CAE Approach in Combustion Modeling on a Classic Internal Combustion Engine and Ignition Timing Modeling

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Abstract. The homologation process of the internal combustion engines (ICE) is becoming more difficult with the passing of time. Coupled with the need for massive investments in new technologies, such as hybrid or hydrogen powertrains or 100% electrical vehicles, the economy in the traditional automotive manufacturing is stretched to its limits. There is stringent need to limit as much as possible the development cost of the new ICEs set for the Euro7 emissions standards. The last 15 years have seen a giant leap forward in computer-aided design (CAD) and computer-aided engineering (CAE) in the imagining and validation of different structures. This software can now reproduce with outstanding precision the aeraulic phenomena in the air intake, in the combustion chamber and in the exhaust of ICEs. The aim of this paper is to present a general overview on system modeling in ICEs, with an emphasis on the use of predictive and non-predictive combustions models. Thus, a special attention was given to the SITurb predictive combustion method, which was employed to build a virtual engine. By modifying the models' parameters, the spark timing was corelated with the values issued from the engine test bench. Some of the tuning activities usually done on the engine test bench could be done on the SITurb model, with similar results to the tests done on physical test supports. Using this approach could lead to economic advantages, in terms of time and material resources employed in the development of the ICEs.

Keywords: Simulation · SITurb model · Spark timing

1 General Information

1.1 Internal Combustion Engines

Internal combustion engines are currently employed in all human activities. Diesel (CI – Compression Ignition engine) and gasoline (SI – Spark Ignition engine) engines are fundamentally different. The SI engine needs a spark to ignite the fuel-air mixture. The combustion is homogenous and ideally takes place at constant volume (Fig. 1.a). On the other hand, in CI engines, the fuel is injected late in the compression stroke, which mixes fuel with air. Hence the combustion is heterogeneous and takes place at constant pressure (Fig. 1.b) [1].

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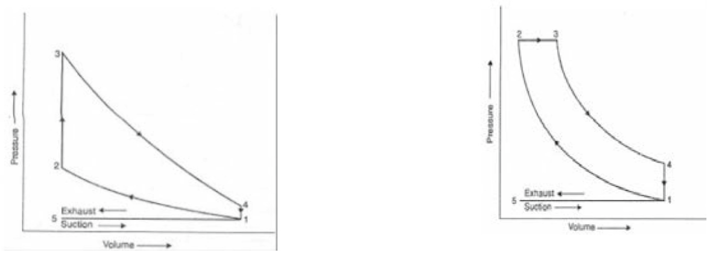

a) SI b) CI

Fig. 1. Most popular internal combustion engines [2]

Theoretically SI engines have a higher thermal efficiency compared to CI engines for a given compression ratio, as shown in Eq. (1) and (2) [3]. However, the SI engine is sensible at the Knock phenomenon, thus limiting the compression ratio and, subsequently, the efficiency. The CI engine is less sensible to Knock and therefore it runs at a significant higher compression ratio, which improves its efficiency [4].

$$\eta_{th} = 1 - \frac{1}{R_c^{\gamma - 1}} \tag{1}$$

$$\eta_{th} = 1 - \frac{R_c^{1-\gamma}\left(R_c^{\gamma} - 1\right)}{\gamma(R_c - 1)} \tag{2}$$

Whatever the engine (CI or SI), the combustion modeling using modern approaches is the key element in reducing the costs with the technical definition choice, as well as with component improvement or engine tuning.

1.2 Computer Aided Engineering

The computer-aided engineering is a software-based tool used to simulate the impact of different conditions on the design of a product/system or to mimic structural loads and constraints. The CAE software is used to analyze and improve the designs created with CAD software. A short classification of CAE tools includes finite element analysis (FEA), computational fluid dynamics (CFD) and multi-disciplinary design optimization (MDO). Several design iterations could be done without the use of physical prototypes. This beneficially impacts both the time and money used for product development and leads to design to quality [5].

