

# LES Multi-Cycle Analysis of the Combustion Process in a Small SI Engine

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## ABSTRACT

Large eddy simulations (LES) of a port-injected 4-valve spark ignited (SI) engine have been carried out with the emphasis on the combustion process. The considered operating point is close to full load at 3,500 RPM and exhibits considerable cyclic variation in terms of the in-cylinder pressure traces, which can be related to fluctuations in the combustion process. In order to characterize these fluctuations, a statistically relevant number of subsequent cycles, namely up to 40, have been computed in the multi-cycle analysis. In contrast to other LES studies of SI engines, here the G-equation (a level set approach) has been adopted to model the premixed combustion in the framework of the STAR-CD/es-ICE flow field solver. Tuning parameters are identified and their impact on the result is addressed. Based on the motivation of decreasing the computational cost for multi-cycle LES and thus bringing such calculations closer to industrial applications, different meshing strategies are subsequently investigated. These include two grid resolutions, and in particular the impact of local refinements in the spark plug vicinity since the predictions showed a high sensitivity of the maximum in-cylinder pressure on the early stage of combustion. Although questionable from an LES fundamental perspective, the impact of employing only one half vs. a full cylinder representation has also been explored. The predictions have been validated by means of pressure and heat release evolutions and in general, good agreement with the experiment in terms of the fluctuation bandwidth has been found. However, the extreme cycles are over predicted by the model resulting in an increased standard deviation of the pressure trace. The dependency of the mean pressure trace on the mesh representation using the Damköhler flame speed close has been illustrated. In addition statistical methods have been applied to characterize the observed cycle to cycle variations (CCV) in the simulation and experiment. The highly resolved results of the LES are probed for correlations between flow variables such as velocity magnitude, subgrid turbulent kinetic energy as well as temperature and residual gas and characteristic combustion parameters. Thus local dependencies have been pointed out which primarily influence the early combustion stage.

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## INTRODUCTION

For the past decades, engine development goals have been largely targeted towards meeting regulations concerning pollutants, e.g. unburned hydrocarbons, carbon monoxide, NO<sub>x</sub> as well as particulate matter, mandating a continuous decrease in emissions levels. More recently, environmental concerns have led to the adoption of limits concerning also CO<sub>2</sub> and hence directly linked to engine efficiency. Concepts such as downsizing of powertrains are nowadays common practice for engine developers to further improve fuel economy. To maintain a given power output, downsizing is routinely combined with turbocharging leading to engine operation considerably closer to the design limits compared to conventional concepts. These limits are not defined by the mean cycle but rather by the extreme cycles. As discussed e.g. in [1], fast burning cycles are the ones most prone to knock and hence limit both compression ratio as well as spark advance. Slow burning cycles on the other hand are

responsible for unburned hydrocarbons in the exhaust gas. Understanding the nature and origin of fluctuations which lead to such extreme cycles can hence be viewed as a prerequisite for advances in future engine design.

Unfortunately, cycle-resolved diagnostics in reciprocating engines are very challenging, in particular if optical access is required and localized effects need to be detected. Additional difficulties arise in the identification of influences from changes to individual parameters since many processes exhibit strong coupling. Numerical tools have therefore received increasing attention in recent years towards complementing experimental investigations. For the calculation of average engine cycles, Reynolds Averaged Navier Stokes (RANS) based simulation approaches are now routinely used in the engine development process; the ensemble averaged nature of the predictions lends itself ideally to a direct comparison with ensemble measurements such as pressure traces. More recently Large

Eddy Simulation (LES) - which has seen numerous application to flames, gas turbine and aircraft engine combustors with fixed geometries using a broad variety of combustion modeling approaches as reviewed in [2] - is emerging also for time-varying geometries in IC engines or engine-like configurations. Since LES applies spatial filtering in lieu of averaging, part of the turbulence spectrum is resolved and LES hence has shown considerable promise to capture cycle-individual phenomena. A review paper by Celik et al. [3] addressed the opportunities of LES in analyzing in-cylinder flows. The feasibility of LES for engine simulations was underlined and crucial problems were identified, e.g. computational cost for multiple cycle calculations as well as the need for better subgrid models in order to improve the quality of predictions. A recent review paper presented by Rutland [4] addressed the suitability and potential for the most important submodels in the context of reactive engine LES, and provided recommendations which have been widely followed by this paper. As pointed out in [4] improved LES submodels which allow the use of RANS like grids show considerable potential with respect to computational cost as opposed to the usually adopted fine resolved grids necessary for simple submodels.

