

Modeling Cycle-to-Cycle Variations in 0-D/1-D Simulation by Means of Combustion Model Parameter Perturbations based on Statistics of Cycle-Resolved Data

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ABSTRACT

The presented paper deals with a methodology to model cycle-to-cycle variations (CCV) in 0-D/1-D simulation tools. This is achieved by introducing perturbations of combustion model parameters. To enable that, crank angle resolved data of individual cycles (pressure traces) have to be available for a reasonable number of engine cycles. Either experimental data or 3-D CFD results can be applied. In the presented work, experimental data of a single-cylinder research engine were considered while predicted LES 3-D CFD results will be tested in the future. Different engine operating points were selected - both stable ones (low CCV) and unstable ones (high CCV).

The proposed methodology consists of two major steps. First, individual cycle data have to be matched with the 0-D/1-D model, i.e., combustion model parameters are varied to achieve the best possible match of pressure traces - an automated optimization approach is applied to achieve that. Second, the combustion model parameters (obtained in previous step) are statistically evaluated to obtain PDFs and cross-correlations. Then such information is imposed to the 0-D/1-D tool by means of both PDFs and cross-correlations of combustion model parameters to mimic pressure traces CCV.

Statistics of different engine operating points are compared. The main conclusion is that the CCV effects are overestimated if cross-correlations are not taken into account. It was confirmed that turbulence properties and initial flame kernel development are the dominant factors in terms of CCV. These factors are not independent nor random - they seem to be correlated.

CITATION: Vitek, O., Macek, J., Poetsch, C. and Tatschl, R., "Modeling Cycle-to-Cycle Variations in 0-D/1-D Simulation by Means of Combustion Model Parameter Perturbations based on Statistics of Cycle-Resolved Data," *SAE Int. J. Engines* 6(2):2013, doi:10.4271/2013-01-1314.

INTRODUCTION

The objective of the task presented in this paper was to develop efficient 0-D/1-D combustion model(s) for direct-injection spark-ignition engine (DI SI-engine) simulation in order to include CCV effects, to parameterize the 1-D models based on cycle-resolved experimental and multi-scale simulation result data (from LES simulations), and finally to validate the 1-D model(s) for DI SI-engine application based on available experimental data.

The work was organized according to the following steps:

- 1-D CFD model definition considering approaches of different complexity levels, such as e.g. an extended Vibe-based approach and a fractal-based combustion model in

order to account for cycle-to-cycle variations on a physical basis;

- Extension of the fractal combustion model by adopting statistically adapted parameter maps in order to take into account cyclic combustion variations;
- Implementation of the extended combustion model in the 1-D CFD code AVL BOOST; parameterization of the model for the relevant range of statistical variations on the basis of experimental data and the LES/PANS results;
- Test and validation of the performance of the developed combustion model and model parameterization by performing calculations of the engine configuration and operating

conditions and by comparison of the calculated results with the experimental data.

Background and Motivation

The operation of spark-ignition engines is characterized by a non-repeatability of the instantaneous combustion rate of the individual engine cycles at nominally identical engine operating parameters, commonly referred to as cyclic combustion variability. For CCV amplitudes (measured in terms of standard deviation of IMEP, normalized by the mean value) exceeding 5%, individual cycles behave very differently from the statistically-averaged mean engine cycle. As a result, predictions of fuel consumption or emissions over a number of cycles may differ quite substantially from the one based on the mean engine cycle. These differences increase with magnitude of CCV, and may reach extreme levels in the case of misfire or knocking cycles.

Experimental studies allow identifying CCV for a given engine, and they can help test control strategies to limit their occurrence and impact. But they are only applicable once the engine is available in hardware, i.e. in a phase of the engine development process in which modifications of the initial design are difficult to realize due to cost and time constraints.

In the above context and in view of the increasingly stringent legislative demands on fuel consumption, CO₂ production, and pollutant emissions, it becomes mandatory to be able to predict and control individual engine cycles, and thus to address the occurrence and the impact of CCV on fuel consumption and hence CO₂ emissions. Being able to predict CCV in the early engine design phase is essential to exploit the full potential of promising new SI-engine technologies, such as e.g. direct-injection, down-sizing, extreme charging, etc. under real operation. This leads to an increased need to make such capabilities available in system simulation tools which are increasingly used in the engine development and optimization process.

Hence, the objective of the present task is to develop a fast and efficient 0-D/1-D combustion modeling approach applicable to DI SI-engines, including CCV effects, to parameterize the 0-D/1-D model based on cycle-resolved experimental and/or multi-scale simulation result data, and finally to validate the selected approach.

