American Journal of Science, Engineering and Technology
2025, Vol. 10, No. 4, pp. 175-184
https://doi.org/10.11648/j.ajset.20251004.12

Research Article

Minimization of Fuel Consumption in Four Stroke Six Cylinder Engines Using Idle Speed Control

Chinonso Francis Nkeoma Ubbaonu*, Damian Obioma Dike, Raymond O. Opara, Chiedozie Francis Paulinus-Nwamuo, Ernest Ezugwu

Electrical and Electronics Engineering Department, Federal University of Technology Owerri, Owerri, Nigeria

Abstract

In a distressed economy with increasing cost of fuel, it is necessary to achieve control of automobile engine over transient behavior and meet performance objectives within the planned life of the vehicle. In this paper the objective is to design a model predictive controller (MPC) that will minimize fuel consumption in a four stroke cylinder engine using idle speed control. Engine speed is most often affected by unexpected disturbances, and the higher the engine speed at idle the higher the consumption of fuel. Non linear dynamic equations of an idle speed engine were obtained and later transformed to equivalent linear equation by linearization method. The designed MPC was integrated with a linearized equation of the plant. The MPC approach was used to forecast and control the four stroke six cylinder engine model in the presence of unexpected disturbance to achieve the desired objective. Simulations were performed in Matlab/Simulink and the results obtained showed that at different instances when disturbances entered the system, with MPC controller in the loop, the instability introduced by the disturbance was eliminated and the referenced idle speed tracked. There by maintaining the engine idle speed at 600 RPM.

Keywords

Cost Minimization, Engine Model, Idle Speed, Model Predictive Control

1. Introduction

Idle speed is the rotational speed of an engine when it is not coupled to the drive train and no force is applied to the throttle pedal. At idle condition, it is necessary to maintain engine speed at a defined set-point in the presence of varying load and perturbations. These disturbances include change in torque arising from 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. Other inputs are throttle position, vehicle speed, feed forward indicators from automatic transmission, air conditioning, power steering and battery charging system. Some environmental measurements such as engine coolant temperature and barometric pressure also play minor roles. The main control output is the amount of air supplied to the engine. Improved disturbance rejection permits lower idle speed which is directly related to fuel consumption reduction. An average passenger car spends a significant amount of its time at idle condition while driving in cities during heavy traffic, particularly in most developing countries. The persistent emphasis to adequately address idle speed problem is compounded by the desire to minimize fuel consumption in the light of the current harsh economic

*Corresponding author: Chinonso.ubbaonu@futo.edu.ng (Chinonso Francis Nkeoma Ubbaonu)

Received: 10 April 2017; **Accepted:** 14 June 2017; **Published:** 14 October 2025

conditions prevalent in the country and increasingly stringent legislative emissions regulations in developed countries.

Engine models and control design methodologies that have been utilized for the idle control include time domain, crank angle domain mean value models, cycle-accurate models capturing periodic engine speed variations due to torque actuations and multivariable control [1-5]. Different controllers have been used to solve idle speed problems, such as Fuzzy control on the basis of equality relations with idle speed control [6]. This showed that equality relations turn out to be the natural way to represent such vague environments and developed suitable interpolation methods to obtain a control function. Continuous action reinforcement learning automation On-line PID tuning has been used for engine idle-speed control [7]. A model-based control design methodology for engine idle speed control in a mild hybrid-electric vehicle based on integrated starter-alternator architecture has also been implemented [8]. An adaptive idle speed control strategy for a V2, 1000 cc four-stroke, water-cooled, port injection SI engine has been developed and a non-dimensional engine model including charging and torque dynamics was established [9].An XU7-1761cc gasoline engine data has been used within idle state to develop a combined fuzzy-PID controller for idle speed control of internal combustion using the approach of model based controller [10]. An ON/OFF control and PID control have been used as a method of decreasing the engine idle speed for the engine of a race car, the PID control was able to stabilize the idling compared with the ON/OFF control [11].An Attempt has been made to tune a controller without the knowledge of mathematical model for the system to be controlled. The Optimization algorithm of MATLAB/ Nonlinear Control Design Blockset (NCD) was used for this purpose for anOn-line tuning in an automotive engine [12]. The theory of adaptive interaction has been used successfully for the active damping of engine idle speed oscillation by applying Adaptive PID control [13].The adaptive PID control scheme tuned the PID controller parameters by using the theory of adaptive interaction.A procedure that uses the Measured Indicated Mean Effective Pressure (IMEP) at different speed at a constant fuel rate and throttle position has been used to develop successfully an Idle speed engine model using in-cylinder pressure data and an idle speed controller for a small capacity port fuel injected spark ignition engine [14]. Pole placement technique has been used to design an adaptive multiple input single output controller-based on self-tuning regulator (STR) for idle speed control [15]. A theoretical basis for further application of gas engine using Fuzzy PID Model and responsiveness of engine idle speed control to disturbances have been investigated using Sliding Mode Control [16, 17].

