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A study of the control of the blank holding force using an MR damper in a drawing press †

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Abstract

The predominant failure modes (wrinkling and tearing) must be avoided during the drawing process in sheet metal forming. These defects may be eliminated by using a controllable device for which the blank holding force (BHF) is adjustable. The purpose of this research is to verify the possibility of using a magneto-rheological (MR) damper for obtaining an almost constant BHF in drawing presses. The damper force is controlled by adjusting the current that is applied to the MR fluid, which is a functional material. To realize this aim, a prototypical press system is manufactured. A control test using a closed-loop PID controller is carried out for achieving the objective where by a constant BHF is retained at a constant prescribed force, while the press slide translates at a constant velocity. The results show that the BHF of the drawing press can be controlled effectively by using the proposed MR damper.

Keywords: Blank holding force; Closed-loop control; Damper; Drawing press; Magneto-rheological fluids

1. Introduction

In drawing presses, the force that is used to clamp the blank between the die and the blank holder is called the blank holding force (BHF). The BHF must be controlled to achieve many desirable objectives, including prevention of the occurrence of tearing or wrinkling in an optimum manner. Kergen and Jodogne claimed that among the significant parameters in deep drawing, the BHF is one of the most important; obviously, the main purpose of BHF is to avoid wrinkles between the die and blank holder [1]. According to Ahmetoglu et al. the major factor affecting the occurrence of defects in sheet metal parts is the BHF [2]. Not enough of a force tends to cause wrinkling of the drawn part, but excessive force causes tears. To avoid these major problems, it is important to appropriately control the BHF. The system for controlling the BHF can be classified into two different types in terms of the operating fluid: gas-spring systems and hydraulic-control systems, which use the BHF as a feedback parameter.

Gas-spring systems have been widely applied in drawing presses for sheet metal forming to build up the BHF, as shown in Fig. 1. These gas-spring systems have several advantages: relatively low cost, compactness of construction, and easy insertion in every press as required. However, there are also several disadvantages in using these systems. One is that gas-springs represent the progressive behavior of the force versus the stroke. For deep drawing, a constant and often retrogressive BHF is desired [3]. To control the BHF in gas-spring systems, Siegert and Hohnhaus developed a gas spring that controls the BHF over the ram stroke, which can avoid tearing in the sheet that is caused by excessively high normal pressure [4]. The valves for controlling the gas flow into the upper gas chamber were effectively used and then controlled the spring force over the spring displacement. However, time delays and valve dynamics have been found to significantly affect system performance.

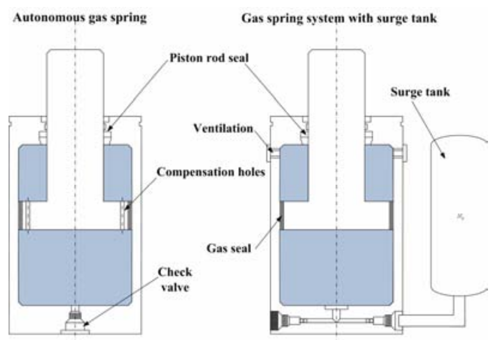

Fig. 1. Schematic design: a gas spring without (left) and with an accumulator (right).

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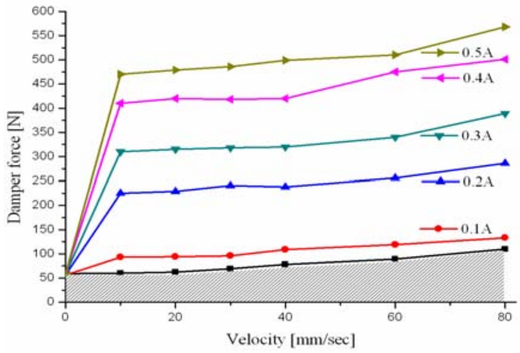
Gunnarsson et al. developed a new blank holder system that consists of inter-connected active and passive gas springs that move in parallel during forming [5]. With this system, retrogressive, constant, and progressive blank holding force trajectories were pre-determined using special software. Today, modern drawing presses have hydraulic blank holders that are usually installed in the press table as die cushions that build up the BHF through the blank holders [6-9]. Hydraulic cushions with proportional or servo valves that are used to achieve accuracy in BHF control in press systems have proven to be effective. This makes it possible to run specific blank-holding forces over the stroke for each hydraulic cylinder. However, these systems have complex structures and are very costly. All these studies indicate that BHF control systems such as controllable gas-springs and hydraulic systems can effectively control the BHF over the stroke with modern control approaches and yield improved results in sheet metal forming. However, few contributions in the prior literature have dealt with new and viable solutions to effectively control a constant BHF.

