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PID-LIKE FLC FOR FOUR CYLINDERS MEAN VALUE GASOLINE ENGINE MODEL IN IDLE MODE

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ABSTRACT

Automatic control of automotive engines provides benefits in the engines performance like emission reduction and fuel economy. Due to high dropping in the rotational speed in the presence of load torque and disturbance, which may lead to engine stalling, an electronic control unit (ECU) has to keep the engine speed at the reference idling speed.

In this paper, the four strokes, four cylinders, gasoline, port fuel injection engine is studied and a Mean Value Engine Model (MVEM) is used to simulate the nonlinear model of the engine. The problem of preserving the engine speed with minimum settling time, minimum undershoot and minimum overshoot in the presence of load torque or disturbance at idle speed mode is studied. PID-like Fuzzy Logic Control (PID-FLC) with minimum structure is designed to solve this problem. Comparisons among conventional PID controller and fuzzy controller is made. Simulation results of this PID fuzzy controller show good improvement over the conventional PID controller the idle speed response.

The simulation results with PIDFC show that the dropping in the engine speed is reduced around (33%) compared with PID controller. The peak overshoot is reduced around (19%) compared with PID controller for the same value of the load torque. The settling time is reduced by (50 %) compared with the PID controller.

KEYWORDS: Spark ignition engine, four cylinders gasoline engine, and Mean value engine model

1. INTRODUCTION

Most automotive machines are supplied with gasoline Port Fuel Injection (PFI) engine. Usually this engine is Spark Ignited (SI), four strokes and the chemical energy for the Otto cycle converted to mechanical work. These engines run in different operating conditions during their life cycle such as cold start, idling, cruising…etc. These operating conditions are different from each other in objectives and performance. At idling mode, that the speed of engine between (750-1400) Revolution per Minute (RPM) (Panse, 2005). Idle speed control is one of the general and difficult problems in the engines of automotive, because it is complex, nonlinear and time delays. So, a controller that reparation about error in speed in the existence the torque of load. The controller should supply accurate and fast to reach the required speed with reduced emissions, amended fuel economy and secured the stability combustion. And in the final decades have been suggested methodology of control to idle speed problem. Different papers that discuss idle speed control techniques, with the corresponding control oriented model used were published. This includes control techniques based on classical methods such as PID controller, as well as different modern control alternatives such as LQ and fuzzy logic techniques. (Feldkamp and Puskorius, 1993) proposed in 1993 simulation based training the fuzzy controller using neural network. They found that the operation of training progress more slowly than similarly carried out the simpler plants of training. (Dotoli, 1997) proposed in 1997 a MVEM for Austin metro engine. Two PID controllers have been developed to regulate the idle speed, one for controlling the bypass valve and the other for controlling the timing of the spark. Panse (2005) proposed in 2005 a dynamic control oriented Mean value engine model of a PFI engine. The idle mode is progressing that uses the throttle to detect the speed based PID controller. Santis et.al (2005) proposed in 2005 the idle speed control of SI engine as the problem of calculating the maximal safe for the modeling of system. Simulation results showed increasing efficiency of the proposed approach. Gibson et.al. (2006) proposed in 2007 the analyses of lead restitution, feedforward and disturbance supervisor the techniques design for idle speed controller (ISC). Simulation results show that a 30 percent reducing in the maximum drop of the engine speed was obtained compared with an ISC with no lead disturbance supervisor compensation. Amir and Shamekhi (2015) presented in 2010 to presage steady-state properties of a variable ratio of compressing and diesel engine based applied two neural networks. Yoon et al., (2012) presented in 2012 applied a NARX neural network to presage the pressure cylinder. Their inaccuracy in neglect the Bias difference problem and over-determining the network effected worse results. Uzun presented in 2015 applied a neural network to complete the results obtained from experimental data of engine.

In this paper, Mean Value Models is intended to combine and the focus is solely on using intelligent control techniques in idle speed regulation. Furthermore, a classical linear PID controller was developed to show advantages of the intelligent controller. The control methods discussed above used the engine speed information for feedback in a feedback control system, and the throttle valve as an actuator. This paper will build an engine model which is capable of closely indicating dynamic engine performance and pollutantion emissions.

2. MEAN VALUE ENGINE MODLE OF SI ENGINE

The model is nonlinear model of SI engine is studied and used as a preliminary step toward engine controller development .It described in this section comprises states for air, fuel and rotational speed of the engine.

