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Idle Speed Control of an Engine Model Using PID Control System

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Abstract

An engine model has been developed and controlled by the system to prevent stalling conditions when the automobile is at rest. The main input for the engine is throttle position in degree angle and the main control output is the engine speed in rpm. The engine speed is controlled by the amount of air supplied to the engine. Idle speed control is an automotive issue in which fuel gets consumed more during idling state of an automobile while driving and so its predictive rate will increase in the future based on the increase in traffic requirements. The PID control technique is used to tune the parameters of non-linear plant model into linearized conditions for stable equilibrium. Through this intelligent technique, we are able to define control parameters for the Idle Speed Control system of the engine model.

Keywords: *PID Control, Tuning, Intelligent Control, Engine Model*

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1. Introduction

The speed of vehicle engine at idle state is called an idle speed. An idle speed is basically a rotational speed of an engine when the engine is uncoupled to the drivetrain and the throttle pedal is up to the combustion engine. The idle state condition is basically a state in which the motion of the vehicle is at rest when the engine is in a stage of motion.

According to theory, 30% of the average fuel is consumed in an idle state of motion and based on its increasing demand rate in the future, it has become a serious confrontation for automobile industries and researchers. This issue is tagged as a control system error for all engines called idle speed control. The aim of idle speed control is to maintain the desired engine speed or rpm to avoid any sort of disturbances such as torque disturbance which arises due to accessory loads on an engine such as air conditioning, power steering/alternators, automatic transmissions, etc. The desired main input of the idle speed controller is the engine speed. In addition, inputs are described as throttle position, vehicle speed, feedforward indicators from automatic transmission, air conditioning, power steering and battery charging system with some environmental measurements such as engine coolant temperature and barometric pressure. The main control output is achieved by controlling the amount of air supplied to the engine.

To optimize vehicle and powertrain operations at idle conditions a control has to be established especially when there are conflicting requirements such as improved fuel economy, reduced emissions, and stable combustions. The complete idle speed control problem travels from different operational phases such as the transition to/from idle, target speed tracking, regulation of the desired engine speed. Our objective is to apply a PID Tuning control technique to regulate the speed of an

engine.

2. Literature Review

According to the facts, idle speed is measured on the no. of revolutions per minute which should be lower as much as possible to get good results in fuel saving economy. The reduced speed is not going to affect the noise, vibration and harshness quality of the vehicle. But some issues are going to be addressed in relation to speed control like engine stalling at all operating conditions, a transition state from/to idle speed in a smooth and controlled manner. Mostly in production and implementation of an engine, the amount of air controlled by a throttle bypass valve turns the intake manifold air towards the closed primary throttle plate. During the stages of sudden de-acceleration air, the bypass valve supplies some additional amount of air in the start and end of damper work which prevents stalling of the engine and provides a smooth transition from higher speed rpm to idle speed rpm. The air controlled path is considered relatively slow due to intake manifold dynamics and subsequent intake of power related issues. Hence a much faster actuation path is provided through spark control.

Sometimes, the air-fuel ratio is included in the idle speed control (ISC) design for emission control purposes to use the complementary strengths of air and spark control paths. The ISC problem is very well suited for closed loop control system. The various control system techniques used as classical and modern alternatives are LQ, H, μ- synthesis and other optimization based methodologies. Due to the accessibility of neural network and fuzzy-logic based control system, many control systems can use the same plant dynamics phenomenon to select a model for Idle Speed Control design. But our main focus for plant dynamics is the engine. The current observation is

focussed on internal combustion engine operating at idle speed ranges.

Earlier back in 1970-80's, engine models were developed to control and maintain the idle speed conditions of an automobile. The first engine model for ISC design was developed by Powell (1979), Dobner (1980), Powel and Powers (1981) and Coates and Fruechte (1983), based on simple fundamental principal models by Hazell and Flower (1971). According to the developers, the first thing we need is to start our model designing work from a nonlinear descriptive engine combustion torque dynamics. Hence, these models have to be derived from a combination of first principle, physical law, and various identification techniques. These techniques will provide the data and dynamic responses for engine mass air flow rate and torque-functions of rpm, throttle or bypass valve opening area, manifold pressure, air-fuel ratio and spark (by Powell and Cook, 1987).