Current Status of CCV Modeling

The phenomenon of cycle-to-cycle variations (CCV) of internal combustion engine (ICE) is well known for many decades, hence the researchers have been dealing with it for a long time. However, the origin and nature of CCV is still not well understood. Spark ignition (SI) engines seem to be more affected by this effect, though compression ignition (CI) engines also feature certain level of CCV. It is obvious that CCV leads to lower engine efficiency while increasing production of pollutants. This is not desired, hence a lot of effort is made to minimize CCV effects. The main sources of CCV have been identified - in-cylinder turbulence, variations

in mixture composition (air excess and EGR), initial flame kernel development, variations of ignition/injection system, etc. Based on that, combustion process is the most important phase of ICE cycle when dealing with CCV - this is certainly true, however motored regimes also exhibit CCV. It seems that there are differences (in terms of CCV) among the engines - they are caused by the design (direct/indirect injection, liquid/gaseous fuel, ignition system, etc). This makes systematic comparison even more difficult as every engine has its unique features.

It is interesting to note that the available information regarding CCV is relatively limited - the papers are usually focused on a particular feature related to CCV, however a systematic approach to assess CCV from general point of view is missing. This is most likely caused by the fact that the experiments which are aimed at systematic evaluation of CCV sources, are difficult to perform. Moreover local properties (under real engine operating conditions) are needed to understand physical processes. A detailed overview of many CCV related issues is given in [3], however this information is relatively old.

Based on that, the modeling approach was also applied to improve the knowledge of CCV phenomena. Initially, simple thermodynamic models (together with large database of experimental data) were applied - these models were usually based on 0-D approach to an ICE cylinder and the main target was to evaluate the influence of certain parameter variations on ICE performance. The advantage of such model is the ability to estimate the CCV influence on a system level, however such models cannot predict CCV. As the computer power increased, it was possible to apply a more detailed modeling approach - using the 3-D CFD. It enables to take into account local properties (which is not possible by 0-D models). However, a common practice of 3-D CFD modeling is to use the RANS approach, which deals with average data - this is obviously not desired when dealing with CCV, hence the potential to properly model CCV is limited. Fortunately, the LES approach enables to resolve individual cycles (not the average one as is the case for RANS), which enables better understanding of CCV effects. Moreover, LES is supposed to resolve the large scales of turbulence, hence the non-linear interaction among them is directly captured - this is a big advantage as it is believed that non-linear turbulent interactions are mainly responsible for CCV effects. Based on that, the 3-D CFD LES approach is supposed to be capable of predicting the CCV.

The following text mainly deals with modeling of SI engine by using 0-D/1-D tools since system simulation is of main interest and the CCV has to be imposed using either experimental data or 3-D CFD ones. Such approach is relatively old - it was applied in [7, 8, 10, 11, 12, 13] and it is also applied in the presented work. These papers deal with different phenomena, however the common issue is the combustion CCV. This is not a surprise as the combustion process is believed to be the main source of CCV - c.f. [3, 11, 13]. Typical way to impose CCV in a system simulation is to

perturb certain parameters of a particular combustion model - this is supposed to model all CCV effects. The paper [13] proposes an innovative approach to use the knowledge from 3-D CFD LES simulations (of motored regime) to impose perturbations of certain physical properties (mainly tumble kinetic energy) - the applied combustion model (which is derived from 3-D version by means of 'coordinate reduction' to formulate 0-D version) is able to take into account those perturbations, hence predicting combustion CCV.

Regarding the CCV source identification, the following can be stated. In [6, 9] was concluded that CCV seems to be a random process - if any 'organized' behavior appears, it is usually a consequence of incomplete combustion (when close to ignition limit of very lean mixture) or knocking. Regarding mixture composition (taking into account air excess and EGR), the conclusions in [7, 10, 11] suggest that it is not the most dominant factor, moreover paper [10] shows that mixture composition mainly influences initial phase of a combustion process (especially wrinkling of flame front). The work presented in [11] leads to the statement that the most dominant factor in terms of CCV is turbulence intensity, which varies a lot among individual cycles. This conclusion is confirmed by [10], moreover it adds that the turbulence properties are almost independent of mixture composition (air excess and EGR). Turbulence properties strongly influence both initial flame kernel development and free flame propagation. Initial flame kernel development is a critical phase which can be responsible for up to 50% of total CCV - this is stated in [14] and echoed in [4, 5]. Finally, [11] presents that there is almost no influence of preceding cycles.