2.1. Mean Value Engine Model

Modeling of the engine can greatly simplify the design of the control and diagnoses systems, as they can be used to simulate the engine instead of performing extensive tests in a car. There are large type of models in SI engine. The model used to characterize these phenomena in specifics is expensive, complex and is not much useful for design of control system of the real time, while the model has been used for the purpose of control is less detailed in comparison. Such models don’t necessarily reflect all phenomena in an engine (Panse, 2005). For control purposes it is often eligible to have a model with a little parameters and low order to obtain easy tuning to a given application so, the PID-like fuzzy logic controller will used to control this engine. In this paper, the automotive engine is modeled using a class of models called Mean Value Engine Models, MVEMs. These models can successfully be used during the development of control system.

2.2. Engine Modeling

It has air dynamic, fuel dynamic and rotational dynamic as well as includes delay ingrained in the engine. The basic configuration model of the engine has three parts in SI engine. These parts are:

1- Intake air dynamics.

2- Fuel dynamics.

3-Crankshaft dynamics.

Fig. 1 shows the state diagram of the engine model

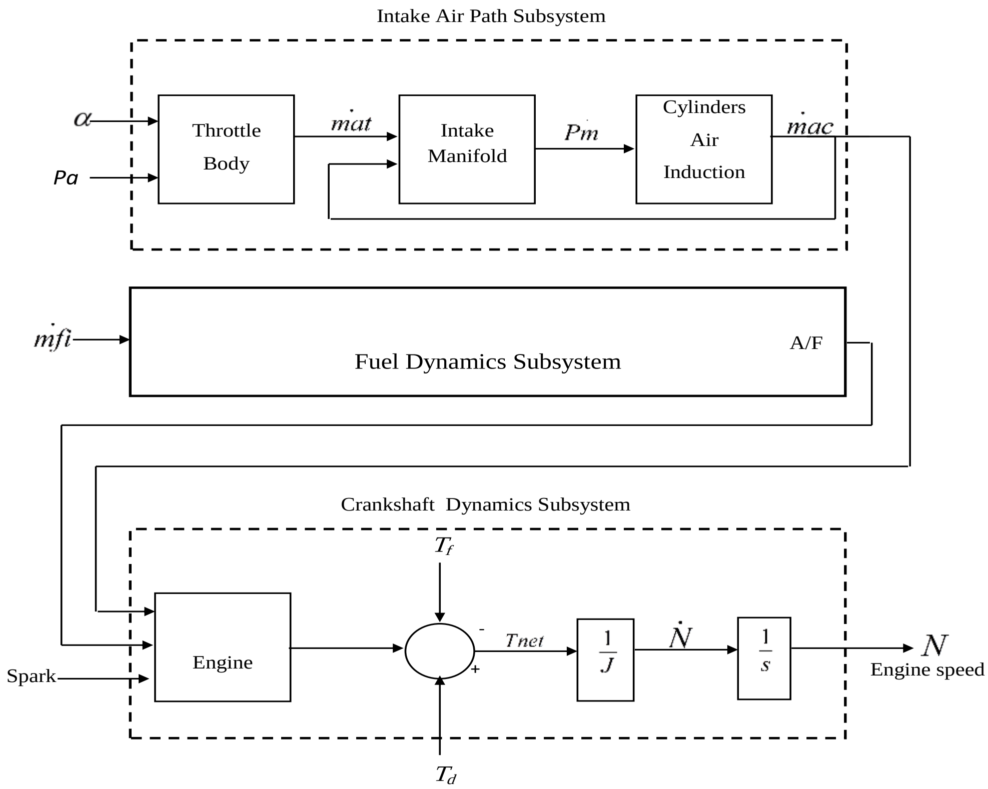


Fig. 1. Block diagram of the engine model.

1- Intake air path dynamics:

It is had three parts that have to be modeled. These are:

A-Throttle body:

The mass flow of air in cylinders so, the output of engine is controlled based throttle valve. The air mass flow through the throttle valve can be modeled as a gas flow through an orifice. This is calculated as:

$$\dot{m}at(t) = Cd\,\frac{Pa(t)}{\sqrt{RTa}}\,\phi(Rp(t))A(t) \qquad (1)$$

Where: Cd: the discharge coefficient; Pa: ambient pressure (N/m^2).

