An Electrohydraulic Position Servo Control Systems Using the Optimal Feedforward Integral Variable Structure Controller

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Abstract: An Optimal Feedforward Integral Variable Structure or FIVSC approach for an electrohydraulic position servo control system is presented in this paper. The FIVSC algorithm combines feedforward strategy and integral in the conventional Variable Structure Control (VSC) and calculating the control function to guarantee the existence of a sliding mode. Furthermore, the chattering in the control signal is suppressed by replacing the sign function in the control function with a smoothing function. The simulation results illustrate that the purposed approach gives a significant improvement on the tracking performances when compared with some existing control methods, like the IVSC and MIVSC strategies. Simulation results illustrate that the purposed approach can achieve a zero steady state error for ramp input and has an optimal motion with respect to a quadratic performance index. Moreover, Its can achieve accurate servo tracking in the presence of plant parameter variation and external load disturbances.

Keywords: Electrohydraulic, Variable Structure Control.

1. INTRODUCTION

Processes requiring large driving forces or torques are often actuated by hydraulic servo system. The dynamic characteristics of such systems are complex and highly nonlinear due to the flow pressure relationship of the hydraulic components. For a practical control system, it is usually desired to have a fast accurate response with a small overshoot. Due to the nonlinear dynamic property of hydraulic servo valves, it is not easy to design the control system of hydraulic position servos with a simple linear controller.

In certain case, a variable structure control (VSC) systems or sliding mode control (SMC) make use of linear control law to a linear or nonlinear systems. The linear law is defined by a linear function and switch function both from a linear combination of sates such that it defines the desired performance measure of the closed loop system. The performance measure is maintained by keeping a switching function as close as possible to zero by dynamically switched feedback gains. The linear function defines a discontinuity plane, termed as the sliding mode and any derivations from it are switched to direct the motion towards and thus ensure a desired system performance. Thus the VSC is invariant to system parameter variations and disturbances when the sliding mode occurs [1-3]. Because of its simple construction, high reliability and fast response without overshoot. The sliding mode operation results in a control system that is robust to model certainties, parameter variations and disturbances. Although the conventional VSC approach has been applied successfully in many applications, but it may result in a steady state error when there is load disturbance in it. In order to improve the problem, the integral variable structure control (IVSC) is presented in [4-6], combines and integral controller with the variable structure control. The IVSC approach comprises an integral controller for achieving a zero steady state error under step input and a VSC for enhancing the robustness. However, its performance when changing, e.g., ramp command input, the IVSC gives a steady state error. The Modified Integral Variable Structure Control (MIVSC), proposed in [7], uses a double-integral action to solve this problem and improve the dynamics response for command tracking. Although, the MIVSC method can give a better tracking performance than the IVSC method does at steady state, its performance during transient period needs to be improved.

In this paper, The design and simulation of an electrohydraulic position servo control systems using the Optimal Feedforward Integral Variable Structure or FIVSC approach is described. The design of a FIVSC system involves : 1) the choice of the control function to guarantee the existence of a sliding motion and 2) the determination of the switching function and the integral control gain such that the system has desired properties. The advantage of this approach is that the error trajectory in the sliding motion can be prescribed by the design. Also, it can achieve a rather accurate servo tracking and is fairly robust to plant parameter variations and external load disturbances. As a simulation results, the tracking performance can be remarkably improved and is fairly robust to plant parameter variations and external load disturbances.

2. DESIGN OF THE FIVSC SYSTEM

The structure of the FIVSC is shown in Fig. 1. It combines the conventional VSC with an integral compensator and a feedforward path from the input command. The FIVSC system can be described as follows [8]

$$\dot{X}_i = X_{i+1} \qquad ,i = 1,\ldots,n\text{-}1 \tag{1a}$$

$$\dot{X}_n = -\sum_{i=1}^{n} a_i X_i + bU - f(t) \tag{1b}$$

$$\dot{Z} = (r - X_1) \tag{1c}$$

where X_1 is the output signal,
r is the input command,
k_I is the gains of the integral compensator,
U is a piecewise linear function,
a_i and b are the plant parameters and
$f(t)$ are disturbances.

