Proportional Actuator from On Off Solenoid Valve using Sliding Modes

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***Abstract*— Solenoid valve is a non-linear actuating device used as on-off control in many hydraulic industrial applications. This paper presents conversion of on-off solenoid valve into proportional actuator using sliding mode controller (SMC). A mathematical model of a solenoid valve is developed using first principle approach. As the model is nonlinear, design of proportional actuator becomes challenging. Initially force control strategy is used to ensure the desired plunger position using two PI controllers. Simulation results show deterioration of the performance in presence of parametric variations and disturbances even for the best tuned PI gains. Hence, a sliding mode controller is investigated for accommodating robustness. Gao's power rate and constant rate reaching law is used for SM control design.**

Keywords – Solenoid Valve; Non Linear Magnetic Force; Proportional Actuator; Sliding Mode Control.

I. INTRODUCTION

A solenoid valve is a device used to control the flow of liquid or gas in a system. It is usually powered by electromagnetic energy in a coil. Hence, solenoids are widely used as switching actuators. They are simple in construction, rugged and relatively cheap to produce. For these reasons they can be found in many industrial and domestic apparatus in which limited stroke, on-off mechanical movements are required. In the hydraulics industry, the term proportional valve refers to a specific type of valve in which displacement of plunger occurs in proportion to current flowing through the solenoid coil. The function of proportional valves is to provide a smooth and continuous variation in flow or pressure in response to an electrical input.

Mathematical modeling and control of valves is reported by very few researchers in literature. N.C.Cheung in [1] considered piecewise linear approximation to model nonlinear magnetic characteristics of the solenoid valve. Viktor Szente in [2] performed experimental and computational studies on the dynamic behavior of a pneumatic solenoid valve and pointed out that the plunger causes repetitive flexible collision (bouncing) at its opened end-position. In this paper, a method for reducing vibrating motion of the plunger by using advanced modeling environment simulations tool box has been described. N.C.Cheung in [3] described the conversion of a switching solenoid into proportional device which has been tested out by simulation and implemented on digital signal processor. A method of estimating the position of a solenoid plunger by indirect observation of incremental inductance in the excited coil has been described in [4]. Fitzgerald [5] explains unidirectional solenoid valve movement with electrical excitation and variation of inductance of coil with plunger movement. Sliding mode control applied to solenoid valve operated systems are explained in papers [11], [12], [10]. The design of sliding modes and the reaching laws are described in [13], [14], [17]. Performance and design analysis of solenoid valve are explained in [15], [16].

A. Motivation

The growth of electronic control system, automation, robotics for agriculture and construction industry demands less expensive and robust proportional actuator using solenoid valve. This proportional actuator has very vast industrial application such as fluid flow controller in hydraulic servo systems, grasping motion in robot fingers, robot joints, positioning systems, machine tools drives, etc. Also it has numerous advantages over other mechanical or electrical systems.

At present, classical controllers like PI controllers are being used for development of proportional solenoid actuator. These controllers are easy to design and implement. However precise tuning of PI gains is the challenging task. Moreover they are not robust to the parametric uncertainties which adversely affect the system performance even for the best tuned PI gains. Hence there is a need of such controller that would give robust performance even if there are parameter variations. This parametric variation may be due to changes in spring constant and coefficient of friction, variations in inductance and resistance of the coil etc. To eliminate this problem SMC is investigated. SMC is a robust technique. Order reduction, ease of implementation and insensitive to matched disturbance are the main features of SMC.

B. Paper Structure

This paper is organized as follows. In Section II mathematical modeling of solenoid valve is presented. Section III describes conversion of on off solenoid valve into proportional actuator using two PI controllers. Section IV describes implementation of proportional actuator using SMC and comparative study of PI controller and SMC. Section V concludes the paper.

II. MODELING OF SOLENOID VALVE

Fig.1 shows schematic of a limited travel solenoid valve in which ferromagnetic plunger is surrounded by solenoid coil with a paramagnetic guide tube that provides path for the plunger motion.