R: ideal gas constant (J/kg.K); Ta: ambient temperature (K)

$\phi(Rp)$ is calculated as follows :

$$\phi(Rp(t)) = \sqrt{\gamma\left[\frac{2}{\gamma+1}\right]^{\frac{\gamma+1}{\gamma-1}}} \qquad \text{For } Pm\ (t) < Pc\ (t) \qquad (2)$$

or

$$\phi(Rp(t)) = (Rp(t))^{\frac{1}{\gamma}} \sqrt{\frac{2\gamma}{\gamma-1}\left[1-(Rp(t))^{\frac{\gamma-1}{\gamma}}\right]}$$ For *Pm (t) > Pc (t)* (3)

where Pc is the critical pressure and calculated as follows:

$$Pc(t) = \left[\frac{2}{\gamma+1}\right]^{\frac{\gamma}{\gamma-1}} \cdot Pa(t) \tag{4}$$

and the area *A*(t) equal to the cross sectional so:

$$A(t) = \frac{\pi}{4} Dth^2 (1 - \frac{\cos\alpha}{\cos\alpha_{leak}}) + A_{leak} \tag{5}$$

where A_{leak} is the flow area when the throttle angle equals α_{leak}

B- Intake manifold dynamics:

It can be expressed as below:

$$\dot{Pm}(t) = \frac{RTm}{Vm} \dot{ma}(t) \tag{6}$$

where $\dot{ma}(t) = \dot{mat}(t) - \dot{mac}(t)$ (7)

Tm : manifold air temperature (K); Vm : manifold volume (m^3)

C- Air induced in cylinders

The mass flow rate to exhaust gas and identical mixture of air is calculated using this concept where its mathematical expression is as follows (Boveie et al., 1994) and (Kruse et. al., 1994):

$$\dot{mac}(t) = \frac{Pm(t)}{4RTm} \cdot \frac{N(t)}{\pi} z \cdot Vd \cdot \eta_V \tag{8}$$

Where:

$$\eta_V = \eta_N(N(t)) \cdot \eta_{Pm}(Pm(t)) \tag{9}$$

$$\eta_N(N(t)) = \gamma_\circ + \gamma_1 N(t) + \gamma_2 (N(t)) \tag{10}$$

$$\eta_{Pm} = \frac{Vc+Vd}{Vd} - \frac{Vc}{Vd} \cdot \left(\frac{Pex}{Pm(t)}\right)^{\frac{1}{\gamma}} \tag{11}$$

where

$\gamma_\circ, \gamma_1, \gamma_2$: empirical constants related to specific engine.

Vc: clearance volume of the cylinder (L) *Pex* : the exhaust gas pressure (N/m^2)

2- Fuel Path Dynamics

It is important to calculate the air-fuel ratio. In general, it is difficult to accurate of model the fuel dynamic, because the parameters of the model depend on the fuel properties and temperature of engine during its operation. The fuel is injected fuel by injector when the valve of intake is close. This fuel film on the intake port evaporates continuously, so, the net flow rate of fuel entering to cylinders consists of the injected fuel and the fuel evaporated from the intake port and no impinging part of walls Fuel dynamics can be calculated as follow (Pfeiffer ,2009); (Stefanopoulou, 1996); (Yoon , Park , and Sunwoo , 2012):

$$\dot{m}ff(t) = x \cdot \dot{m}fi(t) - \frac{1}{\tau_f} mff(t) \tag{12}$$

$$\dot{m}fv(t) = (1-x)\dot{m}fi(t) \tag{13}$$

$$\dot{m}f(t) = \dot{m}fv(t) + \frac{1}{\tau_f} mff(t) \tag{14}$$

Where:

$\dot{m}fi$: Mass flow rate of the injected fuel (kg/sec)

$\dot{m}ff$: Mass flow rate of the fuel film (kg/sec)

$\dot{m}fv$: Mass flow rate of the fuel vapor (kg/sec)

$\dot{m}f$: mass flow rate of fuel that enters the cylinder (kg/sec)

$$x = 0.3 + \frac{0.7}{90}\alpha(t) \tag{15}$$

$$\tau_f = 0.05 + \frac{2.25}{N(t)} \tag{16}$$

The air fuel ratio is assumed to be kept at the stoichiometric value (14.7) (Deur et al., 2010).

3-Crankshaft Dynamics

The interesting of crankshaft is expressed revolution speed, which depended on the production torque by cylinders. and included three parts as following:

A) The Model of Production Torque

The torque generated through cylinders during the strokes of expansion, according to a nonlinear behavior of the air loaded mass to the cylinders, air fuel ratio taken from the stroke of intake earlier (Stefanopoulou, 1996). The model described here is a mean value model. It uses the notation of mean effective pressure (mep), since it depends on the same variables that effect on the torque of engine, as well as, the torque function is just an expansion of the break mean effective pressure (bmep) function. Also, the model derived includes empirical relationships of independent and measurable engine parameters (Panse, 2005) .The theoretical thermal power progressing by the fuel flow combustion is converted into mechanical power through the mechanical efficiency. This will be used to drive the loads and overcome pumping and friction losses (Dotoli, 1997). This can be expressed as follows:

$$\eta_{i-t} = \frac{imep}{mep_{fe}} \qquad (16)$$

Where $bmep = imep - mep_{mec\_losses}$ (17)

