

Control of a Camless Engine Valve Actuation System

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ABSTRACT: This paper focuses on the motion control of a camless engine valve actuation system during both steady state and transient state engine operation. The precise tracking performance obtained using controllers based on the internal model principle for the constant speed case motivates the investigation under engine speed transients. Modelling of the camless engine valve actuation system is performed using linear time varying differential equation method. The system model is simulated in Matlab software using step input. A PID controller is used to track the valve motion trajectory during steady state operation and transient state operation of the engine. Advanced robust controllers can be used to improve the performance of the system.

KEYWORDS: Camless engines, electrohydraulic actuation, internal combustion (IC) engine, PID controller

I. INTRODUCTION

The recent trend of replacing the traditional mechanical systems in the automotive internal combustion (IC) engine with mechatronic systems has been primarily motivated by the decreasing supply of fossil fuels and the increasing concerns about the environment. The removal of the camshaft and the use of individual actuators for opening and closing the engine valves will enable the ultimate flexibility in the air-management system of an IC engine [1]. This flexibility can be used to optimize the air delivery to the cylinders for different speed and loading conditions to improve the efficiency, reduce emissions, and also enable advanced combustion concepts [2], [3].

However, as the engine valves are no longer mechanically connected to the crankshaft, active control is required for precise engine valve positioning to satisfy the performance requirements. These include ensuring repeatable valve motion profiles for precise air delivery, avoiding collision between the engine valve and the piston, and minimizing closing impact velocities, which can lead to excessive noise and wear. With cam-based systems, proper mechanical design automatically guarantees all these properties. The high-speed operation and the demanding performance requirements make the design and control of the actuation systems challenging.

One commonly adopted approach is to sacrifice the ability to fully influence the shape of the valve motion profile to simplify the control requirements. Examples include electrohydraulic actuation systems based on digital valves [4], [5] and electromechanical systems based on solenoids [6], [7]. Advanced controllers [8] – [10] have also been deployed on these systems to ensure the performance characteristics mentioned previously.

It is still desirable to have systems, which can completely control the shape of the valve motion profile. For example, systems with the ability to simulate various cam profiles [11] can significantly accelerate the research and development of cam-based variable valve actuation systems, which are currently being deployed in large quantities. In addition to realizing an actuation system with sufficient bandwidth, the key challenge is the development of control algorithms, which can enable the required tracking performance.

The application of a linear time-invariant (LTI) robust repetitive controller to use the periodicity of the engine valve's motion profile during constant speed operation of the engine was first presented in [12]. However, during engine speed transients, the valve motion profile is no longer periodic with respect to time and thus the traditional LTI controllers are not applicable. Advanced robust controllers can be used to track aperiodic signal.

Paper is organized as follows. Section II explains the the system description. Section III describes the problem definition. Section IV presents the system modelling using linear differential equation method. Section V presents the controller design. Section VI shows the simulation results using matlab software. Finally, Section VII presents conclusion.

II. SYSTEM DESCRIPTION

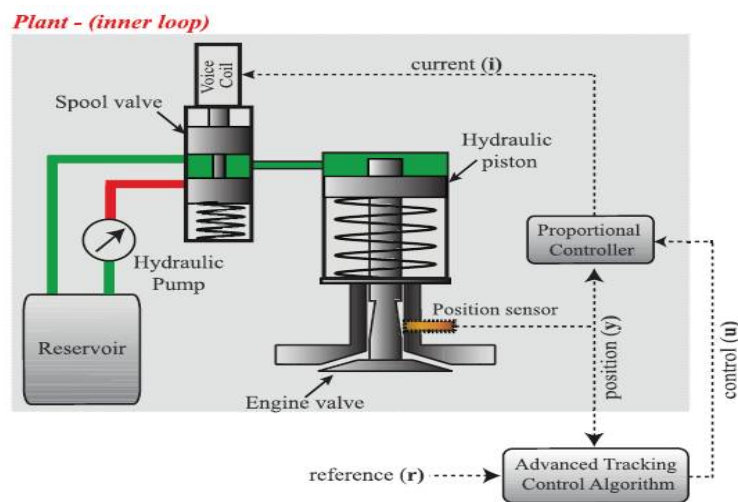


Fig. 1. Schematic diagram of the electrohydraulic camless engine valve actuation system