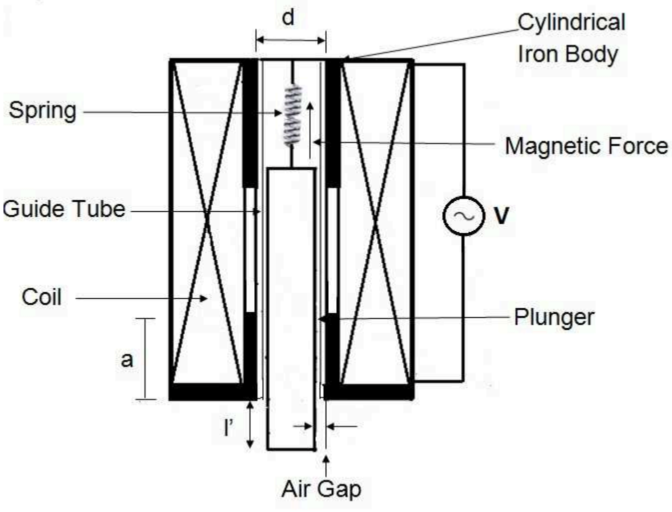

Fig. 1: Limited travel solenoid valve.

A ferromagnetic plunger is allowed to move linearly within the guide tube whenever the current starts flowing through the coil. As soon as the current i starts flowing through the coil, depending on mmf Ni and the reluctance of magnetic path, magnetic flux is set up. Thus the strong magnetic field is produced in upward direction which pulls the plunger in upward direction. As the plunger moves in upward direction, the cross section area A (through which flux linkage occurs between plunger and cylindrical iron strips of valve) increases which results in the decrease of reluctance hence, increase in inductance of coil [6].

Following are the parameters and constants of limited travel solenoid.

v : Input voltage (V).
M : Mass of the plunger (kg).
R : Resistance of coil (Ω).
K : Spring constant (N/m).
i : Current flowing through coil (A).
F_{mag} : Magnetic force (N).
R' : Total reluctance of coil (Ampere-Turns/Weber).
x : Displacement of the plunger (m).
i_d : Desired current flowing through coil (A).
ϕ : Flux (wb).
e : Emf induced in coil (V).
N : Number of turns of coil (Turns).
λ : Flux linkage $= N\phi$ (Turns-Weber).
F : Magnetomotive force mmf $= Ni$ (Turns-A).
d : Diameter of guide tube (m).
b : Coefficient of friction.
g : Thickness of guide tube (m).
λ : Flux linkage $= N\phi$ (Turns-Weber).
F : Magnetomotive force mmf $= Ni$ (Turns-A).
d : Diameter of guide tube (m).
L : Inductance of coil (H).
b : Coefficient of friction.
g : Thickness of guide tube (m).
l : Length of solenoid (m).
l' : Length of plunger outside the valve cabinet (m).
L' : Constant.
a : Length of iron strips through which flux passes (m).
dx : Differential change in plunger displacement (m).
g_r : Acceleration due to gravity ($\mathrm{m/s^2}$).
μ_0 : Permeability of free space ($\mathrm{N/A^2}$).
A : Cross sectional area for iron core and air gap ($\mathrm{m^2}$).

Following are the assumptions considered for the given model:

- Spring force is linearly proportional to the displacement of plunger and spring constant.
- Frictional force is linearly proportional to the velocity of the plunger and the coefficient of friction b.
- All the reluctance is offered by the air gap g.
- The effect of magnetic leakage and the reluctance of the steel are negligible.

By applying KVL to Fig.1 we get,

$$v = Ri + \frac{d\lambda}{dt}, \tag{1}$$

where, the flux linkage λ is a variable dependent on the current of the coil i and the air gap distance x.

$$v = Ri + L(x)\frac{di}{dt} + i.\frac{dL(x)}{dx}.\frac{dx}{dt}, \tag{2}$$

there are three terms in (2). The first term is the resistive voltage drop. The second term is the inductive voltage due to change of current. The third term is known as 'back emf' or 'motion emf' and is caused by the motion of the plunger. Magnetic force can be calculated from the co-energy W'_{fld},

$$F_{mag} = \frac{\partial W'_{fld}\,(i,x)}{\partial x}. \tag{3}$$

The co-energy can be estimated from the integration of flux linkage against current.

