Quasi-dimensional model of an optically accessible spark ignition engine

Paolo Gobbato\textsuperscript{a}, Nicola Saccon\textsuperscript{b} and Adrian Irimescu\textsuperscript{c}

\textsuperscript{a} Department of Industrial Engineering, University of Padova, Padova, Italy, paolo.gobbato@unipd.it
\textsuperscript{b} Department of Management and Engineering, University of Padova, Vicenza, Italy, saccon.n@gmail.com
\textsuperscript{c} Istituto Motori – CNR, Naples, Italy, a.irimescu@im.cnr.it

Abstract:
Efficiency and torque maximization of internal combustion engines (ICEs) requires extensive experimental testing and complex instrumentation for in-cylinder measurements. On the other hand, engine modelling represents a concrete resource for designers to get information about engine operation. Early academic models have evolved into detailed multidimensional models able to provide reliable predictions of thermo-fluid dynamic phenomena occurring within engine cylinders and manifolds. However, if the objective of the model is to evaluate a large range of conditions, perform parametric studies and predict optimum engine settings, zero- and quasi-dimensional models still remain the preferable choice. In addition, several phenomenological sub-models required to complete the set of solving equations have been proposed in the literature for different engine designs and operating conditions.

This paper provides a skeletal set of equations to calculate in-cylinder quantities. The model includes a quasi-steady sub-model to simulate mass exchange across the intake and exhaust valves as well as phenomenological sub-models to calculate the heat release provided by combustion and the heat transfer to engine coolant. Calculated pressures are validated against experimental data acquired on a turbocharged single-cylinder spark ignition (SI) engine featuring a special piston with a quartz-windowed crown. Numerical results show that the model is able to capture the measured pressure trace with a good degree of agreement. The comparison suggests that the predictive capability of the model can be improved by considering blow-by and, secondarily, unsteady phenomena within the intake manifold.

Keywords:

1. Introduction
The design and development of internal combustion engines have been traditionally supported by rig testing. However, in recent decades, simulation tools have partly substituted experimental tests thanks to an increase of computational power. In addition, numerical models allow analysis in detail of the complex thermo-physical processes occurring within the combustion chamber without performing costly experimental in-cylinder investigations.

According to [1], numerical approaches can be classified (in order of increasing complexity and CPU demand) into thermodynamic and dimensional models. In thermodynamic models energy and mass conservation equations are solved to calculate the in-cylinder pressure and the work transferred to the piston. Additional sub-models are required to describe geometric parameters of the cylinder and valve ports, physical properties of the operating fluid, mass and heat transfer across the system boundary and the combustion process. These sub-models are frequently based on empirical or semi-empirical relations and for this reason are termed ‘phenomenological models’. On the other hand, in dimensional models (usually multidimensional models, such as computational fluid dynamic models, i.e., CFD), momentum equations complete the set of governing equations in addition to energy and mass conservation. To complete the classification of numerical approaches, it is also worth mentioning the quite new hybrid models, which combine some features of thermodynamic phenomenological models and fluid dynamic models [2–4].
In dimensional models, equations depend on the spatial coordinates of the system, whereas in zero-dimensional thermodynamic models, conservation of mass and energy are only dependent on time. Improved zero-dimensional models are sometimes referred to as quasi-dimensional [5,6]. This additional distinction appears confusing, because authors do not attribute the same meaning to the terms ‘zero’ and ‘quasi’. In the present context, zero-dimensional models are considered to have become quasi-dimensional models when sub-models account for specific geometric features of the system, although they retain independence from spatial coordinates. This means that the solution also depends on some specific geometric parameters of the engine (e.g., the area of the cylinder liner surface). The latter distinction especially concerns combustion sub-models. In zero-dimensional combustion models, the law of mass burning rate is derived from fitting of heat release experimental data (e.g., using a Wiebe function) or semi-empirical relationships [7]. When experimental data are unavailable, the fuel burning rate must be predicted by analytical and/or empirical relations [8,9]. Combustion models based on this approach can be considered quasi-dimensional, because they rely on estimation of turbulent length scales or flame development [5]. As an exception, the reaction rate in homogeneous charge compression ignition (HCCI) simulations is currently calculated by (zero-dimensional) chemical kinetics [10-12].

