www.s2is.org
International Journal on Smart Sensing and Intelligent Systems
ISSN 1178-5608

REVIEW ON MODELING AND CONTROLLER DESIGN OF HYDRAULIC ACTUATOR SYSTEMS

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Submitted: Oct. 30, 2014 ***Accepted:Jan. 15, 2015*** ***Published:Mar. 1, 2015***

Abstract- The purpose of this paper is to review the literature on modeling and previous control strategies of the hydraulic actuator system proposed by most of the researchers around the world. Before comes to the main discussion, some background information related to hydraulic actuator will be presented. This review includes a short summary and conclusion for hydraulic actuator system. The repercussion of this review is for future inventions of a better and robust hydraulic actuator system.

Index terms: hydraulic actuator system, motion control, load variation, nonlinear system, modeling, controller.

I. INTRODUCTION

Electro-Hydraulic Actuator (EHA) system is one of the important drive systems in industrial sector and most engineering practice due to its high power to weight ratio, stiffness response, high and good and smooth fast. Recently, with the research and development of mathematics, control theory, computer technology, electronic technology and basic theory of hydraulic, hydraulic control technology has been developed and has been widely used in many applications such as manufacturing systems, materials test machines, active suspension systems, mining machinery, fatigue testing, flight simulation, paper machines, ships and electromagnetic marine engineering, injection molding machines, robotics, and steel and aluminium mill equipment [1]. Due to its applications, the highest performance of the electro-hydraulic actuators on position, force or pressure is needed. However, the system is a highly nonlinear one due to factors such as friction, load variation and leakage [2].The main body of this review is consists of (II) hydraulic actuator motivation, (III) hydraulic actuator modeling, (IV) hydraulic actuator controller design, and (V) short summary and conclusion for hydraulic actuators will be discussed in the last section.

II. HYDRAULIC ACTUATOR MOTIVATION

Hydraulic actuator is one of the major drives in industrial sector and engineering practice due to its redundancy (high power-to-weight ratio, fast and smooth response, high stiffness and good positioning capability) in certain applications [3-4]. This has made the hydraulic actuator a focused study and a variety of control algorithms have been proposed in order to overcome its nonlinear dynamic behavior [5-6]. The hydraulic actuator~~s~~ system is the most appropriate choice for an active suspension system [6], due to its low construction, maintenance cost and high power-to-weight ratio. The hydraulic actuator system also has the ability to produce a very large force and torque in any system [7].Some examples of the hydraulic actuator system application that require a large force and torque are electro hydraulic positioning system, industrial hydraulic machines[8], robot manipulators [9], hydraulic elevator [10],etc. Due to high precision position controllers, hydraulic actuator systems are applied in specialized manufacturing equipments or test equipments such as simple shear apparatus utilized for soil testing [11-12]. Although there are a number of advantages and applications that utilize hydraulic actuator systems, there are some weaknesses that complicate the development of hydraulic actuator system controller since the system is a highly nonlinear system. Aside from the nonlinear behavior, the hydraulic actuator system also suffers from a large extent of model uncertainties [4]. The uncertainties can be classified into two main groups which are parametric uncertainties and uncertain nonlinearities. Example of parametric uncertainties is large changes in load and the large variations in hydraulic parameter such as bulk modulus due to component wear or temperature change. Meanwhile, external disturbance, leakage and friction are called uncertain nonlinearities [4]. These uncertainties can cause hydraulic actuator system controller to be unstable or to have degradation in its performance.

III. HYDRAULIC ACTUATOR SYSTEM MODELING

Modeling for hydraulic actuator system can be done via system identification technique or theoretical mathematical analysis. Various approaches have been introduced and used for hydraulic actuators modeling. System modeling can be based on system physical law or system identification method which formally known as a black box identification. System physical law that is performed requires expert knowledge and understanding about the system itself. Thus, this makes many researchers try to avoid using this method as an option in developing a system model. In contrast with system identification, it only requires a set of stimulus response data and no prior knowledge about the system in order to construct the model. A number~~s~~ of researchers use this technique or method to build the model of hydraulic actuator.

a. Theoretical Mathematical Analysis

The diagram of Electro-hydraulic servo system equipments involve servo valve, hydraulic cylinder and load attached at the end of the piston can be represented as shown in Figure 1 [13].The actuator is responsible to deliver force and motion to the external load or the output device of the hydraulic actuator system.

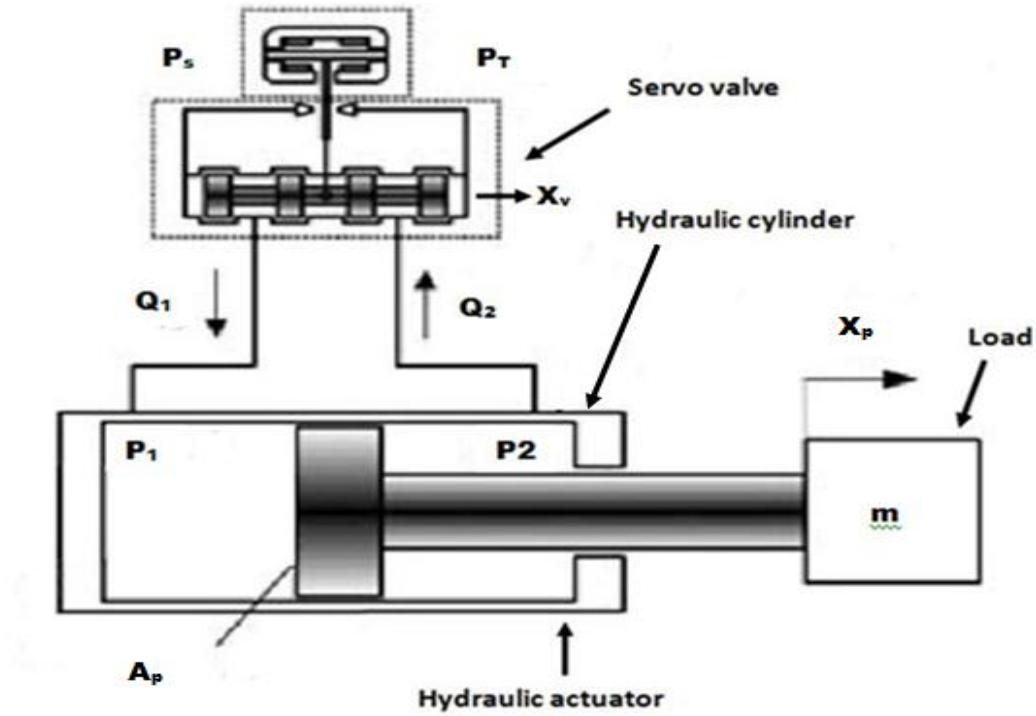

Figure 1. Electro-hydraulic servo valve.