Fig. 1. shows the schematic diagram of an electrohydraulic valve actuator capable of fully flexible operation. It consists of a hydraulic piston that is in contact with the spring-loaded engine valve. The motion of the hydraulic piston is controlled using a voice-coil actuated three-way spool valve to vary the flow of high-pressure fluid into and out of the hydraulic piston. The output of the system, i.e., the position of the engine valve is measured using a noncontact type displacement sensor.

A proportional controller is first used to stabilize the system, i.e., control the current in the voice-coil of the spool valve based on the difference between the commanded valve position (u) and the actual position (y). This stabilized loop is the plant for which the advanced motion control algorithm is then designed to generate the command (u) to ensure that the actual position of the engine valve (y) tracks the valve reference signal (r).

III. PROBLEM DEFINITION

The objective is to control the camless valve actuator for tracking this profile as it repeats itself for each cycle of the engine. The tracking problems for steady-state operation, i.e., constant engine speed and fixed valve event have been solved using the LTI repetitive controller using the time periodicity of the reference signal.

However, during engine speed transients, as the valve reference trajectory becomes aperiodic in the time domain, the controllers based on the LTI internal model principle such as repetitive control are no longer applicable. To precisely track the aperiodic signal, an advanced robust controller that explicitly considers the time-varying nature of the signal is needed.

IV. SYSTEM MODELLING

Block diagram of the proposed valve actuation system is shown in Fig. 2. Main components include a pump, an on/off solenoid, an internal feedback spool (IFS), two on/off valves, a hydraulic piston, and an engine valve. The pump is used

to provide high pressure fluid. The solenoid valve is a two-position three-way on/off valve that connects the valve actuation system to either a high pressure fluid or the tank. The internal feedback spool is a three position two-way proportional valve, which is the most important component in this system. The flow to and from the main fluid chamber must pass through the IFS and is controlled by an orifice inside the IFS. The IFS orifice area can be changed continuously as the IFS moves up or down, as shown in Fig. 2. The first and second on/off valves are two-position two-way on/off valves and are used to enable and disable the internal feedback. The hydraulic piston acts on the engine valve to open or close it. The IFS, together with the on/off valves and the hydraulic piston, constitutes the internal feedback mechanism that will be explained in the following paragraph.

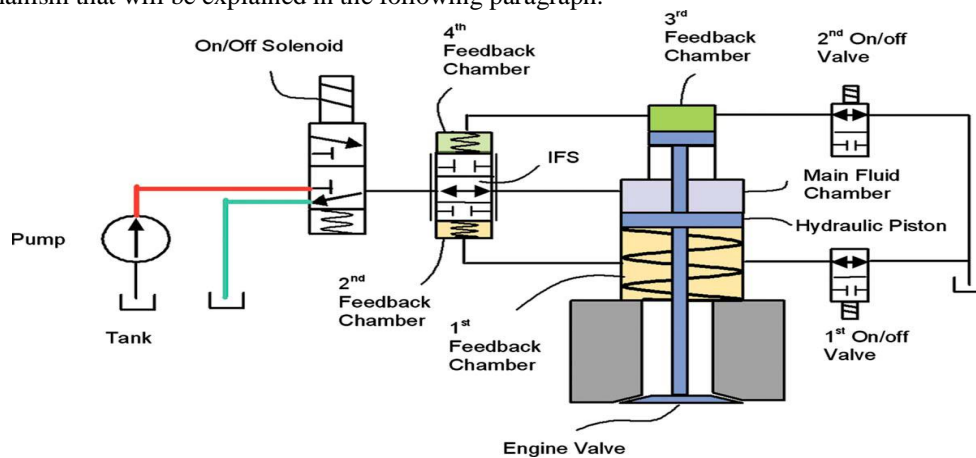


Fig 2. Structure of the valve actuation system

The following is the procedure for operating the system.

