A Review on Engine MVEM Models and AFR Control Methods to Developing a New Engine Model and Ziegler Nichols PID Fuzzy controller: Applied to SI Engines

Mohammad Javad Nekooei, Jaswar, Agoes Priyanto, Zahra Dehghani

Department of Aeronautic, Automotive and Ocean Engineering, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, Johor Bahru, Malaysia.

ABSTRACT

Background: Because of the growing demand of governments and customers, it is an attempt of the vehicles manufacturers for significantly reducing the fuel consumption and pollution emissions while optimum performance of the engine should be maintained. The stoichiometric value of AFR (14.7) is considered as an essential parameter of combustion control. Objective: To develop a new engine MVEM models which is include all of engine dynamics parts such as, engine speed, torque, injection time, AFR and etc. To design and optimize a PID controller by Ziegler Nichols (Z/N) tuning method to control AFR and comparative study with proposed controller method. To develop and simulate an AFR Z/N PID Fuzzy (Hybrid Fuzzy) controller by MATLAB/SIMULINK. Results: A new engine MVEM structure model and a AFR Z/N PID Fuzzy (Hybrid Fuzzy) simulated model. Conclusion: The proposed engine and control model after validation can easily use for any kind of SI engine to modeling and AFR control purpose.

Key word: SI Engine, Air to Fuel ratio, Emission, Fuzzy Controller, Z/N PID controller

Introduction

In 2010 it was appraised that 1,016,760,000 automobiles are in use in the world (Sousanis, J., 2011). This number is corresponding to 165 cars for each 1000 people or it can be said that every 6 persons own a car. This is followed by the rate of motorization in Europe which is for every 1000 people, 559 vehicles (Statistics, P., 2011). On the other hand, the environment can be more polluted by the emissions from ships and boats which are used in international trade in the seas surrounding ports. As an example, in Europe, 2.3 million tonnes of sulfur dioxide and 3.3 million tonnes of nitrogen oxides have been spread in the air in 2000 (Statistics, P., 2011). It is significant for the researchers to consider the impact of environmental conditions to the system. It is not a true belief that increasing vehicles ownership may bring convenient transportation for the people around the world. This happening will consume so many resources and touching the environment in a negative way. Since un-renewable fossil fuel is the main of traditional car fuel, fuel consumption will drastically increase by increasing number of vehicles. Additionally, petroleum resources will be reduced and even raise the petroleum exhaustion possibility. Moreover, consumption of fossil fuel in the cars generates toxic and dangerous emissions. Hydrocarbons (HC), Nitrogen oxide (NOX), Carbon dioxide (CO2) as well as particulate matter (PM) are some of these emissions (Koto, J. and Y. Ikeda, 2002). Because of the growing demand of governments and customers, it is always an attempt of the vehicles manufacturers for significantly reducing of emissions and fuel consumption while optimum engine performance should be maintained. In order to reach this goal, so many different variables must be controlled. Engine torque, engine speed, timing of fuel injection, timing of spark ignition, air intake, AFR are the main variables to be controlled. They are complicatedly related to each other. Furthermore, several different operating modes are defined for the SI engines such as startup, idle, running and braking so that made the engine dynamics very nonlinear and multivariable due to those factors (Balluchi, A., et al., 1999). AFR, between the engine control variables, is considerably interconnected to, emission reduction, fuel efficiency and drivability improvement. It is important to maintain air fuel ratio to be equal to the stoichiometric value (14.7) by which most excellent equilibrium can be obtained between output power and fuel consumption (Kilagiz, Y., et al., 2005). The effect of emission control can be also influenced by AFR due to its stoichiometric value that guarantees the highest three way catalysts (TWC) efficiency. According to (Benninger, N. and G. Plapp, 1991), variation of AFR should be within ± 0.2%. More than 1% variation below 14.7 may significantly rise emission of CO and HC. As shown in Figure 1, production of NOx up to 50%, could be results of deviation greater than 1% (Kilagiz, Y., et al., 2005).
The best balancing between the power output and consumption of fuel can be reached by maintaining AFR to be the stoichiometric value (14.7). The excess air factor, or lambda is frequently defined as the AFR. At normal temperature and pressure, a lambda factor of unity is consistent with an AFR of 14.7:1 which is called the stoichiometric ratio. This is correlated to the air and fuel proportions by which a complete combustion can be formed. A lambda of less than unity is happen with greater proportion of fuel which called rich mixture while greater air proportion results a lambda of greater than unity which is named weak (lean) mixture. When lambda is around 0.86, it is possible to reach the maximum power. With a lambda around 1.05 the fuel consumption is minimum (Lee, S.H., et al., 2007) AFRs curve versus power is illustrated in Figure 2.

