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REVIEW ON SLIDING MODE CONTROL AND ITS APPLICATION IN ELECTROHYDRAULIC ACTUATOR SYSTEM

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ABSTRACT

The purpose of this paper is to review the literatures related to the modeling of sliding mode control strategies of hydraulic actuator systems proposed by various researchers in order to design a high performance nonlinear controller in the presence of uncertainties. Sliding mode controller (SMC) is one of the nonlinear robust controllers which can be used in nonlinear dynamic systems with uncertainty. Before the main discussion, background information related to the hydraulic actuators will be presented. This review is concluded with a short summary and conclusion of hydraulic actuators.

Keywords: *Hydraulic Actuator, Variable Structure Control,Terminal Sliding Mode, Integral Sliding Mode, Super Twisting Sliding Mode.*

1. INTRODUCTION

An Electro-Hydraulic Actuator (EHA) system is one of the important drive systems in industrial sectors and most engineering practices due to its high power to weight ratio and stiffness response being good, smooth and fast. Recently, with the research and development of mathematics, control theory, computer technology, electronic technology and basic theory of hydraulics, hydraulic control technology has been developed and used widely in many applications such as manufacturing systems, material testing machines, active suspension systems, mining machineries, fatigue testing, flight simulation, paper machines, ships and electromagnetic marine engineering, injection moulding machines, robotics, and steel and aluminium mill equipment[1]. Due to its applications, the highest performance of the electro-hydraulic actuators in terms of position, force or pressure is needed. However, the system is highly nonlinear and nondifferentiable due to many factors, such as leakage, friction, and especially, the fluid flow expression through the servo valve[2]. In this paper, a review of sliding mode control (SMC) for EHA has been presented. Compare to others paper review, the main issues surmounted by each types of sliding mode control are also highlighted. The main body of the review deals with (II) hydraulic actuator motivation, (III) modeling of electro-hydraulic actuator, (IV) Sliding Mode Control, (V) Sliding Mode Control Implementation in EHA System and the final part (VI) is the short summary and conclusion.

2. MOTIVATION

Hydraulic actuators are used in many engineering applications due its redundancy (high power-to-weight ratio, fast and smooth response, high stiffness and good positioning capability), therefore, developing advanced control methods for these systems is relevant[3]. This has made the hydraulic actuator a focus of study whereby a variety of control algorithms have been proposed in order to overcome its nonlinear dynamic behaviour. The hydraulic actuator system is the most appropriate choice for an active suspension system[4], due to its low construction and

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maintenance cost with high power-to-weight ratio. The hydraulic actuator system also has the ability to produce a very large force and torque in any system[5]. Some of the examples of hydraulic actuator system application that require large force and torque are electro-hydraulic positioning system, industrial hydraulic machines[6], robot manipulators [7] and hydraulic elevators[8]. Due to high precision position controllers, hydraulic actuator systems are applied in specialized manufacturing equipment or test equipment such as simple shear apparatus utilized for soil testing[9]. Although there are a number of advantages for applications that utilize hydraulic actuator systems, there are some weaknesses that complicate the development of hydraulic actuator system controller as the system is a highly nonlinear system. Aside from the nonlinear behaviour, the hydraulic actuator system also suffers from a large extent of model uncertainties. The uncertainties can be classified into two main groups which are parametric uncertainties and uncertain nonlinearities. Examples of parametric uncertainties are large variations in load and hydraulic parameters such as bulk modulus due to component wear or temperature change. Meanwhile, external disturbance, leakage and friction are called uncertain nonlinearities[10]. These uncertainties can cause the hydraulic actuator system controller to be unstable or to have degradation in its performance.

3. MODELLING OF EHA

Hydraulic actuator can be modeled from theoretical mathematical analysis or system identification. Many approaches have been proposed for hydraulic actuator modeling. System modeling can be based on the system of physical law or system identification method which is formally known as black box identification. Performing a system of physical law requires expert knowledge and understanding about the system itself. This section presents the review for theoretical mathematical model of EHA.