Based on the above it can be stated that CCV is a complex physical phenomenon, which is not easy to understand and most likely difficult to control. Intensive research is needed to improve the knowledge of it, which is supposed to provide information regarding possibilities to minimize CCV effects.

ENGINE AND EXPERIMENTAL SETUP

Engine and Engine Operating Conditions

The DI SI-engine configuration adopted for the activities within the present work is based on the AVL Single Cylinder Research Engine (SCRE), [Figure 1](#), which is a fully flexible single-cylinder base engine configuration that can be equipped with a large variety of cylinder heads, valve trains and injection systems. The engine has a displacement of 500 cc at a bore and stroke of 86×86 mm. In its present configuration the engine has a nominal compression ratio of 12.5 and is equipped with a four-valve cylinder head and a side mounted gasoline injector with double injection capability. In the present studies the engine is equipped with intake and exhaust cam-phasers allowing flexible valve timing and valve overlap strategies. Main engine parameters are summarized in [Table 1](#).

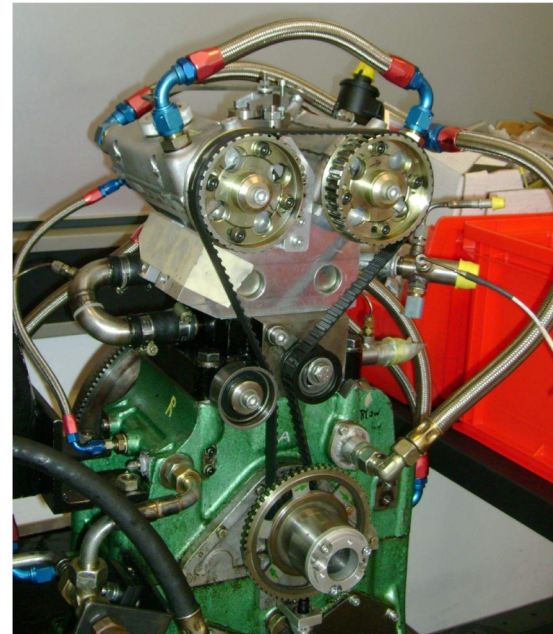


Figure 1. AVL Single Cylinder Research Engine.

Table 1. Main parameters of the AVL SCRE engine.

Bore	[mm]	86
Stroke	[mm]	86
Compression Ratio	[-]	11.5
Engine Speed	[rpm]	1250 - 4000
Fuel Injection		Direct
Intake Valves		2 (VVT)
Exhaust Valves		2 (VVT)

Test bed measurements such as e.g. cycle-resolved in-cylinder pressure indication, low pressure indication at the intake/exhaust ports, cycle-averaged tail-pipe emission measurements, etc. are available for the entire speed/load range adopting broad variations of the intake/exhaust valve phasing. For the present studies two operating conditions are selected, namely a baseline one with zero valve overlap (labeled as LP 2504) and hence exhibiting rather low cycle-to-cycle variations, and one with maximum valve overlap (labeled as LP 2735) and consequently large cycle-to-cycle combustion variations ([Figure 2](#)). Additional information regarding considered operating points can be found in [Table 2](#).

