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Design of a Robust PID Controller for Improved Transient Response Performance of a Linearized Engine Idle Speed Model

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Abstract: *One of the advantages of automatic control system is its application to the regulation of internal combustion engines. An area of significance in fuel energy reduction and efficiency improvement is the control engine idling speed. This paper has presented a control system for improved transient response performance of a linearized engine idle speed model. In order to realize the aim of the work, nonlinear dynamics of engine idle speed are obtained and were later linearized. A proportional integral and derivative (PID) controller was designed using a robust response time tuning of the PID to give the Robust PID controller. The designed controller was integrated with the linearized engine idle speed model. Simulations were conducted using Matlab software for when no step input disturbance entered the control loop and when step input disturbance entered the control loop to ascertain the robustness of the designed controller. The results obtained comfirmed the robustness of the controller.*

Keywords: *Engine, Idle Speed, Control System, PID, Controller*

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I. INTRODUCTION

One of the advantages of automatic control system is its application to regulation of internal combustion engines. This has provided numerous benefits such as emission reduction, fuel efficiency improvement, and power delivery [1]. A significant area of application is in fuel efficiency management of vehicles when the engine is in idle state. This is the state when the engine is not directly coupled to the drive and no application of force to the throttle pedal. This condition is associated with a certain level of engine rotational speed known as idle speed. One obvious requirement to improve engine fuel consumption is to maintain its idle speed to a desirable level during idle condition in the presence of known and unknown load disturbances. One of such disturbances is the load torque disturbance which could be caused by accessory loads on an engine such as air conditioning, power steering/alternators, and automatic transmission. Idle speed control is aimed at maintaining a desired speed of an engine (usually expressed in revolution per minute) in the presence of disturbances. It is required that the idle speed controller should be able to operate with close bounds, time delay allowance, inverse response, ability to change control objectives and sensor failure, and essentially reduced the fuel consumption so as to improve cost of operation.

The control system for engine idle speed has been evolutionary. This was as result of the need to improve fuel efficiency and economy as well as reducing emissions. Idle speed is measured on the number of revolutions per minute; this should be as low as possible to get good results in a fuel saving economy. It is increasingly important to achieve control over transient behavior and meet performance objectives over the life of engines. 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 exists at different operational phases such as target speed tracking and regulation of a desired engine speed.

One main effect of idling speed of automobile engine is that it adversely affects the fuel economy as it increases. For this reason, it must be maintained to an appreciable level to save fuel energy and economy. In order to achieve this, is required that a control system should be implemented so as to ensure proper regulation of the idling speed and maintaining it at an optimal level. In this paper, a simplified linearized engine idle speed model is presented. A robust proportional integral and derivative (PID) controller is developed for the control system, which is aimed at tracking a referenced idling speed of an engine with improved transient response performance. Hence, the primary aim of this paper is to design a robust control system for a linearized engine idle speed model with improved tracking and transient response performance.

