The indicated thermal efficiency ($\eta_{i-t}$) is modeled as a function of the engine speed ($N$), air fuel ratio ($A/F$) and spark advance ($\delta$) [10].

$$\eta_{i-t} = \varphi(N(t), A/F(t), \delta(t)) \quad (25)$$

By assuming that the air fuel ratio is held tightly around stoichiometric, and the spark timing maintained at MBT value [1], ($\eta_{i-t}$) is expressed as a function of engine speed only [10]:

$$\varphi(N(t)) = \eta_\circ + \eta_1 \cdot N(t) \quad (26)$$

where ($\eta_\circ, \eta_1$) are empirical coefficient parameters of the function, related to a specific engine.

Finally, the mechanical losses which are denoted by ($mep_{mec-losses}$), are described by the following equation [10]:

$$mep_{mec-losses} = \psi(N(t), Pm(t)) \quad (27)$$

$$\psi(N(t), Pm(t)) = \beta_\circ + \beta_1 (N(t))^2 \cdot \frac{4\pi}{Vd} + (Pa - Pm(t)) \quad (28)$$

where the variables ($\beta_\circ$, $\beta_1$) are empirical constants related to a specific engine.

Now, by substituting equations (24), (25) and (27) in equation (21) the final form of the *bmep* expression will be as follows [10]:

$$bmep(t) = \frac{mac(Pm(t), N(t)) \cdot \varphi(N(t))}{(A/F) \cdot N(t)} \cdot \frac{Q_{HV} \cdot 4\pi}{Vd} - \psi(N(t), Pm(t)) \quad (29)$$

To get an expression that comprises only the *bmep* and the break torque (Tb), the bmep is defined as a constant hypothetical pressure that is assumed to act on the piston during expansion stroke and produces the same amount of work that the real engine does in two crankshaft revolutions, as expressed mathematically below [10]:

$$bmep = \frac{\text{canstant hypothetical force acting on piston face}}{\text{piston area}}$$

$$= \frac{\dfrac{\text{work produced during one cycle}}{\text{piston displacement during expansion stroke}}}{\text{piston area}}$$

$$= \frac{(\text{brake torque (Tb)}) \cdot (\text{angular duration of one cycle})}{\text{stroke volume (Vd)}}$$

$$\text{So, } bmep = \frac{Tb \cdot 4\pi}{Vd} \quad (30)$$

By the substitution of equation (30) into (29) and making the suitable simplifications, an expression for the break torque is produced as given below:

$$Tb = \frac{mac(N(t), (Pm(t)) \cdot \varphi(N(t))}{A/F(t) \cdot N(t)} \cdot Q_{HV} - \psi(N(t), Pm(t)) \frac{Vd}{4\pi} \quad (31)$$

### 2) Process Delays Model

The process of torque production is discrete depending on the engine speed. The model described here is continuous, two delays are included in the model [10, 15]:

a) Intake to torque production delay ($T_{it}$): this is the delay time between intake stroke to expansion stroke.

$$T_{it} = \frac{2\pi}{N(t)} \quad (32)$$

b) Spark to torque production delay ($T_{st}$): this is delay time from sparking to torque production.

$$T_{st} = \frac{\pi}{N(t)} \quad (33)$$

Incorporating the above delays in equation (31) gives:

$$Tb = \frac{mac(t - T_{it}) \cdot \varphi(N(t))}{A/F(t) \cdot N(t)} \cdot Q_{HV} - \psi(N(t), Pm(t)) \frac{Vd}{4\pi} \quad (34)$$

# Simulation of Mean Value Engine Model for A Single Cylinder Spark Ignition Engine


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## Abstract

Automatic control of automotive engines provides benefits in the engines performance like emission reduction, fuel economy and drivability. To ensure better achievement of these requirements the engine is equipped with an electronic control unit (ECU) that is a microprocessor based system. This control unit continually monitors the engine state using several sensors and selects better control actions to achieve what is demanded from an engine under different defined operating modes. 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.

In this paper, the automotive engine a gasoline powered, four strokes, Port Fuel Injection (PFI), Spark Ignition (SI) engine is modeled and simulated using a class of models called Mean Value Engine Models (MVEMs). This model can successfully be used during the development of control system that will be implemented later in the ECU of a car. The model includes air, fuel and rotational dynamics as well as process delays inherent in the four stroke cycle engine. The model is validated for parameters of an engine, to an accuracy of about (95%).



**Keywords:** Engine model, spark ignition, modeling, mean value.


## 1. Introduction

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 that will be implemented later in the ECU of a car. MVEM represents a basis for the development of different engine control strategies [1]. It is defined as follows:

"MVEMs are models where the signals, parameters and variables are averaged over one or several cycles", [2]. MVEMs offer simplified description for the processes taking place in the engine using physically based equations, and are acceptable for the purposes of control synthesis [3]. These models are continuous time, nonlinear, low frequency, phenomenological with uniform homogeneous charge and lumped parameters of breathing and rotational dynamics [4].

## 2. Engine Modeling:

It will be assumed that the model simulates a gasoline powered, four stroke, PFI, Spark Ignition (SI) engine. The model includes air, fuel and rotational dynamics as well as process delays inherent in the four stroke cycle engine. It is found that when modeling such a complex system, as the SI engine, it is beneficial to divide it into distinct subsystems [5].

The basic configuration of the engine model has three basic subsystems, describing the main phenomena take place in SI engine [6, 7]. These subsystems are shown in fig (1):

a- Intake air path subsystem.
b- Fuel dynamics subsystem.
c- Crankshaft dynamics subsystem.

Fig. (1) shows the block diagram of the engine model.

