Robust Velocity Control of Water Hydraulic Servomotor System via Linearization Approach

Kazuhisa Ito

Abstract—**Water hydraulic servo system is attractive technology for safety environment. This paper deals with water hydraulic servomotor system. To overcome the highly nonlinearity of flow equation, we linearize and parametrize it. Considering the effect of parameter perturbation, *e.g.* load fluctuation and friction parameters, a method of Lyapunov-based recursive design is applied to construct a robust rotational velocity controller. The effectiveness of the proposed controller is examined numerically.**

Keywords—**Water hydraulic servomotor, Rotational velocity control, Nonlinear system, Parameter uncertainty, Robust control, Lyapunov-based recursive design**

I. Nomenclature

c_e : coefficient of viscous friction
c_f : coefficient of Coulomb friction
c_s : coefficient of motor leacage
c_d : coefficient of discharge
I : moment of inertia
D : motor displacement
ω : angular velocity
x : valve spool displacement
μ : viscosity of water
ρ : density of water
w : valve port width
V_0 : average volume of piping between valve and motor
K : bulk modulus of water
P_S : supply pressure
K_v : valve input voltage gain
τ : valve time constant

II. Introduction

In A last decade, water hydraulic applications have increased and continues to increase. Water as a pressure medium gives many benefits compared to mineral oil, including easy maintenance, good availability and so forth. In the early stage of development, many literatures reported the comparison between water and oil on physical and dynamical properties in use[1]-[5] while these days' reports give us the results of control techniques of water hydraulic devices[6]-[12]. Linjima *et.al.* studied the position displacement control of water hydraulic cylinder taking advantage of the low compressibility of water and low-cost low-pressure valves[6]-[9]. In [10], the position displacement and velocity control of water hydraulic cylinder are realized introducing the profile generator which generates smooth control signal to attenuate a large overshoot. The rotational angle control and the speed control of water hydraulic servomotor are also tested with conventional PID controller in [11],[12]. These results are mainly based on the conventional technique for mineral oil hydraulic system and there is a need to be discussed to consider the properties of water hydraulic devices. These days some nonlinear robust control methods are applied to hydraulic devices, however, almost all such papers are on a control of the mineral oil hydraulic systems. An adaptive controller, which compensates uncertainties of inertia load and some physical parameters, is also proposed in [13]. A feedback linearization method and backstepping design of mineral oil hydraulic motor are proposed in [14] while the parameter uncertainty treatment are open question in the paper.

Recently several papers meet these requests with the application of modern control theory. In [15], the controller for water hydraulic pressing machine is constructed by applying the H_∞ mixed-sensitivity problem to compensate parameter uncertainties for the spool clearance and the coefficient of flow rate of servo valve and the bulk modulus of water.

In this paper, we present a development of a design of a robust rotational velocity controller of water hydraulic servomotor for the uncertain parameters and the load fluctuation. And we linearize the flow equation which contains nonlinear state variables to apply a robust control theory. To be more precise, we use the Lyapunov-based recursive design method to construct the virtual input and Lyapunov-like function to ensure stability of each subsystem compensating the effect of parameter uncertainty. Generally speaking, we can achieve the exponential stability for such error system by using the sign function to design a controller. The resulting controller, however, generates non-smooth signal, therefore it requires large discontinuous control input which may occur undesirable behaviour for system. The proposed controller ensures to drive the state error to the set specified by the designer *a priori* which contains the origin without using sign function. The ultimate set to which error converges can be chosen arbitrary small. This is called the uniformly ultimately bounded stability (UUBS)[16] of the system. This scheme can be extended to the rotational angle control system.

In Section 3, first of all, we derive mathematical model of water hydraulic servomotor by linearizing the nonlinear flow equation which contains control variable. Then we define the problem to be solved. In Section 4, we construct a robust controller for uncertain parameters and load fluctuation with Lyapunov-based recursive design. We show the simulation results of the proposed controller in Section 5. Finally we summarize our results.

K.Ito is with the Fluids Engineering Laboratory, Mechanical Engineering Department, Faculty of Science and Technology, Sophia University, Tokyo, Japan. E-mail: kazu-ito@me.sophia.ac.jp .