II. RELATED WORKS

Honey and Best [2] carried out an online PID tuning for engine idle speed control using continuous action reinforcement learning automata. It employed continuous action reinforcement learning automata (CARLA) for automating the tuning process. It stated that the parameters of the PID controller can be tuned on-line to minimize a performance objective. The method was demonstrated on engine idle-speed control. The control law was initially applied to a nominal engine model, and then to a practical application using a Ford Zetec engine in a test cell. It concluded that the CARLA provided marked performance benefits when compared to Ziegler-Nichols tuned controller in the application. Abhishek [3] presented an idle speed control of an Engine model using PID control system. It used a PID control technique to tune the parameters of a linearized model of engine idle speed for stable equilibrium conditions. Feng-chi et al [4] presented a paper on adaptive idle speed control for spark ignition engine. It proposed an adaptive multi-input single-output (MISO) controller based on self-tuning regulator (STR) for idle speed engine. It stated that the spark timing and idle air control were simultaneously applied as control inputs so as to maintain the desired speed. These control inputs were designed using STR of proportional (P) and proportional integral (PI) respectively. It employed the Recursive Least Square technique to identify the engine model as a first order MISO linear model. In order to design the adaptive MISO controller, pole placement technique was employed. The performances of the proposed controller were evaluated using a nonlinear engine model in Matlab/Simulink. Robustness analysis was also performed also using 10% uncertainties of the system parameters. It concluded that results obtained from the simulation performed showed significant reduction of speed deviations under the presence of torque disturbances and model uncertainties. Sharma et al [5] conducted a research on real time model predictive idle speed control of ultra-lean burn engines: experimental results. It applied a linear time varying model predictive approach for idle speed control of ultra-lean burn internal combustion engines. It asserted that unlike conventional gasoline engines, the method used fuel flow as the primary control variable to make up for sudden fluctuations in engine load resulting in fuel assisted idle speed control strategy. It also stated the spark angle was simultaneously maintained at a value to give optimal brake torque as well reducing of emissions to negligible low level with the engine operation constrained to ultra-lean burn mode. It concluded that the method was demonstrated on an inline prototype 4-litre, 6-cylinder hydrogen fuelled internal combustion engine. Yang et al [6] presented a research work on responsiveness improvement of idling speed control for automotive using sliding mode control (SMC). A Mean-Value Engine Model (MVEM) with disturbances and parameters perturbations was investigated using SMC as a form variable structure control. This was done to address idle speed control process instability. It stated that the simulation results showed that stability of idle speed for an engine subjected to disturbances, parameter variations and background noise was greatly improved by the SMC compare to conventional proportional integral (PI) controller. Xiaocheng et al [7] conducted a research on fuzzy proportional integral derivative (PID) model-based study on idle control of gas engine. It analyzed the algorithm of PID controller, algorithm of fuzzy control, and algorithm of fuzzy PID control. These controllers were designed based on algorithm principles, and were applied to idle control process of the automobile gas engine. It concluded that the fuzzy PID controller showed higher superiority to the three algorithms considered in the process of cooling high idle start, switching high idle to low idle and low idle to high idle. Kang et al [8] in their work on idle speed controller based on active disturbance rejection control in diesel engine, proposed an idle speed controller to compensate for changes in engine load and friction torque in passenger car diesel engines. It applied an active disturbance rejection control (ADRC) framework to an idle speed controller that would compensate for disturbances such as load friction torque. It also designed a feedforward compensator into the ADRC to enhance the disturbance rejection performance. It asserted that the proposed controller was validated by engine and vehicle experiments, which were compared with a commercial controller. Wong et al [9] conducted a research on engine idle speed system modeling and control optimization using artificial intelligence. It proposed a novel modeling and optimization approach for steady state and transient performance tune-up of an engine at idle speed. An engine idle speed model based on experimental sample data was developed using Latin hypercube sampling and multiple-input and multiple-output (MIMO) least-square support vector machines (LS-SVMs). A generic algorithm (GA) and particle swam optimization (PSO) were applied to an optimal electronic control unit (ECU) which automatically set under different constraints defined by user. It further added a conventional multilayer feedforward neural network (MFN) to develop the engine idle speed so as to illustrate the advantages of the MIMO LS-SVM. It compared the modeling accuracies of MIMO LS-SVM and MFN. It stated that the predicted results using the estimated model from LS-SVN were in conformity with the actual test results. It asserted that both the GA and PSO optimization results showed a good improvement on the performance of a test engine idle-speed, but the PSO was more efficient.

III. SYSTEM MODELING AND CONFIGURATION

(a) Air System Equation

The air system equation is of two parts: the air flow which passes through the throttle and the air flow and pressure characteristics of the intake manifold.

The rate of flow of air mass into the engine or precisely into the intake manifold is maintained and regulated by the throttle. The throttle being a metal plate is positioned by a positioning servo known as throttle servo. In idling state, the opening of the throttle valve for air mass flow can be modeled as in Eq.(1) using choked flow equation [3]:

$$\dot{w}_{ath} = A_{th} \frac{P_{amb}}{\sqrt{2RT_{amb}}} \tag{1}$$

where $\dot{w}_{ath}$, is the flow rate of air mass through the throttle; A_{th} , is the area of the throttle which is a function of the throttle angle, α . In idle state, the throttle displacement is very small though the throttle area is a non linear function of the throttle position. It will be assumed in this context that the relationship existing between the effective area and position of the throttle is linear.

