Proceedings of the 17th World Congress
The International Federation of Automatic Control
Seoul, Korea, July 6-11, 2008

Control System Design for a New Servo Press

W. K. Yeung, J. P. Li, K. He, Y. X. Luo, C. T. Kong, R. Du

Institute of Precision Engineering
The Chinese University of Hong Kong
Shatin, N. T., Hong Kong, CHINA
(e-mail: wkyeung@mae.cuhk.edu.hk)

Abstract: A control system is developed for controlling a newly designed 300-ton servo press. The control goal is to drive the punch to track various desired trajectories, and to ensure the accurate repeatability of its bottom-dead-centre (BDC). Two major difficulties are encountered in designing the system. The first one is the nonlinear kinematics of the force-amplification mechanism. The second is the tight synchronization requirement of two servomotors. To cope with these problems, techniques such as kinematics buffers, cascaded feedback loops and cross-coupling method have been used. The control algorithms are implemented using six PID channels on a Turbo PMAC2 motion control card and are tested with experimental models. Satisfactory performance is obtained from the test results.

1. INTRODUCTION

For its low cost and high productivity, stamping operation is widely used to manufacture the components for the products we use nowadays. To fulfill various requirements, many different kinds of stamping presses have been developed (Wagener, 1997). Examples include conventional mechanical press, multi-link mechanical press, hybrid driven press, hydraulic press and servo press. Among these stamping technologies, servo presses possess a lot of advantages (Miyoshi, 2003). For example, they have programmable stoke and slide velocity, the ability to dwell in the stroke, and bottom-dead-centre (BDC) accuracy in microns (Landowski, 2004). Besides, they can offer high productivity, low maintenance, and reduced snap-through loads. More importantly, all these can be achieved with only little energy loss. For these reasons, recently a lot of research effort has been put on the development of servo presses. Representatives of these presses include Aida (On-line catalog) and Komatsu (On-line catalog). However, as high-power servomotors are usually required to overcome the peak torque during the stamping operation, these servo presses are relatively energy inefficient and expensive to build.

In view of the above weaknesses, we have designed an energy-efficient yet low-cost 300-ton servo mechanical metal forming press. Besides the mechanical design, another key element of the press design is the control system. In order to make full use of the capability of the press, a competent and reliable control system must be developed. This paper aims at introducing the major difficulties faced in the design process and our corresponding solutions.

The rest of this paper is organized as follows. The design of the press is presented in Section 2. The techniques used in our control system design are discussed in Section 3. The experimental setup and test results are presented in Section 4 and Section 5 respectively. Finally, a brief conclusion is given in Section 6.

2. DESIGN OF THE SERVO PRESS

Fig. 1 shows the assembly drawing of our new servo press. It is a close-frame, double-point press with a nominal capacity of 300-ton. Its overall dimensions is around 5m(H) x 3.5m(W) x 2m(D).

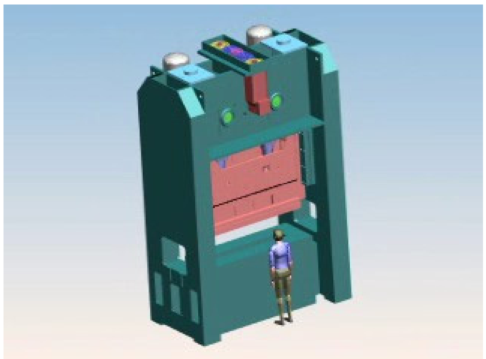

Fig. 1. Design of the 300-ton servo press

The core inside the press is a force amplification mechanism. Its main components are shown in Fig. 2. This nonlinear mechanism provides a two-level force amplification that effectively lowers the torque requirement of the servomotors, and makes our design more energy efficient when compared with other servo presses of the same tonnage. The slider block is actuated by two ball-screws that are driven respectively by two 51kW servomotors. As the slider moves up and down, the punch is driven to move correspondingly via a linkage system. As will be illustrated in Fig. 4, when the punch is closed to its BDC, it moves very slowly even if the slider moves fast. This action enables the press to produce a maximum of 300-ton punching force at the BDC. In order to guarantee the high accuracy of the BDC, a precise linear encoder is installed on the punch for measuring its actual position.

