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SWITCHING FROM VELOCITY TO FORCE CONTROL FOR THE ELECTRO-HYDRAULIC SERVOSYSTEM BASED ON LPV SYSTEM MODELING

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Abstract: In the hydraulic system, it is often that switch the control mode over from one to the other, such as from position to velocity or from velocity to force. In this paper, propose a flow calculation formula for the flow control valve in order to have an LPV system representation. Then, design gain scheduled controllers for the velocity and the force individually. During a switching mode, a control is generated by adding two controller outputs with appropriate ratios. Usefulness of this approach is shown by the experiment results, which are obtained from Injection molding machine application.

Keywords: Injection molding, Electro-hydraulic systems, LPV System, H-infinity, Switching algorithms

1. INTRODUCTION

Hydraulic control system is used in various industrial applications for the power in size, high durability. In most cases, the electro-hydraulic servosystem is applied to obtain the high reproducibility and the fine dynamic performance of the position, velocity or force control in the hydraulic system. The hydraulic plant has many nonlinear factors and components. For example, asymmetric cylinder, mechanical friction, deadband, complex flowpassage relates to hydraulic dynamics, the effective control flow which depends on load condition and hysteresis. In addition to the above, plant parameters are not constant, such as the bulk modulus depends on the containing air quantity or viscosity varies according to the temperature. Also, it is often in indistrial hydraulic applications that a control mode swicthes over from one to the other sequentially, such as from the position to pressure or from the velocity to force. These factors make a controller design complicated. However, because of the electro-hydraulic servosystem capability, a number of studies have been done in designing the controller. Approaches reported recently are adaptive control(Bobrow and Lum, 1996),(Plummer and Vaughan, 1996)and sliding mode control(Ha *et al.*, 1995). There, controller structure becomes complicated and it is not easy to realize the smooth operation and the fast response. The robust control design by H_∞ framework (Tunay *et al.*, 2001) is also applied. The controller design approach, which is based on the linear model, make the closed-loop system stable locally. When the load condition changes significantly, there is a limitation in appling this approch. Here, we examine gain scheduling control of the electro-hydraulic servosystem for the velocity and force control in order to guarantee stability and performance under the significant plant parameter variation.

It is important that we take a varying load condition and a complex flow characteristic of the

Fig. 1. Injection Molding Machine appearance

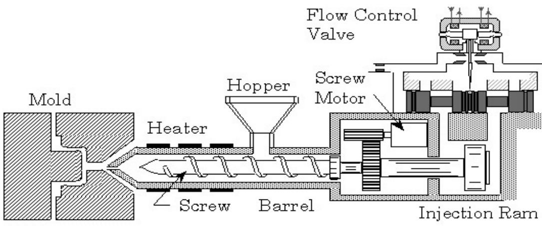

Fig. 2. Test equipment

control valve around the null in consideration to construct a plant model. One of the reasons to make the situation difficult is the discontinuity of the control flow calculation formula around the null. In this paper, we propose a formula, which interpolates the turbulent and the laminar flow in the flow control valve so that the linear model becomes continuous at the boundary for the precise force control. Then, we compose the linear plant model as an LPV (linear parameter varying) system with a scheduling parameter which depends on load force. Based on LPV plant models, we design gain scheduling controllers for the velocity and force. One of the contributions of this paper is that present the way to design the gain scheduled controller of the electro-hydraulic servosystem. The other contribution is that we study switching behavior from velocity to force control mode with gain scheduled controllers applying to the Injection molding machine. Here, we attempt to add up the outputs from the velocity and the force controller according to the weighting coefficient which is determined by the ratio of actual force and the force reference value.