The actuator dynamic equation of electro-hydraulic actuator servo system is expressed as[11]

$$m\ddot{x}_p = SP_L - f\dot{x}_p - kx_p - F_L \qquad (1)$$

Where, m is load at the rod of the system, x_p is the displacement of the piston, P_L is the difference in pressure between two chambers, k is the coefficient of aerodynamic elastic force, f is the coefficient of viscous friction, S is the piston area and F_L is the external disturbance injected into the system's actuator. With the assumption that a high-response servo valve is used in the system, the control applied to the spool valve is proportional to the spool position. Its equation is given as:

$$x_v = k_v u \qquad (2)$$

Where x_v is the opening of the valve, k_v is the coefficient of the servo valve and u is the input voltage.

Assume that the system is a symmetrical cylinder, therefore, both piston area and volume for each port are similar. Thus, the dynamics of cylinder oil flow can be expressed as follows:

$$Q_L = \dot{P}_L + \frac{2\ \ s}{v}\dot{x}_p \qquad (3)$$

Where Q_L is the difference between supplied flow rate to the chambers, v is the volume of the chamber and is the effective bulk modulus of the fluid. Thus the difference of the flow rate to the chambers is given as:

$$Q_L = \frac{2}{v}\left[c_d w\sqrt{\frac{P_a - P_L}{\circ}}x_v - k_l P_L\right] \qquad (4)$$

Where c_d is the coefficient of the volumetric flow of the valve port, P_a is the supply pressure, $\circ$ is the oil density and k_l is the coefficient of internal leakage between the cylinder chambers. Let $x_1 = x_p$, $x_2 = \dot{x}_p$ and $x_3 = P_L$.

$$\dot{x}_1 = \dot{x}_p = x_2 \qquad (5)$$

$$\dot{x}_2 = \ddot{x}_p \qquad (6)$$

Referring (1),

$$\ddot{x}_p = \frac{s}{m}P_L - \frac{f}{m}\dot{x}_p - \frac{k}{m}x_p - \frac{F_L}{m} \qquad (7)$$

Thus,

$$x_2 = \frac{s}{m}P_L - \frac{f}{m}\dot{x}_p - \frac{k}{m}x_p - \frac{F_L}{m} \qquad (8)$$

$$\dot{x}_3 = \dot{P}_L \qquad (9)$$

From (3),

$$P_L = Q_L - \frac{2\ \ s}{v}\dot{x}_p \qquad (10)$$

Substituting (4) into (10), thus (10) becomes

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$$\dot{P}_L = \frac{2}{v} c_d w \sqrt{\frac{P_a - P_L}{\circ}} x_v - \frac{2}{v} k_l P_l - \frac{2\ s}{v} \dot{x}_p \qquad (11)$$

As a result, the differential equations governing the dynamics of electro-hydraulic actuator servo system with external disturbance injected to its actuator is given as

$$\dot{x}_1 = x_2 \qquad (12)$$

$$\dot{x}_2 = -\frac{k}{m} x_1 - \frac{f}{m} x_2 + \frac{s}{m} x_3 - \frac{F_L}{m} \qquad (13)$$

$$\dot{x}_3 = -\frac{s}{k_c} x_2 - \frac{k_l}{k_c} x_3 + \frac{c}{k_c} \sqrt{\frac{P_a - x_3}{2}} k_v \qquad (14)$$

Where x_1 is the displacement of the load, x_2 is the load velocity and x_2 is the differential pressure $^{\circ}{}_1 - {}^{\circ}{}_2$ between the cylinder chambers caused by load. F_L is the external disturbance given to the system and it can be constant or a time varying signal. $\sqrt{\frac{P_a - x_3}{2}} k_v u$ is the challenge of this system where input depends on the square root of the state x_3. Table 1 shows the parameters of electro hydraulic actuator servo system which are represented by equations (12), (13) and (14).

Table 1: Parameter of electro-hydraulic actuator servo system

Parameter	Value
Load at the EHA rod, m	$0.33 Ns^2 / cm$
Piston Area, S	$10 cm^2$
Coefficient of viscious friction, f	$27.5 Ns / cm$
Coefficient of aerodynamic elastic force, k	$1000 N / cm$
Valve port width, w	$0.05 cm$
Supply pressure, P_a	$2100 N / cm^2$
Coefficient of volumetric flow of the valve port, c_d	0.63
Coefficient of internal leakage between the cylinder chambers, k_l	$2.38 \times 10^{-3} cm^5 / Ns$
Coefficient of servo valve, k_v	$0.017 cm / V$
Coefficient involving bulk modulus and EHA volume, k_c	$2.5 \times 10^{-4} cm^5 / N$
Oil density, $\circ$	$8.87 \times 10^{-7} Ns^2 / cm^4$