Haworth [5, 6] was one of the first in 1999 who used LES to simulate the piston-cylinder assembly experimentally studied by Morse et al. [7]. The results showed a better agreement in terms of mean and RMS velocities compared to RANS simulations. A cold-flow LES simulation of a motored optical single cylinder engine setup was published by Enaux et al. [8], reporting good agreement of the mean flow field and cyclic variations compared to PIV measurements. A study which investigates the cold flow and hollow-cone gasoline spray injection was presented by Goryntsev [9]. It was found that the cyclic fluctuations were mainly attributed to the intake flow compared to the influence of the fuel injection. Early attempts in simulating the combustion process of an IC-engine in the LES context have been reported 2007 by Thobois [10]. The premixed combustion has been modeled using a thickened flame LES model (TFLES), where the reaction zone is artificially thickened by modifying turbulent transport coefficient as well as the chemical source term which allows to fully resolve the propagating flame front, which otherwise would not be possible for reasonable grid sizes. Since the TFLES model has its origins in the study of aeronautical turbine combustion which takes place at almost constant pressure and temperature for the unburned mixture, the model had to be adapted in order to account for the changing temperature and pressure during the combustion stroke in an IC-engine. This has been realized by using a reduced chemical scheme in order to calculate the adiabatic flame temperature and laminar flame speed for a range of pressure and temperature. The ignition of the premixed gas was handled by a simple energy deposition model (ED). Although only 4 cycles were simulated, cyclic variations of the pressure trace due to combustion have been reported, demonstrating the feasibility LES for premixed IC-engines. The TFLES model in combination with the ED ignition has also been used by Enaux et al. [11] in order to further study cycle-to-cycle variations (CCV). An SI engine

specifically designed towards generation of LES validation data was considered by computing 26 consecutive cycles. The individual cycles were found to be independent of their preceding cycle in terms of peak in-cylinder pressure in the experiment as well as in the simulation. The early combustion phase defined by 2% burned fuel mass already expressed a considerable amount of CCV as well as a clear relation to the overall combustion at 90% converted fuel. These variations of the early flame stage were found to be primarily controlled by the local variations of the velocity field around the spark plug. Based on the same modeling approach, Granet et al [12] compared 25 and 50 cycles for a stable and an unstable operating point, respectively, to experimental data. The importance of the early flame propagation was further underlined while the CCV of the simulation showed good agreement with the experiment for both operating points. The unstable operating points lead to cycles with incomplete combustion which has been addressed in detail.

The origins of CCV in a production type SI engine were addressed by Vermorel et al in [13] using an extended coherent flame model for LES (ECFM) for the modeling of the premixed combustion. The CFM model is based upon the solution of a transport equation for the flame surface density. Other than the commonly employed ED ignition, an Eulerian arc and kernel tracking ignition model (ATKIM) was used which allowed to track the otherwise unresolved initial flame kernels. For the considered 10 cycles the tumble motion generated during the intake and its breakdown influencing the turbulence intensity at ignition was identified as the root cause for the observed CCV in the given setup. Local variations in the mixture did not reveal a significant impact on the combustion. Fontanesi et al. compared different ignition treatment strategies in the ECFM context for a DISI engine in [14]. The ATKIM approach on a normal mesh was compared to a locally refined mesh where the ECFM model was used to treat the kernel growth from the very beginning. The ATKIM model overpredicted the ignition whereas the no ignition model approach showed better agreement with the measured pressure traces.

Other than the commonly used and established ECFM and TFLES models used for the modeling of premixed combustion, the G-Equation combustion model has been adopted in this study to assess its capability of reproducing the cyclic behavior in small-scale production type engine. First results with the G-Equation in the LES context have been reported by Schmitt et al. in [15].

