MATHEMATICAL THROTTLE MODELLING AND ENGINE RPM CONTROL USING SIMDRIVELINE

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Abstract

Mathematical modeling of physical systems has several advantages most important one is to simulate the system under various operating conditions. It is effectively required design and test the controller algorithm on the computer before actually implementing in the real world. However there are many control design used in automobile which are essential required better performance. The purpose of this project is the model generic engine subsystem is taken and a standard linear PID controller is put to control engine rpm by varying throttle angle as the controllable input variable. PID gains are automatically tuned using PID tuner software. The resultant PID algorithm is tested with sine wave and staircase target variations. The result shows that PID controller gave very satisfactory control performance for controlling engine rpm.

Keywords Engine plant model, PID controller, Throttle, MATLAB Simulink.

I. INTRODUCTION

Mathematical modeling of a physical system, writing the governing differential and algebraic system of equations which allows us to predict the system response (outputs) over a certain duration for a given inputs at a specified time (initial conditions). Mathematical modeling has several advantages 1) To understand how the system behaves at various input combinations during the preliminary design phase itself. 2) To have a better overall understanding of the behavior of the system under various conditions. 3) To create time histories of output variables for a given time histories of input variables. 4) To understand the impact of several properties of the system on the overall output. 5) To evaluate various control algorithms and choose the best one virtually on the computer itself before even building the physical model. 6) To design and the tune the controller equations at all zones of operation of the component. 7) To identify and conflicting requirements, or impossible scenarios early in the design. 8) To avoid costly errors by thoroughly doing what-if studies using system's mathematical model on the computer.

To design a control algorithm for controlling engine rpm by varying throttle angle. The simplest and most widely used linear control algorithm viz., PID (Proportional- Integral-Derivative) controller is used to control the actual engine rpm to desired engine rpm by varying engine throttle appropriately. The PID gains are obtained by automatic tuning which is done by using PID Tuner software add on of Mat-lab/Simulink. The efficiency of PID controller which is measured by engine rpm response to sine wave and staircase target changes are shown.

This paper is organized mathematical modelling of the throttle in section II the proposed design simulation under mat lab Simulink are described in section III simulation results are demonstrated in section IV followed by a conclusion in section v.

II. MATHMATICAL MODELLING OF THROTTLE

The function of a Throttle body is air flow into the engine to control the output torque. The air flow through throttle valve is isentropic flow of compressible fluid (Hendricks & Sorenson, 1991). The model of throttle body is based on the one-dimensional isentropic compressible flow equation for flow across an orifice. The rate of air mass flow through throttle body along with coefficient of discharge is (Heywood, 1988; Guzzella & Onder, 2010).

$$m_{at} = C_d A\,(\,\theta)\frac{\rho_0}{\sqrt{T_0}}\left(\frac{\rho}{\rho_0}\right)^{\frac{1}{k}}\left\{\frac{2k}{k-1}\left[1-\left(\frac{\rho}{\rho_0}\right)\right]^{\frac{k-1}{k}}\right\}^{\frac{1}{2}}\ \frac{\rho}{\rho_0}\ \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

$$C_d A\,(\,\theta)\frac{\rho_0}{\sqrt{T_0}}k^{\frac{1}{2}}\left(\frac{2}{k+1}\right)^{\frac{k}{2(k-1)}}\quad \frac{\rho}{\rho_0}\ \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

where: m_{at}- rate of Air mass flow through throttle valve[g/sec], C_d- coefficient of discharge, $A(\theta)$- throttle opening cross-sectional area [m^2], ρ_0-stagnation pressure of up-stream [kPa], ρ-throat pressure of throttle[k Pa], T_0-stagnation temperature of up-stream [K], Ideal gas constant for air 287×10−5 [bar$\cdot\frac{\mathrm{m3}}{\mathrm{kg}}\cdot$K],