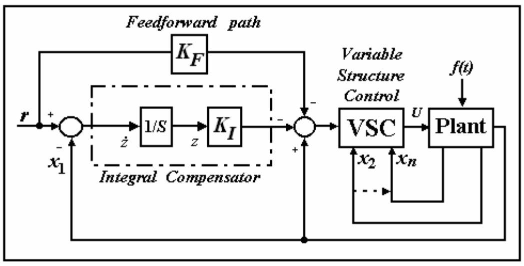

Fig. 1. The structure of FIVSC system.

The switching function, σ is given by

$$\sigma = c_1(X_1 - K_I z - rK_F) + \sum_{i=2}^{n} c_i X_i \tag{2}$$

where $C_i > 0$=constant, C_n=1 and K_I is the integral control gain.
The design of such system involves :

1) the choice of the control function, U so that it gives rise to the existence of a sliding mode control
2) the determination of the switching function, σ and the integral control gain K_I such the system has the desired eigenvalues
3) the elimination of chattering phenomena of the control signal using the smoothing function.

The control signal, U can be determined as follows. From Eq. (1) and Eq. (2), We have

$$\dot{\sigma} = -c_1(K_I\dot{Z}) + \sum_{i=2}^{n} c_{i-1}X_i - \sum_{i=1}^{n} a_i X_i + bU - f(t) \cdot \tag{3}$$

Let $a_i = a_i^0 + \Delta a_i$; $i = 1,\ldots,n$
and $b = b^0 + \Delta b$; $b^0 > 0$, $\Delta b > -b^0$
where a_i^0 and b^0 are nominal values of a_i and b and Δa_i and Δb are the variations of a_i and b, respectively.

The control signal can be separated into

$$U = U_{eq} + \Delta U \tag{4}$$

where the so called equivalent control U_{eq} is defined as the solution of Eq. (4) under the condition where there is no disturbances and no parameter variations,

that is $\dot{\sigma} = 0$, $f(t) = 0$, $a_i = a_i^0$, $b = b^0$ and $U = U_{eq}$.

This condition results in

$$U_{eq} = \left\{c_1(K_I\dot{Z}) - \sum_{i-2}^{n-1} c_{i-1}X_i + \sum_{i=1}^{n-1} a_i^0 X_i + (c_{i-1} - a_n^0)(-X_n)\right\} / b^0 \cdot \tag{5}$$

But under a sliding mode $\sigma = 0$, from Eq. (3) we have,

$$X_n = -c_1(X_1 - K_I Z - rK_F) - \sum_{i=2}^{n-1} c_i X_i \tag{6}$$

Substitution of Eq. (6) into Eq. (5) yields

$$U_{eq} = \left\{ c_1(K_I\dot{Z}) - \sum_{i-2}^{n-1} c_{i-1}X_i + \sum_{i=1}^{n-1} a_i^0 X_i + (c_{i-1} - a_n^0)\left[c_1(X_1 - K_I Z - rK_F) + \sum_{i-2}^{n-1} c_i X_i\right] \right\} / b^0 \tag{7}$$

ΔU is required to guarantee the existence of the sliding mode under the plant parameter variations in Δa_i and Δb and the disturbances $f(t)$.

ΔU can be obtained from Eq. (2) where σ and C_i are replaced by ΔU and φ respectively.