$$W'_{fld}\,(i,x) = \int_0^i \lambda\,(i,x)di = \frac{1}{2}L(x)i^2. \tag{4}$$

Total reluctance is the sum of reluctance due to the upper and lower gap. Inductance of coil varies due to variation in total reluctance which can be written as

$$L(x) = \frac{N^2}{R'} = \frac{\pi d \mu_0 a N^2}{g}\left(\frac{x}{x+a}\right) = L'\left(\frac{x}{x+a}\right), \tag{5}$$

from (2), (3) and (5) magnetic force can be calculated as

$$F_{mag} = \frac{i^2}{2}\frac{aL'}{(a+x)^2}. \tag{6}$$

The dynamic equation which describes the motion of the plunger can be written as

$$F_{Mag} = M\ddot{x} + Kx + Mg_r + b\dot{x}. \tag{7}$$

To obtain state-space model define,
$x = x_1 =$ Displacement of plunger, $\dot{x}_1 = x_2 =$ Velocity of plunger, $x_3 = i =$ Current flowing through solenoid coil.

$$\dot{x}_1 = x_2, \tag{8}$$

$$\dot{x}_2 = \frac{1}{M}\left(\frac{{x_3}^2}{2}\frac{aL'}{(a+x_1)^2} - Kx_1 - Mg_r - bx_2\right), \tag{9}$$

$$\dot{x}_3 = \left(\frac{a+x_1}{x_1 L'}\right)\left(v - Rx_3 - \frac{aL'}{(x_1+a)^2}x_3 x_2\right). \tag{10}$$

A. Simulation Results for Open Loop Stable System

The differential equations (8), (9) and (10) are nonlinear dynamic equations of solenoid valve. Open loop stable system is simulated and results are reported in Fig. 2 and Fig. 3. The values of constants and parameters are obtained from the geometrical model of solenoid valve which have been tabulated in Table I. If sinusoidal input voltage is applied to solenoid valve, it is observed that the plunger exhibits sinusoidal variation which is shown in Fig. 2. Also inductance of a coil varies with variations of plunger displacement is shown in Fig. 3.

TABLE I: Nominal values used in simulations are given in the table.

S/No.	Parameters	Values
1	v	230 (V)
2	R	5540 (Ω)
3	d	0.008 (m)
4	g	0.0012 (m)
5	b	4.25
6	K	90.5 (N/m)
7	N	3750 (Turns-Weber)
8	M	0.0035 (Kg)
9	a	0.006 (m)
10	f	50 (Hz)
11	g_r	9.8 ($\mathrm{m/s^2}$)

