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Large Eddy Simulations and Tracer-LIF Diagnostics of wall film dynamics in an optically accessible GDI research engine

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Abstract

Large Eddy Simulations (LES) and tracer-based Laser-Induced Fluorescence (LIF) measurements were performed to study the dynamics of fuel wall-films on the piston top of an optically accessible, four-valve pent-roof GDI research engine for a total of eight operating conditions. Starting from a reference point, the systematic variations include changes in engine speed (600; 1,200 and 2,000 RPM) and load (1000 and 500 mbar intake pressure); concerning the fuel path the Start Of Injection (SOI=360°, 390° and 420° CA after gas exchange TDC) as well as the injection pressure (10, 20 and 35 MPa) were varied. For each condition, 40 experimental images were acquired phase-locked at 10° CA intervals after SOI, showing the wall-film dynamics in terms of spatial extent, thickness and temperature. The simulation framework was developed as follows: first, the spray model was calibrated using spray morphology evolution data of the same injector, characterized in a constant volume spray chamber by high-speed shadow imaging. In a second step, the wall impingement and film models were calibrated using the reference condition. With the model constants now fixed, simulations were run for the remaining operating points and results were compared with the experimental data. The simulations captured all trends observed experimentally, with good quantitative in the transient films' spatial extents and thicknesses.

Introduction

In view of the global efforts to reduce CO₂ [1] higher efficiency SI engine concepts are needed. The majority of contemporary SI engines employ Direct Injection (DI) due to thermodynamic benefits (charge cooling leading to reduced knock propensity allowing for higher compression ratios, see e.g. [2]; in addition stratified operation can be employed in different parts of the engine map, enabling de-throttling and thereby additional efficiency gains [3]. However, direct admission of fuel into the chamber brings about issues due to limited time available for mixture formation and potential interaction of the spray plumes with cylinder walls. Numerous study are reported hereto in both wall-guided and spray-guided engines using multi-stream, hollow-cone, pressure-swirl and air-assisted injector configurations, see e.g. [4, 5, 6, 7, 8, 9, 10, 11, 12, 13][14, 15, 16, 17] and references therein. Aside lubrication problems, fuel wall films with fuel-rich regions in their vicinity are considered important sources of soot

formation [18, 19, 20]. A detailed optical investigation in a wall-guided optical SI engine [21] revealed two main mechanisms responsible for soot: 1) early during combustion due to the flame passing through regions with rich mixtures, much of this soot is however oxidized later on, and, 2) pool fires in the vicinity of walls lasting until late in the cycle when oxidation deteriorates [22, 23]. Concerns with respect to the respiratory health effects of soot [24, 25] have led to enforcement of corresponding legislation in many countries worldwide. This clearly mandates further investigation into the origins of soot over the entire engine map in order to optimize operational parameters towards reduction of engine-out soot levels. In this study, a spray-guided optical SI engine is therefore examined both experimentally and numerically towards improved understanding of the formation and evolution of wall films for a systematic variation in both fuel and air path. The findings obtained and the proposed LES platform can be seen as a pre-requisite for later combustion and emissions formation investigations.

Methods

In the present section, the numerical and experimental methods are presented. In the first part, the basic conservation equations, the models and the sub-models employed to simulate the gaseous phase and the liquid spray evolution are described. We start with the gaseous phase, by presenting an overview over the Large Eddy Simulations (LES) and related sub-models, and next we move to the liquid phase, by presenting the basic models and sub-models for the Lagrangian treatment, the droplet-wall interaction and the liquid film evolution. In the second part of this section, the experimental setup and method employed to measure liquid film thickness are briefly presented. For the latter, first the LIF technique is introduced followed by an overview of the calibration strategy.