That is,
$$\Delta U = \varphi_1(X_1 - K_I Z - rK_F) + \sum_{i=2}^{n} \varphi_i X_i + \varphi_{n+1} \tag{8}$$

where

$$\varphi_1 = \begin{cases} \alpha_1 & \text{if } (X_1 - K_I Z - rK_F)\sigma > 0 \\ \beta_1 & \text{if } (X_1 - K_I Z - rK_F)\sigma < 0 \end{cases}$$

$$\varphi_i = \begin{cases} \alpha_i & \text{if } X_i\sigma > 0 \\ \beta_i & \text{if } X_i\sigma < 0 \end{cases} \quad , i = 2,\ldots,n$$

and

$$\Phi = \begin{cases} \gamma & \text{if } \sigma > 0 \\ \delta & \text{if } \sigma < 0 \end{cases} .$$

The condition for the existence of a sliding mode is known to be

$$\sigma\dot{\sigma} < 0 . \tag{9}$$

Substitute Eq. (5) and Eq. (8) into Eq. (4) to obtain

$$\begin{aligned} \dot{\sigma} = & -\sum_{i=1}^{n} \Delta a_i X_i - f(t) + (c_{n-1} - a_n)X_n + (c_{n-1} - a_n^0) \\ & \times [c_1(X_1 - K_I Z - rK_F) + \sum_{i=2}^{n-1} c_i X_i \\ & + \frac{\Delta b}{b^0}\Big\{ c_1(K_I\dot{Z}) - \sum_{i=2}^{n-1} c_{i-1}X_i + \sum_{i=1}^{n-1} a_i^0 X_i + (c_{n-1} - a_n^0) \\ & \times [c_1(X_1 - K_I Z - rK_F) + \sum_{i=2}^{n-1} c_i X_i]\Big\} \\ & + b[\varphi_1(X_1 - K_I Z - rK_F) + \sum_{i=2}^{n} \varphi_i X_i + \varphi_{n+1}] \end{aligned} \tag{10}$$

and then

$$\begin{aligned} \dot{\sigma}\sigma = & [-\Delta a_1 + a_1^0 \Delta b / b^0 + c_1(c_{n-1} - a_n^0)(1 + \Delta b / b^0) \\ & + b\varphi_1](X_1 - K_I Z - rK_F)\sigma \\ & + \sum_{i=2}^{n-1}\big\{[-\Delta a_i + a_i^0 \Delta b / b^0 - c_{i-1}\Delta b / b^0 \\ & + c_i(c_{n-1} - a_n^0)(1 + \Delta b / b^0) + b\varphi_i]X_i\sigma\big\} \\ & + [-\Delta a_n + (c_{n-1} - a_n^0) + b\varphi_n]X_n\sigma \\ & + [N + b\varphi_{n+1}]\sigma \end{aligned} \tag{11}$$

where $N = -(K_I Z)(\Delta a_1 - a_1^0 \Delta b / b^0) + \Delta b / b^0 [c_1(K_I\dot{Z})] - f(t) \cdot$

In order for Eq. (9) to be satisfied, the following conditions must be met,

$$\varphi_i = \begin{cases} \alpha_i < \text{Inf } [\Delta a_i - a_i^0 \Delta b / b^0 + c_{i-1}\Delta b / b^0 \\ \qquad - c_i(c_{n-1} - a_n^0)(1 + \Delta b / b^0)] / b \\ \beta_i > \text{Sup}[\Delta a_i - a_i^0 \Delta b / b^0 + c_{i-1}\Delta b / b^0 \\ \qquad - c_i(c_{n-1} - a_n^0)(1 + \Delta b / b^0)] / b \end{cases} \tag{12a}$$

where $i = 1,\ldots,n$-1, $c_0 = 0$

$$\varphi_n = \begin{cases} \alpha_n < \text{Inf } [\Delta a_n + a_n^0 - c_{n-1}] / b \\ \beta_n > \text{Sup}[\Delta a_n + a_n^0 - c_{n-1}] / b \end{cases} \tag{12b}$$

and where
$$\Phi = \begin{cases} \gamma < \text{Inf } [-N] / b \\ \delta > \text{Sup}[-N] / b \end{cases} . \tag{12c}$$