Many studies have been designed engine model but in most, of then all of dynamics part of engine are not used for modeling or many of them used heavy mathematical equations which will increase the modeling time, load of computational and delay of the model response when a controller is applied. Several different control techniques have been employed for controlling air to fuel ratio. They are Fuzzy control, PID, Predictive control, Switching frequency control, Sliding mode (SM) control, etc. Some techniques of control are based on Neural Network (NN).

It is so expensive and time consuming to test new controllers on real engine. Fortunately the engine simulation model brings a chance to carry out investigation on precise and effectual ratio controllers. But, it is still required for developing an engine simulation model in order to solve a wide range of control problem. So, at first in this paper an new engine model developed based on the reviewed MVEM engine model and then a
new Z/N PID with using fuzzy control (Hybrid Fuzzy) to tuning the Z/N PID gains has been developed based on the reviewed AFR controller methods.

In next section the engine MVEM model and AFR controller methods have been reviewed and then the proposed engine model and control method have been explained.

II. Summary of MVEM models:

A model named the mean value engine model (MVEM) was planned and got additional development by different scientists (Cook, J. and B.K. Powell, 1988). Finally, (Hendricks, E. and S.C. Sorenson, 1990) systematically summarized the mean model. Generally, for describing the dynamic process of the engine, the mean value of variables involved in cycle system of the engine is used in this model. Therefore, the engine dynamic characteristics can be correctly reflected in the transient conditions. Subsequently, scientists and researchers developed and enhanced the MVEM predominantly in the oil film as well as the torque models.

Together with the science and technology improvement, many scientists enhanced the MVEM; they have applied hybrid models and intelligent control as well. The scope of the MVEM application has been spread by (Kellerer, H., et al., 1996) since he applied this model to a turbocharged gasoline engine. The air/fuel impact and spark angle have been taken into account by (Yoon, P., et al., 2000) on the output torque. For modeling of gasoline engine, a hybrid model was established by (Balluchi, A., et al., 1999).

In this section the main important MVEM models which developed by (Alippi, C., et al., 1998; Wang, S., et al., 2006; Yoon, P., et al., 2000; Cook, J. and B.K. Powell, 1988), have been reviewed and summarized. Several different simulation model structures are exist that are not included in this literature review. This is because they may be similar to the discussed models or lack sufficient details.

Figure 3, 4, 5 and 6 illustrates the structure of engine simulation structures proposed by (Alippi, C., et al., 1998; Wang, S., et al., 2006; Cook, J. and B.K. Powell, 1988) respectively.

The model of (Alippi, C., et al., 1998) which have some essential and basic constituting blocks. There are six engine model inputs such as Speed of engine (N), Angle of the throttle (\(\alpha\)), External temperature (\(T_e\)), Eternal pressure (\(P_e\)), Temperature of engine manifold (\(T_m\)), Time of fuel injection (\(T_i - \tau_{con}\)) and the outputs is AFR. Throttle body dynamic and time delay are not include this model.

![Fig. 3: Block of Engine Simulation (Alippi, C., et al., 1998)](image)

Figure 4 shows the model of engine simulation which is presented by (Wang, S., et al., 2006). There are two input variables (open angle of throttle (\(u\)) and flow rate of fuel (\(m_{fi}\))), and one output (AFR (air fuel ratio)) in this model of engine simulation. The outputs represent AFR, this model are not include throttle body and fuel injection dynamics.
A large number of types of SI engines can be simulated by an engine model (a nonlinear dynamic model) which is introduced by (Yoon, P., et al., 2000). Different variables which are included in the engine simulation model are illustrated in figure 5. The inputs are angle of throttle ($\alpha$), flow rate of fuel ($m_f$) and spark timing (SA) and outputs are pressure of intake manifold ($P_{man}$), speed of engine (N) and AFR time delay ($\lambda_e$). This model is not include the fuel injection time dynamics and the manifold pressure is the output.

Based on the engine simulation models reviewed above, a new model of engine simulation structure can be designed. All of the dynamic parts can be simulated well in the (Wang, S., et al., 2006)'s engine simulation model. However, throttle body is not taken into account in this model. Because of this, we will combine this model with...
throttle body dynamic model. We used whole box of intake manifold dynamics instead of manifold pressure and temperature dynamics. This new simulation model includes three input variables: throttle angle (\( \alpha \)), engine speed (N), injection fuel rate (\( m_f \)) and two outputs AFR ratio and engine torque Figure 7 illustrates proposed engine simulation structure.

**Fig. 7: Proposed Engine Simulation Structure**

With regard to the above structure of engine system there are five main parts in an engine simulation system: Throttle body model, dynamics model of Intake manifold, dynamic model of Lambda, dynamic model of Injection and dynamic model of Crankshaft. Every dynamic part has specific mathematical equation which it’s necessary to simulation the engine model by MATLAB/SIMULINK function. Each dynamic part has explained in next section.