In above mentioned equation lower throttling input area $A(\theta)$ interrelated discharge pressure improved values are over looked (harrington&bolt,1970). And input manifold pressure was comparatively taking values of plenum pressure. with respect to discharge coefficient can be observed that throttling is one dimensional orifice. The discharging coefficient is nonlinear capacity of throttled angle as well as pressure of throttling (Heywood,1998) The steady-state flow tests show that these two factors are independent of engine operating conditions (Guzzella & Onder 2010) The area of cross-sectional of the throttle body is a nonlinear function of the throat diameter. The throttle rod diameter will affect cross-sectional area of the throttle body when the throttle opening angle is at big position. The cross-sectional area of throttle opening is represented by equation as below (Harrington & Bolt, 1970; Guzzella & Onder, 2010).

$$A\,(\,\theta) = \frac{dD}{2}\left[1-\left(\frac{d}{D}\right)^2\right]^{\frac{1}{2}} + \frac{dD}{2}\left[1-\left(\frac{d}{D}\frac{\cos\theta o}{\cos(\theta o+\theta)}\right)^2\right]^{\frac{1}{2}}$$

$$+\frac{D^2}{2}\sin^{-1}\left\{\left[1-\left(\frac{d}{D}\right)^2\right]^{\frac{1}{2}}\right\}\frac{D^2}{2}\frac{\cos(\theta o+\theta)}{\cos\theta o}\sin^{-1}\left\{\left[1-\left(\frac{d}{D}\frac{\cos\theta}{\cos(\theta o+\theta)}\right)^2\right]^{\frac{1}{2}}\right\}$$

Where is d- shaft diameter of throttle [m], D – bore diameter of throttle [m], θ -open angle of throttle [deg], θ_0- angle for minimum leakage area [deg]. When the throttle opening cannot increase its effective area or ($\theta \geq \cos^{-1}(d/D - \cos\theta_0) - \theta_0$), the cross-sectional area of throttle opening is represented by equation as below (Harrington & Bolt, 1970).

$$A\,(\,\theta) = \frac{D^2}{2}\sin^{-1}\left\{\left[1-\left(\frac{d}{D}\right)^2\right]^{\frac{1}{2}}\right\}\frac{dD}{2}\left[1-\left(\frac{d}{D}\right)^2\right]^{\frac{1}{2}}$$

At the small throttle opening, there is an error between the ideal cross-sectional area and the actual area due to machining variation (Carpenter & Ramos, 1985), the minimal leak area of empirical data can be obtained by a large number of experiments, the error correction for this can be written by $\cos(0.91\theta_0 - 2.59)$ instead of $\cos\theta_0$.

III. SIMULATION UNDER MATLAB SIMULINK

3.1 Engine Plant Model

It is attempted to control engine rpm to the desired target rpm profile by varying throttle position accordingly. For this PID control algorithm is used as it is a simple yet very powerful, and most commonly used linear control algorithm used in a vast majority of control applications. Fig. 1-1 shows the engine plant model Simulink Diagram.

