Control for Straightening Process of Seamless Pipe*

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Seamless pipes have some terminal end bends which must be eliminated before shipment. This paper describes a real-time control system for a press straightening process. The straightening process must account for elastic springback, which is a strong function of the material properties of the pipe. On-line measurements of reactive forces and deflection of the pipe during the early part of the bending process are performed to form an elastic-plastic model in a microcomputer. The final correct die position to yield the desired unloaded shape is predicted using the elastic-plastic model. Through a series of experiments on different materials and different sizes and shapes, the resulting straightness of the pipes was within 0.18 mm, which is sufficiently smaller than product allowance.

Key Words: Press Working, Computer Control, Fluid Power System, Elastic-Plastic Model, Observer, Straightening, Seamless Pipe

1. Introduction

Quality requirement on the seamless pipe is becoming more and more strict along with the deeper drilling of oil and gas wells, and development of oil fields in regions of colder climate.

The terminal end of a seamless pipe produced by a conventional process usually has several millimeter out-of-straightness parameter, which must be eliminated before shipment. The allowance of the out-of-straightness is usually 0.5 mm~1.0 mm and smaller by precise class specifications. Hence a highly effective and accurate straightening of pipe ends is required. The straightening process must include an additional bending to cancel the springback after unloading. As the springback depends on the tube wall thickness and the process condition, the straightening often becomes inaccurate when the springback is predicted by the theoretical bending moment. Although

the springback of each pipe will be accurately predicted when off-line test bending is performed for each pipe⁽¹⁾, the productivity will be lowered.

To construct a highly productive straightening process for the seamless pipe, the authors propose an automatic control system which compensates the springback without requiring preliminary information on the material properties (constituents, outer diameter, wall thickness, Young's modulus, condition of process, etc.). The proposed control system includes an observer to predict the springback using an on-line measured value of reactive force and deflection of the pipe. This method does not require complex procedures such as preserving the data of materials or learning the springback through off-line experiments. For adaptive bending control, there are some investigations on sheets^{(2)~(4)}, but few concerning pipes.

The proposed control process was verified by experiments using a 200-ton hydraulic press machine controlled by an electrohydraulic servovalve. Seamless pipes of different materials, different sizes (STKM13A [JIS G3444] and STO-J [JIS G3439]) and out-of-straightness of 0.1~2.0[mm] were repaired to out-of-straightness within 0.18[mm]. Thus, the experimental result demonstrates the validity of the developed control system.

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Fig. 4 Schematic diagram of control system

because the process proceeds observing the F - δ relationship, the straightening velocity is very slow. Hence the dynamic characteristics of the servovalve and compressibility of oil can be neglected. The equations of continuity for the cylinder chambers are given by

$$Q_1 - A_1 v = 0, Q_2 - A_2 v = 0. \quad (6)$$

Where A_1 , and A_2 are areas of the piston, and v is the velocity of the piston. The equation of motion of the piston is as follows:

$$M \frac{dv}{dt} - Mg + \text{sgn}(v) \cdot F_c + F = A_1 P_1 - A_2 P_2, \quad (7)$$

where M is the mass of the moving parts, F_c is Coulomb friction, and g is the acceleration of gravity.

The velocity of the piston is expressed by

$$d\delta/dt = v, \quad (8)$$

where F is the displacement of the piston.

Figure 5 shows a deflection process model. This is a piecewise linear approximation of the real deflection process explained in Fig. 2. The lines C_1 and C_2 represent the deflection-increasing process by loading, while C_3 represents the deflection-decreasing process by unloading. The yield point is (δ_o, F_o) in Fig. 5. The deflection of the pipe reaches δ_o at time $t = t_o$. The loads on these lines are expressed as follows:

(i) when $0 < \delta < \delta_o$ and $d\delta/dt > 0$ (on C_1),

$$F = E_o \delta, \quad (9\cdot a)$$

(ii) when $\delta > \delta_o$ and $d\delta/dt > 0$ (on C_2),

$$F = F_o + E_1(\delta - \delta_o), \quad (9\cdot b)$$

(iii) when $\delta > \delta_o$ and $d\delta/dt < 0$ (on C_3),

$$F = F_o + E_1(\delta_m - \delta_o) + E_o(\delta - \delta_m). \quad (9\cdot c)$$