The aim of idle speed control is to maintain a desired speed of the engine (rpm) in the presence of disturbances. The controller should be able to operate closer to constraints, time delay allowance, inverse response, ability to change control objectives and sensor failure (predictive), and most importantly improve operating cost by reducing the consumption of fuel. In this paper the objective is to design a model predictive controller (MPC) that will minimize fuel consumption in a four stroke cylinder engine using idle speed control. Non linear dynamic equations of an idle speed engine were obtain and were later transformed to equivalent linear equation by linearization method.

2. Overview of Idle Speed Control

2.1. Concept of Engine Idle Speed

In a vehicle, an engine idle speed may be set by adjusting the screw on the throttle body. This adjustment sets the air flow through a bypass passage from the intake air side to the manifold side of the throttle body. However, if more loads are placed on the engine during idle condition, its idle speed may fall to a level where the engine stalls. High load can be caused by electric loads from head lights or cooling fans. Shifting an automatic transmission into a drive range or engaging the air conditional compressor clutch will also cause a drop in the idle speed.

The extra air needed for a cold engine can come from an auxiliary air device. The control unit reacts to this additional air by metering additional fuel. This makes more air-fuel mixture available during the warm up period. The amount of air that bypasses the throttle can be controlled by electronic control unit (ECU). It receives data on idle speed and idle conditions and uses it to provide an output to a solenoid tapper valve. The valve varies the opening of the bypass passage way and changes the idle speed to sooth. This control unit is a PID controller, Fuzzy- logic controller or hybrid controller design with model predictive control.

It was established that about 30% of fuel is consumed due to idling process in busy roads [18]. Most roads in major cities in Nigeria could be classified as busy and therefore idle conditions become a serious issue in the country. In car gasoline-fuelled vehicles emission test, idle amount of CO and hydrocarbon emissions are usually accounted for about 70% of the total emissions. The idle speed will increase fuel consumption leading to wastages. When idle speed is too low to make obvious reinforcement effect of exhaust gas dilution of mixture in cylinder, a slight change of load or resistance will cause instability or even shut down engine operation.

PID controllers, fuzzy controllers and some hybrid controllers have been the traditional methods of tackling engine idle speed problems. This paper is proposing a more robust controller which is the model predictive controller. This predictive controller improves operating cost, allows time delays, allows nonlinearities, and increases stability in idle speed of engines.

2.2. Model Predictive Control

Model predictive control (MPC) has gain wide acceptance in recent times in advanced control technology. Many varieties of MPC controllers exist both in theory and practice. They all share common feature that an explicitly expressed plant model is used to forecast and optimize future plant behaviour.

In MPC, calculations are done per sampling time. The calculations are performed with respect to current measurement and future output values predictions. There are two types of primordial computation in a MPC controller. They are: set point calculations and control calculations. The main objective of a MPC controller is such that a sequence of control moves in the manipulated variables are determined so as to track and maintain the system at its set point in an optimal trend.

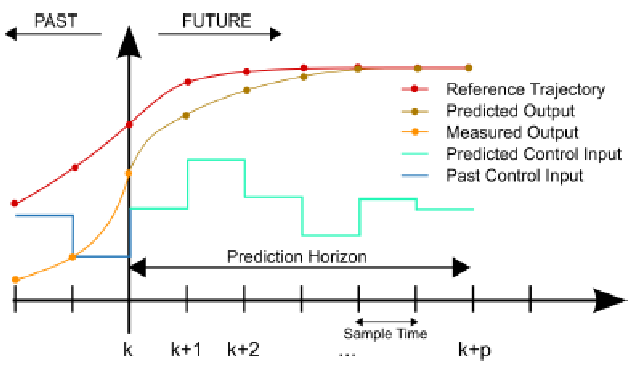

Figure 1. *Basic Concept of MPC [19].*

The basic concept behind a MPC controller is shown in Figure 1. From the figure, it can be seen that at each time step k through an objective function, an optimization problem is solved based on output predictions over a prediction horizon o time steps. In order to minimize this objective function, a selection of manipulated variables moves over a control horizon of M control moves. It should be noted that even though at each step time a group of M moves is calculated, only the first one u_k is implemented. After this step, the measurement at the next time instant y_{k+1} is obtained, followed by a correction due to model error, and then a new optimization problem solved again. These procedures are performed for each time step k .

The prediction output for an instant time k is calculated using the equation:

$$\hat{y}_k = \sum_{i=1}^{N-1} S_i \Delta u_{k-i} + S_N u_{k-N} \quad (1)$$

Where S_i is the model coefficient of the output values at each time step for $(i = 1, 2, 3, ...N)$.

3. Mathematical Model

3.1. Air Dynamics

There are two parts to be noted, the air flow past the throttle plate and the air flow and pressure characteristics of the intake manifold. These areas will be modeled in this section. In this paper the control input is the throttle angle in degrees and the output is the engine speed in revolution per minute (rpm).

3.1.1. Mass Flow Dynamics of Throttle

The rate at which air mass flow 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 Equation (2) using choked flow equation [20].

$$\dot{w}_{ath} = A_{th} \frac{P_{amb}}{\sqrt{2RT_{amb}}} \quad (2)$$

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 nonlinear 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.

