Modeling of electro-hydraulic servo valve and Robust Position Control using Sliding Mode Technique

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***Abstract*—Electro-hydraulic servo valve plays vital role in control of systems which are controlled hydraulically. A detailed mathematical model of a hydro-actuation system is developed using first principle approach. Dynamics of electro-hydraulic systems is highly non-linear. Moreover it is subjected to parametric perturbations which hamper the performance of controller. PID controllers do not ensure robust performance in presence of these complexities. A sliding mode control is investigated to ensure robustness. A new sliding surface is proposed. Controller is developed using Gao's reaching law. The method is validated in simulations.**

Keywords: Electro-Hydraulic Servo Valve (EHSV), non linear model, position control, flow forces, robust performance, Sliding Mode Control (SMC).

I. Introduction

Hydraulic systems are widely used in industries ranging from high power applications to precision tooling operations. Hydraulic control systems are popular in industry by virtue of their ruggedness, reliability, flexibility and quick response. To accomplish effective hydro-actuation, servo valve plays a major role. Electrical excitation to valve establishes smooth and easier control over output of valve and brings about subsequent hydraulic actuation precisely. It is the only component which can be controlled directly to have required behavior. Therefore modeling of valve is very crucial. Dynamics are well captured by classical first principle approach. To have complete overview of system, modeling of valve-actuator assembly and plant is explained in detail in subsequent sections.

Solenoid, which receives electrical signals, plays an important role to accomplish flexible and accurate actuation. Fitzgerald [4] explains unidirectional solenoid valve movement with electrical excitation and variation of inductance of coil with spool movement. Spool movement controls orifice dimension. The dynamically changing orifice area decides the amount of pressurized fluid flows in the piston chamber to drive it. The load dynamics are also taken into account. The fundamental dynamic equations are explained by Merrit [5]. Flow force acting on the spool of any valve is one of the most important dynamics need to modeled as precisely as possible for better controls. Master thesis by Okungbowa [6] throws light on flow force modeling. Also paper by Perry Li et al [8] explains flow forces nicely. Andrew Alleyne proposes simplified approach for modeling [3]. Zheng Li [2], John Reid [1] and Baoquan Jin et al [22] give an brief account on modeling and simulation of EHSV systems.

Many researchers have investigated control techniques to accommodate robustness [7] to [23]. Adaptive control provides robust performance however it needs accurate model information [23], [10], [9], [7]. QFT is another robust control reported [7]. But the control becomes complex. SMC is one of the best known robust control technique. SMC in various forms are applied to EHSV systems as explained in papers by R.Ghazali et al [12], Defu Cheng et al [16], Yan Kung et al [19], Rui Tanga et al [20], Wang Wu, [21] and Yang Shi et al [15] etc.

A. Motivation

Electro-hydraulic systems have numerous advantages over other mechanical or electrical systems. Hydraulic actuators have high force to power ratio. They are used in situations where high power needs to be exerted. They can maintain high loading capabilities (high pressures, flow rate) for longer period of time, unlike electrical actuators. This is one of special attributes where hydraulic systems have upper hand than any other systems. They are used in applications ranging from heavy duty manipulators to precision machine tools.

With the advances in computers and control systems, it is easier to control hydraulic systems precisely and easily. Most of existing electro-hydraulic control techniques use PID controllers. These control methods are giving satisfactory performance over limited range. They cannot meet specified requirements in circumstances like parametric variations and external disturbances acting on system. Hence robust control techniques are needed to get satisfactory performance under all conditions.

To overcome the problem of parametric uncertainties, Sliding Mode Control (SMC) [14] is adopted. The base of sliding mode control is Variable Structure Control (VSC). The applications of more than one control strategies to the same system are implemented in VSC. The structure of control algorithm is changed as per the requirement. The use of more than one control techniques achieve stability which might not be achieved in use of single specific control law. Thats how robustness is achieved in the system.

B. *Paper Structure*

This paper has been organized in the following manner, In section II major assumptions used in the modeling and equations used are explained. Section III discusses Problem Definition, state space model of system and lists nominal values of parameters used in system. In Section IV implementations of PID and SMC controllers are given. In section V simulation results with both strategies in presence parametric uncertainties are compared. Section VI concludes the paper.

