SLIDING MODE FORCE CONTROL DURING DRAWING PROCESSES IN PRESSES

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Abstract: Nowadays high-tonnage drawing presses need a controlled cushion so that the force is controlled during the drawing process. Although with a classic force control algorithm, consisting of a non-linear feedforward loop and a closed loop PID control, the results are normally quite good, there are times when tuning up is difficult and performance diminishes significantly at high rates. A new force control has therefore been developed applying sliding mode techniques, which is more robust than the previous control, especially in problem cases.

Keywords: Sliding-mode control, force control, feedforward, robust control, PID.

1. INTRODUCTION

In recent years, high-tonnage drawing presses have been coming to the fore in the manufacturing industry, especially in the automotive sector. Double-action presses on press lines are being replaced by single-action presses with a controlled hydraulic cushion. If a deep drawing has to be carried out the controlled cushion is now a basic requirement for obtaining parts with the quality required.

The hydraulic cushion on a press has a two-fold function: on the one hand, it holds the sheet metal to be drawn until the slide arrives and drawing begins. On the other hand, during the drawing process, the cushion controls the force exerted against the slide, in order to help material flow and improve part quality.

The Fagor Arrasate and Ona-Pres companies, mechanical and hydraulic press manufacturers respectively, developed a controlled hydraulic cushion several years ago in collaboration with Ikerlan. It is being fitted in their presses when required by the deep drawing process. An algorithm based on a feedforward loop of the slide speed during drawing is used for force control in the hydraulic cushion designed. But increasingly more demanding production requirements mean that rates increase and therefore the slide speed during drawing. As a result, sometimes the performance of the force control is significantly diminished.

The aim of the work carried out and presented here, was to design a new force control algorithm for the drawing process that would be more robust than the one currently used.

The electro-hydraulic systems are highly non-linear and their models contain both parametric and non-structured uncertainties. Recent articles (Bonchis *et al.*, 2001; Pommier *et al.*, 2002) have demonstrated that the application of robust control techniques is necessary for optimum control in certain applications. Among these techniques, variable structure control, known as sliding mode control, has attracted a considerable amount of attention due to its stability properties, and the performance obtained, faced with the imprecision of the model or disturbances. Although in most of the sliding mode control references in electro-hydraulic systems it is used for

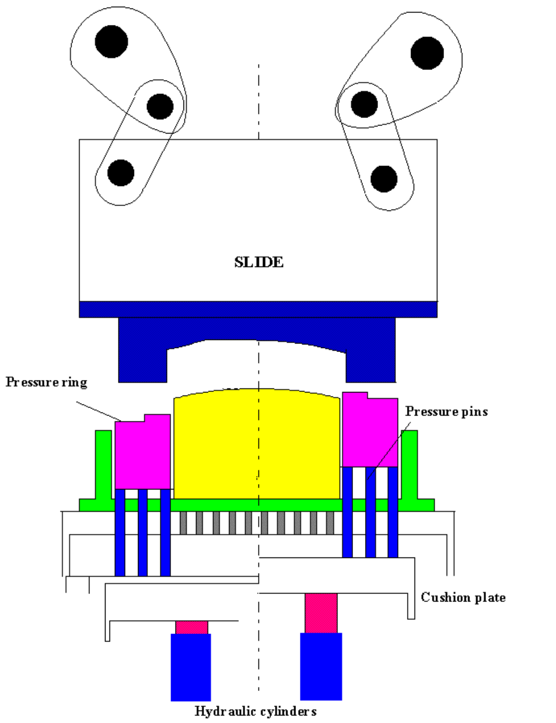

Fig. 1. Diagram of a single-action mechanical press and hydraulic cushion.

position control (Mihajlov *et al.*, 2002), there are also cases in which it has been used for pressure control (Fink and Singh, 1998).

This paper describes the application of sliding mode techniques for force control in a hydraulic cushion during the drawing process in presses. It starts by formulating the problem to be solved, describing the hydraulic cushion and the force control algorithm implemented and used until now. It goes on to present the design of the sliding mode force controller. The results obtained in simulation with the previous algorithm are compared to those obtained with the newly designed one. Although the new algorithm has been implemented in a hydraulic cushion industrial controller, it has not been tried experimentally on a mechanical press yet. Instead it has been tried out with the virtual prototype of a new programmable TRY-OUT press. Some experimental data are presented.