2 System Modeling

System modeling, commonly referred to as 0D modeling, regroups many models that are typically designed from empirical or semi-empirical approaches. The software is designed to substitute the hardware, but quality is a key aspect of the process. The physical tests will still be used for machine learning and model validation. The complexity of an

0D model can vary from a simple mathematical model to a phenomenological model, including the physics of phenomena through reduced state or transport equations.

The 0D modeling can evaluate and analyze the performance of a complex system and at the same time, it can control its behavior through the ‘Model Based Design’ approach. In the automotive industry, the practicality of this kind of tool may vary depending on the systems that need to be modeled. For example, in the powertrain division, the pollutants, air filling or the gas flow in the intake/exhaust could be modeled using the 0D approach. With the evolution of the computer processing from the last decade, the simulation time has gone down and is now comparable with the time needed to perform real tests. Fast processing, low budget, repeatability, and adaptability of the 0D models explain the hype around the use of this type of development in the automotive industry.

The principle of the system modeling is to break down the macro-system into smaller subsystems. Each constitutive element of the system should be represented as faithfully as possible according to the compromise between the simulation time/precision requested. The second phase consists in tuning and setup of each submodel. Then, the subsystems will be assembled into the bigger frame, taking into consideration all possible interactions. Calibration should be done by validation of each subsystem individually, as well as the whole system assembly.

System modeling is therefore a particular special tool for the automotive industry, as it includes many disciplines. The product of this industry - the automobile - must face new kinds of powertrains, new legislation restrictions, increasing product diversity on the market and interconnections both between all the control functions of the vehicle and between itself and the infrastructure. 0D modelling is already used in the first part of the product development, in choosing the best technical solution to meet up the specifications, but it can now be employed even further, in cutting down development costs and shorten the time between the concept freeze and the selling of the product [6].

2.1 Combustion Modeling

There are two types of combustion modelling:

– Predictive combustion models;
– Non-predictive combustion models.

The 0D combustion model presented in the chapter above is a non-predictive (or fixed burn rate) combustion model, thus making it impossible to evaluate the benefits of using different injection patters, VVT, EGR amount etc.

To benefit from the state-of-the-art technologies, CAE developers have created preventive combustion models which can predict the combustion rate based on the in-cylinder conditions. Once again, the quality of the results depends on the calibration made starting from the real test results done on the engine test bench [7].

The choice between predictive and the non-predictive combustion models must be made according to the project needs and the time constraints.

Multi zone models are distinguished from 0D models by inclusion of certain geometrical parameters in the basic thermodynamic approach. This usually involves the radius of a thin interface (the flame) separating burned from unburned gases, resulting in a two-zone formulation [8].

2.2 The Non-predictive Combustion Model

The non-predictive combustion model imposes the burn rate as a function of the crank angle. The simulation follows this prescribed burnt rate, assuming that there is enough fuel regardless of the cylinder conditions. So, during the simulation, the residual fraction (unburnt fuel) and the injection and spark timing do not influence the burn rate. This approach can be used to simulate parameters that are not influenced by the burning rate. This mathematical approach could be based on polynomial or cosinus equations, but the best known is the **Vibe** law (Eq. 3) (known as the **Wiebe** law in English literature) [9].

The Spark Ignition Wiebe model calculates the burn rate from the function Eq. (3):

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right] \tag{3}$$

where:

- θ is the crank angle at the instant moment;
- θ_0 is the start of combustion;
- $\Delta\theta$ is the total combustion duration;
- a and m are adjustable parameters.

The non-predictive combustion model in GT-Power software is based on this SI Wiebe equation. The calibration of the combustion model using this approach is done by correlating the duration of the combustion to physical parameters like engine speed, CA50 (the crankshaft angle were fifty percent of the air-fuel mixture is burned), GBR, EGR etc.

Gamma Technologies, a tech leader company in the fast-growing domain of system modeling, uses the Wiebe formula in the form presented below Eq. (4):

$$Combustion(\theta) = CE\left\{1 - \exp\left[-WC(\theta - SOC)^{E+1}\right]\right\} \tag{4}$$

where:

- θ is the instantaneous crank angle;
- CE is the fraction of fuel burned;
- WC is the Wiebe constant;
- SCO is the start of combustion;
- E is the Wiebe exponent.