3.1.2. Intake Manifold

The intake manifold functions primarily to distribute the combustion mixture to different intake ports in the cylinder head. The dynamic equation of the intake manifold pressure is modeled based on based on the isothermal conditions [20] as:

$$\frac{dp_{im}}{dt} = \frac{RT_{im}}{V_{im}} \left(\dot{w}_{ath} - \dot{w}_c \right) \quad (3)$$

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:

$$\dot{w}_{meancyl} = \eta_{vol} \frac{P_{im} v_d N_e}{4\pi} \quad (4)$$

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

$$\dot{w}_c = \frac{\dot{w}_{meancyl}}{1+\varphi\left(\frac{F}{A}\right)_s} \quad (5)$$

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

3.2. Engine Torque Generation

Generation of torque is generally 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 given by:

$$T_{eng} = g\left(N_e, \dot{w}_{meancyl}, \varphi, SA\right) \quad (6)$$

3.3. Rotational Dynamics of Engine

The equation describing the rotational dynamics of the engine crank shaft with moment of inertia, J is given according to Newton's second law [17] as:

$$J\frac{dN_e}{dt} = \frac{30}{\pi}(T_{eng} - T_{Load}) \quad (7)$$

T_{Load} 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.

3.4. Linearized Model

In this section, a linearized model of the engine idle speed dynamics which be used in this context is presented using the method in [21]. Hence Equations (2) and (3) are given as (8) and (10).

$$\dot{w}_{ath} = k_\alpha \Delta\alpha \quad (8)$$

Equation (8) can be represented as a function of $\Delta\alpha$ by:

$$\dot{w}_{ath} = g_1(\Delta\alpha) \quad (9)$$

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, equation (3) can be expressed as follows after linearization.

$$\Delta\dot{P}_{im} = k_{P_{im}}\left(\frac{\partial \dot{w}_{ath}}{\partial p} - \frac{\partial \dot{w}_c}{\partial p}\right)\Delta P_{im} + k_{P_{im}} k_\alpha \Delta\alpha - k_{P_{im}} k_{N_e} \Delta N_e \quad (10)$$

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 defined for a transfer function of the intake manifold from (10) as:

$$\tau_{P_{im}} = \frac{-1}{k_{P_{im}}\left(\frac{\partial \dot{w}_{ath}}{\partial p} - \frac{\partial \dot{w}_c}{\partial p}\right)} \quad (11)$$

Hence, equation (9) can be generally represented as a function of ΔP_{im}, ΔN_e, and $\Delta\alpha$ by:

$$\Delta\dot{P}_{im} = g_2\left(\Delta P_{im}, \Delta N_e, \Delta\alpha\right) \quad (12)$$

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

$$\Delta T_{eng} = H_p \Delta P_{im} + H_f \Delta f_d + H_\delta \Delta\delta + f_N \Delta N_e \quad (13)$$

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, equation (7), can be linearized by substituting equation (13) into equation (7). This gives:

$$\Delta\dot{N}_e + \frac{30}{J\pi}\left(\frac{2N_e}{k_j^2} - f_N\right)\Delta N_e = \frac{30}{J\pi}\left(H_p\Delta P_{im} + H_f\Delta f_d + H_\delta\Delta\delta - \Delta T_{Load}\right) \quad (14)$$

Equation (14) can be expressed as a function as follows:

$$\Delta\dot{N} = g_3\left(\Delta p_{im}, \Delta N_e, \Delta\delta, \Delta f_d, \Delta T_{Load}\right) \quad (15)$$

If $k_{RN} = \frac{30}{J\pi}$ a transfer function is taken from equation (14) which gives a time constant as:

$$\tau_{RN} = \frac{1}{k_{RN}\left(\frac{2N_e}{k_j^2 - f_N}\right)} \quad (16)$$

In order to determine the model parameters by testing in Matlab/Simulink environment, typical nominal values of the parameters for a four stroke, six-cylinder engine at N_e = 600 RPM [19] is used. The control sampling time, T, for a six-cylinder engine is equal to 120 degrees.

3.5. 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, Equations (9), (12) and (15) can be represented by the following state-space matrix by taking their partial derivative:

$$\begin{bmatrix}\dot{w}_{ath}\\ \Delta\dot{P}_{im}\\ \Delta\dot{N}_e\end{bmatrix} = \begin{bmatrix}\frac{\partial g_1}{\partial \dot{w}_{ath}} & \frac{\partial g_2}{\partial \dot{w}_{ath}} & \frac{\partial g_3}{\partial \dot{w}_{ath}}\\ \frac{\partial g_1}{\partial P_{im}} & \frac{\partial g_2}{\partial P_{im}} & \frac{\partial g_3}{\partial P_{im}}\\ \frac{\partial g_1}{\partial N_e} & \frac{\partial g_2}{\partial N_e} & \frac{\partial g_3}{\partial N_e}\end{bmatrix}\begin{bmatrix}\dot{w}_{ath}\\ \Delta P_{im}\\ \Delta N_e\end{bmatrix} + \begin{bmatrix}\frac{\partial g_1}{\partial \alpha} & \frac{\partial g_2}{\partial \alpha} & \frac{\partial g_3}{\partial \alpha}\\ \frac{\partial g_1}{\partial \delta} & \frac{\partial g_2}{\partial \delta} & \frac{\partial g_3}{\partial \delta}\\ \frac{\partial g_1}{\partial f_d} & \frac{\partial g_2}{\partial f_d} & \frac{\partial g_3}{\partial f_d}\end{bmatrix}\begin{bmatrix}\Delta\alpha\\ \Delta\delta\\ \Delta f_d\end{bmatrix} + \begin{bmatrix}\frac{\partial g_1}{\partial T_{Load}}\\ \frac{\partial g_2}{\partial T_{Load}}\\ \frac{\partial g_3}{\partial T_{Load}}\end{bmatrix}\Delta T_{Load} \quad (17)$$