The hydraulic cylinder above consists of a single rod and a single ended piston with a double acting cylinder. From the figure, P_s is the hydraulic supply pressure, P_T is the return pressure, P_1 and P_2 are the fluid pressure in the upper and lower cylinder and x_v is the spool valve displacement. The hydraulic cylinders will extend or compress when a difference between P_1 and P_2 exists. Hydraulic actuator dynamic which include load and servo valve dynamics manage to describe characteristic and behavior of the hydraulic actuator system [5].And the dynamic equation of above system can be written as:

$$\dot{x}_p = v_p \dot{\mathrm{x}}_{\mathrm{p}} = \mathrm{v}_{\mathrm{p}} \tag{1}$$

$$m\dot{v}_p = F_a - F_f - d_u \tag{2}$$

The hydraulic actuating force, F_a and the hydraulic friction force, F_f are the commonly derived forces in hydraulic system. Since load environment, control input voltage, cylinder pressure, etc.

can influence the hydraulic actuation force, F_a, the actuating force becomes a nonlinear function [13]and can be represented as:

$$F_a = A_p P_l \tag{3}$$

The derivative of the load pressure, P_L or the pressure across the actuator piston is given by the total load flow through the actuator divided by the fluid capacitance [6]:

$$\frac{V_1}{4\beta_e}\dot{P}_L = Q_L - C_T P_L - A_P \dot{X}_p \tag{4}$$

$$Q_L = C_d w x_v \sqrt{\frac{2(P_S - sgn(x_v)P_L)}{\rho}} + Q_s \tag{5}$$

Equation (5) shows the relationship between spool valve displacement, x_v and load flow, Q_L. Therefore from (2) and (5), hydraulic dynamic of actuating force cylinder can be written as:

$$\dot{P}_L = -\alpha v_P - \beta P_L + \gamma(C_a \sqrt{\frac{2(P_S - sgn(x_v)P_L)}{\rho}} x_v + Q_s) \tag{6}$$

where,

$$C_a = C_d w\,, \qquad \alpha = \frac{4A_P\beta_e}{V_t}\,, \qquad \beta = \frac{4C_T\beta_e}{V_t}\,, \qquad \gamma = \frac{4\beta_e}{V_t}\,.$$

The input servo valve, u controls the spool displacement dynamic equation of a servo valve, x_v [13].The relationship can be simplified as:

$$\dot{x}_v = \frac{1}{T_v}(-x_v + k_v u) \tag{7}$$

From (1) to (7), state equation of the hydraulic actuator system can be represented as:

$$\begin{bmatrix}\dot{x}_1\\\dot{x}_2\\\dot{x}_3\\\dot{x}_4\end{bmatrix}=\begin{bmatrix}x_2\\\frac{1}{m}(A_P x_3-F_f-d_u)\\-\alpha x_2-\beta x_3+\gamma\left(C_a\sqrt{\frac{2(P_s-sgn(x_4)P_L)}{\rho}}x_4+Q_s\right)\\-\frac{1}{T_v}x_4-\frac{k_v}{T_v}u\end{bmatrix} \tag{8}$$

if the selected state variables are $x=[x_1, x2, x3, x4]^T=[x_p, v_p. P_L. x_v]^T$ [13].

Figure 2 shows schematic diagram of another example of single rod, single ended hydraulic cylinder as the same as that in [4]. The goal of the work in [14], is to have the inertia load to track any specific motion trajectory as close as possible as a machine tool axis in [15].

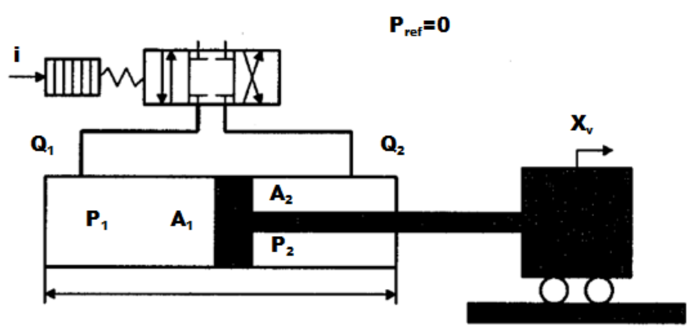

Figure 2. Schematic diagram of single rod, single ended hydraulic actuator.

Dynamic inertia load of the above system can be described as:

$$m\ddot{x}_L = P_1A_1 - P_2A_2 - b\dot{x}_L - F_{fc}(\dot{x}_L) + \tilde{f} \tag{9}$$

where, b is a combined coefficient of the modeled damping and viscous friction forces on the load and cylinder rod, F_{fc} is modeled Coulomb friction force and $\tilde{f}(t, x_L, \dot{x}_L)$ is lumped uncertain nonlinearities due to external disturbance, un-modeled friction forces and other hard-to-modeled terms.

As stated in [7], Q_1, the supply flow rate to the forward chamber and Q_2, the return flow rate to the return chamber, have a relationship with the spool of valve displacement of the servo valve, x_v. The dynamics of cylinder oil flow can be written as (10) if the external leakage cylinder is neglected [7]:

$$Q_1 = k_{q1} x_v \sqrt{\Delta P_1} \qquad \Delta P_1 = \begin{cases} P_s - P_1 & for\ x_v > 0 \\ P_1 - P_r & for\ x_v < 0 \end{cases}$$

$$Q_2 = k_{q2} x_v \sqrt{\Delta P_2} \qquad \Delta P_2 = \begin{cases} P_2 - P_r & for\ x_v > 0 \\ P_s - P_2 & for\ x_v < 0 \end{cases} \tag{10}$$