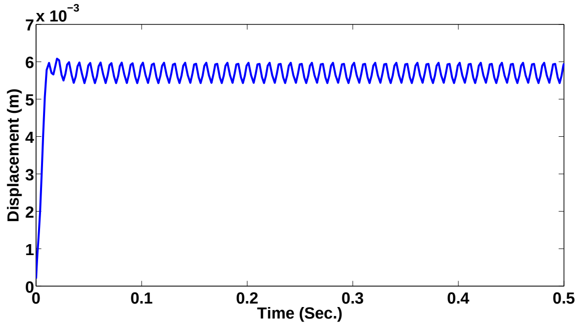

Fig. 2: Displacement of plunger

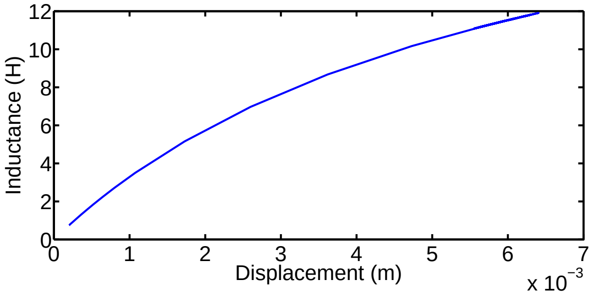

Fig. 3: Variation in inductance as function of plunger displacement

III. PROPORTIONAL ACTUATOR FROM ON OFF SOLENOID VALVE USING PI CONTROLLERS

Proportional valve is used to describe any action where one parameter varies in some proportion to another. The term proportional valve refers to a solenoid activated valve in which plunger or spool displacement occurs in proportion to magnetic force. This magnetic force depends directly on square of current flowing through coil and inversely proportional to the displacement of plunger. The conversion of on/off switching solenoid into a proportional actuator has been described in [8]. As the model is nonlinear and therefore control design is challenging task. Force control strategy is used to ensure the desired plunger position. A PI controller is designed to yield desired force using expression (7). To ensure the desired force another PI controller is designed to ensure the necessary control in a current control loop which is shown in Fig. 4. Two PI controllers were implemented for the control of x and i. Proportional gain K_p in PI parameter is tuned initially to track desired plunger displacement, keeping integral gain K_i zero. After that K_p is kept constant, K_i parameter is tuned to improve tracking further. For nominal system, the gains of PI controller are chosen as $K_p = 17$ and $K_i = 153$ for position control and $K_p = 0.1$ and $K_i = 4$ for current control. Simulation results in Fig.5 and Fig.6 shows the deterioration of performance for overall system response due to parametric variation even for best tuned PI gains. PI gains are kept constant when subjected to parametric variations of $\pm 25\%$ of nominal system. Hence, there is need of robust SMC technique.

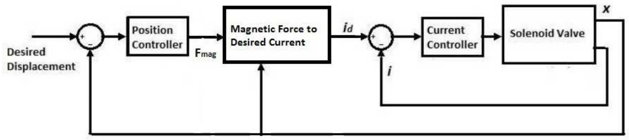

Fig. 4: Proportional actuator from on off solenoid valve using PI controllers

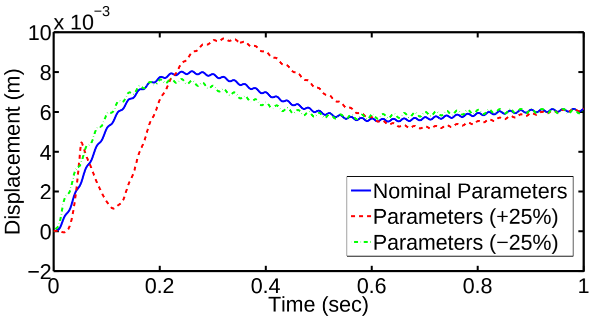

Fig. 5: Displacement of plunger

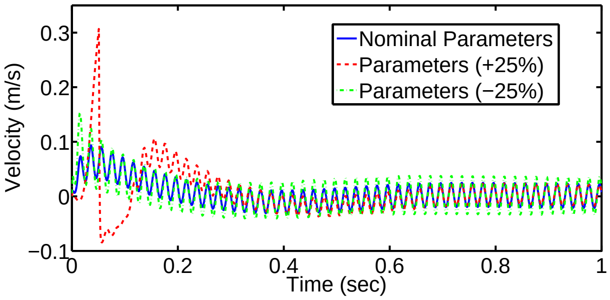

Fig. 6: Velocity of plunger

IV. PROPORTIONAL ACTUATOR FROM ON OFF SOLENOID VALVE USING SMC

Conventional control strategies such as PI controller is being used on a large scale to develop proportional actuator from on off solenoid valve. These strategies are easy to design, however property of robustness to system uncertainties and external disturbances is a major shortcoming. Also, the system performance is affected as system parameters like R and L of solenoid coil vary. These imperfections call for the robust control techniques. The sliding mode controllers provide the robust performance against system uncertainties and matched disturbance. In Fig. 4 both the PI controllers such as position and current controllers have been replaced by SMC. Therefore, design of proportional actuator using SMC has been divided in two parts namely

- Design of Position Controller
- Design of Current Controller

A. Design of Position Controller

Conventional SMC design methodology includes two steps: First, design of a stable sliding surface such that the systems motion along the surface meets the specified performance; second, design a (discontinuous) control law, such that the systems state is driven toward the surface and stays there regardless of disturbances or uncertainties [14]. The dynamic equation (7) which describes the motion of plunger can be rearranged as

$$F_{mag} - Mg_r = M\ddot{x} + Kx + b\dot{x}\ . \tag{11}$$

Assume $u = F_{mag} - Mg_r$, therefore (11) can be written as

$$u = M\ddot{x} + Kx + b\dot{x} \tag{12}$$

To obtain state-space representation of (12) define, x_1 is displacement of plunger and x_2 is velocity of plunger. In matrix form,

$$\dot{\mathbf{x}} = A\mathbf{x} + \mathbf{b}u + \mathbf{d}. \tag{13}$$

where, $\mathbf{x} \in \Re^{2\times1}$, $\mathbf{b} \in \Re^{2\times1}$ and $\mathbf{d} \subset \Re(\mathbf{b})$.