Zero-dimensional combustion models appear in the literature coupled to either single-zone or multi-zone representations of the combustion chamber. For instance, in [13,14] a Wiebe function is fitted to experimental data to determine the rate of mass fraction burned whereas the combustion chamber is treated as a single homogeneous system. In [7,15] and [6], the authors set a Wiebe function as model input, but consider the combustion chamber to be constituted by two and three separated open systems, respectively. Pressure traces calculated by single-zone and multi-zone approaches are compared in [10,12,16]. Although single-zone models lead to a certain over-prediction of in-cylinder pressure and temperature peaks, they provide a satisfactory calculation of the power cycle. However, at least a two-zone description of the combustion chamber is required when prediction of pollutant formation (e.g., nitric oxides) is one of the main goals of the model [12,17].

This paper presents a quasi-dimensional thermodynamic spark ignition (SI) engine model in which phenomenological sub-models are used to close the conservation equations of mass and energy. Of course, the approach is not original to the literature of SI engine modelling, especially when traditional fuels are considered. In fact, early single- or multi-zone combustion chamber models were proposed some decades ago [18,19]. Single-zone models are currently used for compression ignition (CI) [13,14] or HCCI engines [10-12], whereas two-zone modelling, at least (e.g., [5-7]), is usually adopted for SI engines. The originality of the present model lies in its particular application. Indeed, the model has been set up to analyse the operation of a research engine that allows optical investigations of combustion [20-22]. This single-cylinder engine was obtained from a four-cylinder passenger car engine by substituting the piston, adapting the cylinder to acquire flame images and boosting the intake air pressure.

Considering the modified design and materials of the combustion chamber, a “step-by-step” approach has been adopted to develop the model. The version presented in this paper includes a skeletal set of equations applied to the in-cylinder volume. In particular, the model includes a basic single-zone combustion sub-model based on a single Wiebe function, and a literature heat transfer sub-model. The gas-dynamic of intake and exhaust manifolds, cylinder blow-by, and abnormal combustion phenomena are neglected for now. However, unlike many other more complex modelling approaches proposed in the literature that focus their attention on the in-cylinder system alone [5-7,15], the present model is able to calculate the pressure trace of the complete cycle, because it also considers the mass exchanges at inlet and exhaust valves. The study is aimed at a critical evaluation of the prediction capabilities of the model in its skeletal form.

The paper is organised as follows. The engine and its experimental characterization, required to validate the numerical model, are briefly described in Section 2. Assumptions and model equations are presented in Section 3. Finally, in Section 4, the predicted pressure trace is compared with indicated measurements acquired during gasoline operation of the engine.
2. Experimental activities and data processing

Experimental activities on the SI engine analysed in the present study are extensively documented in many publications [20-22]. Essential information about the experimental facility and testing procedure are provided below to present the design and the operating conditions of the system modelled.

2.1. Engine design

The engine is a single-cylinder port fuel injection (PFI) SI engine that includes a cylinder head of a commercial turbocharged engine. The head has four valves and a centrally located spark plug. The original piston was replaced by a flat elongated piston with a quartz-windowed crown to make the combustion chamber optically accessible (Fig. 1). The piston is equipped with self-lubricating teflon-bronze composite rings to avoid window contamination by lubricant oil.

![Fig.1. Simplified schematic of the engine modelled.](image)

Table 1. Engine specifications (CAD: Crank angle degree; BBDC: Before bottom dead centre; TDC: Top dead centre; BTDC: Before top dead centre; ABDC: After bottom dead centre).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displaced volume, $V_{cu}$</td>
<td>399 cc</td>
</tr>
<tr>
<td>Bore, $B$</td>
<td>79 mm</td>
</tr>
<tr>
<td>Stroke, $C$</td>
<td>81.3 mm</td>
</tr>
<tr>
<td>Connecting rod, $b$</td>
<td>143 mm</td>
</tr>
<tr>
<td>Compression ratio (CR)</td>
<td>10:1</td>
</tr>
<tr>
<td>Number of valves</td>
<td>4</td>
</tr>
<tr>
<td>Exhaust valve open</td>
<td>27 CAD BBDC</td>
</tr>
<tr>
<td>Exhaust valve close</td>
<td>TDC</td>
</tr>
<tr>
<td>Inlet valve open</td>
<td>3 CAD BTDC</td>
</tr>
<tr>
<td>Inlet valve close</td>
<td>36 CAD ABDC</td>
</tr>
</tbody>
</table>