**II.1 Engine dynamics mathematical equations:**

**A. Throttle body model:**

**A.1 Area of throttle:**

The filter side and intake side are separated by the throttle plate which is considered as air-flowing valve. A particular amount of air is allowed to flow into the intake manifold by the throttle provided flow area. (Heywood, J.B., 1988) demonstrated that the throttle angle can adjust the flow area as following:

\[
A_{th}(\alpha) = \frac{\pi D^2}{4} \left( 1 - \frac{\cos \alpha}{\cos \alpha_0} \right) + \frac{D^2}{2} \left( \frac{K}{\cos \alpha} (\cos^2 \alpha - K^2 \cos^2 \alpha_0)^2 - \frac{\cos \alpha}{\cos \alpha_0} \sin^{-1}\left( \frac{K \cos \alpha_0}{\cos \alpha} \right) - K(1 - K^2)^{1/2} + \sin^{-1}K \right) \]

Where \( \alpha_0 \), D, d and \( A_{th}(\alpha) \) denoted the open angle of throttle, angle for minimum leakage area, Diameter of throttle bore, Diameter of throttle lot and Area of throttle.

**A.2 Air mass posterior to the throttle:**

By opening the throttle plate, captured air before the throttle is possible to move into the intake manifold. [38] used a differential equation in order to calculate that how much air volume is passed. There are several parameters in a relationship to be known by which the flow rate of air mass can be calculated. They are: discharge coefficient, the area of the throttle, pressure and temperature before the throttle, pressure of intake manifold, gas constant, and specific heat ratio. This relationship is as following:

\[
\dot{m}_{ath} = \frac{C_{d} A_{th} P_0}{\sqrt{R T_0}} \left( \frac{2 \gamma}{\gamma - 1} \right) \left( \frac{P_i}{P_0} \right)^{\frac{2}{\gamma - 1}} - \left( \frac{P_i}{P_0} \right)^{\frac{\gamma + 1}{\gamma - 1}} \right)^{1/2} \]

(1.2)
Where \( C_d \), \( \rho_0 \), \( T_0 \), \( R \), \( \gamma \), and \( p_t \) denoted the Discharge coefficient (0 \( \leftrightarrow \) 1), pressure before throttle, temperature before throttle (Kelvin), Gas constant (287 J/kg * K)), Specific heat ratio and Intake manifold pressure (KPa). A linear regression is used in this study based on the outcomes from the experiment done by (Anderson, P., 2005) in order to establish an equation for the dynamic discharge coefficient. A third-order polynomial can describe above-mentioned equation as

\[
C_d(\rho_0, p_t) = -1.46 \left( \frac{\rho_0}{\rho_0} \right)^2 + 1.06 \left( \frac{\rho_0}{\rho_0} \right)^2 - 0.21 \left( \frac{\rho_0}{\rho_0} \right) + 1.01.
\]  

(1.3)

It is noteworthy to remember that the dynamic behaviors of the air which is passed through the throttle cannot be clearly represented by the present literature. As shown in equation 1.3, this discharge coefficient is now explicitly defined in our study. According to (Heywood, J.B., 1988), in a four-stroke engine, the volumetric efficiency \( \left( \frac{L}{M} \right) \) is a vital parameter. It can be described as a ratio between the real mass flow of air into the cylinder and the mass flow of air used from theoretical volume. Preferably, there are some parameters involved in the volumetric efficiency. In an ideal world, the \( \eta_v \) is described based on the following variables: air mass, engine speed and so on (Gnanam, S.R., et al., 2006).

\[
\eta_v = (24.5.N - 3.14 \cdot 10^4) m_a^2 + (-0.167.N + 222) m_a + (8.1 \cdot 10^{-4}.N + 0.352).
\]  

(1.4)

Where \( m_a \) mass of air and can be express by following

\[
m_a = \frac{M_a p_t V_m}{R T_i}.
\]

Where \( M_a \), \( V_m \), \( T_i \) and \( N \) denoted the air molecular mass kg/kmol, manifold volume, temperature of intake manifold (Kelvin) and engine speed (rad/sec).

A. Intake manifold dynamic model:

The amount of air mass which is passed into the cylinder is greatly impacted by an important parameter which is the intake manifold density, \( \rho_{at} \). This parameter can be expressed by following

\[
\rho_{at} = \frac{\rho_t}{RT_i}.
\]

(1.5)

Where

\[
\rho_t = \frac{RT_i}{V_m} m_a - \frac{\eta_v N T_i p_t}{120 m}.
\]

(1.6)

From Eqs. (1.2), (1.4) and (1.6), for the air flow system, the real amount of air passing into the cylinder \( \dot{m}_{ac} \) can be calculated by

\[
\dot{m}_{ac} = \eta_v N T_i p_t \frac{m_a}{2}.
\]

(1.7)

Where \( V_d \) is displacement volume \( \left( m^3 \right) \).