The primary function of the intake manifold is to distribute the combustion mixture to the various intake ports in the cylinder head. The dynamic equation of the intake manifold pressure is modeled based on based on the isothermal conditions [3] as:

$$\frac{d_{pim}}{dt} = \frac{RT_{im}}{V_{im}}(\dot{w}_{ath} - \dot{w}_c) \tag{2a}$$

where p_{im} , V_{im} , T_{im} ' R , and $\dot{w}_c$ manifold pressure, volume, temperature, universal gas constant, and air mass flow into the cylinder. The mean value of fuel-air mass mixture flow into the engine cylinders is approximately given by:

$$\dot{w}_{meancyl} = \eta_{vol} \frac{P_{im} v_d N_e}{4\pi} \tag{2b}$$

where η_{vol} , V_d , and N_e , are the volumetric efficiency , the displaced volume, and the engine speed in revolution per minute (RPM). The mass flow rate of air into the cylinders can be obtained using:

$$\dot{w}_c = \frac{\dot{w}_{meancyl}}{1 + \phi\left(\frac{F}{A}\right)_s} \tag{2c}$$

where $\left(\frac{F}{A}\right)_s$ and ϕ represent the stoichiometric fuel-to-air ratio and equivalence ratio.

(b) Engine Torque Equation

The torque generated by the engine is a non-linear function of engine speed, mass flow rate into the engine cylinder, equivalence ratio and spark advance. The output torque of the engine is given by:

$$T_o = g\left(N_e, \dot{w}_{meancyl}, \phi, SA\right) \tag{3}$$

(c) Engine Rotational Equation

The equation describing the rotational dynamics of the engine crank shaft with moment of inertia, J is given according to Newton's second law [10] as:

$$J\frac{dN_e}{dt} = \frac{30}{\pi}(T_o - T_L) \tag{4}$$

T_L is the external load (or disturbance) torque which can come from air-conditioning compressor. So far the dynamic equations of the component parts of the idle speed engine have been obtained.

3.1. MODEL LINEARIZATION

A linearized model of an engine idle speed equation is presented using the approach in [10]. Hence, Eq. (1) and (2a) are given as Eq. (5) and (6):

$$\dot{w}_{ath} = k_{\alpha} \Delta \alpha \tag{5}$$

Eq. (5) can be as a function of $\Delta \alpha$ by:

$$\dot{w}_{ath} = g_1(\Delta \alpha) \tag{6}$$

where α is the throttle angle, k_{α} is the linearized rate of airflow rate sensitivity, and $\Delta \alpha$ is the increment in throttle angle from the operating point at which the system is linearized.

The intake manifold pressure, Eq. (2a), can be expressed as follows after linearization

$$\Delta \dot{P}_m = k_{pim} \left(\frac{\partial \dot{w}_{ath}}{\partial p} - \frac{\partial \dot{w}_c}{\partial p} \right) \Delta P_{im} + k_{pim} k_{\alpha} \Delta \alpha - k_{pim} k_{N_e} \Delta N_e \tag{7}$$

where $k_{p_{im}} = \frac{p}{\dot{w}_{ath} - \dot{w}_c}$, $k_{N_e} = \frac{\partial \dot{w}_c}{\partial N_e}$. A time constant, $\tau_{p_{im}}$ can be represented as a transfer function of the intake manifold from Eq. (7) as:

$$\tau_{p_{im}} = \frac{-1}{k_{p_{im}} \left(\frac{\partial \dot{w}_{ath}}{\partial p} - \frac{\partial \dot{w}_c}{\partial p} \right)} \tag{8}$$

Hence, Eq. (7) can be generally represented as a function of ΔP_{im}, ΔN_e, and $\Delta \alpha$ by:

$$\Delta \dot{P}_{im} = g_2(\Delta P_{im}, \Delta N_e, \Delta \alpha) \tag{9}$$

The engine torque or output torque is again represented in this context as linear model given by:

$$\Delta T_{eng} = H_p \Delta P_{im} + H_f \Delta f_d + H_{\delta} \Delta \delta + f_N \Delta N_e \tag{10}$$

where Δf_d is the increment in fuel delayed by induction-to-power (IP) lag, ΔN_e is the increment in engine speed, and $\Delta \delta$ is the increment in the ignition timing in degree before top dead centre (TDC). H_p is the influence of delayed pressure on torque, H_f is the influence of delayed fuel on torque, H_{δ} is the influence of spark advance on torque, and f_N is the engine friction. The terms H_p, H_f and H_{δ} give the combustion torque.