Design details of the engine are listed in Table 1. The value of the compression ratio (CR) shown in the table is calculated without considering crevices. Crevice volume is currently neglected when performing engine simulations, because it represents a small percentage of the clearance volume in stock production engines. Particular engines (typically the ones modified for research purposes) can feature special geometries with relatively high crevice volume. For the present study an actual CR of 9.2 was used for cycle simulation. This value agrees with the estimations provided in [23] for the same engine.

2.2. Experimental tests

Experimental tests were performed to acquire cycle-resolved flame images of gasoline and gasoline/butanol mixture combustion [20,21]. In addition, real-time in-cylinder pressures were measured every $\Delta \theta_u = 0.2$ crank angle degree (CAD) using a quartz pressure transducer flush-installed in the region between the intake-exhaust valves at the side of the spark plug. Each operating condition was tested as follows. After warm-up, the engine was worked in firing conditions for 300 consecutive cycles. Then, the engine was switched to motored conditions for another 100 cycles. An external device allowed control of the intake air pressure and temperature, within a range of 1000-2000 mbar and 290-340 K, respectively. Further details of the testing procedure are provided in [20].
In the present work, pressure traces recorded during a specific gasoline test are considered for model validation. The test was performed at a spark timing of 14 CAD before top dead centre (BTDC). This value avoids the occurrence of abnormal combustion phenomena with a slight safety margin (the knocking limit was evaluated around 16 BTDC [20]). Other operating conditions are specified in Table 2.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed, ( n )</td>
<td>2000 rpm</td>
</tr>
<tr>
<td>Throttle position</td>
<td>Wide open</td>
</tr>
<tr>
<td>Start of injection (SOI)</td>
<td>300 CAD BTDC</td>
</tr>
<tr>
<td>Duration of injection (DOI)</td>
<td>148 CAD</td>
</tr>
<tr>
<td>Fuel injected per cycle, ( m_f )</td>
<td>45.7 mg</td>
</tr>
<tr>
<td>Intake temperature</td>
<td>333 K</td>
</tr>
<tr>
<td>Intake pressure</td>
<td>130,000 Pa</td>
</tr>
</tbody>
</table>

2.3. Data processing

Pressure traces were processed to identify a mean cycle among the last 100 firing cycles measured. The pressure values of the mean cycle are the closest to the in-cylinder pressures that have the highest probability to occur during a cycle. For each \( \Delta \theta \), data were processed as follows:

- Calculation of maximum and minimum value of the \( N_c=100 \) cycle traces:
  \[
  p_{\text{max},N_c} = \max\{p_i; p_{N_c}\} \quad p_{\text{min},N_c} = \min\{p_i; p_{N_c}\}
  \]
  where \( p_1 \ldots p_{N_c} \) are the in-cylinder pressures of cycles 1… \( N_c \).

- Calculation of the pressure range between the maximum and minimum values:
  \[
  \Delta p_{N_c} = \frac{p_{\text{max},N_c} - p_{\text{min},N_c}}{N_c}
  \]

- Division of the pressure range previously identified into \( N_r=10 \) sub-ranges, each one of amplitude \( \Delta p_{N_r} = \frac{\Delta p_{N_c}}{N_r} \)

- Calculation of the number of cycles \( f_k \) having the trace included in the \( k \)-th pressure sub-range.