II. MODELING OF ELECTRO-HYDRAULIC SERVO SYSTEM

The voltage applied to the solenoid coil results in the current flowing through the coil. The current is function of resistance and inductance of coil. Inductance in turn depends on spool position of the valve. (spool and plunger are connected). The spool movement generates the orifice which causes the flow of fluid in the chambers of cylinders. Pressure in the chamber grows and pressure difference is created in cylinder. The pressure differential of fluid at piston of cylinder drives the load. The block diagram of complete system is drawn in following manner,

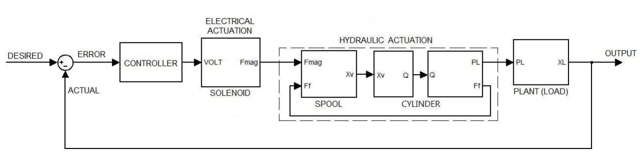

Fig. 1: Block diagram of system

Following are the parameters and constants used in system model.
v : applied voltage(V),
i : current through valve (A),
R : resistance of coil (Ω),
L : inductance of coil(depends on x_v)(H),
x_v : spool displacement (m),
a : stroke length(m),
m : mass of plunger+spool assembly(Kg),
b : coefficient of friction for spool and plunger (Ns/m),
k : spring constant of valve (N/m),
F_{flow} : flow force (opposing force) (N),
A_v : orifice area (m^2),
Q_L : flow rate in system(m^3/s),
c_d : discharge coefficient of valve assembly,
ω : area gradient (m^2/m),
P_s : supply pressure of system (Pa),
P_1 : pressure in chamber 1 (Pa),
P_2 : pressure in chamber 2 (Pa),
P_L : load pressure(Pa),
ρ : density of hydraulic fluid used (Kg/m^3),
V(t) : instantaneous volume of driving chamber (m^3),
V_0 : initial volume of driving chamber (m^3),
β : bulk modulus of hydraulic fluid (N/m^2),
A : cross sectional area of piston (m^2),
x_L : load displacement (m),
m' : mass of plant (load)(Kg),
b' : damping coefficient of load (Ns/m),
k' : spring constant of load (N/m).

A. *Major Assumptions used in the modeling are as follows*

1) Values of parameters are assumed to vary within known boundaries of nominal values (in table).
2) Frictional force between cylinder wall and piston is neglected.
3) Leakage flow of fluid in cylinder and valve is neglected.
4) Spool in the valve is critically lapped.

B. *Electrical Actuation of Electro-Hydraulic Valve*

When voltage (DC) is applied to the solenoid valve, the plunger will move under the influence of magnetic field induced by the current. As plunger moves, inductance associated with position also changes in non linear fashion. Inductance of valve coil is function of spool displacement. At equilibrium point, plunger becomes stationary.

Solenoid coil is basically R-L circuit. Application of KVL to the solenoid valve yields following equation,

$$v = iR + L(x_v)\dot{i} + i\frac{dL(x_v)}{dx_v}\dot{x_v} \tag{1}$$

Considering valve specifications and simplifying,

$$v = iR + L'\left(\frac{x_v}{a + x_v}\right)\dot{i} + \frac{iaL'}{(a + x_v)^2}\dot{x_v} \tag{2}$$

Generally magnitude of electro-magnetic force acting on the plunger is given by,

$$F_{mag} = \frac{i^2}{2}\frac{dL(x_v)}{dx_v} \tag{3}$$

the electromagnetic force depends on rate of change of inductance w.r.t. spool displacement and current through solenoid.

$$F_{mag} = \frac{i^2 a L'}{2(a + x_v)^2} \tag{4}$$

from equation (4) and (3) motion equation for spool-plunger assembly is,

$$\frac{i^2 a L'}{2(a + x_v)^2} = m\ddot{x}_v + b\dot{x_v} + kx_v + F_{flow} \tag{5}$$

As plunger moves, spool coupled with plunger is also displaced. The spool displacement opens the orifice at a rate decided by the controller. Pressure starts building up through the orifice, in the cylinder. Consequently flow force acts on the spool which tries to oppose the motion of spool. The flow forces are taken into account in the equations and by proper application of control force on the spool, they can be nullified.