2. INJECTION MOLDING PROCESS

In the injection molding process, there is an injection velocity control and a holding pressure control mode. The velocity profile is generated with respect to the mold shape so that the velocity between melt plastic and mold surface becomes constant. The force control of the holding pressure mode makes the plastic stress uniform in order to minimize the deformation of the product. The electro-hydraulic servosystem is adapted to both of the velocity and force control for the high power, fast response and fine reproducibity. The

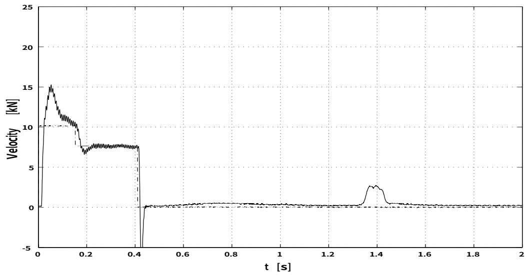

Fig. 3. Velocity behavior (Case 1)

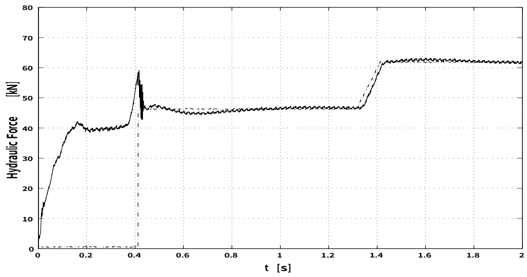

Fig. 4. Force behavior (Case 1)

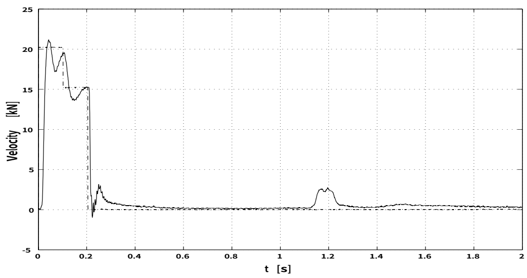

Fig. 5. Velocity behavior (Case 2)

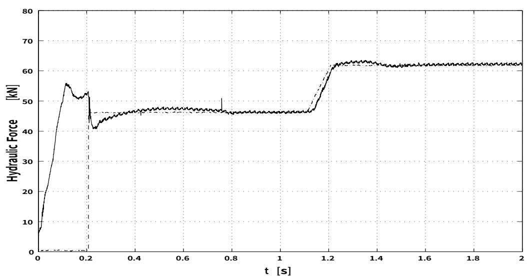

Fig. 6. Force behavior (Case 2)

appearance of the Injection molding machine is shown in Figure 1 and the structure scheme is shown in Figure 2. Main components are a mold, heaters, a screw, a hydraulic injection cylinder, a flow control valve and transducers for the velocity and force. The purpose to design the gain scheduled controller here, is to have stable and good performance under significant palnt parameter variation, caused by the load force change. Also, the switching behavior from velocity to force is important. To make it smooth, we generate a control by adding up two controller outputs, during the transition from velocity to force control. Figures 3,4 and figures 5,6 show the typical behaviors, which we see often, when switch the controlled variable from velocity to force without any measures to make the transition smooth. In figures 3,4, switching occurs at 7.5 cm/s velocity. Figure 3 shows a rapid velocity change and an revrse direction velocity at 0.4 sec. This means that the injection ram moves to backward. This phenomenon should be avoided in the process. In Figure 4, there is a pressure peak at the just before switching happens. In figures 5, 6, switching takes place at 15 cm/s velocity. In this case, the reverse velocity behavior and pressure peak are small, but still exist.

3. FLOW CALCULATION FORMULAS

In the flow control valve, which has a sleeve and spool, there are two typical flow conditions, called the turbulent flow at the metering orifice opening and laminar flow in the clearance between sleeve and spool. Now, think about the flow at the metering orifice A out of four orifices in Figure 7, as an example. There are commonly used flow calculation formulas (1), (3), with lap l_a and clearance c_r, for each conditions. However, these two equations are not continuous at the boundary. Hence, adopt equation (2) in order to interpolate flow and derivation of the turbulent and the laminar conditions.