In this paper, the application of magneto-rheological (MR) dampers to control the BHF of press systems during drawing operations is considered. A number of analytical and experimental studies have clearly established the benefits regarding the potentially superior performance of MR fluid-based dampers in various systems, such as vehicle suspensions, vibration control devices, active buffers, and seismic applications. The principal aim of the investigation described in this paper is to evaluate, through experiments, the feasibility and practicability of implementing MR dampers and understand the potential benefits. First, the introduction of MR fluids and the characteristics of the commercial MR fluid-based damper used in this work are presented together with the test results that were carried out to obtain the force-velocity curve. After the test setup — which basically consists of a hydraulic cylinder, MR damper, and data acquisition system — are described experiments pursued to evaluate the system response for several values of the piston velocity and the applied current level. Finally, a PID feedback controller is developed to address the problem where the damper force is retained at a constant prescribed current level while the shaft translates at a constant velocity. The results obtained show that the basic performance of the system using an MR damper is satisfactory for the control of the constant BHF.

2. MR dampers and their modeling

MR fluids are a special class of rheological fluids whose yield stress varies with the applied magnetic field. The fluids consist of suspensions of micro-sized ferromagnetizable particles that are immersed in a carrier fluid such as silicon oil or water. The primary advantage of MR fluids stems from their high dynamic yield strength due to the magnetic energy density that can be established in the fluids. In the absence of magnetic fields, these fluids exhibit Newtonian behavior. The

(a)



(b)

Fig. 2. (a) The magneto-rheological (MR) damper used. (b) The force-velocity curve.

application of an external magnetic field causes the particles to become aligned with the field, and dramatically changes the effective viscosity of the fluid. Depending on the strength of this applied magnetic field, the viscosity (or yield strength) of MR fluids can reach that of Bingham solids, making them well-suited for semi-active applications. Several commercial applications of this kind of semi-active actuator can be found in the literature; see for example the works by Chrzan and Carlson [10], Dyke et al. [11], Breese and Gordaninejad [12] and others described in Carlson [13, 14].

Two important characteristics of MR fluids are as follows. First, they exhibit linear responses, i.e., the increase in stiffness is directly proportional to the strength of the applied magnetic field. Second, they provide fast responses, i.e., MR fluids change from the fluid state to a near-solid state within milliseconds of exposure to a magnetic field. Fig. 2(a) shows a commercially available MR damper manufactured by Lord Corporation that is used in this study [15]. The length of the damper is 20.8cm in its extended position. It has a 5.3cm stroke. The typical properties of this damper are listed in Table 1. The force-velocity curve is commonly used to characterize the performance of MR dampers and depicts the range of controllability that is applicable for MR dampers. The force-velocity characteristics of the damper used in this study are experimentally tested. The characteristics of the MR damper used are shown in Fig. 2(b), where the hatched region in the first quadrant refers to the physical constraints, i.e., the MR damper cannot generate a force depending on the input current.

Table 1. Typical properties of the used damper.

Property	MR damper (RD-1005-3)
Body diameter [mm]	41.4
Shaft diameter [mm]	10
Damper force [N]	> 2224 (5cm/sec @1A)
Operating temperature [$°C$]	71
Input current [Amp]	1max (continuous for 30 seconds)
Resistance [Ω]	5 @ ambient temperature