The rotational dynamics model, Eq. (4), can be linearized by substituting Eq. (10) into Eq. (4). This gives:

$$\Delta \dot{N}_e + \frac{30}{J\pi} \left(\frac{2N_e}{k_j^2} - f_N \right) \Delta N_e = \frac{30}{J\pi} \left(H_p \Delta P_{im} + H_f \Delta f_d + H_{\delta} \Delta \delta - \Delta T_L \right) \tag{11}$$

Equation (11) can be expressed as a function as follows:

$$\Delta \dot{N} = g_3(\Delta p_{im}, \Delta N_e, \Delta \delta, \Delta f_d, \Delta T_L) \tag{12}$$

If $k_{RN} = \frac{30}{J\pi}$ a transfer function is taken from Eq. (11) which gives a time constant as:

$$\tau_{RN} = \frac{1}{k_{RN} \left(\frac{2N_e}{k_j^2} - f_N \right)} \tag{13}$$

In order to determine the model parameters by testing using Matlab software, a typical nominal values of the parameters for a four stroke, six-cylinder engine at N_e = 600 RPM [11] is used. The control sampling time, T for a six-cylinder engine is equal to 120 degrees. Typical parameters of an engine idle speed used in this context; which is that of six-cylinder engine model is presented in Table 1.

Table 1. Typical parameters of six-cylinder-engine model at N_e= 600 RPM [11]

Parameter	Unit	Value
k_α	(lb/hr)/deg	20.000
$\tau_{p_{im}}$	Sec	0.210
$k_{p_{im}}$	lbf-h/(lbm-in^2-sec)	0.776
H_p	ft-lbf/psi	13.370
H_δ	ft-lbf/deg	10.000
T	Sec	0.033
τ_{RN}	Sec	3.980
k_{RN}	Rpm/(ft-lbf-sec)	67.200
k_{N_e}	lbm/(rpm-hr)	0.080
H_f	ft-lbf/lbm	36.600

3.1.2. STATE-SPACE REPRESENTATION OF THE LINEARIZED ENGINE IDLE SPEED MODEL

A state space representation of the system is realized by choosing a state matrix which has three elements: the mass flow rate through the throttle ($\dot{w}_{ath}$), the intake manifold pressure (P_{im}) and the engine speed (N_e). The inputs are the change (or increments) in throttle angle, spark timing, delayed fuel, and the load torque. Hence, Eq. (6), (9) and (12) can be represented by the following state-space matrix by taking their partial derivative:

$$\begin{bmatrix} \dot{w}_{ath} \\ \Delta \dot{P}_{im} \\ \Delta \dot{N}_e \end{bmatrix} = \begin{bmatrix} \frac{\partial g_1}{\partial \dot{w}_{ath}} & \frac{\partial g_2}{\partial \dot{w}_{ath}} & \frac{\partial g_3}{\partial \dot{w}_{ath}} \\ \frac{\partial g_1}{\partial P_{im}} & \frac{\partial g_2}{\partial P_{im}} & \frac{\partial g_3}{\partial P_{im}} \\ \frac{\partial g_1}{\partial N_e} & \frac{\partial g_2}{\partial N_e} & \frac{\partial g_3}{\partial N_e} \end{bmatrix} \begin{bmatrix} \dot{w}_{ath} \\ \Delta P_{im} \\ \Delta N_e \end{bmatrix} + \begin{bmatrix} \frac{\partial g_1}{\partial \alpha} & \frac{\partial g_2}{\partial \alpha} & \frac{\partial g_3}{\partial \alpha} \\ \frac{\partial g_1}{\partial \delta} & \frac{\partial g_2}{\partial \delta} & \frac{\partial g_3}{\partial \delta} \\ \frac{\partial g_1}{\partial f_d} & \frac{\partial g_2}{\partial f_d} & \frac{\partial g_3}{\partial f_d} \end{bmatrix} \begin{bmatrix} \Delta \alpha \\ \Delta \delta \\ \Delta f_d \end{bmatrix} + \begin{bmatrix} \frac{\partial g_1}{\partial T_{Load}} \\ \frac{\partial g_2}{\partial T_{Load}} \\ \frac{\partial g_3}{\partial T_{Load}} \end{bmatrix} \Delta T_{Load} \quad (14)$$

The coefficients of the matrices are evaluated at the given nominal operating point of either of the following: $\dot{w}_{ath}$, p_{im} , N_e, α , δ , f_d , T_{Load} .