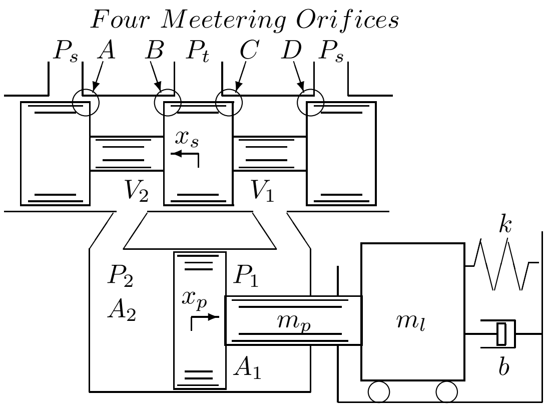

Fig. 7. Configuration of the plant

Then, continuous flow in whole operating range, is given as follows

$$q_{2int} = K_t\sqrt{(x_s - l_a)^2 + c_r^2}\sqrt{P_s - P_2} \quad ... \; l_a \le x_s \quad (1)$$

$$q_{2ini} = \left(\frac{3}{K_l}\right)^3 \left(\frac{K_t c_r}{4}\right)^4 \frac{(x_s - l_a)^3}{P_s - P_2} + K_t c_r\sqrt{P_s - P_2} \quad ... \; l_a + x_{sa} \le x_s < l_a \quad (2)$$

$$q_{2inl} = K_l \frac{P_s - P_2}{-(x_s - l_a)} \quad ... \; x_s < l_a + x_{sa} \quad (3)$$

here, $x_{sa} = -4C_r\sqrt{Ps - P_2}/(3K_t K_l)$

According to these equations (1),(2) and (3), the flow from control valve highly depends on load pressure (or force) and is nonlinear. Because of such a characteristics, a fixed controller at the one operating condition can not satisfy the performances in the over all operating range. $P_2(t)$, $P_1(t)$ are bore and rod pressure, A_2, A_1 are effective piston areas of the cylinder. m_p and m_l are masses, $x_p(t)$ is piston displacement. K_t, K_l are coefficients of the turburant and the laminar flow. Linearization of the equations (1),(2) and (3) at the arbitrary operating point (x_{s0}, P_{20}). We have

$$\delta q_{2int} = K_t\left(\frac{x_{s0} - l_a}{\sqrt{(x_{s0} - l_a)^2 + c_r^2}}\sqrt{P_s - P_{20}}\delta x_s - \frac{\sqrt{(x_{s0} - l_a)^2 + c_r^2}}{2\sqrt{(P_s - P_{20})}}\delta P_2\right) \quad (4)$$

$$\delta q_{2ini} = \left(\frac{3}{K_l}\right)^3 \left(\frac{K_t c_r}{4}\right)^4 \frac{3(x_{s0} - l_a)^2}{P_s - P_{20}}\delta x_s + \left(\left(\frac{3}{K_l}\right)^3 \times \left(\frac{K_t c_r}{4}\right)^4 \frac{(x_{s0} - l_a)^3}{P_s - P_{20}} - \frac{K_t c_r}{2\sqrt{P_s - P_{20}}}\right)\delta P_2 \quad (5)$$

$$\delta q_{2inl} = K_l\left(\frac{P_s - P_{20}}{-(x_{s0} - l_a)}\delta x_s - \frac{1}{-(x_{s0} - l_a)}\delta P_2\right) \quad (6)$$

For the simple presentation such as (7), choose the corresponding equation from (4), (5) or (6) according to spool displacement $x_s(t)$.

$$\delta q_{2in} = K_{2inxs}\delta x_s + K_{2inp2}\delta P_2 \quad (7)$$

The same way as above, derivative equations of the metering orifice B, C and D are described below, respectively.