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

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

$$\begin{bmatrix}\dot{w}_{ath}\\ \Delta\dot{P}_{im}\\ \Delta\dot{N}_e\end{bmatrix} = \begin{bmatrix}0 & 0 & 0\\ 0 & -1/\tau_{P_{im}} & -k_{P_{im}}k_{Ne}\\ 0 & k_{RN}H_P & -1/\tau_{RN}\end{bmatrix}\begin{bmatrix}\dot{w}_{ath}\\ \Delta P_{im}\\ \Delta N_e\end{bmatrix} + \begin{bmatrix}k_\alpha & 0 & 0\\ k_{P_{im}}k_\alpha & 0 & 0\\ 0 & k_{RN}H_\delta & k_{RN}H_f\end{bmatrix}\begin{bmatrix}\Delta\alpha\\ \Delta\delta\\ \Delta f_d\end{bmatrix} + \begin{bmatrix}0\\ 0\\ -k_{RN}\end{bmatrix}\Delta T_{Load} \quad (18)$$

Evaluating the values of the coefficients of the matrices in equation (18) using Table 1 gives:

$$\begin{bmatrix} \dot{w}_{ath} \\ \Delta\dot{P}_{im} \\ \Delta\dot{N}_e \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -4.762 & -0.062 \\ 0 & 898.464 & -0.251 \end{bmatrix} \begin{bmatrix} \dot{w}_{ath} \\ \Delta P_{im} \\ \Delta N_e \end{bmatrix} + \begin{bmatrix} 20 & 0 & 0 \\ 15.52 & 0 & 0 \\ 0 & 672 & 2460 \end{bmatrix} \begin{bmatrix} \Delta\alpha \\ \Delta\delta \\ \Delta f_d \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ -67.20 \end{bmatrix} \Delta T_{Load} \quad (19)$$

The state space equation (19) can be represented in form of a state equation as:

$$\dot{x} = Ax + Bu + bv \quad (20)$$

$$y = Cx + Du \quad (21)$$

Where

$$x = \begin{bmatrix} \dot{w}_{ath} \\ \Delta P_{im} \\ \Delta N_e \end{bmatrix} u = \begin{bmatrix} \Delta\alpha \\ \Delta\delta \\ \Delta f_d \end{bmatrix} \text{and } v = \Delta T_{Load}$$

$$A = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -4.762 & -0.062 \\ 0 & 898.464 & -0.251 \end{bmatrix}, \ B = 1, \ b = \begin{bmatrix} 0 \\ 0 \\ -67.20 \end{bmatrix}$$

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

3.6. Transfer Function Representation of the Linearized Engine Idle Speed Model

In order to carry out simulation in Matlab/Simulink environment, the state-space equations (20) and (21) are transformed into Laplace form using the formula:

$$G_p(s) = \frac{C adj(sI - A)B}{\det(sI - A)} \quad (22)$$

Substituting the Matrices A, B, and C yields the closed loop transfer function of the plant as in equation (23).

$$G_p(s) = \frac{2460s^2 + 1.171x10^4 s}{s^3 + 5.013s^2 + 56.97s} \quad (23)$$

4. Flow Chat and Implementation

4.1. Algorithm Development

In order to minimize fuel consumption by controlling the engine idle speed, a controller has been designed. The first thing we did was to identify the control variables (the air dynamics and the intake manifold), then the rotational dynamics of the engine using the engine torque generation. When we have obtained the model dynamics of the idle speed, the system configuration was established. In other to design a predictive controller, we had to understand how the predictive controller works and get a comfortable set point for the engine idle speed. A set point where the idle speed will neither be to high nor too low in the presence of unknown disturbances causing high rate of fuel consumption or engine stalling respectively.

