ONLINE SYSTEM IDENTIFICATION OF HYDRAULIC SERVO ACTUATORS BY THE SELF-EXCITED OSCILLATION METHOD (APPLICATION TO ANGULAR VELOCITY CONTROL SYSTEM)

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ABSTRACT

It has been well known that hydraulic servo actuators can often be approximated with a standard second order transfer function when the controller is designed for these systems. Earlier research developed a simple method utilizing the self-excited oscillation caused from the hydraulic servo actuators to directly estimate the dynamic parameters such as the damping ratio and undamped natural frequency. The advantage of this method is an online identification ability that is able to identify these parameters while the operation conditions are continually changing. Although this method was confirmed to be very useful, it is available only when the spool valve is close to the neutral position, which corresponds to the operation of position control systems. In the practical situations, the spool valve sometimes operates at displaced position from the neutral center position such that a hydraulic motor speed is controlled. This paper proposes a revised self-excited oscillation method for this system. The experimental works are conducted by giving the various system pressures and angular velocities so as to validate the method. The resulting frequency characteristics of these identified transfer functions are then compared with those of the measured data by the frequency characteristics method. In addition, in order to demonstrate the effectiveness of the self-excited oscillation method, the dynamic parameters of two practical devices such as a motion seat and aircraft tail surface control simulator are identified and compared with the results from the frequency response method.

Keywords: hydraulic servo actuator system, system identification, limit cycle, second order transfer function

1 Introduction

Hydraulically driven actuators that are controlled by a servovalve have been employed in various industry fields because they have several advantages such as high power density, compactness, and so on. The hydraulic servo actuators have non-linear characteristics in their dynamic behavior. These dynamic characteristics are strongly affected by the system operating condition such as supply pressure, load, actuator speed, etc. For the control design of these hydraulic servo actuator systems, it is very important to know a precise knowledge of the system dynamic behavior and the corresponding mathematical descriptions. However, development of the precise mathematical model is not easy work in a practical situation. Hence, this is often treated as simplified descriptions by linearized approximation for analyzing the system dynamics. Their dynamic characteristics are generally dealt with as a second order transfer function (e. g. Watton, 1989). It should be noted that the hydraulic servo actuator system acts as a linear system only at an operating point.

In order to design a controller of hydraulic servo actuators adequately, an estimation of unknown parameters will be required. One of the practical ways to obtain these parameters is the use of an experimental identification method. Therefore numerous identification methods have been proposed to estimate unknown parameters. Jelali and Kroll introduced and reviewed various identification methods for hydraulic servo systems using ARMA, Fuzzy, Neural and so on (Jelali and Kroll, 2003). The frequency response method and the step response method are well known as classic identification methods for an estimation of two dynamic parameters of the second order transfer function, which are the damping coefficient ζ and the undamped natural frequency ω_n. They are still often used as the practical methods. Utilizing the limit cycle of the

This manuscript was received on 08 December 2010 and was accepted after revision for publication on 10 April 2010

identified system is other way to estimate the unknown parameters (Fujii et al., 1968). For instance, in the process control fields, the automatic tuning technique of PID controller utilizing the limit cycle generated in the relay feedback system has been already applied to the industry (Astrom et al., 1995). For the hydraulic servo applications, no report has been published for the method which identifies the dynamic parameters ζ and ω_n of the standard second order transfer function by utilizing the limit cycle. The limit cycle is generally considered to be harmful to the hydraulic servo system. It may cause damage if the amplitude of the oscillation is large and the oscillation continues for a long time. However, some application would be allowed to oscillate if the oscillation is small for a very short period of time. Our research group proposed a simple method utilizing the self-excited oscillation of the hydraulic servo actuator system to directly estimate the dynamic parameters ζ and ω_n (Konami et al., 1996 and Konami et al., 1997). The usefulness and the validity of the method were described. In these reports, the controller of the self-oscillation system was arranged by an analog electric circuit. This method was expanded to an online identification system, which was realized by using a digital signal processor (DSP) (Ichiyanagi et al., 2003). In addition, the amplitude and frequency correction factors were analytically obtained from the described function in the position control system (Nishiumi et al., 2005). After these works, some papers referred this self-excited oscillation method and applied it to hydraulic and pneumatic servo systems (Hwng and Cho, 2002 and Noskievic, 2005).

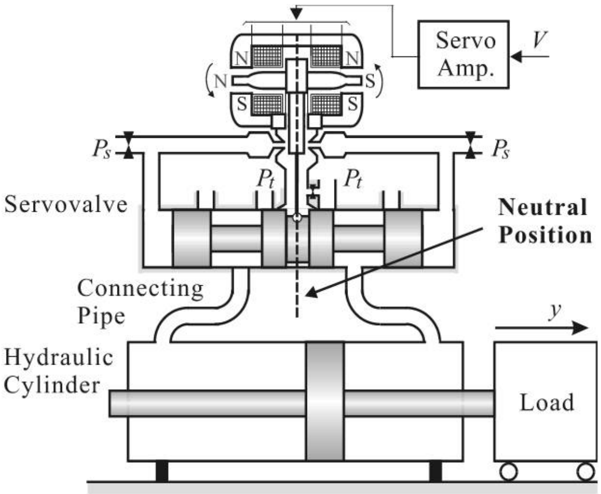


Fig. 1: *Hydraulic servo cylinder*

Even though the previous reports proved the usefulness of the proposed method, this method only identifies the dynamic parameters when the servovalve spool is close to the neutral position. In other words, the method was only available for position controlled hydraulic servo actuators. But in practical operating conditions, if the actuator is a hydraulic motor and if the motor rotational speed is controlled, the servovalve is operated as the valve spool is displaced from the neutral position. The previous method could not be applied to this velocity controlled hydraulic servo actuator system. This paper addresses the revised self-excited oscillation method which can identify the dynamic parameters of a velocity controlled hydraulic servo motor. In this method, the angular velocity self-excited oscillation around a constant rotational speed is utilized to identify the dynamic parameters. In addition, in order to evaluate the self-excited oscillation method as applied to some systems for hydraulics education, the dynamic parameters of two applications are experimentally identified and examined. These applications are a two degree of freedom hydraulic motion seat and an aircraft tail control surface simulator.