By substituting equation (16) into equation (17), equation (18) is generated. Which is another expression for the bmep as shown below:

$$bmep = mep_{fe} \cdot \eta_{i-t} - mep_{mec-losses} \qquad (18)$$

(mepfe) is expressed in the following equations:

$$mep_{fe} = \frac{M_f \cdot \mathrm{Q_{HV}}}{Vd} \qquad (19)$$

The mass of fuel (M_f) is calculated as following:

$$M_f = \frac{mac\,(\text{for one cycle})}{A/F} \qquad (21)$$

$$M_f = \frac{m\dot{a}c(t)}{A/F(t)} \cdot \frac{4\pi}{N(t)} \qquad (22)$$

From equations (19) and (22) the (mep_{fe}) is produced.

$$mep_{fe}(t) = \frac{m\dot{a}c(t) \cdot \mathrm{Q_{HV}}}{A/F(t) \cdot Vd} \cdot \frac{4\pi}{N(t)} \qquad (23)$$

The indication of the thermal efficiency (η_{i-t}) is [1]

$$\eta_{i-t} = \varphi(N(t), A/F(t), \delta(t)) \qquad (24)$$

and $\varphi(N(t)) = \eta_\circ + \eta_1 \cdot N(t)$ (25)

where ($\eta_\circ$ and η_1) are empirical coefficient parameters, related to engine specification

lastly, the mechanical losses of engine in equation (18) above are described by following equations:

$$mep_{mec-losses} = \psi(N(t), Pm(t)) \quad (26)$$

where $$\psi(N(t), Pm(t)) = \beta_\circ + \beta_1 (N(t))^2 \cdot \frac{4\pi}{Vd} + (Pa - Pm(t)) \quad (27)$$

The variables ($\beta_\circ$ and β_1) are constants of empirical, related to the engine specification.

Yet, by sub (23), (24) and (26) in (18) the bmep is (Panse, 2005) :

$$bmep(t) = \frac{mac(Pm(t), N(t)) \cdot \varphi(N(t))}{(A/F) \cdot N(t)} \cdot \frac{Q_{HV} \cdot 4\pi}{Vd} - \psi(N(t), Pm(t)) \quad (28)$$

and

$$bmep = \frac{Tb \cdot 4\pi}{Vd} \quad (29)$$

By substituting equation (29) into (28), the torque of break is:

$$Tb = \frac{mac(N(t), (Pm(t)) \cdot \varphi(N(t))}{A/F(t) \cdot N(t)} \cdot Q_{HV} - \psi(N(t), Pm(t)) \frac{Vd}{4\pi} \quad (30)$$

B) Process Delays Model

The production torque process is discrete according to the speed of engine. This model produced is continuous. So, to add the discrete behavior of the engine to this model, two delays are included in the model (Polonskii, 2000) and (Panse, 2005) :

1- Intake to delay the production of torque (T_{it}): it is a delay between taking a stroke to expand the stroke time.

$$T_{it} = \frac{2\pi}{N(t)} \quad (31)$$

2-Spark to delay the production of torque (T_{st}): it is a delay of stirring to produce a torque time.

$$T_{st} = \frac{\pi}{N(t)} \quad (32)$$

So, from (31) and (32) in equation (30) gives:

$$Tb = \frac{mac(t - T_{it}) \cdot \varphi(N(t))}{A/F(t) \cdot N(t)} \cdot Q_{HV} - \psi(N(t), Pm(t)) \frac{Vd}{4\pi} \quad (33)$$

C) Rotational Dynamics Model

It is get from the application of Newton's second law of motion from rotation (Panse, 2005):

$$Tnet = J\,\dot{N} \quad (3$$

$Tnet$: the torque used for vehicle acceleration (N.m).

J: the total inertia of the engine (Kg.m^2).

$Tnet$:is obtained from the variance between the production torque and the summation of the torque load. This includes the mechanical friction torque calculated in the static model for the produced torque, and the load torque (Td) which acts as disturbance on the output side as a result of the load the engine is exposed to. The load torque (Td) value usually varies around (30N.m) for low and medium range of engine speeds (Polonskii, 2000)

2.3. Simulation of the Mean Value Engine Model

The model is simulated using Matlab/Simulink software environment. Equations of every subsystem are implemented as sub blocks in Simulink and the sub blocks are then connected to obtain a complete simulation model. This simplifies changing, correcting or developing the model.

2.3.1. Simulation of Intake Air Path Dynamics

The intake air path dynamics block simulates equations from (1) through (11). As mentioned before, the intake air path dynamics consists of three subsystems: throttle body subsystem, intake manifold subsystem and cylinders air induction subsystem.