Simulation

Large-eddy simulations formulation

The STAR-CD flow field solver version 4.28 [26] is used in conjunction with the es-ICE plugin to handle mesh motion. The LES equations, employed for the simulation of the gaseous phase, are derived from the Navier-Stokes equations by applying a spatial filter

in such a way as to resolve only the large-scale turbulent motions, while the smaller scales are modelled by suitable sub-models [27]. The resulting LES equations for compressible flows can be written as

$$\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial \bar{\rho} \tilde{u}_i}{\partial x_i} = 0, \quad (1)$$

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

where Eq. 1 and Eq. 2 are the conservation of mass and momentum respectively, $\bar{\rho}$ is the mean density, $\tilde{u}$ is the Favre averaged velocity vector, $\bar{p}$ is the pressure, $\bar{\sigma}_{ij}$ is the shear stress tensor, given by

$$\sigma_{ij} = 2\mu(T)S_{ij} - \frac{2}{3}\mu(T)\delta_{ij}S_{kk}, \quad (3)$$

where $\mu(T)$ is the dynamic viscosity, T the temperature and $S_{ij} = \frac{1}{2}(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i})$ the rate of strain tensor. Finally, $\tau_{ij}^r = \widetilde{u_i u_j} - \tilde{u}_i \tilde{u}_j$ is the sub-grid stress-tensor for the Favre-averaged momentum field. In order to model sub-grid scales, in this work the sub-grid scales tensor is modelled via an eddy-viscosity type closure, which assumes a linear relationship between the sub-grid stress-tensor and the rate of strain tensor:

$$\tau_{ij}^r = -\mu_t \tilde{S}_{ij} + \frac{1}{3} \bar{\rho} \widetilde{u_k u_k} \delta_{ij}, \quad (4)$$

where μ_t is the turbulent viscosity, which depends on the chosen sub-model. In this work, the $\kappa - l$ model is employed, due to its simplicity and reduced computational cost [28] and considerations put forward in [29] assessing different LES models in the IC engines context.

The turbulent viscosity can then be written as:

$$\mu_t = C_2 \bar{\rho} \Delta \kappa_{SGS}^{0.5}, \quad (5)$$

where C_2 is a model constant set to 0.05, Δ is the cell characteristic size and κ_{SGS} is the turbulent kinetic energy, for which an equation of the form

$$\frac{\partial \bar{\rho} \kappa_{SGS}}{\partial t} + \frac{\partial \bar{\rho} \tilde{u}_j \kappa_{SGS}}{\partial x_j} = -\bar{\sigma}_{ij} \tilde{S}_{ij} - C_1 \bar{\rho} \frac{\kappa_{SGS}^{\frac{2}{3}}}{\Delta} + \frac{\partial}{\partial x_j} \left[C_2 \bar{\rho} \Delta \kappa_{SGS}^{0.5} \frac{\partial \kappa_{SGS}}{\partial x_j} \right], \quad (6)$$

is derived, and where C_1 is a second model constant set to 0.1 here. In order to simulate compressible flows, conservation of energy has to be taken into account. To do so, the chemico-thermal static enthalpy equation of the form [26]

$$\frac{\partial \rho h}{\partial t} + \frac{\partial}{\partial x_j} (\rho h u_j + F_{h,j}) = \frac{\partial p}{\partial t} + u_j \frac{\partial p}{\partial x_j} + \tau_{ij} \frac{\partial u_i}{\partial x_j} + s_h, \quad (7)$$

is solved in its Favre-averaged filtered version with the sub-models of the same type as for the momentum equation. Here, the static enthalpy is defined as $h \equiv \bar{c}_p T - c_p^0 T_0 + \sum Y_m H_m$, where $\bar{c}_p$ is the mean constant-pressure specific heat at temperature T , c_p^0 is the corresponding reference value at reference temperature T_0 , Y_m the mass fraction of specie m , and H_m the heat of formation of species m .

Discretization and solution algorithm

Spatial discretization is carried out by a finite volume formulation, while the Pressure Implicit with Splitting of Operators (PISO) algorithm for the pressure solution is employed together with a first-order implicit Euler scheme for time discretization. Mass and momentum equations are differentiated by a central-differences scheme, and the energy conservation equation by the Monotone Advection and Reconstruction Scheme (MARS). For more information about the discretization procedure the reader is referred to the Star-CD methodology [26]. A fixed time-stepping is employed, with a base of 0.1 °CA and reduced at begin and end of valve motion and spray injection until a minimum time-step of 0.025 °CA. For droplets discussed next, the solver employs a sub-stepping methodology which has been used with standard values.