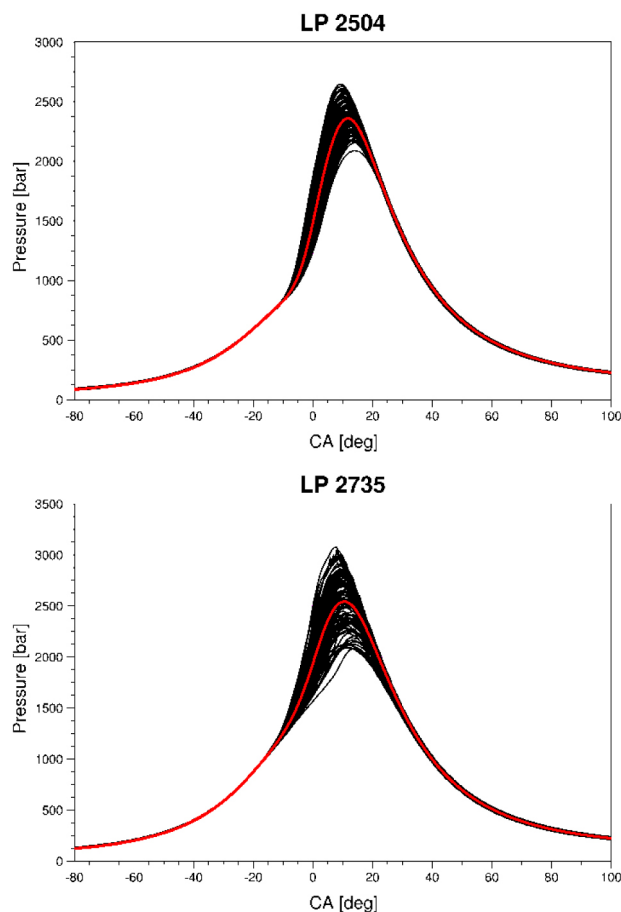


Figure 2. Measured cylinder pressure traces for selected operating conditions; black: cycle-resolved data, red: cycle-averaged data.

Table 2. Main parameters of engine operating points.

		LP 2504	LP 2735
Engine Speed	[rpm]	2000	3000
IMEP of HP Phase	[bar]	3.6	3.2
Ignition Timing	[degCA]	-17	-29
Exhaust Valve Open	[degCA]	134	187
Exhaust Valve Close	[degCA]	346	400
Intake Valve Open	[degCA]	381	347
Intake Valve Close	[degCA]	592	558

Measurement Data Processing

Analysis of the measured in-cylinder and low-pressure indication data is performed by application of the AVL Gas Exchange and Combustion Analysis Tool (GCA). Basically, GCA is an integration of the 1-D cycle simulation tool BOOST into the test bed environment. Based on a simple cylinder configuration, GCA enables the analysis of the engine high pressure cycle adopting the low pressure indication data as boundary conditions at the intake and exhaust side. Adopting GCA it is hence easily possible to

analyze cycle-resolved data directly on the test bed online during the measurements or separately as an offline application based on stored measurement data in the office (Figure 12 located in Appendix).

0-D/1-D MODELING

A full 0-D/1-D cylinder/flow model of the entire engine was set in order to properly model the tested engine configuration including all intake and exhaust ductworks and thus to provide proper 1-D pressure, temperature and mass initial conditions in the combustion chamber at the time of inlet-valve closure. The model consists of the engine cylinder itself plus all relevant pipes and volumes in the intake and exhaust line. Parameterization of the flow coefficients and other relevant parameters is achieved on the basis of the measured / GCA processed data at selected monitoring locations of the system.

1-D Cycle Simulation

When dealing with system simulation, a standard approach is to apply 0-D/1-D tools (e.g., [2]) which provide relatively fast computations while being able to keep a sound physical basis. They are suitable for modeling of the whole system, e.g., all important thermodynamic parts of the engine. That is why it is important to be able to impose CCV phenomenon into those tools to understand its influence at a system level. That is why it was decided to model (in tool [2]) the whole experimental setup of the SCRE engine (Figure 13 in Appendix).

Combustion Modeling in SI Engines: The Fractal Combustion Model

In the present work the in-cylinder combustion is modeled by a physically based, predictive combustion model. This model takes into account both engine configuration (e.g. combustion chamber geometry) as well as operating parameters (e.g. turbulence characteristics, ignition timing, charge state at IVC,...) on a 0-D basis.

The fractal combustion model is of the type of flamelet models. In these models, the flame front is considered an infinitely thin area, which is highly wrinkled due to the effects of turbulent eddies of different length scales.

Therefore, the combustion model required as an input the in-cylinder turbulence level which is calculated as proposed in the following sections.

Flamelet Approach

The fractal combustion model for SI engines, predicts the rate of heat release in a homogeneous charge engine. The calculation is based on a two zone discretization of the cylinder. The combustion model calculates the gas conditions of the combustion products (i.e. the burned zone) and the remaining fresh charge (i.e. the unburned zone).

It is well established that the flame front propagating within the turbulent flow field occurring inside the

combustion chamber of an internal combustion engine is a very thin and highly wrinkled surface. This turbulent flame area A_T , is much higher than the one occurring in a laminar burning process. The laminar flame area A_L , is considered as a smooth and spherical surface centered in the spark plug. The increase in the flame surface (A_T/A_L), is then primarily responsible for the increase in the turbulent burning rate S_T with respect to the laminar one (S_L). The mass burning rate can be expressed as:

$$\frac{dm_b}{dt} = \rho_u \cdot A_T \cdot S_L = \rho_u \cdot \left(\frac{A_T}{A_L}\right) \cdot A_L \cdot S_L$$

where $\left(\frac{S_T}{S_L}\right) = \left(\frac{A_T}{A_L}\right)$ (1)

This expression, introduced by Damköhler in [19], represents a definition of the turbulent burning speed.