Flow forces are defined as,

$$F_{flow} = 0.43\omega(P_s - P_L)x_v \tag{6}$$

C. *Generation of Flow rate*

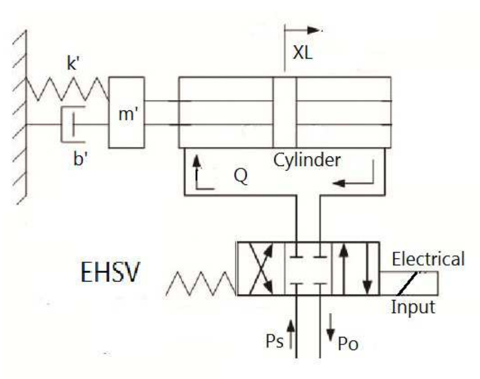

Fig. 2: Electro-Hydraulic Valve with Actuating Cylinder

The flow rate depends on applied pressure across the valve-actuator assembly and spool displacement.

$$Q_L = c_d \omega x_v \sqrt{\left(\frac{P_s - \mathrm{sgn}(x_v).P_L}{\rho}\right)} \tag{7}$$

Max. stroke length of spool is 1cm in positive and -1cm in negative direction. The variations of orifice areas with respect to spool displacement as given.

1) Flow Generation in Valve: Pressurized fluid flows through dynamically changing orifice area. The relation between the spool displacement x_v and orifice area A_v is given as

$$A_v = \omega x_v \tag{8}$$

ω is area gradient which is constant for given valve. It tells about area available for flow per unit spool displacement.

At neutral position valve is completely closed and flow is seized. From neutral position to stage 1 (upto 5mm), flow rate increases, it is given as follows.

$$Q_L = c_d \omega x_v \sqrt{\left(\frac{P_s - \mathrm{sgn}(x_v).P_L}{\rho}\right)} \tag{9}$$

From stage 1 to stage 2 (5mm to 1cm).
when spool displacement increases, area of pressure ports starts decreasing. Flow reduces and goes to zero when spool has reached at stroke length limit (1cm). This phenomenon is captured by,

$$Q_L = C_d \omega (0.005 - (x_v - 0.005)) \sqrt{\left(\frac{P_s - \mathrm{sgn}(x_v)P_L}{\rho}\right)} \tag{10}$$

where (5 mm) is the width of pressure ports of cylinder.
flow reverses in similar fashion when spool has negative motion.

D. *Development of Hydraulic Head*

The flow rate establishes load pressure on piston, within driving chamber of cylinder.

$$\dot{P_L} = \frac{4\beta}{V(t)}\left(Q_L - A\,\dot{x_L}\right) \tag{11}$$

$$V(t) = V_0 + Ax_L \tag{12}$$

$$P_L = P_1 - P_2 \tag{13}$$

E. *Displacement of Load under Hydraulic Head*

Load driving force comprises of cross section area of piston multiplied by load pressure.

$$AP_L = m'\ddot{x}_L + b'\,\dot{x_L} + k'x_L \tag{14}$$

F. *Complete Dynamics of System*

System equations at glance will be,

$$\begin{aligned} v &= iR + L'\left(\frac{x_v}{a + x_v}\right)\dot{i} \\ &\quad + \frac{iaL'}{(a + x_v)^2}\dot{x_v} && (15) \\ \frac{i^2 a L'}{2(a + x_v)^2} &= m\ddot{x}_v + b\dot{x_v} + kx_v + && (16) \\ &\quad 0.43\omega(P_s - P_L)X_v \\ \dot{P_L} &= \frac{4\beta}{V(t)}\left(Q_L - A\dot{x_L}\right) && (17) \\ AP_L &= m'\ddot{x}_L + b'\dot{x_L} + k'x_L && (18) \end{aligned}$$

the term Q_L will assume appropriate formula as per variation in x_v.