1. At the closed position of the engine valve, the solenoid is de-energized, which connects the IFS to the fluid tank, as shown in Fig.2. The first and second on/off valves are open, so that all the feedback chambers are exposed to the low pressure fluid in the tank. The springs inside the IFS keep it at the center position, which connects the main fluid chamber to the tank. The engine valve spring keeps the valve at seat.
2. To open the engine valve, the solenoid is energized, which connects the IFS to the high pressure fluid source. High pressure fluid enters the main fluid chamber through the IFS, overcoming the engine valve spring force and pushes the engine valve open.
3. To stop the engine valve at a desired lift position, the first on/off valve is closed, which cuts off the fluid connection between the first feedback chamber and the tank. As the engine valve continues to move downward, it pushes the fluid in the first feedback chamber into the second feedback chamber, which drives the IFS upward. As the IFS moves upward, it gradually decreases the IFS orifice area, which will reduce the pressure inside the main fluid chamber and slow down the engine valve. This synchronized motion between the engine valve and the IFS continues until the IFS orifice area completely closes and cuts off the fluid connection between the main fluid chamber and the pump. The engine valve then reaches the equilibrium point and stops. This constitutes the internal feedback mechanism for lift control.
4. To close the engine valve, the solenoid is de-energized, which again connects the IFS to the tank. The first on/off valve opens, so that the IFS is returned to the center position by its springs. High pressure fluid in the main fluid chamber will then exhaust into the tank through the IFS. The engine valve spring pushes the valve upward. Low pressure fluid will fill the first feedback chamber as the engine valve moves upward.
5. To land the engine valve at the seat with a desirable seating velocity, the second on/off valve is closed, which cuts off the fluid connection between the third feedback chamber and the tank. As the engine valve continues

to move upward, it pushes the fluid in the third feedback chamber into the fourth feedback chamber, which drives the IFS downward. As the IFS moves downward, it gradually decreases the IFS orifice area, which will raise the pressure inside the main fluid chamber and slow down the engine valve. This motion continues until the IFS orifice area reduces to a minimum opening or cuts off the fluid connection between the main fluid chamber and the tank. The engine valve reaches the valve seat with a desirable seating velocity and stops. This constitutes the internal feedback mechanism for seating control.

The control block diagram of the valve actuation system is shown in Fig. 3. The inner loop represents the internal feedback mechanism that consists of the IFS, the hydraulic piston, and the on/off valve. The outer loop represents the external feedback that controls the solenoid to achieve desired valve position. Without the internal feedback mechanism, the external feedback is the only choice for the valve actuation system. With the internal feedback mechanism, the external feedback becomes optional, as shown in dashed line, in other words, the outer loop could be operated in an open-loop fashion. There are three key advantages of this valve actuation concept. First, as shown in Fig. 3, the lift and seating control are realized through the internal feedback, which bypasses the complex dynamics and time delay associated with the solenoid as otherwise must be faced by the external feedback controller in the outer loop. Once the internal feedback is triggered through the on/off valves, the feedback action only involves a very stiff hydramechanical system consisting of the IFS and the hydraulic piston. This enables a more precise and robust lift and seating control. Second, before the internal feedback mechanism is triggered, the IFS is in the center position and the IFS orifice area remains fully open so that the system is under throttleless operation. This enables the system to operate in an energy efficient manner and addresses the second challenge. Third, with this concept, the external control is only responsible for the triggering timing of the on/off valves to enable and disable the internal feedback mechanism, so the control/calibration effort is greatly simplified and reduced.