Some models like the non-linear engine models for idle speed control (by Butts et al, 1995) were used to assess the response of engine and provide adequate data to improve the idle speed of the engine. This type of models has the engine torque motion as rotational dynamics which typically includes flywheel and impeller inertia.

By using certain techniques, a model can be rebuilt into simple linear, modified and discreet version at a constant air/fuel ratio. In these models, a good correlation is established between the experimental test data and dominating mode frequency range. In these models, some tuning is established of pre-calculated model parameters range due to the given assumptions such as linear model structure and a constant air fuel ratio (Mill's, 1992).

The development of non-linear steady state torque map is a necessity in linearizing the system. This map analyses the torque production with mass air/fuel parameter and engine speed to maintain a constant speed at the idle situation. In studies, many models were investigated like cycle accurate model in which engine speed varies by torque fluctuations. The pure torque production delay happens when the intake to combustion stroke delays in a travelling path when the crank angle set from 180-360 degrees of the crankshaft. For this ISC problem, it's important not to neglect this delay because it constitutes strong dynamism for a closed loop system.

There are various control techniques which exist for controlling of an engine such as feedback control for the spark loop, PID control for the air loop, multiple feedforward control which makes the use of accessory load information, along with some adhoc compensation schemes for environmental conditions. A sensitive-guided design (Kokotovic and Rhode, 1986; Hrovat and Jhonson, 1991) was mentioned as a better way to tune idle speed PID controllers.

The performance cost function was reduced by making use of control gain sensitivities based on the error of engine speed which focuses on a fixed control structure. LQ based optimization was popular among several researchers. The advantages of spark feedback were established by Powell and Powers (1981) using a five-state, continuous time engine model. An LQ based controller was demonstrated superior over a traditional PID based controller by Abate and DiNunzio (1990) by conducting experimental implementation.

An extension of LQ techniques was also applied, H methods, in order to build strong control systems (William et al., 1989; Carnevale and Moschetti, 1993). Some trial and error adjustments were done on the frequency weighting functions in case of system sensitivity and excess actuator efforts. The result suggested a frequency shaped PI controller and a PD controller.

The stability of large engine model dissimilarities could not be maintained by H methods. So to incorporate uncertainties, Hrovat and Bodenheimer (1993) implemented the μ synthesis technique which resulted in a controller which could sustain completely different two idle speed operation points. However, the engine with this technique was unstable when spark advance was subjected to hard constraints. Hrovat and Zheng (1994) tried to get rid of this problem by applying Model

Predictive Control (MPC) technique. With this, however, the slight downfall was seen in the unconstrained performance but stability was maintained in the constrained performance for both conditions. Although many optimizing control techniques for regulating the idle speed control was still ruled based with some classical PID controllers, the modern optimization techniques have modified the early model into a simple and more refined models to solve the problem. By regulating the idle speed controller through PID controlling loop we can improve the fuel economy and reduce the emissions of a vehicle.

3. Geometric Model

The standard plant model for an Idle Speed Control is explained in this section. The control input is taken as throttle angle in degrees and the output is taken as engine speed in revolution per minute (rpm). The modelling requirements are discussed below:

3.1 Throttle Mass Flow:

The air mass flow through the throttle valve opening during idling can be modelled using choked flow equation:

$$\varpi_{th} = A_{th} \frac{Pa}{\sqrt{(2RT)}}$$

Where $_{th}$ is the air mass flowrate passing through throttle opening, A_{th} is the effective area of throttle, P_a is the ambient pressure, T is the ambient temperature, R is the universal gas constant. The throttle area is a non-linear function of throttle position but given that during idling the throttle movement is very small, hence a linear relationship will be assumed between the throttle position and throttle effective flow area.