The physical interpretation of the Wiebe equation from above is that the combustion starts at 0° crank angle (0% burned) and finishes when 100% of the mixture is burned [10].

Jaine and collaborators [11] have presented the low predictiveness of this pure mathematical approach and the need to do a systematic readjustment of the parameters on each engine operating point. Although it is a time-consuming process, in the end, the output of this method is very useful for the development of the engine control strategies [6].

2.3 The Predictive Combustion Models

In the predictive combustion models, the burn rate changes accordingly to any modification of the parameters that have an influence on it. The input for the burn rate calculation are the test results from the engine test bench. The computation time of this kind of models is significantly higher than the non-predictive approach due to the complexity of the calculations [12].

For the fuel burn rate calculation, the GT-Power software divides the combustion chamber into two separate volumes. The first volume represents the unburned air-fuel mixture, which is the mixture that is being injected in the combustion chamber at that instant plus the residuals at IVC (inlet valve closure). The second compartment (the burned volume) progressively increases in subsequent time steps as burning occurs in the first (unburned) compartment. The burn rate is defined as the percentage of air-fuel mixture transferred from the unburned compartment to the burned volume [7].

The engine performance depends on combustion efficiency, which is defined by burn rate. Yet directly measuring the burn rate during combustion is challenging, so predictive combustion models use cylinder pressure to predict burn rate. Calibration of these models using test data from the engine test bench is essential to obtain reliable results [12].

Depending on the engine type, Gamma Technologies have designed four different predictive combustion models:

- The Spark-Ignition Turbulent Flame Model (SITurb);
- The Direct-Injection Diesel Multi-Pulse Model (DI-Pulse);
- The Direct-Injection Diesel Jet Model (DI-Jet);
- The Homogeneous Charge Compression Ignition Model (HCCI) [12].

More details of the SITurb model are presented further in this article.

2.4 Spark-Ignition Turbulent Flame Model (SITurb)

The SITurb combustion model predicts the burn rate for spark ignition engines with homogeneous charge.

The combustion process inside the cylinder is governed by the engine burn rate, for which cylinder pressure is used as a surrogate measure in the engine test bench. In the GT-Power software, by knowing the pressure within the cylinders, the burn rate can be calculated and vice versa. The software uses a ‘reverse run’ simulation to estimate the burn rate from the values of the cylinder pressure and a ‘forward run’ simulation during which, using the burn rate, the cylinder pressure is calculated [13]. The air-fuel mixture is transported from the unburned zone to the burned zone as specified by the burn rate and the cylinder pressure is the result of the energy released from combustion. The calibration of the combustion model is always done using the ‘reverse run’ simulation. The two methods (‘forward run’ and ‘reverse run’) use the same equations as the two-zone combustion process [7].

The two-zone combustion model takes into consideration the cylinder geometry, the spark locations and timing, the air dynamics, and the fuel properties. This turbulent flame entrainment-based combustion model assumes that the fresh mixture at the flame front Eq. (5) is entrained into small eddies and then burned up in a characteristic time Eq. (6). The equations Eq. (5), Eq. (6) and Eq. (7) illustrate the flame entrainment and burn-up processes [14]:

$$\frac{dM_e}{dt} = \rho_u A_e (S_T + S_L) \tag{5}$$

$$\frac{dM_b}{dt} = \frac{(M_e - M_b)}{\tau} \tag{6}$$

$$\tau = \frac{\lambda}{S_L} \tag{7}$$

where:

- M_e is the estimated mass of unburnt mixture;
- t is the time;
- ρ_u is the unburned density;
- A_e is the entrainment surface area at the edge of the flame front;
- S_T is the turbulent flame speed;
- S_L is the laminar flame speed;
- M_b is the burned mass;
- τ is the time constant;
- λ is the Taylor microscale length.