B. Fuel injection dynamic model:

An electromechanical device which is controlled electrically which is used for activating a solenoid valve is named fuel injector. The fuel amount which is injected inside the intake manifold is stated by

\[
\dot{m}_{fi} = \frac{k}{2} (t_{inj} - t_a).
\]

(1.8)

Since each cycle equals to two crankshaft rotation, 1/2 factor is presented in the above equation. In equation 1.8, \( k \) is a conversion factor from fuel injection time to fuel mass flow and is small constant around 0.014. Since the injected time \( t_{inj} \) is commanded by the ECU, in the equation (1.8), it is considered as a dynamic parameter. According to (Chang, C.-F., et al., 1995) a mechanical delay is represented by the solenoid response time \( (t_a) \). This delay has a small constant value of 0.41 ms. Dynamic process of fueling is started right after the fuel is injected (Chang, C.-F., et al., 1995).

The simplest fuel-film-flow model can be described by

\[
\dot{m}_{ff} = \frac{1}{\tau_f} \left[ (-X_f \dot{m}_{ff} + X_f \dot{m}_{fi} \right] \quad \text{(1.9)}
\]

\[
\dot{m}_{fv} = (1 - X_f) \dot{m}_{fi} \quad \text{(1.10)}
\]

\[
\dot{m}_{fc} = \dot{m}_{fv} + \dot{m}_{ff} \quad \text{(1.11)}
\]

The dynamics of fuel flow in manifold injection engine is represented in this model. It is considered that, in the intake manifold, the fuel evaporation occurs. There are two parameters involved in this model. One is the constant of time for fuel evaporation \( \tau_f \) and the other one is the fuel proportion which is placed on the intake
manifold or near to the intake valves $X_f$. They are point dependent parameters and stated based on the different conditions of the model as follows:

$$\tau_f(p_i, N) = 1.35(-0.672N + 1.68)(p_i - 0.825)^2 + (-0.06N + 0.15) + 0.56$$

Combining Equation 1.9, 1.10 and 1.11 yields:

$$\dot{m}_{fc} = \frac{1+(1-S)\tau_f}{1+\tau_f} \dot{m}_{fi}$$

C. Crankshaft dynamic model:

The system output is the engine torque so that it is requires calculating the velocity of crankshaft. The crankshaft system operation is based on the relationship between two parameters, engine speed and pressure (Heywood, J.B., 1988). Velocity of the crankshaft is possible to be computed through an integral from the consequent torque divided by inertia of the engine and total torque can be calculated by multiply of Velocity of the crankshaft to inertia of the engine (Ahmed, Q. and A.I. Bhatti, 2011), and can be expressed by

$$\dot{n} = \frac{1}{I}(T_c - T_f - T_p - T_l)$$

$$T_{total} = \dot{n} \cdot I$$

Where

$\dot{n}$: Speed of engine, $T_c$: combustion torque after Sparks, $T_f$: friction torque while the piston goes up and down, $T_p$: pumping torque, $T_l$ torque load and $I_{eng}$: engine inertia.

Actually, with utilizing the term of pressure or the term of mean effective pressure, the velocity of crankshaft can be expressed as

$$\dot{n} = \frac{1}{I}[\frac{V_d(\text{imep}-\text{tfmep})}{4\pi} - T_l]$$

Where $\text{imep}$ is the net indicated mean effective pressure (IMEP) for a four-stroke engine without a supercharger, and it is a consequence of subtraction between the gross IMEP ($g\text{mep}$) and pumping IMEP ($p\text{mep}$).

$$\text{imep} = g\text{mep} - p\text{mep}$$

And the $\text{imep}$ can then be computed by

$$\text{imep} = \frac{120\eta_f \dot{m}_f Q_{HV \text{min}} \text{min}(\lambda, 1)}{V_d N}$$

Where $\eta_f$ efficiency of the fuel conversion is, $Q_{HV}$ is the fuel low heat value and $\dot{m}_f$ is the flow rate of fuel mass. It is the fuel type that strongly defines the two first parameters. $\text{tfmep}$ Is the friction MEP and can be calculated by the summation of the mechanical friction MEP ($\text{mfmep}$) and accessory mean effective pressure (afmep). The $\text{tfmep}$ is expressed here as follows:

$$\text{tfmep} = \text{afmep} + \text{mfmep}$$

There are effects and variables used for calculating $\text{tfmep}$ as follows: Mechanical friction MEP is relative to the friction of friction of journal-bearing, friction of rings and piston, as well as the friction of valve train. The engine oil viscosity and purity can directly impact the friction of journal-bearing. The created scratch between ring pack and piston skirt with the inside wall of the cylinder caused the friction of piston and rings. Three different parts are involved in creating the friction of valve train. They are valve components, pivot rockers and overhead camshaft. Water and oil pump together no charging alternator friction are involved in the accessory MEP. When these effects are combined, it is possible to write the friction MEP base on the speed of engine speed as follows:

$$\text{tfmep} = 0.97 + 0.15 \left(\frac{N}{1000}\right) + 0.05 \left(\frac{N}{1000}\right)^2$$