- Identification of the \( k \)-th sub-range including the maximum number of cycles:
  \[
  k_{\text{max}} : f_{k_{\text{max}}} = \max\{f_k\}
  \]

- Calculation of the mean pressure of cycles included in the sub-range \( k_{\text{max}} \):
  \[
  \bar{p}_{k_{\text{max}}} = \frac{\sum_{j=1}^{f_{k_{\text{max}}}} p_j}{f_{k_{\text{max}}}}
  \]

- Calculation of the squared deviation of each \( i \)-th cycle pressure (with \( i=1 \ldots N_c \)) from the mean value \( \bar{p}_{k_{\text{max}}} \):
  \[
  s_i^2 = (\bar{p}_{k_{\text{max}}} - p_i)^2
  \]
  Thus, for each \( i \)-th cycle, the total squared deviation is given by:
  \[
  S_i^2 = \sum_{i=1}^{N_c} s_i^2
  \]
where \( N_\theta = \frac{720}{\Delta \theta_d} \). The \( \ell \)-th cycle showing the lowest value of \( S^2 \) is identified as the mean cycle. For the set of cycles here considered, the mean cycle results to be the number 83. Its pressure trace is shown in Fig. 2. The start of combustion (slightly before TDC) and the exhaust valve opening (between 480 and 540 CAD) are clearly visible. Pressure fluctuation during intake is compatible with the inertia of the fresh charge entering the cylinder (valves open at 717 CAD and close at 216 CAD).

By applying the method originally proposed in [24] and formalised in [25], the mass fraction burned (MFB) can be calculated as a function of the CAD from in-cylinder pressures of cycle 83. Solid markers displayed in Fig. 3 show the crank angles at which a fuel mass fraction of 10%, 50% and 90% is burned, respectively. MFB data were used to determine the coefficients \( a \) and \( m \) of the Wiebe function, which is expressed as:

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

where \( x_b \) is the mass fraction burned, \( \theta \) is the crank angle, \( \theta_0 \) is the start of combustion, \( \Delta \theta_b \) is the combustion duration (\( x_b = 0 \) to \( x_b \approx 1 \)), and \( a \) and \( m \) are adjustable parameters, respectively. The function was fitted by requiring that the curve passes through the 50% of MFB and assuming that combustion starts at the ignition timing (14 CAD BTDC) and ends 60 CAD after the ignition. The calculated profile plotted in Fig. 3 was fitted with \( a = 16.16 \) and \( m = 3.7 \). It is worth noting that the combustion process in the actual engine shows a fast initial phase and a slower end-flame compared to a typical combustion process occurring within gasoline-fuelled engines [1]. As a consequence, a single Wiebe function cannot describe the MFB profile along the whole combustion process. However, as it was fitted, the Wiebe function is able to reproduce well the rate of heat release up to 50% of MFB at least, that is the fraction of the entire combustion phase that determines the magnitude and the crank angle of peak pressure.

**Fig. 2. In-cylinder pressure measured during the 83\textsuperscript{th} cycle (identified as the mean cycle).**
3. Engine model

The model is essentially based on the first law of thermodynamics and mass conservation with the addition of two empirical sub-models for combustion and heat transfer modelling. The control volume is limited to the cylinder, and the combustion chamber is treated as an homogeneous system. The geometry of the valves and cylinder is considered to calculate the mass transfer at system boundaries and heat transfer through the cylinder walls, therefore the model can be considered as quasi-dimensional. The set of equations presented in the following were implemented and solved in Matlab-Simulink® environment using the Dorman-Prince solution algorithm and setting the time step $\Delta t$ equal to $5 \cdot 10^{-5}$ s.

3.1. Model hypotheses

This section specifies the main hypotheses on which the thermodynamic model is based.

a) Pressure, temperature and composition of the fluid contained in the combustion chamber are assumed to be uniform at each simulation time step, as a straight consequence of the single-zone modelling approach.

b) The in-cylinder fluid is considered to be a perfectly mixed ideal gas composed of isoctane vapour, nitrogen, oxygen, carbon dioxide and water. Specific heats, enthalpy and internal energy of each chemical species are assumed to be a function of temperature and are calculated using the relations proposed in [18]. The thermodynamic properties of the fluid are calculated by mass weighted averaging performed at each time step.

c) Back flows from both the intake and the exhaust valves are admitted and modelled according to a perfect displacement hypothesis, whereas by-pass of fresh charge directly from intake to exhaust valves is neglected (this hypothesis does not penalise the model, because the engine under analysis features a very small valve overlap, see Table 1).