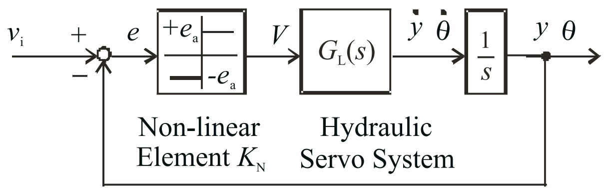


Fig. 2: *Position control self-excited oscillation system*

The paper describes the fundamental principle of the self-excited oscillation method for both the position control system and the proposed velocity control system. The experimental apparatus of identified hydraulic servo motor system and two practical hydraulic servo actuator systems are explained. The online experimental identifications for the hydraulic servo motors are carried out by providing the various supply pressure and angular velocity. The identified results derived from the conventional frequency response and proposed self-excited oscillation methods are compared to confirm the validity of the proposed method. In order to demonstrate the usefulness of this method, the dynamic parameters of the hydraulic motion seat and the aircraft tail control surface simulator are identified and also compared with the results from frequency response method.

2 Self-excited Oscillation Method

2.1 Position Control System

A schematic diagram of an identified hydraulic servo cylinder is illustrated in Fig. 1. The system includes a hydraulic cylinder, load mass, servo amplifier, and servovalve. In the case of the position feedback control system, a spool of the servovalve moves around the neutral position. The relationship between the input voltage of the servo amplifier V and the velocity of the cylinder position $\dot{y}$ can be simply described by Eq. 1 as a linear transfer function, provided that connecting pipes are treated as lumped element characteristics and the dynamic characteristics of the servovalve are negligible, i.e. the servovalve dynamics are considerably faster than the hydraulic actuator response.

$$G_L(s) = \frac{\dot{y}(s)}{V(s)} = \frac{K_L \omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{1}$$

where K_L, ω_n and ζ are the gain of the hydraulic servo actuator system, the undamped natural angular frequency and the damping ratio, respectively.

The self-excited oscillation method is able to directly identify the system dynamic parameters ω_n and ζ of this second order transfer function by using the self-excited oscillation of the identified system. Figure 2 shows the block diagram of the position (angular position) self-excited oscillation system that was utilized to identify the linear transfer function G_L. This self-excited oscillation system is realized by putting a non-linear element K_N in the forward side of the hydraulic servo system. Since the non-linear element, which is an ideal relay, has variable gain characteristics expressed by Eq. 2, the system causes the limit cycle at the point of stability limit.

$$K_x = \frac{e_a}{|e|} \tag{2}$$

where e_a is the setting voltage of non-linear element. This non-linear element outputs the setting voltage $V = \pm\, e_a$ which corresponds to plus or minus of the error signal e as shown in Fig. 3. When the error signal becomes close to zero, the variable gain K_x increases and leads to an instability of the system. Then once the system enters the unstable region, the error signal becomes large in turn and the system goes to the stable region again. Finally this causes limit cycle. The exact frequency of this limit cycle varies continuously within its cycle. Hence using the average angular frequency ω_s and the amplitude V_s, the wave shape of this limit cycle is approximated to the sinusoidal wave as shown in Eq. 3.

$$u(t) \approx V_s \sin \omega_s t \tag{3}$$

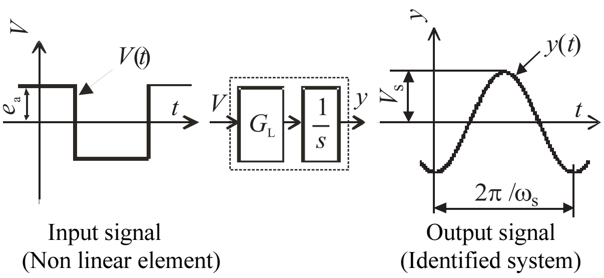


Fig. 3: *Input and output of identified system*

Since the non-linear element is variable gain characteristics, the gain at stability limit can be obtained in Eq. 4 by using Routh-Hurwitz stability criterion.

$$K_{Nc} = \frac{e_a}{e_c} = \frac{2\zeta\omega_n}{K_L} \tag{4}$$

The undamped natural angular frequency ω_n of the identified system can be defined as Eq. 5 by introducing the frequency correction factor ξ, because this value is equal to the angular frequency of the stability limit.

$$\omega_n = \frac{\omega_s}{\xi} \tag{5}$$

Then the damping coefficient of the identified system is derived from Eq. 4 and Eq. 5

$$\zeta = \frac{\Gamma K_L e_a}{2 V_s \omega_n} \tag{6}$$

where Γ is the amplitude correction factor defined as the ratio of the error signal e_c at the stability limit and the amplitude V_s of the limit cycle wave, i.e. self-excited oscillation wave.