Under these assumptions it is possible to develop a quasi-dimensional combustion model, based on the concept of the fractal geometry. In this approach, an initially smooth flame surface of spherical shape (A_L) is wrinkled by the presence of turbulent eddies of different length scales. The interactions between the turbulent flow field and the flame determine the development of a turbulent flame surface A_T , which propagates at the laminar flame speed. If a self-similar wrinkling is assumed within the length scales interval [L_{min} , L_{max}] then the flame front presents the characteristics of a fractal object and its flame surface can then be computed by:

$$\left(\frac{A_T}{A_L}\right) = \left(\frac{L_{max}}{L_{min}}\right)^{D_3-2}$$

(2)

The above expression, substituted in the Equation 2, allows computing the burning rate once the laminar flame surface A_L and speed S_L as well as the wrinkling scales L_{min} and L_{max} and fractal dimension D_3 are known. The upper limit of the wrinkling scale L_{max} is defined by the characteristic length L_I , calculated by eq. 11, while L_{min} is given by the Kolmogorov length scale.

$$\left(\frac{dm_b}{dt}\right)_{fractals} = \rho_u \left(\frac{L_{max}}{L_{min}}\right)^{D_3-2} A_L S_L$$

(3)

Turbulence

Based on the physical hypothesis mentioned above, the computation of the wrinkling scales L_{min} and L_{max} as well as the fractal dimension D_3 must depend on the characteristics of the turbulent flow field inside the cylinder. Its evaluation within a zero-dimensional model is challenging.

The genuine implementation of the fractal combustion model used the two-equation modified K - k approach given in [20] for the calculation of in-cylinder turbulence. By performing detailed comparisons with CFD results it has shown, that even better results in terms of agreement of TKE characteristics can be achieved with the refined K - k approach by Bozza given in [22]. The most important equations are given below.

Definitions:

$$K = \frac{1}{2} \cdot m \cdot U_f^2, \quad k = \frac{3}{2} \cdot m \cdot u'^2$$

(4)

K is the kinetic energy of the mean flow field (U_f) - whose production and destruction is mainly related to the intake and exhaust flow rates ($\dot{m}_{in}$ and $\dot{m}_{ex}$). k is the kinetic energy of the turbulent flow field (assumed isotropic). Equations (5) and (6) are integrated all over the engine cycle and a turbulent production term due to the in-cylinder unburned density variation during the compression and expansion stroke is included in both K and k balance equations. These balance equations do not consider the piston motion as a source of kinetic energy, which is especially relevant at higher engine speeds.

$$\frac{dK}{dt} = c_{in} \cdot \left(\frac{1}{2} \cdot \dot{m}_{in} \cdot u_{in}^2\right) + c_{ex} \cdot \left(\frac{1}{2} \cdot \dot{m}_{ex} \cdot u_{ex}^2\right) - P + K \cdot \frac{\dot{m}_{ex}}{m} + K \frac{\dot{\rho}}{\rho}$$

(5)

$$\frac{dk}{dt} = P - D + k \cdot \frac{\dot{m}_{ex}}{m} + k \cdot \frac{\dot{\rho}}{\rho}$$

(6)

P represents a turbulent production term which characterizes the energy transfer between the mean and the turbulent flow field. D is the dissipation rate

$$P = c_p \cdot \frac{K}{t_p} \quad \text{with } c_p = 0.064$$

(7)

$$D = c_D \cdot \frac{k}{t_D} \quad \text{with } c_D = 0.64$$

(8)

The timescales t_p and t_D are calculated from eq. (9) and (10). In eq. (7) a smooth transition between the two formulations is specified.

$$t_p = \frac{L_I}{U_f} \quad (9)$$

$$t_D = \begin{cases} \frac{L_I}{u'} \dots \text{for high pressure and intake} \\ \frac{l_k}{u'} \dots \text{for exhaust phase} \end{cases} \quad (10)$$

The model also gives the possibility to estimate the Kolmogorov length scale l_k which, under the hypothesis of isotropic turbulence, assumes the expression (12). L_I is the integral length scale, assumed proportional to the characteristic length estimated from the cylinder volume assuming spherical shape.

$$L_I = c_L \cdot \sqrt[3]{V_{cyl}} \quad (11)$$

$$l_k = \frac{L_I}{(u' \cdot L_I / \nu)^{0.75}} \quad (12)$$

For all calculations the following values were fixed:

$$c_{in} = 0.156 \quad \text{and} \quad c_{ex} = 0.06$$

The parameter c_L was used as a calibration parameter to adjust the model to CFD results.