### a- Intake Air Path Dynamics

In this section a Mean Value Model for the intake air path of SI engine is described. It is divided into three subsystems that have to be modeled, these are:

1) Throttle body.
2) Intake manifold.

3) Cylinder air induction.
However, Fig. (2) shows the Schematic diagram of the intake air path.

### 1) Static Model of the Throttle Body

The air mass flow into the cylinders, and thereby, the output power of the engine is controlled by the throttle valve. Its opening depends on the pedal position [6].

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

Where:

$\dot{mat}(t)$: is the air mass flow rate through the throttle valve.
Cd: is the discharge coefficient which is an experimentally determined constant and it relates the effective throat area to the actual throat area. It is roughly equals to 0.7 [1, 8].
Pa: ambient pressure ($N/m^2$).
R: ideal gas constant (J/kg.K).
Ta: ambient temperature (K)

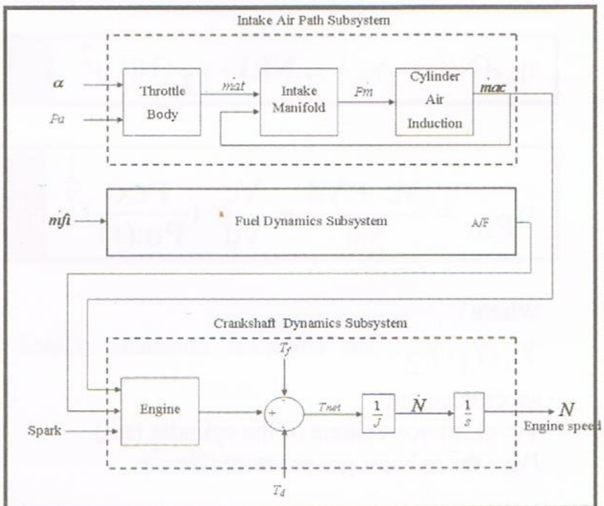

Fig. (1): Block diagram of the engine model.

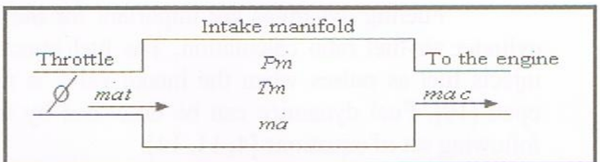

Fig. (2): Schematic diagram of the intake air path

$\phi(Rp(t))$: Function of pressure ratio across the throttle position, which depends on the flow conditions, where under chocked flow, $\phi(Rp)$ is constant and is calculated as follows [8, 9]:

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

Where:
$\gamma$ : denotes the ratio of specific heats of air which is assumed to be (1.35) [10].
Using this value of $\gamma$, equation (2) above is calculated to be equal to (0.6761).
Rp: represents the ratio of the manifold pressure (Pm) to the ambient pressure (Pa).
Moreover, during sonic flow $\phi(Rp)$ is a function of time and is calculated with the following equation [8, 9]:

$$\phi(Rp(t)) = (Rp(t))^{\frac{1}{\gamma}}\sqrt{\frac{2\gamma}{\gamma-1}\left[1-(Rp(t))^{\frac{\gamma-1}{\gamma}}\right]} \qquad (3)$$

For $Pm(t) \ge Pc(t)$

Where:
Pc: is the critical pressure where the flow reaches sonic condition in the narrowest part and is calculated as follows:

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

The area available for the flow ($A(t)$) equals the cross sectional area of the channel less the area blocked by the throttle plate. The blocked area depends on the throttle angle ($\alpha$) measured from fully closed position in radian as shown in Fig. (3) [10].

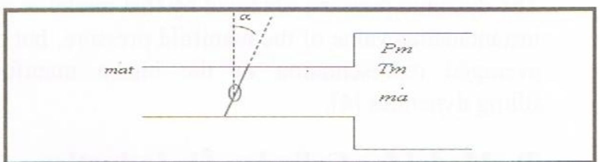

Fig. (3): Air flow past the throttle plate

If the channel and the throttle plate are assumed to be circular in shape with diameter (Dth), the available area for the flow can be expressed as [10]:

$$\dot{A}(t) = \frac{\pi}{4} Dth^2 (1 - \cos(\alpha(t))) \quad (5)$$

This equation is modified to the following form to account for the leakage mass flow rate. Deriving from the fact that when the engine throttle is fully closed, it holds $\alpha = \alpha_{leak}$ [10, 11, 12].

$$\dot{A}(t) = \frac{\pi}{4} Dth^2 (1 - \frac{\cos\alpha(t)}{\cos\alpha_{leak}}) + A_{leak} \quad (6)$$

where $A_{leak}$ is the flow area when the throttle angle equals $\alpha_{leak}$.

### 2) Dynamic Model of the Intake Manifold

The intake manifold dynamics can be described by a first order differential equation that is based on the mass conservation equation as given below [6, 10, 11]:

$$\dot{P}m(t) = \frac{RTm}{Vm} \dot{m}a(t) \quad (7)$$

Where

$$\dot{m}_a(t) = \dot{m}_{at}(t) - \dot{m}_{ac}(t) \quad (8)$$

$Tm$ : Manifold air temperature (K)
$Vm$ : Manifold volume ($m^3$)

$\dot{m}_{ac}(t)$: Mass flow rate of the homogeneous mixture of air and exhaust gas.

$\dot{m}_{at}(t)$ : Air mass flow through the throttle valve.

$\dot{m}_a(t)$ : Intake manifold mass of air.

The dynamic pressure obtained by this model is not instantaneous value of the manifold pressure, but an averaged representation of the intake manifold filling dynamics [4].

