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A Study on Combustion Modelling of Marine Engines Concerning the Cylindrical Pressure

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ABSTRACT

Background: This research expands the combustion model of SI engines using the mass fraction burned (MFB) based on the Wiebe function approach. **Objective:** The goal of this study is to build a combustion model specially cylindrical pressure model suitable for control oriented model. **Results:** The cylinder pressure signal can be constructed as a function of crank angle over an engine operational map. Then the cylindrical pressure model parameters have been calibrated with a s-curve matching techniques with experimental results. **Conclusion:** In this research we explained the most significant dynamics modelling of SI engine by Wiebe function such as cylindrical pressure which is very important to designing SI engine controller.

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INTRODUCTION

The SI engine is designed to make power from the energy that is contained in its fuel. Specifically, its fuel contains chemical energy and together with air, this mixture is burned to output mechanical power. There are several types of fuels that can be used in SI engines such as petroleum, bio-fuels, and hydrogen. Modeling of a whole SI engine is an extremely significant and difficult procedure since nonlinear, multi inputs-multi outputs and time variant engines. Precise modeling aims to keep expansion expenses of real engines and decreasing damaging risk of an engine during controller designs validation. However, a small model can be designed, implemented and validated and then can be used for a bigger problem. (Boiko, *et al* 2007). SI engines dynamic modeling is deployed to explain the performance of this system aligned with model based controller designing and also for simulation.

The association among nonlinear output formulation to electrical or mechanical source and the meticulous dynamic impacts to system behavior can be described using dynamic modeling.

A four cylinder SI engine has been selected for the case study. Experimental test will complete with E-dynamometer, suitable sensors and data gaining equipment is employed for the collection of data set for the calibration of the dynamics modeling.

Engine Operating Cycle:

It is common for an internal combustion engine that a piston goes up and down in a cylinder transferring power using a connecting rod connected to a crank shaft. Engine cycle is known as frequent piston motion and crank shaft rotation when fuel and air go in and out from the cylinder using the Ingestion and tire out valves. Otto engine developed by Nicolaus A. Otto in 1876 is the first and best internal combustion machine (Priyanto, A., & Nekooei, M.J. 2014). Otto create a single engine cycle containing of four piston strokes. These strokes are:

1. Intake stroke
2. Compression stroke
3. Expansion stroke
4. Exhaust stroke

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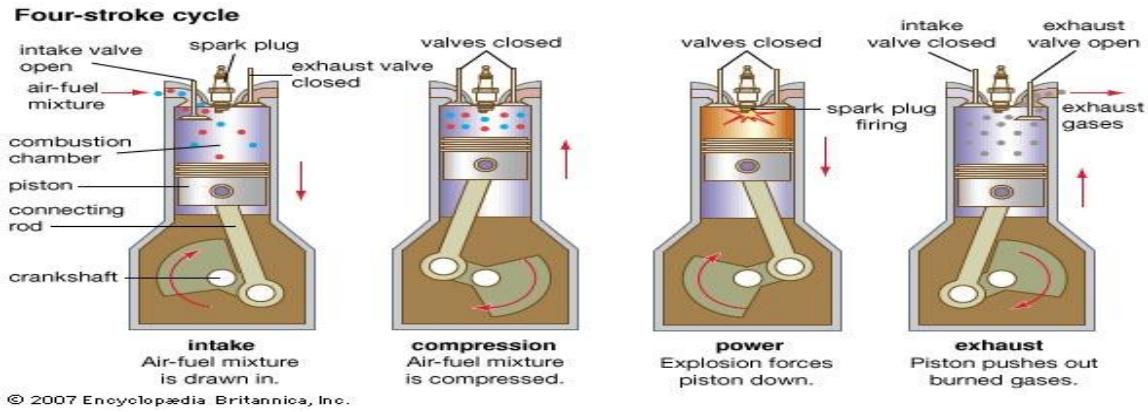


Fig. 1: The four stroke engine cycle. (Boiko, *et al* 2007).