The deflection δ is controlled by input current (i), which is calculated from δ_r , $\bar{\delta}$ and other parameters. It takes a rather long time to reach the reference ($\delta_o + \delta_r$) by a proportional compensation, where the error is proportional to the control signal, i.e., the input

$$F = f(\delta)$$

Fig. 5 Load characteristics with hysteresis

current to the servovalve, because the velocity becomes slow when the output approaches the reference. The steady-state error (offset) remains if the control is stopped on the way to the reference. The following simple control algorithm is applied instead of proportional compensation to solve the above problem.

$$\left. \begin{aligned} i &= I \quad (\delta_r - \bar{\delta} > 0) \\ i &= -I \quad (\delta_r - \bar{\delta} \leq 0), (I: \text{const.}) \end{aligned} \right\} \quad (10)$$

$$\bar{\delta} = \delta - F/E_o \quad (11)$$

From Eqs. (4) ~ (6), we have

$$\left. \begin{aligned} P_1 &= P_s - \left(\frac{A_1}{K_v i} \right)^2 v^2, P_2 = \left(\frac{A_2}{K_v i} \right)^2 v^2 \\ \text{for } i &> 0 \\ P_1 &= \left(\frac{A_1}{K_v i} \right)^2 v^2, P_2 = P_s - \left(\frac{A_2}{K_v i} \right)^2 v^2 \\ \text{for } i &< 0. \end{aligned} \right\} \quad (12)$$

When the load is far larger than other forces ($|F| > |Mdv/dt - Mg \pm F_c|$), the relation between F and δ is approximated by

$$F \equiv f(\delta) = A_1 P_1 - A_2 P_2, \delta = f^{-1}(F). \quad (13)$$

A block diagram of the proposed control system is shown in Fig. 6.

3.2 Displacement of the piston

The control system starts from a stationary state where $\delta = 0$ and $i = 0$. Let the reference straightening displacement δ_r ($\delta_r = aU(t)$, $U(t)$: the unit step function) be given at time $t = 0$.

(i) When $0 < \delta < \delta_o$ and $d\delta/dt > 0$, it holds that $\bar{\delta} = 0$ in this interval. Then from Eqs. (9•a), (12) and (13), we have $i = I$, and

$$\frac{d\delta}{dt} = K_v I \sqrt{\frac{A_1 P_s}{A_1^3 + A_2^3}} \sqrt{1 - \frac{E_o \delta}{A_1 P_s}}. \quad (14)$$

Solving this differential equation, the time t_o when $\delta = \delta_o$ is given by

$$t_o = \frac{2}{\alpha} \cdot \frac{x_o}{1 + \sqrt{1 - x_o}}, \quad x_o = \frac{E_o \delta_o}{A_1 P_s}, \quad (15)$$

where

$$\alpha = \frac{K_v I E_o}{A_1 P_s} \sqrt{\frac{A_1 P_s}{A_1^3 + A_2^3}}, \quad x = \frac{E_o \delta}{A_1 P_s}.$$

(ii) When $\delta > \delta_o$ and $d\delta/dt > 0$, it holds that $\bar{\delta} \neq 0$ in this interval. Substituting Eq. (9•b) into the equation in the block diagram of Fig. 6, we have the following expression for $t > t_o$:

$$\begin{aligned} \frac{d\delta}{dt} &= K_v I \sqrt{\frac{A_1 P_s}{A_1^3 + A_2^3}} \\ &\times \sqrt{1 - \frac{E_o \delta_o}{A_1 P_s} - \frac{E_1}{A_1 P_s} (\delta - \delta_o)}. \end{aligned} \quad (16)$$