Ignition

The complex phenomena occurring after spark occurrence, plasma formation and subsequent flame kernel evolution are described in detail in [23]. Kernel initiation process ends about 200 ms after spark at a critical flame radius of about 2 mm (it is an estimate only - the flame kernel radius was not fixed at value of 2 mm during the course of simulation work, i.e., it was one of the considered parameters to be varied). During this period burning speed is very high, depending on energy released by the ignition system, then it reaches a minimum to values similar to the laminar flame speed and subsequently it increases again, as a consequence of the flame surface wrinkling described previously.

As the phenomena described above cannot be included in detail in a 0-D model, it is assumed to start the computation at the end of kernel initiation process with a stable and spherically-shaped smooth flame of about 2 mm radius. The flame wrinkling process then starts at a rate which increases with both the instantaneous flame radius and the turbulence intensity. The following expression is proposed in [21] for the computation of a non-dimensional flame wrinkling rate:

$$\omega_{wr} = \frac{r_f}{r_{f,ref}} \frac{n}{n_{ref}} \quad (13)$$

In the model, the reference radius $r_{f,ref}$ is used as a tuning parameter which is usually in the order of 10 mm. A careful tuning of parameter $r_{f,ref}$ is required to match the experimental heat release at each engine operating condition. The reference speed n_{ref} is set to 1000 rpm.

Wall Combustion

When the flame front reaches the combustion chamber walls, the solid wall limits gas expansion, constrains all flows, and forms a relatively low-temperature solid boundary that cools the gas. All of these factors change the fundamental behavior of the combustion compared to that of a flame propagating freely across the chamber, for which the described fractal mechanism of flame propagation is no longer valid.

According to [24], wall-combustion burning rate can be described by an exponential decay. In the derived equation (14), m_b is the mass of fuel burned and τ is the characteristic time scale of the above process, derived from the characteristic length scale L_I and characteristic velocity u' .

$$\left(\frac{dm_b}{dt} \right)_{wall-combustion} = \frac{m - m_b}{\tau} \quad (14)$$

The overall burning rate can be consequently defined as a weighted mean of the two described combustion rates:

$$\begin{aligned} \left(\frac{dm_b}{dt} \right)_{overall} &= (1 - w_2) \cdot \left(\frac{dm_b}{dt} \right)_{fractals} \\ &+ w_2 \cdot \left(\frac{dm_b}{dt} \right)_{wall-combustion} \end{aligned} \quad (15)$$

Under usual conditions, a great portion (30-40%) of the unburned mixture burns in this particular 'wall-combustion' mode. The mass fraction (mf_{wall}) of fuel burned at which wall combustion starts is model parameter defined by the user.

METHODOLOGY OF PARAMETER IDENTIFICATION AND STATISTICAL ANALYSIS

0-D/1-D Model Calibration and Validation

As the SCRE is a single cylinder engine, the intake and exhaust system is relatively simple. The engine is equipped with variable valve timing (VVT) and a direct-injection combustion system. The latter fact brings special challenges to modeling (especially 0-D approaches) due to the more complex geometry of the DI piston bowl. All important information regarding geometry (including valve train properties) was well known and it was applied consistently in both 0-D/1-D tool [2] and 3-D CFD one [1]. Based on that,

the model was calibrated in detail to match with experimental data for the average cycle of both LP 2504 and LP 2735.

The calibration procedure included the following steps:

1. High Pressure Analysis
2. Calibration of Gas Exchange
3. Calibration of Turbulence based on CFD Data
4. Calibration of the Combustion Model Parameters for individual Engine Cycles

Ad 1: The high pressure analysis calculates the in-cylinder heat release based on energy balance including sub-models for heat transfer and blow-by. For the present study a 2-zone analysis was selected, as it matches the thermodynamic approach of the fractal combustion model and therefore allows a direct comparison of heat release curves in the further steps. The first step also includes calibration of heat transfer and blow-by model as well as the adaption of the motored pressure curve. This calibration was done using the tool labeled GCA Burn [25].