### 3) Model for Cylinder Air Induction

The gas in the intake port is assumed to be a homogeneous mixture of air, residual gases are left in the cylinder after the exhaust phase has taken place and the exhaust gas recycled to the intake port through the Exhaust Gas Recycled (EGR) system, with the same temperature for all the mixture.

The mass flow rate of the homogeneous mixture of air and exhaust gas is calculated using this concept where its mathematical expression is as follows:

$$\dot{m}ac = mac(Pm(t), N(t)) \quad (9)$$

$$\dot{m}ac(t) = \frac{Pm(t)}{RTm} \cdot \frac{N(t)}{2\pi} \cdot Vd \cdot \eta_v \quad (10)$$

where ($N$) is the rotational speed of the engine in radian per second (r/s) and Vd is the displacement volume of the engine cylinder in ($m^3$).
In this paper, the fraction of EGR is assumed to be 0.1 of the air induced to the cylinder. The effect of engine speed and the manifold pressure on the volumetric efficiency is modeled separately in an individualistic manner as follows [10]:

$$\eta_v = \eta_N(N(t)) \cdot \eta_{Pm}(Pm(t)) \quad (11)$$

$$\eta_N(N(t)) = \gamma_o + \gamma_1 . N(t) + \gamma_2 . (N(t))^2 \quad (12)$$

$$\eta_{Pm} = \frac{Vc + Vd}{Vd} - \frac{Vc}{Vd} \cdot (\frac{Pex}{Pm(t)})^{\frac{1}{\gamma}} \quad (13)$$

Where :
$\gamma_o, \gamma_1, \gamma_2$: are empirical constants related to specific engine.
Vc: clearance volume of the cylinder ($m^3$).
*Pex* : the exhaust gas pressure ($N/m^2$).

## b. Fuel Path Dynamics

Fueling dynamics are important for the in cylinder air-fuel ratio calculation. The fuel injector injects fuel as pulses when the intake valve is not open [10]. Fuel dynamics can be described by the following set of equations [4, 11, 14]:

$$\dot{m}ff(t) = x \cdot \dot{m}fi(t) - \frac{1}{\tau_f} mff(t) \quad (14)$$

$$\dot{m}fv(t) = (1 - x)\dot{m}fi(t) \quad (15)$$

$$\dot{m}f(t) = \dot{m}fv(t) + \frac{1}{\tau_f} mff(t) \quad (16)$$

where:

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

$\dot{m}ff$ & *mff*: mass flow rate (kg/sec) and mass of the fuel film (kg) respectively.

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

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

*x*: fraction of the injected fuel which is deposited on the manifold or intake port as fuel film, and is calculated using the following equation [4]:

$$x = 0.3 + \frac{0.7}{90} \alpha(t) \quad (17)$$

$\tau_f$ : fuel evaporation time constant which is calculated as [14]:

$$\tau_f = 0.05 + \frac{2.25}{N(t)} \quad (18)$$

Ideal fuel delivery is assumed, i.e., the air fuel ratio is assumed to be kept at the stoichiometric value (14.67) [1]. The fuel injection $\dot{m}fi$ is set according to the evolution of the air charge. The air charged to the cylinder is divided by stoichiometric A/F which dented by ( □□□in the Simulink model to provide the feed-forward fuel flow command.

## c. Crankshaft Dynamics

The crankshaft variable of interest is the revolution speed expressed in revolution per minute (RPM) or rad. per second (r/s), which depends on the torque produced by cylinder during expansion stroke. Three important models must be explained clearly in the crankshaft dynamics model. These are:

1) Torque production.
2) Process delays.
3) Rotational dynamics of the engine.

### 1) Torque Production Model

The torque produced by the cylinder during expansion stroke depends in a nonlinear fashion on the mass of air loaded to the cylinder, air fuel ratio taken during the earlier intake stroke, spark timing and mass of residual gases left after the earlier exhaust stroke [4]. This can be expressed using the mean effective pressure (mep) notation as follows:

$$\eta_{i-t} = \frac{imep}{mep_{fe}} \quad (19)$$

$$bmep = imep - mep_{mec\_losses} \quad (20)$$

$$bmep = mep_{fe} \cdot \eta_{i-t} - mep_{mec-losses} \quad (21)$$

Where ($mep_{fe}$) is the (bmep) that will be produced if the engine is fully efficient, as expressed mathematically in the following equations:

$$mep_{fe} = \frac{\text{const.hyp. force on piston if engine 100\% efficient}}{\text{piston area}}$$

$$= \frac{\dfrac{\text{work produced in one cycle with 100\% effeciency}}{\text{piston displacement during expansin stroke}}}{\text{piston area}}$$

$$= \frac{(\text{fuel mass inducted in cycle } (M_f)) \times (\text{fuel heating value } (Q_{HV}))}{\text{stroke volume (Vd)}}$$

$$\text{So,} \quad mep_{fe} = \frac{M_f \cdot Q_{HV}}{Vd} \quad (22)$$

The mass of fuel inducted during one cycle ($M_f$) is calculated as follows [4]:

$$A/F = \frac{\dot{m}ac}{\dot{m}f} \quad \Rightarrow \quad \dot{m}f = \frac{\dot{m}ac}{A/F}$$

and for one cycle:

$$M_f = \frac{\dot{m}ac\ (\text{for one cycle})}{A/F}$$

$$M_f = \frac{\dot{m}ac(t)}{A/F(t)} \cdot \frac{4\pi}{N(t)} \quad (23)$$

Using equations (22) and (23), the final form of the ($mep_{fe}$) is found:

$$mep_{fe}(t) = \frac{\dot{m}ac(t) \cdot Q_{HV}}{A/F(t) \cdot Vd} \cdot \frac{4\pi}{N(t)} \quad (24)$$

### 3) Rotational Dynamics Model

The rotational dynamics of the engine crankshaft is obtained by applying Newton's second law for rotational motion [4].

$$T_{net} = J\dot{N} \qquad (35)$$

where:

$T_{net}$: is the net torque used for vehicle acceleration (N.m).