As δ increases, $\bar{\delta}$ reaches a at time t_m . The time t_m is calculated by the following equation deduced from Eqs. (9•b) and (11):

$$a = (\delta_m - \delta_o)(1 - E_1/E_o) \quad (17)$$

$$\delta_m = \delta_o + \frac{a}{1 - E_1/E_o}, \quad F_m = E_1(\delta_m - \delta_o) + F_o \quad (18)$$

$$\begin{aligned} t_m &= t_o + \frac{2E_o}{\alpha E_1} \\ &\times \{ \sqrt{1 - x_o} - \sqrt{1 - x_m - (E_1/E_o)(x_m - x_o)} \}, \\ x_m &= x_o + \frac{E_o a}{A_1 P_s} \cdot \frac{1}{1 - E_1/E_o}, \end{aligned} \quad (19)$$

where δ_m and F_m are displacement and force at time

Fig. 6 Block diagram of control system

t_m .

(iii) When $\bar{\delta}$ reaches δ_r at time t_i , it holds that $i < 0$.

$$\frac{d\delta}{dt} = -\frac{K_v I}{\sqrt{A_1^3 + A_2^3}} \sqrt{P_s A_2 + E_o \delta - F_m} \quad (20)$$

$$F = \frac{A_1^3 + A_2^3}{K_v^2 I^2} \left(\frac{d\delta}{dt} \right)^2 - A_2 P_s = 0 \quad (21)$$

The time of detachment t_i , and displacement (δ_i) at $t = t_i$ are given by the following expressions deduced from Eqs. (9•a-c) and (20):

$$\delta_i = (1 - E_1/E_o)(\delta_m - \delta_o), \quad (22)$$

$$\begin{aligned} \alpha(t_i - t_m) &= 2 \left\{ \sqrt{\frac{A_2}{A_1} + x_m - \frac{F_m}{A_1 P_s}} \right. \\ &\quad \left. - \sqrt{\frac{A_2}{A_1} + x_i - \frac{F_m}{A_1 P_s}} \right\}, \end{aligned} \quad (23)$$

(iv) When the piston is unloaded at time t_i , the piston begins to move upward, and the motion is expressed by

$$\delta = -\frac{K_v I}{A_1^3 + A_2^3} \sqrt{A_2 P_s} (t - t_i) + \delta_i. \quad (24)$$

The control system gives a stop signal (equivalent to $i = 0$) by a limit switch at a suitable point in region $\delta \leq 0$.

The one cycle of the straightening process is thus completed. To solve Eq. (13) considering the gravity and the coulomb friction ($F - Mg \pm F_c$) is somewhat complicated, but the result is the same, namely, the residual displacement a remains in the pipe.

4. Experimental Results and Discussion

4.1 Experimental system configuration and method of experiment

This section describes experimental results obtained by using a 200-ton hydraulic press machine (see Fig. 4). The system is controlled by a microcomputer. The program language is composed of an assembly language and BASIC.

The actuator is composed of a hydraulic cylinder and a servovalve. The rated flow of the servovalve is $83.3 \times 10^{-6} [\text{m}^3/\text{s}]$ (at $\Delta P = 6.87 [\text{MPa}]$), the rated pressure is $15.7 [\text{MPa}]$, and the rated current is $30 [\text{mA}]$. The internal diameter, rod diameter and stroke of cylinder are $0.4 [\text{m}]$, $0.22 [\text{m}]$ and $0.6 [\text{m}]$, respectively.

The load is calculated by detecting pressures in the head side and the rod side of the hydraulic cylinder chambers. Strain-gauge-type pressure sensors are used. The pressure range of 0 to $49 [\text{MPa}]$ is transformed to voltage range of 0 to $5 [\text{V}]$. The linearity of the sensors for full range is within 0.3 percent. The displacement of the pipe is measured with a dial gauge as illustrated in Fig. 7, and is transformed to analogue voltage.