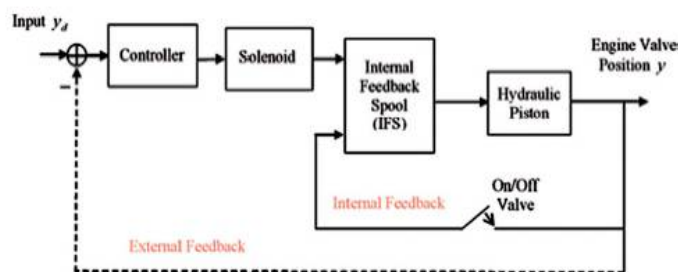


Fig.3. Control block diagram of the valve actuation system

A. Dynamic Range Capability

The dynamic range capability of the valve actuation system refers to its ability to function across the whole speed range of the ICE, typically from 700 rpm to 7000 rpm. As the engine speeds up, the engine valve travels at a higher velocity when it approaches the peak lift or the seat. To stop the engine valve smoothly on either end, the valve actuation system needs to have enough deceleration capability. For any spring-returned electrohydraulic valve actuator, the deceleration capability is limited by the spring stiffness on the opening side and the hydraulic fluid stiffness on the closing side. So for a given engine valve speed, we need to evaluate the minimum lift or stroke required to stop the engine valve with the deceleration capability prescribed by the physical system.

On the opening side, once the internal feedback is triggered, the IFS will start to move upward in synchronization with the engine valve motion. The IFS orifice area will decrease to reduce the pressure inside the main fluid chamber. However, even the pressure inside the main fluid chamber drops to zero, it can only stop accelerating the engine valve, but not provide a decelerating force that is required to stop the valve. Since the spring force is the only resistance to the valve motion, the minimum lift is determined by the stroke that is required to convert the engine valve kinetic energy into the spring potential energy so that the valve speed is zero when it reaches the desired lift.

$$\frac{1}{2}mv^2 = \frac{1}{2}Kx_d^2 - \frac{1}{2}Kx^2 + F_{preload}(x_d - x) \quad (1)$$

where m is the valve train reciprocating mass, K and $F_{preload}$ are the engine valve spring stiffness and preload, respectively, x and v are the engine valve lift and speed, respectively, when the internal feedback is triggered, and x_d is the desired lift.

On the closing side, once the internal feedback is triggered, the IFS will move downward in synchronization with the engine valve motion. The IFS orifice area will decrease to raise the pressure inside the main fluid chamber. The minimum lift is determined by the stroke that is required to convert the engine valve kinetic energy and the spring potential energy into the hydraulic resistance so that the valve reaches the seat with a desirable seating velocity

$$Fx = \frac{1}{2}Kx^2 + F_{preload}x + \frac{1}{2}mv^2 - \frac{1}{2}mv_{seating}^2 \quad (2)$$

where F is the average hydraulic force acting on the hydraulic piston, $v_{seating}$ is the valve seating velocity, and all other parameters are the same as defined in Eq. (1)

The hydraulic force F is going to induce some fluid deflection, which can be calculated as follows:

$$P_A = \frac{F}{A_{piston}} = \frac{\beta}{V_{main}} \Delta V_{main} \quad (3)$$

where P_A is the pressure acting on hydraulic piston, A_{piston} is the hydraulic piston area, β is the fluid bulk modulus, V_{main} is the volume of the main fluid chamber, and ΔV_{main} is the fluid deformation.

And we know that

$$\Delta V_{main} = A_{piston} \Delta h \quad (4)$$

where Δh is the fluid deflection.

Combining Eqs. (3) and (4), we have

$$\Delta h = \frac{FV_{main}}{A_{piston}^2 \beta} \quad (5)$$

B. Internal Feedback Spool Area Schedule

The area schedule of the internal feedback spool refers to the relationship between the IFS displacement and the IFS orifice area. This area schedule is critical to shape the curvature of the engine valve motion for both lift and seating control. Here, we show that for any given engine valve profile, the area schedule can be designed analytically to generate such desired motion.

During the engine valve closing event, the flow going through the IFS is

$$Q = A_{piston} v \quad (6)$$

where Q is the flow rate exiting the main fluid chamber.

The pressure required at the hydraulic piston to generate the desired acceleration is

$$P_A = \frac{ma + F_{preload} + Kx}{A_{piston}} \quad (7)$$

where a is the engine valve acceleration.