Lagrangian treatment

The liquid phase dispersed in the gaseous phase, arising during liquid spray injection, is modeled here with a Lagrangian approach. As a consequence, assumptions and simplifications are needed. In particular, neither the exact nozzle geometry nor the exact flow field inside the nozzle are simulated in detail, but the droplets are simulated only from the moment they exit the nozzle. However, this approach allows to simulate quite accurately at low computational cost. In the Lagrangian approach considered here, the injected liquid is divided into elements for which mass, momentum and energy conservation equations are solved [30]. Since the number of elements may be large for a realistic gasoline direct injection spray, a statistical approach is employed here: only a limited number of computational parcels is used, each one representing a limited number of elements (droplets ensembles) having the same properties.

Basic conservation equations

Every parcel representing a group of droplets obeys the conservation of mass, momentum and energy equations, given respectively by

$$\frac{dm_d}{dt} = -A_s F_m, \quad (8)$$

$$m_d \frac{d\mathbf{u}_d}{dt} = \mathbf{F}_{dr} + \mathbf{F}_p + \mathbf{F}_{am} + \mathbf{F}_b, \quad (9)$$

and

$$m_d c_{p,d} \frac{dT_d}{dt} = -A_s \dot{q}_d'' + h_{fg} \frac{dm_d}{dt}, \quad (10)$$

where m_d is the droplet mass, A_s is the droplet surface area, F_m is the mass transfer rate per unit area, $\mathbf{u}_d$ is the droplet velocity vector, $\mathbf{F}_{dr}$ is the drag force, $\mathbf{F}_p$ is the pressure force, $\mathbf{F}_{am}$ is the “virtual mass” force, $\mathbf{F}_b$ is the general body force, $c_{p,d}$ the droplet specific heat, T_d the droplet temperature, $\dot{q}_d''$ the surface heat flux and h_{fg} the latent heat of phase change.

Explicitly defined droplet injection

In the Lagrangian framework, two main approaches to define the droplets’ characteristics at nozzle exit exist: while one tries to model the nozzle internal flow and the atomization process at the nozzle exit, the other explicitly defines the droplet size distribution at nozzle exit and computes the droplets’ speed according to the nozzle diameter.

Here we employ the second approach, as in common practice in GDI multistream atomization simulations [26]. The droplets' size distribution employed here is the Rosin-Rammler distribution:

$$Q(D) = 1 - \exp\left[-\frac{D^q}{X}\right], \quad (11)$$

where Q is the mass fraction of droplets with diameter less than D , q is the distribution parameter and X is related to the SMD of the droplets at nozzle exit through:

$$SMD = \frac{X}{\Gamma\left(1 - \frac{1}{q}\right)}, \quad (12)$$

where Γ is the Gamma function. Sensitivity to the choice of both the distribution parameter and the SMD will be discussed below.

Break-up and collision model

Break-up models are required in order to simulate droplet instability and subsequent break-up into smaller droplets. The break-up model employed here is the Kelvin-Helmholtz Rayleigh-Taylor model (KHRT) [31]. The general formulation for break-up models and thus for droplet decay into smaller droplets is of the form:

$$\frac{dD_d}{dt} = -\frac{(D_d - D_{d,stable})}{\tau_b}, \quad (13)$$

where D_d is the actual droplet diameter, $D_{d,stable}$ is an estimate of the stable droplet diameter and τ_b is the characteristic time scale of the droplet break-up. A break-up model is designed in such a way as to find $D_{d,stable}$ and τ_b . In the KHRT model, the Kelvin-Helmholtz aerodynamics instabilities and Rayleigh-Taylor instabilities arising due to droplet deceleration are taken into account and estimated. The two physical processes compete with each other, and the fastest determines the onset of break-up. In the case of the KH instability, the stable diameter and characteristic time are given by

$$D_{KH} = 2B_0\Lambda_{KH}, \quad (14)$$

$$\tau_{KH} = \frac{3.726B_1D_d}{2\Lambda_{KH}\Omega_{KH}}, \quad (15)$$

with B_0 and B_1 being model constants which depend on the operating conditions, Λ_{KH} the wavelength of the growing unstable surface wave, and Ω_{KH} the growing rate of the fastest growing wave. For the case of the RT model, the stable diameter and characteristic times are given by

$$D_{RT} = C_3\Lambda_{RT}, \quad (16)$$

$$\tau_{RT} = \frac{C_\tau}{\omega_{RT}}, \quad (17)$$

where C_3 and C_τ are model constants, Λ_{RT} is the RT fastest growing wave wavelength, and $\omega_{RT} = \omega(2\pi/\Lambda_{RT})$ is the maximum growth rate.