3.2 Intake Manifold:

Based on the isothermal conditions, the pressure dynamics can be modeled as:

$$\frac{d}{dx} P_m = \frac{RTm}{Vm} (\varpi_{th} - \varpi_{eng})$$

Where Pm, Tm, Vm are manifold pressure, temperature and volume respectively and ϖ_{eng} is the air mass flow rate exiting the intake manifold and entering the engine.

3.3 Engine Air mass Flow:

The mean value of fuel-air mixture flow rate entering the engine cylinders can be approximated using the following equation:

$$\varpi_{mix} = \boldsymbol{\eta_\upsilon} \frac{p_m}{RT_m} \frac{\upsilon d\, \varpi e}{4\pi}$$

Where ηv is the volumetric efficiency, $\boldsymbol{\upsilon}_d$ is the displacement volume and ω is the engine speed in radians-per-second. Air mass flow rate entering the cylinders can be found using the formula

$$\varpi_{eng} = \frac{\varpi mix}{[\, 1 + \phi \left(\frac{F}{A}\right)_S \,]}$$

Where $\left(\frac{F}{A}\right)_S$ and Φ represent the stoichiometric fuel-to-air ratio and fuel-to-air ratio normalized by the stoichiometric fuel-to-air ratio, respectively. Φ is referred as equivalence ratio.

3.4 Torque Generation

In general, generation of torque is a non-linear function of engine speed, mass flow rate into the engine cylinder, equivalence ratio, and spark advance:

$$Te = f\,(N, \varpi_{mix}, \Phi, \ SA)$$

Where SA represents the spark advance. Note that the induction of power (IP) delay enters into system dynamics through above equation as torque depends on the delayed value of the mass flow rate into the engine cylinders.

3.5 Engine Rotational Dynamics

The eq. of engine rotational dynamics is as follows:

$$\frac{d}{dt}\,\varpi_e = \frac{1}{J}\,(T_e - T_1)$$

Where J is the inertia of engine in neutral state and T_1 is the load torque on the engine including internal engine friction.

3.6 Idle Speed Control Model

For ISC model, a nonlinear behaviour engine model based on the above set of models was linearized at a nominal speed of 900 rpm to obtain a linear plant model. Considering deviation in throttle position in degrees as input and deviation in engine speed in rpm as the output for this model.

The transfer function of this model is:

$$G\,(s) = K\,\frac{s^2+n_1 s+n^2}{s^3+d_1 s^2+d_2 s+d_3}\ e^{-0.15sec}$$

Note that the time delay is 3rd order and degree one function for the plant transfer function. K, s, d_1, d_2, d_3 are the various parameters of the combustion engine model. Their values are maintained as nominal values at operating conditions for the engine.

The nominal constant parameter values are as follows:

$K=29.8$; $s=1$; $n_1 = 50$;

$n_2 = 833$; $d_1 = 21.2$;

$d_2 = 51.3$; $d_3 = 189.5$

The Induction of Power delay at a nominal speed of 900 rpm is 125 ms assuming this delay as the result of 360^0 of crank rotation or 1 revolution of the crankshaft. However, we are taking this revolution only as an approximation because we don't receive maximum torque calculation at the top dead center of crankshaft rotation. Hence we receive an overall delay of 150ms by the combination of actuator
delay and computational delay.

4. Controller

The PID control system is used in this Idle Speed Control problem to address the speed changes in the system for a continuous-time linearized plant model.