2. PROBLEM FORMULATION

2.1 Description of the hydraulic cushion.

Figure 1 shows the sketch of a single-action mechanical press and hydraulic cushion as a whole. Figure 2 shows the typical positions of the slide and the cushion during a mechanical press cycle. Before contact, the cushion preaccelerates so that, at the moment of contact, the relative speed between the slide and the cushion is the programmed value (typically between 30-50% of the speed of the slide at the contact point). This reduces vibrations, improves part quality and prevents acoustic contamination.

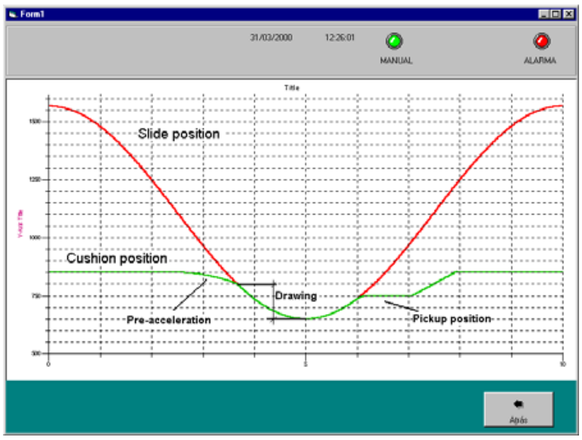

Fig. 2. Typical slide and cushion positions.

After impact, the drawing process starts. The slide moves the cushion and the cushion carries out force control during the whole process, until the slide reaches the bottom position. It can also do so whilst the slide goes up to its top position if it has been programmed to accompany it. In this case, only a minimum force is exerted to ensure contact.

The hydraulic cushion developed by Fagor Arrasate and Ona-Pres in collaboration with Ikerlan, consists of a certain number of hydraulic cylinders, each controlled by a high flow rate servovalve with good dynamic characteristics. Figure 3 shows the hydraulic diagram for each cushion cylinder. As can be seen, it is an asymmetrical cylinder where the top chamber pressure is the constant supply pressure. By means of the servovalve, oil is removed from the bottom chamber when the cushion descends, that is to say, when it is preaccelerating or during drawing. When the cushion ascends in a controlled way, the servovalve delivers oil from the supply pumps.

2.2 Speed-feedforward based force algorithm.

Force control of a hydraulic circuit like the one shown in Figure 3 is not simple. A good performance is not usually achieved with PID type algorithms,

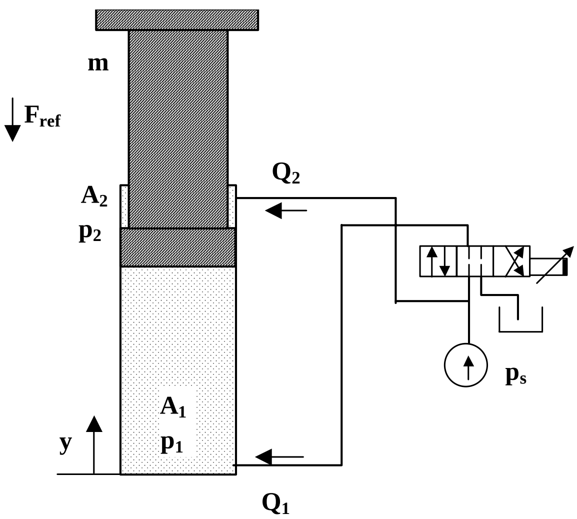

Fig. 3. Hydraulic circuit for each cushion cylinder.

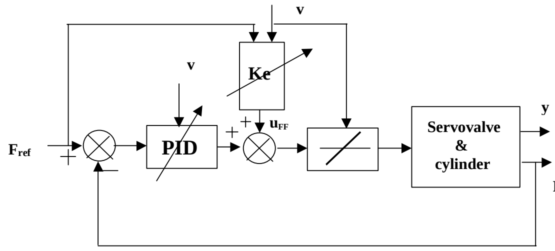

Fig. 4. Feedforward based force algorithm.

mainly due to the high speed of the slide during the drawing process. The algorithm finally implemented is shown in Figure 4. The key part is the feedforward loop, which taking into account the characteristics of the servovalve and the theoretical speed of the piston (which can be estimated at each point), calculates the theoretical reference value for the amount of oil to come out or go in due to the movement of the slide and the programmed force to be exerted. A closed force control loop offsets the errors introduced by the feedforward loop. Normally it is a proportional control with a different gain when the slide is going down (drawing) or going up.