## SETUP

### Experimental Setup

Measurements have been carried out on a Swissauto Wenko 250cc single cylinder four-stroke spark ignition engine for which specifications are provided in Table 1. The combustion chamber has a pent roof with the spark plug centered over the four valves, slightly off center. The in-cylinder pressure-

(144 cycles per operating point) is recorded every 0.2°CA with a piezo-electric pressure sensor (Kistler 6052) while absolute values of the intake and exhaust pressures are measured with a Kistler 4075 and 4049 transducer, respectively. Fuel consumption is determined using a coriolis mass flow meter by Endress+Hauser Cubemass with a tolerance of  $\pm 1\%$ . The engine is connected to an APIcom FP75 eddy current brake. The ECU is a Bosch MS4 Sport. The temperatures in the intake and exhaust are measured using radiation protected thermocouples located at the same position as the pressure transducers.

All measurements are carried out using regular gasoline (RON 95). All operating points have a stoichiometric (lambda-controlled, Bosch LSU 4.9) homogeneous mixture. For the considered simulations the operating point is at 3,500 rpm and 9.9 bar BMEP which is close to full load.

Table 1. Characteristic data for the investigated engine

| Swiss Auto 250ccm   |                             |
|---------------------|-----------------------------|
| Displaced volume    | 250 cc                      |
| Stroke              | 56 mm                       |
| Bore                | 75 mm                       |
| Connecting Rod      | 95 mm                       |
| Compression ratio   | 12.5:1                      |
| Number of Valves    | 4                           |
| Number of Cylinders | 1                           |
| Fuel Injection      | Port Fuel Injection (4 bar) |

## Numerical Setup

Commonly used and simple models were used as far as possible in order to assess the possibility of pushing LES towards industrial applications by lowering the computational cost as well as modeling effort and tuning. This procedure allows investigating the performance and validity of such a simplified LES and can be used as a baseline for further studies using more sophisticated modeling.

## Solver

The flow field solver STAR-CD in conjunction with the es-ICE plugin v4.18 by CD-Adapco is used to solve the compressible Navier-Stokes equations and model the engine under investigation. The momentum equations as well as the subgrid-scale turbulent kinetic energy equation are discretized with a second order monotone advection and reconstruction scheme (MARS). The PISO algorithm is used for the pressure velocity coupling. The time step has been set to 0.1°CA throughout most of the cycle. Due to high pressure gradients during the exhaust valve opening, the time step has been considerably reduced towards 0.02° CA during this period. The time step size has further been monitored in terms of averaged and maximum CFL number to ensure numerical stability.

## Turbulence Model

By applying the spatial LES filtering to the Navier-Stokes equations, the so called subgrid turbulence stress tensor  $\tau_{ij,SGS}$  arises as an additional term in the filtered momentum equation which is not determined by the present equations [16].

$$\frac{\partial \tilde{u}_i}{\partial t} + \frac{\partial \tilde{u}_i \tilde{u}_j}{\partial x_j} = -\frac{1}{\bar{\rho}} \frac{\partial \bar{p}}{\partial x_i} + \nu \frac{\partial^2 \tilde{u}_i}{\partial x_j \partial x_j} - \frac{\partial \tau_{ij,SGS}}{\partial x_j} \quad (1)$$

The subgrid stress tensor represents the influence of the unresolved scales on the resolved scales. In order to close the filtered Navier-Stokes equations,  $\tau_{ij,SGS}$  needs to be modeled.

For the modeling of the subgrid turbulence in the course of this paper the one-equation k-model is chosen. According to the review presented in [4], this model was found to incorporate more physical processes, namely the convection, production and dissipation of the SGS turbulent kinetic energy ( $k_{SGS}$ ) compared to the commonly used Smagorinsky model. Good results were also reported in [5] comparing predictions of a valve-piston assembly using the same model to LDA measurements from [7].

In the k-equation model a transport equation for  $k_{SGS}$  is solved:

$$\frac{\partial \bar{\rho} k_{SGS}}{\partial t} + \frac{\partial \bar{\rho} \tilde{u}_j k_{SGS}}{\partial x_j} = -\tau_{SGS,ij} \tilde{S}_{ij} - C_1 \rho \frac{k_{SGS}^{3/2}}{\Delta} + \frac{\partial}{\partial x_j} \left[ C_2 \bar{\rho} \Delta k_{SGS}^{1/2} \frac{\partial k_{SGS}}{\partial x_j} \right] \quad (2)$$

where  $C_1$  and  $C_2$  are model constants and  $\Delta$  denotes the filter width.