$$\Gamma = \frac{V_s}{e_c} \tag{7}$$

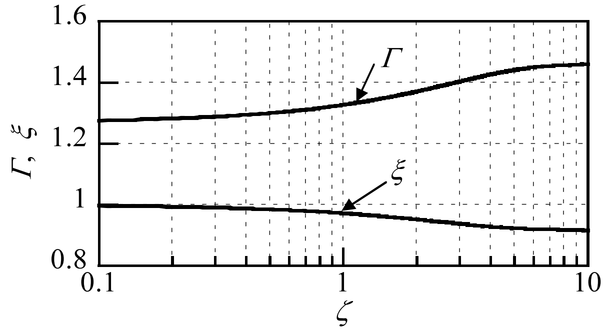


Fig. 4: *Amplitude and angular frequency correction factors*

Therefore in this identification method, two dynamic parameters can be easily obtained from Eq. 5 and Eq. 6 by measuring the amplitude V_s and the angular frequency ω_s of the measured self-excited oscillation wave. Two correction factors for the angular frequency and the amplitude are expressed by the following equations.

$$\left.\begin{aligned} \xi &= 1.0 - 0.0315\zeta + 0.00415\zeta^2 - 0.000185\zeta^3 \\ \Gamma &= 1.27 + 0.0647\zeta - 0.00762\zeta^2 + 0.000307\zeta^3 \end{aligned}\right\} \tag{8}$$

These correction factors are obtained from simulation of the position control self-excited oscillation system shown in Fig. 2 using MATLAB/Simulink®. The simulation was conducted with changing the parameters K_L, ω_n, ζ, e_a. Equation 8 is derived from the interpolation of these simulation results which are shown in Fig. 4. The amplitude and angular frequency correction factors are dependent on the damping coefficient ζ and independent of K_L, ω_n, e_a.

2.2 Velocity Control System

For the application of a rotary actuator system, the rotational speed of the hydraulic motor is often controlled. It should be considered a condition where the hydraulic motor is driving at a constant rotational speed. In this case, the servovalve spool is displaced x from the neutral position as shown in Fig. 5 and therefore the dynamic characteristics of the identified system are varied from the spool is around the neutral position, i.e., the position control system. In order to apply the self-excited oscillation method to this hydraulic motor angular velocity control system, the angular velocity ω is used for the feedback signal to make up the angular velocity self-excited oscillation system shown in Fig. 6 (a). It is clear from Fig. 2 and Fig. 6 (a) that the open loop transfer functions of both the angular velocity self-excited oscillation system and the position self-excited oscillation system are the same.

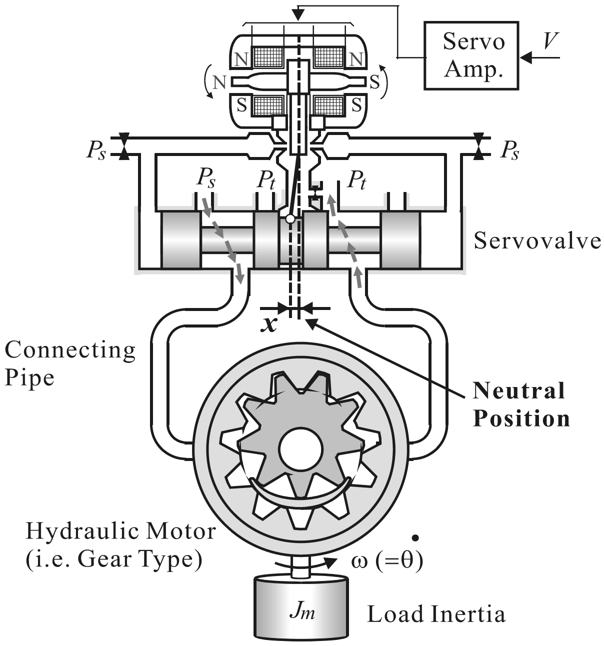
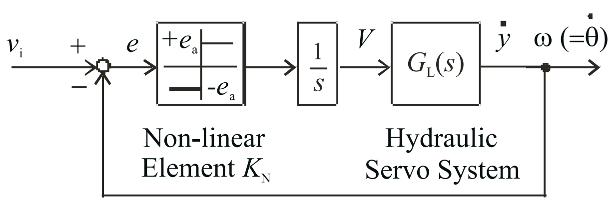
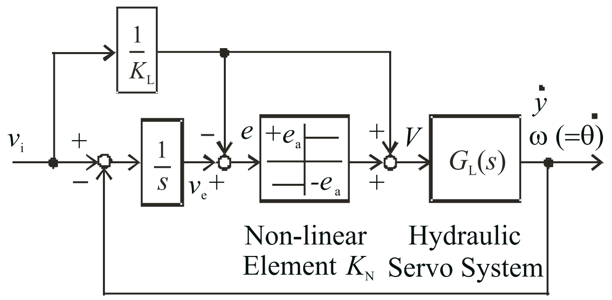