The formula that this feedforward loop K_e implements is as follows:

$$u_{FF} = \frac{10 \cdot A_1 \cdot v}{A_0 \cdot Cd \cdot \sqrt{\frac{2 \cdot |\Delta p|}{\rho}}} \qquad (1)$$

where $\Delta p = p_s - p_{1ref}$ when v>0 (piston going up) and $\Delta p = p_{1ref}$ when v<0 (piston going down).

In the equation (1) u_{FF} is the feedforward control signal for the servovalve, A_1 is the lower chamber area, $v = \dot{y}$ the piston speed, A_0 is the maximum flow area of the control servovalve, C_d is the discharge coefficient, ρ is the mass density of the fluid, p_s is the inlet pressure of the oil into the servovalve and $p_{1\text{ref}}$ is the pressure set-point value.

The $p_{1\text{ref}}$ value is calculated from the force set-point value F_{ref} ($A \cdot p_{1ref} = F_{ref} + A_2 \cdot p_2$), where it is supposed that the weight mg is included in F_{ref}.

Moreover, the algorithm only allows the oil to go out and not come in during drawing. Otherwise, when a minimum force is being applied whilst the slide goes up, the oil can only go in.

2.3 Problems in high-speed drawing processes.

In a hydraulic press the drawing speed is constant and normally low, between -20 y -50 mm/s. In a mechanical press, where the slide follows a pure cosine type path (figure 2) or a smoothed out cosine (link-drive press), the slide speed is a sine type function. So for deep drawing and high rates, slide speed during drawing can be very high. Fagor Arrasate's specifications for their mechanical presses indicate maximum slide speeds during drawing of up to -500 mm/s. Obviously, this speed occurs during the initial moments of drawing, after impact, and falls to zero when drawing is over.

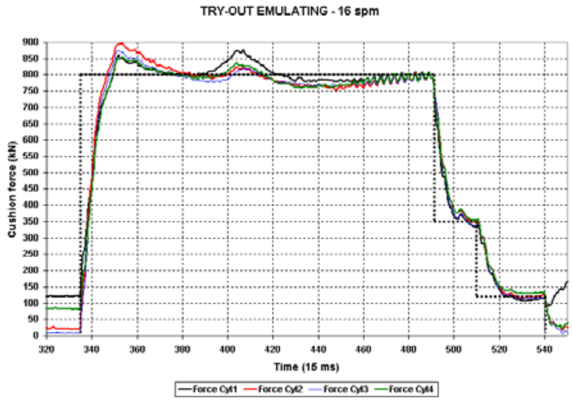

Fig. 5. Experimental force results in a special press.

The force control exerted with the algorithm presented above is very good for low or medium speeds (up to about -300 mm/s), but some times the performance degenerates at higher speeds, depending a great deal on each installation. Although very good results are always obtained in simulation, in reality there are times when the performance is usually worse, that is to say, the algorithm is not robust enough faced with non-linearities and unmodelled uncertainties. There are times when vibrations occur, and therefore the proportional gain has to be reduced until they are eliminated. This means that the response is slower and that the steady-state error may be quite large. For example, and although it is a relatively extreme case, Figure 5 shows the experimental force obtained with a reference value of 800 kN in a programmable hydraulic TRY-OUT press, emulating a mechanical link-drive press, with a maximum speed at the start of drawing of -350 mm/s. A characteristic of this press is that the slide path is programmable and the drawing conditions of any mechanical press can be emulated. This type of press is used to set up dies before they go into production. In this press, due to disturbances in slide speed and the interaction between the slide and the cushion, the force control results are quite a lot worse than those obtained with the hydraulic cushions in conventional mechanical presses. As can be seen, there is a considerable overshoot and some not very desirable vibrations. The response degenerates a lot more when rates are increased and slide speed is about –500 mm/s.

The force exerted by each cylinder is programmable between 100 and 1000 kN. Normally the indicated algorithm is tuned up for a constant force of 500 kN, at a rate corresponding to a maximum drawing speed of -250 mm/s. With this tuning up, the response is worse for smaller forces (typically 200 kN) and for bigger forces (up to 1000 kN). Owing to these effects, different closed loop proportional gains are often programmed for small forces and for big forces.

These problems, and the certain difficulty of tuning up the controller in actual cases, led to the design of a new robust control using the sliding mode theory.