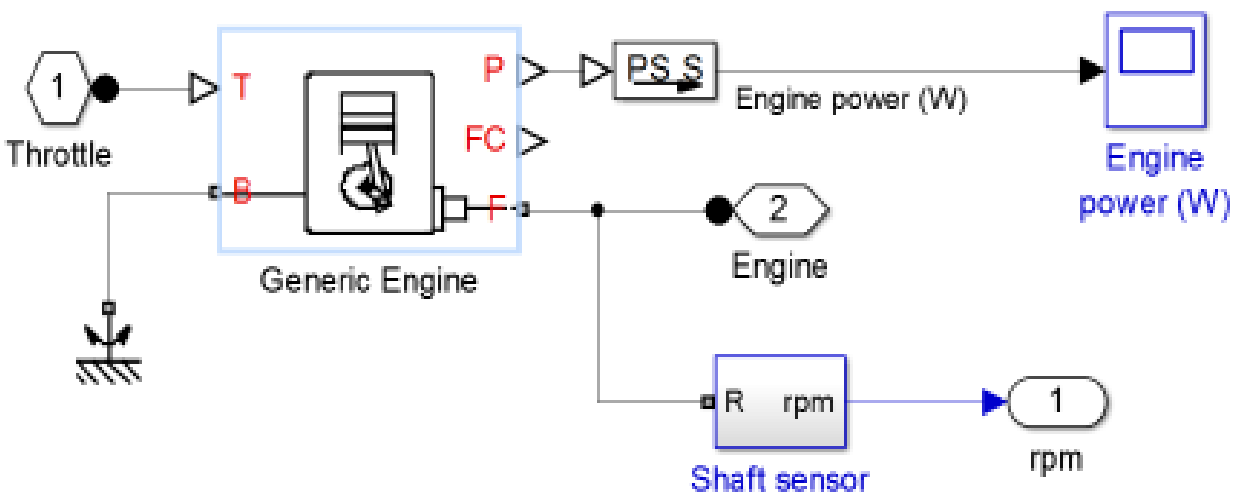

Fig. 1-1: Engine Plant Model Simulink Diagram

The output shaft of engine model is connected to inertia and damper blocks which act as sinks absorbing the energy generated by the engine. The engine plant model has only one controllable input, viz., throttle position which varies from 0 to 1 (0 = throttle off, 1 = wide open throttle).

3.2. PID Controller on Engine Plant Model

On to this engine plant model, PID controller is modeled in Simulink using the standard PID Controller block. Fig. 1-2 shows the Simulink diagram with PID controller and Table 1-2 shows the block parameter values for PID Controller block.

PID algorithm uses error which is the difference between the actual value of the output and the desired target value of the output to decide on the actuator response. In this case actual engine rpm is the actual output, and the desired target engine rpm which is specified in the signal builder or by using a sine wave is the target output.

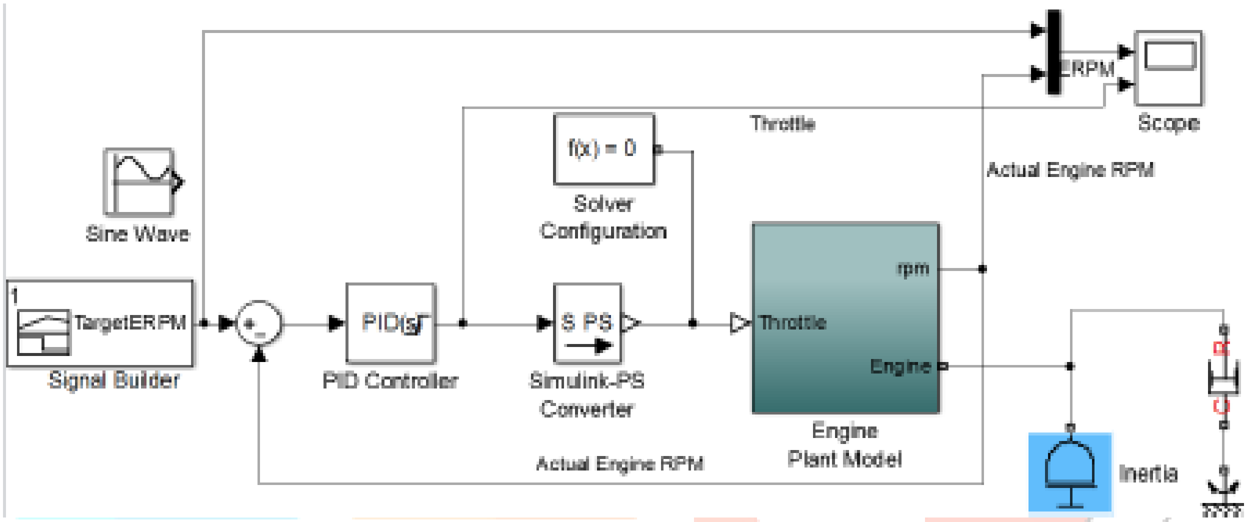

Fig.1-2: PID Controller on Engine Plant Model Simulink Diagram

$$\text{Throttle} = K_p E + K_i \int_o^t E\, dt + K_d \left(\frac{dE}{dt}\right)$$