$$\delta q_{2out} = K_{2outxs}\delta x_s + K_{2outp2}\delta P_2 \quad (8)$$

$$\delta q_{1out} = K_{1outxs}\delta x_s + K_{1outp1}\delta P_1 \quad (9)$$

$$\delta q_{1in} = K_{1inxs}\delta x_s + K_{1inp1}\delta P_1 \quad (10)$$

4. MODELING

4.1 *Linear System*

Figure 8 presents the block diagram of the linear system from the input δv_{sig}, which is applied to the flow control valve, to the controlled variable hydraulic force V_{F_h} and piston velocity $V_{x_{pv}}$. v_{sig} is the control. r_{sig} is the rated signal and r_{str} is the rated spool displacement. w_v and ζ_v represents the control valve dynamics as the second order transfer function. The pressure in the actuator chamber is calculated from the hydraulic compressibility, called as bulk modulus β, and the effective volume change caused by flow in and out from the chamber, plus piston displacement. A mass, an equivalent viscous resistance and a spring compose the load. In the actual injection process, the viscous resistance and spring rate change under the various operating conditions. Here, suppose that they are constant in the velocity and force control mode. Here, take state vector δx as

$$\delta x = (\delta x_{sv}, \; \delta x_s, \; \delta P_2, \; \delta P_1, \; \delta x_{pv}, \; \delta x_p)^T \quad (11)$$

From the linear block diagram in Figure 8, the linear state space equation is represented in equation (12) and (13).

$$\delta\dot{x} = \begin{pmatrix} -2\zeta_v\omega_v & -\omega_v^2 & 0 & 0 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & a_p(3,2) & a_p(3,3) & 0 & a_p(3,5) & 0 \\ 0 & a_p(4,2) & 0 & a_p(4,4) & a_p(4,5) & 0 \\ 0 & 0 & \dfrac{10^4 A_2}{mp+m_l} & -\dfrac{10^4 A_1}{mp+m_l} & \dfrac{10^2 b}{mp+m_l} & \dfrac{10^2 k}{mp+m_l} \\ 0 & 0 & 0 & 0 & 1 & 0 \end{pmatrix} \delta x + \begin{pmatrix} \dfrac{r_{str}}{r_{sig}} \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{pmatrix} \delta v_{sig} \quad (12)$$

$$\begin{pmatrix} V_{Fh} \\ V_{x_{pv}} \end{pmatrix} = \begin{pmatrix} 0 & 0 & \dfrac{A_2}{457.78} & -\dfrac{A_1}{457.78} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0.1 & 0 \end{pmatrix} \delta x \quad (13)$$

Some of the matrix elements, such as $a_p(3,2)$, are not constant. Here,

$$a_p(3,2) = \beta \frac{K_{2inxs} - K_{2outxs}}{V_2 + A_2(L_n + x_{p0})}$$
$$a_p(3,3) = \beta \frac{K_{2inP_2} - K_{2outP_2}}{V_2 + A_2(L_n + x_{p0})}$$
$$a_p(3,5) = -\beta \frac{A_2}{V_2 + A_2(L_n + x_{p0})}$$
$$a_p(4,2) = \beta \frac{K_{1inxs} - K_{1outxs}}{V_1 + A_1(L_n - x_{p0})}$$
$$a_p(4,4) = \beta \frac{K_{1inP_1} - K_{1outP_1}}{V_1 + A_1(L_n - x_{p0})}$$
$$a_p(4,5) = \beta \frac{A_1}{V_1 + A_1(L_n - x_{p0})}$$

L_n is a half with total piston stroke.

4.2 *LPV System*

As mentioned above, some of the elements in the state space equation vary according to the operating conditions. To have linear state space equation, we adopt a LPV system presentation, which has the parameter that is the function of the load force. Equation 12 has the form as

$$\delta\dot{x} = \frac{\partial}{\partial x} f(x_0)\delta x + B_p \delta v_{sig} \quad (14)$$
$$\delta y = C_p \delta x \quad (15)$$

Two matrices B_p and C_p are constant and all elements in these matrices are decided based on the mechanical specifications. x_0 is a state at the equilibrium point. But, it is difficult to have the solution of x_0 from the implicit function of $f(x_0)+B_p v_{sig0} = 0$. Now, suppose that x_0 is given and then y_0 is calculated by $y_0 = C_p x_0$. Addition to this, suppose that the scheduling parameter θ is a smooth function of the y_0, such as $\theta = \varphi(y_0)$. The linear state space equation, which depends on the scheduling parameter θ, represents the plant behavior in the neighborhood of the equilibrium x_0 (Uchida, 1995), (Rugh and Shamma, 2000). By the way, $a_p(3,2)$, $a_p(4,4)$ and so on, are decided when x_{s0}, P_{20}, P_{10} and x_{p0} are fixed. But, as explained already, it is hard to decide these values from the related equations.