3. DESIGN OF THE SLIDING MODE FORCE CONTROLLER

In the hydraulic circuit for the cushion shown in Figure 3, the following equations can be proposed for balancing the flow rates (Pommier, 2002):

$$\begin{aligned} Q_1(u, p_1) &= \frac{Cd \cdot A_0}{10} \sqrt{\frac{2 \cdot |\Delta p|}{\rho}}\, u \\ Q_2(u, p_2) &= -Q_1(u, p_1) \end{aligned} \tag{2}$$

where A_0 is the servovalve's maximum oil pass area, Cd is the discharge coefficient and ρ is the mass density. The signal u (±10 V) is the servovalve input. The servovalve dynamics are not modelled, although they are of decisive importance.

Moreover
$$\begin{aligned} \Delta p &= p_s - p_1 \quad if \quad u > 0 \\ \Delta p &= p_1 \quad\quad\;\; if \quad u \le 0 \end{aligned}$$

The cylinder is modelled using the thermodynamic equation that gives the pressure behaviour

$$\frac{V}{B}\frac{dp}{dt} + \frac{dV}{dt} = Q \tag{3}$$

where B is the bulk isotherm modulus, V is the volume of each cylinder chamber, p is the chamber pressure and Q is the mass flow rate in the chamber. For the hydraulic circuit under study (Figure 3), the mass flow rates Q_1 and Q_2 are given by (2). Thus the pressure behaviours in the two cylinder chambers are described by

$$\dot{p}_1 = \frac{B}{A_1 \cdot y}\left(Q_1 - A_1 \cdot \dot{y}\right) \qquad \dot{p}_2 = 0 \tag{4}$$

Substituting (2) in (4) the pressure in the lower chamber can be put in the form

$$\dot{p}_1 = f(y, \dot{y}) + b(y, p_1) \cdot u \tag{5}$$

where

$$f = -\frac{B \cdot \dot{y}}{y} \qquad b = \frac{B \cdot Cd \cdot A_0}{y \cdot A_1 \cdot 10}\sqrt{\frac{2 \cdot |\Delta p|}{\rho}} \tag{6}$$

The system dynamics f and the control input gain b can be estimated by $\hat{f}$ and $\hat{b}$, where the estimation errors are assumed to be bounded by some known functions F and β:

$$\left|f - \hat{f}\right| \le F \qquad \left|b - \hat{b}\right| \le \beta$$

In this case the state error vector is the scalar e_{p1}, so the sliding surface s=0 is defined as

$$s = e_{p1} = p_{1ref} - p_1 \tag{7}$$

The best approximation $\hat{u}$ of a continuous control law would be achieved from the condition $\dot{s} = 0$, resulting in (Slotine and Li, 1991):

$$\hat{u} = \hat{b}^{-1}(\dot{p}_{1ref} - \hat{f}) \tag{8}$$

To accommodate the estimation errors, a discontinuous term is added to (8), where k can be used as a parameter to be tuned up:

$$u = \hat{u} + k \cdot \mathrm{sgn}(s) \tag{9}$$

Normally, to avoid the chattering phenomenon, the *signum* function is replaced by other functions that smooth the control function (9), but in this case it is not necessary due to the limited bandwidth of the servovalve, the dynamics of which were not taken into account in the model.

Examining the expression (8), it can be seen that when the set-point value of the force to be exerted during drawing is constant ($\dot{p}_{1ref} = 0$), $\hat{u}$ coincides with the feedforward term (1) used in the classic algorithm. Moreover, the force profile to be exerted during drawing, although programmable, is normally made up of sections of constant forces.

4. RESULTS IN SIMULATION

Fagor Arrasate's hydraulic cushion is made up of a variable number of cylinders (normally between 4 and 8), with a hydraulic circuit like the one shown in Figure 3. In a typical configuration, the cylinders are 350 mm, with diameters of 250 and 200 mm, and each cylinder is controlled by a Rexroth 4WRDE32V-600L servovalve. These servovalves have a bandwidth of 20 Hz for a control signal step of 100% and 70 Hz for a step of 25%. The supply pressure p_s is 90 bars. The force that each cylinder can exert during drawing can be programmed between 100 kN and 1000 kN. The force profile is programmable, although it normally consists of sections of constant values.