Based on  $k_{SGS}$  the turbulent viscosity is computed using an algebraic relationship as:

$$\nu_T = C_k \Delta \sqrt{k_{SGS}} \quad (3)$$

The subgrid-scale stress tensor  $\tau_{SGS,ij}$  is closed using the Boussinesq assumption

$$\tau_{SGS,ij} = -2\nu_T \tilde{S}_{ij} + \delta_{ij} \frac{2}{3} k_{SGS} \quad (4)$$

As is evident from the inspection of Equation (2), the subgrid turbulent kinetic energy strongly depends on the filter width  $\Delta$ . As is common practice with finite volume codes, the LES filter width is directly related to the cell volume  $V$  by

Therefore a grid dependency of  $k_{SGS}$  is introduced by the fact that a higher spatial grid resolution leads to a smaller filter width and thus to more resolved and less unresolved kinetic energy, reflected by lower values for  $k_{SGS}$ .

## Combustion Model

In order to account for the premixed combustion, the G-Equation model is deployed. A nonreacting scalar  $G$  is introduced. The isosurface  $G = G_0$  divides the domain into an unburned region with  $G < G_0$  and a burned region with  $G > G_0$  which is called a level set approach. Williams [17] introduced a transport equation for  $G$  valid on the  $G = G_0$  isosurface, whereas Peters [18] provided a closure for the G-Equation in the RANS context. Pitsch [19] presented a filtered equation for the LES context:

$$\bar{\rho} \frac{\partial \tilde{G}}{\partial t} + \bar{\rho} \tilde{u} \cdot \nabla \tilde{G} = \bar{\rho} s_T |\nabla \tilde{G}| - \bar{\rho} D_t \tilde{\kappa} |\nabla \tilde{G}| \quad (6)$$

Outside of the  $G = G_0$  isosurface the  $G$ -field is assigned as a signed distance function. An additional transport equation is solved for the variance  $G'$  which is used to calculate the turbulent flame brush thickness.

In the current implementation, the turbulent flame speed  $s_T$  is by default modeled according to Damköhler, based on the laminar flame speed  $s_L$  and the fluctuation velocity  $u'$  as a turbulent scaling factor:

$$s_T(\phi) = s_L(\phi) \left[ 1 + A \left( \frac{u'}{s_L(\phi)} \right)^{5/6} \right] \quad (7)$$

The so called flame speed coefficient  $A$  is a model parameter which needs to be adjusted; in a RANS context  $A$  usually takes values between 2 and 3. In the present LES framework, the fluctuation velocity  $u'$  is related to the *subgrid-scale* turbulence kinetic energy by the assumption of isotropic turbulence:

$$u' = \sqrt{\frac{2}{3} k_{SGS}} \quad (8)$$

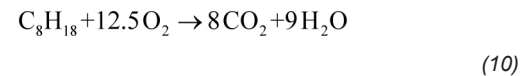
Due to the smaller amount of unresolved scales on the subgrid in a LES,  $A$  has been adjusted to a constant value of 8.5 for all considered configurations in this paper.

The laminar flame speed  $s_L$  is based on the empirical correlation of Gülder [20] which is valid for a broad range for IC engine applications and, aside the fuel type, considers also the influence of the local equivalence ratio, the temperature, pressure as well as the presence of any residual gas.

$$s_L(\phi) = ZW\phi^\eta \exp(-\xi(\phi - 1.075)^2) \cdot \dots \cdot \left( \frac{T_u}{T_{ref}} \right)^\alpha \left( \frac{P}{P_{ref}} \right)^\beta (1 - ffY_{EGR}) \quad (9)$$

The fuel-dependent constants for gasoline can be found in [Table 2](#), while the model constant  $ff$  is set to a default value of 2.3.

Iso-octane has been chosen as a single-component surrogate fuel for gasoline. The tabulated reference enthalpy has been adjusted in order to represent a lower heat of combustion of 42.5 MJ/kg in order to meet the same amount of released heat as in the experiment with gasoline. In order to minimize the modeling effort and computational cost the fuel is converted in a basic global one step reaction:



No wall quenching effects have been considered since the default implementation of the G-Equation has been used.