III. Problem Formulation

The schematic diagram of the water hydraulic servo motor system is shown in Fig.1. Under the following assumptions, the mathematical model of the servo system was derived:

1. There are no external leakage from the servo valve and the motor.
2. There are no overlap between the valve body and the spool.
3. Spool displacement is linear to the input voltage applied to the servo amplifier.
4. The density and the viscosity of the medium are dependent on the temperature, but are assumed to be constant during the movement of the plant.
5. The supply pressure from the power unit is constant.
6. The motor displacement is constant.

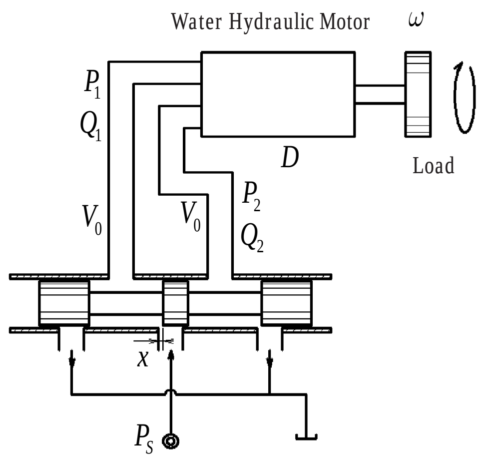

Fig. 1. Diagram of water hydraulic servomotor

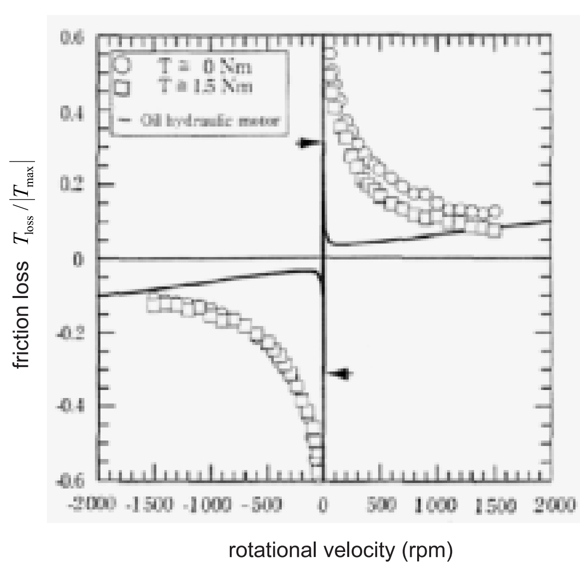

Fig. 2. torque loss of water hydraulic motor

In general, the stark differences between mineral oil hydraulic system and water hydraulic one are the strong mechanical friction (Coulomb friction) and impulsive surge pressure. The former depends on lower viscosity while the latter on higher bulk modulus of water. Low viscosity requires strong liquid seal to prevent the leakage and this leads to the larger starting torque. Fig.2 shows the torque loss of water hydraulic motor under load torque. We see from this figure that the Coulomb friction of water hydraulic motor is very large comparing to the mineral oil hydraulic motor.

From Fig.1, the equation of motion of the water hydraulic servomotor are expressed as follows.

$$\begin{aligned} \dot{\omega} &= -c_e \tfrac{D\mu}{2\pi I}\omega + (1-c_f)\tfrac{D}{2\pi I}P_L \\ \dot{P}_L &= -\tfrac{DK}{\pi V_0}\omega - c_s P_L + \tfrac{2K}{V_0}Q_L \end{aligned} \tag{1}$$

where $P_L (= P_1 - P_2)$ is load pressure. The equations for load discharge Q_L and the dynamics of servo valve are given by

$$Q_L = c_d w x \sqrt{\tfrac{P_S - \mathrm{sgn}(x) P_L}{\rho}} \tag{2}$$

$$\dot{x} = -\frac{1}{\tau}x + \frac{K_v}{\tau}u \tag{3}$$

where u is the input voltage to the servo valve. By linearization of the flow equation (2), we have

$$Q_L = k_q x - k_c P_L \tag{4}$$

where parameters k_q, k_c are obtained experimentally around an operation point of the system. We introduce the scaling factor S_p, S_x because the rotational speed ω, the load pressure P_L and valve spool displacement x are completely different order :