A system like the one mentioned was modelled using the SABER package. For the slide a sinusoidal path

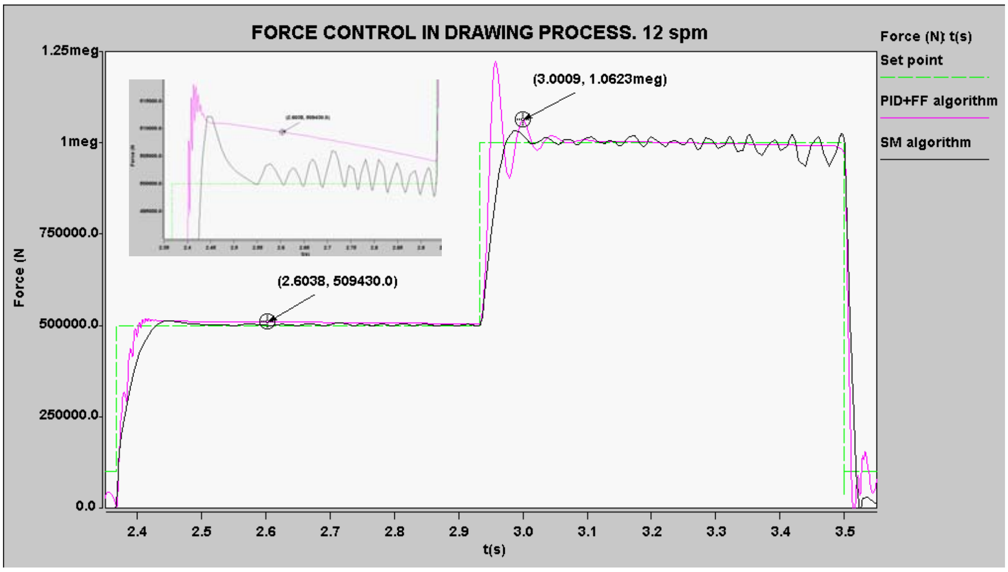

Fig. 6. Results in simulation with FF based algorithm and sliding mode algorithm at a rate of 12 spm.

with amplitude of 350 mm was supposed. Normally, the controller is tuned up at an average slide speed - about 250 mm/s, that in a sinusoidal press corresponds to a rate of 8 spm (strokes per minute) - and an average force (typically 500 kN). The preacceleration distance was 30 mm, the drawing height 220 mm and the speed relationship at the moment of impact 50%.

Figure 6 shows the force response to a force profile made up of two sections of 500 and 1000 kN, at a rate of 12 spm. With this rate the maximum slide speed in the specifications is reached (500 mm/s during drawing). The results shown in Figure 6 for the classic algorithm (PID with feedforward) and the sliding mode algorithm designed, are very similar to those obtained at the tuning speed (8 spm) and at other intermediate ones. That is to say, in principle, both algorithms are robust. This robustness has been tested by varying the model's parameters. In the classic algorithm, in addition to the feedforward term (1), a closed loop with a proportional gain of 0.004 was considered, with the force set-point values in kN. If this classic algorithm is tuned up at an intermediate force (500 kN), the response degenerates for small forces (200 kN) and big forces (1000 kN). This is clear from the response shown in Figure 6. The figure also shows that the steady-state error is small. In the response obtained with the sliding mode algorithm according to the control (9), for a k value of 0.3, oscillations typical of this sort of controller can be observed, which maintain the force's response in a band around the reference value. The zoom that appears in the top left-hand part of the figure shows a detail of the response with the two algorithms to a set-point value of 500 kN. The oscillations of the sliding mode algorithm are about 20 kN, which are really very small. In the response with both algorithms to the 1000 kN section of the set-point profile, it can be seen that the important initial overshoot obtained with the classic algorithm is offset with the sliding mode algorithm, although the typical oscillations are maintained during the whole section. As a conclusion, both algorithms perform well and are robust in simulation.

Figure 7 shows the control signals for force control during the drawing process. The corrective effects of the sliding mode algorithm can be observed.

5. RESULTS IN A VIRTUAL PROTOTYPE OF A TRY-OUT PRESS

The sliding mode algorithm (9) has been programmed in the industrial controller of the hydraulic cushion and it will be experimentally tested on a real press. For the time being the actual industrial controller has been tested on a virtual prototype of a TRY-OUT press at Ikerlan (Landaluze *et al.*, 2004). According to the experiments carried out when the TRY-OUT press was designed and built, the results obtained with the real press and the virtual prototype were similar. Therefore, the results obtained with the cushion controller can be considered representative of what would happen with a real press.