Here, we use a value of the variable which is obtained from the following simulation. Apply PI controller to close the velocity loop in the nonlinear plant model. Then, apply relatively slow enough ramp velocity reference signal ($\delta v_{sig} = 0$) so that we are able to assume all state variables are close enough to the equilibrium states. Now, we have a set of the equilibrium points for the specified operating range in both velocity control. Using these values, whose could be considered as the equilibrium set, figure out the elements of equation (12), such as $a_p(3,2)$ so on, with respect to corresponding scheduling parameter that is described as equation (19) or (20). In the case of force plant model, the way to have PLV representation is same as mentioned above. Then, we represent these elelments by the polynomial approximation, as follows

$$a_p(\theta) = a_{p0} + a_{p1}\theta + a_{p2}\theta^2 + a_{p3}\theta^3 \quad (16)$$

The state space equations (12), (13) are rewrited as equations (17), (18) with the scheduling parameter.

$$\delta\dot{x}_p = A_p(\theta)\delta x_p + B_p \delta v_{sig} \quad (17)$$
$$\delta y_p = C_p \delta x_p \quad (18)$$

Considering the flow characteristic which depends on $\sqrt{\triangle P}$, define the scheduling parameter θ_v for the velocity and θ_n for the force control as

$$\theta_v = \frac{\dfrac{1}{\sqrt{1 - \frac{F_x}{F_{max}}}} - 1}{\dfrac{1}{\sqrt{1 - \frac{F_{v \cdot rated}}{F_{max}}}} - 1}, \ [0 \le \theta_v \le 1] \quad (19)$$

$$\theta_n = \frac{\dfrac{1}{\sqrt{1 - \frac{F_x}{F_{max}}}} - 1}{\dfrac{1}{\sqrt{1 - \frac{F_{n \cdot rated}}{F_{max}}}} - 1}, \ [0 \le \theta_n \le 1] \quad (20)$$

F_{max} is maximum force, $F_{v \cdot rated}$ and $F_{n \cdot rated}$ are the rated force in the velocity and force control mode, F_x is actual force which is measured as V_{F_h}.

5. GAIN SCHEDULED CONTROLLERS

Designing gain scheduled controllers for the volocity and force, we take following issues in consideration. In the velocity loop, the rise time is within 15ms to the step reference and minimizes steady state error. In the force controller design, the force follows to the ramp reference signal, which reaches to the rated force with 15 ms and zero steady state error. In order to construct the generalized plant for the H_∞ controller design framework, two weighting functions, $W_s(s)$ for the sensitivity