where, k_{q1} and k_{q2} represent flow gain coefficient of the servo valve, P_s is supply pressure of the fluid and P_r is tank or reference pressure. Constant scaling factors S_{c3} and S_{c4} are introduced to the pressure and valve opening factors in order to facilitate the gain turning process and to minimize the numerical error. Then the scaled pressure becomes $\overline{P_1} = \left(\frac{1}{S_{c3}}\right) P_1$, $\overline{P_2} = \left(\frac{1}{S_{c3}}\right) P_2$, $\overline{P}_s = \left(\frac{1}{S_{c3}}\right) P_s$, $\overline{P}_r = \left(\frac{1}{S_{c3}}\right) P_r$, $\overline{x_v} = \left(\frac{1}{S_{c4}}\right) x_v$. The state variable is defined as; $x=[x_1, x_2, x_3, x_4, x_5]^T=[x_L, \dot{x}_L, \overline{P}_1, \overline{P}_2, \bar{x}_v]^T$. Thus, the entire system can be expressed as [7]:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_5 \end{bmatrix} = \begin{bmatrix} x_2 \\ \frac{S_{c3}A_1}{m}\left(x_3 - \bar{A}_c x_4 - \bar{b} x_2 - \bar{F}_{fc}(x_2)\right) + d(x_1, x_2, t) \\ h_1 x_1 \left[\frac{\beta_e S_{c4} k_{q1}}{\sqrt{S_{c3}} V_{h1}}\left(-\bar{A}_1 x_2 + g_3\left(x_3, sgn(x_5)\right)x_5\right) - \frac{C_{tm}\beta_e}{V_{h1}}(x_3 - x_4) - \frac{C_{em1}\beta_e}{V_{h1}}(x_3 - \bar{P}_r)\right] \\ h_2 x_1 \left[\frac{\beta_e S_{c4} k_{q1}}{\sqrt{S_{c3}} V_{h1}}\left(\bar{A}_2 x_2 + g_4\left(x_4, sgn(x_5)\right)x_5\right) - \frac{C_{tm}\beta_e}{V_{h1}}(x_3 - x_4) - \frac{C_{em2}\beta_e}{V_{h1}}(x_4 - \bar{P}_r)\right] \\ -\frac{1}{T_v} x_5 + \frac{\bar{K}_v}{T_v} u \end{bmatrix} \tag{11}$$

where, $\bar{A}_c = \frac{A_2}{A_1}$, $\bar{b} = \frac{b}{S_{c3}A_1}$, $\bar{F}_{fc} = \frac{1}{S_{c3}A_1} F_{fc} x_2$, $d = \frac{1}{m}\tilde{f}(t, x_1, x_2)$, $h_1 x_1 = \left({}^{1}\!/_{1 + \bar{A}_{h1} x_1}\right)$, $\bar{A}_{h1} = \frac{A_1}{V_{h1}}$, $\bar{A}_1 = \frac{A_1}{k_{q1} S_{c4} \sqrt{S_{c3}}}$, $h_2 x_1 = \frac{1}{V_{hc}} - \bar{A}_{h2} x_1 \mathrm{h_2 x_2} = \frac{1}{\mathrm{V_{hc}}} - \mathrm{\bar{A}_{h2} x_1}$, $\bar{A}_{h2} = \frac{A_2}{V_{h1}}$, $\bar{A}_{hc} = \frac{V_{h2}}{V_{h1}}$, $\bar{A}_2 = \frac{A_2}{k_{q1} S_{c4} \sqrt{S_{c3}}}$, $\bar{K}_{v2} = \frac{K_v}{S_{c4}}$ and nonlinear function is defined as [7]:

$$g_3 = \sqrt{\overline{\Delta P_1}} \qquad \overline{\Delta P_1} = \begin{cases} \bar{P}_s - x_3 & for\ x_5 > 0 \\ x_3 - \bar{P}_r & for\ x_5 < 0 \end{cases}$$

$$g_4 = \frac{k_{q2}}{k_{q1}} \sqrt{\overline{\Delta P_2}} \qquad \overline{\Delta P_2} = \begin{cases} x_4 - \bar{P}_r & for\ x_5 > 0 \\ \bar{P}_s - x_4 & for\ x_5 < 0 \end{cases} \tag{12}$$

Many researchers have already discussed about the effect of servo valve dynamic [16] as this process require additional sensor in order to obtain the spool position. And since minimal improvement is achieved for position tracking, researchers tend to neglect servo valve dynamic as in [17].

Dynamic equation that describing actuator movement can be represented as [18]:

$$m\ddot{y} + d\dot{y} = p_i A_i - p_o A_o + f_d \tag{13}$$

where y is piston's displacement, m is inertia of the moving part, d is equivalent viscous damping coefficient, A_i and A_o represent piston effective area, p_i and p_o are input output line pressure. From [19], hydraulic actuator movement given by Newton's second law can be written as:

$$J\ddot{\theta}_p = D_m(P_1 - P_2) - B\dot{\theta}_p - T_l \tag{14}$$

Meanwhile the nonlinear equation that describe the fluid flow in the valve can be written as [18]:

$$q_i = c_d w x_{sp} \sqrt{\frac{2}{\rho}(p_s - p_i)}\ ; \qquad q_o = c_d w x_{sp} \sqrt{\frac{2}{\rho}(p_o - p_e)}\ ; \qquad x_{sp} \geq 0\ (extension)$$

$$q_i = c_d w x_{sp} \sqrt{\frac{2}{\rho}(p_i - p_e)}\ ; \qquad q_o = c_d w x_{sp} \sqrt{\frac{2}{\rho}(p_s - p_o)}\ ; \qquad x_{sp} < 0\ (retraction) \tag{15}$$

where q_i and q_o represent the fluid flow into and out of the valve, c_q is orifice coefficient of discharge, ρ is mass density of the fluid, p_s and p_e are pump pressure and return pressure respectively, ω is area gradient that relates the spool displacement and x_{sp} is orifice displacement. Equation (14) can be linearized by using a Taylor series expansion about zero spool displacement opening point and neglecting the higher order terms [18] and can be written as:

$$\begin{aligned} q_i &= K_s^i x_{sp} - K_p^i p_i \\ q_o &= K_s^o x_{sp} - K_p^o p_i \end{aligned} \tag{16}$$

Flow and pressure sensitivity gains are represented by $K_s^i(K_s^o)$ and $K_p^i(K_p^o)$ respectively, where these two variables are load-dependent and pressure dependent variables [18].

By neglecting the leakage flow across the actuator's piston, continuity equation oil flow through the cylinder can be given as [18]:

$$\begin{aligned} q_i &= A_i \frac{dy}{dt} + \frac{V_i}{\beta_e}\frac{dp_i}{dt} \\ q_o &= A_o \frac{dy}{dt} + \frac{V_o}{\beta_e}\frac{dp_o}{dt} \end{aligned} \tag{17}$$

where V_i and V_o represent volume of fluid trapped at the side of the actuator and β_e is the effective bulk modulus of the hydraulic fluid. The effective bulk modulus is highly dependent on load condition, air contains in the oil and also the oil temperature. The relationship between spool displacement, x_{sp} and valve input voltage, u can be described as first-order model as in (18) where this dynamic is adequate for many industrial application [18].

$$\dot{x}_{sp} = \frac{-1}{\tau} x_{sp} + \frac{k_{sp}}{\tau} u \tag{18}$$

where, k_{sp} and τ represent the valve gain and the time constant respectively. By transforming equation (13)-(18) into Laplace domain, the transfer function model of the open-loop system can be written as[18]:

$$Y(s) = P_u(s)U(s) + P_d(s)F_d(s) \tag{19}$$

where,

$$P_u(s) = \frac{k_{sp}K_s(A_i+A_o)}{s(\tau s+1)[(ms+d)(Cs+K_{p)}+A_i^2+A_0^2]} \tag{20}$$

and

$$P_d(s) = \frac{(Cs+Kp)}{s[(ms+d)(Cs+K_{p)}+A_i^2+A_0^2]} \tag{21}$$

Here $K_s^i(K_s^o)$ and $K_p^i(K_p^o)$ are considered as uncertain parameters and they are simply replaced by k_s and k_p.