$$\bar{P}_L = \tfrac{P_L}{S_p},\ \bar{x} = \tfrac{x}{S_x} \tag{5}$$

With scaling factors and (4), we have linearized system equation

$$\begin{aligned} \dot{\omega} &= -c_e \tfrac{D\mu}{2\pi I}\omega + (1-c_f)\tfrac{DS_p}{2\pi I}\bar{P}_L \\ \dot{\bar{P}}_L &= -\tfrac{DK}{S_p \pi V_0}\omega - \left(c_s + k_c \tfrac{2K}{V_0}\right)\bar{P}_L + \tfrac{2Kk_q S_x}{S_p V_0}\bar{x} \\ \dot{\bar{x}} &= -\tfrac{\bar{x}}{\tau} + \tfrac{K_v}{S_x \tau}u \end{aligned} \tag{6}$$

The control objective is to maintain the rotational velocity ω to operation point ω_e specified by the designer. It is shown by straightforward that the equilibrium point $(\omega_e, \bar{P}_{Le}, \bar{x}_e)$ of system is

$$\begin{aligned} \omega_e &: \text{ arbitrary specified by designer} \\ \bar{P}_{Le} &= \frac{\bar{c}_e \mu}{S_p(1-\bar{c}_f)}\omega_e \\ \bar{x}_e &= \frac{\tfrac{DK}{S_p\pi V_0}\omega_e + \left(\bar{c}_s + \tfrac{2K\bar{k}_c}{V_0}\right)\bar{P}_{Le}}{\tfrac{2K\bar{k}_q S_x}{S_p V_0}} \end{aligned} \tag{7}$$

while the control input u_e to keep ω to the equilibrium point ω_e is

$$u_e = \tfrac{S_x}{K_v}\bar{x}_e \tag{8}$$

where the $\bar{c}_e, \bar{c}_f, \bar{c}_s, \bar{k}_q, \bar{k}_c$ stand for the nominal values of these parameters:

$$\begin{aligned} c_e &= \bar{c}_e + \Delta c_e, & 0 < c_e^- \leq \bar{c}_e + \Delta c_e \leq c_e^+ \\ c_f &= \bar{c}_f + \Delta c_f, & 0 < c_f^- \leq \bar{c}_f + \Delta c_f \leq c_f^+ \\ c_s &= \bar{c}_s + \Delta c_s, & 0 < c_s^- \leq \bar{c}_s + \Delta c_s \leq c_s^+ \\ k_q &= \bar{k}_q + \Delta k_q, & 0 < k_q^- \leq \bar{k}_q + \Delta k_q \leq k_q^+ \\ k_c &= \bar{k}_c + \Delta k_c, & 0 < k_c^- \leq \bar{k}_c + \Delta k_c \leq k_c^+ \\ I &= I_0 + \Delta I, & 0 < l_1 \leq I_0 + \Delta I \leq l_2 \end{aligned} \tag{9}$$

Note that it is not necessary to have an information of load inertia to get the equilibrium point x_e.

With (8), we introduce following state variables

$$e_1 = \omega - \omega_e, \quad e_2 = \bar{P}_L - \bar{P}_{Le}, \quad e_3 = \bar{x} - \bar{x}_e \tag{10}$$

and nominal control input

$$u = u_e + \frac{S_x \tau}{K_v} v \tag{11}$$

where v is the robust control input to be decided, then we have the error system

$$\dot{e} = A(\Delta)e + \Delta A + bv \tag{12}$$

where

$$e = [e_1, e_2, e_3]^T, \ b = [0, 0, 1]^T,$$

$$A(\Delta) = \begin{bmatrix} -c_e \frac{D\mu}{2\pi I} & (1-c_f)\frac{DS_p}{2\pi I} & 0 \\ -\frac{DK}{S_p \pi V_0} & -(c_s + \frac{2Kk_c}{V_0}) & \frac{2Kk_q S_x}{S_p V_0} \\ 0 & 0 & -\frac{1}{\tau} \end{bmatrix},$$

$$\Delta A = \begin{bmatrix} -\frac{D}{2\pi I}(\Delta c_e \mu \omega_e + \Delta c_f S_p \bar{P}_{Le}) \\ -(\Delta c_s + \frac{2K\Delta k_c}{V_0})\bar{P}_{Le} + \frac{2K\Delta k_q S_x}{S_p V_0}\bar{x}_e \\ 0 \end{bmatrix}$$

To clarify the problem to be solved, we introduce following definition [16].