d) Blow-by, abnormal combustion phenomena and other physical processes not explicitly mentioned in the following are neglected.
3.2. Governing equations

3.2.1. Mass and energy equations

The engine cylinder is an open thermodynamic system in which the volume is continuously varied due to the piston motion (Fig. 4). The mass conservation and the first law of thermodynamics for an open system are, respectively:

\[
\dot{m} = \frac{d(\rho V)}{dt} = \dot{m}_i - \dot{m}_e \tag{9}
\]

\[
\frac{dU}{dt} = \dot{m}_i h_i - \dot{m}_e h_e + \dot{q} - \dot{w} \tag{10}
\]

where \(\dot{m}\) is the rate of change of the total mass \(m\) within the control volume \(V\), \(\rho\) is the mass density, \(\dot{m}_i\) and \(\dot{m}_e\) are the mass flows at the inlet and outlet sections of the system, respectively, \(U\) is the internal energy of the mass \(m\), and \(h_i\) and \(h_e\) are the sensible enthalpies of the inlet and the outlet flows, respectively. In (10), \(\dot{q}\) is the total heat transfer rate into the system across the boundary, and equals the difference between the heat release rate given by combustion, \(\dot{q}_c\), and the heat transfer to cylinder walls per unit time, \(\dot{q}_l\). On the other hand, \(\dot{w}\) is the work-transfer rate out of the system across the boundary and equals \(p\left(\frac{dV}{dt}\right)\) when the piston is displaced.

![Fig.4. Schematic of the engine as an open thermodynamic system.](image)

The equation of state completes the set of basic equations of the model:

\[
\rho = \frac{p}{RT} \tag{11}
\]

where \(p\) and \(T\) are the in-cylinder pressure and temperature, and \(R\) is the gas mixture constant. Substituting (11) into (9), an equation for pressure can be derived:

\[
\frac{dp}{dt} = p\left(\frac{\dot{m}_i - \dot{m}_e}{m} - \frac{1}{V} \frac{dV}{dt} + \frac{1}{T} \frac{dT}{dt}\right) \tag{12}
\]

where \(m\) is the mass within the cylinder.

Considering now the following equation, in which \(u\) is the internal energy per unit mass:

\[
\frac{dU}{dt} = \frac{d(mu)}{dt} = \frac{d(\rho Vu)}{dt} = \frac{d(\rho V)}{dt} \frac{du}{dt} = \frac{\dot{m}_i \dot{u}_i - \dot{m}_e \dot{u}_e}{m} + \rho V \frac{du}{dt} = (\dot{m}_i - \dot{m}_e)u + \rho V \frac{du}{dt} \tag{13}
\]

and combining (10) with (13), an expression for temperature can be obtained:

\[
\frac{dT}{dt} = \left[ (\dot{m}_i h_i - \dot{m}_e h_e) - (\dot{m}_i - \dot{m}_e)u + \dot{q} - p \frac{dV}{dt} \right] \left/ \left(\frac{p}{RT} V c_v\right) \right. \tag{14}
\]

where \(c_v\) is the specific heat at constant volume for the in-cylinder mass. The solution of equations (9), (12) and (14) provides the state of the open system as a function of time once the initial state of
system is specified and additional sub-models for calculation of $dV/dt$, $\dot{m}_i$, $\dot{m}_e$, $\dot{q}_c$ and $\dot{q}_i$ are provided. Here an in-cylinder initial pressure and temperature are fixed at 130,000 Pa and 323.15 K, respectively, to which corresponds an initial air mass of $m_0 = 6.208 \cdot 10^{-4}$ kg.

### 3.2.2. Geometry sub-model

The cylinder volume at any crank position $\theta$ is given by [6]:

$$V(\theta) = V_c \left[ 1 + 0.5 (r_c - 1) \left[ 1 + R_c - \cos \theta - \sqrt{R_c^2 - \sin^2 \theta} \right] \right]$$  \hspace{1cm} (15)

where $V_c$ is the combustion chamber volume at TDC, $r_c$ is the engine compression ratio and $R_c$ is the ratio of connecting rod length to crank radius. Remembering that $\theta = \omega t$, (15) can be derived to obtain the rate of change of the cylinder volume against time.