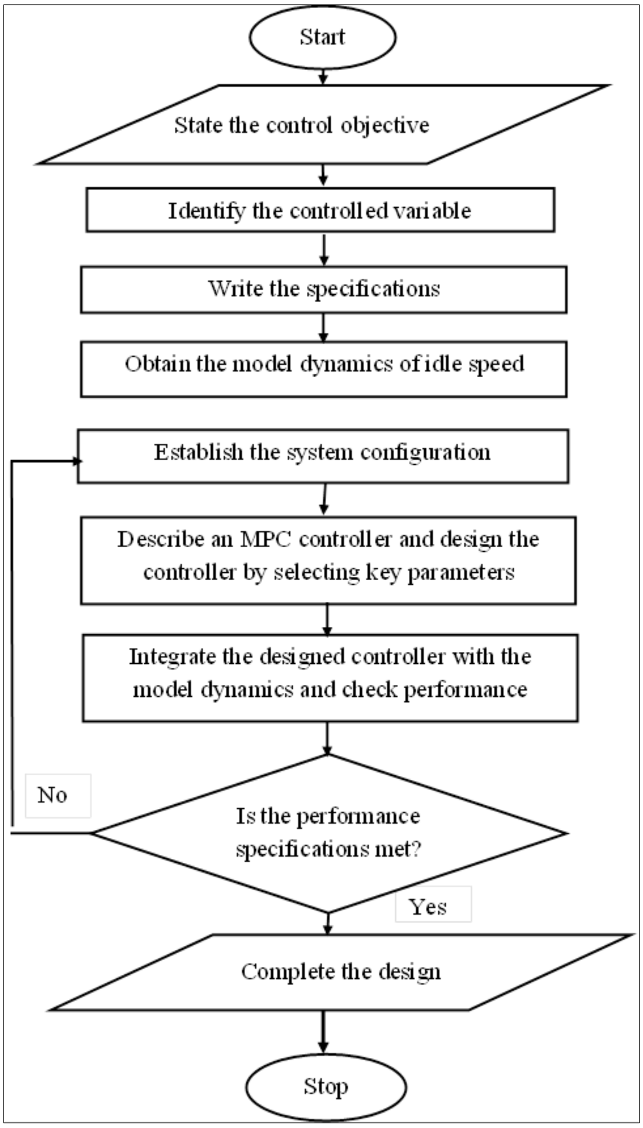

Figure 2. *Element of the control system design process.*

The control loop is a standard single input single output

(SISO) system and was implemented in Matlab/Simulink environment. The plant model dynamics were initially in a nonlinear form and were later linearized and transformed to equivalent linear forms. The linearized dynamics where combined to a single transfer function in continuous time. The transfer function was integrated with a MPC designed using Simulink block to create a closed loop control system for engine idle speed.

4.2. Simulation Data

Table 1*. Typical parameters of six-cylinder-engine model at N_e= 600 RPM [19].*

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

4.3. System Configuration and Controller Design

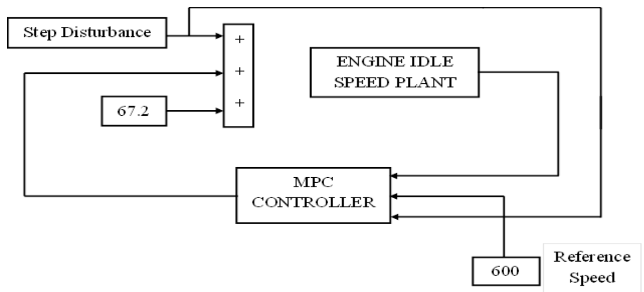

Figure 3*. System configuration for engine idle.*

Figure 3 shows the structure of the system to be considered in this context. It is a single input and single output (SISO) linearized engine idle speed close loop control system. The controller and the plant are all in the forward path.

4.4. Controller Design

The controller design in this context is performed in Matlab/Simulink environment. The procedures taken for realizing the suitable model predictive controllers (MPC) have been concisely outlined in this section. A mode predictive controller block in Simulink is used. The controller design and simulation steps are as outlined in *Parameter Selection*

In order to design a model predictive controller (MPC), certain parameters have to be chosen. The parameters are the different horizons, set-points, output weight, and constraints.

(a)Setting the model and horizons

Table 2*. Selected Horizons.*

Horizons	Selected Value
Control Intervals	0.2
Prediction Horizons	10
Control Horizon	2

Selected the horizons of the MPC designed in this context. To design the controller, the horizons must be set to values that will meet the desired set-point of the control system. For stability purpose for the closed loop system, the prediction length as well as the control horizon are essential because the controller when conditioning the control signals to perform certain changes can subject the system into unst The horizon determine how far the controller could predict the plant or process behaviour in future. Table 2 shows able mode. The relationship between the prediction horizon (H_p) and the control horizon (H_c) is such that H_p should be far greater than H_c.

(b)Set-point selection

This is the same as the nominal value of the input and output system. The set-points give the target values for both input and output signals. There are nominal values which have to be selected for the input and output. The selected set-points used in the simulation performed in this context are shown in Table 3.

Table 3*. Selected Set-Points.*

Signal	Type	Set-Point
Torque	Manipulated Variable	67.2Nm
Step-Disturbance	Measured disturbance	0
Engine Idle Speed	Measured output	600RPM

(c)Setting the constraints

The constraints tab makes it possible to specify the minimum and maximum values for the manipulated variable (input) and control variable (output). It is important to set these components when designing model predictive controller (MPC) because this approach is able to take care of the process constraints in an optimal fashion. Hence, for the evaluation of the closed loop system for the engine idle speed, different responses were tested and evaluated in this context by selecting different values for the constraints until a response which gives the desired output was realized. Table 4 shows the selected values for the constraints.