The unburned mixture is entrained into the flame front through the flame area (A_e) at a rate proportional to the sum of the turbulent (S_T) and the laminar (S_L) flame speeds.

The burn rate (M_b) is proportional to the amount of unburned mixture behind the flame front, divided by a time constant which, in term, is calculated by dividing the Taylor Microscale (λ) by the laminar flame speed (S_L) Eq. (7) [14]. Figure 2 presents a sketch of the equations Eq. (5) and Eq. (6).

Model calibration requires measurements on the engine test bench which evaluate the impact of the turbulence intensity and the length scale on the model; for this purpose, the software adjusts the turbulent flame speed and Taylor micro-scale length parameters [12].

The initialization of the SITurb model is done via the Wiebe model for the first engine cycles. The goal is to have the airflow already converged to a steady state before the start of the predictive combustion model [10].

Gamma Technologies uses two methods to estimate the burn rate: TPA-three pressure analysis and CPOA-Cylinder Pressure Only analysis, both using a ‘reverse run’ approach. For this present study, the TPA method was used.

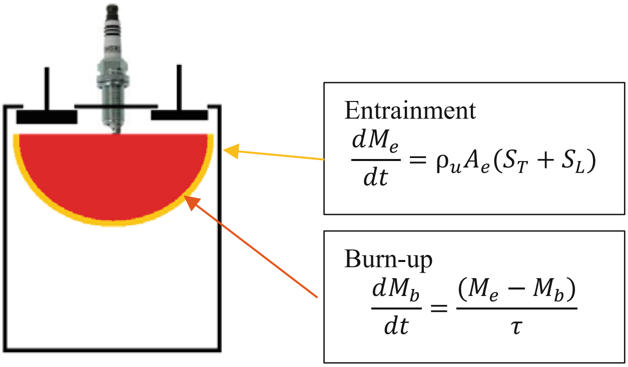

Fig. 2. The two-zone combustion model: entrainment and burn-up

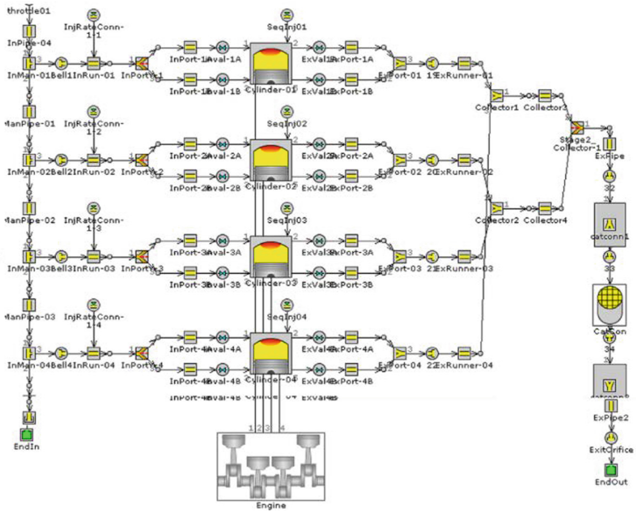

Fig. 3. Example of a four-cylinder engine model using GT-Power [15]

The TPA model needs experimental values for the intake pressure, the in-cylinder pressure and the exhaust pressure. The disadvantage of this method is that the whole engine must be modeled in GTPower in order to use the TPA calibration model. In exchange, this method delivers the trapping ratio at IVC and the residual EGR fraction values as well. An example of an engine modeled in GTPower is depicted in Fig. 3.

For the SITurb combustion model to be as close as possible to the real engine, four multipliers could be modified [12]:

– the dilution multiplier;
– the Flame Kernel Growth Multiplier;
– the Tylor factor multiplier;
– the turbulent flame speed multiplier.

The goal is to obtain a single set of multipliers that could be used on a wide range of engine operating points.