Furthermore, the damper force is not strictly linear with respect to the control input. Although many models have been developed for predicting the nonlinear dynamic of MR dampers, in this study, the Bingham model proposed by Stanway et al. is used to determine the damper force [16]. In this model, the force generated by an MR damper is given by:

$$F_{mr} = f_c \,\mathrm{sgn}(\dot{x}) + c_0 \dot{x} \tag{1}$$

where $\dot{x}$ denotes the velocity that is attributed to the external excitation, c_0 is the damping coefficient, and f_c is the frictional force, which is related to the fluid yield stress. The interested reader is directed to a review of MR damper models, as presented by Spencer et al. [17]. The Bingham model accounts for the behavior of MR fluids beyond the yield point, i.e., for fully developed fluid flow or sufficiently high shear rates. Note that the magnitude of the hatched area in the first quadrant indicates the frictional force between the piston rod and the cylinder housing; it is lumped into the MR damper model. The force, f_c, is bounded and proportional to the applied current, $i(t)$:

$$\begin{aligned} &f_0 \le f_c(t) \le f_{\max} \quad \mathit{for}\ \ 0 \le i(t) \le i_{\max} \\ &f_c(t) = f_0 + \gamma i(t). \end{aligned} \tag{2}$$

In Eq. (2), γ is the constant that linearly scales the force resulting from the applied current $i(t)$. The desired BHF that is determined by the control algorithm must be converted to the coil control current. Thus, the control input current that is required for achieving the desired MR damper force is given by:

$$i(t) = (f_c(t) - f_0) / \gamma. \tag{3}$$

3. Experimental setup and experiments

To test the performance of the MR damper used, as well as experimentally validate its theoretical model and perform closed-loop force control experiments, the experimental setup shown in Fig. 3 has been developed. This setup makes it possible to run a specific BHF over the stroke for a hydraulic cylinder. The system is capable of a reproducible BHF curve over the stroke and can be monitored by appropriate sensors during the cylinder stroke. One of the damper shafts is attached to the load cell under proper alignment and the other to

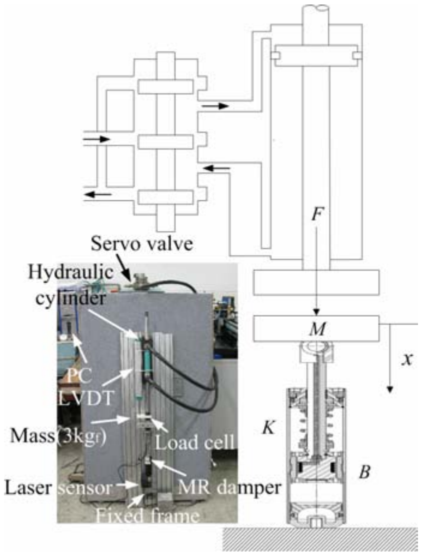


Fig. 3. Experimental setup for using an MR damper to control the BHF.

a fixed base frame. The other side of the load cell is connected to a mass, $m = 3kg$, for measuring the BHF output against the cylinder. This double-rod cylinder is controlled by a servo valve with good dynamic characteristics. A linear motion transducer is mounted on the exciting cylinder rod and the table frame to control the displacement. A laser displacement sensor is mounted on the damper to measure the relative displacement.

The velocity of the damper is obtained by differentiation of a low-pass filtered displacement with a cut-off frequency of 8Hz. In this experimental setup, two parameters can be changed by the user: the velocity of the hydraulic actuator and the input current that is applied to the damper. The spring inside the damper is preloaded for storing potential energy that is sufficient to return the damper; hence, the return stroke is governed mostly by the stored spring energy. In the setup, a PC augmented with a multifunction I/O MF624 card is used in the closed-loop experiments. This card provides an interface between the MATLAB Real Time Toolbox and the system, and sends or receives signals pertaining to the force, displacement and velocity. Also, the system is controlled by command blocks in the MATLAB Real Time Toolbox, which cooperates with an MF624 card where hardware support generated output signals are implemented enabling real-time control over the experiment. All the data were sampled at a sampling frequency of 5 kHz. The control and simulation platforms are implemented in the Matlab/Simulink environment. The associated lists are tabulated in Table 2. First, to examine the mobility of the MR damper, a triangular wave signal is used; it is shown in Fig. 5(a).

The objective of using this signal is to realize a constant velocity. In an actual press, the movement of the press slide is analogous to a sine-wave form, as shown in Fig. 4. If a sine-wave input acts upon the MR damper, the damper force will vary continuously according to the changing velocity of the

Table 2. The equipment used.