Fig. 5: *Hydraulic servo motor system*



(a) Velocity control self-excited oscillation system



(b) Revised velocity control self-excited oscillation system

Fig. 6: *Velocity control self-excited oscillation system*

Therefore, all equations described in the position control system can be applied to the velocity control system. Figure 6 (b) shows the revised angular velocity self-excited oscillation system. The self-excited oscillation system shown in Fig. 6 (a) is transformed into the system shown in Fig. 6 (b) because there is a problem in the measurement of angular velocity. The noise of high order harmonic frequency is often superimposed onto the measured angular velocity. This causes the inaccurate estimation of the dynamic parameters. Hence, the integrator element is placed forward to the non-linear element as a noise filter. In this case, the input signal of the non-linear element is oscillated around v_i/K_L. The non-linear element always outputs $+e_a$ and then the angular velocity self-excited oscillation would not be occurred. Therefore, v_i/K_L must be subtracted from v_e so that the input signal of the non-linear element is oscillated around zero. Now the non-linear element can output the signal $+e_a$ and $-e_a$ periodically. Then v_i/K_L is added to the output signal of the non-linear element $\pm e_a$ i.e., the input signal of the servo amplifier. This makes the revised angular velocity self-excited oscillation system is equivalent to the system shown in Fig. 6 (a).

2.3 Online Identification Procedure

Hydraulic servo actuators have been utilized in various industrial motion control fields. In the past decade, the control technology of these systems is rapidly developing due to the recent progress related to the electrical and electronic instruments including computers and sensors. Therefore the latest hydraulic servo actuator systems are often equipped with the intelligent and advanced control strategy functions. The self-excited oscillation method can also be a suitable software application of these modern electro-hydraulic servo actuator systems, because this method is able to estimate the system dynamic parameters online. In this report, a digital signal processor is used to compute the algorithm of the self-excited oscillation method. The online parameter estimation procedure consists of the next four processes.

(1) Measurement of the self-excited oscillation wave

The position (angular position) or angular velocity signals are acquired from appropriate sensors using an analog to digital (A/D) converter.

(2) Low pass filter

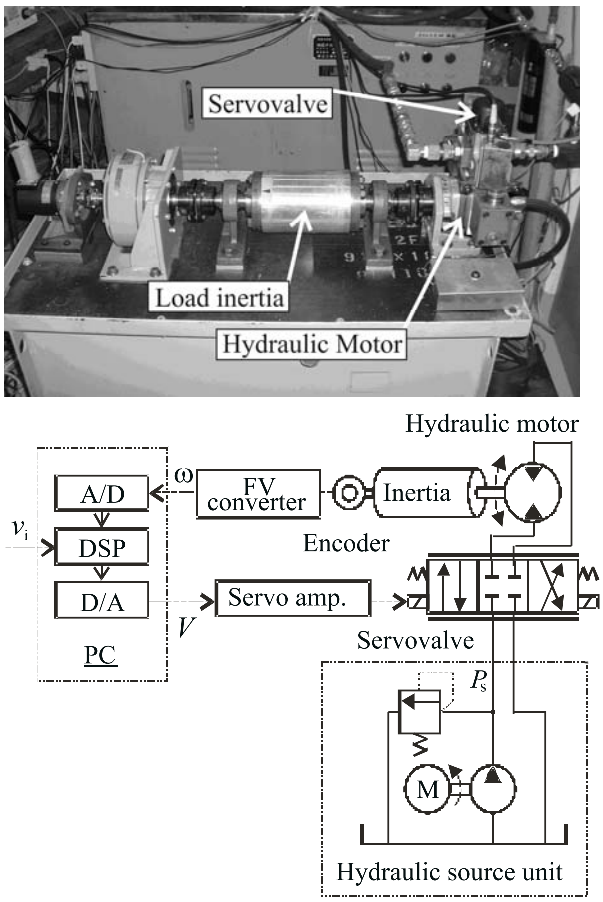


Fig. 7: *Experimental apparatus of Hydraulic servo motor*

For the position feedback systems, a low pass filter will be required if the self-excited oscillation wave of identified systems contain some high frequency noise components. The design of the filter is dependent on the noise frequency of the identified system.

(3) Analyze the measured wave

This is the process that finds the amplitude V_s and angular frequency ω_s of the self-excited oscillation wave. The wave signals of the self-excited oscillation are stored in the computer memory. Then the present wave data $y(i)$ is compared with the past data $y(i\text{-}1)$ and $y(i\text{-}2)$ where the i-1, i-2 are the time delay for one previous and two previous sampling time. The amplitude V_s can be derived from the difference between the positive peak data ($y(i\text{-}2) < y(i\text{-}1) > y(i)$) and the negative peak data ($y(i\text{-}2) > y(I\text{-}1) < y(i)$). The angular frequency ω_s is also obtained from the time data at the positive and negative peaks.

(4) Computation of dynamic parameters

From the amplitude V_s and angular frequency ω_s information obtained the previous process, the damping ratio ζ and undamped natural frequency ω_n are calculated from Eq. 5 to 8. Since these equations contain the amplitude and frequency correction factor that is function of damping ratio ζ, the calculation must be done iteratively. In the iterative calculation, the first values of the correction factors are set to $\xi_1 = 0.95$ and $\Gamma_1 = 1.4$. Then the first value of the damping ratio ζ_1 can be estimated and is used for the second values of the correction factors ξ_2 and Γ_2. By iterating until the error becomes insignificant, the precise dynamic parameters ζ and ω_n are finally obtained. Since this iterating procedure usually takes only several iterations, the computing time is very short.

The DSP processes these procedures every sampling period. It should be noted that the least time of the parameter estimation in this method becomes the self-excited oscillation period, since the dynamic parameters can be obtained after analyzing the angular frequency of one self-excited oscillation wave.