However, the term $N(t)$ may not be neglected in the presence of in put commands, plant parameter variations and/or external disturbance. Hence once the effect of term $N(t)$ exceeds the sum of other terms in Eq. (11) such that inequality Eq. (9) is violated, then the sliding mode breaks down and the system gives rise to a limit cycle. Fortunately, by increasing the control gain φ_i, the effect due to the term $N(t)$ can be arbitrarily suppressed so that the magnitude of the limit cycle can be reduced to within a tolerable range, the validity of the assumption can be shown by the simulation and a quasi-ideal sliding motion can be obtained. Under the sliding motion, the system described by Eq. (1) can be reduced to

$$\dot{X}_i = X_{i+1} \qquad ,i = 1,\ldots,n\text{-}2 \tag{13a}$$

$$\dot{X}_{n-1} = -\sum_{i=1}^{n} c_i X_i + c_1 K_I Z \tag{13b}$$

$$Z = (r - X_1)\,. \tag{13c}$$

or, in the matrix form,

$$\dot{X} = AX + BV + Er \tag{14a}$$

$$V = GX \tag{14b}$$

where

$$X = \begin{bmatrix} Z \\ X_1 \\ \vdots \\ X_{n-1} \end{bmatrix}_{n\times 1}, A = \begin{bmatrix} 0 & -1 & 0 & \cdots & 0 \\ 0 & 0 & 1 & & 0 \\ \vdots & & & & \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}_{n\times n}$$

$$B = \begin{bmatrix} 0 \\ 0 \\ \vdots \\ 1 \end{bmatrix}_{n\times 1}, E = \begin{bmatrix} 1 \\ 0 \\ \vdots \\ 0 \end{bmatrix}_{n\times 1}$$

and

$$G = \begin{bmatrix} c_1 K_I & -c_1 & -c_2 & \cdots & -c_{n-1} \end{bmatrix}_{1\times n}.$$

In order to find the optimal gain matrix G by means of the optimal linear regulator technique, the quadratic index I as shown in the following equation must be minimized :

$$I = \frac{1}{2}\int_{s}^{\infty}\left(X^T Q^T X + V^T R V\right) \tag{15}$$

where $Q = Q^T \rangle 0$ and $R = R^T \rangle 0$ are weighting matrices and t_s is the time for the sliding mode begins.

The weighting matrix Q can be chosen as

$$Q = D^T D \tag{16}$$

where D is a 1xn vector and the pair (A,D) is observable.

Then the optimal gain matrix G is given

$$G = -R^{-1} B^T P \tag{17}$$

where P is the solution of the matrix Riccati's equation

$$PA + A^T P - PBR^{-1} B^T P + Q = 0. \tag{18}$$

Let φ_i ,$i = 1,\ldots,n+1$, be chosen as $\varphi_i = \alpha_i = -\beta_i$.

Finally, the control function of FIVSC approach for simulate is obtained as

$$\begin{aligned} U = \Bigg\{ & c_1(K_I \dot{Z} - \sum_{i=2}^{n-1} c_{i-1} X_i + \sum_{i=1}^{n-1} a_i^0 X_i \\ & + (c_{n-1} - a_n^0)\Big[c_1(X_1 - K_I Z - rK_F) + \sum_{i=2}^{n-1} c_i X_i\Big]\Bigg\} / b^0 \\ & + \Big(\varphi_1 \left|X_1 - K_I Z - rK_F\right| + \sum_{i=2}^{n} \varphi_i \left|X_i\right| + \varphi_{n+1}\Big) sign\,(\sigma) \end{aligned} \tag{19}$$

where $\varphi_i \langle -\text{Sup}\left|\begin{matrix}[\Delta a_i - a_i^0 \Delta b / b^0 + c_{i-1} \Delta b / b^0 \\ -c_i (c_{n-1} - a_n^0)(1 + \Delta b / b^0)] / b\end{matrix}\right|$

$$\varphi_n \langle -\text{Sup}\left|[\Delta a_n + a_n^0 - c_{n-1}] / b\right|$$

and $\varphi_{n+1} \langle -\text{Sup}\left|[N] / b\right|$.