### 3.2.3. Intake and exhaust flow sub-model

Intake and exhaust valve groups are considered as time-dependent flow restrictions defined by their instantaneous geometry. The corresponding discharge coefficients were determined empirically under steady-state conditions. The gas flow rate through valves is computed using steady one-dimensional flow equations, assuming the actual flow to be quasi steady. Thus, mass flow rates $\dot{m}_i$ and $\dot{m}_e$ into and out of the system can be calculated by:

$$\dot{m}_{i/e} = \frac{C_D A_i p_s}{\sqrt{RT_S}} \left( \frac{p_T}{p_s} \right)^{\gamma y} \left[ 1 - \left( \frac{p_T}{p_s} \right)^{(y-1)/y} \right]^{\gamma (y-1)/2(y-1)}$$

when $\left( \frac{p_T}{p_s} \right) > \left( \frac{2}{\gamma + 1} \right)^{(y-1)/y}$ \hspace{1cm} (16)

$$\dot{m}_{i/e} = \frac{C_D A_i p_s}{\sqrt{RT_S}} \gamma^{\gamma/2} \left( \frac{2}{\gamma + 1} \right)^{(y+1)/2(y-1)}$$

when $\left( \frac{p_T}{p_s} \right) \leq \left( \frac{2}{\gamma + 1} \right)^{(y-1)/y}$ \hspace{1cm} (17)

Equation (17) is solved instead of (16) when flow at the valve port is choked. In the expressions above, $C_D$ is the discharge coefficient (values are provided in Appendix A), $A_i$ is the reference port area used to make dimensionless the $C_D$ values, $p_s$ and $T_s$ are the upstream stagnation pressure and temperature, $p_T$ is the downstream static pressure, $R$ is the gas constant, and $\gamma$ is the specific heat ratio. Intake pressure is estimated by the experimental in-cylinder pressure traces, whereas intake temperature was iteratively adjusted taking into account the in-cylinder temperature provided by the model for the intake stroke. Back pressure at the exhaust is assumed equal to 95,000 Pa. In (16) and (17), the flow area is given by:

$$A_v = \frac{\pi D_v^2}{4} n_v$$  \hspace{1cm} (18)

where $D_v$ is the valve inner seat diameter (equal to 23.5 mm and 18.2 mm for the intake and exhaust, respectively) and $n_v$ is the valve number (equal to 2 for both the intake and exhaust).

### 3.2.4. Combustion sub-model

Combustion heat release is calculated by using the following expression:

$$\dot{q}_c = \dot{m}_f H_u = \frac{dx_b}{dt} m_f H_u$$  \hspace{1cm} (19)

where $dx_b/dt$ is calculated by deriving (8), $m_f$ is the fuel mass per cycle (see Table 2) and $H_u$ is the fuel lower heating value (43 MJ/kg).
3.2.5. Heat transfer to cylinder walls

Heat transfer to cylinder walls in SI engine is mostly due to convection. Thus, the following expression can be used:

\[ \dot{q}_i = K \cdot A \cdot \Delta T_i \]  \hspace{1cm} (20)

where \( K \) is the total heat transfer coefficient, \( A \) is the entire area of the walls exposed to gases and \( \Delta T_i \) is the difference between the in-cylinder temperature, \( T \), and the coolant temperature \( T_c \) (assumed constant and equal to 363 K). The total heat transfer coefficient can be expressed as:

\[ K = \left( \frac{1}{\alpha_g} + \frac{s}{\lambda} + \frac{1}{\alpha_c} \right)^{-1} \]  \hspace{1cm} (21)

in which \( \alpha_g \) is the gas convective heat transfer coefficient, \( s \) is the mean thickness of cylinder walls (assumed equal to 0.07\( B \), where \( B \) is the cylinder bore), \( \lambda \) is the wall thermal conductivity (8000 W/mK) and \( \alpha_c \) is the coolant convective heat transfer coefficient (150 W/m²K). The gas convective heat transfer coefficient is calculated using Eichelberg’s correlation [26]:

\[ \alpha_g = \left(7.67 \cdot 10^{-3}\right) \cdot S_p^{0.33} \cdot (pT)^{0.5} \]  \hspace{1cm} (22)

where \( S_p \) is the mean piston speed, and \( p \) and \( T \) are the in-cylinder pressure and temperature. In (22), speed, pressure and temperature are expressed in m/s, Pa and K, respectively. The formulation proposed by Woschni [27] is not suitable for the analysis of the present engine because it results in underestimation of heat losses.