III. Problem Formulation

A. State Space Representation

With help of system equations in previous section, states will be defined as
$x_1 = i$ = current in EH valve, $\dot{x_1} = \dot{i}$,
$x_2 = x_v$ = spool displacement, $\dot{x_2} = \dot{x_v} = x_3$ = spool velocity, $\dot{x_3} = \ddot{x}_v$ = spool acceleration,
$x_4 = P_L$ = load pressure, $\dot{x_4} = \dot{P}_L$,
$x_5 = x_L$ = load displacement, $\dot{x_5} = \dot{x_L} = x_6$ = load velocity, $\dot{x_6} = \ddot{x}_L$ = load acceleration
$v = u$ = control input

The system equations in state space representation are

$$\dot{x_1} = \left(\frac{a+x_2}{L'x_2}\right)\left(u - x_1R - \frac{x_1aL'}{(a+x_2)^2}x_3\right) \tag{19}$$

$$\dot{x_2} = x_3 \tag{20}$$

$$\dot{x_3} = \frac{{x_1}^2aL'}{2m(a+x_2)^2} - \left(\frac{k}{m}\right)x_2 - \left(\frac{b}{m}\right)x_3 \tag{21}$$

$$\dot{x_4} = \frac{\beta}{V_0+Ax_5}(c_dwx_2\sqrt{\left(\frac{P_s - \text{sgn}(x_2).x_4}{\rho}\right)} - Ax_6) \tag{22}$$

$$\dot{x_5} = x_6 \tag{23}$$

$$\dot{x_6} = \left(\frac{A}{m'}\right)x_4 - \left(\frac{k'}{m'}\right)x_5 - \left(\frac{b'}{m'}\right)x_6 \tag{24}$$

In the system, parameters which undergo (bounded) variations are given as, Resistance of valve(R), Inductance of valve(L'), Bulk modulus of fluid(β), Density of fluid (ρ), Coefficient of Flow Force

Nominal values used in simulations are given in the table.

B. Problem Definition

The parametric variation affects the performance of the system hence Sliding Mode Controller (Robust Controller) needs to be developed to control performance of system in presence of parametric variations.

IV. Design of Controllers

In this section two controllers i.e. PID and SMC are designed and their performance are compared.

TABLE I: Parameters of system

Sr.No.	Parameters	Values	units
1	a	0.01	m
2	L	0.05	H
3	R	20	Ω
4	b	540	Ns/m
5	k	20	N/m
6	m	0.5	Kg
7	b'	1200	Ns/m
8	k'	410	N/m
9	m'	9.5	Kg
10	B	1.64×10^9	Pa
11	ρ	839.612	Kg/m^3
12	C_d	0.7	-
13	ω	0.024	m^2/m
14	P_s	3.5×10^4	Pa
15	A	0.001855	m^2
16	V_0	$3.1 \times 10{-}4$	m^3

A. PID controller

There are many methods of tuning PID gains such as Zeiglers Nichole, Cohen Choon and many more. We tuned K_p first optimally for the desired settling time of about 2 Sec. Further K_i was tunned to reduce steady state error. Finally K_d was tuned to get desired transient response. MATLAB also yields optimally tuned PID Gains. Tuned PID gains are K_p = 130, K_i= 50, K_d = 10.

B. Sliding Mode Control

A new sliding surface is proposed which is linear combination of only two states as below

$$s = a_se_{x_1} + b_se_{x_5} \tag{25}$$

$$\dot{s} = a_s\dot{e}_{x_1} + b_s\dot{e}_{x_5} = -K|s|^\alpha \text{sgn}s \tag{26}$$

where, s is surface, as $a_s$and b_s are positive scalars. It may be noted that e_{x_1} and e_{x_5} are errors in the states x_1 (current) and x_5 (load position). Gaos power rate reaching law [13] has been used to establish the sliding mode control as shown in subsequent equations.