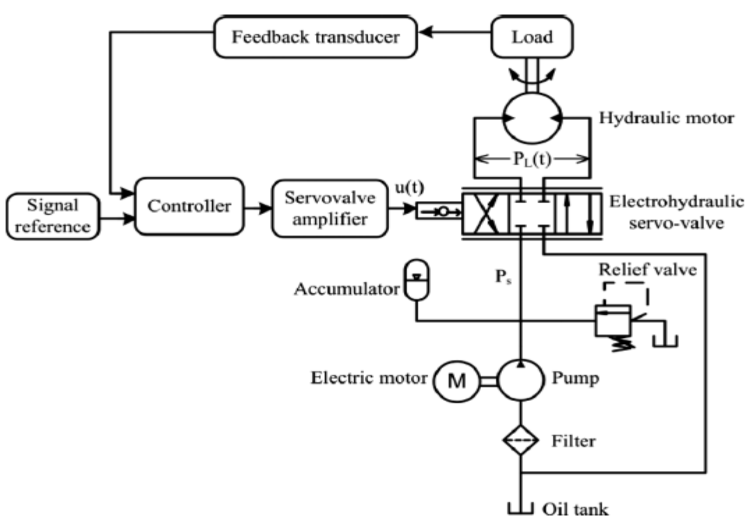

Figure 3. Electro hydraulic system schematic diagram.

Figure 3 shows schematic diagram of electro hydraulic system that has been utilized for experiment in [11][20][21][22]. The oil stored in the tank feeds the system pump. The supply pressure, P_s is kept constant by relief valve and accumulator; however, assuming that the variation in the spring behavior of the relieve valve resulting the non-constant value of P_s. Electro hydraulic servo valve is moving by the power of electrical control input. And the motion of the oil flow from the pump through the hydraulic motor is controlled by the spool motion. And the bidirectional hydraulic motor is driven the load to the desired control objectives. A sensor will measure the angular position which is the output signal of the system. The electro hydraulic system can be described by below fourth-order nonlinear state space model:

$$\dot{x}_1(t) = x_2(t) \tag{22}$$

$$\dot{x}_2(t) = \frac{1}{J}\left(D_m x_3(t) - Bx_2(t) - T_F sgn\big(x_2(t)\big) - T_L\right) \tag{23}$$

$$\dot{x}_3(t) = \frac{4\beta C_d}{V_m\sqrt{\rho}} x_4(t)\sqrt{P_s - sgn\big(x_4(t)\big)x_3(t)} - \frac{4\beta C_m}{V_m} x_2(t) - \frac{4\beta C_{sm}}{V_m} x_3(t) \tag{24}$$

$$\dot{x}_4(t) = \frac{1}{\tau}\left(Ku(t) - x_4(t)\right) \tag{25}$$

$$y(t) = x_1(t) \tag{26}$$

where,x_1, $x_2(t)$, $x_3(t)$ and $x_4(t)$ are angular displacement, angular velocity, motor pressure difference due to the load and servo valve opening area respectively. In (22) – (26), $u(t)$ is control current input, $y(t)$ is system output, J is total inertia of the motor, D_m is volumetric displacement of the motor, B is viscous damping coefficient, T_F is Coulomb friction coefficient, T_L is load torque which we assume to be constant and unknown, β is fluid bulk modulus, V_m is total oil volume in the two chamber of the actuator, C_d is flow discharge coefficient, ρ is fluid mass density, C_{sm} is leakage coefficient, P_s is supply pressure, K is servo valve amplifier gain and τ is the servo valve time constant. The continuous differentiable sigmoid function is used to approximate non-differentiable sign function in (22)-(26) and result in:

$$sgn(\big(x(t)\big) \approx sgm\big(x(t)\big) = \frac{1-e^{-\delta x(t)}}{1+e^{-\delta x(t)}}; \quad \delta > o \tag{27}$$

By doing this, electro hydraulic system can be differentiated and the use of feedback linearization approach is allowable [21]. This linear feedback linearization can help to ensure stability and a good performance of the system if certain conditions of the system are met [11]. This feedback linearization approach is used in [21][23], and has shown a good improvement in stability and system performance.

From [24], hydraulic fluid can be represented by (28) where V is chamber volume.

$$\beta = -V(\frac{d_p}{d_v}) \tag{28}$$

Since hydraulic fluid is compressible to some limit, it should be taking into account of the actuator dynamic. Thus, we can write compressibility equation as follows [19]:

$$\frac{V}{2\beta}\dot{P}_L = \frac{C_d A_v}{\sqrt{\rho}} g(.) - D_m\dot{\theta} - C_L P_L \tag{29}$$

where,

$$g(\cdot) = \sqrt{P_s - sigm(A_v)P_L} \tag{30}$$

In [25-26], the linear differential equation that describe actuator-valve dynamic can be derived from (2) and (6):

$$\dot{v}_p = \frac{1}{m}(-kx_p - bv_p + AP_L - F_r - D) \tag{31}$$

$$\dot{P}_L = \frac{4\beta_e}{V_1}(-A_c v_p - C_T P_L + C_d w x_v\sqrt{\frac{P_s - sgn(x_v)P_L}{\rho}}) \tag{32}$$

where b is the viscous damping of the load and D is the external disturbance of hydraulic system. Laplace transform of (31) and (32) for zero initial condition will produce~~d~~ input output relationship of the hydraulic system as:

$$U_p(s) = H(s)X_v(s) + H_l(s)F_L(s) \tag{33}$$

where,

$$\begin{aligned} H(s) &= \frac{4\beta A K_f}{(ms+b)\left(Vs+4\beta K_{tp}\right) - 4bA^2} \\ H_L(s) &= \frac{-4\beta K_{tp} - sV}{(ms+b)\left(Vs+4\beta K_{tp}\right) + 4bA^2} \end{aligned} \tag{34}$$

b. System Identification

Another approach of getting the mathematical model of a system is by using the system identification technique. System identification is the process to obtain system model through system's input-output (stimulus-response) data. Term identification was first introduce by Zadeh [27], referring to problem of identifying the input-output relationships based on experimental data sets. In contrast to physical law, this technique does not require any expert knowledge about system under study. Thus, this makes this technique more popular due to easy application.