Since the straightening displacement is minute and the velocity of the cylinder is very slow, the

Tabel 1 Experimental conditions

Work piece No.	Outside diameter d_o mm	Pipe thickness t_w mm	Span length l mm	E_o by theory N/m	Number of samples	Reference displacement δ , mm
(1)	73	5.51	400	10.5×10^7	11	0.10~2.00
(2)	127	4.50	870	5.0	5	0.10~1.00
(3)	73	7.01	400	12.6	10	0.10~2.00
(4)	127	9.00	870	8.9	5	0.10~1.00
(5)	73	9.00	400	14.9	12	0.10~2.00
(6)	73	5.16	400	10.0	10	0.10~2.00

inertial force ($M\dot{v}$) is neglected in the calculation. The equation of motion for the straightening process is reduced to

$$F = A_1 P_1 - A_2 P_2 + Mg - \text{sgn}(v) \cdot F_c \quad (25)$$

where $A_1 = 0.1257[\text{m}^2]$ and $A_2 = 0.877[\text{m}^2]$.

The gravitational load (Mg) and the Coulomb friction ($\text{sgn}(v) \cdot F_c$) was assumed to be constant during low velocity drive. Measured values of the resistive force under the no-load condition are $2.06 \times 10^{-4}[\text{N}]$ during downward motion (loading direction), and $5.96 \times 10^{-4}[\text{N}]$ during upward motion (unloading direction). The load F is calculated at real-time by substituting the above data and measured pressures P_1 and P_2 into Eq. (25). The pressures are averaged for 50 measurements to eliminate noise. The required computing time for one control step is 0.31[s].

As samples for the experiment, straight pipes of 1.2[m] in length were used to easily measure the residual displacement. Pipe materials were STKM13A (JIS G3444, tensile strength: over 38[kgf/mm²]) and STO-J (JIS G3439, tensile strength: over 53[kgf/mm²]). Each sample was exposed to load, and the residual displacement was measured. The experimental conditions are listed in Table 1. Samples No.(2),(4),(5) and (6) are STKM13A, and Nos.(1) and (3) are STO-J.

4.2 Experimental results and discussion

Figures 8 and 9 show the examples of the F - δ relationship for No.(1) and No.(4) samples. In Fig. 8, the reference and the output displacements are 2.0[mm] and 2.0[mm], respectively. In Fig. 9, they are 0.5[mm] and 0.62[mm], respectively. These results justify the elastic-plastic model introduced in Section 2. Reference values and measured values are compared in Figs. 10 and 11. The reference is expressed as 45-degree inclined lines in each picture. Measured values are slightly larger than those of the respective reference; however, they are sufficiently close to the reference and satisfy production tolerance. Absolute differences between reference and measured values are not dependent upon the magnitude of the reference. The standard deviation of the difference (σ) for

Fig. 7 Schematic diagram of displacement measurement

Fig. 8 Example of straightening experiment of sample No. (1)

Fig. 9 Example of straightening experiment of sample No. (4)

Fig. 10 Experimental result of sample No. (2)

all experimental results was 0.0844[mm], and the maximum difference between the measured and the reference values ($\max|a - \delta_i|$) was 0.18[mm]. These values are well within the production tolerance of steel pipe straightness.

The experimental and the theoretical values of E_0 are also compared in Figs. 10 and 11. Although the measured value varies with the support condition, the difference between the reference and the measured values ($a - \delta_i$) is always stable and small because E_0 is measured in real-time in the control loop. The average value of the measured E_0 of each material is shown in Fig. 12. The result coincides well with the predicted value. Hence the theoretical value of E_0 can be used to estimate the reaction force and the load on the hydraulic system. The experimental values of E_0 in Figs. 10 and 11 scatter around the theoretical value. As the scatter is random, it may be supposed that its causes were secondary factors such as small elliptic deformation of the pipe cross section and clamp condition of the dies. Although the gradients of the F - δ relationship at loading and at unloading are not exactly parallel, the experimental result revealed that it had little influence on the accuracy of the straightening process developed in this paper.

5. Conclusions

An automatic control method for elimination of out-of-straightness appearing at terminal ends of a seamless pipe was developed. The developed automatic control system was able to compensate the springback independent of material characteristics. Through a series of experiments on seamless pipes of different materials and different sizes and shapes by using a 200-ton hydraulic press machine, the resulting straightness of the pipes was within 0.18[mm]. Thus

Fig. 11 Experimental result of sample No. (5)

Fig. 12 Comparison of E_0 by theory and experiment

the good performance of the control system was verified. This control method can also be applied to sheets for small adaptive bending.

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