

Knock and Cycle by Cycle Analysis of a High Performance V12 Spark Ignition Engine. Part 2: 1D Combustion and Knock Modeling

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ABSTRACT

The results of the experimental analyses, described in Part 1, are here employed to build up an innovative numerical approach for the 1D modeling of combustion, cycle-by-cycle variations and knock of a high performance 12-cylinder spark-ignition engine. The whole engine is schematized in detail in a 1D framework simulation, developed in the GT-Power™ environment. Proper “in-house developed” sub-models are used to describe the combustion process, turbulence phenomenon, cycle-by-cycle variations (CCV) and knock occurrence. In particular, the knock onset is evaluated by a chemical kinetic scheme for a toluene reference fuel, able to detect the presence of auto-ignition reactions in the end-gas zone. In a first stage, the engine model is validated in terms of overall performance parameter and ensemble averaged pressure cycles, for various full and part load operating points and spark timings. Then, the correlation regarding the maximum in-cylinder pressure distribution developed in Part 1 is here applied to predict representative *faster-than-average* and *slower-than-average* cycles, miming the effects of the experimentally observed CCV. A proper knock index is introduced and evaluated with reference to the above *faster-than-average* cycle. An automatic procedure is implemented to identify the Knock Limited Spark Advance (KLSA), based on the same threshold level utilized in the experimental knock analysis of Part 1. The numerical and experimental KLSA presents an excellent agreement, denoting the accuracy of the proposed combustion and knock modeling.

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INTRODUCTION

As widely discussed in Part 1, the engine operation is always affected by a certain cyclic dispersion. In the case of spark ignition (SI) engines, cycle-by-cycle variations (CCV) are mainly a consequence of the early flame development conditions, that deeply influence the subsequent combustion phase [1,2,3].

Nowadays, many automotive manufactures make an extensive use of 1D codes, which are able to describe the behavior of the whole engine starting from the intake mouth up to the exhaust tailpipe.

Several models are available in current literature for the description of the combustion process. Since many years, two-zone quasi-dimensional models have been proposed [4,5] and some of them are usually employed in a number of proprietary or commercial 1D codes. They are based on different theoretical approaches, trying to describe the flame development inside a turbulent flow field.

A popular theory is the entrainment of turbulent eddies in the flame front. These eddies burn inwards from peripheral ignition sites to be consumed in a characteristic time with the laminar burning speed. The entrainment velocity is given by an algebraic equation from the flamelet modeling relations. The entrainment theory is rather a mathematical modeling that agrees with the S-shaped burned mass fraction, which is observed in experiments [6,7].

Among the different approaches, the “fractal” combustion model appears to be the one which better describes the turbulent flame propagation phenomenon inside an internal combustion engine [8, 9, 10]. It is based on the hypothesis that aerodynamic turbulence enhances the burn rate by increasing the surface of the flame front with respect the laminar extent, without significantly altering the inner flame structure. All turbulent length scales are hence assumed to be larger than the flame front thickness (flame is in the so called “wrinkled flamelet” regime). While the above approaches are successfully adopted to reproduce the “ensemble averaged” engine cycles, their employment is not straightforward for the modeling of sequence of consecutive cycles, taking into account cycle-by-cycle variations.

A basic method [11] for the CCV simulation consists in modeling of the combustion by means of a Wiebe function, whose coefficients are randomly varied to reproduce the experimental cyclic dispersion. A more “physical” approach, proposed in [12], is based on phenomenological models for the description of the in-cylinder turbulence and combustion processes. CCV are introduced through a random modification of the turbulence production during the intake phase. Indeed, in [13, 14, 15], CCV are imposed through a perturbation of the parameters that control the combustion phasing and duration (turbulence intensity, air-to-fuel ratio, residual fraction, etc.). However, the above approaches, while providing encouraging results in CCV simulation, of course require an increased computational effort because of the necessity to compute a large number of cycles to obtain a statistically significant test sample. This may undermine the typical simplicity and efficiency of 1D modelling.