1) Simulation of Throttle Body Dynamics

The throttle body block simulates equations (1) through (5). This includes simulation of the pressure across the throttle valve and simulation of the area available for the flow.

2) Simulation of Cylinders Air Induction Dynamics Cylinders air induction block simulates equations (8) through (11). This includes simulation of volumetric efficiency equations and simulation of the Cylinders air induction main equation.

3) Simulation of Intake Manifold Dynamics

This includes simulation of equation (6-7) which represents the mass conservation equation.

2.3.2. Simulation of Fuel Path Dynamics:

The fuel path dynamics block simulates equations from (12) through (16). This includes simulation of the non-inhibitor of fuel and fuel evaporated from the intake port walls.

2.3.3. Simulation of Crankshaft Dynamics

he crankshaft dynamics block simulates equations from (17) through (34) this includes mep_{fe} which is the bmep produced with full efficient engine, indicting thermal efficiency η_i, and mechanical losses .

2.4. Simulation Results

In this paper, the simulation is done for a Mitsubishi motors. Engine model 4G64 .The 4G64 engine is a single overhead camshaft, 4 cylinders, 4 valves per cylinder gasoline powered, four stroke, PFI, SI engine prototype with idle speed equals to 955 RPM**.** All the physical parameters used in the model are listed in Table 1**.** On the other hand, engine geometry Parameters needed in the model are listed in Table 2**.** The coefficients of the imperical data used in the model also listed in Table 3**.**

Table 1. Physical constants.

Parameter	Value
R	287 J/Kg.K
Ta	298 K
Pa	10^5 N/m^2
γ	1.35
Tm	340 K
Cd	0.7
Pex	$1.08*10^5$ N/m^2

Table 2. Engine parameters.

Parameter	Value
α_0	0.1379 rad
Dth	$86.5*10^{-3}$ m
A_{leak}	$5.3*10^{-6}$ m^2
Vm	$2.351*10^{-3}$ m^3
Vd	$2.77*10^{-3}$ m^3
Vc	$0.277*10^{-3}$ m^3
J	0.3 Kg.m^2

Table 3. Equations coefficients.

Parameter	Value
$\gamma_\circ$	0.45
γ_1	$3.42*10^{-3}$ sec
γ_2	$-7.7*10^{-6}$ sec^2
$\eta_\circ$	0.16 J/kg
η_1	$2.21*10^{-3}$
$\beta_\circ$	15.6 N.m
β_1	$0.175*10^{-3}$

The open loop dynamics for the nonlinear engine model has been obtained for step change in the angle of throttle without and with applying variable value of the load torque (Td) as explained below:

1) Step Change in the Angle of Throttle with no Load Torque:

A step change in the angle of throttle is applied to the model within a range of 9.1558 (degree) to 9.1959 (degree) at 10 seconds (Panse, 2005). The result was a step change in the engine rotational speed as shown in the open loop dynamics for the engine is in Fig. 2.

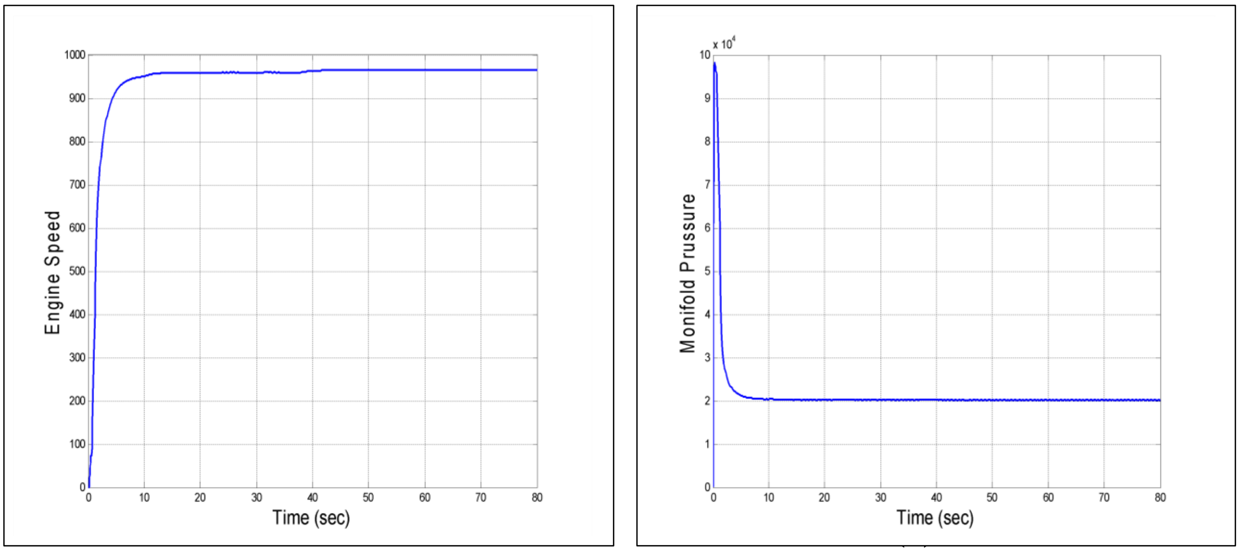


Fig. 2. Open loop dynamics of the MVEM with no load applied.