The transfer function when the system is on the sliding surface can be shown as

$$H(s) = \frac{X_1(s)}{R(s)} = \frac{c_1 K_I}{s^n + c_{n-1}s^{n-1} + \ldots + c_1 s + c_1 K_I}\,. \tag{20}$$

Using the final value theorem, it can be shown from Eq. (20), that the steady-state tracking error due to a ramp command input is zero. This is the result of the integral action. Furthermore, the zeros of the transfer function Eq. (20), which is the result of the feedforward path will give rise to the improvement on tracking performance during the transient period. The transient response of the system can be determined by suitably selecting the poles of the transfer function.

Let $\qquad S^n + \alpha_1 S^{n-1} + \ldots + \alpha_{n-1} S + \alpha_n = 0 \qquad (21)$

be the desired characteristic equation(closed-loop poles), the coefficient C_1 and K_I can be obtained by

$$C_{n-1} = \alpha_1,$$
$$C_1 = \alpha_{n-2},$$
$$K_I = \alpha_n / C_1.$$

Normally, the sign function, sign (σ) in Eq. (19), direct application of such a control signal, U to the plant will give rise to chatterings. In order to reduce chattering phenomena in the switching action, the term sign (σ) can be replaced by a continuous function [6], given by

$$M_\delta(\sigma) = \frac{\sigma}{|\sigma| + \delta_0 + \delta_1 |e_1|} \tag{22}$$

and on the basis of the practical view, will be defined as a function of $|e_1|$ instead of a constant, that is

$$\delta = \delta_0 + \delta_1 |e_1|$$

where δ_0, δ_1 are positive constants.

3. DYNAMIC MODELING OF AN ELECTROHYDRAULIC POSITION SERVO CONTROL SYSTEMS

The block diagram of the electrohydraulic position servo control systems to be studied is shown is Fig. 2. The relation between the valve displacement X_v and the flow rate Q_L is described as [9]

$$Q_L = X_v K_j \sqrt{P_s - sign(X_v) P_L} = X_v K_s \tag{23}$$

where K_j is a constant for a specific hydraulic motor;
P_s is the supply pressure;
P_L is the load pressure and
K_s is the valve flow gain that varies under different operating points.

The flow continuity property of the motor chamber yields

$$Q_L = D_m \omega_c + K_{ce} P_L + \left(\frac{V_t}{4\beta} \right) \dot{P}_L \tag{24}$$

where D_m is the volumetric displacement;
K_{ce} is the total leakage coefficient;
V_t is the total volume of the oil;
β is the bulk modulus of the oil and
ω_c is the velocity of the motor shaft.

The torque balance equation for the motor is given by

$$D_m P_L = J \dot{\omega}_c + B_m \omega_c + T_L \tag{25}$$

where B_m is the viscous damping coefficient;
J is the inertia of the motor and
T_L is the load disturbance.

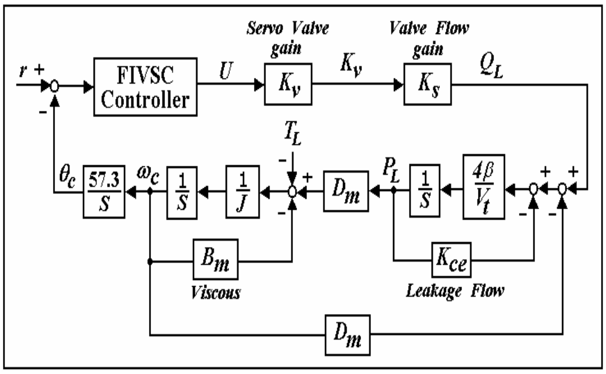

Fig. 2. The electrohydraulic position servo systems using FIVSC controller.