978-1-1234-7890-2/08/$20.00 © 2008 IFAC 10.3182/20080706-5-KR-1001.0063

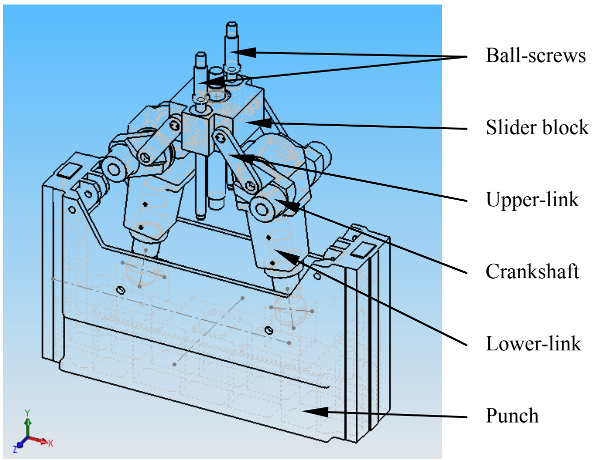

Fig. 2. Force amplification mechanism

To possess a 300-ton capacity a very high-power servomotor is required. However, even the largest commercially available ball-screw cannot withstand the large torque produced by a single motor. For this reason, two, instead of one, ball-screws are used in the design. Although two lower-power servomotors can now be used, using two motors introduces the problem of synchronization. In common gantry systems, as the two motors are usually far away from each other, the synchronization requirement is not so high. In contrast, since the two ball-screws are put quite close together in our design, a tight synchronization of servomotors becomes necessary.

In spite of the synchronization requirement, the design has the following advantages. With the servomotors and ball-screws installed vertically at proper positions, there is no lateral force acting on the servo input guides. Consequently, not only smaller rolling guides and hence fewer spaces are required, but the friction wear is also reduced. Further, without flywheel, clutch or huge gears, the structure of our design is simple and only easy maintenance is required.

3. CONTROL SYSTEM DESIGN

Our control goal is to drive the punch to track some desired trajectories and to maintain the repeatability of the punch BDC. To achieve this goal, some difficulties need to be overcome when designing the control system for the press. The first one is the nonlinear kinematics of the force-amplification mechanism. The second is the tight synchronization requirement of two servo motors. Also, to ensure the accurate repeatability of the BDC, an outer feedback loop is required. To solve these problems, techniques such as kinematics buffers, cascaded control loops and cross-coupling method are used. They are discussed in details below.

3.1 Nonlinear Kinematics Buffers

To handle the nonlinear kinematics of the force amplification mechanism, an approach similar to robotic applications is used. First of all we need to derive the exact kinematics relationship between the punch position and slider position. A forward and an inverse kinematics buffers are then setup. Fig. 3 shows a schematic diagram of the mechanism. Since the mechanism is symmetrical about its centre line, only a half-model is considered.

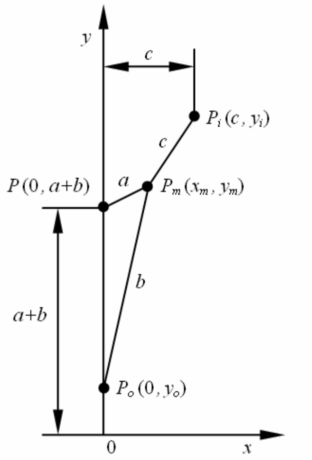

Fig. 3. Schematic diagram of the mechanism

In the figure, a, b and c are respectively the lengths of the crankshaft, the lower link and the upper link. P, P_m, P_i and P_o are the coordinates of a fixed and 3 moving joints respectively. Let the lowest possible point of P_o be the origin of the coordinates system, and P, P_o on the y-axis. The slider position y_i and the punch position y_o are then related by 3 equations:

$$x_m^2 + (y_m - a - b)^2 = a^2 \tag{1}$$

$$x_m^2 + (y_m - y_o)^2 = b^2 \tag{2}$$

$$(x_m - c)^2 + (y_m - y_i)^2 = c^2 \tag{3}$$

Since the mechanism has only one degree-of-freedom, both the forward and inverse kinematics are one-to-one mappings. By eliminating simultaneously the unwanted variables, the exact relationship between y_i and y_o is obtained. Due to limited space, details of the solutions are omitted here. Nevertheless, the relationship between y_i and y_o is plotted as Fig.4. Note that the slope of the curve keeps changing and is almost zero when y_o is close to zero.