The coefficients of the matrices at operating point for a six-cylinder engine at nominal engine speed N_e = 600 RPM are presented below:

$$\begin{bmatrix} \dot{w}_{ath} \\ \Delta \dot{P}_{im} \\ \Delta \dot{N}_e \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -1/\tau_{P_{im}} & -k_{P_{im}} k_{Ne} \\ 0 & k_{RN} H_P & -1/\tau_{RN} \end{bmatrix} \begin{bmatrix} \dot{w}_{ath} \\ \Delta P_{im} \\ \Delta N_e \end{bmatrix} + \begin{bmatrix} k_\alpha & 0 & 0 \\ k_{P_{im}} k_\alpha & 0 & 0 \\ 0 & k_{RN} H_\delta & k_{RN} H_f \end{bmatrix} \begin{bmatrix} \Delta \alpha \\ \Delta \delta \\ \Delta f_d \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ -k_{RN} \end{bmatrix} \Delta T_L \quad (15)$$

Evaluating the values of the coefficients of the matrices in equation (15) using Table 1 gives:

$$\begin{bmatrix} \dot{w}_{ath} \\ \Delta \dot{P}_{im} \\ \Delta \dot{N}_e \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -4.762 & -0.062 \\ 0 & 898.464 & -0.251 \end{bmatrix} \begin{bmatrix} \dot{w}_{ath} \\ \Delta P_{im} \\ \Delta N_e \end{bmatrix} + \begin{bmatrix} 20 & 0 & 0 \\ 15.52 & 0 & 0 \\ 0 & 672 & 2460 \end{bmatrix} \begin{bmatrix} \Delta \alpha \\ \Delta \delta \\ \Delta f_d \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ -67.20 \end{bmatrix} \Delta T_L \quad (16)$$

The state space Eq. (16) can be represented in form of a state equation as:

$$\dot{x} = Ax + Bu + bv \quad (17)$$

$$y = Cx + Du \quad (18)$$

where

$$x = \begin{bmatrix} \dot{w}_{ath} \\ \Delta P_{im} \\ \Delta N_e \end{bmatrix} \quad u = \begin{bmatrix} \Delta \alpha \\ \Delta \delta \\ \Delta f_d \end{bmatrix} \text{ and } v = \Delta T_L \quad (19)$$

$$A = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -4.762 & -0.062 \\ 0 & 898.464 & -0.251 \end{bmatrix}, B = \begin{bmatrix} 20 & 0 & 0 \\ 15.52 & 0 & 0 \\ 0 & 672 & 2460 \end{bmatrix}, b = \begin{bmatrix} 0 \\ 0 \\ -67.20 \end{bmatrix} \quad (20)$$

$$C = \begin{bmatrix} 0 & 0 & 1 \end{bmatrix} \text{ and } D = \begin{bmatrix} 0 & 0 & 0 \end{bmatrix}$$

3.1.3 TRANSFER FUNCTION OF LINEARIZED ENGINE IDLE SPEED MODEL

In order to carry out simulation in Matlab/Simulink environment, the state-space Eq. (17) and (18) are transformed into Laplace form using the formula:

$$G_p(s) = \frac{C adj(sI - A)B}{\det(sI - A)} \tag{21}$$

Substituting the Matrices A, B, and C yields the closed loop transfer function of a second order plant as in Eq. (22).

$$G_p(s) = \frac{2460s + 11700}{s^2 + 5.013s + 56.97} \tag{22}$$

3.2. SYSTEM CONFIGURATION AND CONTROLLER DESIGN

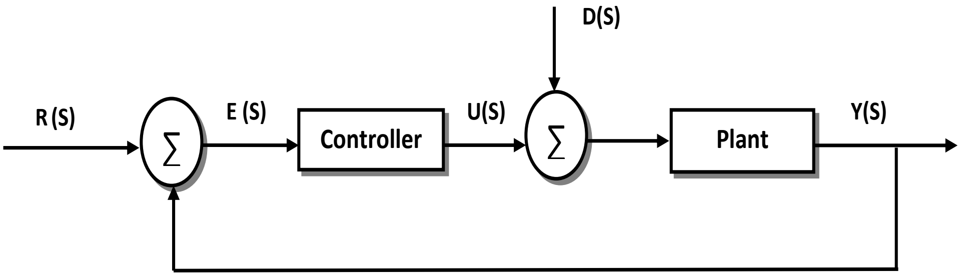

Figure 1: Block diagram of engine idle speed control system

In Fig. 1, R(s) is the desired engine idle speed, E(s) is the error signal, U(s) is the control input, D (s) is the disturbance input, and Y(s) is the controlled engine idle speed. R(s) = 600 RPM and D(s) = unit step disturbance. A unit step disturbance has been deliberately introduced in this context into the loop to serve as load torque disturbance while neglecting the value -67.2 for change in load torque.