4. THE FIVSC SYSTEM FOR AN ELECTROHYDRAULIC SYSTEMS

The nominal values of the electrohydraulic parameters and the FIVSC controller are listed in Table. 1 and Table. 2, respectively. Based on the block diagram as shown in Fig. 2, by combining Eqs. (23)~(25), the following set of state equations can be obtained :

$$\dot{X}_1 = X_2 \tag{26a}$$

$$\dot{X}_2 = X_3 \tag{26b}$$

$$\dot{X}_3 = -a_2 X_2 - a_3 X_3 + BU - f(t) \tag{26c}$$

$$\dot{Z} = r - X_1 \tag{26d}$$

where

$$a_2 = \frac{4\beta}{V_t}\frac{D_m^2}{J} + \frac{4\beta}{V_t}\frac{B_m}{J} K_{ce}$$

$$a_3 = \frac{B_m}{J} + \frac{4\beta}{V_t} K_{ce}$$

$$b = 57.3 K_v K_s \frac{4\beta}{V_t}\frac{D_m}{J}$$

$$f(t) = 57.3 \frac{4\beta}{V_t}\frac{K_{ce}}{J} T_L + 57.3 \frac{1}{J} \dot{T}_L$$

$X_1 = \theta_c$ is the position of the motor shaft and
$r = \theta_r$ is the reference input.

Following the design procedure given in the section 2, one obtains the control function

$$U = U_{eq} + \Delta U = \{ [c_1 K_I (r - X_1) - c_1 X_2 - c_2 X_3 + a_2^0 X_2 + a_3^0 X_3] \} / b^0 \\ + (\varphi_1 | X_1 - K_1 Z - r K_F | + \varphi_2 |X_2| + \varphi_3 |X_3| + \Phi) M_\delta(\sigma) \tag{27}$$

where $\varphi_i \langle - \left| [\Delta a_i - a_i^0 \Delta b / b^0 - c_{i-1} \Delta b / b^0)] / b \right|$
i=1,…,3 c_o=0

and $\Phi \langle - \left| [N] / b \right|$

with the switching function,

$$\sigma = c_1 (X_1 - K_I Z - r K_F) + c_2 X_2 + X_3 \tag{28}$$

and by suitably choosing Q and R, one can obtain the optimal gains of C_1, C_2 and K_I.

The weighting matrices are chosen as

$$Q = \begin{bmatrix} 10^5 & 0 & 0 \\ 0 & 50 & 0 \\ 0 & 0 & 0.1 \end{bmatrix} \text{ and } R = 10^{-5}$$

Then, from Eq. (17), the optimal gain matrix can be obtained as

$$G = \begin{bmatrix} -10^5 & -3565.8 & -97.46 \end{bmatrix}.$$

The robustness of the proposed FIVSC approach against large variations of plant parameters and external load disturbances has been simulated for demonstration.

Table. 1. System parameters for simulation.

Parameter	Value	Dimension
K_s	$0.03\sqrt{P_s - sign(X_v)P_L}$	in^2/s
P_s	2000	psi
β	50000	psi
V_t	2.0	in^3
K_{ce}	0.001	$in^3/s/psi$
D_m	1.0	in^3/rad
J	0.5	$in\text{-}lb\text{-}s^2/rad$
B_m	75	in-lb-s/rad
K_v	20	in/V

Table. 2. Parameters of FIVSC controller.