The dilution effect (**C_{DE}**) of the residual EGR or the external EGR influence the laminar flame speed (S_L) [16]. Increasing **C_{DE}** will reduce the effect of dilution on the S_L and thus increase the burn rate.
The Flame Kernel Growth Multiplier (C_{FKG}) is used to scale the calculated value of the growth rate of the Flame Kernel. This variable influences the ignition delay. Higher values shorten the delay, advancing the transition from laminar combustion to turbulent combustion [17].
The Taylor length scale multiplier (C_{TLS}) is used to scale the calculated value of the Taylor microscale length (λ) of turbulence. The λ modifies the time constant of combustion of fuel/air mixture entrained into the flame zone by changing the thickness of the plume.
The turbulent flame speed multiplier (C_{TFS}) is used to scale the turbulent flame speed. This variable influences the overall duration of the combustion. Higher values increase the speed of the combustion.

The **C_{FKG}** and **C_{TFS}** are magnitude factors of the turbulent flame speed. **C_{FKG}** has an especially greater influence on the initial combustion velocity with smaller flame radius [16]. Recent direct numerical simulation (DNS) [18] studies have shown that the initial Flame Kernel size and the turbulence structure (flow surrounding the spark plasma, at constant length scales and turbulence intensity) are more important than integral length scale or turbulence intensity for flame growth.

3 Correlation Between the Simulated and ECU Command Spark Timing on Engine Test Bench

Most of the tuning activities are based on engine maps done either in standard conditions (90 °C coolant temperature and 23 °C ambient temperature) or in any other conditions. The engine map is done on the engine test bench, and it is defined by a sweep of engine speed from idle to maximum engine speed and a sweep of engine torque from idle to maximum engine torque, as shown in Fig. 4.

The purpose of using the Wiebe or SITurb combustion models is to limit the time needed on the engine test bench for different activities (each of them being both time and money consuming).

In the early stages of a spark ignition engine development (when choosing the technical definition), the Wiebe or SITurb combustion models are used, but the combustion phasing is calculated from the CA10 (the crankshaft angle were ten percent of the air-fuel mixture is burned) and CA50 parameters. In the tuning activities that follow the initial project development, knowing the exact spark timing is essential, with implications in torque estimation, pollutants, and drivability.

This section presents results of research aimed at determining the combustion model parameters needed to simulate the spark timing in virtual engine test bench.

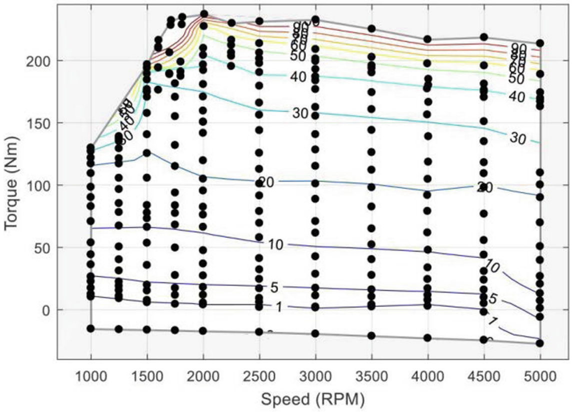

Fig. 4. Engine map [19]

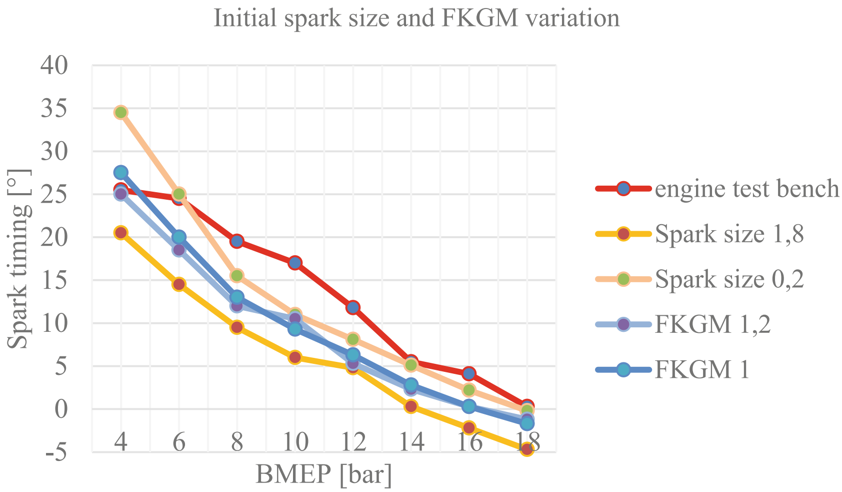

Fig. 5. Initial spark size and FKGM influence on a variation of engine load at constant engine speed