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

and ($\dot{N}$) (rad/sec$^2$) represents the angular acceleration of the engine which will be integrated later to get angular speed of the engine. $T_{net}$ is calculated from the difference between the torque produced by the cylinders during combustion process and the sum of all the load torques placed on the engine.

## 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.

### 3.1 Simulation of Intake Air Path Dynamics

The intake air path dynamics block simulates equations from (1) through (13). The intake air path dynamics consists of three subsystems: throttle body subsystem, intake manifold subsystem and cylinder air induction subsystem. The top level view of the intake air path simulation is shown in Fig. (4):

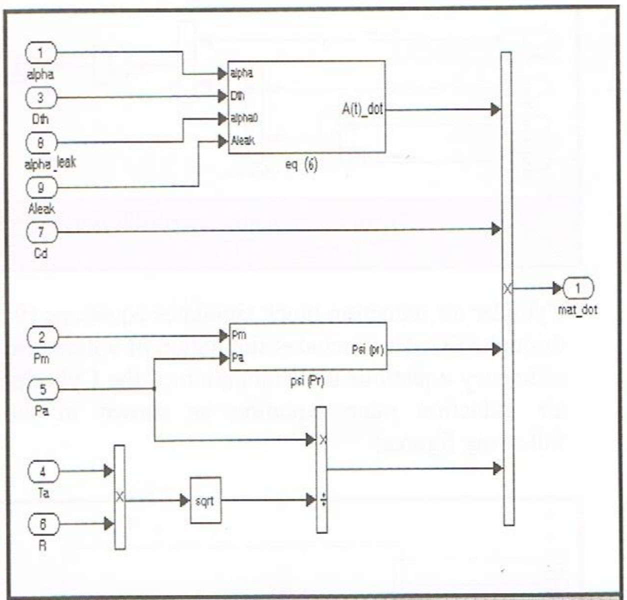


Fig. (4): Simulation of Intake air path dynamics, (equations (1) through (6)).

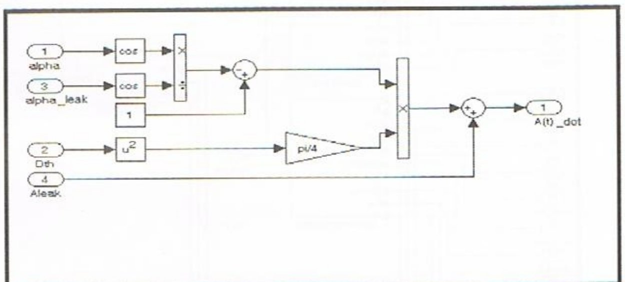


Fig. (5): Simulation of the area available for the flow equation (6).

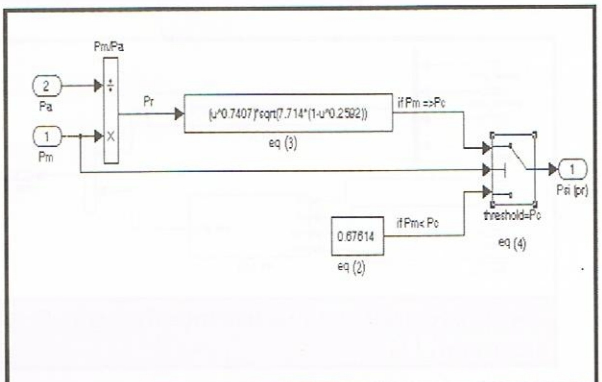


Fig. (6): Simulation of the pressure ratio across the throttle, (equations (2), (3) and (4)).

The simulation of the mass conservation equation described as shown in the following figure:

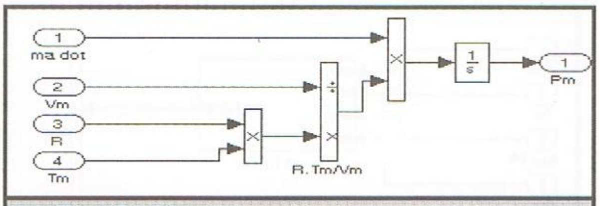


Fig. (7): Simulation of intake manifold block, (equation (7)).

Cylinder air induction block simulates equations (9) through (13). This includes simulation of volumetric efficiency equations and simulation of the Cylinder air induction main equation as shown in the following figures:

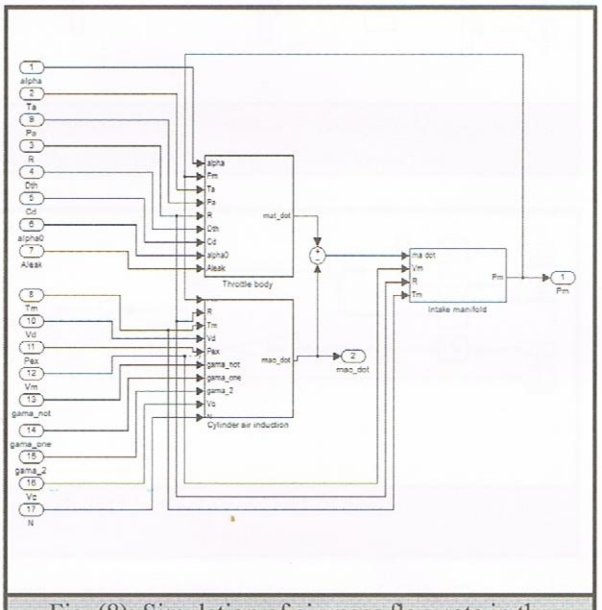


Fig. (8): Simulation of air mass flow rate in the intake manifold, (equation (8)).