Parameter	Value
λ_1	-4.796+29.656i
λ_2	-4.796-29.656i
λ_3	-18.546
λ_4	-4.392
C_1	255.8
C_2	77.46
K_I	18
K_F	43
φ_1	-1
φ_2	-0.01
φ_3	-0.000015
Φ	-0.002
a_2^0	12745
a_3^0	654
b^0	24356
δ_0	5
δ_1	50

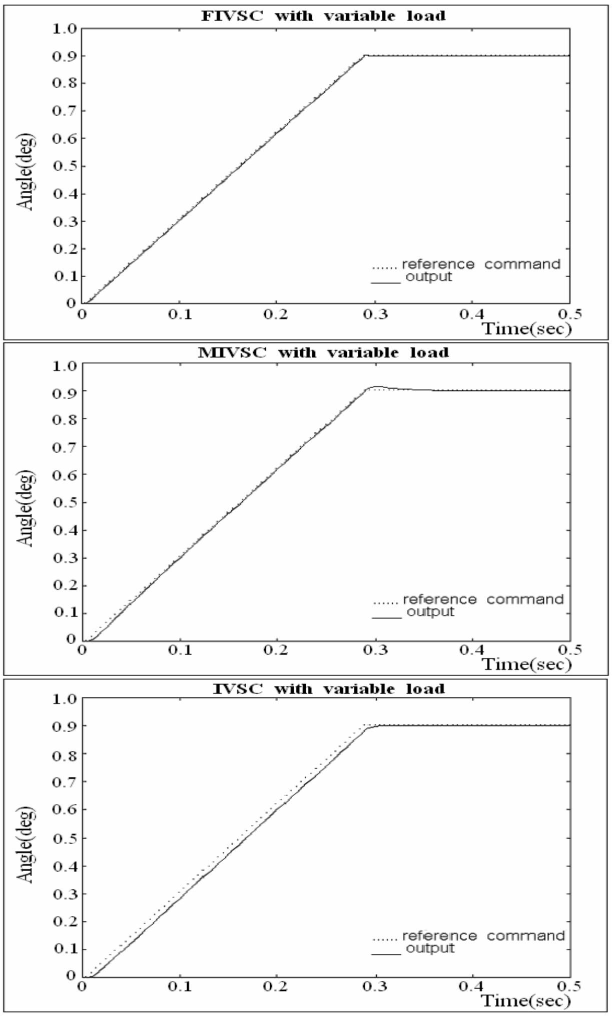

Fig. 3. Comparison of tracking performance under random deviation of load disturbance T_L and variations of plant parameters K_v and J.

5. SIMULATION RESULTS AND DISCUSSIONS

The simulation results of the dynamic responses (angle) are shown in Fig. 3 and Fig. 4, where a ramp command input is introduced. In addition, the electrohydraulic is applied with a shaft angle-dependent external load disturbance T_L and variations of plant parameters K_v and J. The results are compared with those obtained from the IVSC and MIVSC approaches. These curves illustrate the robustness of the FIVSC for electrohydraulic under various loads and abrupt disturbance. It is clear from the figures that FIVSC approach can be maintained almost identically but vary significantly for other approaches. Fig. 4, shows the comparison of tracking errors and control signals under the same testing conditions. The smooth curve of the control function with the proper smoothing function clearly indicates that the smoothing function can eliminate chattering. From the observation, it is obvious that the proposed approach gives the minimum tracking error. That is, it gives a minimal tracking error and it also tracks the command input very closely during the change of the command input. Among them, the IVSC approach performs poorly. It gives a substantially sustained tracking error. Thus, the proposed approach seems amenable for practical implementation.

6. CONCLUSIONS

This paper described a position servo control systems for an electrohydraulic using the FIVSC approach. The system combines the nonlinear integral variable structure control with additional feedforward controller. The control function has derived the conditions that ensure the existence of a sliding mode control. Procedures are developed for choosing the control function for determining the coefficients of the switching plane and the integral control gain such that the resultant system has the desired properties. The application of FIVSC to an electrohydraulic has show that the proposed approach can improved the tracking performance by 65% and 80% when compared to the MIVSC and IVSC approaches. Furthermore, the simulation results demonstrate that the proposed approach can achieve the requirements of robustness in the presence of plant parameter variation, load variations and nonlinear dynamic interactions. It is a robust and practical control law for an electrohydraulic servo systems.