There are two model structures involved in system identification which are "black box model" and "grey box model". For "black box model", a system model is developed without the need of physical model interpretation. Meanwhile for a "grey box model", a system model is developed by considering friction force happened between the piston pressure equations in the actuator chamber [28]. One of the system identification processes is to generate stimulus signal to excite the operating region of the system [29]. The stimulus signal is generated by computer and sent to servo valve of hydraulic system through DAQ card. Servo valve will mode the piston position accordingly by controlling the hydraulic fluid. And draw wire sensor is used to capture position of the piston. The input output data is collected to identify the model. Once the model is obtained, validation of the model is needed to ensure that the obtained model is fits with the observed system behavior. Most common validation processes are done by comparing model performance with real system performance. Best fitting percentage is used as a validation standard, where the higher fitting percentage indicate a more accurate model [25].

In [29], the researcher uses system identification technique with the aid of System Identification Toolbox in MATLAB and System Identification Toolkit in LabVIEW to obtain an electro hydraulic actuator system model. The obtained model is validated with the actual performance of real electro hydraulic actuator system. The process of system identification to obtain system model is as discussed above. The final result shows that nonlinear electro hydraulic system can be modeled by using system identification approach. Both System Identification Toolbox in MATLAB and System Identification Toolkit in LabVIEW perform the same way in identifying electro hydraulic system model.

IV. HYDRAULIC ACTUATOR CONTROLLER DESIGN

Many types of controllers have been used by researchers in order to increase robustness and stability of hydraulic actuator system. This includes robust control, adaptive control [30], state feedback control, variable structure systems control, modern control theories [31][32][33][34]. Previously, a majority of researchers used linear control technique in hydraulic actuator control system [22][35][36]. Examples of linear control technique that have been used before are feedback linearization technique [37] and classic proportional-integral-derivative controller [38][39]. However, linearization technique may degrade some important dynamic information of hydraulic actuator system. Thus, this has made a nonlinear control technique an efficient controller method for hydraulic actuator system. Example of nonlinear control technique that had been proposed before are sliding mode control [40][41], nonlinear back stepping control for position tracking [20] and also for force tracking. This section will give a brief discussion on controller methods that have been used or proposed as hydraulic actuator system controllers.

a. PID Controller

A suitable controller needs to be designed in order to acquire the highest performance of the electro-hydraulic actuator. Many researchers have used advanced control strategies to improve the system performance mainly in tracking control and motion control ability. However, their studies show that the PID control laws are sufficient to control the hydraulic actuator as desired. In [42], the feedback control system design using PID controller has been adopted because it is simple and robust when applied within specified operating range. To ensure a good performance of the controller, appropriate values for each parameter *Kp*, *Ki* and *Kd* must be tuned optimally. PID tuning approach such as Ziegler-Nichols[43] and Nelder-Mead [44] requires information of ultimate gain and ultimate period of oscillation in order to calculate the controller parameters. By using Ziegler-Nichols , [43] shows the increment of the best fit of the model from 92% to 98%. But, a slight different between input and output happened because the electro-hydraulic system which is nonlinear model is modeled in linear model and some nonlinearity and uncertainties characteristic are ignored. While in [44], the Nelder-Mead have been applied to tune the PID parameters. The PID controller seems feasible to control the electro- hydraulic according to desired reference signal but the speed of the response can be improved further for better tracking control.

b. Fuzzy controller

Since the first introduction of fuzzy controller by Mamdani [45], a lot of researchers have applied this controller method in their research study especially in controlling hydraulic actuator system [46][47]. In addition, fuzzy controller has been widely applied to industries and small application around us, such as washing machines, elevators, automobiles, etc.

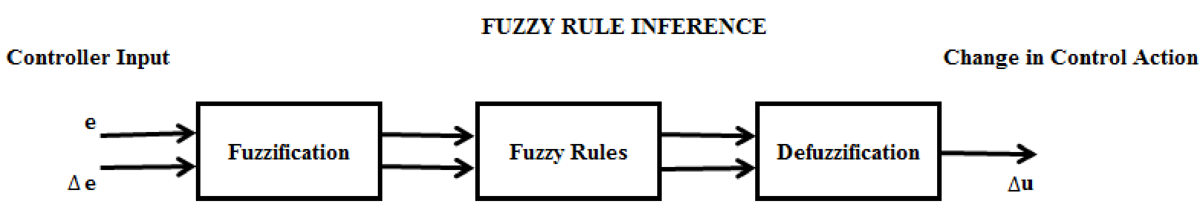

Figure 4. Schematic diagram of a typical fuzzy logic controller

A typical fuzzy controller consists of three basic parts as shown in Figure 4 [1] ;

1) Input signal fuzzification, which transform continuous input signal into linguistic fuzzy variable such as small, medium and large.

2) Fuzzy rules, consists of linguistic control rules which are conditional linguistic statements of the relationship between input and output, so that the property of fuzzy controller emulate the behavior of a human operator [48].
3) Defuzzification converts inferred control action back to a continuous signal. And this makes fuzzy logic sometimes referred to as continuous logic [49]. In fuzzy controller, output error, *e* and change on output error, *ce* of a system are taken as the controller inputs and defined as [48]:

$$e(k) = sp(k) - y(k) \tag{35}$$

$$ce(k) = e(k-1) - e(k) = y(k) - y(k-1) \tag{36}$$

where *sp* and *y*are set point and plant output respectively, *k* and *k-1* indicate present state and previous state of a discrete time system respectively. Control rules in fuzzy controller reflect operators understanding of the system process, so that fuzzy controller can be an expert in controlling the system. Completeness of fuzzy control is when a proper control action for any fuzzy state is generated [48].

In 1994, a multiregion fuzzy logic controller is developed for nonlinear process control [50]. The process that needs to be controlled is divided into four fuzzy regions which is high gain, low gain, long time constant and small time constant. Then a fuzzy logic controller is designed based on the information from each region. Auxilliary process variable is used to detect region operating process. Other than control error and change in control error as input, another variable input is auxiliary variable. And nonlinear relation of fuzzy controller can be described as:

$$\Delta u = FLC(e, \Delta e, AV) \tag{37}$$

where AV represent Auxilliary variable. A fuzzy rule for multiregion fuzzy controller can be described as:

$$\{if\ AV\ is\ A_i\ and\ e\ is\ B_i\ and\ \Delta e\ is\ C_i\ then\ make\ \Delta u D_i\} \tag{38}$$

where A_i, B_i, C_i and D_i are adjective for AV, e, Δe and Δu respectively. These adjective can be a descriptor to the auxiliary variable. To achieve smooth transition between regions, the tuning procedures listed below are to be followed:

1) Scaling factor is tuned for the low gain region.
2) Position of the inner membership function for Δu is tuned for high gain region.
3) Membership function of the auxiliary variable over all regions is tuned to achieve smooth transition of control.
4) Membership function and fuzzy rules are tuned to achieve desired control performance.