Definition III.1 (Uniformly Ultimately Bounded Stable) Consider the system $\dot{x} = f(x)$, $x(t_0) = x_0$. A solution $x\colon \mathbf{R}^+ \to \mathbf{R}^n$, is said to be uniformly ultimately bounded stable (UUBS) with respect to a set $W \in \mathbf{R^n}$ containing the origin if there is a nonnegative constant $T = T(x_0, W) < \infty$, possibly dependent on x_0 and W but not on t_0, such that for all $t \geq t_0 + T(x_0, W)$

$$\|x(t_0)\| < \eta \quad \Longrightarrow \quad x(t) \in W \tag{13}$$

▮

Here we put a natural assumption for uncertain parameters.

Assumption III.1: For uncertain parameters in (9), each upper and lower bound, that is, $(.)^-$ and $(.)^+$ are known. ▮

In Section 3, we construct the robust feedback controller assuring uniformly ultimately bounded stability of the origin of error system (12) under the *Assumption (III.1).*

IV. Controller Design

In this section, we construct a robust controller for uncertainties and load fluctuations with recursive Lyapunov-based design. To be more specified, we design the virtual input and Lyapunov-like function to ensure UUB stability of each sub-system recursively compensating the effect of uncertain parameters. Before designing controller, we set some controller parameters evaluating some bounds of elements in (12).

$$\begin{aligned} &\varepsilon_0^- := (1-c_f^+)\frac{DS_p}{2\pi l_2}, \quad \varepsilon_0^+ := (1-c_f^-)\frac{DS_p}{2\pi l_1} \\ &\varepsilon_1 := \frac{D}{2\pi l_1}\left[(c_e^+ - c_e^-)\mu\omega_e + (c_f^+ - c_f^-)S_p\bar{P}_{Le}\right] \\ &\varepsilon_2 := \frac{DK}{S_p \pi V_0}, \quad \varepsilon_3 := c_s^- + \frac{2Kk_c^-}{V_0} \\ &\varepsilon_4^- := \frac{2Kk_q^- S_x}{S_p V_0}, \ \varepsilon_4^+ := \frac{2Kk_q^+ S_x}{S_p V_0} \\ &\varepsilon_5 := \left(c_s^+ - c_s^- + (k_c^+ - k_c^-)\frac{2K}{S_p V_0}\right)\bar{P}_{Le} \\ &\qquad\qquad + (k_q^+ - k_q^-)\frac{2KS_x}{S_p V_0}\bar{x}_e \end{aligned} \tag{14}$$

[Step 1] Defining new state variable $\tilde{e}_1, \tilde{e}_2$ and positive definite function V_1

$$\tilde{e}_1 = e_1, \quad \tilde{e}_2 = e_2 - \beta_1(\tilde{e}_1), \quad V_1 = \frac{1}{2}\tilde{e}_1^2 \tag{15}$$

the time derivative of V_1 along the trajectories of the system (12) satisfies

$$\begin{aligned} \dot{V}_1 \ \leq \ -c_e\frac{D\mu}{2\pi I}V_1 + \tilde{e}_1 \Big[&(1-c_f)\frac{DS_p}{2\pi I}(\tilde{e}_2 + \beta_1) \\ &- \frac{D}{2\pi I}(\Delta c_e \mu\omega_e + \Delta c_f S_p \bar{P}_{Le})\Big] \end{aligned} \tag{16}$$

For positive constant ξ_1, we choose virtual input β_1 for e_1-subsystem as

$$\beta_1(\tilde{e}_1) = -\frac{\varepsilon_1}{\varepsilon_0^-}\frac{\tilde{e}_1}{|\tilde{e}_1| + \xi_1} \tag{17}$$

This implies

$$\dot{V}_1 \leq -m_1 V_1 + f_1(\tilde{e}_1) + (1-c_f)\frac{DS_p}{2\pi I}\tilde{e}_1\tilde{e}_2 \tag{18}$$

where

$$m_1 = c_e^-\frac{D\mu}{2\pi l_2}, \quad f_1(\tilde{e}_1) = \frac{\xi_1\varepsilon_1|\tilde{e}_1|}{|\tilde{e}_1| + \xi_1}$$

The last term in the right hand side of (18) will be canceled in Step 2.