Table 2. Fuel-dependent constants of the Gülder correlation for the laminar flame speed, depicted for gasoline

| Z | W      | $\eta$ | $\xi$ | $\alpha$ | $\beta$ |
|---|--------|--------|-------|----------|---------|
| 1 | 0.4658 | -0.326 | 4.48  | 1.56     | -0.22   |

## Ignition Treatment

In the present study, ignition is treated numerically by initializing a flame kernel sphere with predefined radius and flame brush thickness during the ignition time step at the ignition location. The ignition radius and thickness are set to constant values of 0.5 mm and 0.1 mm, respectively. Such 'energy deposition' approaches have also been followed in similar studies, e.g. [12, 21]. No ramp-up is used, therefore the laminar to turbulent transition is neglected and the flame is propagating with turbulent burning speeds from the very beginning on. While more elaborate approaches have been presented in [22] and specifically for G-equation e.g. in [23], here this simplifying assumption has been adopted due to the previously mentioned target of minimizing the model parameter tuning and computational cost.

## Geometry and Simulation-Setup

### Geometry

The simulation domain consists of the cylinder with moving piston and moving inlet and exhausts valve as well as parts of the inlet and exhaust channel as represented in [Figure 1](#). Due to the location of the spark plug on the cylinder axis, the problem domain is symmetrical in terms of the combustion chamber and piston as well as the intake and exhaust port and valves which are not expected to introduce any form of swirl. In

order to reduce the computational cost and thus trying to push LES towards industrial applications, the possibility of utilizing the symmetrical character of the engine has been investigated by using also a half cylinder model versus the full cylinder representation.

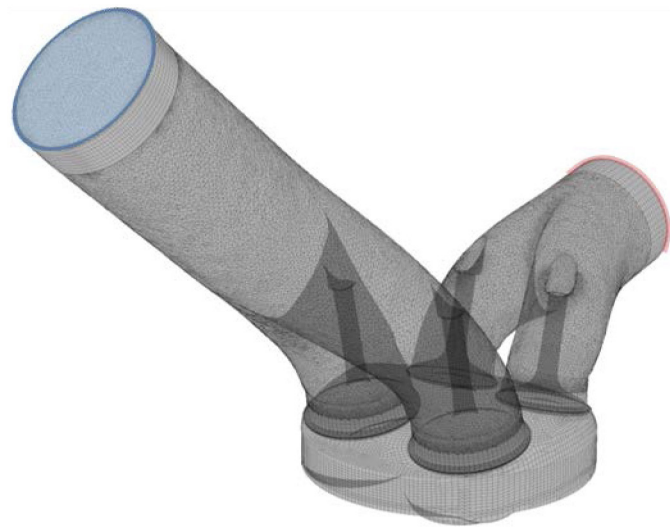


Figure 1. Full model representation of the simulation domain consisting of cylinder, inlet and exhaust channel as well as valves at 40°CA BTDC. Inlet boundary indicated blue, outlet boundary red.

## Mesh

A nominal mesh resolution with an average cell size of 1.2 mm has been chosen for the full cylinder and half cylinder, referred to as the full normal and half normal mesh, respectively. In order to study the impact of mesh resolution on the simulation results, a mesh with a local refinement around the spark zone as well as a mesh refined throughout the entire computational domain have been generated for the half cylinder, named half refined and half fine, respectively. The characteristics of those meshes are indicated in Table 3 whereas a detail of the spark zone can be seen in Figure 2 where the crosshair indicates the ignition location.

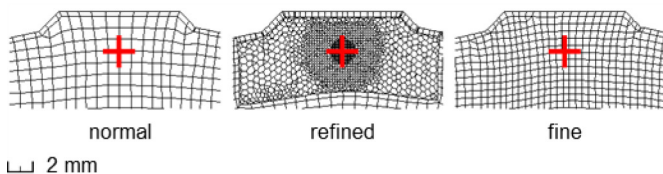


Figure 2. Detail of ignition location for the half normal, half refined and half fine mesh

The aforementioned local refinement consists of polyhedral cells at a size of 0.1 mm within a sphere of radius 1 mm around the ignition location and 0.2 mm in an outer sphere of radius 3 mm. The rest of the insert which reaches around 15 mm in width is filled with polyhedral cells with a size of 0.5 mm. Outside of the insert the refined mesh is identical to the normal mesh and therefore coarser than the fine mesh, whereas inside the refinement it is considerably higher resolved compared to the fine mesh. Since both the outer hexahedral mesh and the

polyhedral local refinement are unstructured, no additional influence on the solution is expected other than by the change in the cell size itself.