Where

E= Target ERPM – Actual ERPM

3.3. PID Gain tuning using PID Tune

Determining the best values of the Kp, Ki and Kd gains for this plant model is called as PID gain tuning. Though PID gain tuning is a challenging task, Simulink provides a very useful tool called PID tuner to automate this process. Around the operating point specified by the initial conditions of the plant model, PID tuner first linearizes the non-linear plant model as PID gains can be tuned only on a linear plant model. Linearization is a linear approximation of a nonlinear system that is valid in a small region around the operating point.

After linearization, it applies nonlinear optimization algorithms to decide optimum values of Kp, Ki and Kd to minimize error. In PID tuner, depending upon the control performance required, i.e., limits of overshoot, rise time and steady state error, the gains can be tuned to either make the response time faster or slower, and damped transient behavior as under damped or over damped.

If you are not satisfied with the optimum gains that PID tuner has determined, you can manually vary them by graphically adjusting the sliders for response time and transient behavior in the GUI, and selecting the gains that suit best to your requirement. The effect of response time and transient damping on time domain control responses can be instantly seen in the graph in Fig. 1-3.

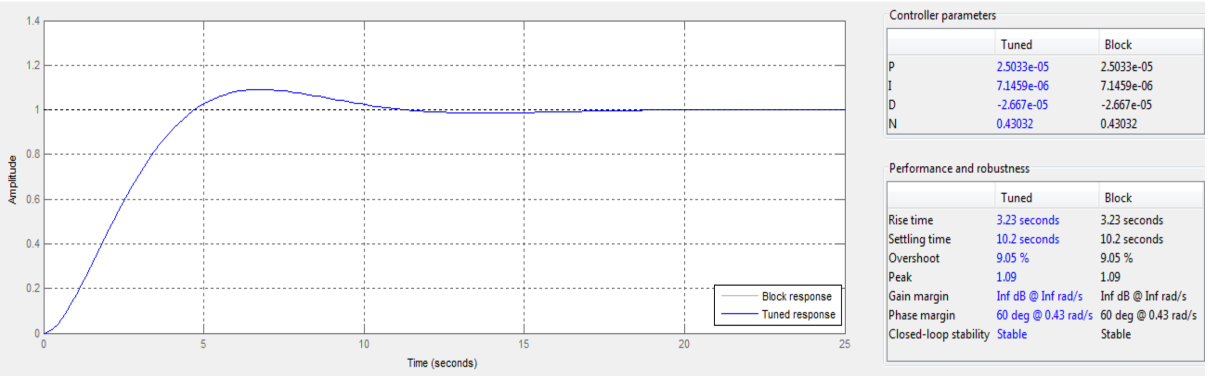

Fig. 1-3: Tuning PID Gains using PID Tuner Software

For this engine plant model, PID tuner has given the PID gains as given in Table 1-1.

PID parameter	Optimum value
Kp	2.5033 e-5
Ki	7.1459 e0
Kd	-2.6670 e-5
N	4.3032 e-1

Table 1.1 Optimum PID Gain Values for Engine Plant Model

IV. SIMULATION RESULTS

The controllability of the system and effectiveness of the control algorithm is tested by varying the target engine rpm sinusoidally and staircase profiles.