Finally, we work out both mass flow rate of air and flow rate of fuel. Therefore, the equation for speed of the engine is expressed as follows:

$$N = \left[60 \left(\int_0^n \dot{n} \, dt\right)\right] / 2\pi$$

D. Air to Fuel ratio:

In SI engines, the air-to-fuel ratio is measured by a device known as an oxygen sensor, or sometimes named a lambda sensor. The sensor is located in the exhaust manifold and its main purpose is to determine how far away from stoichiometry the air-fuel mixture is. Air to fuel actual ratio within the cylinder is possible to be described as:
\[ \text{A/F} = \frac{\text{m}_{\text{ac}}}{\text{m}_{\text{fc}}} \] (1.22)

And the lambda input could be considered concurrently as:

\[ \lambda_i = \frac{\text{A/F}}{\lambda_{\text{sto}} \text{A/F}} \] (1.23)

E.1 Time delays:

Moreover, it is necessary to consider the time delays of injection systems. This time delay may be caused by three reasons for injection systems. Firstly, the delay of the engine between two cycle of fuel injection and the expulsion from the exhaust valves. Secondly, the time that gases in the exhaust need to touch the oxygen sensor as well as the output delay of sensor. It is demonstrated that the effect of the speed of engine has a higher impact on these delays in comparison to the pressure of manifold. Hence, the delays of injection systems can be represented by the following equation (Gnanam, S.R., et al., 2006)

\[ \lambda_m = \frac{\lambda e^{-td}}{ts + 1} \] (1.24)

Note that \( s \) is a complicated variable which is written in frequency domain fashion, \( t_s \) equals to oxygen sensor time constant, and \( t_d \) is the delay of transport between the oxygen sensor of the exhaust gas and the injection point.

\[ t_d = \frac{120}{N} \]

Finally based on the Figure 7 and above equations the proposed engine model could be simulated.

III. Summary of AFR control methods:

There are so many studies performed researches related to the AFR of control systems. Intelligent control modern control theories are recently advanced so that preciseness of the AFR control and robustness of the system are improved. Several different control techniques have been employed for controlling air to fuel ratio. They are Fuzzy control, PID, predictive control, switching frequency control, sliding mode control, etc. Some techniques of control are based on Neural Network.

Feasibility and effectiveness of adaptive NN method has been investigated by (Wang, S. and D. Yu, 2008) based on predictive control for AFR of SI engines.

Several different researchers including, (Kovalenko, O., et al., 2004; Muske, K.R. and J.C.P. Jones, 2008; Rupp, D., et al., 2007; Rupp, D., 2009; Kovalenko, O., et al., 2004) proposed successful adaptive control methods. One of the recent developed techniques is the adaptive posicast control. Plants with large time delays and parametric uncertainties are considered as the target of this approach (Niculescu, S.-I. and A.M. Annaswamy, 2003; Yildiz, Y., et al., 2010).

(Gao, S., et al., 2011) Presented a new NN method for engine control. This method employed an adaptive critic designs in order to provide self-learning control of Spark-Ignition engines. NN learning is used in this design by means of approximate dynamic programming. Based on the control outcomes, it is verified that a strong alternative technique for engine control is the controller.

In order to estimate AFR, a method for identifying an engine dynamic system is investigated by (Alippi, C., et al., 1998) by using neural networks. A single hidden layer with 16 hidden neurons is included in this identified neural model. For predictive control of AFR, it is a reference model.

A/F ratio control specific application which is based on the observer measurements in the intake manifold has been conducted by (Benninger, N. and G. Plapp, 1991).

A development of a feedback linearization strong controller has been done by (Guzzella, L., 1995). Lack of robustness in contradiction of noisy measurements as well as the time delay which is induced by the lambda sensor are the main disadvantages of their method.

An engine modeling with diagonal recurrent neural networks is studied by (Zhai, Y.-J., et al., 2010). They proposed this type of model-based predictive control (MPC) for air to fuel ratio. Their model was made adaptive on-line for dealing with engine time changing dynamics. Hence, there was a great improvement in control robustness performance.

A model-based control for the air to fuel ratio is presented by (Tomforde, M., et al., 2011). Algorithm of linear quadratic regulators is used for designing the controller. In addition, the oxygen storage states model is used for the prediction of the desired values. A good quality of control is exposed by the measurements. There was a reduction of almost 45% in the absolute value integral of AFR deviations in comparison to a general PI-controller.