C. Surface Stability

The sliding surface is stable at $S = 0$ as both e_{x_1} and e_{x_5} are 0. The values observed by the errors of both quantities are either positive or negative at a time. In no case the opposite values are observed by them (Errors vary in same direction). Bringing trajectory on the zero implies that both the errors are made zero. Hence $s = 0$ is true when both the errors are zero. Thus sliding implies convergence of all the states in finite time. Thus the proposed surface is finite time stable.

$$s = a_s e_{x_1} + b_s e_{x_5} \quad (27)$$

$$\dot{s} = a_s \dot{e}_{x_1} + b_s \dot{e}_{x_5} = -K|s|^{\alpha}\text{sgn}s \quad (28)$$

Errors are defined as, $e_{x_1} = x_1 - x_{1_d}$, $e_{x_5} = x_5 - x_{5_d}$, $\dot{e}_{x_1} = \dot{x}_1 - \dot{x}_{1_d}$ and $\dot{e}_{x_5} = \dot{x}_5 - \dot{x}_{5_d}$.

It may be noted that the error e_{x_1} and e_{x_5} both are positive or negative by virtue of the geometry of the valve. The implication are during sliding when $s = 0$, $e_{x_1} = 0$ and $e_{x_5} = 0$. which further implies that $\dot{e}_{x_1} = 0$, $\dot{e}_{x_5} = 0$.

now from state space representation,

$$\dot{x_1} = \left(\frac{a + x_2}{L' x_2}\right)\left(u - x_1 R - \frac{x_1 a L'}{(a + x_2)^2} x_3\right)$$

$$\dot{x_5} = x_6$$

at equilibrium

$$\dot{x}_{5_d} = 0$$

$$\dot{x}_{1_d} = 0$$

Equation (28) becomes

$$\dot{s} = a_s \dot{x}_1 + b_s \dot{x}_5 \quad (29)$$

$$\dot{s} = -K|s|^{\alpha}\text{sgn}s$$

$$-K|s|^{\alpha}\text{sgn}s = a_s \left(\frac{a + x_2}{L' x_2}\right) \left(u - x_1 R - \frac{x_1 a L'}{(a + x_2)^2} x_3\right) + b_s x_6 \quad (30)$$

On simplification, control law is found to be

$$u = \left(\frac{L' x_2}{a_s (a + x_2)}\right)\left(-b_s x_6 - K|s|^{\alpha}\text{sgn}(s)\right) + x_1 R + \frac{a L'}{(a + x_2)^2} x_3 x_1 \quad (31)$$

1) The design parameters of the SMC: coefficients of surface a_s = 1 and b_s = 0.25; switching gain K = 8.5; index of power rate reaching law α = 0.2;

Chattering is the main drawback of SMC. To alleviate chattering sigmoid function is used as it yields continuous approximation of discontinuous signum function. Sigmoid function is defined as $\frac{s}{s+\delta}$ and δ will be decided as per requirement in control law. Where s is the sliding surface. (δ = 0.2)

D. Robust control with variable parameters

The variations in the inductance, resistance of solenoid coil, temperature variations in hydraulic fluid are considered. Roughly ±20% variation in solenoid quantities is considered. Nominal value of L = 50 mH. Variation is 40-60 mH. Nominal value of R = 20 Ω. Variation is 16-24 Ω. Quantities like bulk modulus, density of fluid are dependent on its temperature. Typical applications like aerospace demand large band of variation. Nominal value of temperature is 50°C. Variation from 90°C to -40°C.

Bulk modulus and density of oil is temperature dependent. Variation in bulk modulus (β) is from $1.356 \times 10^9 Pa$ to $2.55 \times 10^9 Pa$ and Variation in density (ρ) is from 812.428 Kg/m^3 to 900.776 Kg/m^3 with respect to corresponding temperatures. Coefficient of flow force which depends on spool geometry and position, is also another variable to be taken into account. Nominal value of coefficient of flow force is said to be 0.43. Variation range is from 0.365 to 0.83.

V. Simulation Results

A. PID control with parametric variation

The responses of PID controller are obtained with nominal value, upper and lower extremes of parameters and plotted on same graph. Figure 3 depicts the load tracking of system. The response of the actuator displacement changes when subjected to parametric variations. Figure 4 explains about voltage applied to the valve by PID controller. The flow forces variations are shown figure 5. Spool movements in different system parameters are shown figure 6. The flow rates in all the 3 cases are plotted in figure 7. Load pressure profiles are as assumed in the figure 8.