[Step 2] We introduce the new variable

$$\tilde{e}_3 = e_3 - \beta_2(\tilde{e}_1, \tilde{e}_2) \tag{19}$$

then we obtain

$$\begin{aligned}\dot{\tilde{e}}_2 &= \dot{e}_2 - \dot{\beta}_1(\tilde{e}_1)\\ &= -\frac{DK}{S_p\pi V_0}\tilde{e}_1 - (c_s + \frac{2Kk_c}{V_0})(\tilde{e}_2+\beta_1)\\ &\quad -(\Delta c_s + \frac{2K\Delta k_c}{V_0})\bar{P}_{Le} + \frac{2Kk_qS_x}{S_pV_0}(\tilde{e}_3+\beta_2)\\ &\quad +\frac{2K\Delta k_qS_x}{S_pV_0}\bar{x}_e - \dot{\beta}_1 \end{aligned} \tag{20}$$

We take the positive definite function for whole error system

$$V_2 = V_1 + \frac{1}{2}\tilde{e}_2^2 \tag{21}$$

The time derivative of V_2 along the trajectories of the system (12) satisfies

$$\begin{aligned}\dot{V}_2 &= \dot{V}_1 + \tilde{e}_2\dot{\tilde{e}}_2\\ &\le -m_2V_2 + f_1(\tilde{e}_1)\\ &\quad +\tilde{e}_2\Big[-\frac{DK}{S_p\pi V_0}\tilde{e}_1 + (1-c_f)\frac{DS_p}{2\pi I}\tilde{e}_1 + \frac{2Kk_qS_x}{S_pV_0}\beta_2\\ &\quad -(\Delta c_s + \frac{2K\Delta k_c}{V_0})\bar{P}_{Le} + \frac{2K\Delta k_qS_x}{S_pV_0}\bar{x}_e - \dot{\beta}_1\Big]\\ &\quad +\frac{2Kk_qS_x}{S_pV_0}\tilde{e}_2\tilde{e}_3 \end{aligned} \tag{22}$$

where $m_2 = m_1 + \varepsilon_3$. Therefore we choose the virtual control input β_2 as

$$\beta_2(\tilde{e}_1,\tilde{e}_2) = -\frac{1}{\varepsilon_4^-}\frac{\tilde{e}_2}{|\tilde{e}_2|+\xi_2}\left[(\varepsilon_0^+ + \varepsilon_2)|\tilde{e}_1| + \varepsilon_5 + |\dot{\beta}_1|\right] \tag{23}$$

where ξ_2 is positive constant design parameter. Then

$$\dot{V}_2 \le -m_2V_2 + f_2(\tilde{e}_1,\tilde{e}_2) + \frac{2Kk_qS_x}{S_pV_0}\tilde{e}_2\tilde{e}_3 \tag{24}$$

where

$$f_2(\tilde{e}_1,\tilde{e}_2) = f_1(\tilde{e}_1) + \frac{\varepsilon_2|\tilde{e}_2|}{|\tilde{e}_2|+\xi_2}\left[(\varepsilon_0^+ + \varepsilon_2)|\tilde{e}_1| + \varepsilon_5 + |\dot{\beta}_1|\right]$$

[Final Step] We take the positive definite function for whole error system

$$V = V_2 + \frac{1}{2}\tilde{e}_3 \tag{25}$$

We obtain the time derivative of V along the trajectories of the system (12) as

$$\begin{aligned}\dot{V} &= -\dot{V}_2 + \tilde{e}_3\dot{\tilde{e}}_3\\ &\le -mV + f_2 + \tilde{e}_3\left[v - \frac{\beta_2}{\tau} + \frac{2Kk_qS_x}{S_pV_0}\tilde{e}_2 - \dot{\beta}_2\right]\end{aligned} \tag{26}$$

where $m = m_2 + \frac{1}{\tau} > 0$. Therefore we choose a robust control input v as