Name	Model name	Company
Servo valve	J076-103 J0345	Moog Co.
Servo valve amp	J121-001 J0547	Moog Co.
Load cell	MNT-500L	CAS Co.
LVDT	LTM 300S	GEFRAN
Laser displacement sensor	LB-1201	KEYENCE
Wonder box current device controller kit	RD-3002-03	LORD Co.
Multifunction I/O card (Real Time Toolbox)	MF624	HUMUSOFT

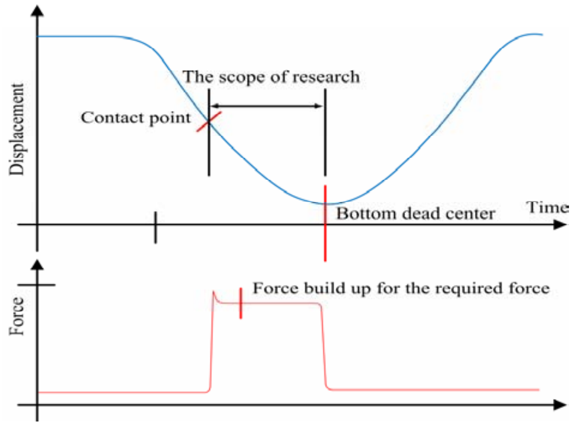


Fig. 4. The displacement of the press slide (upper plot) and a plot of the required drawing force.

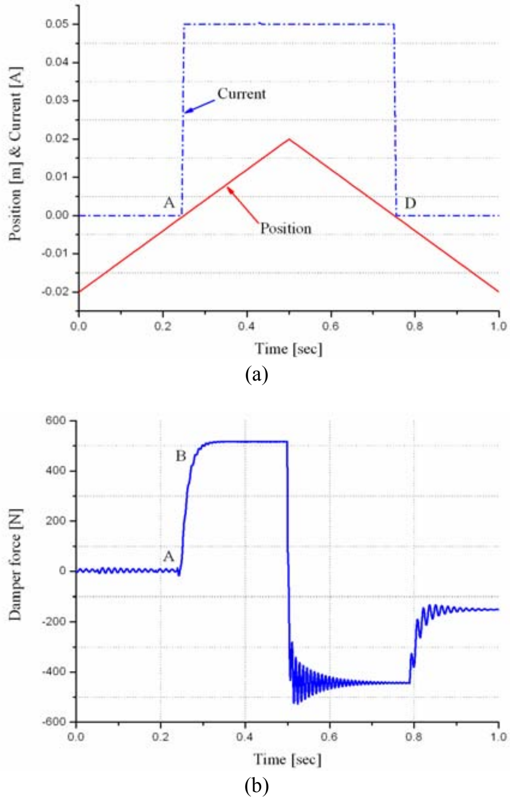


Fig. 5. Triangle input and response: (a) the displacement and applied current and (b) the damper force.

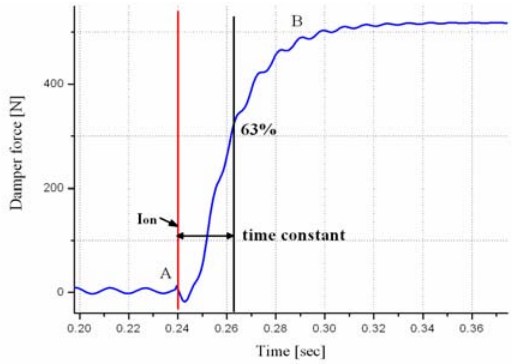


Fig. 6. Damper-force response for each operating current.

input source. Although nonlinearity exists near the bottom dead center, the gradient can be assumed to vary linearly until the slide reaches the bottom position after contact. This test was performed at a relatively fast velocity of 80mm/s and activation current of 0.5A. At point A, a current of 0.5A was applied to the damper and the current fell back again to 0A at point D. The damper-force response is shown in Fig. 5(b). For displaying the force together with the positional curve, the current is represented with a ten-fold reduction. For examining the response time of the damper, the time constants at points A and B are measured for several values of the velocity and the applied current. Fig. 6 shows the results obtained under a velocity of 60mm/s. Fig. 7 shows the response time for each velocity and operating current. After an activation time and change of 63% in the force, the time constant was found to be around 23~150ms. The results mean that the time delay that is associated with the inductance of the coils is not significant. Thus, the effect of the time delay can be disregarded in the control of the BHF in the light of mild variations in drawing presses.