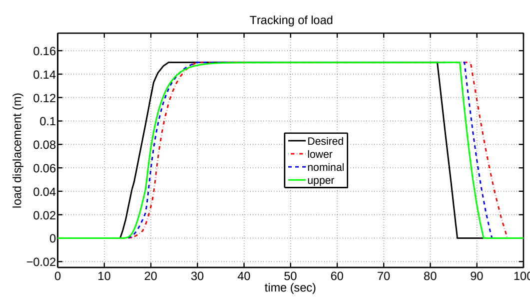

Fig. 3: Load Tracking -PID

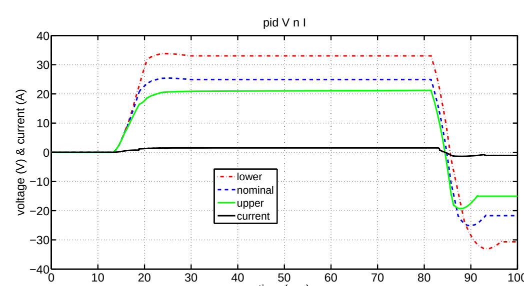

Fig. 4: Control voltages - PID

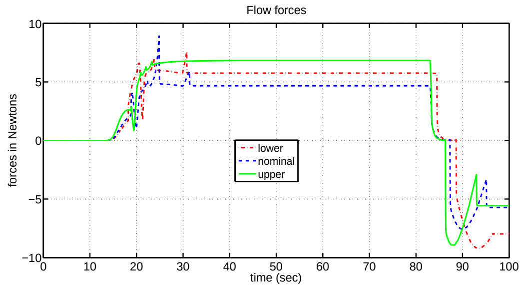

Fig. 5: Forces acting on spool - PID

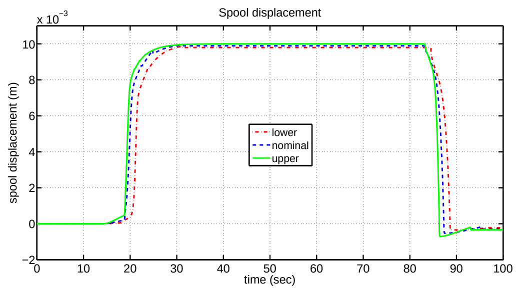

Fig. 6: Spool Displacement - PID

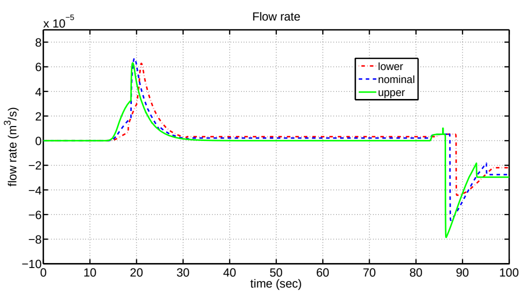

Fig. 7: Flow Rate - PID

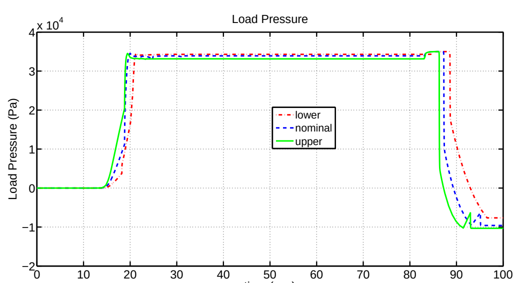

Fig. 8: Load Pressure - PID

B. SMC control with parametric variation

The responses of SM controller are obtained with nominal value, upper and lower extremes of parameters and plotted on same graph. Figure 9 depicts the load tracking of system. Figure 10 explains about voltage applied to the valve by SM controller. Chattering is reduced to a great extent. The flow force profiles are shown in figure 11. Spool movements in varied system conditions are shown figure 12. The flow rates in all the 3 cases are plotted in figure 13. Load pressure profiles obtained are as per in the figure 14.