$$v = \frac{\beta_2}{\tau} + \dot{\beta}_2 - \frac{\varepsilon_4^+\tilde{e}_3}{|\tilde{e}_3|+\xi_3}|\tilde{e}_2| \tag{27}$$

where ξ_3 is positive constant design parameter. Finally we have

$$\dot{V} \le -mV + f_3(\tilde{e}_1,\tilde{e}_2,\tilde{e}_3) \tag{28}$$

where

$$f_3(\tilde{e}_1,\tilde{e}_2,\tilde{e}_3) = f_2(\tilde{e}_1,\tilde{e}_2) + \frac{\varepsilon_4^+\xi_3|\tilde{e}_3|}{|\tilde{e}_3|+\xi_1}|\tilde{e}_2|$$

Applying LaSalle's invariant theorem [17] to this result, we have desired stability property(UUBS). It is easy to show that the residual set

$$D_s = \left\{(\tilde{e}_1,\tilde{e}_2,\tilde{e}_3)\ \Big|\ \frac{\tilde{e}_1^2+\tilde{e}_2^2+\tilde{e}_3^2}{2} \le f_3(\tilde{e}_1,\tilde{e}_2,\tilde{e}_3)\right\} \tag{29}$$

is a compact set. Moreover by taking the design parameters ξ_1, ξ_2 small, the set D_s will be arbitrary small. Then we have main result.

Theorem IV.1: Consider the system (12) under the assumption III.1. The robust controller (11), (27) ensures the UUBS of the origin of (12). Moreover the state error approaches compact set D_s defined in (29). ∎

Remark IV.1: As we see from the equation (12), the coefficient of e_2 which is input signal to the e_1-subsystem has the uncertainty I, c_f. In this case, we construct theoretically the stabilizing controller by means of sgn(·) function in general, however, it requires large control input practically. To avoid the case, we define the virtual input β_1 as smooth function by introducing design parameters ξ_1. For remaining two design steps, we apply same ides. In fact, we have sgn function as boundary function by taking ξ_1, ξ_2 and ξ_3 as 0. ∎

V. Simulation results

To illustrate the proposed robust controller above, simulation results are obtained for a system(Table I).

TABLE I

Physical parameters for numeriacl example

parameter	value
D	1.6×10^{-5}(m^3/rev)
μ	0.797×10^{-3}(Pa·s) at 30°C
ρ	995.7(kg/m^3)
K	2250×10^6(Pa)
P_S	5×10^6(Pa)
V_0	1.272×10^{-4}(m^3)
w	12.5×10^{-3}(m)
k_v	1×10^{-4}(m/V)
τ	0.01(1/s)

It is set that each friction coefficient, leak coefficient and load inertia vary below as:

$$\begin{gathered}1\times10^5 = c_e^- \le c_e \le c_e^+ = 1\times10^7\\ 0.2 = c_f^- \le c_f \le c_f^+ = 0.5\end{gathered}$$

$$
\begin{aligned}
0.1 = c_s^- \le c_s \le c_s^+ &= 0.3 \\
0.9 = k_q^- \le k_q \le k_q^+ &= 1.1 \\
1 \times 10^{-14} = k_c^- \le k_c \le k_c^+ &= 2.5 \times 10^{-13} \\
0.124 = l_1 \le I \le l_2 &= 0.284
\end{aligned}
\tag{30}
$$

which are obtained from estimation. Each uncertain parameter is all periodically time varying, for example

$$c_e(t) = \tfrac{1}{2}(c_e^+ - c_e^-) + (c_e^- - c_e^-)\sin(t)$$

for the coefficient of viscous friction and the frequencies of varying are distinct. We give the nominal values of these parameters as its average. The desired rotational velocity is set to $\omega_e = 20(\mathrm{rad/s})$, then this implies $P_{Le} = 0.124(\mathrm{MPa})$, $x_e = 0.509(\mathrm{mm})$, $u_e = 0.509(V)$. The design parameters are chosen as $S_p = 1 \times 10^6$, $S_x = 1 \times 10^{-5}$, $\xi_1 = \xi_2 = \xi_3 = 0.1$ and the initial condition as $\omega(0) = 0\,(\mathrm{rad/s}), P_L(0) = 0\,(\mathrm{Pa}), x(0) = 0\,(\mathrm{mm})$. And we notice the input saturation of servo valve, $\pm 10(V)$.