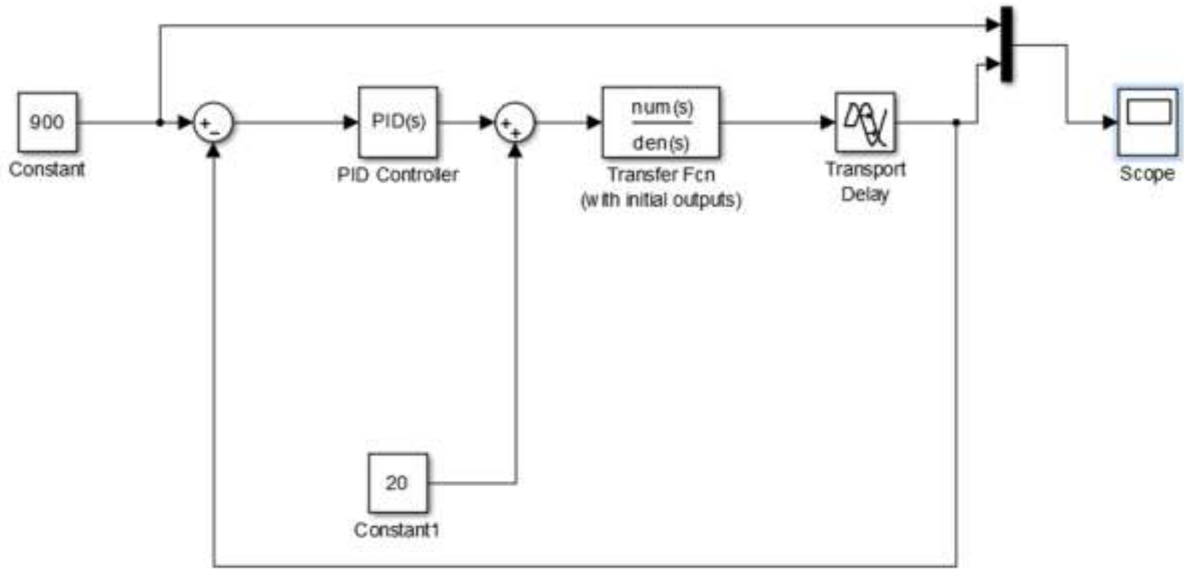

Fig: Simulink Model of Idle Speed Control with PID controller

Proportional

In proportional, a control used is directly proportional to control error with controller gain of the system. A pure proportional control is used as a simplest form of feedback as:

$$u\ (t) = K\ e(t)$$

Where u (t) is the used control; K is the controller gain; e (t) is the control error.

Integral

The main function of Integral is to obtain a zero steady state error. Earlier in Proportional control, we saw that the control requires a control error in order to have a non-zero control signal. But in the Integral controller, a small positive error will always lead to an increasing control signal.

$$u_0 = K\,[e_0 + \frac{e_0}{T}\ t]$$

Where u_0 is the control; K is controlled gain; e_0 is controlled error; T is integral time to eliminate error; t is a function of control.

Derive

This action is used to improve the stability of closed loop in a plant model. It predicts the change in correction for an error of control. The derivative will predict the error change to controller T_d seconds before it can happen in the future. Initially, the checking starts at T_d seconds but when the derivative is too large then it becomes oscillatory.

5. Experimental Results

We are practising some experiments in order to find whether the positions of Proportional, Integral, and Derivative matches correctly to the requirements of the linearized plant model. Since the plant model has a robust behaviour hence we will be practising manually PID tuning methods on various values to get a linear curve.

Case-1:

For P= 0.10; I= 0.00; D=0.00

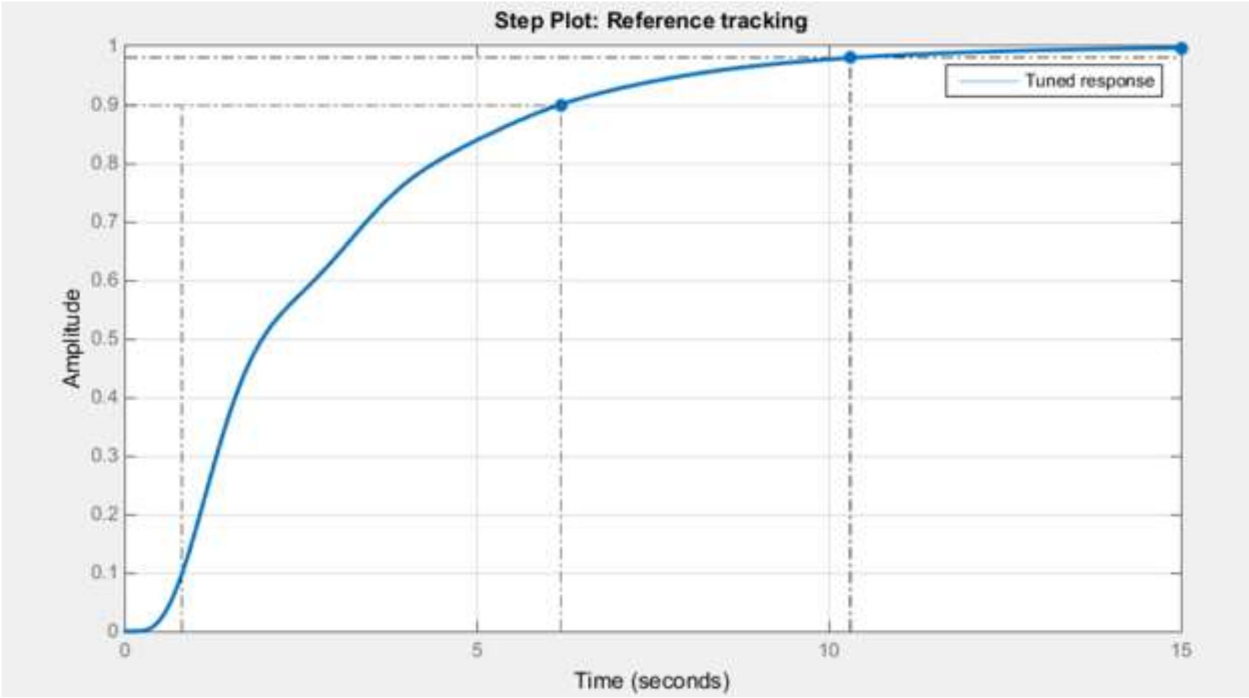

Figure: Step Plot Reference Graph

Controller Parameters

	Tuned
P	0
I	0.007836
D	0
N	100

Performance and Robustness

	Tuned
Rise time	5.37 seconds
Settling time	10.3 seconds
Overshoot	0 %
Peak	1
Gain margin	14.3 dB @ 2.74 rad/s
Phase margin	82.8 deg @ 0.348 rad/s

Figure: PID Control Parameters

Hence, it is clear from this graph that closed loop system with controlled gains defined in the PID block is unstable and not displayed. The rising time of response is 5.37 sec, settling time is 10.3 sec, zero error steady state time is 15 sec at Amplitude one.