(a): speed response and (b): intake manifold pressure response

2) Step Change in the Angle of Throttle with Load Torque:

The load of engine during idle mode is of the nature variable and this could be the source of sudden variance in the speed of engine. So,the nonlinear model is loaded with variable value of load torque. The loads were applied with the same step change in the throttle angle that was used before. As shown in the open loop responses in Fig. 3, the engine rotational speed was dropped at the moment of applying the load torque. This action accompanied with increasing in the intake manifold pressure. When the load torque is removed, the engine rotational speed returns to its original value.

3. IDLE SPEED CONTROLLER DESIGN

The design of intelligent controller for automotive engine is considered. The control of idle speed is produced as a control problem, where procrastination of engine should be prevented despite load varying and acting. So, that the controller should get the idle rotational speed with better performance. The most widely used control structure in the control industry is the PID controller. In spite of PID controller advantages, the use of such controller may not result in the required performance. This is due to large nonlinearities and inneglible disturbance exhibited by some plants (Stefanopoulou, 1996).

One successful control technique is fuzzy control which originates from the human experiences. The objective of the Fuzzy Logic Control (FLC) systems is to control complex process by experience of human being.

The idle speed controller designed in paper is of PID-like FLC to recompense for the descend in the engine rotation speed, it uses the throttle angle only.

3.1. Fuzzy Logic Controller (FLC) Design

It is a PID like Fuzzy Controller PIDFC is developed. This controller uses the discrete from with minimum structure form of the conventional PID controller equation as follows:

$$u = Kpe(k) + KdT\Delta e(k) + \frac{Ki}{T}\sum e(k) \qquad (35)$$

where e(k) is the error signal and the index (k) represents the present sampling instant and the sampling time uses 0.1s. It is clear from the equation that the controller has three inputs. If seven fuzzy sets are used for each input, then a (7*7*7=343) rules will be needed for the controller. Also, each rule will have three conditions in its antecedent part, which is very difficult to design such controller and needs too much work (Polonskii, 2000) .To avoid such problems, the PIDFC is constructed as a parallel structure of PD like Fuzzy Controller PDFC and PI like Fuzzy Controller PIFC. As a result, the equation of the PIDFC will be:

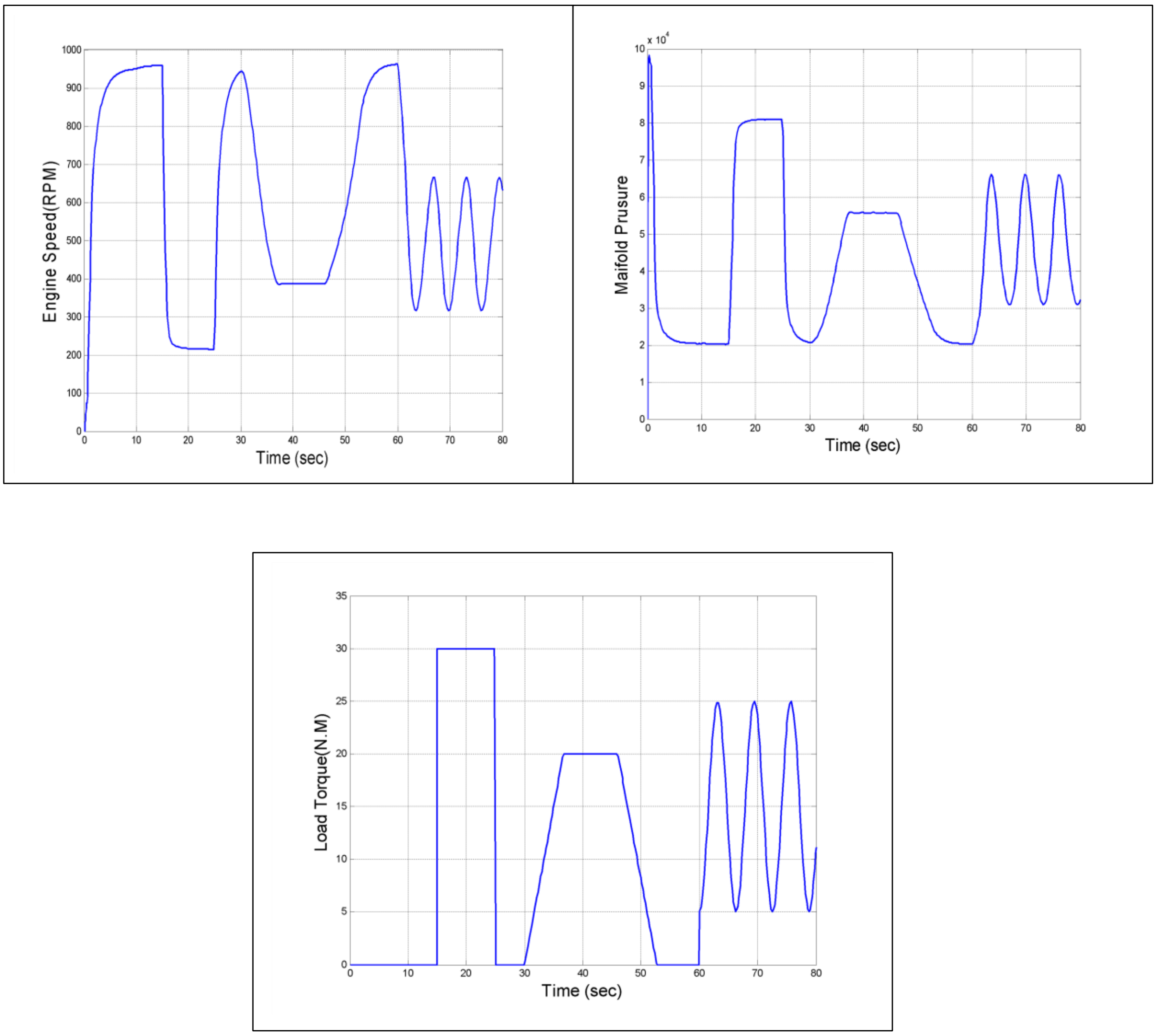
PIDFC=(PDFC+PIFC)Ko (36)

Where

(37)

$$PDFC = Kpe(k) + Kd\Delta e(k)$$

And $PIFC = \sum PDFC$ (38)



(c)

Fig. 3. Open loop dynamics of the MVEM with applying variable value of load torque

(a): Open loop speed response, (b): Open loop intake manifold pressure response

(c): Variable value of the load torque

Each controller will need two inputs only. With seven fuzzy sets for each input, this result in 49 rules which will be needed. So, the PIDFC will need (49) rules only. Reducing the number of rules in a fuzzy controller makes the implementation of the fuzzy controller possible with limited processor throughput (Pfeiffer, 2009).

Moreover, the PIDFC is designed using Mamdani type (Pfeiffer, 2009). It has two inputs e(k) and Δe(k) and one output. The inputs are defined as follows:

(39)

$$e(k) = r(k) - y(k)$$

$$\Delta e(k) = e(k) - e(k-1) \tag{40}$$

Moreover, all the membership functions of the FLC inputs and outputs are defined on the common normalized domain [-1, 1].

The rule base for computing the output (u(k) (Δu(k)))is listed in Table 4. The selection of rules shown is based on the knowledge of the behavior of the error equation. For example if the output response is moving away from the set point with big steps, negative error, then a large positive control signal will be needed to reflect the direction of the output signal as the first rule implies.

In this paper the FLC scaling factors are tuned manually and a set of gains are obtained. The error allowed using these scaling factors shown in Table 5.and a conventional PID controller uses to make the comparison between a PID and PID-FC results. So, a set of PID gains are (Kp=0.005, Ki=0.0015 and Kd=0.000004). However, The closed loop structure simulated in Matlab/Simulink environment is shown in Fig. 4

Table 4. Rule Base of the PIDFC.

	NB	**NM**	**NS**	**Z**	**PS**	**PM**	**PB**
NB	PB	PB	PB	PB	PM	PS	Z
NM	PB	PB	PB	PM	PS	Z	NS
NS	PB	PB	PM	PS	Z	NS	NM
Z	PB	PM	PS	Z	NS	NM	NB
PS	PM	PS	Z	NS	NM	NB	NB
PM	PS	Z	NS	NM	NB	NB	NB
PB	Z	NS	NM	NB	NB	NB	NB

Table 5. PIDFC scaling factors values.