The used cell size of 1.2 mm represents a rather coarse computational mesh compared to the usually used 0.5-1 mm found in literature, whereas the fine mesh represents the commonly deployed grid size. However, the coarse mesh comes at considerably decreased computational costs, as it can be found in Table 3. When moving from the half mesh to the full mesh an increase of the computational cost by a factor of about 3 can be observed counterintuitively. This can be related to smaller time steps during the exhaust stroke in order to maintain a stable solution due to a more fluctuating exhaust flow on the full cylinder mesh. Furthermore it has to be noted that the computational cost does not necessarily scale linearly with the number of cells even when keeping the number of cells per CPU in the optimal range.

The cylinder including piston and valves has been meshed using hexahedral cells using the trimming method and a prism layer at the walls. The static part, namely the inlet and exhaust channels and the local refinement has been meshed using polyhedral cells. While other studies include the details of the spark plug geometry, here the geometric features of the spark plug housing and the electrodes have been neglected similar to the setup seen in [10, 12, 13]. This simplification has been chosen since the ignition modeling is realized in a basic numerical manner as seen before. Additionally small cells can be saved by neglecting the spark plug geometry, following the idea of minimizing the computational cost. The impact of the spark plug geometry can be assessed in a later stage by simply changing the local refinement insert.

Table 3. Characteristics of the considered meshes

| Mesh         | Cylinder | Average Cell Size | Cells at TDC | Cells at BDC | CPUh / Cycle | Cycles |
|--------------|----------|-------------------|--------------|--------------|--------------|--------|
| Full normal  | Full     | 1.2 mm            | 260'000      | 500'000      | 1'024        | 31     |
| Half normal  | Half     | 1.2 mm            | 130'000      | 250'000      | 304          | 45     |
| Half refined | Half     | 1.2 mm            | 170'000      | 290'000      | 416          | 37     |
| Half fine    | Half     | 0.6 mm            | 210'000      | 1'000'000    | 2'560        | 27     |

## Boundary and Initial Conditions

Crank angle resolved and phase averaged measured pressure boundary conditions - representing the mean cycle - were imposed at the inlet and outlet for all cycles. No perturbations in the boundary conditions are imposed in order to study the cyclic variability caused solely by the present turbulent flow field. For the same reason, none of the available synthetic eddy models for the inlet boundary has been activated in the modeling framework. This procedure follows the work presented in [14]. The fresh gas in the intake is initialized as perfectly premixed air ( $N_2$ ,  $O_2$ ) and  $C_8H_{18}$  at stoichiometric conditions ( $\lambda=1$ ) and the same mass fractions of 0.721, 0.219 and 0.060, respectively, have been prescribed at the inlet boundary. This assumption is justified by the fact that the fuel is

injected into the intake port far upstream of the inlet valve for the investigated engine configuration and the presence of a lambda controller ensuring stoichiometric conditions. The temperature in the intake has been adjusted towards 315 K in order to meet the pressure trace of the experiment during compression until ignition. This approach has been chosen since a motored non-fired operating point is not available for the considered engine. The walls are treated isothermal at fixed temperatures. The combustion dome and piston crown regions are set to a temperature of 470 K, whereas the cylinder liner is set to 400 K based on pressure data analysis values calculated by means of an in-house two-zone model [24]. The near-wall region is treated using a fine prism layer and an estimated wall friction velocity  $u_\tau$  which allows for calculation of wall heat losses by means of the correlation of Angelberger as documented in [25].

Between two consequent cycles the state at the end of the preceding cycle is retained for the initial conditions of the next cycle. The initial conditions of the first cycle however are set to homogeneous zero velocity conditions with the thermodynamic state based on the measured mean values in terms of pressure and temperature. Since the simulation is initialized shortly before IVO, the combustion chamber and exhaust port are set to a fully burned state regarding the species composition, while the intake port is set to a fresh mixture composition as reported above. For the post-processing and statistical evaluations the first cycle is discarded in order to remove the influence of these initial conditions.