The pH CSTR (Continuous Stirred Tank Reactor) for pH titration is used for Multiregion fuzzy tester. The pH value is used as auxiliary variable and can be identified as: pH-high region, pH-medium region, and pH-Low region. The three regions can be defined based on the steady state relation as shows in Figure 5.

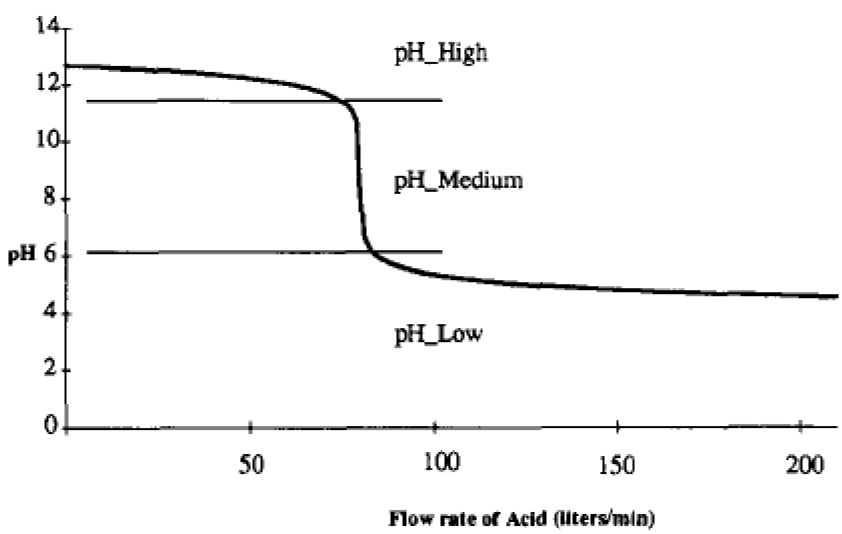

Figure 5. Steady state relation of pH value versus acid flow rate [2] .

To examine how the three region fuzzy controller performs, a disturbance is injected to the system. The source of disturbance that is used here is base concentration (C_2) and figure 6 shows the three region fuzzy controller performance. Both positive and negative disturbance changes are applied while pH value is fixed at 7. It is seen that this controller can perform well with the existence of disturbance.

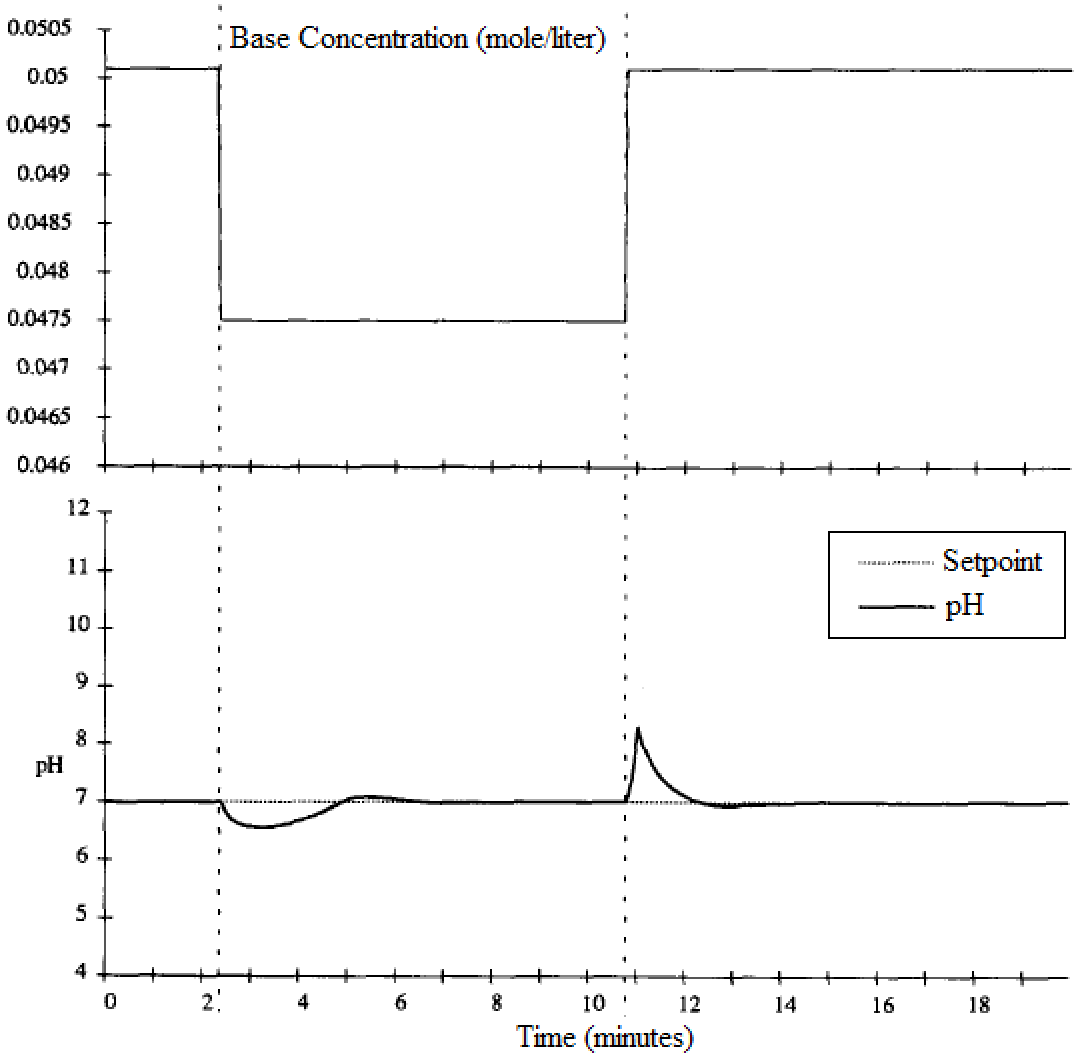

Figure 6. Step response of pH CSTR controller by Multiregion fuzzy controller.