7. ACKNOWLEDGEMENT

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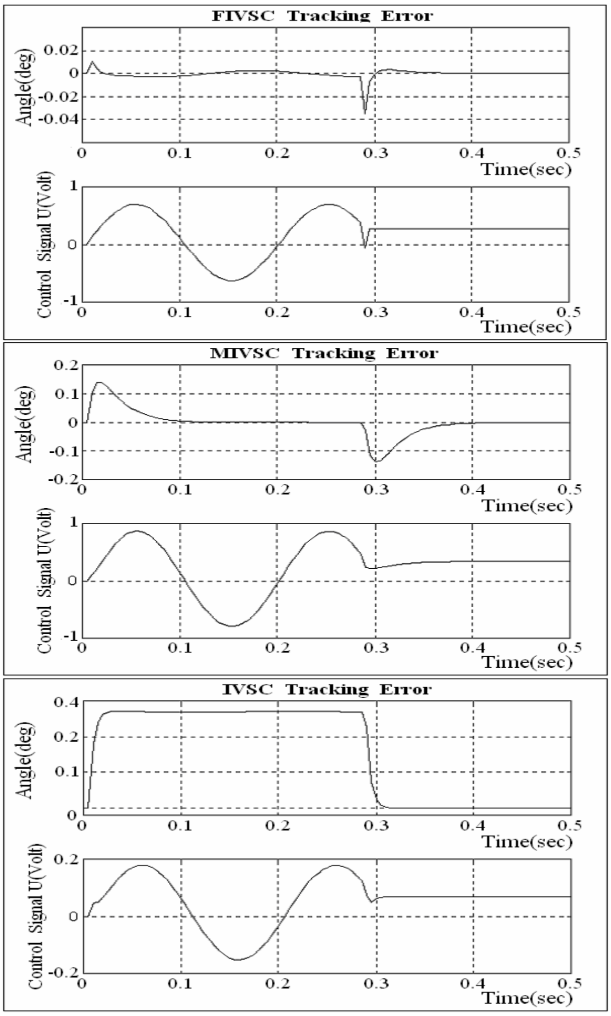

Fig. 4. Comparison of tracking errors and control signals under random deviation of load disturbance T_L and variations of plant parameters K_v and J.

REFERENCES

[1] U. Itkis. "Control Systems of Variable Structure", *John Wiley & Sons*, New York, Wiley, 1976.

[2] V.I. Utkin, J. Guldner and J. Shi, "Sliding Mode Control in Electromechanical Systems", *Taylor & Francis*, London, 1999.

[3] H.Y. John, G. Weibing and C.H. James, "Variable Structure Control : A Survey", *IEEE Transaction on Industrial Electronic*, Vol.40, No.1, pp.1-22, 1993.

[4] T.L. Chern and J. Chang, "DSP_Based Induction Motor Drivers Using Integral Variable Structure Model Following Control Approach", *IEEE International Electric Machine and Drives Conference Record*, pp.9.1-9.3, 1997.

[5] T.L. Chern and Y.C. Wu, "Design of brushless DC position servo system using integral variable structure approach", *IEE proceeding-b*, Vol.140, No.1, pp.27-34, 1993.

[6] T.L. Chern and J.S. Wong, "DSP based integral variable structure control for DC motor servo drivers", *IEE proc.-Control Theory Application*, Vol.142, No.5, pp.444-450, 1995.

[7] S. Nungam and P. Daungkua, "Modified Integral Variable Structure Control for Brushless DC Servomotor", *21st Electrical Engineering Conference (EECon-21)*, Thailand, pp.138-141, 1998.

[8] S.Nungam and P.Phakamach, "Design of a Feedforward Integral Variable Structure Control Systems and Application to a Brushless DC Motor Control", *IEEE International Symposium on Intelligent Signal Processing and Communication Systems*, Thailand, Dec. pp.390-393, 1999.

[9] Merrit, H.E. "Hydraulic Control System", *John Wiley & Sons*, New York, 1967.