In this simulation, we consider a conventional PID controller as well as robust controller to compare the results of these controllers. By try and error, we choose PID controller parameters as

$$
\begin{aligned}
u_{PID} &= u_e - 0.1 \int_0^t e_1 dt - 0.001 e_1 - 0.001 \dot{e}_1 \\
e_1 &= \omega - \omega_e
\end{aligned}
$$

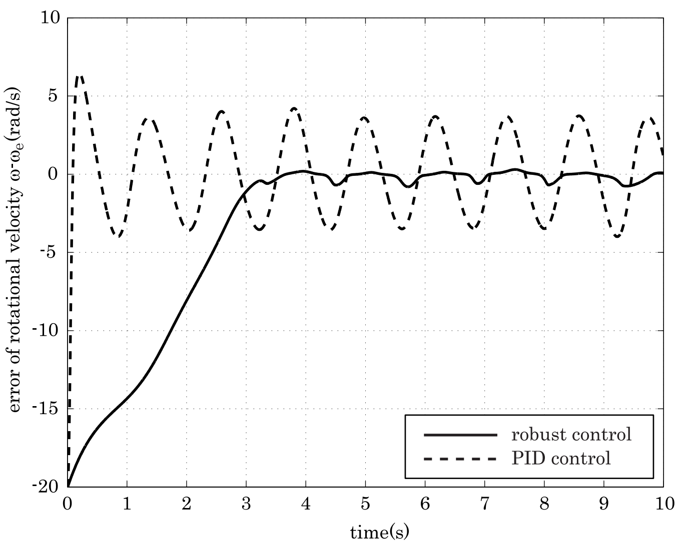

Fig. 3. rotational velocity error $e_1 = \omega - \omega_e(\mathrm{rad/s})$

Fig.3 - Fig.5 are simulation results and show the rotational velocity error $e_1(= \omega - \omega_e)$, the load pressure error $e_2(= P_L - P_{Le})$ and the spool position error $e_3(= x - x_e)$, respectively. The solid line is for the robust control results and the dotted for PID control. From Fig.3, it is observed that the error of the proposed robust controller is converges to a neighborhood of 0 smooth and attenuates the parameter uncertainties up to a certain level while the error of PID control has large vibration. This amplitude of error band depends on the design parameters ξ_i. In Fig.4 and Fig.5, there remain the pressure and the valve spool displacement vibration for the uncertain parameters change. And the state variable P_L is very sensitive to the uncertainty of flow coefficient k_q because this parameter is multiplied to $\bar{x}$ which is designed to cancel the right hand side of P_L-subsystem so that β_2 is required to be large signal depending on the magnitude of $k_q^+ - k_q^-$. The precise estimation of this value is very important for the application.