In [48], a design of fuzzy controller is presented for real time controlling of a hydraulic servo system. In the control rules, seven linguistic fuzzy set are applied for all the fuzzy input and output variable. These seven linguistic fuzzy set are:

NB: Negative Big

NM: Negative Medium

NS: Negative Small

PB: Positive Big

PM: Positive Medium

PS: Positive Small

ZE: Zero

Once the linguistic fuzzy set is defined, membership function is needed to describe the fuzzy sets for fuzzification. In this research "trapezoidal" membership function is applied to define all the fuzzy set input output (Figure 7). Center point of the fuzzy sets NB, NM, NS, ZE, PS, PM, PB are -6,-6,-2, 0, 2, 4, 6 respectively. And the membership function can represent as[48]:

$$x < c_p - 2, \qquad \mu(x) = 0$$

$$c_p - 2.0 \le x < c_p - 0.5, \qquad \mu(x) = 1 + \frac{x - (c\_p - 0.5)}{1.5}$$

$$c_p - 0.5 \le x \le c_p + 0.5, \qquad \mu(x) = 1 \tag{39}$$

$$c_p + 5.0 < x \le c_p + 2.0, \qquad \mu(x) = 1 - \frac{x - (c_p + 0.5)}{1.5}$$

$$x > c_p + 2.0, \qquad \mu(x) = 0$$

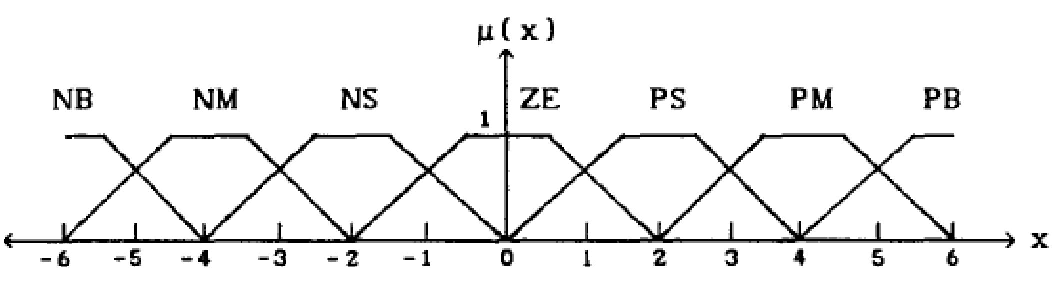

Figure 7. "Trapizoidal" membership function.

Then fuzzy inference "max-product" inference method is adopted to compute membership value and it is defined as [48]:

$$\mu(u^*) = max_{U^i}\{\mu_{E^i}(e^*) \cdot \mu_{CE^i}(ce^*) \cdot \mu_{U^i}(u^*)\} \tag{40}$$

where e^* and ce^* are given control situations; E^i, CE^i, and U^i are the linguistic fuzzy set of rule i i corresponding to input variable e, ce and u respectively. $\mu(u^*)$ is the membership value of the center point u^*. Then center-of-gravity defuzzification method is use to generate a numerical controller output. This method is defined as:

$$u = \frac{\sum_{U^*} \mu(u^*) \cdot u^*}{\sum_{u^*} \mu(u^*)} \tag{41}$$

where *u* is numeric inference result. To generate a proper controller output, (40) and (41) is used. Tuning of scaling factors and quantization procedure are applied in this controller development. The details of the research above can be found in [48]. And finally the designed fuzzy controller is applied to the real time control of a hydraulic servo system with the load and disturbance taken into account. And the output result shows the outstanding performance in control application.

c. Adaptive Robust Controller

Adaptive robust controller is a valid technique to solve for system uncertainties. Many kind of adaptive control schemes have been introduced in hydraulic control system in order to compensate its uncertainties behavior. In linear adaptive controller, assumption are made in such that original control volume between servo valve and cylinder, including volumes between the servo valve, pipelines and cylinder chamber, are certain and known [7]. As in previous adaptive robust controller, it is always assumed that the system's unknown parameters occur linearly, however in practical this situation is impossible. To prevent this, nonlinear adaptive robust controller is presented. As in [7], experiment is done to test the performance of nonlinear adaptive robust controller and the result had proven that the proposed controller manage to obtain a better performance in position signal tracking trajectories, even with the existence of nonlinear parameter compared to linear adaptive robust controller.

In recent years, indirect adaptive controller schemes have been widely used to achieve stability and convergence of the controller system. Example of indirect adaptive controller is in [19] to control the position of electro hydraulic servo system. This controller is chosen among other adaptive controller because it is able to identify the real system value. To test this, the output from this controller is compared with the real time non-adaptive back stepping controller. And the results show that during parameter variation, indirect adaptive controllers are able to track the desired reference signal compared to non-adaptive back stepping controller. In [51], indirect adaptive controller is proposed to control velocity of an electro hydraulic servo system subjected to un-modeled dynamic and load disturbance. A series of simulations are done. The controller is fairly robust to control system and manages to increase performance characteristic of electro hydraulic servo system compared to a conventional PID controller. Thus, it is proven that the indirect adaptive controller has successfully controlled and monitors electro hydraulic servo system. Figure 8 shows flow diagram of an indirect adaptive robust controller. State variable, x_1, x_2, x_3, x_4 and control signal, u are sent to parameter identification block. Once the system parameters are identified, they are sent to back stepping control block to generate control signal. Then the control signal is sent to electro hydraulic system and hydraulic actuator is forcing to track the desired trajectory x_{ref}. This flow is repeated for each sampling time.

