-servo system, the conventional small press, and the proposed press.

6 Conclusion

We have proposed a small mechanical press with an adjustable driving system that has a slide and a double-axis servo system for manufacturing small metal products. This design not only provides smaller servo presses with the free-motion function of regular servo presses for use in all types of manufacturing processes, but also provides the advanced ability (similar to some larger servo presses) to adjust the parallelism of the slide in real time. This is expected to improve the quality of the product and extend the life of the die. Reducing the wear caused by imbalance of the slide reduces the frequency of maintenance. Therefore, press machines with these characteristics should be suitable for the mass production of small metal products, particularly when requiring high-precision and stable quality. On the other hand, because we can correct the parallelism of the slide—caused by the tolerance of the mechanism—by means of the multi-servo control system, not all of the components and parts of the press machine need to be manufactured to high precision (<10 μm). This means that the cost of the press machine can be reduced efficiently.

In this paper, we also proposed to use the speed command rather than the position command, which is widely used in the most common servo presses to control the free-motion function. Using the speed command means that it can not only improve the response of the servo control but also avoids the speed and torque drop of motors caused by servo-tuning of the position control. Furthermore, we used the B-spline theory of curves to divide the curve of free-motion into hundreds of segments. With this, the controller is able to replace the original rough speed command with a more detailed speed command, and so the motors run smoothly with no sudden rises or drops in speed.

In future work, we will run further manufacturing tests such as forging, fast cutting, and progressive die manufacturing to verify the performance of the proposed servo press. Moreover, the reliability of the proposed servo press should also be tested prior to mass production.

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