Through the intake stroke, the piston begins at top-dead-center (TDC) and ends at bottom dead-center (BDC). An air and gasoline mixture enters the cylinder through the intake valve and in some instances this valve opens slightly prior to the intake stroke begins to permit more air-fuel mixture to the cylinder (Priyanto, A., & Nekooei, M.J. 2014). Throughout the compression stroke, the intake and exhaust valves are closed and the mixture is compressed to a really small fraction of its initial volume. The compressed mixture is then ignited by a spark evoking the pressure to go up very rapidly. through the expansion stroke, the piston begins at TDC. Because of the high pressure and temperature gases in the cylinder, the piston has become pushed down, evoking the crank to rotate. Whilst the piston approaches BDC the exhaust valve opens. through the exhaust stroke, the burned gases exit the cylinder as a result of high cylinder pressure and low exhaust pressure and also as a result of piston moving up towards TDC. The cycle starts again following the exhaust valve closes (Boiko, *et al*, 2007). With models for each one of these processes, a simulation of complete engine cycle could be developed and be analyzed to supply info on engine performances. These ideal models that describe characteristic of every process are proposed. Though the calculation needs information from each state as shown in Fig. 2

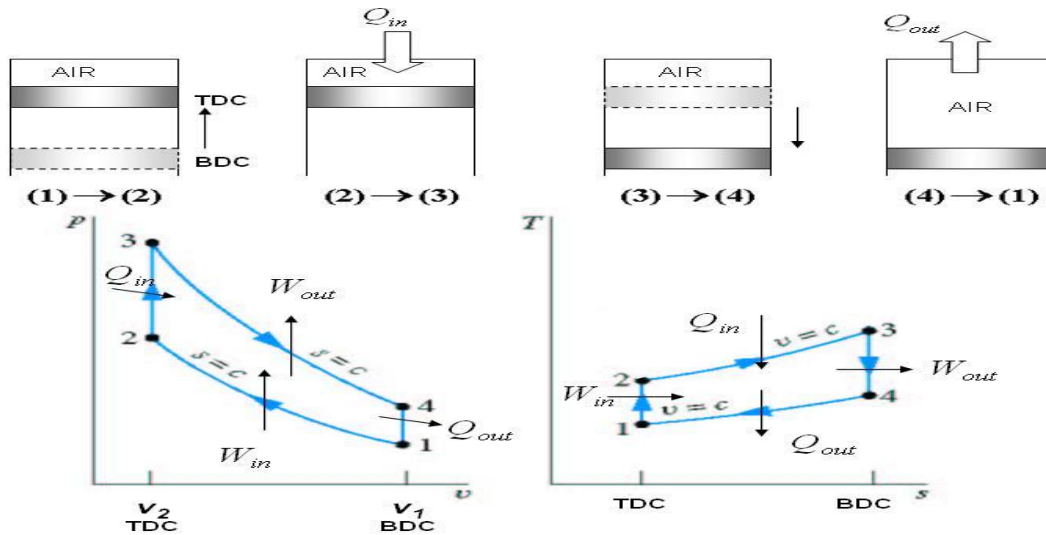


Fig. 2: Pressure-volume and Temperature-Entropy diagram of Otto cycle (Boiko, *et al* 2007).

A entire engine cycle is split into 720 crank angle degrees, where in fact the crank angle is involving the piston connecting rod at TDC and the connecting rod far from TDC. which means that the piston will go up and down in the cylinder twice during one complete engine cycle. because there are two revolutions in a single engine cycle, time duration (in seconds) of just one engine cycle is found given the rotations-per-minute (RPM). As an example, at 1500 RPM, an engine cycle lasts 80 milliseconds (ms) and at 3000 RPM an engine cycle lasts 40 ms.

Although, the Otto cycle was created many years back, it remains a commonly used engine design. As stated, the modeling of the whole process of the SI engine is really a very complicated one, which involves

modeling of thermal dynamics. This research intent is to produce an easy cylinder pressure model that can be utilized in real-time simulation for controller design and validation purposes.

Background of Engines Modelling:

The combustion models can be classified into two main categories: multi-dimensional models and zero-dimensional models. The multi-dimensional models (Priyanto, A., & Nekooei, M.J. 2014). can provide a good spatial description of the fuel spray, which is very important for exhaust gas emissions, but unfortunately they require a very high computational time which makes them unsuitable for our study. Zero-dimensional models can be also classified into 3 different categories: multi-zone models ($n > 2$), two zone models, and single zone model or empirical model.