From the Fig.1, an equation is obtained for the output response as:

$$Y(s) = R(s)\left[\frac{G_c(s) * G_p(s)}{1 + G_c(s) * G_p(s)}\right] + D(s)\left[\frac{G_p(s)}{1 + G_c(s) * G_p(s)}\right] \tag{23}$$

where $G_c(s)$ is the designed robust proportional integral and derivative (PID) controller.

The design method adopted in this context is PID tuning and a robust response time tuning method was adopted for this purpose using the automatic tuning of a single input and single output (SISO) design task of the Matlab control tools. The value of $G_c(s)$ is given by Eq. (24) with a loop gain of 0.21078.

$$G_c(s) = \frac{0.21078 \times (1 + 0.023s)}{(1 + 5.4e-06s)(1 + 0.079s)} \tag{24}$$

The transfer function of the engine idle speed loop when a controller is not in the loop is given by:

$$G_n(s) = \frac{1.476e06s + 7.026e06}{s^2 + 2465s + 1.177e04} \tag{25}$$

The open-loop transfer function of the control loop in Fig. 1 is given by:

$$G_o(s) = G_c(s) * G_p(s) \tag{26}$$

$$G_o(s) = \frac{2.7446e07(s + 4.76)(s + 43.8)}{(s + 12.6)(s + 1.84e05)(s^2 + 5.013s + 56.97)} \tag{27}$$

The expressions in Eq. (23) are represented as:

$$R(s)\left[\frac{G_c(s) * G_p(s)}{1 + G_c(s) * G_p(s)}\right] = G(s) \tag{28}$$

where
$$G(s) = \frac{1.6468e10(s + 4.76)(s + 43.8)}{(s + 1.839e05)(s + 4.835)(s^2 + 162s + 6586)} \tag{29}$$

$$D(s)\left[\frac{G_p(s)}{1+G_c(s)*G_p(s)}\right]=W(s) \quad (30)$$

where $$W(s)=\frac{1.476e06(s+4.76)(s+12.6)(s+43.8)(s+1.84e05)}{(s+1.839e05)(s+43.8)(s+4.835)(s^2+162s+6586)} \quad (31)$$

Equations (29) and (28) are combined to give the closed-loop response as:

$$Y(s)=\frac{1.6468e10(s+4.76)(s+43.8)}{(s+1.839e05)(s+4.835)(s^2+162s+6586)}+\frac{1.476e06(s+4.76)(s+12.6)(s+43.8)(s+1.84e05)}{(s+1.839e05)(s+43.8)(s+4.835)(s^2+162s+6586)} \quad (32)$$

IV. SIMULATION RESULTS AND DISCUSSION

4.1. SIMULATION RESULTS

In this context simulations were performed considering four possible cases. Fig.2 shows the simulation result when the controller is not integrated into the engine idle speed control loop. Fig.3 shows the simulation result when the controller is in the engine idle speed loop, assuming no disturbance enters the system. Fig.4 shows the result of the plant response considering input step disturbance rejection handling of the controller. Fig.5 shows the closed loop step response performance of the engine idle speed system considered in this context. The characteristics of the step response performance of the system are expressed in continuous time in terms of overshoot, settling time, and rise time.