The GT-Power SITurb model was chosen to attempt to model the exact spark timing (CA0), by modifying the initial spark size and the Flame Kernel Growth Multiplier (FKGM). The initial approach was to manually modify the initial spark size and FKGM at a constant engine speed to see the impact when the engine load changes, Fig. 5.

After this first attempt, the spark timing generated by the model was significantly disproportionate compared to the spark timing resulting from the engine test bench data. To decrease the error, two optimization methods were chosen:

– DO (direct optimizer);
– DOE (design of experiments).

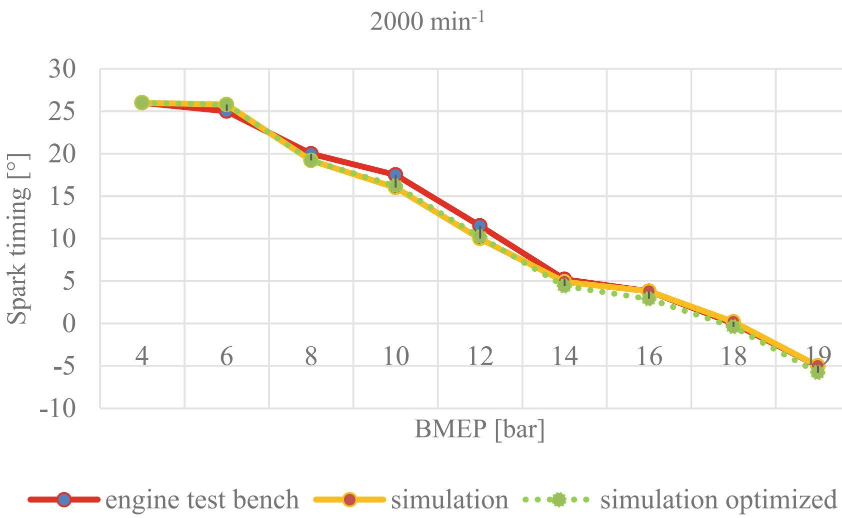

Fig. 6. Real and simulated spark timing by DO approach

The Direct Optimizer is an optimization tool based on an iterative method**.** To determine the spark timing, the DO will keep the engine speed at a constant value, while changing the initial spark size and the FKGM until the difference between the simulated spark timing and the one issued from the engine test bench is lower than a cut-off (± 1° crank angle, as inputted by the user). A major disadvantage of this method is that the convergence could be achieved after many iterations, which is time consuming.

Results of the spark timing depending on the BMEP (break mean effective pressure), after several iterations of initial spark size and FKGM are presented in Fig. 6.

80% of the measurements were within the cut-off of ± 1° crank angle. The simulated data graphic followed the trend and values of the engine test bench data.

The values of Spark Size and FKGM used in the simulation depicted in Fig. 6 are illustrated in Fig. 7.

The initial Spark Size and the FKGM values as compiled from Direct Optimizer vary in a hazardous way, a trend cannot be predicted to use for other engine speeds. Thus, test bench values of the ignition timing will be needed for the other points of the engine map, rendering the simulation gain senseless.

To maximize the use of the CAE tools, the parameters Initial Spark Size and FKGM were modified to limit their variations, as imaged by the dotted lines in Fig. 7. With these new adjusted values, the ignition timing error was within the cut off ± 1° crank angle (green dotted line in Fig. 6).

These values of Initial Spark Size and FKGM were validated at a different engine speed (4000 min^{-1}), when the ignition timing error was also within the criteria of ± 1° crank angle (Fig. 8).