By substituting all the parameters into equations (12), (13) and (14), the system equations can be expressed as a product of matrix as below:

$$\dot{x}_{3x1} = A_{3x3} x_{3x1} + B_{3x1} u + C_{3x1}$$

Where

$$\dot{x} = [\dot{x}_1 \quad \dot{x}_2 \quad \dot{x}_3]^T$$

$$A = \begin{bmatrix} 0 & 1 & 0 \\ -3030.3 & -83.3 & 30.3 \\ 0 & -40000 & -9.52 \end{bmatrix}$$

$$x = [x_1 \quad x_2 \quad x_3]^T$$

$$B = \left[0 \quad 0 \quad 3196 \sqrt{\frac{2100 - x_3}{2}} \right]^T$$

$$C = \left[0 \quad -3.03 F_L \quad 0 \right]^T$$

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Figure 1 shows a schematic diagram of another example of a single rod, single ended hydraulic cylinder similar as that in[12]. The electro-hydraulic actuator system modelled in this paper consists of two main parts, i.e. the valve and the cylinder. The cylinder is modelled as a double acting single rod or single-ended piston, with a single load attached at the end of the piston.

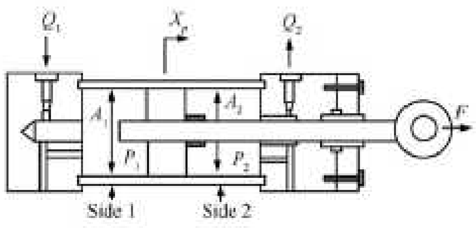

Figure 1: Electro-hydraulic servo system[12]

In Figure 1, X_p is the piston position, F denotes the applied load to the cylinder, Q_1 and Q_2 are the fluid flows to and from the cylinder, respectively. P_1 is the fluid pressure within side 1 and P_2 is the fluid pressure within side 2. The pressurized areas on side 1 and side 2 are A_1 and A_2 , respectively. The cylinder will retract or extend when a pressure difference between P_1 and P_2 occurs. The dynamics of the system are expressed by:

$$\dot{X}_p = V_p \tag{15}$$

$$ma_p = F_a - F_f \tag{16}$$

Where V_p is the piston velocity, a_p is the piston acceleration, and m is the piston and load mass. There are two forces in equation (16) that influence the system named the hydraulic actuating force, F_a and the friction force, F_f which are functions of nonlinearities that will significantly influence the system. The parameters that affect F_a are the control input voltage, environment load, cylinder pressure, friction force and leakage. Hence, it can be represented by:

$$F_a = A_p P_l \tag{17}$$

Therefore, the force balance equation of the cylinder is represented by:

$$ma_p = A_p P_l - F_f \tag{18}$$

where A_p is the cross section of the hydraulic cylinder, and P_l is the cylinder differential pressure written as:

$$P_l = P_1 - P_2 \tag{19}$$

Equation (18) represents the dynamics of the system. The load pressure P_l is defined to be the pressure across the actuator piston, the derivative of the load pressure is given by the total load flow through the actuator divided by the fluid capacitance as:

$$\frac{V_t}{{}_e}\dot{P}_l = Q_l - C_t P_l - A_p V_p \tag{20}$$

where V_t is the total actuator volume of both cylinder sides, B_e is the bulk modulus of hydraulic oil, C_t is the total leakage coefficient, and Q_l is the load flow. By using equation (20), the flow equality of the servo valve is given in (21), which expresses the relationship between the spool valve displacement X_v and the load flow Q_l .

$$Q_L = C_d W X_v \sqrt{\frac{P_s - \mathrm{sgn}(X_v) P_l}{\circ}} \tag{21}$$

where C_d is the discharge coefficient, W is the spool valve area gradient, P_s is the supply pressure, and $\circ$ is the oil density. Substituting equation (21) into (20), the hydraulics dynamics of the cylinder pressure can be written as

$$\dot{P}_l = \frac{4\,{}_e}{V_t}[-A_p V_p - C_t P_l + \ldots$$

$$\ldots + C_d W X_v \sqrt{\frac{P_s - \mathrm{sgn}(X_v) P_l}{\circ}}] \tag{22}$$

The spool displacement of the servo valve X_v is controlled by the control signal generated by the FLC .