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\frac{K}{M} & -\frac{b}{M} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{M} \end{bmatrix} u + \begin{bmatrix} 0 \\ 33\sin wt \end{bmatrix} d \tag{14}$$

The sliding surface is designed by pole placement technique. Gao's power rate reaching law [13] is used for control law synthesis.

1) Design of Sliding Surface: Consider a sliding surface s as below.

$$s = \mathbf{c}^T(\mathbf{x} - \mathbf{x}_d) \tag{15}$$

where, $\mathbf{x}$ is actual state vector, $\mathbf{x}_d$ is desired state vector. $\mathbf{c}^T$ is sliding surface matrix, $\mathbf{c}^T \in \Re^{m\times n}$ where m is number of inputs and n is number of states. Now $\mathbf{c}^T$ can be designed by converting the nominal system $\dot{\mathbf{x}} = A\mathbf{x} + \mathbf{b}u$ into regular form using $\mathbf{Z} \leftrightarrow T_r\mathbf{x}$ such that,

$$T_r\mathbf{b} = \begin{bmatrix} 0 & b_2 \end{bmatrix}^T$$

where, $B_2 \in \Re^{m\times m}$.

$$\begin{bmatrix} \dot{\mathbf{z}}_1 \\ \dot{\mathbf{z}}_2 \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \begin{bmatrix} \dot{\mathbf{z}}_1 \\ \dot{\mathbf{z}}_2 \end{bmatrix} + \begin{bmatrix} 0 \\ b_2 \end{bmatrix} u \tag{16}$$

where, $\mathbf{z}_1 \in \Re^{1\times1}$ and $\mathbf{z}_2 \in \Re^{1\times1}$.
The switching function matrix partitioned compatibly in Z co-ordinate as, $s = c_1\mathbf{z}_1 + c_2\mathbf{z}_2 = 0 \equiv M_s\mathbf{z}_1 + \mathbf{z}_2 = 0$, where $M_s = c_2^{-1}c_1$ and $M_s \in \Re^{1\times1}$. Using this, null space dynamics of (16) can be written as,

$$\dot{\mathbf{z}}_1 = A_{11}\mathbf{z}_1 + A_{12}\mathbf{z}_2 = (A_{11} - A_{12}M_s)\mathbf{z}_1 \tag{17}$$

Here M_s is designed to ensure stable $(A_{11} - A_{12}M_s)$. Without loss of generality c_2=I, therefore design of M_s yields c_1. M_s can be designed using pole placement technique. $A_{11} = 0$; $A_{12} = 1$; $A_{21} = -25857$; $A_{22} = -357$; $b_2 = -285.71$ and $\mathbf{c}^T = [5 \;\; 1]$.
The designed sliding surface s is for nominal system of (13),

$$s = [5 \;\; 1](\mathbf{x} - \mathbf{x}_d). \tag{18}$$

2) Design of SM Control Law: Differentiating (15),

$$\dot{s} = \mathbf{c}^T(\dot{\mathbf{x}} - \dot{\mathbf{x}}_d) \tag{19}$$

Power rate reaching law [13] is used which is given as

$$\dot{s} = -k|s|^\alpha \text{sgn}(s) \tag{20}$$

where k is switching gain. It is chosen to ensure the existence of sliding. The α is tuning parameter such that $0 < \alpha < 1$. By substituting plant dynamics in (19), a control can be synthesized.

$$u = (\mathbf{c}^T B)^{-1}(-\mathbf{c}^T A\mathbf{x} - k|s|^\alpha \text{sgn}(s) + \mathbf{c}^T\dot{\mathbf{x}}_d) \tag{21}$$

Magnetic force profile can be obtained as

$$F_{mag} = u + Mg_r \tag{22}$$

Once the magnetic force is known then, desired current profile can be calculated as

$$i_d = \sqrt{\frac{2F_{mag}(a+x)^2}{(aL')}} \tag{23}$$

B. Design of Current Controller

The desired current profile called i_d is calculated and a current controller is designed to pass i_d through solenoid coil for the required plunger displacement. This current controller is cascaded with position controller as discussed earlier.