And then the IFS orifice area for seating control A_s can be determined as

$$A_s = \frac{Q}{c_d \sqrt{\frac{2(P_A - P_{back})}{\rho}}} \quad (8)$$

where P_{back} is the hydraulic back pressure, C_d is the discharge coefficient, and ρ is the hydraulic fluid density.

Similarly, we can calculate the IFS orifice area for lift control A_l for any desired engine valve motion. Equations (6) and (7) remain the same, and Eq.(8) becomes

$$A_l = \frac{Q}{c_d \sqrt{\frac{2(P_{pump} - P_A)}{\rho}}} \quad (9)$$

where P_{pump} is the pump supply pressure.

C. Internal Feedback Spool Dynamics

As shown in Fig. 2, the internal feedback spool itself is a spring-mass-damper system. It is connected to the hydraulic piston through hydraulic feedback chambers. Since the IFS area schedule is controlled through the IFS displacement, it is crucial to synchronize the IFS motion with the hydraulic piston. When the internal feedback is triggered, the IFS is stationary while the engine valve/hydraulic piston travels at a certain speed. So there will be an impact force acting on the IFS. We need to design the system so that there is no resonance mode close to the IFS operating range. In this section, we examine the dynamics associated with the IFS and the hydraulic feedback chambers.

The IFS dynamics can be modelled as follows

$$\dot{\xi}_1 = \xi_2 \quad (10)$$

$$\dot{\xi}_2 = \frac{A_{IFS} \xi_3 - K_{IFS} \xi_1 - C \xi_2}{m_{IFS}} \quad (11)$$

$$\dot{\xi}_3 = -\frac{\beta}{V_{fb}} A_{IFS} \xi_2 + \frac{\beta}{V_{fb}} u \tag{12}$$

where ξ_1 is the internal feedback spool displacement, ξ_2 is the internal feedback spool velocity, ξ_3 is the hydraulic pressure in the feedback chamber, A_{IFS} is the internal feedback spool feedback area, K_{IFS} is the internal feedback spool spring stiffness, C is the internal feedback spool damping coefficient, m_{IFS} is the internal feedback spool mass, V_{fb} is the feedback chamber volume, and u is the flow rate entering the feedback chamber.

Let $y = \xi_1$ as the internal feedback spool displacement, we can obtain the transfer function

$$\frac{y}{u} = \frac{\frac{\beta A_{IFS}}{V_{fb}}}{s^3 m_{IFS} + Cs^2 + \frac{s\beta A_{IFS}^2}{V_{fb}} + sK_{IFS}} \tag{13}$$