4.1. Sinewave Target RPM Response

For sinewave target, the actual engine rpm tracking depends on the frequency of the sinewave. At very slow target variation frequencies (0.05 Hz) the actual rpm tracks target rpm very good as shown in Fig. 1-4. In the top figure red curve is the target rpm, green is the actual rpm and bottom figure is the throttle opening variation. It can be observed that throttle also varies sinusoidally to match a sinusoidal target rpm. The error increases as the frequency of target increases, because the engine system being a mechanical system having time lags and inertia cannot track targets at higher frequencies. The response of actual rpm tracking at frequencies of 0.2 Hz and 1 Hz are shown in Fig 1-5 and Fig 1-6. As can be seen actual rpm is not able to catch up with the extremely fast variations of target rpm.

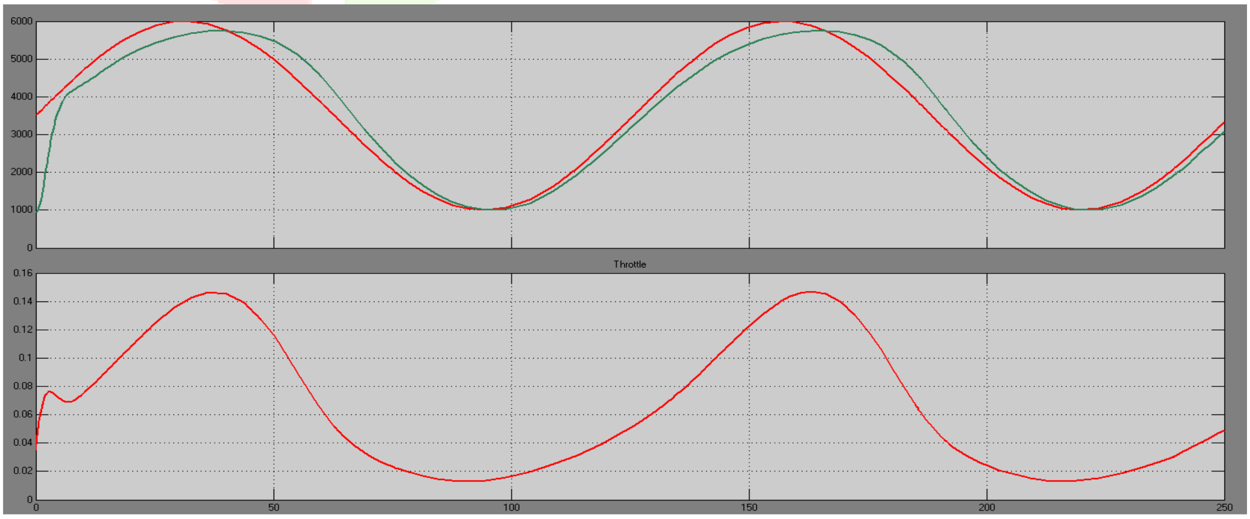

Fig. 1-4: Target Sinewave Frequency 0.05 Hz

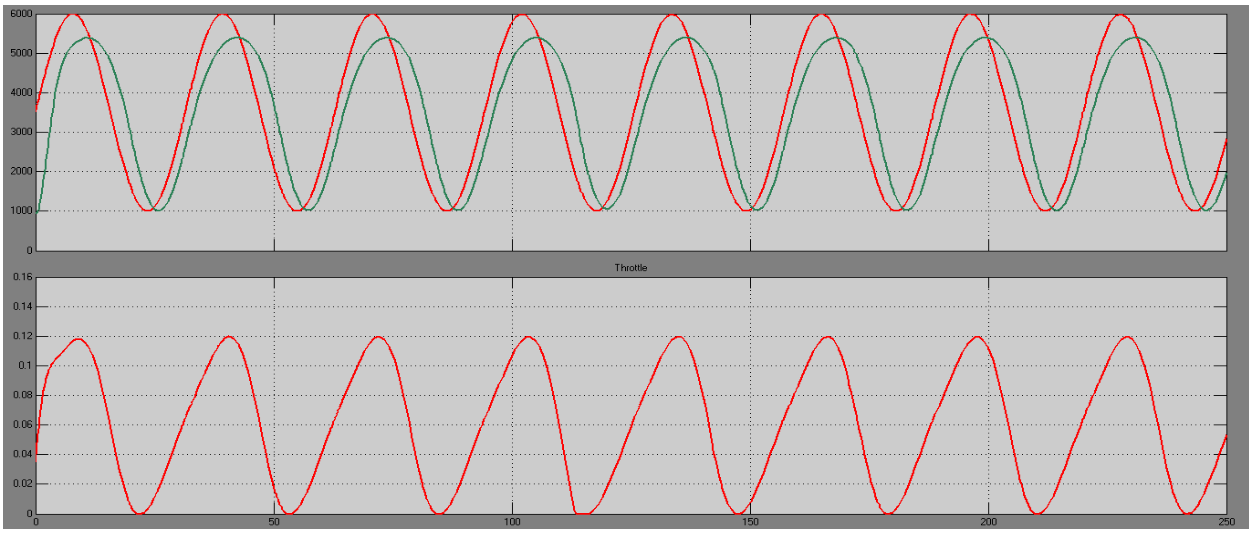

Fig. 1-5: Target Sinewave Frequency 0.2 Hz

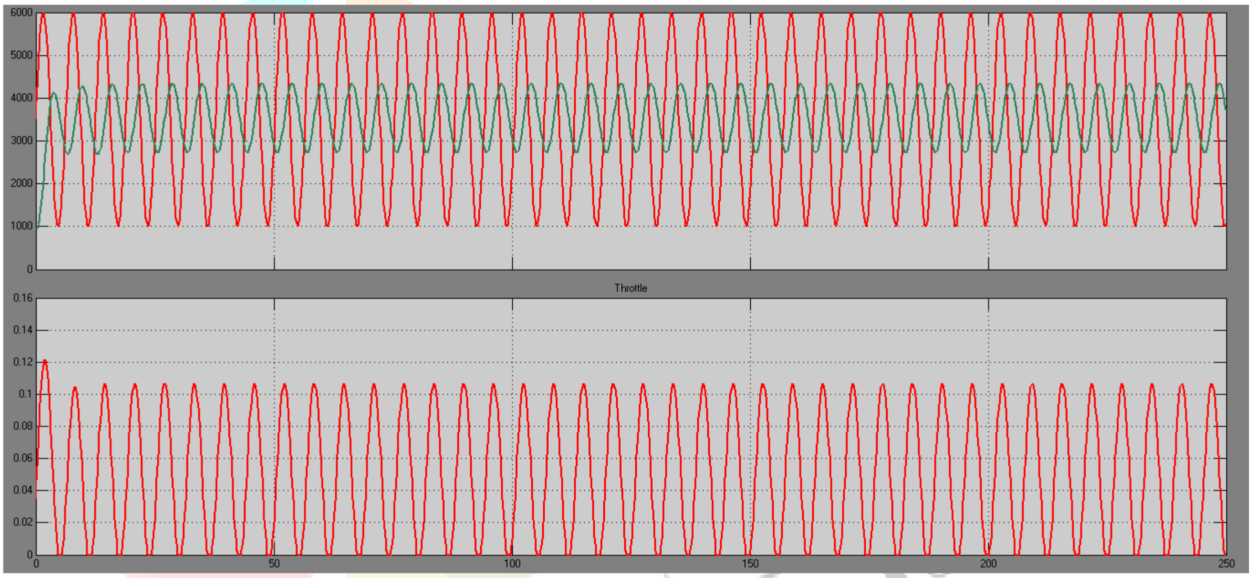

Fig. 1-6: Target Sinewave Frequency 1 Hz

4.2. Staircase Target RPM Response

Using signal builder tool of Simulink, the staircase target engine rpm of both rising up from 1000 to 6000 rpm and falling down back from 6000 rpm to 1000 rpm, in steps of 1000 rpm is shown in Fig. 1-7. At each rpm, target is held constant for 100 sec. The result of actual engine rpm responses are shown in Fig. 1-8. Fig. 1-9 shows the throttle variation during staircase response. As can be seen from Fig. 1-8, the PID controller does a good job in tracking closely the target rpm very well.