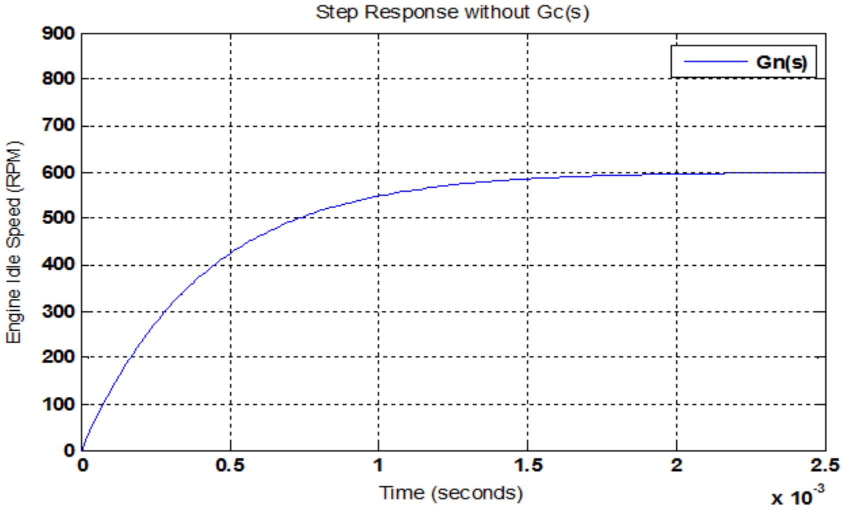

Fig. 2: Step response without controller in the loop

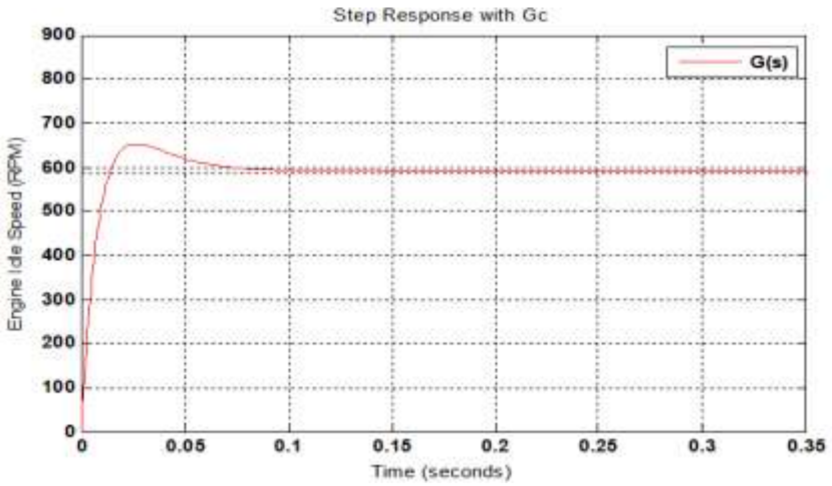

Fig. 3: Step response with controller in the loop (disturbance = 0)