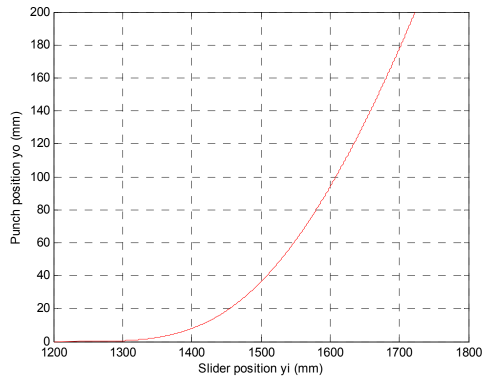

Fig. 4. Relationship between y_i and y_o

The forward kinematics buffer uses the slider positions as input, and converts them to punch coordinates. In order to establish the starting coordinates for the first programmed move, this calculation is required at the beginning of a sequence of moves programmed in punch coordinates. The inverse kinematics buffer uses the punch positions as input, and converts them to slider coordinates. This calculation is required for the end-point of every move that is also programmed in punch coordinates. If the path to the end-point is important, calculations must be done at periodic intervals during the move as well (Delta Tau Data Systems Inc., 2006a).

3.2 Cascaded Feedback Control Loop

Fig. 5 shows the block diagram of the overall control system. As the slider position is directly proportional to the rotational angle of the servomotor, the relationship between the slider motion and the servomotor motion is linear. Hence, a standard servo loop can be used directly to drive the slider to track its commanded position. Details inside the servo block, highlighted in the figure, will be further elaborated in next session.

Given a desired punch trajectory, its corresponding theoretical slider position can be obtained through the inverse kinematics calculation. In theory, since the kinematics calculation is exact, if the servo loop is well tuned, the punch should also track the desired trajectory well. However, in the actual case, there are some factors that may deteriorate the tracking accuracy of the punch position. These factors include backlashes inside the joints among the links, thermal expansion of the structures and distortion of the press components, etc. The last two factors are in particular significant when the press has been operated for a prolonged period and when a large impact force is produced under full-load condition.

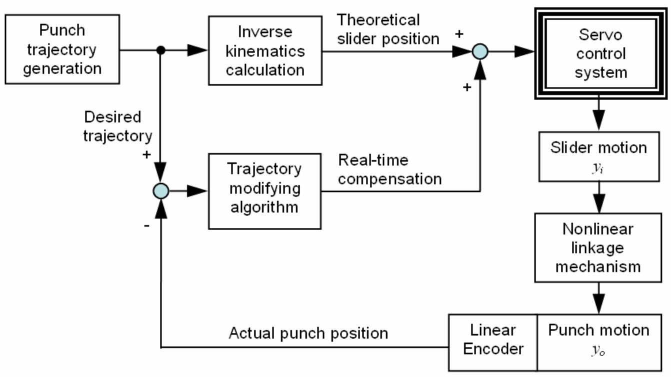

Fig. 5. Block diagram of the overall control system

To guarantee the high accuracy of the BDC, an outer feedback loop making use of the punch's encoder signal is introduced. As illustrated in Fig. 5, if the actual punch position deviates from the desired punch position, the error signal would drive the trajectory modifying algorithm to generate a real-time compensating signal. The commanded slider position is then superimposed by this compensation value before passing to the servo filter of the inner servo loop.