1) Design of a Sliding Surface: Consider a sliding surface s as below.

$$s = \mathbf{x}_3 - \mathbf{x}_d \tag{24}$$

where, $\mathbf{x}_3$ is a state which represents the actual current flowing through coil, $\mathbf{x}_d$ is also a state which represents the desired current.

2) Design of SM Control Law: Differentiating (24) to get

$$\dot{s} = \dot{\mathbf{x}}_3 - \dot{\mathbf{x}}_d \tag{25}$$

from (2),

$$\dot{x}_3 = \left(\frac{1}{L}\right)\left(v - Rx_3 - x_3x_2\frac{dL}{dx}\right). \tag{26}$$

Constant rate reaching law is given as

$$\dot{s} = -k_1 \text{sgn}(s) \tag{27}$$

from (25) and (27)

$$\dot{\mathbf{x}}_3 - \dot{\mathbf{x}}_d = -k_1 \text{sgn}(s) \tag{28}$$

where k_1 is switching gain. It is chosen to ensure the existence of sliding. By substituting plant dynamics in (28), a control can be synthesized.

$$v = L(-k_1 \text{sgn}(s) + \dot{\mathbf{x}}_d + \frac{Rx_3}{L} + \frac{x_2x_3}{L}\frac{dL}{dx}) \tag{29}$$

To reduce chattering in the control law, instead of signum function, sigmoid function has been used. Sigmoid function is defined as $\frac{s}{s+\delta}$ and δ will be decided as per requirement in control law.

C. Simulation Results

Desired current and magnetic force profile is shown in Fig.7 and Fig.8. The performance of SMC is verified and compared with PI controller in simulation. Design parameters of SMC are $\mathbf{c}^T = [5 \;\; 1], k = 1.5, \alpha = 0.9$ for position controller and k_1 = 9 for current controller. Following Figures elaborate superior performance of SMC over PI controller. Tracking of plunger displacement shows less peak overshoot for SMC than PI controller is shown in Fig.9. Velocity of plunger and current flowing through the coil are shown in Fig. 10 and Fig.11 respectively. Control efforts needed for position and current controller are shown in Fig.13 and Fig.15 respectively. Sliding surfaces shown in Fig.12 and Fig.14 reveal that reaching time is 0.08 sec. and 0.06 sec for position and current control respectively.

