Reviewed on Combustion Modelling of Marine Spark-Ignition Engines

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ABSTRACT

This research reviewed combustion modelling of marine spark-ignition engine using mass fraction burned based on Wiebe function approach. In the paper, firstly, the cylindrical pressure is discussed and correlated function of engine operation condition. The cylinder pressure signal was constructed as a function of crank angle over an engine operational map. The parameters of the cylindrical pressure model were calibrated with s-curve matching techniques. The objectives of the research was to construct a combustion model on the cylindrical pressure suitable for control oriented model.

KEY WORDS: *SI Engine; Cylindrical Pressure; Wiebe Function.*

NOMENCLATURE

SI	Spark Ignition
TDC	Top Dead Center
BDC	Bottom Dead Center
RPM	Revolution-Per-Minute
MBT	Maximum Brake Torque
MFB	Mass Fraction Burned

1.0 INTRODUCTION

The emissions from ships engaged in international trade in the seas surrounding ports take a part in the environmental pollution. Exhausts caused by marine engines contains high level of carbon dioxides (CO_2), nitrogen oxides (NOx), sulfur dioxide (SOx), and other of particles due to the heavy fuel oil used for combustion. For example in Europe, AirClim was estimated to have been 2.3 million tonnes of sulfur dioxide and 3.3 million tonnes of nitrogen oxides a year in 2000. The standard improvement on combustion of engine is required to control exhaust emissions from the engines and evaporative emissions from fuel.

Jaswar (2002) stated that Environment in forms of air pollution due to NOx, and SOx, global warming due to CO_2, and water pollution becomes an essential issue on societies' point of view. The governments in industrialized and developing countries consider not only economical issue, but also an environmental issue. Antonić et.al (2011) reported that The Kyoto Protocol (1997) has been a turning point for the future economic and environmental policies in both industrialized and developing countries.

The impact of environmental sea conditions to the system is significant to be considered by researchers which were starting from the phases of ship construction and building to the stage of ship's control system design and implementation. The task becomes crucial for design optimization of the single propulsion system and the whole ship as in the fact that the propulsion system is a key aspect of the global behavior of all the ships. This paper reviewed combustion of SI marine engine.

2.0 SPARK-IGNITIONI MARINE ENGINE

The spark-ignition marine engines are mostly using gasoline, methanol, compressed natural gas, liquid petroleum gas, bio-

fuels, hydrogen and propane. The SI marine engine is designed to make power from the energy that is contained in its fuel. The fuel contains chemical energy and together with air, this mixture is burned to output mechanical power. 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. Boiko, *et al.*, 2007 stated that a small model can be designed, implemented and validated and then can be used for a bigger problem. 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.

2.1 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 [Nekooei, 2013]. These strokes are:, Intake stroke, Compression stroke, Expansion stroke and Exhaust stroke as shown in figure.1.
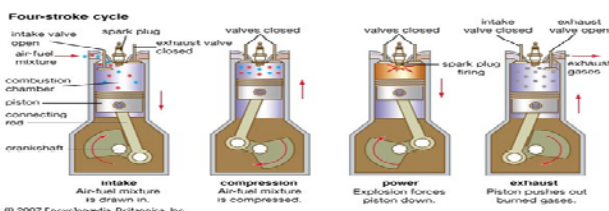

Figure 1: The four stroke engine cycle (Nekooei, 2013).

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 [Xiao, 2013].

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 [B. Siciliano, 2008].

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
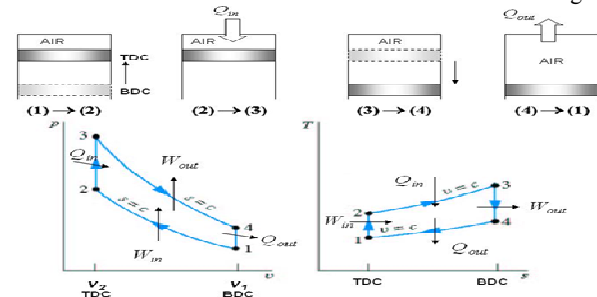

Figure 2: Pressure-volume and Temperature-Entropy diagram of Otto cycle [B. Siciliano, 2008]

An 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 revolution-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.

2.2 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 is known as Minimal advance for the maximum brake torque (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 reduce output torque or power.

The prefect 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 [Xiao, 2013].
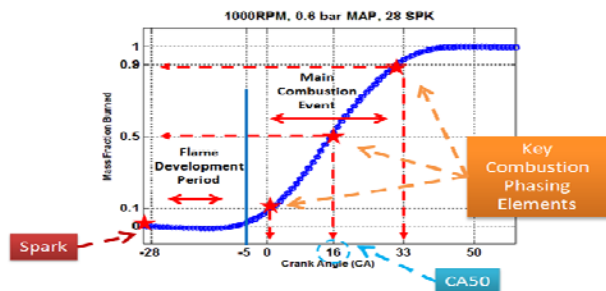

Figure .3: Combustion phasing description [Xiao, 2013]

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 [Nekooei, 2013]. 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.

3.0 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.

3.1 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 [Hsueh, et al., 2009]. A normal MFB vs. crank angle includes a smooth curve that's **"s-**shaped**."** Figure.4 shows a normal MFB vs. crank angle curve.
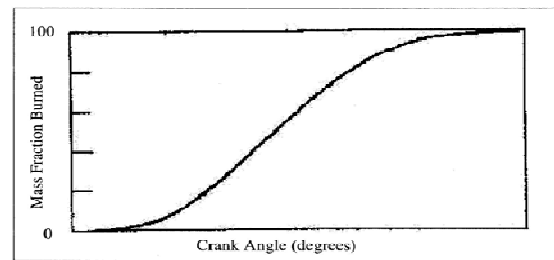

Figure 4: A typical MFB vs. crank angle curve [Hsueh, et al., 2009]

The Wiebe function is given by the formula,

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

Where x_b is the mass fraction burned, θ_0 is the start of the combustion, $\Delta\theta$ is the combustion duration ($x_b = 0$ to $x_b = 1$), and a and m are calibration parameters [B. Siciliano, 2008]. 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.
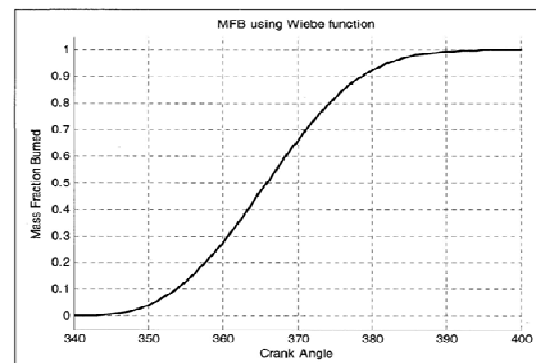

Figure 5: MFB vs. crank angle (Wiebe function: $\theta_0 = 340$, $\Delta\theta = 50$, $a = 5$, $m = 2$) [B. Siciliano, 2008]

3.2 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 has to be 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.

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 determine 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.

3.3 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.

4.0 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 marine engine by wiebe function such as cylindrical pressure which is very important to designing SI engine controller.

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