3 Identified Experimental Apparatus

In order to verify the usefulness of the self-excited oscillation method, the dynamic characteristics of three different hydraulic servo actuators are identified. These are (a) test hydraulic servo motor, (b) hydraulic motion seat, and (c) aircraft tail control surface simulator. These applications are mainly used for the educational purpose of our academy to learn the mechanism of the hydraulic components and the integrated system of electronics and mechanics. They are also very useful experimental apparatus to learn how the control theory is applied to a real system.

(a) Hydraulic servo motor

Figure 7 shows the picture and the schematic diagram of test hydraulic servo motor used in this study. This is the laboratory test bed for the hydraulic rotary actuator applications. The hydraulic source unit includes a pump, electric motor, relief valve and main tank. The supply pressure is adjusted by this relief valve. The identified hydraulic servo motor consists of a servovalve, hydraulic motor and load inertia. The main specification of this system is shown in Table 1 (a). The gain of the hydraulic motor system K_L has been obtained in advance by measuring the static characteristics. This gain shows a dependency characteristic on the supply pressure and is expressed in Eq. 9.

$$K_L = 4.25 + 2.72 \times 10^{-6} P_s - 0.134 \times 10^{-12} P_s^2 \tag{9}$$

where the supply pressure unit of this equation is [Pa]. The angular velocity ω is detected by the encoder and sent to the PC through a frequency to voltage (F/V) converter as a feed-back signal.

(b) Two degree of freedom hydraulic motion seat

The motion simulator is one of appropriate applications for the hydraulic servo actuator system since it requires higher forces at fast response speed. Figure 8 shows a hydraulically driven motion seat which has the two degree of freedom. The seat can be driven in the pitch and roll axis. The figure also shows the schematic diagram of the experimental system. The main structure of the two degree of freedom hydraulic motion seat is divided into two parts; the hydraulic actuator and controller parts. The controller part in this system is managed by a personal computer with a single board type digital signal processor (DSP). The hydraulic actuator part mainly includes two hydraulic actuators and two servovalves. The hydraulic unit which supplies the pressure P_s is the same circuit as shown in Fig. 7. Roll movement is provided by the hydraulic motor, while pitch movement is driven by the hydraulic cylinder. The cylinder translational movement is converted to the rotational movement by the mechanical links. The angle of roll movement is measured by the encoder. The angle of pitch movement is obtained from the cylinder position measured by the linear scale. Both signals are sent to the DSP through the analog to digital (A/D) converter. The main specifications of this system are shown in Table 1(b).

Table 1: Specification of identified experimental apparatuses

(a) *Test hydraulic servo motor*

Motor displacement	1.75×10^{-6} m^3/rad
Load inertia	1.92×10^{-2} kgm^2
Servovalve flow rate	30 L/min
Hydraulic fluid	MIL-H5606C
Fluid temperature	40 ± 2 °C

(b) *Two degree of freedom hydraulic motion seat*

Actuator 1	Roll axis
Hydraulic motor	Displacement 350 cc/rev
Servovalve flow rate	30 L/min
Max.rotation angle	-30 – +30 degree
Actuator 2	**Pitch axis**
Hydraulic cylinder	Piston diameter 40 mm Rod diamter 25 mm
Servovalve flow rate	30 L/min
Max.rotation angle	-30 – +30 degree
Hydraulic fluid	ISO-VG32
Fluid temperature	40 ± 2 °C

(c) *Aircraft tail control surface simulator*

Actuator 1	Rudder
Servovalve flow rate	28.5 L/min
Max.rotation angle	-30 – +30 degree
Actuator 2	**Stabilator**
Servovalve flow rate	19 L/min
Max.rotation angle	-25 – +25 degree
Hydraulic fluid	ISO-VG32
Fluid temperature	40 ± 2 °C

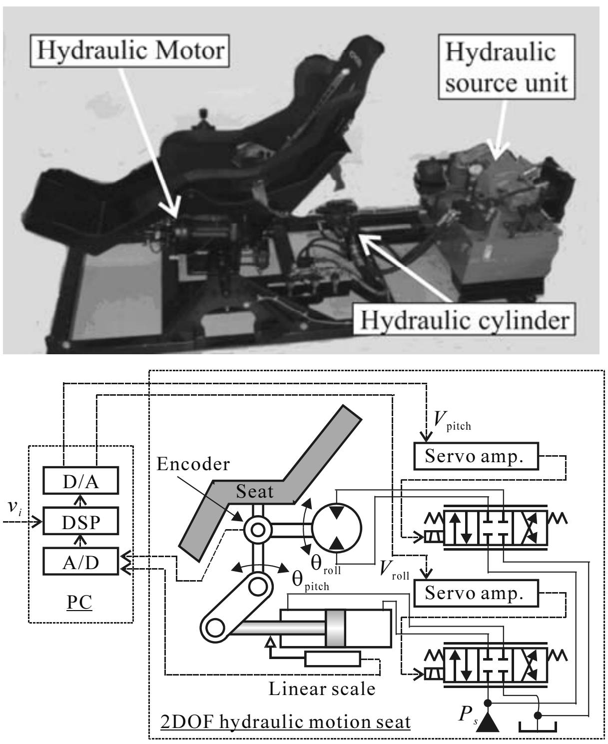


Fig. 8: *Experimental apparatus of two degree of freedom hydraulic motion seat*

Figure 9 shows the picture of our aircraft tail control surface simulator. This devise simulates the movements of the aircraft tail control surface by the inputs of a joystick and foot pedals. The tail control surface consists of the rudder and the stabilator which respectively controls the yaw and pitch direction. The direction of the rudder is manipulated with the movement of foot pedals. The stabilator's direction is controlled by the operator's joystick. Both inputs from the operator are provided electrically to the servovalves through the DSP inserted computer. This is a so called "fly by wire" system. The configuration of this system is basically the same as the hydraulic motion seat shown in Fig. 8. Instead of the hydraulic motor for the roll actuation of the hydraulic motion seat, this tail control surface simulator uses hydraulic cylinders with mechanical links for both rudder and stabilator actuation. The main specifications are shown in Table 1 (c).