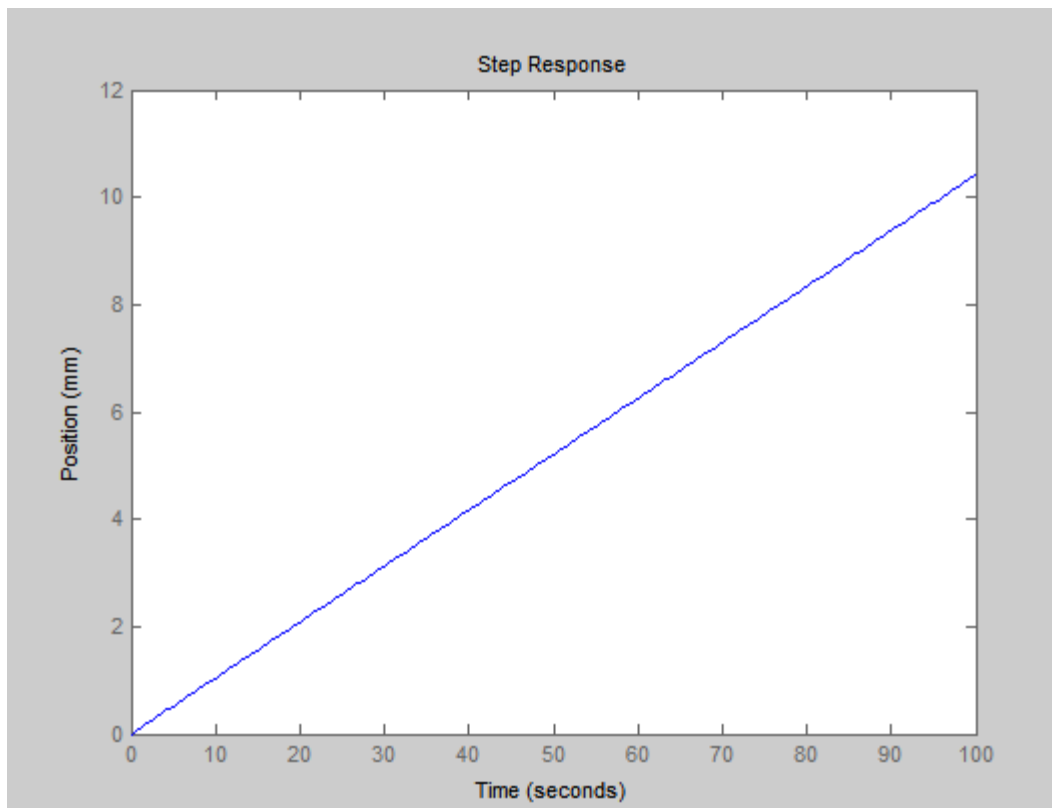


Fig. 4. Step Response of the System

V. CONTROLLER DESIGN

A. PID Controller

The PID controller is the most common form of feedback. It continuously calculates an error value as the difference between a measured process variable and a desired set point. The controller attempts to minimize the error overtime by adjustment of a control variable, such as position of a control valve, a damper or the power supplied to a heating element, to a new value determined by a weighted sum:

$$u(t) = k_p e(t) + k_i \int_0^t e(t) dt + k_d \frac{de(t)}{dt} \quad (14)$$

Where k_p = Proportional gain, k_i = Integral gain and k_d = Derivative gain.

P depends on the present error, I on the accumulation of past errors, and D is a prediction of future errors, based on current rate of change.

The variable (e) represents the tracking error, the difference between the desired input value (r) and the actual output (y). This error signal (e) will be sent to the PID controller, and the controller computes both the derivative and the integral of this error signal. The signal (u) just past the controller is now equal to the proportional gain (k_p) times the magnitude of the error plus the integral gain (k_i) times the integral of the error plus the derivative gain (k_d) times the derivative of the error.

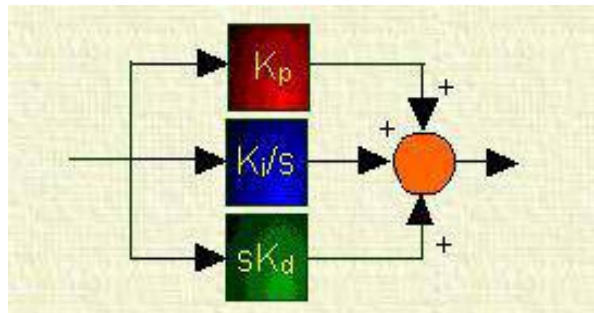


Fig. 5. PID control parameters

B. PID Controller Tuning Using Zeigler-Nichols Method

The Ziegler-Nichols tuning method is a heuristic method of tuning a PID controller. It was developed by John G. Ziegler and Nathaniel B. Nichols. The PID controller has three parameters: proportional gain, integral gain and derivative gain. In industrial control systems more than half of the controllers used have been PID controllers. The performance of PID controllers depends on the relative magnitude of those three parameters. Determination of the relative magnitude of the three parameters is called tuning of PID controllers.

Steps:

1. Find the steady state value of open loop response (K).
2. Find the magnitude of time taken by the response to reach 28.3% of open loop response (t_1).
3. Calculate the magnitude of time taken by the response to reach 63.2% of open loop response (t_2).