Some single zone models published in literature takes into account the spray modeling. The best description of the spray divided into multi-zone is provided by Hiroyasu (Priyanto, A., & Nekooei, M.J. 2014). This approach divides the fuel spray into several packets, which are independent from each other. This kind of models is suitable to predict the combustion process and the pollutant formation. However, to increase the accuracy of the model, the number of spray packets must be increased, leading to a significant CPU time (Priyanto, A., & Nekooei, M.J. 2014). Although their computational time is lower compared to multi-dimensional models, it remains unsuitable for real time modeling.

(Y.C. Hsueh, *et al.*, 2009) has developed two zones model for the spray: one zone to describe the premixed combustion phase and the second zone to describe the diffused combustion phase. This approach needs a low computational time compared to the multi-zone spray approach. Due to their high computational time, the multi-zone spray modeling is not suitable to real time approach. Consequently, the empirical models (i.e; single zone model) are often used for real time models. The combustion process is then described through the well known Wiebe correlation (single or double Wiebe equations (Y. C. Hsueh, *et al.*, 2009) proposed an approach based on the Wiebe model, but instead using a single Wiebe correlation to predict the heat release rate, they coupled two mathematical equations: the first one to describe premixed combustion and the second one for the diffused combustion. In their approach, the combustion duration for both combustion phases is assumed to be a constant. (Y.C. Hsueh, *et al.*, 2009) have used a double Wiebe equations model to describe the combustion behavior of Diesel engine. Both approaches require low computational times, and gives good accuracy compared to experimental data. In the present study, we have used a Wiebe models approach to describe the cylindrical pressure model .

Combustion Process:

In creating a valid engine model of SI engines, the idea of the combustion process must certainly be understood. The combustion process is easy and it begins with fuel and air being mixed together in the intake manifold and cylinder. This air-fuel mixture is trapped inside cylinder following the intake valve(s) is closed and then gets compressed. Thereafter, the compressed mixture is combusted, usually near the end of the compression stroke, because a power discharge from the spark plug. The flame that's produced close to the spark electrode travels through the unburned air-fuel mixture and extinguishes when it hits the combustion chamber walls. This combustion process varies from engine cycle-to-cycle and also varies from cylinder-to-cylinder. The particular combustion of the air-fuel mixture begins before the conclusion of the compression stroke, extends through combustion stroke, and ends following the peak cylinder pressure occurs (Priyanto, A., & Nekooei, M. J. 2014).

Cylinder Pressure:

The pressure in the cylinder is an essential physical parameter that may be analyzed from the combustion process. The pressure in the cylinder reaches a particular point (in the lack of} combustion) since the air-fuel mixture within the cylinder is compressed. Right after the flame develops, the cylinder pressure steadily rises (in the presence of combustion), reaches a maximum point after TDC, and finally decreases through the expansion stroke once the cylinder volume increases. Enough time at that the electrical discharge from the spark plug occurs is essential to the combustion event and should be designed to happen at the peak cylinder pressure which occurs very near to top dead center. This is performed so the maximum power or torque could be obtained. Consequently, this optimum timing known as Minimal advance for the maximum Torque or MBT timing. The spark timing will often be advanced or retarded because of various operating conditions, including engine speed and load, and this can lead to reduced output torque or power.

The perfect spark timing (or MBT timing) can be determined using cylinder pressure signals and mass fraction burned (MFB) based on the cylinder pressure. recently years, two important criteria have now been found using in-cylinder pressure signals: peak cylinder pressure occurs around 15 degrees after TDC and 50% mass fraction burned occurs at 8 to 10 degrees after TDC as shown Fig.3(Nekooei, M. J., Jaswar, J., & Priyanto, A. 2013).