By using predictive controller in which the adaptive expand particle swarm optimization is the base, AFR error can be maintained on 2% by (Choi, S.-B., et al., 1994).
An acceptable performance was reported by (Muske, K.R. and J.C.P. Jones, 2008) within various types of SI engine operation air fuel ratio control by means of an analytical MPC for SI engine air to fuel ratio. For the Spark-Ignition engines the air fuel ratio control is proposed by (Abdi, J., et al., 2011). They utilized a model predictive control method as well as a reference model which is based on neural network. Actually, this method robustness versus the parameters error estimation is significantly high.

The first fuzzy logic controller (FLC) has been developed by (Mamdani, E.H. and S. Assilian, 1975). The FLC was successfully utilized for controlling a steam engine plant in laboratory scale. In fact, the first fuzzy controller shown in (Mamdani, E.H. and S. Assilian, 1975) was equivalent to two-input fuzzy PI (or PI-like) controllers where error and error change, were used as the inputs for the inference.

The most popular and strong method of fuzzy reasoning was introduced by the new work of (Mamdani, E.H. and S. Assilian, 1975). It is named Zadeh–Mamdani min–max gravity reasoning. Similarly, there are several different detailed theoretical and analytical studies have been performed about this structure (Cook, J. and B.K. Powell, 1988).


(Choi, S.-B., et al., 1994) researched about observer-based SMC and Gaussian NN air to fuel ratio controller. Compared to the conventional sliding mode control it is shorter settling. Moreover, it is possible to avoid the time consuming gain tuning process by using the new controller.

Table 1 summarized the reviewed engine AFR controller methods which it showed PID and fuzzy controller widely used to control engine AFR and the AFR results are in acceptable range. Based on table 1 in this paper a PID controller with technique of Z/N combined with a fuzzy controller to on line tuning the Z/N PID controller has been used which it so called Z/N PID fuzzy controller.

### Table: Summarized the Reviewed AFR Control Models

<table>
<thead>
<tr>
<th>Model</th>
<th>Robustness</th>
<th>Settling Time</th>
<th>Capability of Combination</th>
<th>Complexity of Math. Equation</th>
<th>Reduction of Target Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMC</td>
<td>F</td>
<td>H</td>
<td>√</td>
<td>H</td>
<td>F</td>
</tr>
<tr>
<td>(Pieper and Mehrotra (1999) and Souder and Hedrick (2004))</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>NN</td>
<td>F</td>
<td>H</td>
<td>√</td>
<td>H</td>
<td>G</td>
</tr>
<tr>
<td>Alippi and Saraswati and Chand (2010)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MPC</td>
<td>L</td>
<td>L</td>
<td>√</td>
<td>H</td>
<td>VG</td>
</tr>
<tr>
<td>Muske and Jones (2006)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MPC</td>
<td>VG</td>
<td>L</td>
<td>√</td>
<td>H</td>
<td>G</td>
</tr>
<tr>
<td>(Abdi et al. (2011))</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>PID</td>
<td>VG</td>
<td>VL</td>
<td>√</td>
<td>VL</td>
<td>O</td>
</tr>
<tr>
<td>(Gao et al. (2011) and Abdi et al. (2011))</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>FUZZY</td>
<td>G</td>
<td>L</td>
<td>√</td>
<td>VL</td>
<td>VG</td>
</tr>
<tr>
<td>Mamdani and Assilian (1975)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>F= Fair</td>
<td>G=Good</td>
<td>VG=Very Good</td>
<td>L=Low</td>
<td>VL=Very Low</td>
</tr>
</tbody>
</table>

### III.1 Z/N PID and Z/N PID Fuzzy Controller:

Figure 8 shows the structure of Z/N PID fuzzy controller which briefly has been explained in this section.
A. Equation of PID Controller:

In a PID controller, the transfer function has the subsequent form:

$$G_c(S) = K_p + \frac{K_i}{S} + K_dS$$

where $K_p$, $K_i$ and $K_d$ are the proportional, integral, and derivative gains, respectively. One more helpful equivalent structure of the PID controller is

$$G_c(S) = K_p \left(1 + \frac{1}{T_i s} + T_d s\right)$$

where $T_i = K_p / K_i$ and $T_d = K_d / K_p$. $T_i$ and $T_d$ are identified as constants of the integral and derivative time, correspondingly. The discrete-time corresponding expression for PID control which is utilized in our study is identified as

$$u(K) = K_p e(k) + K_i T_s \sum_{i=1}^{K_d} e(i) + \frac{K_d}{T_d} \Delta e(K)$$

where $u(K)$ is the signal of control, $e(k)$ is the error between the reference and the plant output (14.7-AFR_m), $T_s$ is the sample period for the controller, and $\Delta e(K) \triangleq e(k) - e(K - 1)$

$K_p$, $K_i$, and $K_d$ or $K_{th}$ $T_d$ and $T_i$ are the PID controller parameters that can be retouched in order to generate different curves of response from a particular process. It is vital to find the controller’s optimum adjustments for a particular process. $K_{th}$ $K_p$ and $K_i$ or $K_{th}$ $T_d$ and $T_i$ gains required to be tuned in this study for achieving an acceptable performance level. This is our main reason for using fuzzy logic control for tuning the proportional, integral, and derivative gains.