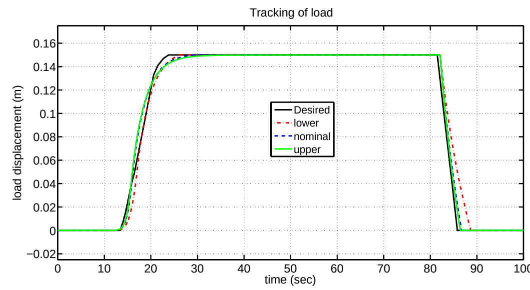

Fig. 9: Load Tracking - SMC

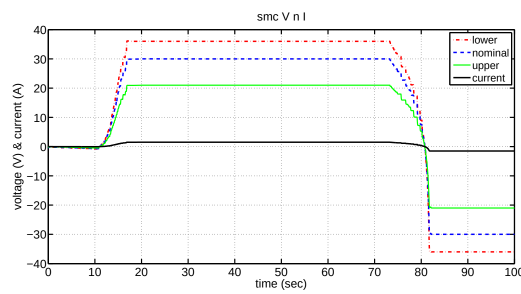

Fig. 10: Control voltages -SMC

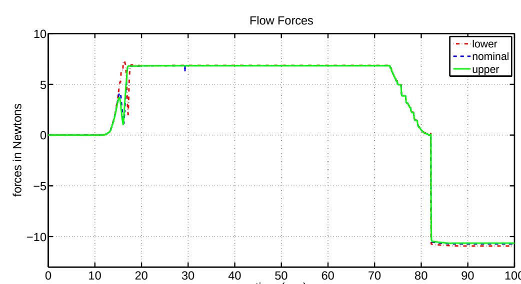

Fig. 11: Forces acting on spool -SMC

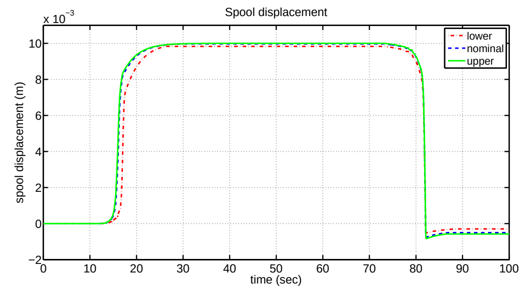

Fig. 12: Spool Displacement - SMC

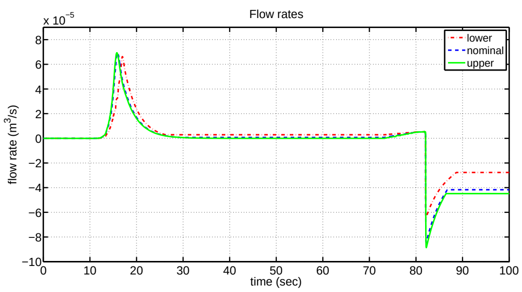

Fig. 13: Flow Rate - SMC

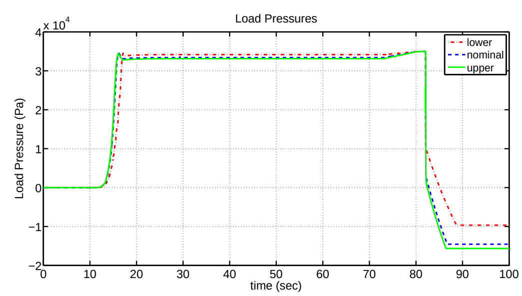

Fig. 14: Load Pressure - SMC

VI. CONCLUSION

In the analysis, PID and SMC controllers are designed and performances of both controllers are observed. On the face of similar parametric variations and disturbances, performance of SMC is found to be more robust than PID. The control efforts applied by both the controllers to valve are measured. Efforts are measured in terms of $\|u\|_2$. Values of control efforts are tabulated as given

SM controller demands lesser control efforts than PID controller. SMC rejects bounded uncertainties and disturbances lying in input channel. SMC gives better and reliable results for position tracking.

TABLE II: Control efforts with varied parameters

Control Efforts	Nominal values	Upper bounds	Lower bounds
PID	977.90	3531.6	775.5
SMC	688.43	942.3	502.6
percentage reduction	29.5 %	73.3%	35.2%

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