Ad 2: The physical modeling of the combustion relies on the estimation of the in-cylinder turbulence level which predominantly depends on the intake and exhaust processes. Therefore it is not enough to focus on the high pressure cycle. Figure 14 (positioned in Appendix) shows the 1-D engine model of the SCRE. It includes all relevant geometrical data of the test-bed engine. In this step the model uses as an input the analyzed heat release from step 1. Focus of the calibration is to match measured mass flow, intake/exhaust temperatures from various sensor position and finally in-cylinder pressure traces.

Ad 3: As soon as gas exchange process is correctly predicted, it is possible to calibrate the turbulence model, which used intake and exhaust mass flow as an input. Target function for the matching is the TKE characteristics from the 3-D CFD [1] results (FIRE [1] is 3-D CFD commercial tool from AVL which was used for all LES/RANS 3-D CFD calculations within EU project LESSCCV, hence the knowledge obtained from these data was applied during the course of the work presented in this paper). As shown in Figure 3, a good agreement of TKE characteristics is achieved, especially during the relevant combustion phase (-20 to $+50$ degCA). One can also see that the intake peak is over-predicted in the 0-D model. Still, if we emphasize that the high flow velocity at IVC is a major source of in-cylinder turbulence, the position of the intake peak would need to occur near IVC which is the case in the 0-D model. The dissipation of turbulent kinetic energy during compression as well as the increase around firing TDC are well in line with the 3-D results. Especially at these low engine speeds, the results of the 0-D model are reasonable.

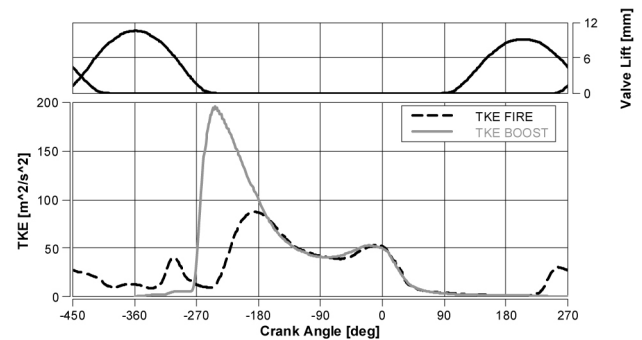


Figure 3. Simulation of TKE Comparison of 3-D and 0-D model.

Ad 4: This step is described in detail in the following section.

Automatic Calibration for Individual Cycles

Once the 0-D/1-D model of the SCRE was calibrated with respect to average cycle data, it was necessary to match each individual engine cycle (in total 200 cycles were processed) - it was verified that perturbation of certain combustion model parameters is sufficient to model CCV phenomenon. The matching itself was achieved by application of optimization (genetic algorithm) - an automatic procedure was necessary due to high amount of data which had to be processed. The optimization consists in finding such values of the combustion model to minimize the difference between measured and predicted in-cylinder pressure (only high-pressure phase of ICE cycle was considered, i.e., all valves are closed).

It was already mentioned above that variation of combustion model parameters are supposed to be sufficient when modeling CCV of SI engine. This means that when performing matching procedure, all model parameters were fixed with the exception of selected parameters of the fractal combustion model. Generally speaking, each combustion model requires different parameters to be varied to model CCV. To determine which parameters were to be varied, a sensitivity study (of many parameters) was performed. After analyzing the results, it was decided that three parameters are sufficient.

The parameters were selected in order to emphasize the individual physical mechanisms of the combustion process, i.e. Turbulence, Ignition Process, and Combustion. This would allow identifying physical effects and their interrelations by a statistical analysis of the parameter distributions and their cross-correlations. For the selection of appropriate parameters, sensitivity analyses were performed.

For the turbulence model, the turbulence length scale parameter c_L from equation (11) (labeled 'Bozza_Cls') was selected as it controls turbulence intensity, hence combustion speed. Higher values lead to faster combustion. Second, ignition delay parameter $r_{f,ref}$ from equation (13) (labeled

'**fract_ign_rad**') is selected, as it controls the delay between ignition and start of combustion ('visible' rate-of-heat-release). Higher value causes longer ignition delay. The flame-front driven combustion process is modeled on a sound physical basis. Therefore it does not include appropriate empirical tuning parameters. What regards the combustion progression, the main simplification of the 0-D approach is the simplified representation of the combustion chamber geometry. The current engine in particular has a quite complex chamber geometry due to the characteristic piston bowl for the direct injection combustion system. Therefore, as the third parameter the wall-combustion mass fraction $m_{f_{wall}}$ (labeled '**fract_mf_wall**') was selected. This enables to shape RoHR curve, especially in the late phase of combustion process.