Table 4. *Selected Constraints.*

Signal	Minimum	Maximum	Max Down Rate	Max Up Rate
Load Torque	-67.2	134.4	-Inf	Inf
Engine Idle Speed	-600	600	0	0

5. Results and Discussion

5.1. Results

In this section, the results obtained when the system is simulated in the open loop and closed loop conditions in the Matlab/Simulink are presented. The simulation results in both conditions show different dynamic behaviours of the control loop for an engine idle speed system.

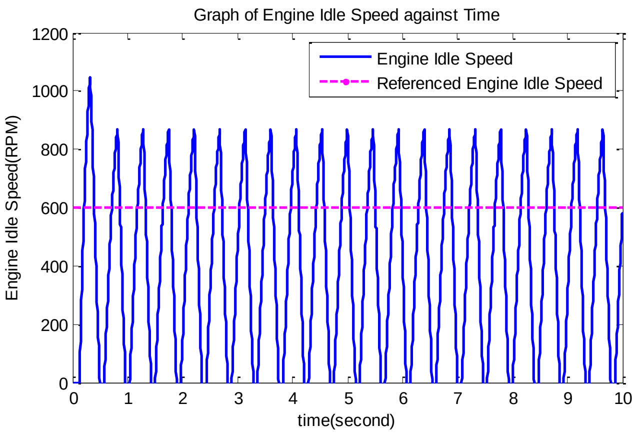

Figure 4. *Open loop simulation response of the engine.*

Figures 4 and 5 show the simulation results performed in this context in open loop for when a unit step input disturbance and a sine wave disturbance enter the system. The solid line represents the referenced speed and the dash line represents the engine idle speed (tracking speed). The open loop case is the simulation without controller in the loop while the closed loop is the simulation with controller in the loop. In both cases the plant responses remain the same as can be seen from the graphs. It can be seen that the system shows high degree of instability without a controller in the loop.

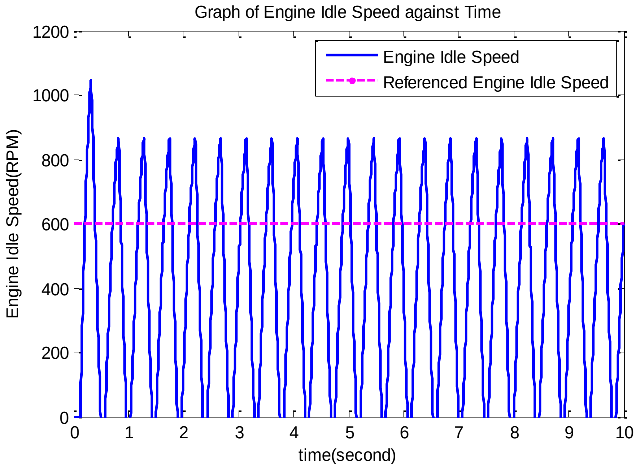

Figure 5. *Open loop simulation response of the engine idle speed system for a unit step disturbance idle speed system for a sine wave disturbance.*

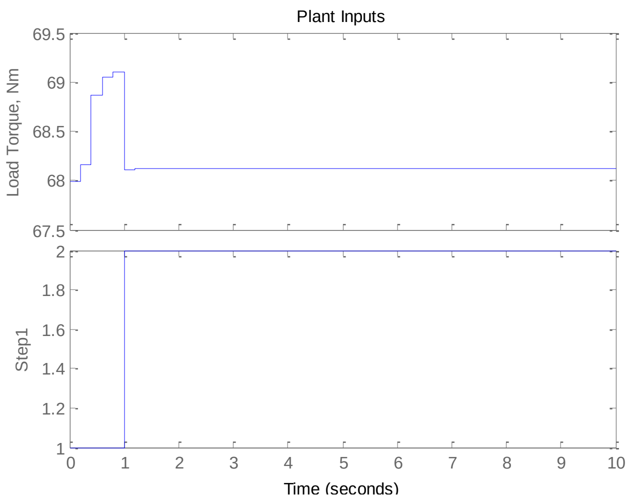

Figure 6. *Plant inputs characteristics with an MPC for a unit step disturbance for a sine wave disturbance.*

Figures 6 and 7 show the plant inputs characteristics with an MPC in the loop. In Figure 6, the manipulated variable (load torque) rises from 68 Nm to 69 Nm and settles at a load torque of about 68.1 Nm which is within the range chosen for the designed MPC, that is -67.2 Nm (minimum) and 134.4 Nm (maximum). In Figure 7, the manipulated variable (load torque) rises from a value of 67.2 Nm to 70 Nm and settles at a load torque of about 68.1 Nm.