The heat exchange area in (20) is the sum of the combustion chamber wall area, \( A_{cc} \), and the cylinder liner area, \( A_l \):

\[ A = A_{cc} + A_l = SF \left( 2 \cdot \frac{\pi B^2}{4} \right) + \left[ \frac{\pi BC}{2} \left( 1 + R_c - \cos \theta - \sqrt{R_c^2 - \sin^2 \theta} \right) \right] \]  \hspace{1cm} (23)

in which \( B \) and \( C \) are the cylinder bore and stroke, respectively, and \( SF \) is a shape factor that accounts for the design of the combustion chamber. In fact, in addition to the piston crown, heat is also transferred by the cylinder head and the cylinder liner not covered by piston stroke (and thus not included in \( A_l \)). The modelled engine features a four-valve pent roof combustion chamber, but, compared to usual gasoline engines, shows a significant portion of cylinder liner always in contact with gases and a quartz-windowed piston (which limits the heat losses). Considering these three contributions, a value of 0.9 is estimated for \( SF \).

4. Validation of predicted in-cylinder pressure

Calculated in-cylinder pressure is compared to measured pressure traces in Fig. 5 and 6. Figure 5 plots pressures in the crank angle range that includes exhaust, intake and most of the compression stroke, whereas Fig. 6 focuses on ignition, combustion and most of the expansion stroke. Numerical values show a good agreement with experimental data, especially during the pressure increase phase. The close correspondence of calculated and measured traces along compression (right-hand side of Fig. 5 and left-hand side of Fig. 6) suggests that trapped mass is correctly estimated by the model and that the heat transfer rate is likely to be close to the actual value, at least up to 30 bar. The pressure trace is also well-captured during the first phase of combustion (Fig. 6). In addition, both the magnitude and the crank angle of the peak pressure match closely the experimental values (49.49 bar at 384 CAD calculated by the model, against 49.86 bar at 383 CAD measured experimentally). This means that the Wiebe function proposed in Subsection 2.3 allows a satisfactorily prediction of the heat release rate up to the maximum pressure.
Fig. 5. Experimental and calculated pressure traces during exhaust, intake and compression stroke. In the measured trace exhaust valves open at 513 CAD and close at TDC, whereas intake valves open at 717 CAD and close at 216 CAD.

Fig. 6. Experimental and calculated pressure traces during combustion and expansion stroke. Ignition occurs at 346 CAD.

A discrepancy up to 5% between numerical values and experimental data is visible along the expansion stroke. Considering that in the present engine the blow-by can achieve 12% of the trapped
mass at 2000 rpm [23], it is likely that the model overestimates the measured pressure because blow-by is neglected. An underestimation of the heat transferred to cylinder walls and a not accurate description of the heat release rate during the ending of the combustion process (see Fig. 3) may contribute to increase the mismatching between the two pressure traces as well.

To compensate for the effect of the mass overestimation on in-cylinder pressure during the exhaust phase, the valve opening was anticipated by 7 CAD with respect to the actual operating value (see Table 1) to encourage the gas outflow (also the valve closing was retarded by 7 CAD for the same reason). Nevertheless, a certain mismatch between the two profiles during the blow-down phase still remains (Fig. 5, 500< \( \theta \)<580 CAD).