For calibration of the RoHR for all 100 cycles of each of the two operating points, an automatic optimization procedure was applied. The criterion of automatic procedure was to minimize integral of square difference between experimental data and simulation - value of the red curve in left-bottom sub-figure of [Figure 15](#) (located in Appendix) at crank angle of 900 degCA; it is defined as

$$\int_{540}^{900} (p - p_{exp})^2 d\alpha$$

The optimization was then done using a genetic algorithm implemented in AVL BOOST Design Explorer. The quality of automatic matching procedure can be estimated from [Figure 15](#) (in Appendix), which shows comparison of experimental data with simulation ones. In general a good agreement of RoHR and cylinder pressure could be achieved for all considered engine cycles.

Statistics of Fractal Combustion Model

This section deals with statistical evaluation of the data obtained from calibration procedure of all individual cycles. Generally speaking, two different types of information were evaluated. First, probability distribution of each of the combustion model parameters was derived (c.f. [Figure 16](#) in Appendix). Second, cross-correlations (c.f. [Figure 4](#)) between any two parameters were evaluated to understand the relations among them.

First information deals with probability distribution functions of the combustion model. They are shown in [Figure 16](#) (located in Appendix). An obvious conclusion is that statistics of stable point (LP 2504) are closer to symmetrical distribution. Moreover the parameter range (horizontal axis) is narrower when compared with unstable operating point (LP 2735).

There is also possibility to look at the correlation among the parameters of the combustion model - this is shown in [Figure 4](#). It seems that when dealing with low CCV case (LP 2504), the correlation coefficients are relatively low - the combustion model parameters are almost statistically independent, hence they can be generated by random number generators independently. On the other hand, high CCV case

(LP 2735) features strong correlation of certain parameters, most noticeably between **Bozza_Cls** and **fract_ign_rad**, and also between **fract_mf_wall** and **fract_ign_rad**. These parameters are definitely not statistically independent (hence they cannot be generated randomly). The correlation interpretation from the physics point of view is the following. The correlation coefficient between **Bozza_Cls** and **fract_ign_rad** is positive, hence when the former one is increased it is very likely that the latter is increased as well. Increasing **Bozza_Cls** means to increase turbulence kinetic energy (which also leads to faster combustion) in the cylinder while increasing **fract_ign_rad** means to increase the 'delay' between spark discharge and start of combustion (identified by pressure increase when compared with motored regime). In other words, the positive correlation means that when in-cylinder turbulence is high (hence faster combustion), the combustion will start later - this may also suggest that high turbulence level slows initial phase of flame kernel development. The negative correlation between **fract_ign_rad** and **fract_mf_wall** means that when combustion starts later, the flame is constrained by the wall earlier in terms of amount of burnt fuel fraction.

LP2504 (low CCV)	Bozza_Cls	fract_ign_rad	fract_mf_wall
Bozza_Cls	1.000	0.220	-0.207
fract_ign_rad	0.220	1.000	-0.102
fract_mf_wall	-0.207	-0.102	1.000

LP2735 (high CCV)	Bozza_Cls	fract_ign_rad	fract_mf_wall
Bozza_Cls	1.000	0.572	-0.173
fract_ign_rad	0.572	1.000	-0.574
fract_mf_wall	-0.173	-0.574	1.000

Figure 4. Correlation coefficients among considered parameters of the combustion model - top table concerns stable point (LP 2504) while the bottom one shows unstable point (LP 2735); each number (in the table) represents the correlation coefficient between selected 2 parameters (their label is printed in the 1st row/column).

Especially the correlation between turbulence (**Bozza_Cls**) and 'ignition delay' (**fract_ign_rad**) is important. It seems that turbulence causes complex interactions in the cylinder - it slows the early flame kernel development phase (i.e., increasing 'ignition delay'). However when there is a stable initial flame, it burns faster (due to higher flame propagation speed which is mainly driven by turbulence level). This conclusion is yet to be verified by detailed 3-D CFD calculations using LES approach. The other cross-correlation (between **fract_ign_rad** and **fract_mf_wall**) seems to have only a minor effect on in-cylinder pressure traces - it may be important to fine-tune the final phase of combustion process.

Imposing of Statistics into 0-D/1-D Tool

Once the statistics are known, they can be imposed into the 0-D/1-D tool. The results (regarding individual cycles) can be imposed directly, however this approach would not