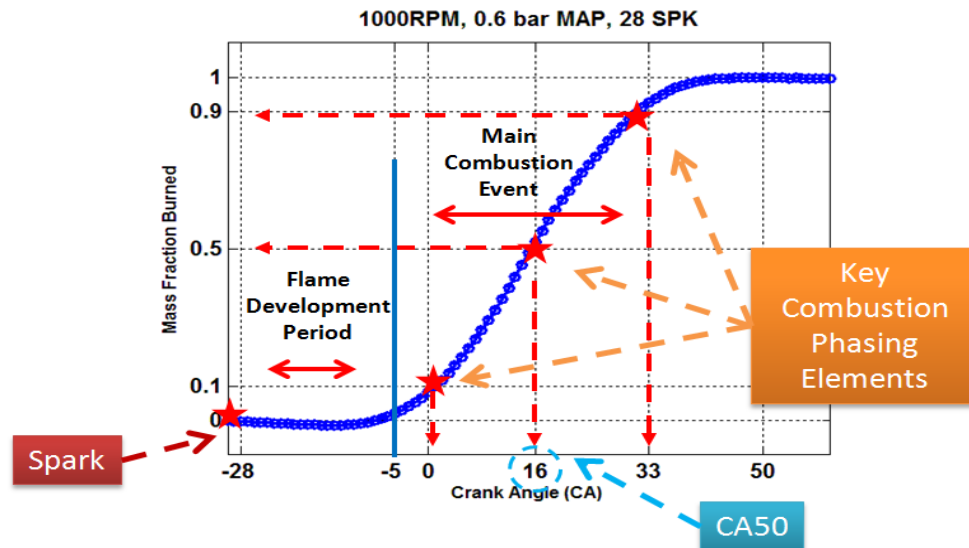


Fig. 3: Combustion phasing description. (Boiko, *et al* 2007).

The velocity and acceleration of combustion could be obtained by taking the very first and second derivatives of the MFB signal, which is often parameterized with a so called Wiebe function. Utilizing the peak cylinder pressure location, 50% MFB location, and maximum acceleration of MFB as a closed loop control criterion, the MBT spark timing could be optimized in real-time.

Cylinder Pressure Model:

Since cylinder pressure is essential to the combustion event and the engine cycle in SI engines, the development of a model that creates the cylinder pressure for every crank angle degree is necessary.

Wiebe Function:

As previously stated, peak cylinder pressure is essential in determining the perfect spark timing occurring during combustion. The optimum timing for MBT is located in accordance with the peak cylinder pressure. If the timing is advanced or retarded out of this peak pressure, the engine will produce lower output power and torque. The combustion process can be looked at as both a chemical and physical process described by the MFB in the cylinder. MFB signifies simply how much and how quickly chemical energy is released throughout the combustion cycle and could be parameterized by the Wiebe function. Thus, the Wiebe function can be used to mathematically represent the MFB vs. crank angle curve and has been recognized to model the engine combustion process perfectly (Y. Li and Q. Xu, *et al* 2010). A normal MFB vs. crank angle includes a smooth curve that's "s-shaped." Figure.4 shows a normal MFB vs. crank angle curve.

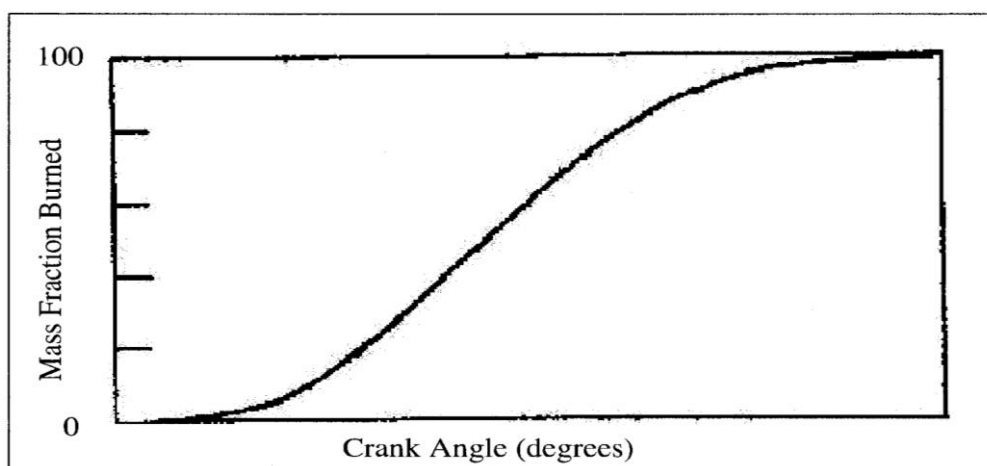


Fig. 4: A typical MFB vs. crank angle curve. (Boiko, *et al* 2007).

The Wiebe function is given by the formula,

$$x_b(\theta) = 1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right]$$

where x_b could be the mass fraction burned, θ_0 could be the start of the combustion, $\Delta\theta$ could be the combustion duration ($x_b = 0$ to $x_b = 1$), and a and m are calibration parameters (Boiko, *et al.*, 2007). Modifying the values of a and m can significantly change the form of the s-curve. The θ_0 is commonly called as spark timing or ignition timing, that will be the time (or crank angle) where in fact the air-fuel mixture is ignited. Figure .5 shows a MFB curve that has been generated by the Wiebe function.