3.3 Cross-coupling of Two Servomotors

Our design requires the two servomotors always doing the same movement. This is similar to the application in gantry systems. Usually, the control of multi-motor gantry system is done by either the classic master-slave or the full coordination method (Tan, e*t al.*, 2004). In classic master-slave, one motor is chosen as the master motor, which is defined to the axis in the coordinates system and executes trajectories directly from the motion program. Its feedback encoder is used as the master encoder for the slave motor. However, the slave motor often operates more roughly than the master motor, in particular in systems where both motors have similar resonant frequencies. In the full coordination approach, the two motors are assigned to the same axis in the same coordinates system. A motion command for that axis then provides identical commanded trajectory to all motors assigned to this axis. In spite of this, as the motors still have independent servo loops, the actual motor positions will not necessarily be the same.

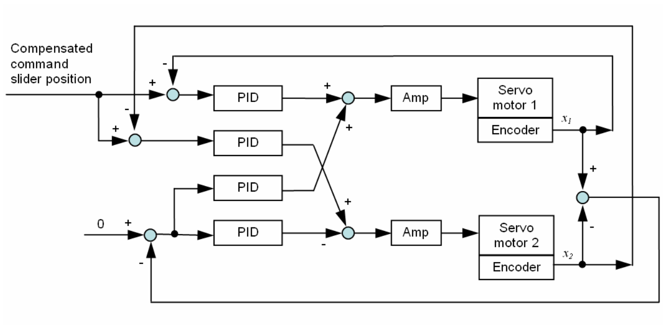

Fig. 6. Details inside the servo control loop

To further improve the performance of the above methods, a cross-coupling algorithm (Delta Tau Data Systems Inc., 2006b), as shown in Fig. 6, is hence proposed. In this approach four motor channels are required for use with two servomotors. Two of them are connected physically to the amplifiers of the motors, as in the full coordination configuration. The remaining two channels are used for de-coupling of the control signal. The two physical motor channels must be the higher number channels. For example, if motor channels 1 to 4 are used in the application, the two servomotors should be connected to channels 3 and 4. This ensures the control signals are sent at the end, but not at the beginning or middle, of each servo cycle.

The difference in position between the two motors are calculated and compared with zero. If it is not zero, then two compensating signals with opposite signs are generated to fine tune the trajectories of the two motors. Eventually, the control outputs are superimposed by these compensation values before passing to the servo amplifiers.

4. EXPERIMENTAL SETUP

4.1 Press Model and Control Hardware

The control algorithms discussed in Section 3 are implemented using the Turbo PMAC2 motion control card (Delta Tau Data Systems Inc., 2006a). This motion control card is capable of controlling eight axes simultaneously. In our application, six PID channels are used. One is for handling the nonlinear kinematics. One is for generating a compensating signal in the cascaded feedback loop. The other four are used in the cross-coupling control of two servomotors. The PID parameters are first adjusted by auto-tuning algorithms and then fine-tuned manually to fit the design specifications.

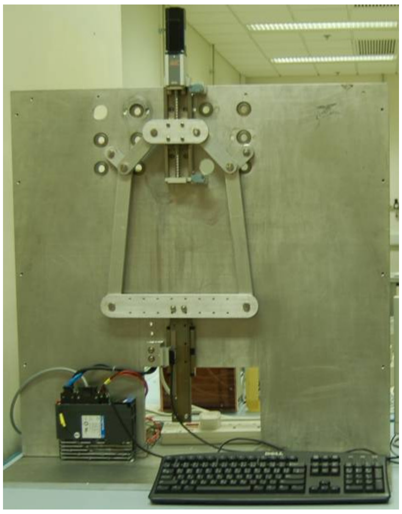

Fig. 7. 1:4 model of the servo press

Two experimental setups have been built to test the performance of the control algorithms. To test the tracking performance of the punch, a 1:4 model of the servo press as shown in Fig. 7 was built. Top above the ball-screw is a servomotor. Two limit-switches are installed at the two ends of the ball-screw to prevent the slider from over-traveling. The upper switch, together with the encoder's index signal, also acts as a reference point for the model's coordinates system. A magnetic linear encoder with 1 micron resolution is used to measure the actual position of the punch.