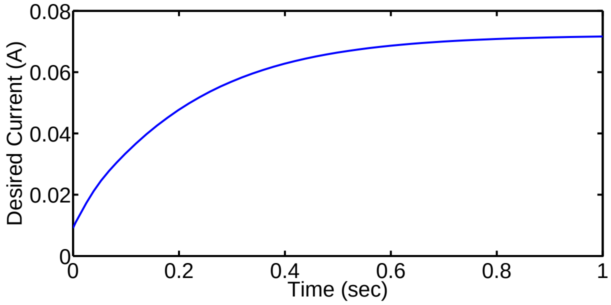

Fig. 7: Desired current profile using SMC

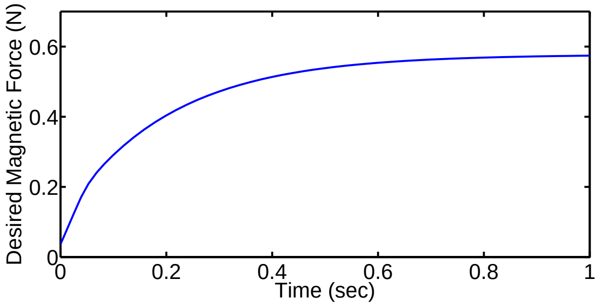

Fig. 8: Desired magnetic force using SMC

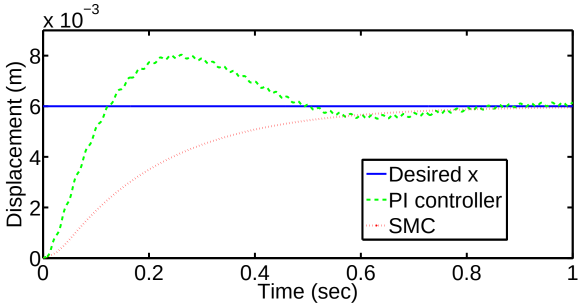

Fig. 9: Tracking of displacement of plunger

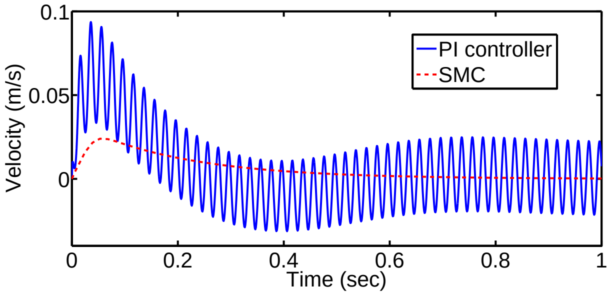

Fig. 10: Velocity of plunger

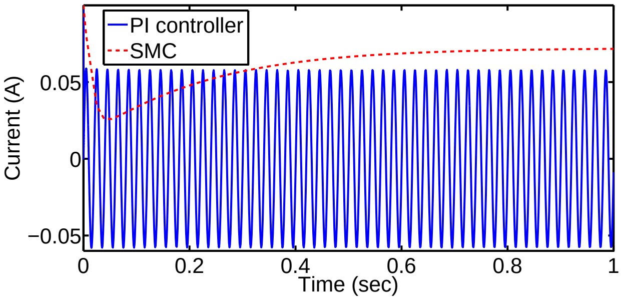

Fig. 11: Current flowing through a coil

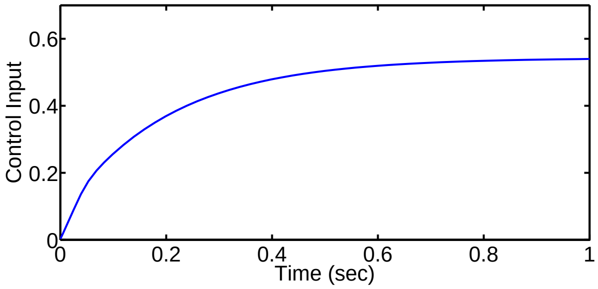

Fig. 13: Control input while designing position controller

D. *External Disturbance*

The disturbance is added in the input channel of the system to check the robustness of the system in presence of external noise. The performance of PI controller and SMC is verified after adding disturbance is shown in the Fig.16 and Fig. 17. This is added in the voltage channel (input channel) of solenoid valve. It is observed that the response remains unaltered even with addition of disturbance in input channel for SMC.