The corresponding relation can be simplified into:

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$$\dot{X}_v = \frac{1}{2_v}(-X_v + k_v u) \tag{23}$$

The servo valve input can also be expressed as a second order equation as:

$$u = \frac{1}{k_v}\left[\frac{1}{W_v^{\ 2}}\ddot{x}_v + \frac{2DR_v}{W_v}\dot{X}_v + X_v\right] \tag{24}$$

where k_v is the servo valve gain, 2_v is the time constant and DR_v is the damping ratio of servo valve. Based on equations (15) – (24), if the state variables are determined as

$$x = [x_1 \quad x_2 \quad x_3]^T = [x_p \quad v_p \quad a_p]^T \tag{25}$$

$$X_v = k_v u \tag{26}$$

Then a third order state equation model for the servo-hydraulic actuator system can be obtained by replacing the valve dynamics (23) with (36).

The following equation can be obtained as

$$\dot{x}_1 = x_2 \tag{27}$$

$$\dot{x}_2 = x_3 \tag{28}$$

$$\dot{x}_3 = \dot{a}_p = \frac{1}{m}\left(A_p \dot{P}_l - \dot{F}_f\right) \tag{29}$$

4. SLIDING MODE CONTROL

The first steps of the sliding mode control theory originated in the early 1950 initiated by S.V. Emelynov. It is started as Variable Structure Control (VSC). At the beginning, a linear second order system modeled was considered as a plant in phase variable form and the trajectories always move toward an adjacent region with a different control structure as fulfill the design requirement of the multiple control structures [13][14]. The most significant advantage of SMC is that once the states of the system reach the predefined sliding surface, the system behaviour depends neither on the system parameters nor the disturbances.This is the reason of the sliding mode control robustness[15][16][17].

The design procedure of SMC consists of defining the sliding mode surface passing through the origin of phase plane to reduce the error equal to zero and determining the equivalent and switching control laws[18]. The implementation of SMC introduces undesired high frequency oscillations or chattering, due to discontinuous switching function (signum) which causes control signal to oscillate around the sliding mode surface. The chattering results in excessive wear and tear of the actuators and even may excite the unmodeled high frequency dynamics of the system. The problem can be overcome by replacing signum function with a smooth function such as saturation or hyperbolic tangent tanh function considering an ultimate boundedness of the error within some predetermined boundary layer[19][20][21][22].

A new control scheme called terminal sliding-mode control (TSMC) is proposed and attracted the attention of many researchers in the past few years to overcome this drawback utilizing nonlinear sliding surface[23][24][25][26]. Nonlinear switching hyperplanes in TSMC can improve the transient performance substantially. Besides, compared with the classical SMC with linear sliding manifold, TSMC offers some superior properties such as faster, finite time convergence, and higher control precision[27]. The terminal sliding mode (TSM) method can be used to design a robust controller that will guarantee a finite time convergence to the origin. This makes it a welcome method for attitude controller designs[28]. The key point of designing a terminal sliding mode controller is the introduction of the nonlinear manifold defined by [24]

$$s = \dot{x} + \sim x + \quad x^{q/p} = 0 \tag{30}$$

where x is a scalar variable, $\sim, \quad > 0$ are constants and $q, p(p > q)$ are odd positive integers. However, TSM controllers suffer from the inherent singularity problem, which means that the control effort requires to be infinite to guarantee the reachability of the pre-selected TSM manifolds.

Digital control technology has grown attention of researchers towards design and implementation of discrete time sliding mode control (DSMC) strategy [19][29][30][31]. However, the discontinuous term in discrete time control law not only introduces chattering but also may lead to instability of systems because of the sampling rate far away from being infinity, which can be resolved by keeping discontinuous term very small. More than that, the DSMC requires process model to derive the control law.

Integral Sliding Mode Control (ISMC) is presented in [32], in which the order of the motion equation is equal to the order of the original system,

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rather than reduced by the number of control inputs. The advantages of ISMC compared to SMC is that the robustness are guarantee for any initial state condition, starting from the initial time instance and since the order of the system has not changed, the performance specification of the system can be used as a guide for designing sliding surface. ISMC also known as Full-Order Sliding Mode Control in [33] and [34]. In [35], the procedure of designing the switching surface has been described for linear time-invariant systems by the placement of an eigenvalues through the solution of Ackermann's equation. The design procedure is defined in terms of the original system rather than in terms of the sliding mode equations.