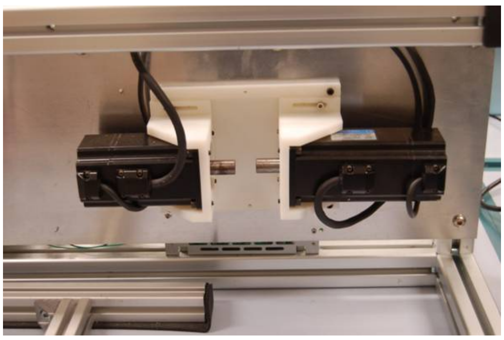

Fig. 8. Setup for testing synchronization performance

To test the synchronization performance, another setup as shown in Fig. 8 was built. The two servomotors have different but similar specifications. To find out the difference in the motor positions, encoders with resolution of 8,000 counts/revolution are installed on both servomotors.

4.2 Locating the Reference Position

The accuracy of the kinematics calculations depend greatly on the precision of the press's coordinates reference. In order to locate accurately a reference position on the press model, a simple movement test is performed. Based on the test result and the kinematics solution, an iterative method is then used to solve for the correct value of the reference point. The procedure is outlined as follows.

1. Command the slider to move upward until it triggers a reference point. In our press model, the upper limit-switch is used.
2. Reset both the slider's and punch's positions to zero.
3. Command the slider to move downward a known distance x_s. For an accurate result, this distance should be taken as long as possible.
4. The distance travelled by the punch, denoted by x_p, is recorded at the same time.
5. Assign a rough possible value y_f for the reference point.
6. Calculate $d = f(y_f) - f(y_f - x_s) - x_p$, where $f(.)$ is the forward kinematics mapping.
7. Plot d out over a possible range of y_f.
8. Take the value of y_f that renders d zero as the correct position for the reference point.
9. Obtain the corresponding punch's reference position via forward kinematics calculation.

As an example, x_s = 104.930mm and x_p = 56.003mm are obtained in a test on our press model. With a = 45mm, b = 300mm and c = 100mm, a series of d are calculated and plotted against a range of y_f as shown in Fig. 9. From the figure, the value 436.060mm is obtained as the slider reference position, as d is zero at this point. Via forward kinematics calculation, the corresponding punch reference position is obtained as 56.308mm.

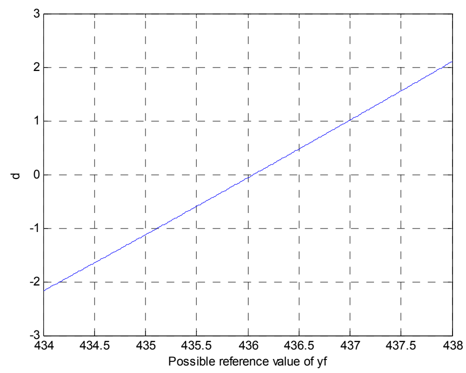

Fig. 9. Plot of d against a possible range of y_f

5. TEST RESULTS

5.1 Testing Trajectory

A typical application of stamping press is drawing operation which is commonly used to change the shape of a sheet metal (Kalpakjian and Schmid, 2006). A typical trajectory of such operation is shown in Fig. 10. In each cycle, the punch moves downward as fast as possible until the molds close. Then it slows down to deform the sheet metal and dwells at the BDC for a short while to prevent the sheet metal from springing-back. Finally, it returns to its starting position as fast as possible. As an illustration, this trajectory is employed in all the tests below.

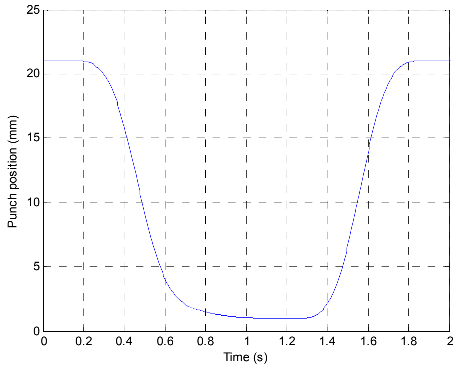

Fig. 10. Typical trajectory of a drawing operation

5.2 Open-loop Test

Given the punch trajectory, the theoretical slider position shown in Fig. 11 is generated via the inverse kinematics buffer. When this slider trajectory is applied, the punch can roughly track the required punch trajectory. However, as shown in Fig. 12, if the outer feedback loop is open, the tracking performance of the press is poor. There is an obvious offset between the actual punch position and the commanded position, in particular when the punch is moving upward or downward quickly. The magnitude of the offset is up to 33 microns at the BDC. This offset is mainly caused by the relatively large backlash in the press model, and thus cannot be eliminated by kinematics calculation.