Gains	**Value**
Kp	0.02
Kd	0.008
Ko	0.15

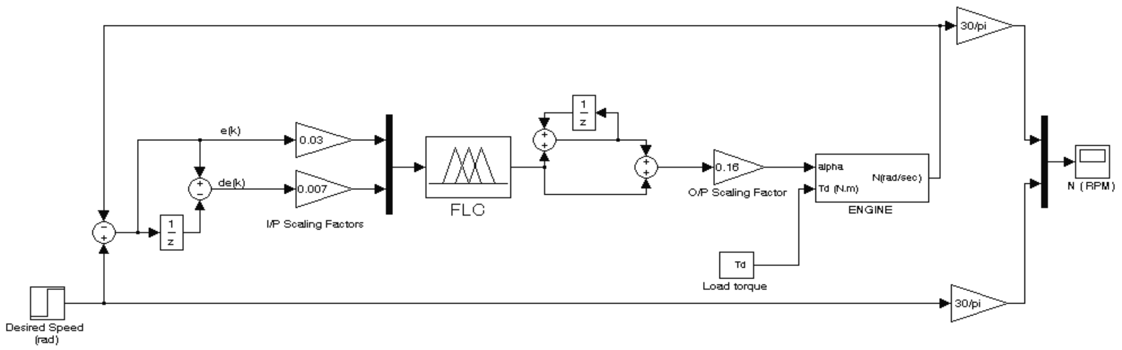


Fig. 4. Closed loop system of the engine model with PIDFC simulated in Matlab/Simulink.

The closed loop system with PIDFC has been simulated with applying load torque with and without disturbance .

a) <u>Applying Variable Value of the Load Torque:</u>

The value of torque load is applied to the close loop system, as show in Fig. 5.

b) <u>Applying Load Torque with Disturbance:</u>

The effect of parameter uncertainties and unmolded dynamics is added to the variable values of the load torque applied before as a disturbance. The disturbance applied to the simulation is a random value ranging between ±10% from the load torque, as shown in Fig. 5.

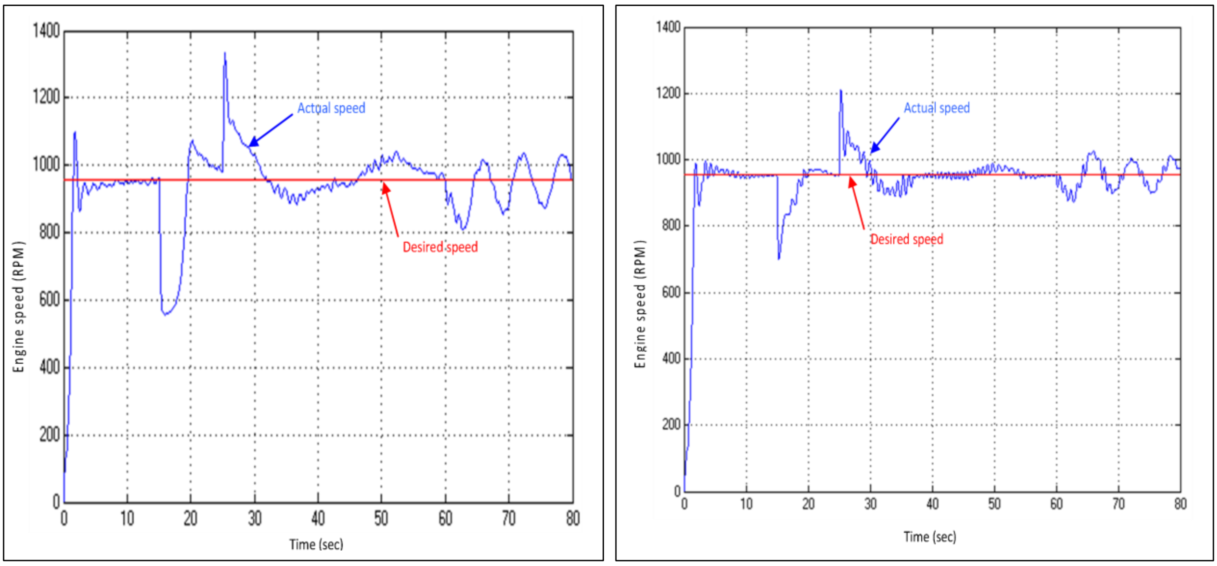


Fig. 5. Simulation result of the closed loop system with applying load torque and with disturbance (a): Engine speed with conventional PID controller, (b): Engine speed with PIDFC with and without disturbance.

4. CONCLUSIONS

In the paper, a Mean Value Engine Model is established by using SIMULINK. In the same time, a PIDFC is designed to control the engine speed in idle mode to reach it to the best performance and compare the results with PID controller.

To achieve an improvement on the speed response, a PIDFC was designed and good improvement has been obtained. The simulation results with PIDFC show that the dropping in the engine speed is reduced around (33%) compared with using PID controller. The peak overshoot is reduced around (19%) compared with using PID controller for the same value of the load torque. The settling time is reduced by (50 %) compared with using the PID controller.

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