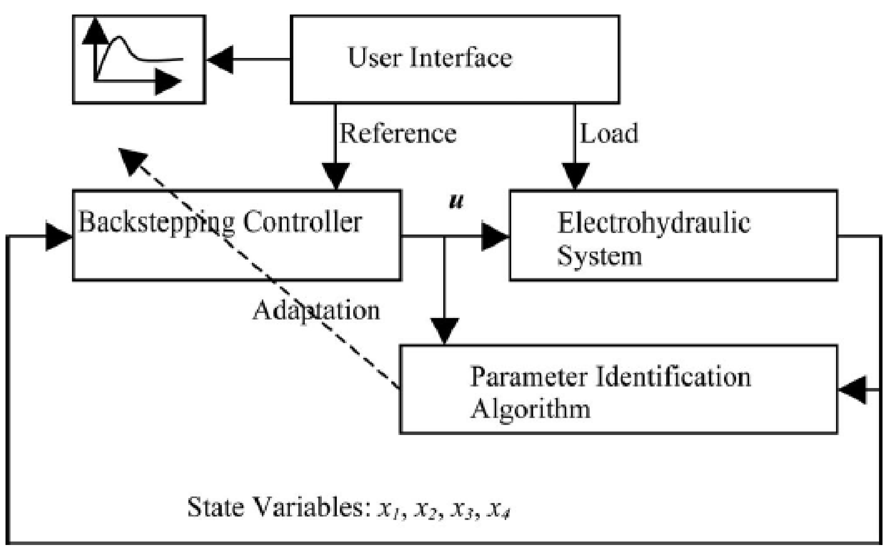

Figure 8. Flow diagram of an indirect adaptive robust controller

Other examples of controller that applies the adaptive robust controller scheme are; 1, A discontinuous projection-based adaptive robust controller that is presented in [4]. This controller considers the effects of parameter variations coming from inertia load and other hydraulic parameters, as well as effects of hard-to-model nonlinearities such as friction force and external disturbance. From the experiment, the controller manages to achieve a guaranteed transient performance, high tracking accuracy, zero error dynamics for tracking any nonzero constant velocity trajectory. This result had verified the high performance nature of discontinuous projection-based adaptive robust controller. 2, High performance robust motion control of single rod hydraulic actuator system is considered in [4], by using discontinuous projection based adaptive robust controller. This research is a continuation from the work done in [4, 56-58]. From the research, it is proven that this controller manages to handle all the uncertain nonlinearities such as external disturbance and uncompensated friction force in hydraulic actuator system and a robust motion controller for hydraulic actuator system is presented. 3, To obtain a better performance in velocity control of nonlinear hydraulic servo system, adaptive model following control is proposed in [22, 59-61]. This controller is designed to track the desired velocity of hydraulic servo system as closed as possible. And from a series of simulations, results show the controller is robust to any unknown disturbance and yields a good performance in following a desired model response.

d. Hybrid Controller

There is several control approaches used in control system design as applied in classical, modern and intelligent systems. The new control strategies have been studied, implemented and suggested in many industrial applications. Every control system technique has its advantages and disadvantages. Thus, the trend nowadays is to implement hybrid systems consisting of more than one types of control technique. The ideal controller would be robust against parameter variations and lead to better performances. Recently, research on fuzzy logic control has been actively done and utilized such as applied hybrid of fuzzy with PID and adaptive PID control using fuzzy [1][38][52][53][54].Their studies show that the hybrid control laws are sufficient to control the hydraulic actuator as desired.

In [1], the development and implementation of self-tuning fuzzy controller to control the position of hydraulic actuator have been discussed. Feedback control system design using PID controller has been adopted in this study due to its simple and robustness when applied within specified operating range. For the result, it shows that the tracking error has been effectively reduced and the self-tuning fuzzy PID controller performance is better than conventional controller. Another implementation of self-tuning fuzzy PID is explained in [53], where the controller is used to control the position of an electro hydraulic actuator system. According to the results presented, when the self-tuning fuzzy PID controller is implied into the system, the response become significantly faster and achieves better tracking response than conventional PID controller. It is indicated from faster rise time, faster settling time, less overshoot and without steady state error. However, the proposed control needs to develop by including disturbance and any others nonlinearity and uncertainties in the design with various frequencies in reference input signals

e. Other Controller Types

Most of the adaptive controllers recently developed use a linearized model for the system. And an adaptive controller that is based on a linearized system model is always unstable [22]. An alternative to the adaptive controller is the variable structure controller. This controller is robust to large parameter changes. However, selection and tuning of the required dead band is a major problem in developing this controller. If too small dead band is selected, the nearly discontinuous control excites un-modeled dynamics present in the system. And if too large dead band is selected, degradation on tracking performance will occur~~s~~. In [24], a nonlinear tracking control law is derived from a Lyapunov function to provide exponentially stable force trajectory tracking in hydraulic system. This control law is similar in [37]. After performance in simulation mode is acceptable, the controller is tested on existing hydraulic test system. And the controller manages to provide excellent force and position tracking even in the presence of system's disturbance.

In [11], feedback linearization-based controller is developed to control supply pressure variations in rotational electro hydraulic servo system. And based on the simulation result, it shown that feedback linearization-based controller is robust for variation in rotational electro hydraulic supply pressure. There are several types of feedback linearization controller such as full-state feedback linearization controller, input-output feedback linearization controller, partial input feedback linearization controller and etc. These controllers have been used in [21][36] for electro hydraulic servo system controller and had shown an improved performance compared to a conventional PID controller.

The robust force controller via nonlinear quantity feedback theory is employed in [55] to control hydraulic actuator force in the presence of system uncertainties and nonlinearities. There are two design methodologies to generate a set of acceptable input-output time history. This input-output time history is necessary in designing a robust quantity feedback theory controller. The first method is based on experimental input-output measurement of an acceptable system response. And the second method, nonlinear mathematical is used for the derivation of input-output histories. The result of the research has clearly shown that the robust force controller via nonlinear quantity feedback theory could provide an effective tool for the control design of hydraulic system. Moreover, this controller is experimentally tested on a real industrial problem, which has been rarely reported in the literature.

In 2006, a robust contact task controller is developed for electro hydraulic actuator system that operates under significant uncertainties and nonlinearities [56]. There are two distinct controllers that are designed individually for position regulation in free space and force regulation during sustain contact. These two controllers are then combined via simple switching law and contact task controller is formed. With the existence of switching, these systems become non-smooth. Then, stability of the controller is analyzed using Lyapunov’s second method under the condition of existence and uniqueness of Filippov’s solution. The advantages of this controller are it is easy to implement, requires a small computation effort, robust with the variation of hydraulic function and environment stiffness, its only requires measurement contact force and actuator position as feedback and have a good performance in transient and steady state period [56].With these advantages, this controller becomes attractive for industrial implementations.

Fault-tolerant controllers for an electro hydraulic servo positioning system are introduced by Niksefat and Sepehri in 2001 [18]. This controller is required to maintain the system’s stability under sensor failure or in the present of faults in servo valve and supply pump. In order to maintain the key properties of closed loop system which is stability and disturbance rejection, this robust controller is designed based on quantitative feedback theory. The feasibility of this controller is tested by implementing it on real hydraulic system. The results show high degree of stability during sensor failure and this controller manage to tolerate with a pump failure when pump pressure dropped ~60% below the normal value [18].

V. CONCLUSION

A hydraulic actuator system that consists of a servo motor and an actuator system has been widely applied in many fields. This is due to its redundancy such as high power to weight ratio, fast and smooth response and good positioning capabilities in many applications. Unfortunately, hydraulic actuator systems are well known for its nonlinearities and it also suffers from a large extent of model uncertainties such as leakage, friction, external disturbance, etc. With all the nonlinearities and uncertainties, stability and performance of the hydraulic actuator system are affected. To overcome this problem, a variety of controller algorithms are proposed. Some of the controllers that have been developed are fuzzy controller, adaptive robust controller, feedback linearize-based controller, robust contact task controller, fault-tolerant controller etc. In order to design an advanced controller with the ability to immune the system's weaknesses, a proper development of system modeling is a must. Hydraulic actuator system modeling can be based on two methods which are system physical law and system identification method. Most researchers choose to use system identification method compared to system physical law as in this method, no prior knowledge about the system is required. As there is still some limitation in current controller development, a continuous study for hydraulic actuator controller needs to carry on for the development of a robust controller for a better performance.

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