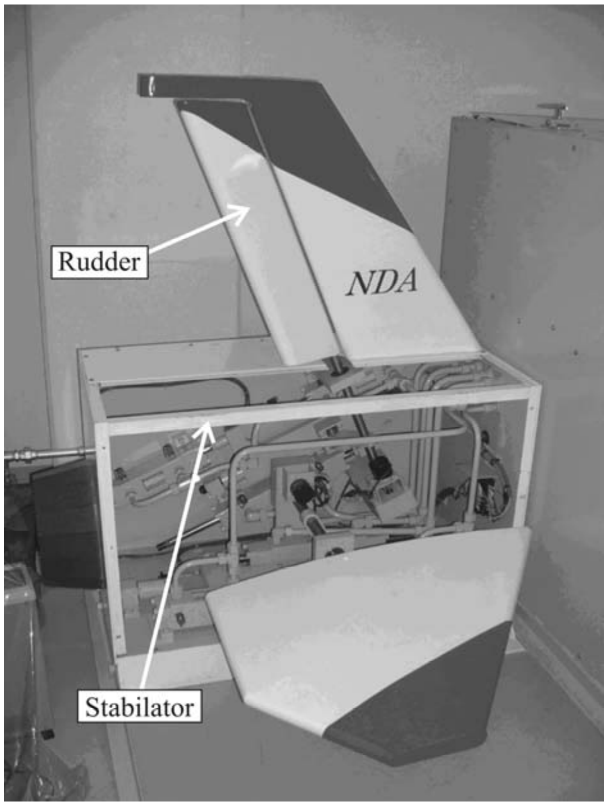


Fig. 9: *Aircraft tail control surface simulator*

These applications of hydraulic servo actuator are all combined with the computer. Hence for the online identification of these systems, the digital signal processer embedded inside a PC is used in this report to analyze the self-excited oscillation wave and calculate the dynamic parameters. The online identification algorithm described in the previous chapter is written in MATLAB/Simulink® model code. Then the program is implemented in the single board type DSP (dSPACE DS1102:TI-TMS320C31 25 MHz processor with 16 bit AD/DA converter). The sampling time is chosen to 0.5 ms. Data acquisition and editing parameter sets are managed by the virtual instrument panel on PC monitor (The software is "Control desk" by dSPACE).

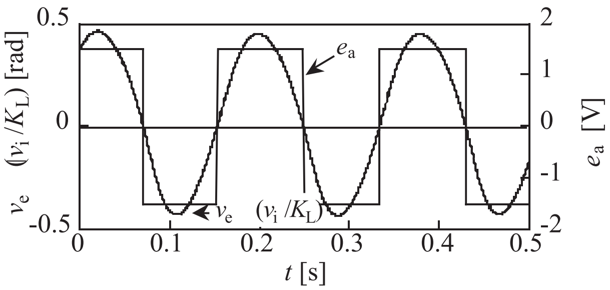


Fig. 10: *Example plots of self-excited oscillation waves*

4 Experimental Results

The verification of the self-excited oscillation method for the angular velocity control system is discussed. This is the case that the self-excited oscillation method applies to the condition where the servovalve spool is constantly displaced from the neutral position. Figure 10 shows the example plot of the measured self-excited oscillation wave. The input and output signals of the non-linear element shown in Fig. 6 (b) are examined. It can be seen from this figure that the noise component imposed on the angular velocity oscillation wave is completely attenuated and the non-linear element outputs the setting voltage e_a corresponding to the angular velocity self-excited oscillation period without suffering from the influence of the noise.

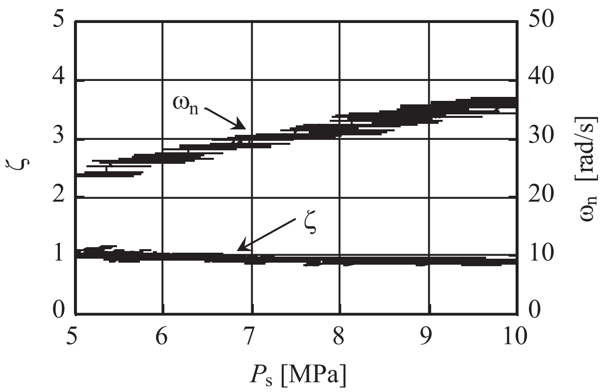


Fig. 11: *Identified dynamic parameters (hydraulic servo motor system, $K_L v_i = 32.5$ rad/s, $e_a = 1.5$ V)*