Figure 8 shows the responses obtained with the actual controller and the virtual prototype at a rate of 12 spm (specifications limit) with both the classic algorithm and the sliding mode algorithm. The force

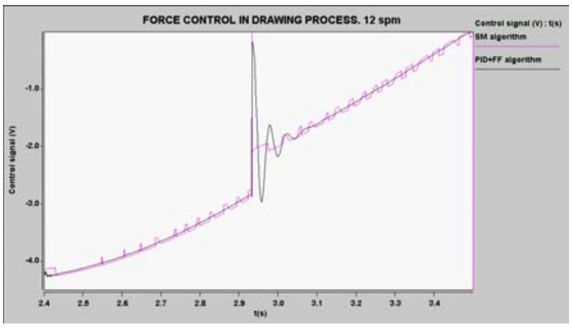

Fig. 7. Control signals with both algorithms.

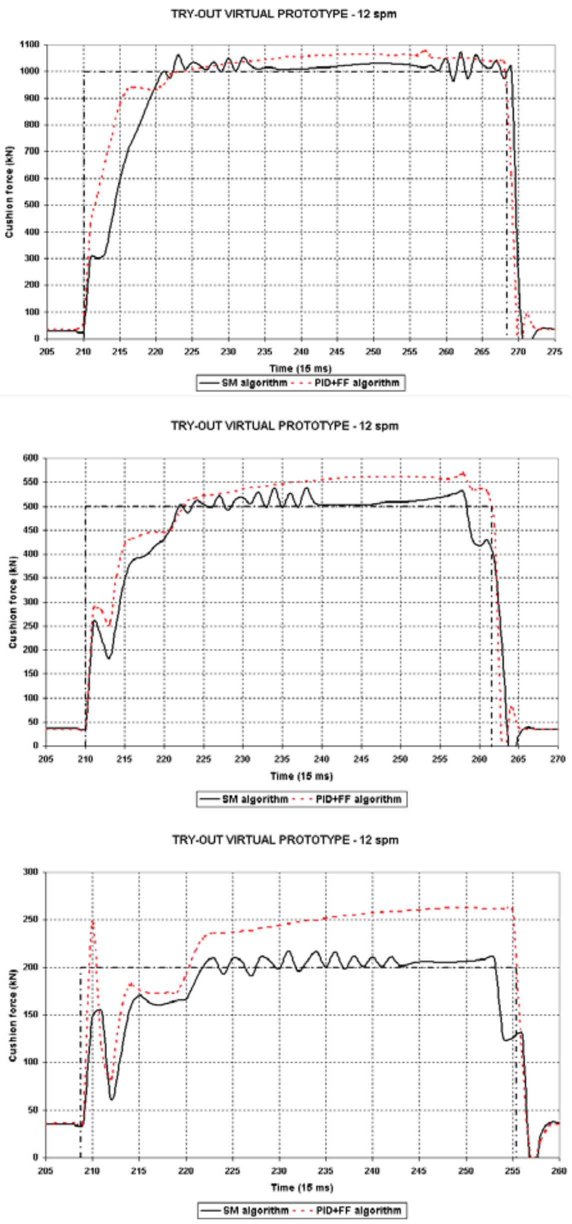

Fig. 8. Force responses obtained in the Virtual Prototype with both force algorithms for low, medium and high set-point forces.

set-point values were 200 kN (lower graph), 500 kN (graph in the middle) and 1000 kN (upper graph). For low forces, it can be seen in the lower graph that although at the beginning, at the moment of impact, there is some disturbance, the classic algorithm gives rise to a large steady-state error, of more than 50 kN, whilst the sliding mode controller better maintains control in respect of the set-point value. The oscillations of about 10 kN are practically insignificant. In the other two graphs, it can be seen that the stationary error is 60 kN for the classic algorithm, and the slide mode algorithm follows the reference value well with its typical oscillations. It should be pointed out that the disturbance that appears at the moment of impact and that does not appear in simulation is due to the particular characteristics of the TRY-OUT press. On this press (Landaluze *et al.*, 2004), the programmed position of the slide is also controlled and the cushion disturbs it at the moment of impact. This disturbance is clear in the force responses shown in Figure 8. But this does not happen with conventional mechanical presses.

6. CONCLUSIONS

In the force control carried out using a controlled hydraulic cushion during the drawing process in presses and using a classic algorithm consisting of a non-linear feedforward loop and a closed loop PID control, the results are normally quite good, in terms of both performance and robustness. But sometimes the performance diminishes significantly at high rates. Therefore, a new force control has been designed using sliding mode techniques. This paper has presented the design process for the control together with the results obtained in simulation with the SABER package. After implementing the controller, it was also tested with a virtual prototype of a programmable TRY-OUT press, which made clear its advantages compared to the results obtained with the classic algorithm.

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