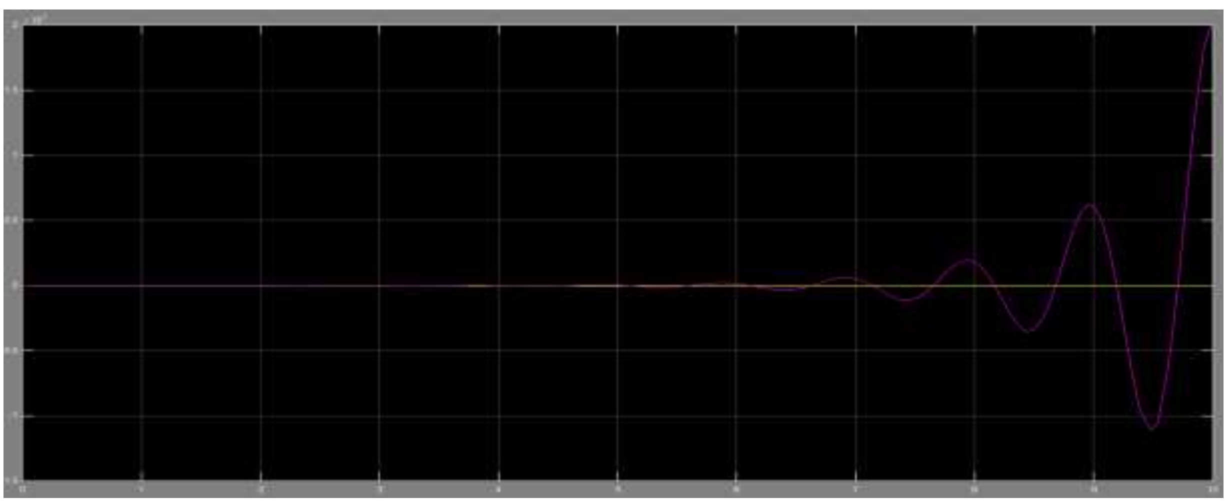

Figure: Simulink of Idle Speed Control of Engine

Through SIMULINK, we receive an unstable curve for Idle Speed control of an engine model. We discover that as the time response increases, the plant model gets highly unstable in a robust control model. Hence, the PID control system of the plant model is not stable at the given points.

Case-2:

For P= 0.0050; I= 0.0020; D= 0

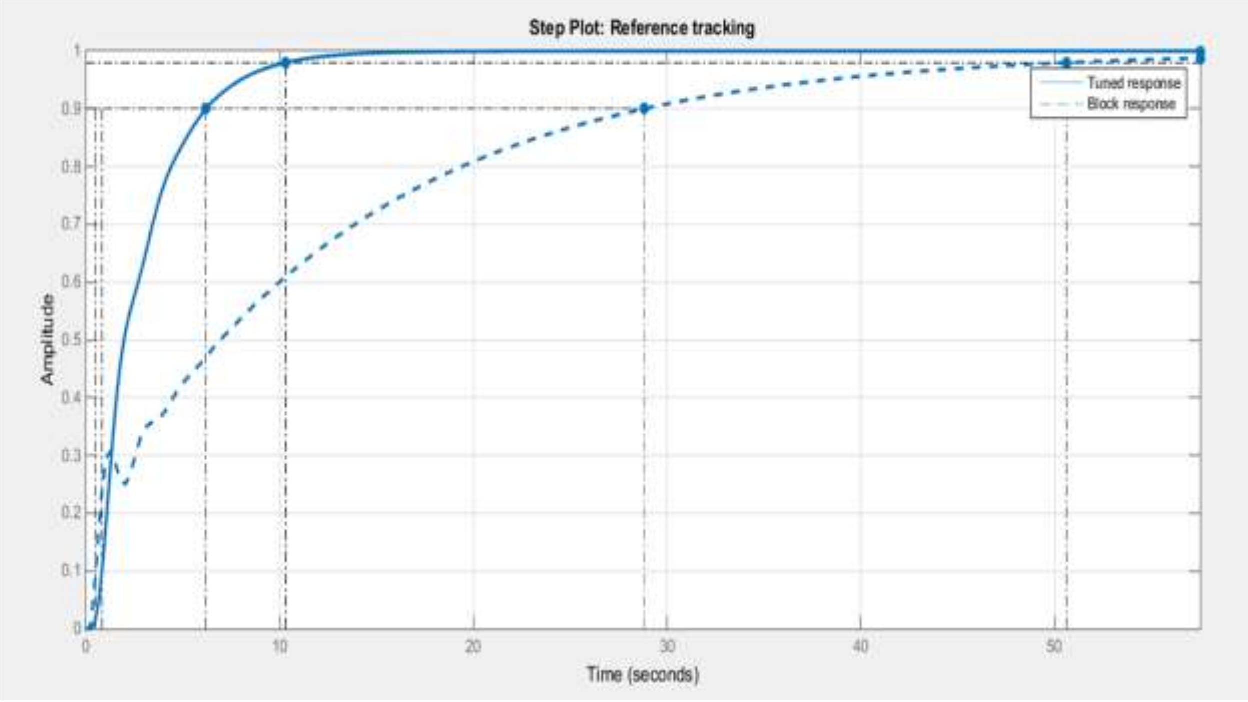

Figure: Step plot Reference Graph

Controller Parameters

	Tuned	Block
P	0	0.005
I	0.007836	0.002
D	0	0
N	100	100

Performance and Robustness

	Tuned	Block
Rise time	5.37 seconds	28.3 seconds
Settling time	10.3 seconds	50.7 seconds
Overshoot	0 %	0 %
Peak	1	0.999
Gain margin	14.3 dB @ 2.74 rad/s	16.7 dB @ 4.58 rad/s
Phase margin	82.8 deg @ 0.348 rad/s	101 deg @ 0.0902 rad/s
Closed-loop stability	Stable	Stable