B. Fuzzy logic control:

Due to in fact that, the complicated nature of engine system, it is not easy to achieve the desired control effect by using conventional PID controllers. Recently, new control methods based on fuzzy logic have been quickly improved. They have been attracted by many researchers in the field of control industry.

Mapping behavior can be manually impacted and known with a feasible way provided by the Fuzzy logic (FL). Generally, simple rules are used in fuzzy logic for explaining the device of interest as opposed to analytical equations. This makes it a user friendly method to be implemented. Based on the Fuzzy logic advantages including its speed and validity, this technique is one of the best ways of solving for modeling of a system and control. There are three main basics defined for An FIS. Stage of fuzzification, rule base, and stage of defuzzification. For transforming supposed input variable values into a fuzzy membership values, Fuzzification stage is implemented. Then, by utilizing “if-then” statements, the rule base processes these membership values. The principles outcomes are then summed and defuzzified into a value of crisp analogue output. Now it is possible to easily understand the results of changes in a FIS parameters and the model calibration will be facilitated (Lee, S.H., et al., 2007).

The device inputs, for instance: $e(K)$=14.7-AFR_m’ and ‘$\Delta e(K)$’ are so called linguistic variables, while ‘high’ and ‘very high’ are linguistic values which are characterized by the membership function (MF). After assessing the rules, the fuzzy membership values will be transformed into a value of crisp output by the defuzzification, for instance, depth of penetration.in our research they are PID controller gains such as $k_p$, $k_d$, $k_i$.

C. Z/N PID Fuzzy Structure:

Based on the full analysis of Z/N PID Fuzzy performance, the speed of response and suppressed overshoot in the composite controllers can be enhanced, eliminating the steady state error. It is decided to use gainscheduling Z/N PID Fuzzy controller in this study. Structure of a Z/N PID Fuzzy control in the SIMULINK is illustrated in Figure 9.
High precision of the PID control and flexibility and adaptability of fuzzy control are inherited in the Z/N PID Fuzzy control. With these two inherited characteristics, the fuzzy control of dynamic and static performance as well as the single PID control are improved. There is also an excellent control effect on the time-varying and nonlinear complicated systems (Gao, S., et al., 2011).

D. Simulation of PID and Z/N PID Fuzzy Using Simulink:

D.1 PID Simulation by Ziegler-Nichols method:

PID controller must be designed for making the tuning easy and collaborating with the toolbox. Figure10 is illustrating a PID controller in which Gain, Integrator and Derivative blocks are used for comprehending the PID control.

After the simulation finished we have to set PID gains to tune the simulation. By Double-click on PID block we can see this windows as shown in fig 11.

As mentioned we have set the PID gains as Ziegler-Nichols (Z/N) method based on table 2 as shown in figure 12.

Table 2: Ziegler-Nichols (Z/N) method
Control Type | $K_p$ | $K_i$ | $K_d$
--- | --- | --- | ---
P | $0.5K_u$ | - | -
PI | $0.45K_u$ | $1.2K_u/T_u$ | -
PD | $0.8K_u$ | - | $2K_u/T_u$
Classic PID | $0.6K_u$ | $2K_u/T_u$ | $K_uT_u/8$

**Fig. 12:** PID Controller Based on Ziegler-Nichols (Z/N) method

**D.2 Z/N PID Fuzzy Simulation:**

**D.2.1 Fuzzy logic Design by MATLAB Fuzzy Tool Box:**

Based on mentioned we have to design fuzzy logic in matlab. At first by writing in command windows fuzzy we will see the fuzzy tools box as shown in Figure 13.

**Fig. 13:** fuzzy logic editor windows

As shown in Figure 13 we add two inputs: $e(K)$ and $\Delta e(K)$ and three out puts $K_p', K_d'$ and $\propto$. In the next part we have to define the membership function based as shown in Figure 14.
As shown in Figure 14 we used 7 triangular functions with the range [-1 1] to define the $e(K)$ and $\Delta e(K)$ membership function and for define the membership function of $K_p, K'_p$ we used 2 Gaussian functions with range of [0 1] as shown in Figure 15.

For define the $\propto$ we used the 4 triangular function instead of singleton function with the range (Mamdani, E.H. and S.A ssilian, 1975) as shown in Figure 16.
After the defined membership function we defined the 49 rules based on table 3 to 5 as shown in Figure 17.