Knock modelling is another open point in the current literature about SI engines. In 1D codes, the knock occurrence is usually detected by the computation of the induction time in the unburned zone [16]. The evaluation of this time is very commonly based on simple Arrhenius-like correlations [17]. The latter do not take into account the negative temperature coefficient, typical of various common fuels. In addition, they usually neglect the effects related to fuel composition, air to fuel ratio or residual fraction.

In this paper, a detailed model of a high performance 12-cylinder engine, extensively tested in the Part 1, is developed in the GT-Power environment, adopting “in-house developed” routines for the modelling of combustion, cycle-by-cycle variations and knock. In particular, the combustion model is based on a “fractal approach” for the description of the flame front propagation [18]. Moreover, a refined in-cylinder heat transfer model is adopted, based on a modified Hohenberg correlation. Wall temperature distributions are computed through a proper specification of the heat transfer coefficients of the cooling water and lubricant oil. In a first stage, the model is validated in terms of global performance (air flow, indicated mean effective pressure and fuel consumption) and “ensemble averaged” pressure cycles, under full load and part load operation for various engine speeds, loads and spark timings. Then, the effects of the experimentally observed cyclic dispersion are introduced through the computation of representative *faster-than-average* and *slower-than-average* cycles. The above approach is realized on the basis of the correlation, presented in the Part 1, between the in-cylinder peak pressure Coefficient of Variation (CoV_{pmax}) and the predicted combustion parameters, namely the combustion phasing (crank angle of 50 % Mass Fraction Burned, MFB_{50}), and duration $\Delta\theta_b$. The same conventional definition of combustion duration of Part 1 is here adopted, that is $\Delta\theta_b = 2 (MFB_{50} - SA)$.

A knock model is finally introduced. The latter describes the species variations inside the “end-gas” unburned zone through the solution of a chemical kinetic scheme for a Toluene Reference Fuel (TRF). The above model is used to compute the Knock Limited Spark Advance (KLSA) that is compared with the experimentally identified values.

COMBUSTION AND TURBULENCE MODELS

Description

As said, the combustion process is modeled through a “fractal approach” [18], sensing both the combustion system geometry and the operating parameters. Flame wrinkling scales and the fractal dimension of the flame area depend on the in-cylinder turbulence evolution, described by a 0D sub-model, extensively reported in [18,19]. Following the referenced hierarchical 1D-3D approach, the tuning constants of turbulence and combustion sub-models are selected in order to fit the in-cylinder 3D-derived turbulent intensity and the in-cylinder pressure cycles, respectively. A special attention is devoted to the heat transfer modelling because of its critical influence on the knock prediction. A modified Hohenberg correlation is adopted, while the piston, cylinder liner, and head temperatures are calculated by the simplified finite element approach implemented in GT-Power. The heat transfer coefficients on the coolants side (water and oil) are derived by dedicated simulations of the coolant circuit and properly adjusted according to the engine speed [20].

Tuning and Validation

The considered engine is carefully schematized in the GT-Power environment. The experimentally actuated air-to-fuel ratio, spark advance and valve timing are imposed as input data in the simulations. Six full load operating points (from 2000 up to 7000 rpm, with step 1000 rpm) are studied. In addition, two part load points (2000 rpm / 6 bar BMEP and 3000 rpm / 10 bar BMEP) are analyzed, as well. For each point, a 2 CAD spark advance sweep is performed, collecting 56 operating points. A window of twelve crank angles around a reference value of the spark timing (labelled in the following as “Base SA”) is investigated. The Base SA corresponds to a first evaluation of the SA at the knock limit identified by heuristic procedures, based on operator experience at the test bench. In the case of part load operation, the throttle is adjusted to match the experimentally derived BMEP, while it is fully opened for the full load cases.