The quantified control response measurement at each target rpm is given in Table1.2. Overshoot is the rise of actual rpm above target rpm, rise time is the time taken to reach from 10% to 100% of the target value, and steady state error is the band of oscillations after steady state is reached. It can be seen that steady state error is zero is all cases, and maximum overshoot is 122 rpm at target 2000 rpm, maximum rise time is 31.8 sec at target 6000 rpm. Also maximum undershoot is 116 rpm at target 1000 rpm and maximum fall time is 26.0 sec at target 5000 rpm.

Up				Down		
Target engine rpm	Overshoot	Rise time	Steady state error	Undershoot	Fall time	Steady state error
1000	0	12.0	0	116	3.7	0

2000	122	3.6	0	91	3.8	0
300	77	3.8	0	13	4.7	0
4000	0	4.6	0	0	10.3	0
5000	0	11.5	0	0	26.3	0
6000	0	31.8	0			

Table 1.2 Staircase Control Responses

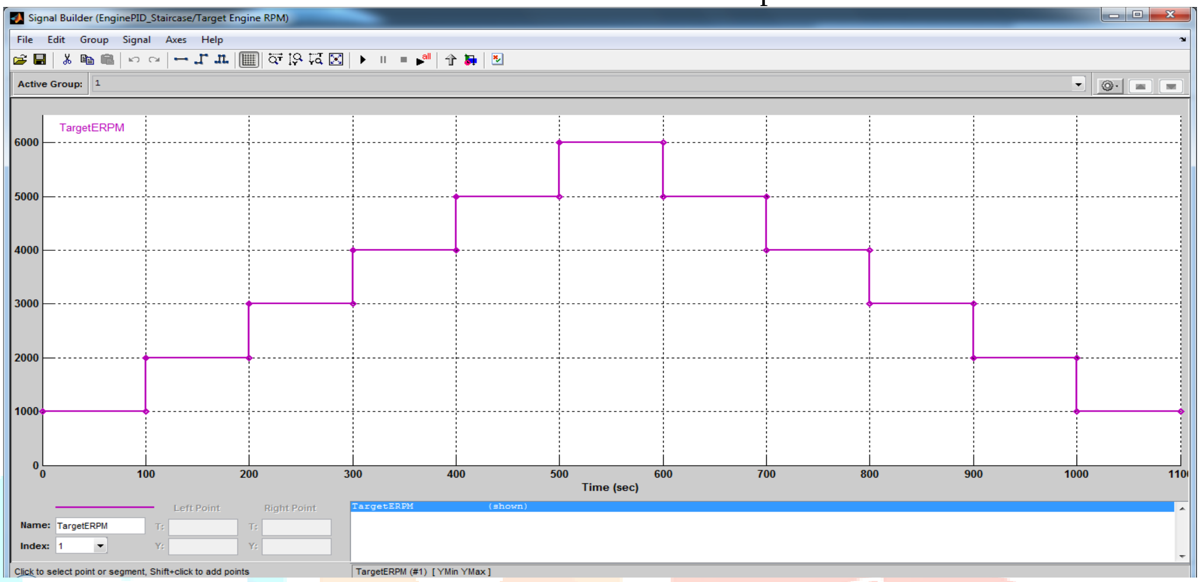

Fig. 1-7: Staircase Up and Down Target Engine RPM Profile

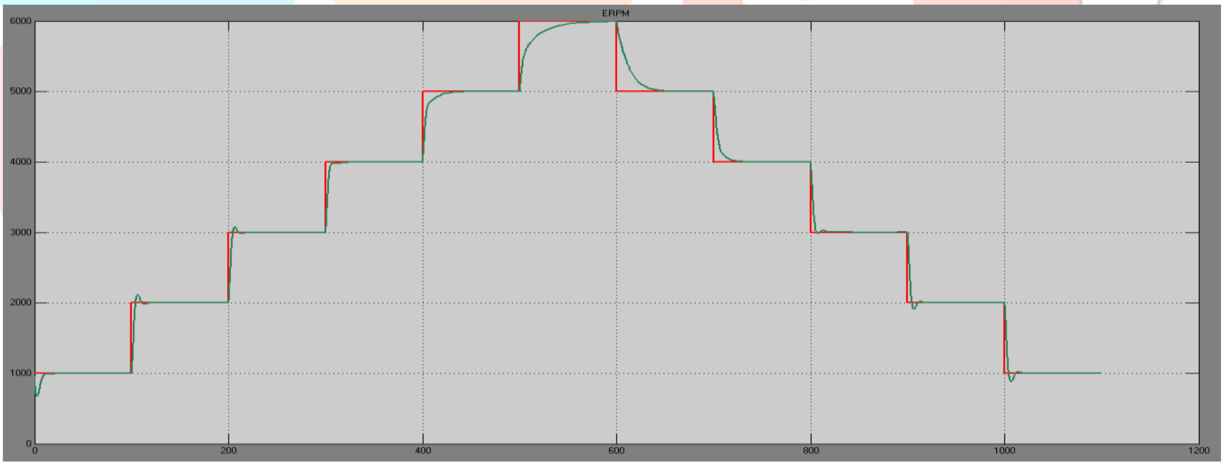

Fig 1-8: Staircase Response for Engine RPM Control

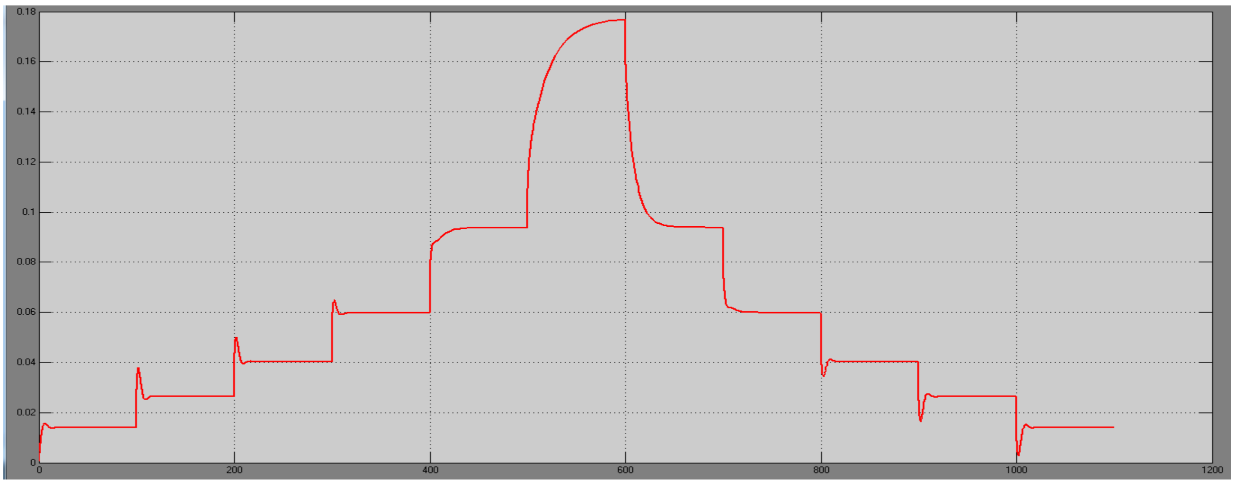

Fig. 1-9: Throttle Variation during Staircase Response

V. CONCLUSION

In this project, it can be observed that the mathematical Simdriveline model captured the vehicle engine phenomena excellently as test cycle results are very much in tune with what is observed in real world. Also PID algorithm controlled engine rpm to a very high level of accuracy with acceptable overshoot and rise times.

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