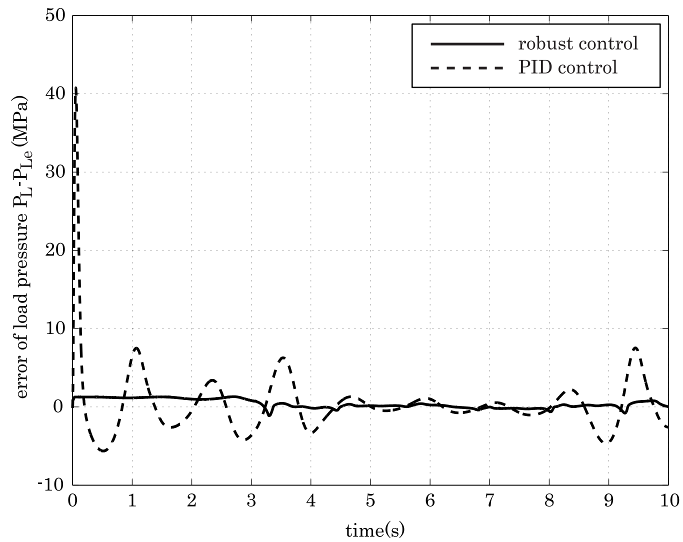

Fig. 4. load pressure error $e_2 = P_L - P_{Le}(\mathrm{MPa})$

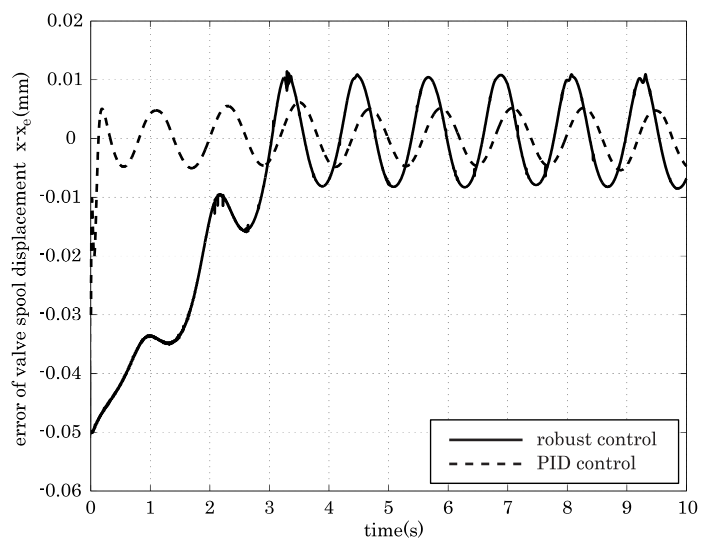

Fig. 5. valve spool displacement error $e_3 = x - x_e(\mathrm{mm})$

In Fig.6, it can be seen that introducing the design parameters $\xi_i (i = 1, 2, 3)$ prevent the robust input v from destabilizing the whole system. In this simulation we give rougher value k_q as it may be estimated and this makes uniformly ultimate set to which the error converges being borderer.

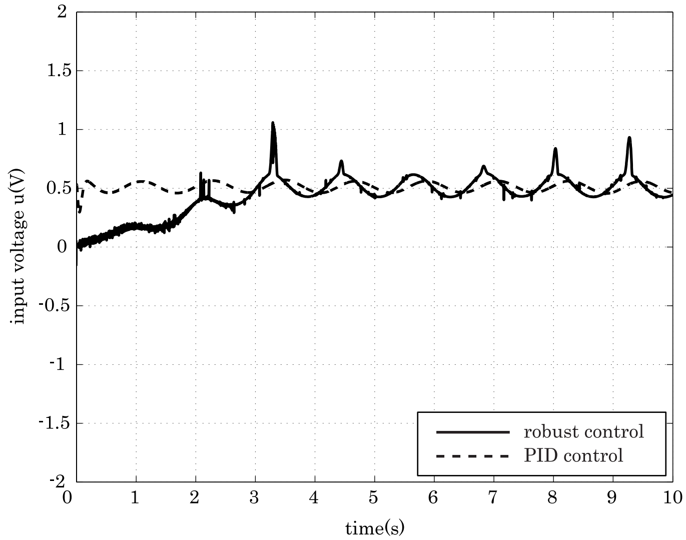

Fig. 6. control input u(V)

VI. Conclusions

In this paper, we proposed a method to construct a robust rotational velocity controller of water hydraulic servomotor. The parameter uncertainties describing viscous and Coulomb friction, leakage of the water hydraulic motor, linearizing error of flow equation and load fluctuation, are considered in controller design. By using Lyapunov-based recursive method, the uniformly ultimately stabilizing problem with input uncertainty is solved. The resulting controller ensures the convergence of the error to the compact set that can be arbitrary small by design parameter. The numerical example shows the effectiveness of proposed robust controller. We examine the controller experimentally in future work.

Acknowledgments

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Kazuhisa Ito received the B.S., M.S and Ph.D. degrees in mechanics from Sophia University (Control Engineering Laboratory), Tokyo, Japan, in 1993, 1995 and 2001, respectively. From 1995, he worked for Komatsu Ltd. in the production department of construction machine for three years. In 2002, he joined the Department of Mechanical Engineering, Sophia University, where he is associate researcher in fluidics laboratory. His research interests include nonlinear systems, robust control theory and its application for mechanical systems, especially the water hydraulic devices.