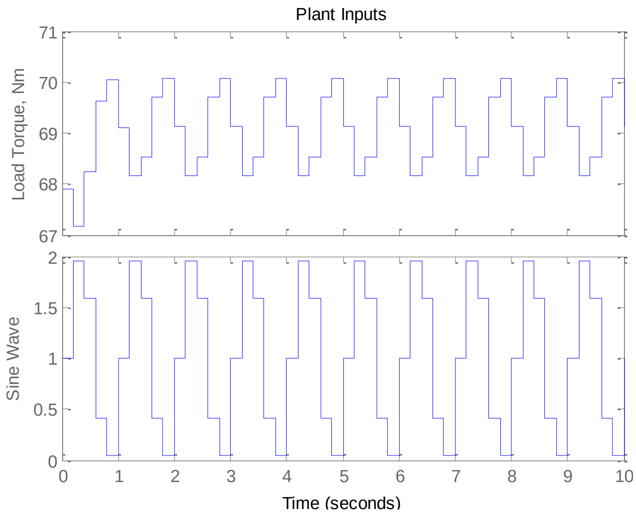

Figure 7. *Plant inputs characteristics with an MPC.*

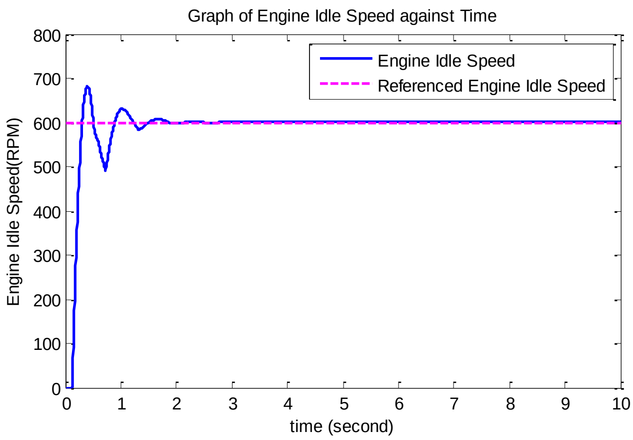

Figure 8. *Closed loop simulation response of engine idle.*

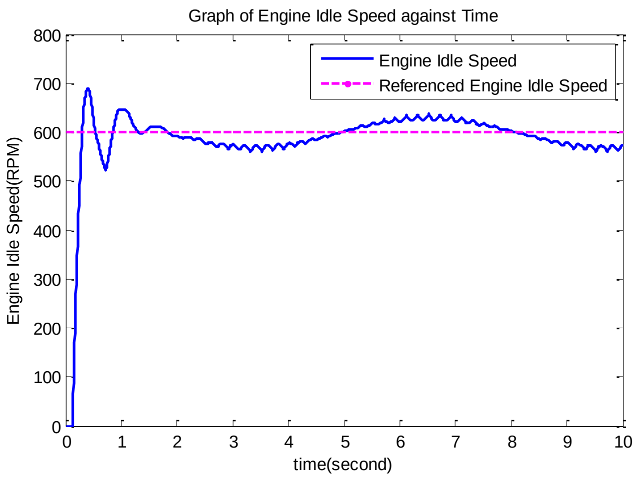

Figure 9. *Closed loop simulation response of engine idle speed system for a unit step disturbance speed system for a sine wave disturbance.*

Figures 8 and 9 show output responses of the engine idle speed control loop when a step input disturbance and sine wave disturbanceentered the system. In both cases the controller is able to track the referenced speed. The solid line represents the referenced speed and the dash line represents the engine idle speed (tracking speed).

5.2. Discussion

Figures 4 and 5 are the open loop simulations of the engine idle speed. It can be seen that the uncompensated system shows continuous cycling. The response of the engine idle speed to a referenced speed of magnitude 600 rpm shows a peak value of above 1000 rpm and continues cycle. Hence this indicates high instability in the open loop condition and leading to increase in fuel consumption by the engine in the idle state. And this requires a controller in the loop to take care of the cycling and maintain stability in the control loop with improved fuel consumption rate.

Figures 8 and 7 are the closed loop simulations with a Model Predictive Controller (MPC) in the loop. It can be seen that the response initially peaks to a value of about 680 rpm and then cycles for 0.04 minute (2.5 seconds) in both cases. In Figure 8, at point 1.0 seconds, a step disturbance was introduced and the response peaks downward and then rises rapidly before tracking the referenced speed at 2.5 seconds. This shows that the designed controller has been able to eliminate the cycling in the open loop and has been able to stabilize the system with a settling time of 2.5 seconds. This shows an optimal and fast response tracking. In Figure 9, at point 1.0 seconds, a sine wave disturbance was introduced and the response peaks downward and then rises rapidly before tracking the referenced speed but with slight variation from the referenced input as the system settles and stabilizes. In both cases the controller is able to eliminate the high instability in the open loop. This leads to improveengine idle speed with the fuel rate consumption greatly minimized. Hence, operating cost of the engine in terms of fuel consumption has been improved to a large extent.

It is obvious from the simulation graphs of Figures 8 and 9 that the designed controller greatly improved the performance of engine idle speed system when compared to the loop without a controller. Hence, it can be seen that the engine idle speed has been considerably minimized and maintained at a desired speed of 600 rpm by the designed MPC controller.