4. Closed-loop force control

The control characteristics of the press system using an MR damper were examined for the problem where the damper force is retained at a constant predefined current while the cylinder rod translates at a constant velocity.

In this study, a conventional proportional-integral-differential (PID) closed-loop control is considered for controlling the MR damper. A procedure similar to that utilized by Chonan et al. [18] was carried out to address the above problem. A schematic of the system with a feedback controller is illustrated in Fig. 8. The input current that is necessary for achieving a reference BHF against the cylinder that is used for mimicking the press slide can be determined from Fig. 2(b). For example, for an input velocity of 20 mm/s, the current that is necessary for achieving a BHF of 300 N is found to be 0.3 A. Also, the friction force, f_0, between the piston rod and the cylinder housing is measured as being about 60N. For use in

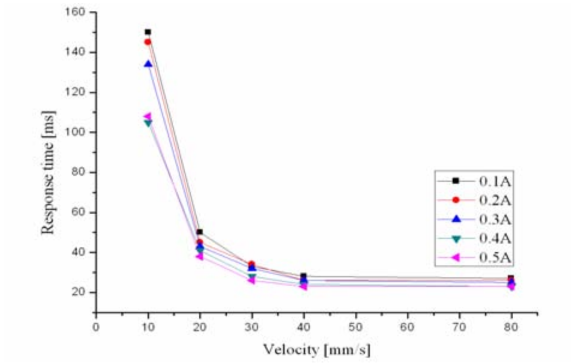


Fig. 7. Response time for each velocity and operating current.

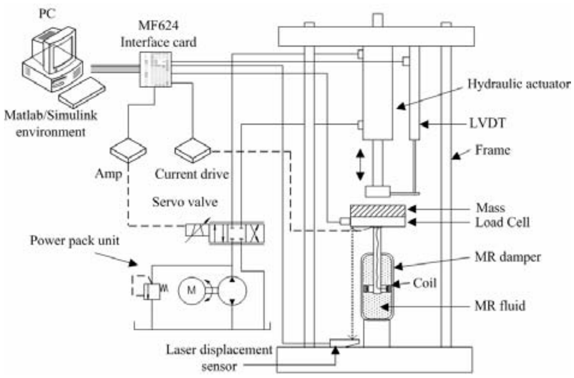


Fig. 8. Schematic of the system with a feedback controller.

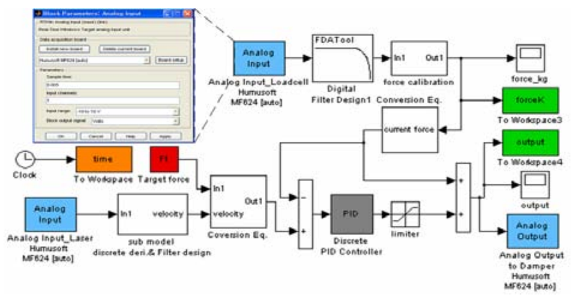


Fig. 9. Closed-loop system controller with the PID control law in MATLAB.

the following control system, the equation for the conversion of the force to the current was formulated beforehand for each individual input velocity. In this study, the control commences after the desired BHF and the input velocity are fixed. The cylinder moves with a constant velocity. A servo-valve was utilized to control the cylinder velocity. Under the specified constant velocity, when the hydraulic cylinder comes into contact with the damper for generating the BHF, the measurement of the damper force, F_m , is commenced by the force sensor. At this moment, the PC collects the measured data at a sampling time of 5ms and then calculates the error, $i_e(=i_t-i_m)$, between the currents corresponding to the reference force F_t , and the measured force, F_m , by using the conversion equation. The calculated error is multiplied by the value of the gain of the PID controller and fed back into the

Table 3. Equations for the conversion of the force to current.