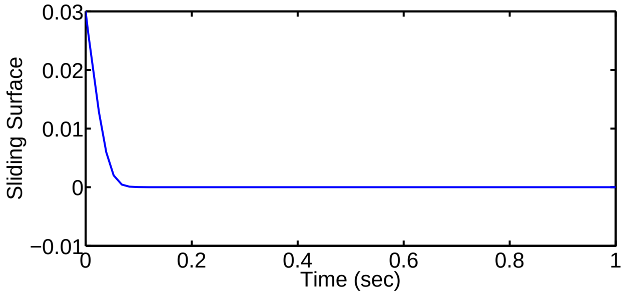

Fig. 12: Sliding surface of position controller

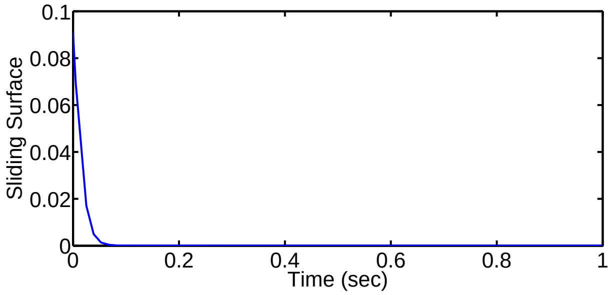

Fig. 14: Sliding surface of current controller

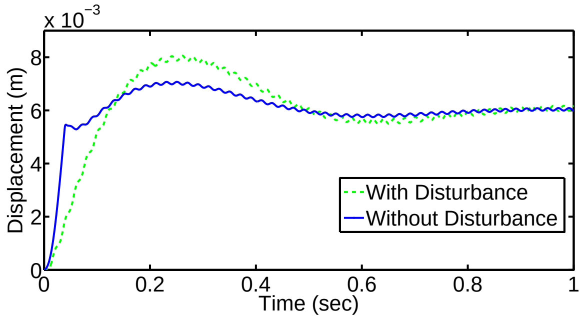

Fig. 16: Displacement of plunger with PI controller

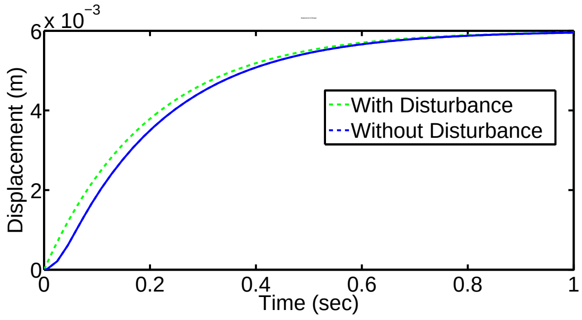

Fig. 17: Displacement of plunger with SMC

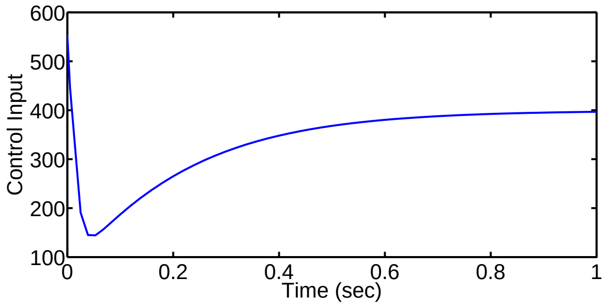

Fig. 15: Control input to the system

E. *Robustness against Parametric Uncertainties*

Sliding mode control (SMC) is known to be robust and can be used to yield robust performance against dynamic model and uncertainties in the parameters. The robust performance of SMC is tested by changing the system parameters by $\pm 25\%$ in the simulation. Controller gains are kept constant as for the nominal system. Fig.18 and Fig.19 represents the robust performance against parametric variation.

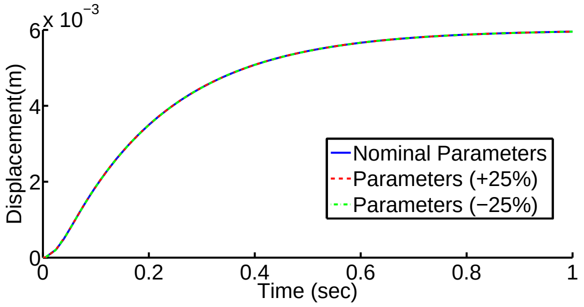

Fig. 18: Displacement of Plunger

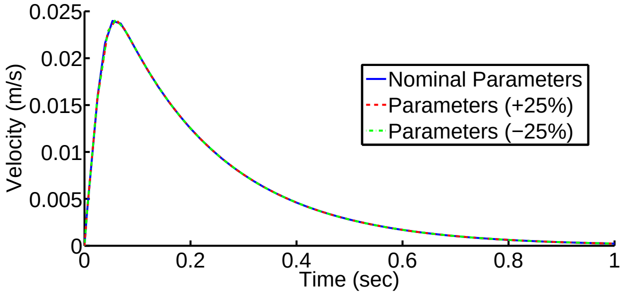

Fig. 19: Velocity of Plunger

Thus, PI and SM controllers have been applied to the solenoid valve to develop proportional actuator. The PI controller stabilizes the proportional actuator asymptotically but requires large input. It gives overshoot and large deviation from the response of nominal plant. On the other hand SMC provides smooth reaching with less energy as compared to PI controller as $\|u\|_2$ = 9772.1 for PI controller and $\|u\|_2$ = 2668.21 for SMC.

V. CONCLUSION

A challenging control problem of proportional actuator has been investigated using sliding modes. Dynamic model of solenoid valve is derived using first principle approach. It has been found from the simulation results that SMC gives better performance over PI controller. PI controller provides the stabilization, however SMC yields the robustness to the parametric uncertainties. SMC demands lesser control efforts than PI controller. SMC rejects bounded uncertainties and disturbances lying in input channel. SMC gives better and reliable results for position tracking.

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