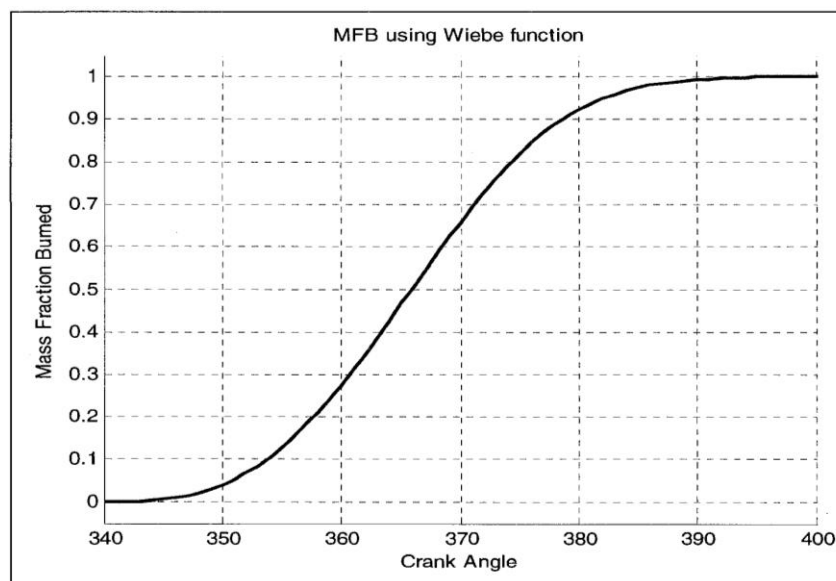


Fig. 5: MFB vs. crank angle (Wiebe function: $\theta_0 = 340$, $\Delta\theta = 50$, $a = 5$, $m = 2$) (Y. C. Hsueh, *et al.*, 2009)

Wiebe Function Calibration:

Although the Wiebe function can be used to represent MFB, it must be calibrated at various engine operational conditions to provide an accurate MFB representation. To achieve this, a Port Fuel injection gasoline engine have to tested at various combinations of engine speeds, loads, and air-to-fuel ratios.

The data from these tests allowed us to calculate the actual MFB for the various operating conditions. The speed of the engine will be measured in rotation per minute (RPM), the load, which is a percent measurement of how hard an engine is working, ranged from zero to one, and finally the λ - (Lambda), which is calculated by dividing air-to-fuel ratio by stoichiometric (14.6).

The actual MFB is then plotted vs. crank angle to obtain "s-curves" that could be represented using the Wiebe function. The θ_0 for each engine test will known and could be used in the Wiebe function equation. The other parameters a , m , and $\Delta\theta$ were unknown and had to be found. To determine these values, the actual MFB plot at a certain engine speed, load, and air-to-fuel ratio was compared to a Wiebe function plot for that operating condition. The correct values for a , m , and $\Delta\theta$ were found by matching the actual MFB curve with the Wiebe function plot, which is a trial and error process done using Matlab.

Once all the parameters for the Wiebe function had to determined for the engine operating conditions, the MFB will known for these engine operating conditions. Thus, by giving the engine speed, load, and air-to-fuel ratio, the MFB (x_b) can be found via the Wiebe function.

Cylinder Pressure Model Simulation:

After locating the equation for the cylinder pressure that's given above, the implementation of the sum total cylinder pressure model in Matlab Simulink had to begun. The target of the model is to calculate the cylinder pressure at each crank angle.

The cylinder pressure model in Matlab had the ability to output the cylinder pressure at any crank angle, engine speed, engine load, A/F ratio, crank angle, and spark timing (θ_0) are used as the inputs.

More specifically, the peak value of the cylinder pressure that is calculated varies over each engine cycle due to the variance of W_b , which causes the combustions variations to be modeled very well.

Conclusion:

SI engines dynamic modeling is deployed to explain the performance of this system aligned with model based controller designing and also for simulation.

The association among nonlinear output formulation to electrical or mechanical source and the meticulous dynamic impacts to system behavior can be described using dynamic modeling.

In this research we explained the most significant dynamics modelling of SI engine by wiebe function such as cylindrical pressure which is very important to designing SI engine controller.

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