The Design of Experiments (DOE) optimization tool is based on a matrix of input elements to achieve the best output solution. Several combinations of engine speed – engine load were chosen. For each such point on the engine map different combinations

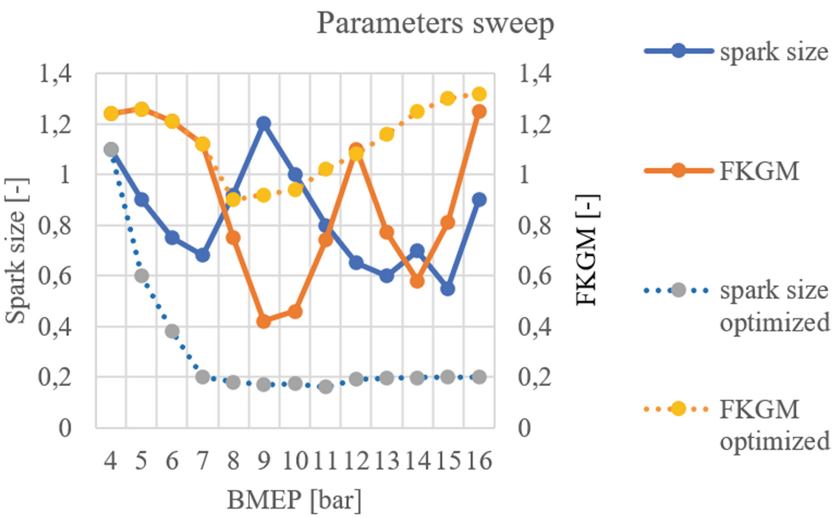

Fig. 7. Initial spark size and FKGM sweep

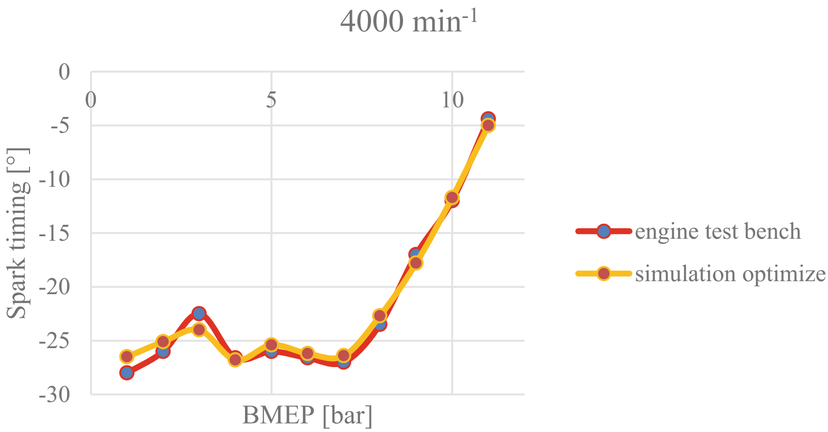

Fig. 8. The ignition timing error using the initial spark size and FKGM found for 2000 min^{-1}, validated at 4000 min^{-1}

of the Initial Spark Size and the FKGM values were tried. Once the spark timing error reduced within the cut off ± 1° crank angle, the Initial Spark Size and the FKGM were noted into a table alongside the corresponding engine load and speed.

The goal was to see that this matrix of engine speed/load – Initial Spark Size/FKGM found for a particular engine validates on other engine types (cylinder size, number etc.). The advantage of this method was that it was quick, but it was finally abandoned because it was prone to error. Although the limits of the values for each parameter were manually established, the tool was keen to doing extrapolations that would influence the results and this was exactly what happened during the trial, motivating its abandoning.

4 Conclusions and Future Work

Simulation of ignition timing using the GTPower SITurb model resulted in a group of parameters (Initial Spark Size and Flame Kernel Growth Multiplier) that matched the cut off $\pm$ 1° crank angle, at an engine speed of 2000 rpm and were validated at 4000 rpm.

Future directions include verifying that the pair of parameters found could be validated for all engine speeds. If successful, this method will prove that tuning activities based on ignition timing could be done in simulation rather than on engine test bench.

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