The super-twisting control law (STW) is one of the most powerful second order continuous sliding mode control algorithms that handles a relative degree equal to one[36]. It generates the continuous control function that drives the sliding variable and its derivative to zero in finite time in the presence of the smooth matched disturbances with bounded gradient, when this boundary is known. Since STW algorithm contains a discontinuous function under the integral, chattering is not eliminated but reduced. The knowledge of the boundaries of the disturbance gradient are required in STW. In many practical cases this boundary cannot be easily estimated. The overestimating of the disturbance boundary yields to larger than necessary control gains, while designing the STW control law. The adaptive-gain STW (ASTW) control law, which handles the perturbed plant dynamics with the additive disturbance or uncertainty of certain class with the unknown boundary, was proposed in [37][38] to overcome this drawback.

5. SMC IMPLEMENTATION IN EHA

This section will give a brief discussion on the sliding mode controller and controller methods combined with SMC that have been used or proposed as hydraulic actuator system controllers.

SMC is recognized as one of the most potential approach in nonlinear control field and has been proved to solve the problem of maintaining the stability for controlling electrohydraulic actuator systems that are subjected to parameter variations and external disturbances[39]. However, as mentioned before in Section 4, the system robustness is not assured until the sliding mode is reached. The main drawback of SMC is the chattering phenomenon which can excite undesirable high-frequency dynamics in position trajectory[40][41]. Moreover, problems relating to simplification of design procedures, control performance enhancement in the reaching mode, and chattering alleviation remain to be fully explored[42]. Variable structure control (VSC) has been studied as an alternative control law for hydraulic servo systems. A VSC with a sliding mode is built on mainly two steps which are proper choice of sliding surface and the choice of reaching law which enforces the closed-loop trajectory to reach the manifold asymptotically[43]. Mohamed A. Ghazy [44] developed a VSC with reaching law for an electro-hydraulic position servo control system to achieve accurate servo tracking in the presence of load disturbance and plant parameter variation while for Bonchis et al.[45], in the presence of important friction nonlinearities. The proposed control technique with reaching law achieves a zero steady-state error for step input with good dynamic transient without chattering in the control input. In [46], similar controller is also designed for the system by considering lumps friction and load as an external disturbance to the system. This work represents the friction in retraction and extension motion of the system. VSC was also proposed to overcome load disturbance and plant parameter variation for electro- hydraulic position servo system [44]. This work considers constant value as disturbance to the system. Y. Liu [47] discusses that the sliding mode control method is suitable for solving the control problem of hydraulic position servos (HPS) with flexible load. However, the research considers the reference model with a flexible load to be treated as nonlinearities. The simulation and experimental results are compared by assessing the performance of the designed controller for the system by means of giving step and sinusoidal input reference. The results showed that a sliding mode controller is strongly robust in maintaining the same level of dynamic performance of the studied servo systems. In [48] and [49], another form of sliding mode control is designed for position tracking of electro-hydraulic servo systems. A new sliding mode control with varying boundary layers is proposed to improve the tracking performance of a nonlinear electro-hydraulic position servo system, which can be found in many manufacturing devices. The proposed control scheme uses varying boundary layers instead of fixed boundary layers, which are usually employed in conventional sliding mode control.