Sliding velocity of the press	Targeted BHF	Corresponding input current
20mm/s	200 [N]	$i(t)=1/857.1\times(F-f_0)$
30mm/s	400 [N]	$i(t)=1/871.8\times(F-f_0)$
40mm/s	600 [N]	$i(t)=1/931.6\times(F-f_0)$

actual current that is applied to the damper. Thus, the current input applied to the damper using the PID controller is given by:

$$i_u=i_m(t)+K_p i_e(t)+K_i\int_0^t i_e(\tau)d\tau+K_d\frac{di_e(t)}{dt}. \tag{4}$$

In Eq. (4), the proportional, integral and derivative gains are represented by K_p , $K_i(=K_p/T_i)$, and $K_d(=K_pT_d)$, respectively. The discrete-time form of Eq. (4) is:

$$\begin{aligned} i_u(t)&=i_m(k-1)+K_p i_e(k)\\ &+K_i\sum_{T=0}^{k} i_e(T)T_s+\frac{K_d}{T_s}[i_e(k)-i_e(k-1)]\,. \end{aligned} \tag{5}$$

The closed-loop system with a PID feedback controller in the MATLAB software is presented in detail in Fig. 9. Here, a limiter (min. current=0.0 and max. current = 1A) was incorporated into the controller to mitigate the effects of saturation. The experiment was undertaken for cylinder velocities of v = 20, 30, and 40 mm/s. The commanded BHF and the corresponding force-current conversion equations are listed in Table 3 for each individual cylinder velocity.

To evaluate the system response, we chose two parameters: the maximum overshoot and the settling time. The maximum overshoot, M_p , and the settling time, t_s , can be defined as:

$$M_p=(F_{\max}-F_t)/F_t\times 100\% \tag{6}$$

$$t_s=e_e/(F_t\times 1.5)\times 100\% \tag{7}$$

where $F_{\max}$ is the maximum force at the peak time, F_t is the target force, and e_e is the absolute error that is integrated from 0 through 1.5s. In the design process, the balance of the response factors, M_p and t_s , is a matter of importance. In the following, we focus on choosing the feedback gains that yield system response with smaller t_s and M_p .

5. Results and discussion

The experimental results with the closed-loop controller are presented in Figs. 10-12. Note that before an experiment is started, the damper is subjected to a constant current for preheating so that the thermal effects are restricted to a minimum. To obtain a well-damped response, the gains were tuned up through a trial-and-error method and Ziegler-Nichols tuning rules. The M_p and t_s obtained for the shaft velocities of v

Table 4. M_p , t_s , and gains for the closed-loop response of Figs. 10-12.

Figure	M_p	t_s	K_p	K_i	K_d
Fig. 10	5.1%	1.0%	1.8	0.6	0.4
Fig. 11	5.5%	1.1%	1.6	0.4	0.3
Fig. 12	4.7%	1.1%	1.2	0.3	0.3

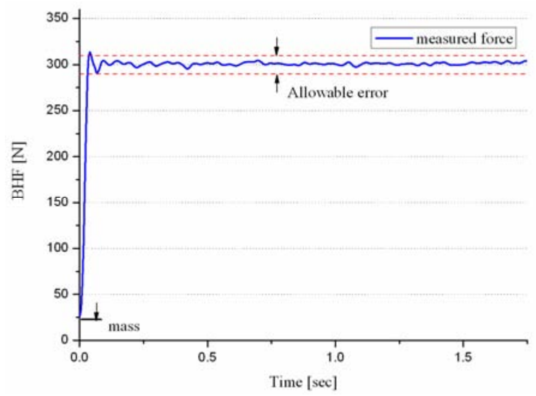


Fig. 10. The time response of the targeted BHF under PID control ($v = 20[mm/s]$ and $F_t = 200[N]$).

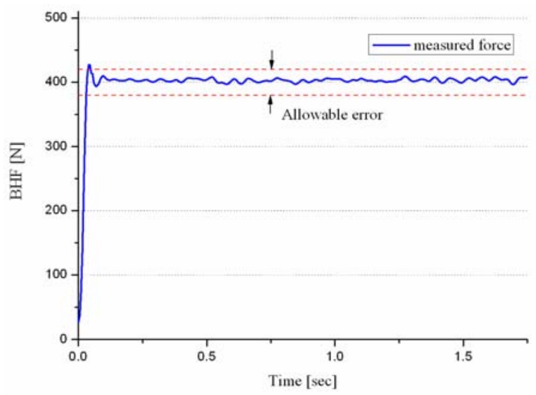


Fig. 11. The time response of the targeted BHF under PID control ($v = 30[mm/s]$ and $F_t = 400[N]$).