4. Calculate the time constant (T).
$$T = (t_2 - t_1) * 1.5 \tag{15}$$

5. Calculate time delay (L).
$$L = t_2 - T \tag{16}$$

6. Find the value of proportional gain (K_p).
$$K_p = (1.2 * T) / (L * K) \tag{17}$$

7. Calculate the value of integral gain (K_i).
$$K_i = K_p / T_i \quad \text{where } T_i = 2 * L \tag{18}$$

8. Calculate the value of derivative gain (K_d).
$$K_d = K_p * T_d \quad \text{where } T_d = 0.5 * L \tag{19}$$

VI. SIMULATION RESULTS AND DISCUSSIONS

Fig. 6. Shows the simulation result of the system. The system settled in 2 seconds with the presence of oscillations.

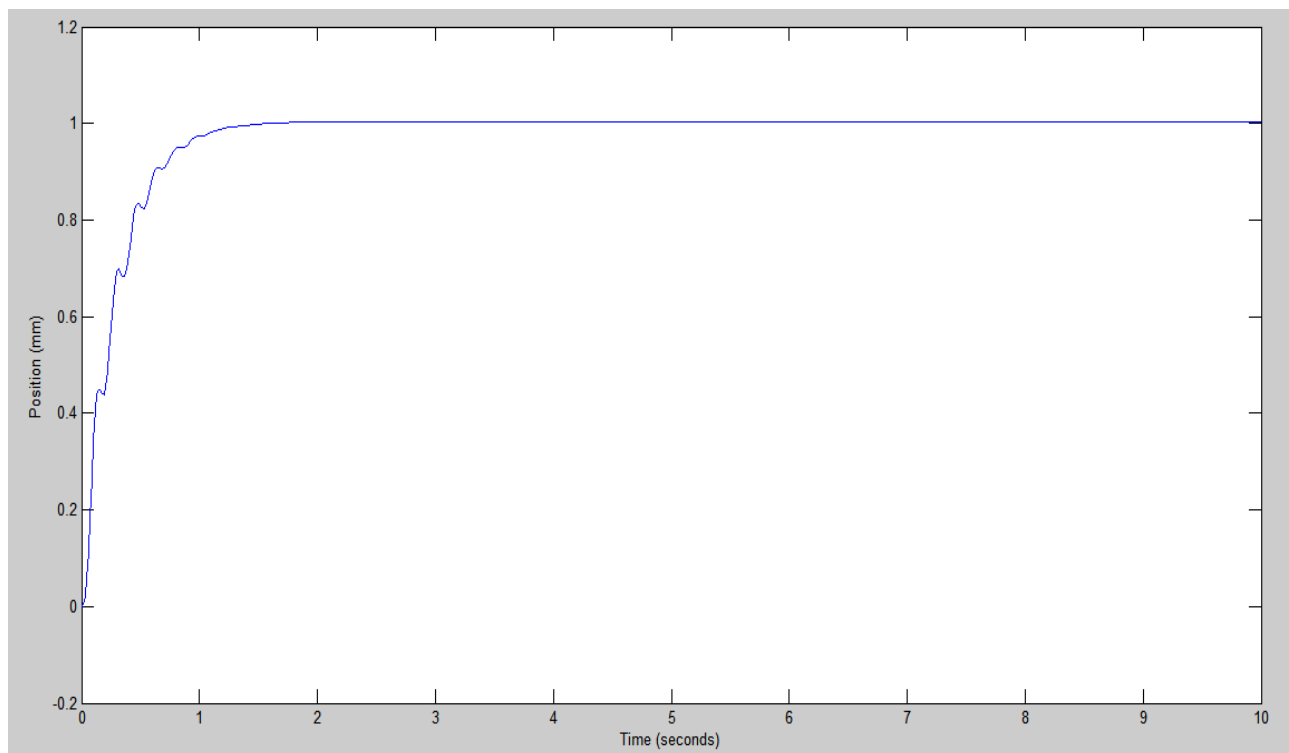


Fig. 6: Response of the system using PID controller

VII. CONCLUSION

This paper focuses on the motion control of a camless engine valve actuation system during both steady state and transient engine operation. Modelling of the system is performed using linear time varying differential equation method. The system model is simulated in the matlab using step input. Designed a PID to track the valve motion trajectory during steady state and transient state of engine operation. Using PID controller the system settled in 2- seconds with oscillations. Advanced robust controllers can be used to improve the performance.

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