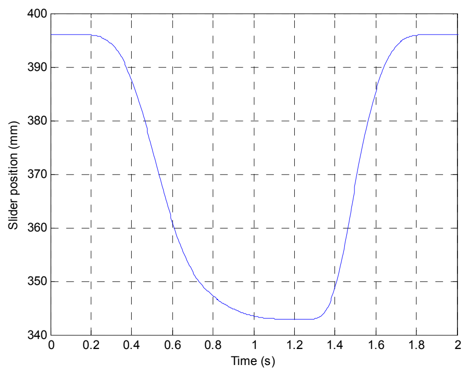

Fig. 11. Corresponding slider position

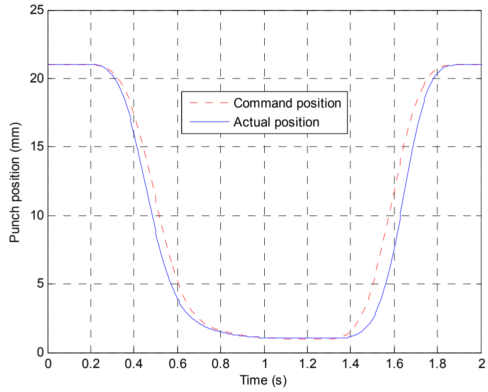

Fig. 12. Performance with outer feedback loop open

5.3 Closed-loop Test

When the outer feedback loop is closed, a real-time compensating signal as shown in Fig. 13 is produced. With this compensation added to the slider command, the tracking performance is much improved. As shown in Fig. 14, the punch can now track the desired trajectory well. The command position and the actual position almost overlap and the offset at the BDC is reduced to 3 microns. In other words, the cascaded feedback loop has effectively increased the repeatability of the punch BDC.

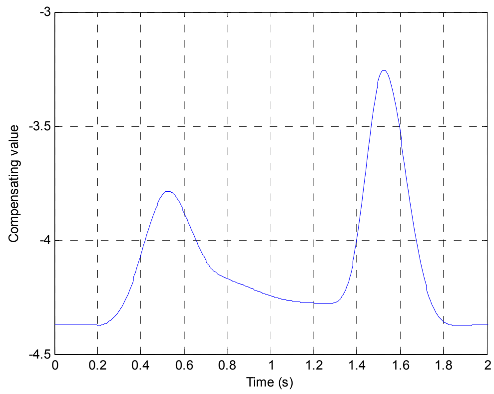

Fig. 13. Real-time compensating signal

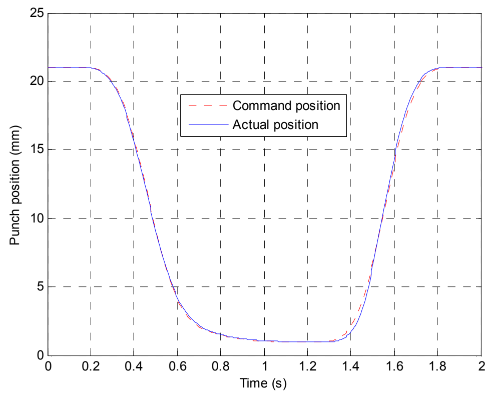

Fig. 14. Performance with outer feedback loop closed

5.4 Synchronization Test

In the synchronization test, the two servomotors are commanded to track the same drawing trajectory. Firstly, the motors are connected in full coordination configuration. As discussed in session 3.3, since the motors have independent servo loops, their actual positions are not necessarily the same. Fig. 15 shows the difference between the two servomotors' actual positions. The magnitude of this difference is up to ±6 resolution counts.