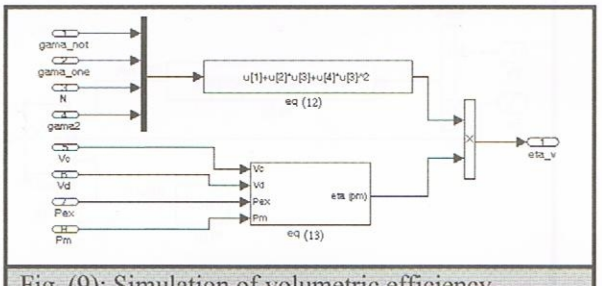


Fig. (9): Simulation of volumetric efficiency, (equation (11)).

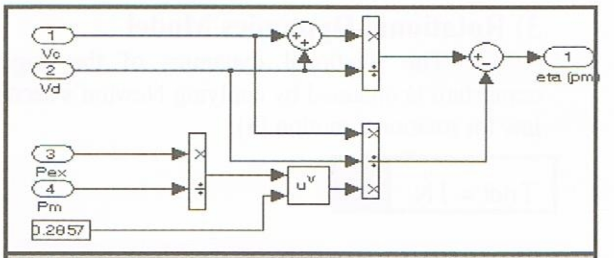


Fig. (10): Simulation of volumetric efficiency as a function of manifold pressure equation (13)).

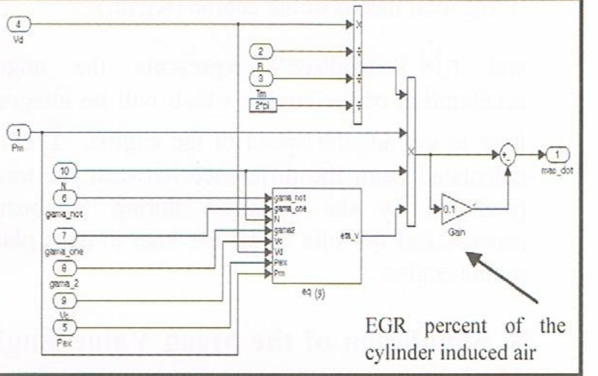


Fig. (11): Simulation of cylinder air induction dynamics, (equation (10)).

### 3.2 Simulation of Fuel Path Dynamics

The top level view of the fuel path dynamics and the details of its blocks is shown in figures below:

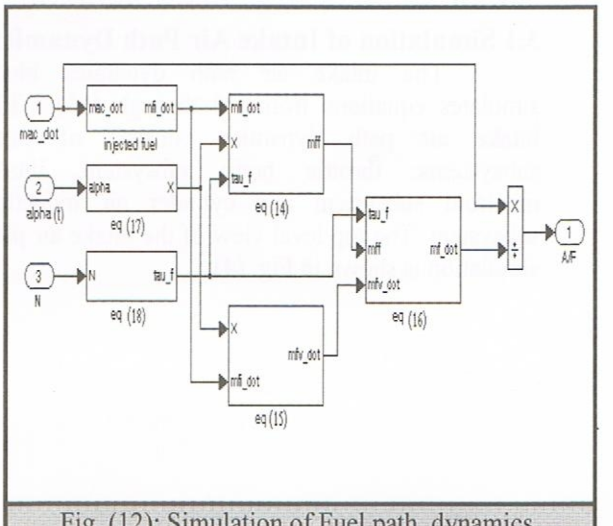


Fig. (12): Simulation of Fuel path dynamics, (equations 14 through 16).

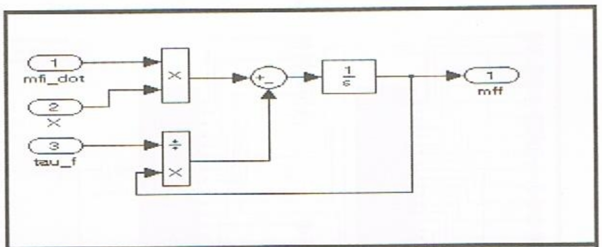


Fig. (13): Simulation of the fuel film part of the injected fuel (equation (14)).

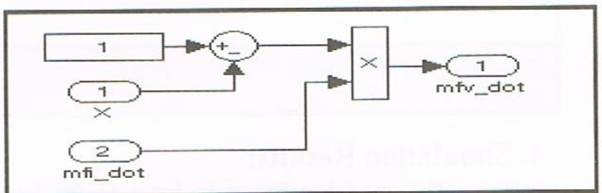


Fig. (14): Simulation of the evaporated part (equation (15)).

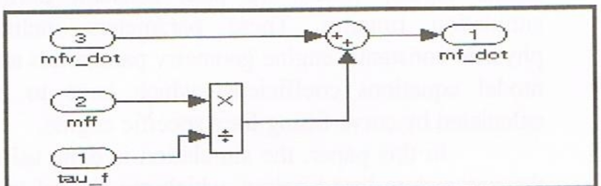


Fig. (15): Simulation of the total fuel which enters the cylinder (equation (16)).