Figure: PID Control Parameters

So, it is clear from the graph that closed loop system with controlled gains defined in the PID block is stable before and after the tuning process. The rising time of response is 9.31 sec, settling time is 16.8 sec, zero error steady state time is infinite sec at Amplitude one.

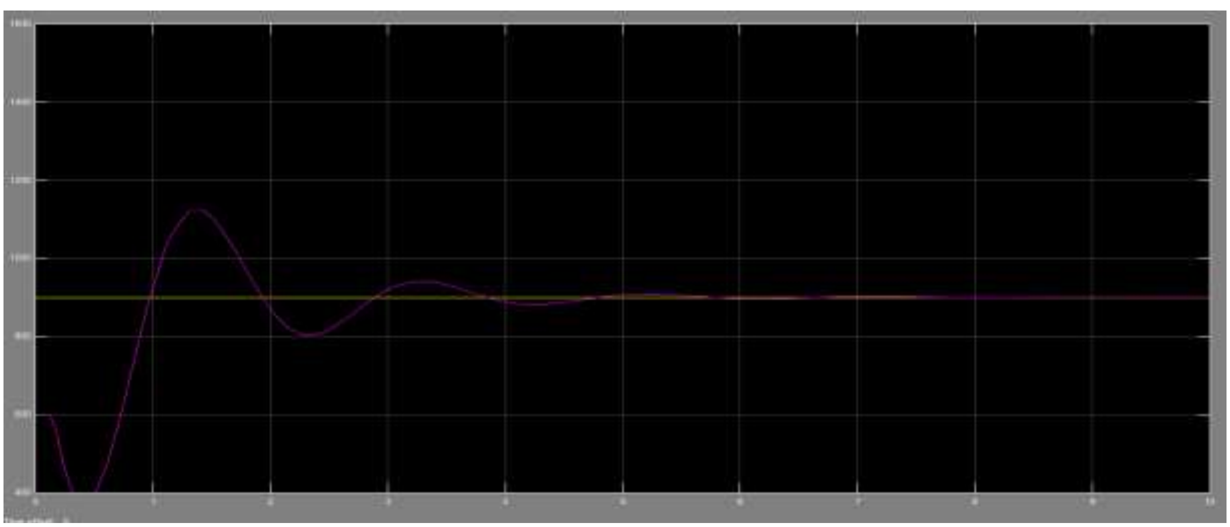

Figure: Final Result of Idle Speed Control Model

In the final Simulink graph, we receive a stable curve for Idle Speed control of an engine model. We discover that the time response increase or decrease doesn’t affect the plant model stability in a robust control model. Hence, the PID control system of the plant model is stable at the given points.

Conclusion

Idle speed control is one of the highest confrontations for the automobile industry and researchers as they were addressing many issues in relation to engine stalling at rest position and fuel saving economy. An engine model plays a major role in defining the correct parameters for the control system. So, we defined the engine model based on certain aspects and analysed it properly to fit in the system.

The control system was introduced to transform the non-linear model into a linearized plant model. This control system was called Proportional, Integral, and Derivative (PID). This system basically controls, integrate and predict the control change in the system for stable equilibrium. Through manual tuning of PID control parameters, we were able to linearize the plant model and determine the correct parameters of the plant model. We conducted various experiments with changing values of PID control system. Through Simulink graphs, we compare various results and were able to find good and correct value for Idle Speed Control of engine model.

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