6. Conclusion

A model predictive control system for minimizing idle speed in four stroke six cylinder engine has been presented in this context. Predictive control of engine's throttle valve has many advantages such as reduction in emissions, improvement in fuel efficiency and power delivery. In this paper, a four-stroke six-cylinder engine model is used; model predictive control approach is used in order to design a robust controller for engine at idle speed. The main objective of this paper is to design a controller that will minimize fuel consumption in idle engine speed. The controller achieved the objective and hence improved fuel economy. The basic design steps can be summarized as linearization of Throttle body dynamics, design of suitable robust controller for stability and tracking problems. Lastly, simulation of control indicates that applying the proposed approach successfully overcomes obstacles for control of engine at idle speed. The choice of a sine wave disturbance after testing with a step input disturbance that could arise from an ignition, was due to the fact that one of the disturbance that affect the engine in idle state is an audio sound from car stereo which is sinusoidal in nature.

Conflicts of Interest

The authors have declared that no competing interest exists.

References

[1] D. Hrovat and J Sun. Models and Control Methodologies for IC Engine Idle Speed control design. Control Engineering practice, vol 5,1997, pp. 1093 – 1100.

[2] K Butts, N. sivashankar, and J. Sun. Application of LI Optimal Control to the Engine Idle Speed Control Problem. IEEE Transactions on Control System Technology, Vol. 7, 1999, pp 258-270.

[3] S. Yurkovich, and M. Simpson. Crank Angle Domain Modelling and Control for Idle Speed. SAE Technical paper970027, 1997, https://doi.org/10.427/970027

[4] D Shim, J park, P. P. Khargonekar and W. B Ribbens. Reducingautomative engine speed fluctuation at idle. IEEETrans. On Control Systems Technonlogy, 1996, vol. 4, no. 4, pp. 404-410.

[5] H. Onder and H. P. Geering. Model-based multivariable speed and air-to-fuel ratio control of a Spark Ignition engine. Technical Report, 930859, SAE, 1993.

[6] F. Klawonn, J. Gebhardt and R. Kruse. Fuzzy control on the basis of equality relations with an example from idle speed control. IEE Trans. Fuzzy System., vol 3, 1995, https://doi.org/10.1109/91.413237 pp. 336 – 350.

[7] M. N. Howell, and M. C. Best. Online PID tuning for engine idle-speed control using continuous action reinforcement learning automata. Control Engineering Practice, 8, no 2, 2002, pp. 147 – 154.

[8] A. Brahma and G. SRizzoni, and Qi Ma. Idle speed control for automative engines with an integrated starteralternator. World congress, vol 15 part 1, 2002, https://doi.org/10.318/20020721-6-ES-1901.01509, pp 1507.

[9] J. Yin, Y. Syu, B. Chen, Y. Wu, Y. Liang and H. Tsai. Application of adaptive idle speed control on v2 engines. SAE International Journal of Engines (9)1, 2016, 458-465.

[10] S. Banarezaie and M. Shalchian. Development of a model based combined fuzzy-pid controller for idle speedcontrol of internal combustion engine. Journal of Engineers, 38(38), 2015, 3-16.

[11] T. Fukui and H. Enomoto. Decreasing of the engine idle speed of a small gasoline engine with feedbackcontrol. SAE technical paper 2097171. 2009.

[12] Ali MajeedMahmood. Design of On-Line Tuned idle speed controller for an automative engine by using NCD. IJCCCE vol 12, No. 2, 2012.

[13] B. Badreddine, A. Zaremba, J. Sun., and F. Lin, "Active Damping of Engine Idle Speed Oscillation by Applying Adaptive Pid Control," SAE Technical Paper 2001-01-0261, 2001, https://doi.org/10.4271/2001-01-0261

[14] P. V. Manvannan, M. Singaperumal, A. Ramesh. Development of an idle speed engine model using in-cylinderpressure data and an idle speed controller for a small capacity port fuel injected SI engine, International Journal ofAutomotive technology 12(1), 2011, https://doi.org/10.1007/s12239-011-0002-3 pp 11 – 20.

[15] F. Hsieh, B. Chen, and Y. Wu, "Adaptive Idle Speed Control for Spark-Ignition Engines," SAE Technical Paper2007-01-1197, 2007, https://doi.org/10.4271/2007-01-1197

[16] GeXiaocheng, XuZhongming, Li Jingboand Zou Bowen. Fuzzy PID Model -Based Study on Idle Control of GasEngine, Open Cybernetics &Systemics Journal, 8, 2014, 660-666.

[17] Y. Zhang, N. Kurihara, H. Yamaguchi. Responsiveness Improvement of Idling Speed Control for.

[18] Automotive Using SMC, JSEA, Vol. 2 No. 5, 2009, https://doi.org/10.4236/jsea.2009.25040 PP. 309-315.

[19] Y. Fengfeng, S. Jiamin and W. Wenwen. Engine Idle speed control research status and developmenttrend. 4th International conference on Mechatronics, Materials, Chemistry and Computer Engineering. 2015.

[20] J. A. Cook; B. K. Powell, Modeling of an internal combustion engine for control analysis, 1988, Page(s): 20–26.

[21] AbhishekChaturvedi. Idle Speed Control of an Engine Model Using PID Control System, Research gate, 2015, https://doi.org/10.13140/RG.2.1.5089.9367 1 – 20.

[22] A. G. Ulsoy, H. Peng andM. Çakmakci. Automotive Control Systems, Cambridge Univ. Press, NY. 2012, Page 54 – 77, page 126.