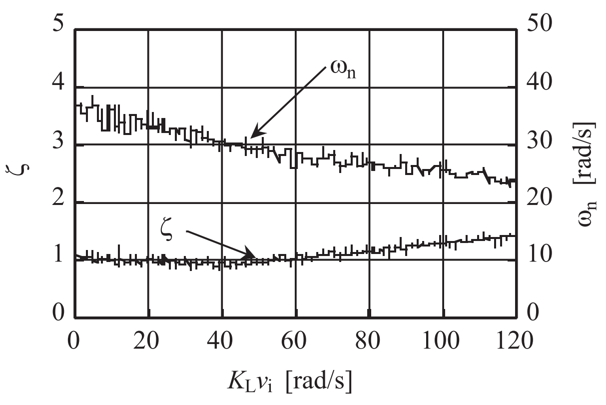


Fig. 12: *Identified dynamic parameters (hydraulic servo motor, $P_s = 8$ MPa, $e_a = 1.5$ V)*

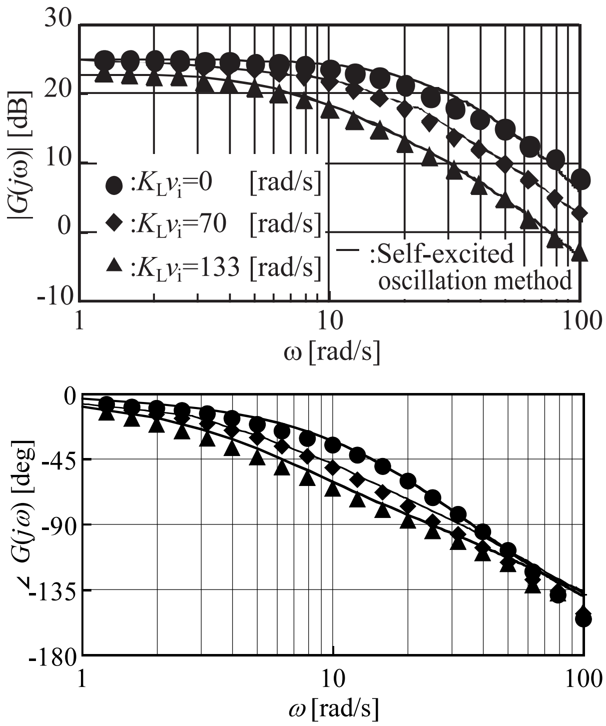


Fig. 13: *Comparison between self-excited oscillation method and frequency response (hydraulic servo motor system, $P_s = 8$ MPa, $e_a = 1.5$ V)*

Figure 11 shows the results of the online identification with changing the supply pressure P_s continuously from 5 MPa to 10 MPa at the reference rotational speed 32.5 rad/s and the setting voltage e_a = 1.5 V. It is obvious that these dynamic parameters are estimated continuously by the proposed identification method. This indicates the realization of online identification. The results also show that the identified dynamic parameters are notably affected by the supply pressure. In particular, the undamped natural angular frequency ω_n increases almost in proportion to the supply pressure P_s. Concerning the setting voltage e_a, if the amplitude of this voltage is too small, the self-excited oscillation does not occur properly because of the actuator's internal and external friction. Therefore, this voltage should be chosen large enough to generate the self-excited oscillation.

The identified dynamic parameters when the servovalve spool is displaced from the neutral position are shown in Fig. 12. The rotational speed of the hydraulic motor is varied from 0 rad/s to 120 rad/s continuously at the supply pressure P_s = 8 MPa and at the setting voltage of the non-linear element e_a = 1.5 V. This rotational speed range approximately corresponds to the displacement range of the servovalve spool. It is noted that the maximum rotational speed in this identified hydraulic servo motor system is 140 rad/s at the maximum spool displacement. The result at 0 rad/s is the same condition as the angular position self-excited oscillation system. It is confirmed that the dynamic parameters as the servovalve spool is displaced, are also identified continuously as well as when the spool is in the neutral position.

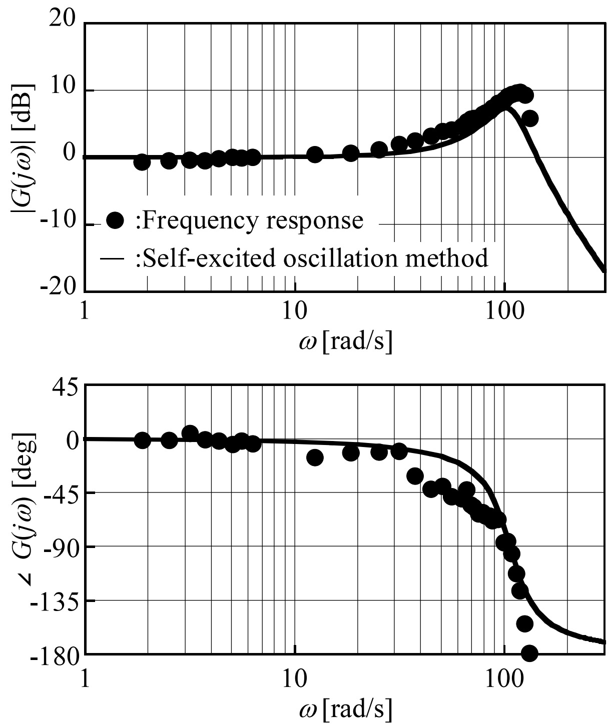


Fig. 14: *Comparison between self-excited oscillation method and frequency response (Roll actuator part of hydraulic motion seat, P_s = 7 MPa, e_a = 0.5 V)*

Table 2: Identified dynamic parameters for two degree of freedom hydraulic motion seat (P_s = 7 MPa, e_a = 0.5 V)