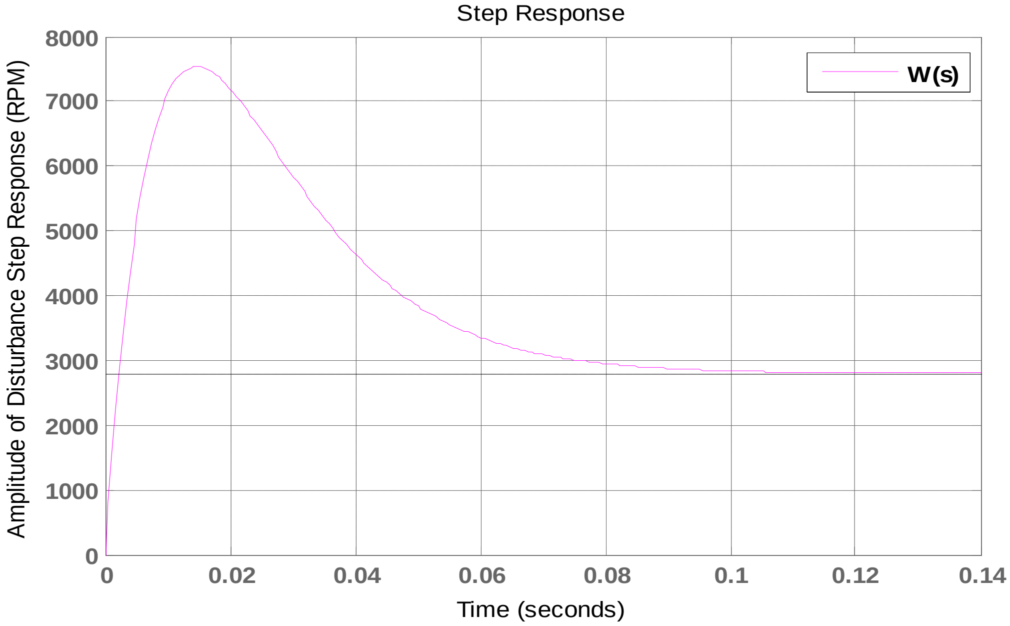

Fig. 4: Input step disturbance rejection handling

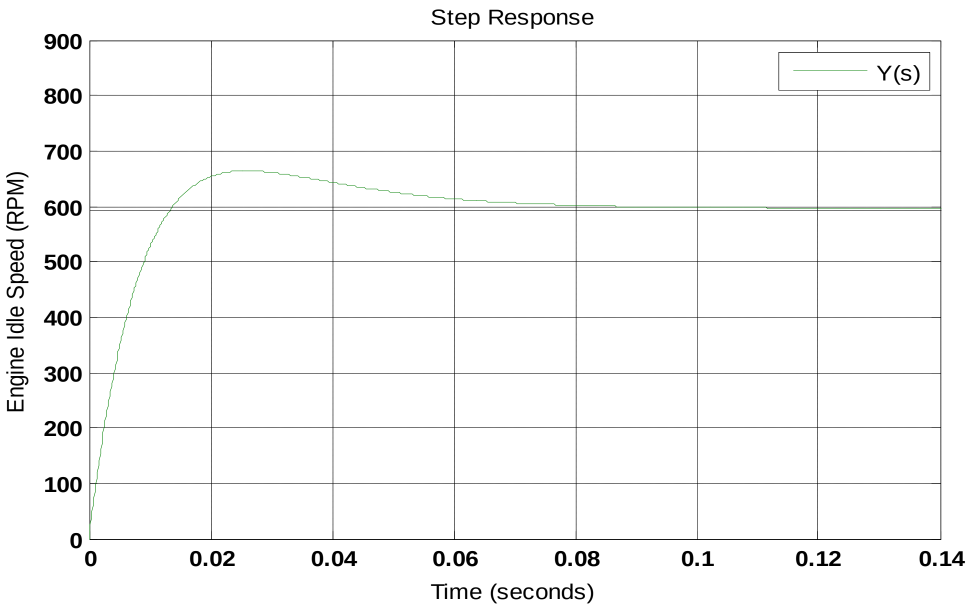

Fig. 5: Improved step response performance of the plant

Performance comparison of the designed proportional integral and derivative (PID) controller on both conditions is presented in Table 2 based on when there is no input disturbance and when input disturbance enters the engine idle speed control loop.

Table 2. *Controller Performance Comparison*

Characteristic	No Input Disturbance	Input Disturbance	Remark
Overshoot	11.4%	12.3%	Robust
Settling time	0.0761s	0.0768s	Robust
Rise time	0.00988s	0.00956s	Robust
Steady state error to a unit step input	0.023	0.015	Improved optimal tracking

4.2. DISCUSSION

Fig. 2 presents the result of the simulation performed when the designed proportional integral and derivative (PID) controller is not integrated into the engine idle speed loop. It can be seen that the step response of the process is sluggish as the output response could not track the desired speed of 600 revolutions per minute (RPM). The characteristics of the step response performance of the loop in this case are: overshoot of 0.3366%, settling time of 0.00151s, and rise time of 0.00876s. Fig.3 is the result of the simulation performed when the designed controller is integrated with the process, assuming no disturbance enters the system. It can be seen that the step response of the engine idle speed control system is largely improved. The output of the plant was able to track the desired speed of 600RPM. The improved step response performance characteristics are: overshoot of 11.4%, settling time of 0.0761s, and rise time of 0.00988s. Also from the result, it is obvious that the steady state error is improved. Fig. 4 shows the input step disturbance rejection handling. As disturbance enters the system, the magnitude of the idling speed of the engine increases as can be seen from the plot. The result is instability and this largely affects the fuel economy of the engine. The performance characteristics are: overshoot of 171%, settling time of 0.0895s, and rise time of 0.00177s. This is undesirable. Fig.5 shows the step response, Y(s), of the engine idle speed control system considered in this context. It can be seen that the designed PID controller is able to improve the system performance and disturbance handling. This gives a robust control system whose performance characteristics are: overshoot of 12.3%, settling time of 0.0768s, and rise time of 0.0956s. A look at Fig.3 and Fig.5 will give an impression of similar performance effect. While Fig.3 is the step response of system assuming no input disturbance, Fig.5 is the step response with step input disturbance entering the plant. Hence this near similarity or almost the same step response performance for both conditions proved the robustness of the designed PID controller. In both cases the controller was able to track the referenced idling speed and maintained the engine idle speed at the desired value of 600RPM.

V. CONCLUSION

The primary aim of this research is to design a robust control system for a linearized engine idle speed model with improved tracking and transient response performance. In order to achieve this aim, non linear dynamic equations of idling speed engine were obtained and then transformed to their equivalent linear forms by considering the nominal operating point of the plant. A robust tuning approach was employed in designing the controller so as to effectively improve the performance handling of the system. Simulations were performed using the robust PID controller to regulate and maintain the output response while rejecting any input disturbance. Table 2 shows that the controller maintained good robust control as the performance characteristics of the closed loop in both conditions of no input disturbance and input disturbance are nearly the same. In terms of steady state error, the controller provides improved tracking performance when disturbance entered the system. This shows that load disturbances can be effectively handled by the designed robust PID controller during idling state of a vehicle engine.

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