Looking at the measured trace after blow-down (Fig. 5, \( \theta \)>580 CAD), it can be noted that pressure remains fairly constant before exhaust valve closing. This suggests that unsteady effects within the manifold do not significantly affect the exhaust process and, therefore, a quasi-steady model for the exhaust flow may be suitable. Instead, intake pressure fluctuation caused by the inertia of the flow cannot be captured by the present model, because it does not account for intake gas dynamic. The good agreement between measured and calculated traces shown in the right-hand side of Fig. 5 was achieved by anticipating the intake valve closing by 11 CAD (in the model the valves close at 205 CAD instead of 216 CAD). The modified valve timing allows limiting the backflow in the intake manifold during the compression stroke and, therefore, reaches the same aim of the flow inertia action.

5. Conclusions

A thermodynamic quasi-dimensional model for SI engines was developed and validated against experimental measurements of a turbocharged laboratory engine set up to perform optical investigations of the combustion process. The engine was tested at constant speed and load using gasoline and gasoline-butanol blends as fuels. Only gasoline operation is considered in this study to carry out a robust validation of the model in its basic form. The model includes semi-empirical sub-models to estimate the combustion heat release rate and the wall heat transfer, and it also allows the calculation of the in-cylinder thermodynamic state during the intake and the exhaust.

Results of the simulation suggest that the model is able to correctly estimate both the heat transfer and the mass trapped in the cylinder during the two high pressure strokes. In addition, the close correspondence between the calculated and the measured peak pressure indicates that a single-zone combustion sub-model coupled to a single Wiebe function is adequate to provide a reliable description of the combustion heat release rate, at least before expansion.

On the other hand, the less satisfactory prediction of the pressure trace during the expansion stroke is likely to be mostly due to the lack of a blow-by sub-model, although the poor representation of last MFB fraction by the single Wiebe function may play a role as well. Therefore, a blow-by sub-model and some alternatives to the single Wiebe function should be considered to improve the model of such research engine.

Finally, the quasi-steady modelling of intake and exhaust flows provides a quite good estimation of in-cylinder pressure during mass exchange. However, when a more accurate description of pressure trace is desired, manifold unsteady effects should be modelled, especially at the intake side.

Acknowledgments

The authors are grateful to Massimo Masi for his precious suggestions.

Appendix A

Intake and exhaust valve lift curves (shown in Fig. 7) allow the calculation of the valve lift, \( L \), for any crank angle \( \theta \). Valve timing is slightly different compared to that listed in Table 1, as specified in Section 4. Once \( L \) is known, the intake and exhaust discharge coefficients can be estimated by interpolating data listed in Table 3.
Fig. 7. Intake and exhaust valve lift curves.

Table 3. Intake and exhaust discharge coefficients as a function of the valve lift.

<table>
<thead>
<tr>
<th>Intake L/Dv</th>
<th>CD</th>
<th>Exhaust L/Dv</th>
<th>CD</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.03927</td>
<td>0</td>
<td>0.07850</td>
</tr>
<tr>
<td>0.03927</td>
<td>0.125</td>
<td>0.04854</td>
<td>0.181</td>
</tr>
<tr>
<td>0.07850</td>
<td>0.265</td>
<td>0.09709</td>
<td>0.363</td>
</tr>
<tr>
<td>0.11774</td>
<td>0.389</td>
<td>0.14563</td>
<td>0.514</td>
</tr>
<tr>
<td>0.15699</td>
<td>0.498</td>
<td>0.19418</td>
<td>0.630</td>
</tr>
<tr>
<td>0.19623</td>
<td>0.570</td>
<td>0.24272</td>
<td>0.699</td>
</tr>
<tr>
<td>0.23549</td>
<td>0.602</td>
<td>0.29126</td>
<td>0.755</td>
</tr>
<tr>
<td>0.27473</td>
<td>0.617</td>
<td>0.33941</td>
<td>0.775</td>
</tr>
<tr>
<td>0.31397</td>
<td>0.630</td>
<td>0.38835</td>
<td>0.792</td>
</tr>
<tr>
<td>0.32322</td>
<td>0.639</td>
<td>0.43689</td>
<td>0.812</td>
</tr>
<tr>
<td>0.34247</td>
<td>0.645</td>
<td>0.48544</td>
<td>0.834</td>
</tr>
<tr>
<td>0.43171</td>
<td>0.648</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

References