Table 3: Fuzzy-tuning role for $K_p$ (Zhao, Z.-Y., 1993)

<table>
<thead>
<tr>
<th>$\Delta \varepsilon(k)$</th>
<th>NB</th>
<th>NM</th>
<th>NS</th>
<th>ZO</th>
<th>PS</th>
<th>PM</th>
<th>PB</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\varepsilon(k)$</td>
<td>NB</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
</tr>
<tr>
<td></td>
<td>NM</td>
<td>S</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>S</td>
<td>S</td>
</tr>
<tr>
<td></td>
<td>NS</td>
<td>S</td>
<td>B</td>
<td>B</td>
<td>S</td>
<td>S</td>
<td>S</td>
</tr>
<tr>
<td></td>
<td>ZO</td>
<td>S</td>
<td>S</td>
<td>B</td>
<td>S</td>
<td>S</td>
<td>S</td>
</tr>
<tr>
<td></td>
<td>PS</td>
<td>S</td>
<td>S</td>
<td>B</td>
<td>B</td>
<td>S</td>
<td>S</td>
</tr>
<tr>
<td></td>
<td>PM</td>
<td>S</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>S</td>
</tr>
<tr>
<td></td>
<td>PB</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
</tr>
</tbody>
</table>

Table 4: Fuzzy tuning role for $K_d$ (Zhao, Z.-Y., 1993)

<table>
<thead>
<tr>
<th>$\Delta \varepsilon(k)$</th>
<th>NB</th>
<th>NM</th>
<th>NS</th>
<th>ZO</th>
<th>PS</th>
<th>PM</th>
<th>PB</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\varepsilon(k)$</td>
<td>NB</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
</tr>
<tr>
<td></td>
<td>NM</td>
<td>B</td>
<td>B</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>B</td>
</tr>
<tr>
<td></td>
<td>NS</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>S</td>
<td>B</td>
<td>B</td>
</tr>
<tr>
<td></td>
<td>ZO</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>B</td>
</tr>
<tr>
<td></td>
<td>PS</td>
<td>B</td>
<td>B</td>
<td>S</td>
<td>B</td>
<td>B</td>
<td>B</td>
</tr>
<tr>
<td></td>
<td>PM</td>
<td>B</td>
<td>B</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>B</td>
</tr>
<tr>
<td></td>
<td>PB</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
</tr>
</tbody>
</table>

Table 5: Fuzzy tuning role for $\propto$ (Zhao, Z.-Y., 1993)

<table>
<thead>
<tr>
<th>$\Delta \varepsilon(k)$</th>
<th>NB</th>
<th>NM</th>
<th>NS</th>
<th>ZO</th>
<th>PS</th>
<th>PM</th>
<th>PB</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\varepsilon(k)$</td>
<td>NB</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>NM</td>
<td>3</td>
<td>3</td>
<td>2</td>
<td>2</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>NS</td>
<td>4</td>
<td>3</td>
<td>3</td>
<td>2</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>ZO</td>
<td>5</td>
<td>4</td>
<td>3</td>
<td>3</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>PS</td>
<td>4</td>
<td>3</td>
<td>2</td>
<td>3</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>PM</td>
<td>3</td>
<td>3</td>
<td>2</td>
<td>2</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>PB</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
</tbody>
</table>
Fig. 17: 49 rules in Matlab
As shown in Figure 18 we can see the surface of rules, and it shows that, the surface has symmetry on the colors because of the true defined the rules in MATLAB.

Fig. 18: Rules surface viewer in Matlab
After design fuzzy model as above mentioned we must to apply this model to PID control to online adjusting the PID gains.

Conclusion And Future Work:
In this study we have reviewed engine MVEM models and presented an engine simulation structure which its include all of dynamics parts of real engine while unnecessary complexity have been removed. For reach this goal mathematical equations of each dynamic parts have been presented then based on the presented equations an engine simulation structure could be simulate by MATLAB/SIMULINK Function blocks. In the second part of this paper an AFR control system for the engine based on the reviewed AFR controller methods has been presented in which Z/N PID and Fuzzy controller are used. Therefore, a PID control by using Z/N technique is presented. Then a fuzzy control based on the Zhao 1999 rule tables have been presented for on line tuning of Z/N PID controller gains. So, propose control method should be evaluated at first and then could use for controlling of any kind of SI engines.
Developed engine model is able for add or remove some of dynamic parts to simulate and test for different engines. For example this model is able to add EGR model for further evaluation and tests. On the other hand the control model also is able to combine with any other controller methods to decrease the delay or any other control performances.

For applying mechanical model at first the proposed mechanical model should be validate under many different conditions such as idle or running to simulate the stable and transient conditions, after then its model could apply for controlling any kind of SI engines.

Acknowledgements

The author would like to a great appreciation to UNIVERSITI TEKNOLOGI MALAYSIA for supporting this research.

References