The results shown in [Figure 1](#), [Figure 2](#), [Figure 3](#) are obtained after the tuning of the combustion and turbulence models. It must be stressed that a single set of tuning constants has been selected for all the considered operating conditions. In particular, the Figures show the numerical/experimental comparison in terms of air mass flow rate, IMEP and BSFC. Due to confidentiality reasons, the scales of the plots are hidden. It can be noted that the air flow rate is satisfactorily predicted (average error of 2.83 %), denoting the good accuracy in the engine geometry schematization. The IMEP prediction is mainly affected by the combustion modeling and an average error of about 1.81% is obtained ([Figure 2](#)), with the majority of points below 5%. The BSFC denotes slightly higher errors ([Figure 3](#)), depending on small inaccuracies in both flow and combustion modelling. However, the average error still lies below 5%.

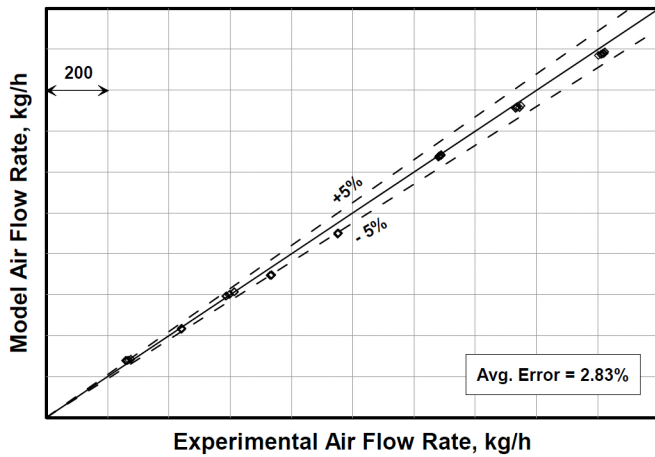


Figure 1. Experimental vs. numerical air flow rate comparison

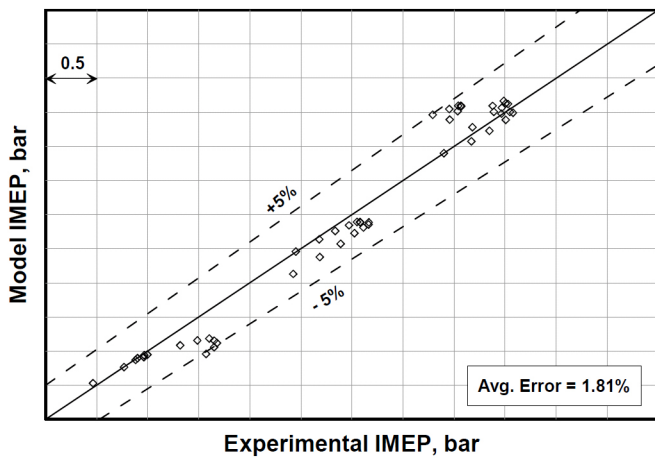


Figure 2. Experimental vs. numerical IMEP comparison

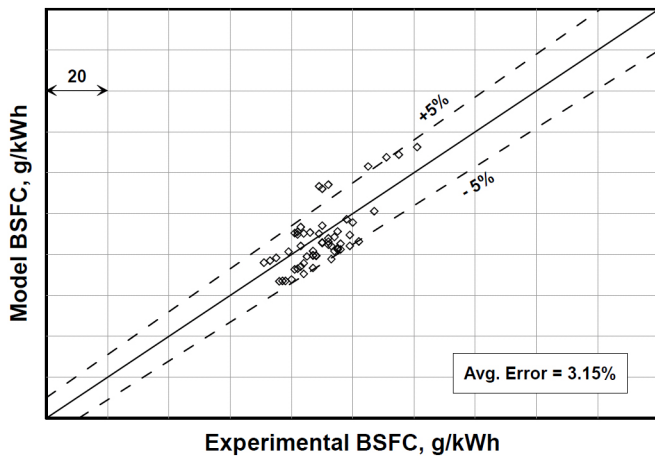


Figure 3. Experimental vs. numerical BSFC comparison

The model accuracy is further proved by the experimental / numerical comparison of the pumping loop of Figure 4. The latter underlines that pressure fluctuations are very well captured by the model, both in terms of shape and pulse amplitude. Minor inaccuracies only occur during the early stage of the exhaust phase.