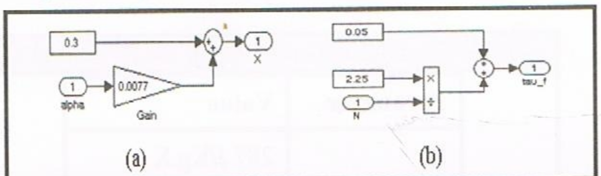


Fig. (16):
Simulation of fuel film fraction (equation (17)).
b)Simulation of the fuel evaporation time constant (equation (18)).

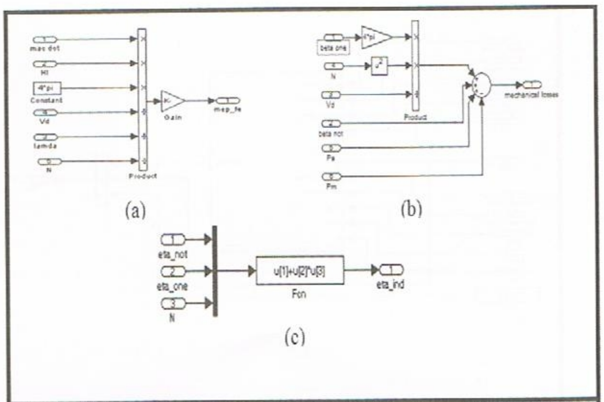


Fig. (17):
a) Simulation of the full efficient engine bmep (equation (24)).
b) Simulation of the mechanical losses (equation (28)).
c) Simulation of indicated thermal fficiency (equation (26)).

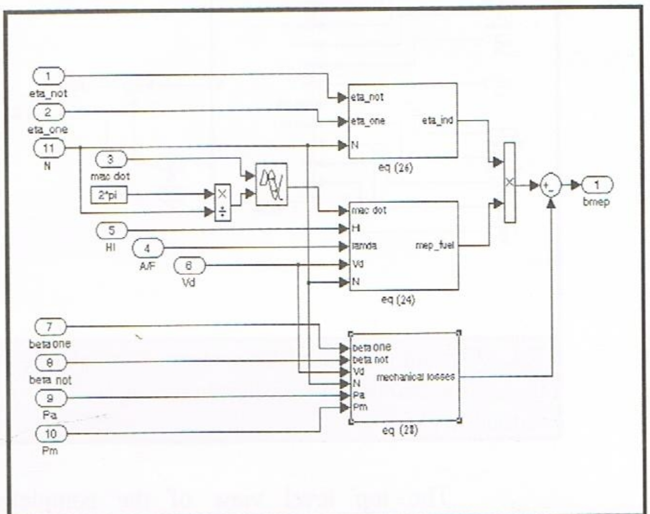


Fig. (18): Simulation of the (bmep) produced by the engine, (equation (29))

### 3.3 Crank Shaft Dynamics Simulation

The crankshaft dynamics block simulates equations from (19) through (34), this includes $mep_{fe}$ which is the (bmep) produced with full efficient engine, indicting thermal efficiency ($\eta_i$), and mechanical losses as shown in the following figures:

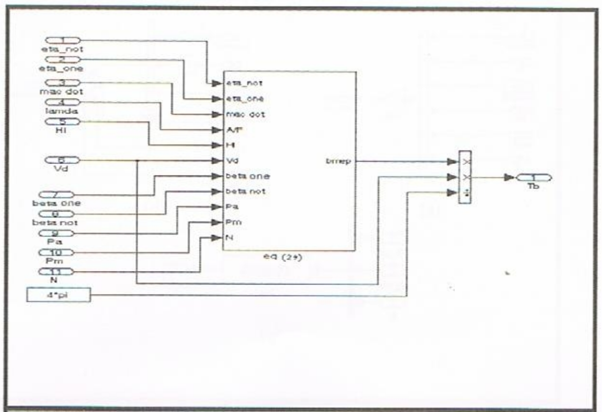


Fig. (19): Simulation of break torque developed by the engine, (equation (34))

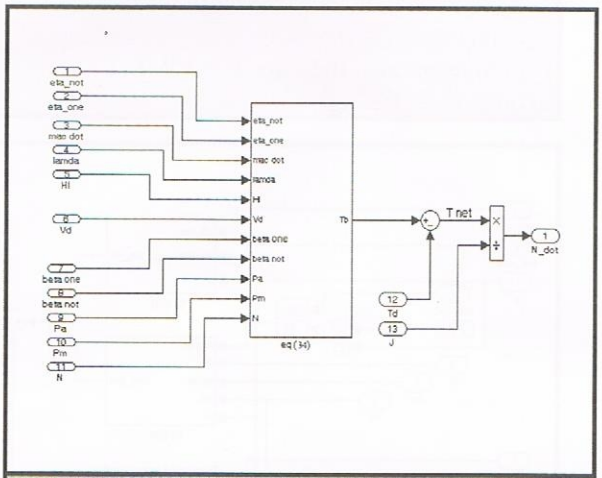


Fig. (20): Simulation of the net torque produced by the engine and used to accelerate the engine (equation (35))

The top level view of the completed simulink model is shown in Fig. (20) below.

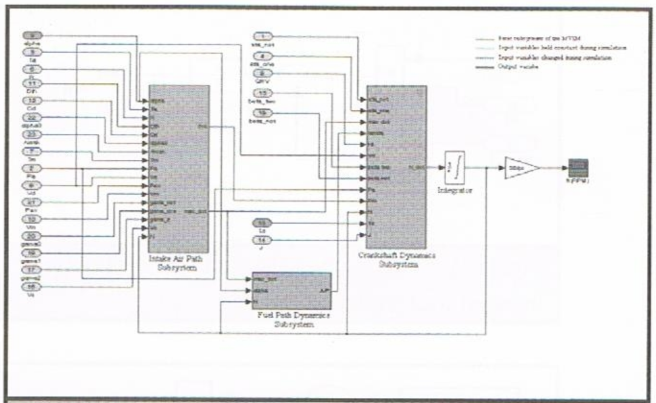

Fig. (21) Top level view of the simulink model

## 4. Simulation Results:

The model parameters have to be listed before the simulation could be run and the results could be obtained. These parameters represent the model inputs, which are held constant during simulation running. These parameters include physical constants, engine geometry parameters and model equations coefficients which have to be calculated by curve fitting for a specific engine.