Axis	ζ		ω_n [rad/s]	
	SEO	FR	SEO	FR
Pitch actuator	0.45	0.41	145	151
Roll actuator	0.21	0.24	117	120

SEO: Self-excited oscillation method
FR: Frequency response method

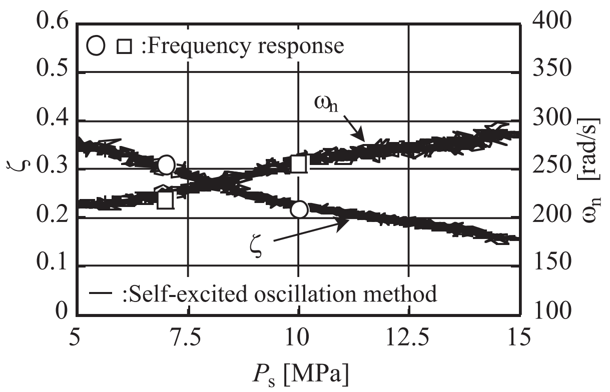


Fig. 15: *Identified dynamic parameters (rudder actuator part of aircraft tail control surface simulator, e_a = 0.1 V)*

In order to investigate the validity of the online identification results for the proposed method, comparison between the self-excited oscillation method and the frequency response method is carried out. Figure 13 shows the Bode plot of the identified hydraulic servo motor system at the supply pressure P_s = 8 MPa, the setting voltage of the non-linear element e_a = 1.5 V and the rotational speed 0, 70, 133 rad/s. In the figure lines show the self-excited oscillation method results obtained from Eq. 1 using the identified dynamic parameters ζ and ω_n; in the second plot, the points show the experimental results from the frequency response approach. Acceptable agreement is achieved between these two methods.

The self-excited oscillation method for the position control system is also applied to the two degree of freedom hydraulic motion seat and the aircraft tail control surface simulator in order to evaluate the method in more practical hydraulic applications.

The results of comparison between the proposed method and the frequency response method are shown in Fig. 14 and Table 2. The Bode diagram in Fig. 14 shows the frequency response of the roll actuator part of hydraulic motion seat at supply pressure P_s = 7 MPa and the setting voltage of the non-linear element e_a = 0.5 V. Table 2 shows the comparison of the obtained dynamic parameters for both pitch and roll actuators. From these results, it is clear that the identified dynamic parameters of the self-excited oscillation method coincide acceptably with the frequency response method for the both actuator axis of hydraulic motion seat.

Figure 15 shows the results of the online parameter estimation for the rudder actuator part of aircraft tail control surface simulator while the supply pressure P_s is continuously changing at the setting voltage e_a = 0.1 V. The two lines correspond to the dynamic parameters ζ and ω_n and the points show the measured results from the frequency response method. It can be again seen that both results are matched in a reasonable manner and the obtained dynamic parameters are dependent on the supply pressure P_s. As it is seen in the results until now, the proposed method enables a continuous identification of parameters, while the frequency response method can only identify the system dynamic parameters at one supply pressure condition at one time. Hence, this method is able to identify the system characteristics without repeated tests, even if the characteristics are varied by certain effects such as oil temperature rise, external force disturbance, and so on. In addition, online identification results may be valuable information for the online tuning of a controller, which optimizes the output of system, if an applied system allows a small oscillation of the present method. It should be noted that only several cycles of the self-excited oscillation wave is enough to identify the parameters. Hence the typical oscillation time for one identification is around 0.05 to 0.8s.

5 Conclusions

This paper has proposed an approach for identification of system dynamic parameters when the spool of the servovalve is not in its neutral position. This approach was applied to several experimental systems and was successful in identifying system dynamic parameters. In the first instance, the method was revised for the hydraulic servo motor system which operates at a constant rotational speed. In this case, the servovalve is always opened to deliver flow and its spool is displaced from the neutral position. The angular velocity self-excited oscillation system was utilized for this system instead of the position self-excited oscillation system used in the previous reports. Then, it was clarified that the proposed identification method was able to identify the dynamic parameters of the hydraulic servo motor system, when the rotational speed was varied continuously. From the results examined for the revised method, it was verified that the proposed identification method has the potential for real time parameter estimation.

The usefulness of this method when applied to a system configuration which is quite common in practice was examined. Two applications which are the two degree of freedom hydraulic motion seat and the aircraft tail control surface simulator were chosen as the plant to be identified. The dynamic parameters for both applications were well acquired with the online identification for the position self-excited oscillation system. It was confirmed that the identified results coincided with the measured data obtained from the frequency response method.

Nomenclature

e	deviation signal	
e_a	setting voltage of non-linear element	[V]
e_c	error signal at stability limit	
G_L	transfer function of hydraulic servo motor system	
G_N	transfer function of non-linear element	
J_m	load inertia	[kg m]
K_L	gain of hydraulic servo motor system	[(rad/s)/V]
K_N	variable gain of non-linear element	
m	load mass	[kg]
P_s	supply pressure	[MPa]
P_t	tank pressure	[MPa]
V	servo amplifier input voltage	[V]
V_s	amplitude of self-excited oscillation wave	[V]
y	cylinder position	[m]
v_i	reference input	
x	spool displacement of servovalve	[m]
Γ	amplitude correction factor	
θ	angular position	[rad]
ζ	dumping coefficient	
ω	angular velocity	[rad/s]
ω_n	undamped natural frequency	[rad/s]
ω_s	angular frequency of self-excited oscillation wave	[rad/s]
ξ	frequency correction factor	

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