For the half cylinder models a symmetry boundary has been introduced in the symmetry plane. This boundary allows no velocity components or gradients in the normal direction of the plane.

## RESULTS

### Cylinder Pressure

Cyclic variations can be characterized by the fluctuating pressure traces for consecutive cycles at constant operating conditions. Figure 3 represents the maximum, mean and minimum of the measured in-cylinder pressure based on 144 consecutive cycles in the experiment. The peak pressure minima and maxima span 27.1 bar.

For the considered setup spark timing is set to 30.4° CA BTDC. The pressure traces of the slowest and fastest burning cycles start to diverge after 15° CA BTDC as a result of the combustion process, whereas they are identical before. This observation suggests, that the cyclic variations in terms of measured in-cylinder pressure can be clearly related to variations in the combustion process.

Figure 4 indicates the predicted maximum, mean and minimum pressure trace for all four considered mesh configurations; the number of calculated cycles differs between the variants and is given in Table 3. As is evident from the large spreads in the

predicted pressure envelopes, the model is capable of reproducing variations in the combustion event from cycle to cycle with all four resolutions. No considerable differences can be seen as far as the maximum pressure traces are concerned for the half cylinder meshes. The discrepancies in the minimum traces are more pronounced, however. In terms of the mean pressure for the half cylinder, the fine mesh expresses slightly lower pressure levels whereas the locally refined mesh is at considerably lower levels compared to the basic mesh. All traces for the full model are clearly shifted towards lower pressures compared to the half models.

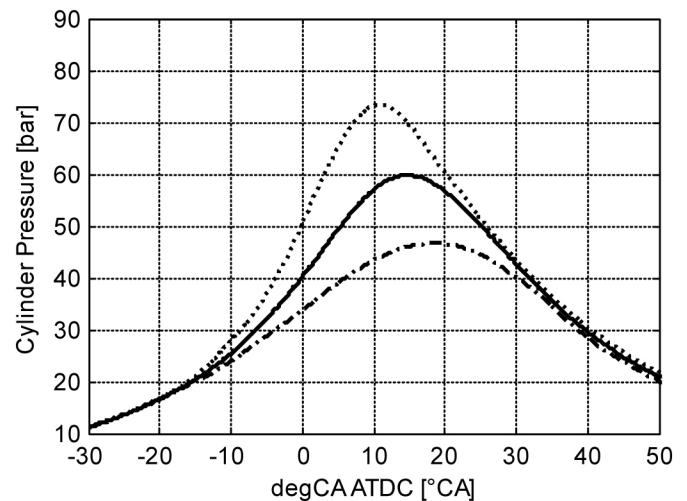


Figure 3. Maximum (dotted), mean (solid) and minimum (dash-dotted) in-cylinder pressure of 144 consecutive cycles in the experiment

An overall good agreement with the experiment in terms of spread between the peak values of the highest and lowest pressure traces can be found when comparing Figure 3 and Figure 4, for which the values can be found in Table 4.

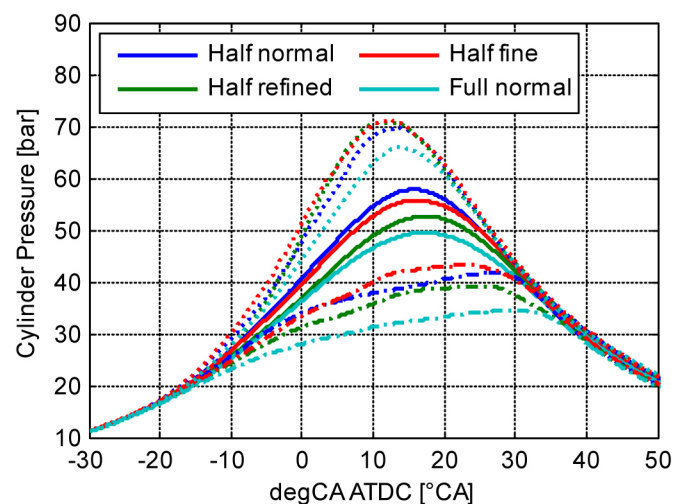


Figure 4. Maximum (dotted), mean (solid) and minimum (dash-dotted) in-cylinder pressure of the simulation for all considered mesh configurations