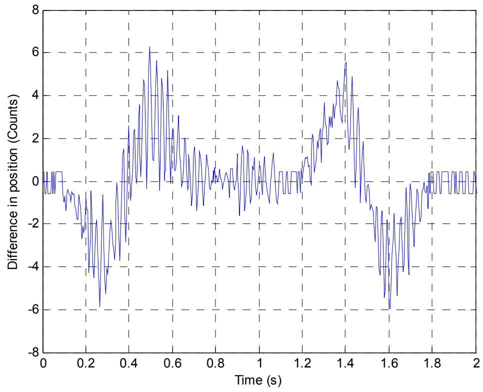

Fig. 15. Difference in servomotor position without cross-coupling

The cross-coupling method is then implemented. The difference between the servomotor positions is recorded as Fig. 16. The magnitude of the difference is reduced to ±3 resolution counts, almost half that without compensation. This illustrates the effectiveness of the method.

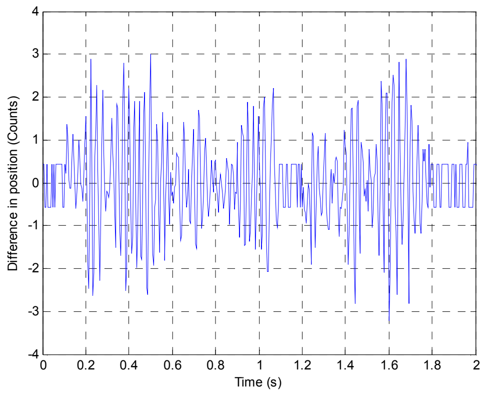

Fig. 16. Difference in servomotor position with cross-coupling

6. CONCLUSIONS

This paper presents the control system design for a new servo press. It utilizes kinematics buffers to handle the nonlinearity of the force-amplification mechanism, cascaded feedback loop to increase the repeatability of the punch BDC and cross-coupling method to improve the synchronization of two servomotors. These techniques are implemented using Turbo PMAC2 motion control card and are tested on experimental models of the press. Test results show that the techniques effectively improve the performance of the press model.

In this paper, six PID filters with fixed parameters are used. However, as the force-amplification mechanism is nonlinear, for optimal performance, different parameter setting may be required at different punch position. Hence, in order to further improve the press performance, a gain-scheduling PID setting is proposed in the future research work.

ACKNOWLEDGEMENTS

This project is supported by the Innovation and Technology Commission of Hong Kong (GHS/008/04). The authors also like to thank Guangdong Metalforming Machine Works Co., Ltd. for building a prototype of the servo press.

REFERENCES

Aida (On-line catalog). NS2-series servo press, http://us.aida-global.com/products/ns2.jsp

Delta Tau Data Systems Inc. (2006a). *Turbo PMAC user manual*, version Nov 2006.

Delta Tau Data Systems Inc. (2006b). *Cross coupled drive of dual-motor gantry system – PMAC application note*, downloadable at http://www.deltatau.com/common/support/technotes/Gantry%20with%20Cross%20Coupling.pdf

Kalpakjian, S. and S. R. Schmid (2006). *Manufacturing Engineering and Technology*. Prentice Hall, Singapore.

Komatsu (On-line catalog). H2F hybrid AC servo presses, http://www.komatsusanki.co.jp/en/tanatsu/eH2F.html

Landowski, J. (2004). Sizing up servo presses – a look at tonnage and energy, *Stamping journal*, **April**.

Miyoshi, K. (2003). Current trends in free motion presses, Metalforming magazine, **37**, 6.

Tan, K. K., S. Y. Lim, S. Huang, H. F. Dou and T. S. Giam (2004). Coordinated motion control of moving gantry stages for precision applications based on an observer-augmented composite controller, *IEEE trans. control systems technology*, **12**, 6, 984-991.

Wagener, H. W. (1997). New development in sheet metal forming: sheet materials, tools and machinery. *Journal of materials processing technology*, **72**, 342-357.