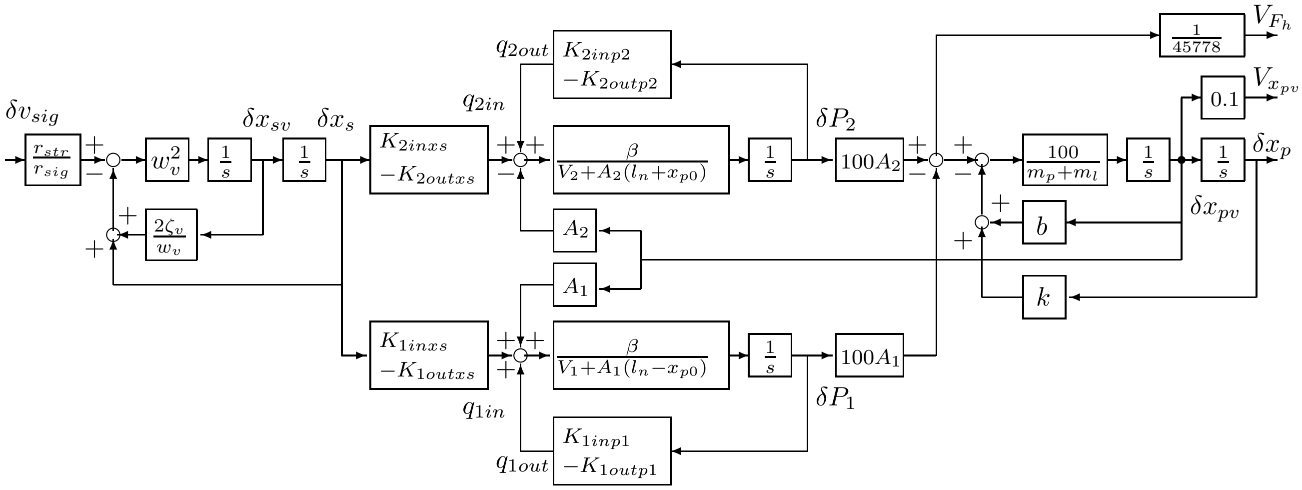

Fig. 8. Block diagram of the linearlized plant

function and $W_a(s)$ for the additive uncertainty at the input of the plant, are specified after the several try and error. For the velocity controller, we use

$$W_{sv}(s) = \frac{0.0025s + 68.75}{s + 0.005}, \quad W_{av}(s) = \frac{5(s+1)}{s + 375}$$

and at the force controller design

$$W_{sn}(s) = \frac{0.4s + 5}{s + 0.01}, \quad W_{an}(s) = \frac{25(s+1)}{s + 75}$$

Beside these weighting functions, $(0.1s + 0.015)/s$ is added in series to the plant in the velocity control. Also, $(50s + 1)/(s + 0.01)$ is in series to force plant to improve the response. When solve H_∞ controller design problem with LMI formulation, the two positive definite matrices, in many cases described as $\mathcal{X}$ and $\mathcal{Y}$, also let the function of the scheduling parameter, such as

$$\mathcal{X}_v(\theta_v) = \mathcal{X}_{v0} + \mathcal{X}_{v1}\theta_v + \mathcal{X}_{v2}{\theta_v}^2 + \mathcal{X}_{v3}{\theta_v}^3$$

so that minimize the conservatives of the controller. As the results, the generalized plant becomes function of the continuous scheduling parameter and has to solve infinite number of LMIs. In order to reduce this problem to finite number of constraints, a technique that proposed by Azuma (Azuma *et al.*, 2000) to construct a convex hull is introduced. For the more details of the gain scheduled controller for the velocity and force control, see (Sugiyama *et al.*, 2000), (Sugiyama and Uchida, 2001) and (T. Sugiyama and K. Uchida, 2002).

6. SWITCHING SCHEME AND TEST RESULTS

In this experiment, the rated velocity is 200 mm/s, the rated force is $F_{n \cdot rated} = 160$ kN and the maximum force $F_{max} = 183$ kN are defined by the specific product and mold capability. The actual load force in the velocity control mode becomes 40 to 50 kN , which is used in calculation of the scheduling parameter θ_v. In the force control mode, the controlled force comes up to 60 kN in the process. The way to add up the outputs form the velocity and force controller is described in Figure 9.