= 20, 30, and 40 mm/s are presented in Table 4. Note that the offset forces in these figures are due to the mass that is used. Fig. 10 shows the experimental result when the slide velocity was 20mm/s. The targeted force is successfully controlled to a value of 200N. It is seen that the damper force rapidly converges to the target force without prominent residual oscillations. This is because of the damping effect of the highly viscous MR fluids. The steady-state error is less than 3% and the overshoot is about 5.1%. Other experiments were repeated for two different speeds of the slide shaft. Figs. 11 and 12 show the experimental results for the desired forces of 300N and 400N, respectively, with the parameters of the PID controller listed in Table 4. Parameters for the case where the slide velocity was 20mm/s were tuned because the feedback gains vary depending on the shaft velocity. It is clear that the steady-state error was less than 3%, the overshoot around 4.7-5.5%, and the damper response rapid with a time constant of approximately 20-50ms. All the experimental results described in this study were carried out at a room temperature of about 24 °C . MR dampers can experience temperature changes as a result of heating caused by energy dissipation, but our systems are designed without consideration of this fact. The temperature rise may be affected as a disturbance of the system. Hence, the temperature effect must be considered in applications where the dampers are subject to large temperature variations.

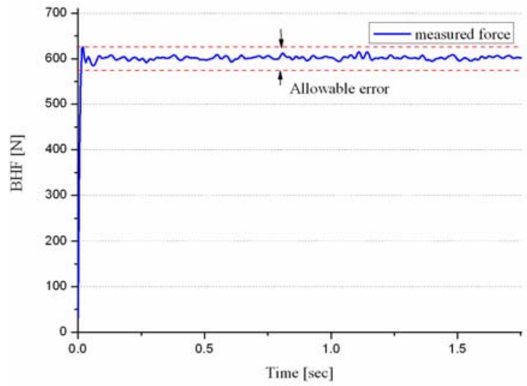


Fig. 12. The time response of the targeted BHF under PID control ($v = 40[mm/s]$ and $F_t = 600[N]$).

Although the temperature effect was not taken into consideration in this work, it can be seen from these results that the control system is minimizing the disturbance effect over small temperature variations caused by repeated tests. Our prime concern in this research endeavor was to maintain a constant BHF.

Other possible modes such as retrogressive and progressive BHF trajectories were not considered. The BHF can be varied to achieve many desired objectives, from preventing the occurrence of tearing or wrinkling. However, these modes require information on BHF curve trajectories in order to improve part quality in sheet metal forming. Since these modes should be accounted for in MR dampers when they are compared to conventional devices, further research work is required to investigate their performance when they are used in press systems where a control system is implemented to modulate the damping force according to estimated the BHF trajectories.

6. Conclusions

An MR damper has been proposed for drawing presses to retain a constant BHF during drawing operations and its feedback control characteristics have been experimentally evaluated. The key technical contribution of this work lies in the fact that the MR damper can be applied to new and viable solutions to effectively control the BHF. Experiments were set up to investigate the feasibility and practicability of MR dampers and understand their potential benefits. First, the force-velocity curve of the MR damper under study was determined experimentally using different current levels. For use in control, the equation of conversion of the damper force to the current source was formulated beforehand for each individual input velocity. The mobility of the damper was examined. It has been shown that the time constant of the damper force following the change in the applied current is around 23~150ms. From the obtained results, the damper mobility is confirmed as being suitable for active control of the blank-holding force in press systems. A closed-loop controller was then applied to the problem where the damper force is retained at a constant prescribed current while the slide piston that is used for mimicking the press slide translates at a constant velocity. The Ziegler-Nichols method was used to select the initial controller parameters, and the ultimate parameters were determined after some tuning necessitated by the characteristics of the damper. The obtained results show that the damper force rapidly settles to the command force without a significant overshoot and oscillation when the feedback gains are selected appropriately. The MR damper proved to be effective in controlling the constant BHF. The results that are presented in this study are still preliminary for the development of press systems that use MR dampers for controlling the BHF. For practical applications, the following studies will be undertaken in the near future:

A study on a declining trajectory of the BHF for increasing the drawing depth.

The development of a single drawing press using an MR damper that should lock the blank holder for avoiding deformation of the formed part.

A study on a robust approach to improve the performance of the system.

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