In Figure 5, Figure 6, Figure 7, Figure 8 the “ensemble averaged” measured pressure cycles are compared with the computed ones, for the full load points at 2000, 4000 and 7000 rpm (labelled as

2000WOT, 4000WOT and 7000WOT, respectively) and the part load point at 3000 rpm (labelled as 3000@10). For each condition, three spark timings are considered: the base spark advance, and both a 6 CAD delayed and a 6 CAD advanced timing (labelled as “Base SA + 6 CAD” and “Base SA - 6 CAD”, respectively).

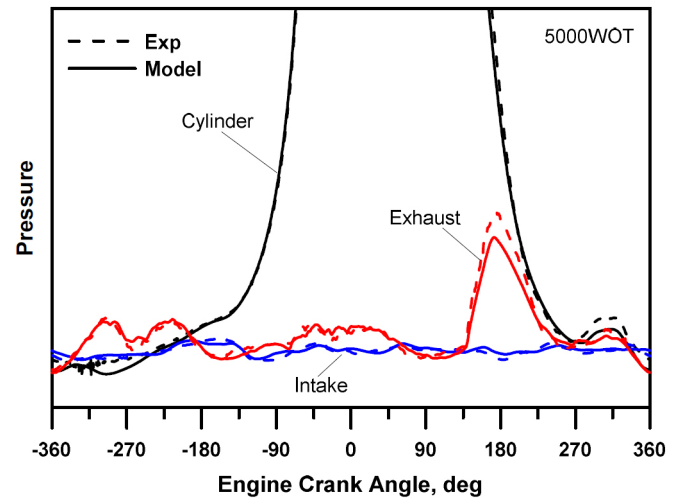


Figure 4. Experimental vs. numerical pumping loop

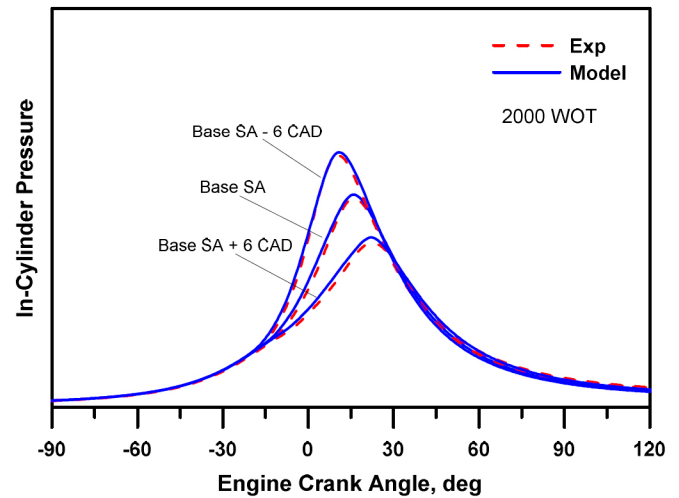


Figure 5. In-cylinder pressure comparison at 2000 WOT

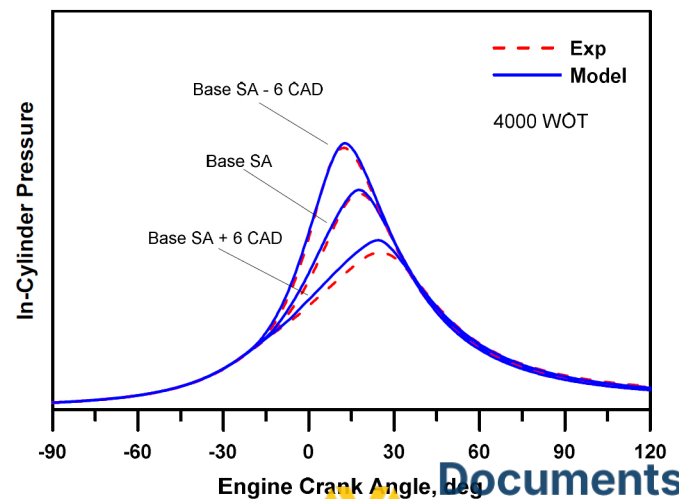


Figure 6. In-cylinder pressure comparison at 4000 WOT