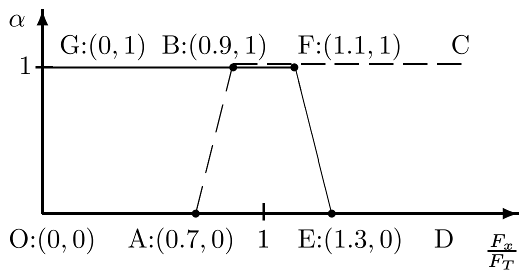

Fig. 9. Explanation of the switching method

Summing up control depends on the force level and whether it overs a set point or not. If the force F_x over the set force level F_r when the switching occurs. In this case, follows the line "D-E-F-G" in Figure 9. Moreover, set the velocity reference as zero. And, the force crosses the point "F", comes from the direction of "E", switch to the force control completely. In the other case, it means that the force level is below to a set point, follows the line "O-A-B-C" and keep to use the velocity reference as it is. In the range "A" to "B" or "E" to "F", we use the add up control. Let's say, u_v is the velocity controller output and u_f is the one of the force controller. The switching control u_t is calculated as

$$u_t = (1 - \alpha)u_v + \alpha u_f \tag{21}$$

In Figure 10,11 and 12,13 show the transitional behaviors that occurs in 7.5 and 15 cm/s velocity. At the beginning of the velocity control, the big overshoot or fluctuation is observed. This is caused by dead time that we do not consider in gain scheduled controller design, so far. But, there is not significant influence on switching operation. Looking at the switchig behaviors, obtain

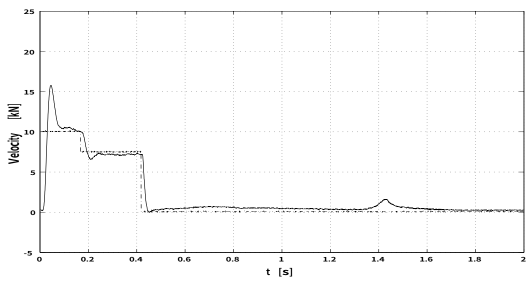

Fig. 10. Velocity behavior with control (Case 3)

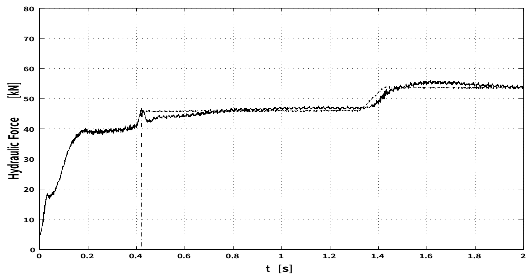

Fig. 11. Force behavior with control (Case 3)

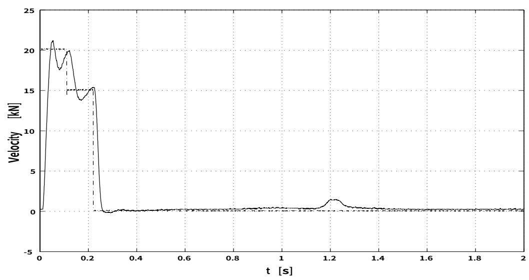

Fig. 12. Velocity behavior with control (Case 4)

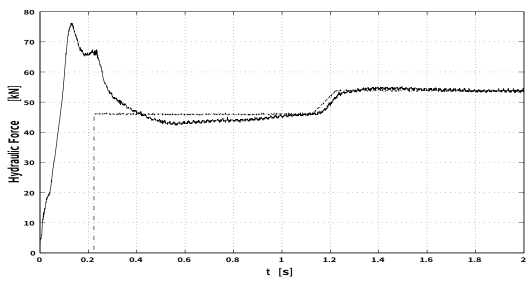

Fig. 13. Force behavior with control (Case 4)

reasonable results by the switching control principle, mentioned in this paper. It is obvious that the velocity comes down near to zero level very smoothly in the both of cases. Force is changing quite smooth, too. The smooth transition is achieved.

7. CONCLUSION

In this paper, we propose the flow calculation formula to have continues flow between the turbulent and laminar flow so that compose the linear plant model with a scheduling parameter. Then, we obtained the values of the system variables by the simulation in order to design the gain scheduled controller for the velocity and force control based on LPV system. The usefulness of the controller design method, described here, is confirmed to the electro-hydraulic servosystem. Also, the way described here to switch from velocity to force makes the transition behavior smooth successfully.

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