In this paper, the simulation is done using the parameters listed below, which are related to a gasoline powered, four stroke, single cylinder, PFI, SI engine prototype with idle speed equal to 955 RPM [16]. All the physical parameters used in the model are listed in Table (1) as shown below:

Table (1): Physical constants of the MVEM [10]

| Parameter | Value |
|---|---|
| R | 287 J/Kg.K |
| Ta | 298 K |
| Pa | $0.98*10^5$ N/m$^2$ |
| $\gamma$ | 1.35 |
| Tm | 340 K |
| Cd | 0.7 |
| Pex | $1.08*10^5$ N/m$^2$ |

On the other hand, engine geometry parameters needed in the model are listed in Table (2) below:

Table (2): Engine geometry parameters [10]

| Parameter | Value |
|---|---|
| $\alpha_0$ | 0.1379 radians |
| Dth | $58.7*10^{-3}$ m |
| $A_{leak}$ | $5.3*10^{-6}$ $m^2$ |
| Vm | $5.8*10^{-3}$ $m^3$ |
| Vd | $2.77*10^{-3}$ $m^3$ |
| Vc | $0.277*10^{-3}$ $m^3$ |
| J | 0.2 $Kg.m^2$ |

The coefficients of the curve fitting equations used in the model are listed in Table (3) below [10]:

Table (3): Model's equations coefficients parameters

| Parameter | Value |
|---|---|
| $\gamma_o$ | 0.45 |
| $\gamma_1$ | $3.42*10^{-3}$ sec |
| $\gamma_2$ | $-7.7*10^{-6}$ $sec^2$ |
| $\eta_o$ | 0.16 J/kg |
| $\eta_1$ | $2.21*10^{-3}$ |
| $\beta_o$ | 15.6 N.m |
| $\beta_1$ | $0.175*10^{-3}$ |

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

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

A step change in the throttle angle is applied to the model from 0.1589 radian to 0.1605 radian at 10 seconds [10] the result was a step change in the engine rotational speed as shown in Fig. (21).

## 2) Step Change in the Throttle angle with Load Torque:

The engine load during idling is of variable nature and this could be the source of sudden drop or rise in engine speed [10], where the nonlinear model is loaded with variable value of load torque. The load torque was applied first as a step change form to represent the loads that affect suddenly [17], then a trapezoidal signal used to represent the loads that increased gradually [12, 18], and finally a sine wave signal were also applied to represent the load torque [15]. The loads were applied in the presence of the same step change in the throttle angle that was used before and the open loop responses are shown in Fig. (22). 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.

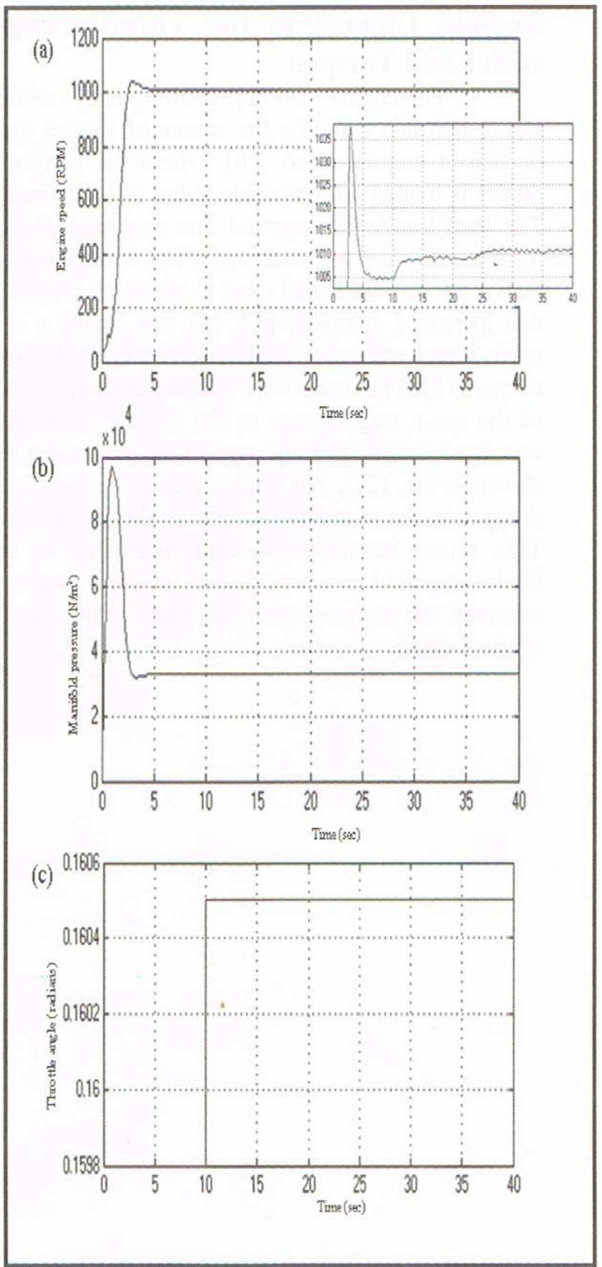


Fig. (22): Open loop dynamics of the MVEM with no load applied:
a): Open loop speed response.
b): intake manifold pressure response.
c): Step change in the throttle angle