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Fuzzy control using linguistic information possesses several advantages such as being model-free, robust, applying universal approximation theorem and rule-based algorithm. However, the huge amount of fuzzy rules for higher order systems makes the analysis more complex. Therefore, some researchers proposed fuzzy sliding mode controllers (FSMC), which integrated fuzzy set theory and SMC into the controller design to acquire stability and consistent performance. However, it is hard to establish the exact fuzzy rule. In [41] and [50], the variable universe fuzzy self-learning sliding mode control (FSSMC) is proposed to overcome this problem. Fei Cao [41] introduced a variable universe adaptive fuzzy control. It has the ability to adjust the universes of input variables and the membership functions of the conclusion part in rules on-line. It ensures the kinematical state parameters to surpass the sliding mode surface at a low speed thus providing the possibility to minimize chattering. While in [50], an adaptive fuzzy controller with on-line self-tuning fuzzy sliding-mode compensation (AFC-STFSMC) is proposed. This control strategy employs the adaptive fuzzy approximation technique to design the equivalent controller of the conventional sliding-mode control (SMC). Additionally, the fuzzy sliding-mode control scheme with self-tuning ability is introduced to compensate the approximation error of the equivalent controller for improving the control performance. Meng Jun Xia [51] designed a fuzzy sliding mode controller, which uses fuzzy logic to obtain equivalent control signal after the system states reached sliding manifold. High-frequency chattering caused by switching action is weakened effectively and the equivalent control can restrain unknown external disturbances well. In order to design a good fuzzy controller without professional experience, membership functions of the output linguistic variable and control rules are optimized by means of a genetic algorithm simultaneously. Miroslav Mihajlov [52] presented the integration of fuzzy sliding mode and PI controller. This type of fuzzy controller has been chosen to alleviate the problems in calculating the switching gain and improve operating efficiency in the presence of additional external disturbance that is not taken into account in the design for SMC. In this research, the fuzzy PI controller is added in the feedforward branch of the closed-loop, in parallel with the SMC with fixed boundary layers. The model of the system covered unmodeled dynamics, parameter of uncertainties and LuGre friction as external disturbances. In addition to the common nonlinearities that originate from the compressibility of the hydraulic fluid and valve flow-pressure properties, most electro-hydraulic systems are also subjected to hard nonlinearities such as dead-zones due to the valve spool overlap. The presence of a dead-zone can lead to performance degradation of the controller and limit cycles or even instability in the closed-loop system. In order to overcome this problem, an adaptive fuzzy sliding mode controller is developed [53] and [54]. The proposed design attenuates the chattering problem while preserving fast convergence. In comparison with other known fuzzy logic-based approaches to eliminate the chattering problem the proposed method is more transparent and interpretable.

The sliding mode control technique combined with adaptive method has been proposed by Cheng Guan [55] and Lu Xinliang Jia [56]. In [55], the adaptation laws were added to compensate for the system uncertain nonlinearities, linear uncertain parameters, and especially for the nonlinear uncertain parameters caused by the various types of the original control volumes. The proposed control method and adaptation schemes can obtain good performance when the position trajectories are tracked even with nonlinear uncertain parameters. Ning-Bo Cheng [57] proposed a position controller to handle three problems in the control of an electro-hydraulic servo system. The first two problems are about the nonlinearities and the uncertainties of the electro-hydraulic servo system, and the third one is the chattering problem caused by adopting sliding mode control. The proposed controller is called a Switch Controller, for it controls the output switches between the outputs of two other controllers, i.e. a nonlinear controller and a conventional linear controller. The nonlinear controller is actually a nonlinear sliding mode controller and is focused to deal with nonlinearities and uncertainties. The linear controller is designed to improve the transient performance near the steady state.

From the literature, it can be concluded that sliding mode control is a significant nonlinear controller. However, discontinuous control action on SMC results chattering phenomenon which reduces tracking accuracy of the EHA system. Therefore, the suitable method should be proposed to eliminate chattering phenomenon. One way to reduce chattering is to use the boundary layer technique where the sign function is replaced with a saturation function and the sliding function converges and remains within the sliding layer.

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However, this method degrades robustness of the controller. Based on the needs of research in this area, optimal super twisting sliding mode control is the suitable method to overcome the chattering issues. From the literature, a systematic method for obtaining the switching vector and optimum feedback of the SMC was not discussed. The parameters were selected by using trial and error method. The propose method formulates the design of Super Twisting Sliding Mode Control as an optimization problem and utilize Partical Swarm and Gravitional Search Algorithm Optimization algorithm to find the optimal feedback gains and switching vector values of the controller. Two performance functions will be used in the optimization process to demonstrate the system dynamical performance and SMC chattering reduction. The effectiveness of the SM controller integrated with two different optimization technique will be compared in terms of tracking error and the response of the system's output.

6. CONCLUSION

Position tracking performance of an electro-hydraulic actuator can be assured when its robustness and tracking accuracy are guaranteed. The existence of friction, internal leakage and other system behaviors might cause the degradation of robustness and tracking accuracy. There are many works in designing SMC of EHA system previously based on continuous-time. As a conventional sliding mode controller is severely affected by chattering, the GSA-STSMC and PSO-STSMC has been proposed. Two performance functions will be used in the optimization process Apart from the chattering phenomenon, lower control energy consumption is another drawback for future investigation.

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