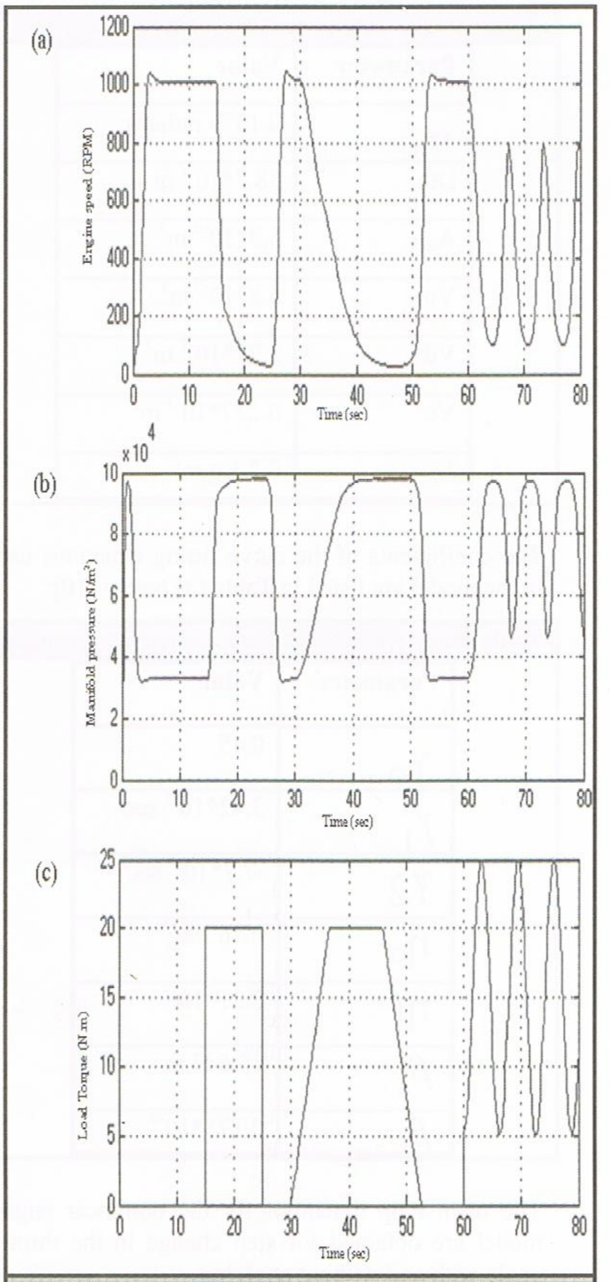


Fig. (23): Open loop dynamics of the MVEM with applying variable value of load torque:
a): Open loop speed response with $\alpha = 0.1604$ rad.
b): Intake manifold pressure response
c): Variable value of the load torque with $\alpha = 0.1604$ rad

## 4. Conclusions

The following points can be concluded from this work:

- A MVEM is used in the engine modeling phase. The MVEM is control oriented, continuous time, nonlinear and physically based model. This model describes the dynamic engine variables like engine speed or intake manifold pressure as mean value rather than instantaneous values on a long time scale.

- The model is validated, by comparing it with the results of open loop dynamics obtained referring to [10] for the same engine parameters, to an accuracy about (95%).
- The open loop dynamics of the engine shows that the dropping in the engine rotational speed is nearly equal to (99%) when (20 N.m) of load torque is applied and this is very large value and will, surly, cause engine stalling.

Finally, it can be concluded that the MVEM can be used in simulating four strokes, PFI, SI, single piston engine. It's only required to provide the model with the values of some physical parameters and engine geometry parameters as well as make curve fitting to torque production subsystem and the volumetric efficiency equation.

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# محاكاة موديل معدل القيمة لمحرك إشعال الشرر أحادي الأسطوانة

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**الخلاصة:**

إن السيطرة الالية على المحركات الذاتية الحركة يحقق فوائد عديدة في خصائص هذه المحركات مثل خفض الانبعاث واقتصاديه الوقود واداء المحرك. ولضمان أفضل انجاز لهذه الخصائص فقد جهزت هذه المحركات بوحدة سيطره الكترونيه، وهي عبارة عن نظام يعمل بمعالج دقيق . تراقب وحدة السيطرة وبصورة مستمرة حالة المحرك باستخدام بعض المتحسسات وتقوم بإختيار أفضل إجراء لإنجاز ما مطلوب من المحرك ضمن انماط تشغيل مختلفه. أن عمل نموذج رياضي لهذا النوع من المحركات يجعل عملية تصميم منظومة السيطرة والتشخيص مبسطه لكون موديل المحرك سيكون بديلا عن أجراء تجارب مكثفه على السيارة .

تم في هذا البحث بناء ومحاكاة موديل رياضي لمحرك بنزين ذو أربعة أشواط يعمل عن طريق أشعال الشرر بإستخدام الحاقن المنفذي عن طريق بناء مجموعه بناء من الموديلات تسمى نماذج معدل القيمة للمحرك . ويمكن إستخدام هذا الموديل الرياضي وبنجاح لغرض تطوير منظومة السيطرة وتنفيذها لاحقا في وحدة التحكم الايكترونية للسيارة . إن الموديل مكون من ديناميكيات الهواء والوقود والدوران وكذلك على أزمان التأخير المتضمنة في محرك الأربعة أشواط، وقد تم أثبات صحة الموديل بإستخدام عناصر موديل حقيقي لمحرك وكانت دقة النتائج حوالي 95 %.