The collision model employed by Star-CD is the O'Rourke model [32], which distinguish mainly three types of interactions: coalescence, separation and bouncing. While the separation and bouncing mechanisms involve only a momentum transfer, coalescence needs to take into account also mass and energy transfer.

Droplet-wall interaction model

The model employed for droplet impingement is the Bai's model [33], which takes into account a number of parameters associated with the droplet and the wall, such as droplet velocity and angle with respect to the wall, droplet size and temperature-dependent thermo-physical properties, wall temperature and roughness, wetted or dry wall, liquid-film thickness, as well as near wall gas conditions. The model accounts for three wall temperature regimes, each one distinguishing between a dry or wetted wall, while droplet to wall-film transition is governed by a number of criteria discussed below.

Two characteristics, non-dimensional numbers are used to characterize the droplets; these numbers are the Weber number,

$$We = \frac{\rho|\mathbf{u} - \mathbf{u}_d|^2 D_d}{\sigma_d}, \quad (18)$$

and the Laplace number,

$$La = \frac{\rho_d \sigma_d D_d}{\mu_d^2}, \quad (19)$$

where ρ_d is the droplet density, σ_d is the surface tension of the droplet, and μ_d is the droplet viscosity.

Two characteristic temperatures are employed: the rebound temperature,

$$T_s^* = B_s T_{boiling}, \quad (20)$$

and the sliding temperature,

$$T_{LEID}^* = B_l T_{Leidenfrost}, \quad (21)$$

where B_s and B_l are multiplicative factors of the boiling and Leidenfrost temperatures respectively, here set to 1. **The multiplying factors can be adjusted in order to better match the different conditions of the wall and the material properties of the droplets.**

Three wall temperatures regimes are taken into account here, each-one distinguishing between a dry or wetted wall. In the following, the three regimes are briefly presented.

Range 1: $T_w \leq T_s^*$

In this range of temperature, a quite large number of regimes may occur. These may be: stick, in which the impinging droplet adheres to the wall in a nearly spherical form; spread: the droplet spreads forming a liquid film or merges with an already existing liquid film; rebound: the impinging droplet rebounds away from the wall after the impact; splash: the droplet breaks-up into smaller droplets, some of which are reflected from the wall; and break-up: the droplet breaks-up into smaller droplets, which are all reflected from the wall.

1. Dry wall: in case of dry wall, two regimes are taken into account: the first is the adhesion, which includes both stick and spread regimes and belongs to the range

$$We_d \leq A La^{-0.18},$$

where A is a coefficient depending on surface roughness, named Bai_1 in later on. The second is the splashing, which is set for

$$We_d > A La^{-0.18}.$$

2. Wetted wall: in case of wetted wall, three regimes are considered. The rebound

$$We_d \leq 5,$$

the spread

$$5 \leq We_d \leq A_w La^{-0.18},$$

and the splash

$$We_d > A_w La^{-0.18},$$

where A_w is an empirical coefficient, named Bai_3 later on and which is here set to $Bai_3 = 300$.

Range 2: $T_s^* < T_w \leq T_{LEID}^*$

In this range, droplet deposition cannot occur due to the presence of a vapor film. Thus, the mechanisms that may appear are distinguished as follows:

1. Rebound:

$$We_d \leq We_1^T,$$

2. Break-up and rebound:

$$We_1^T \leq We_d \leq We_2^T,$$

3. Break-up and spread:

$$We_2^T \leq We_d \leq We_a,$$

4. Splash without deposition:

$$We_d > We_a,$$

where We_1^T , We_2^T and We_a are characteristic Weber numbers.

Range 3: $T_w > T_{LEID}^*$

In this case, wall contact is also avoided due to the presence of a vapor film; the regimes that are taken into account are thus:

1. Spread:

$$We_d \leq We_1^T,$$

2. Break-up and spread:

$$We_1^T \leq We_d \leq We_a,$$

3. Splash without deposition:

$$We_d > We_a.$$

For details on the post-impingement droplet characteristics the reader is referred to Star-CD manual [26], and where an additional tuning parameter Bai_2 is related to wall friction, and which is set here to $Bai_2 = 0.005$.

Wall-film modeling

Wall-film modeling employs conservation equations governing liquid film dynamics, as well as a criterion for the transition from the droplet mode to the wall-film mode. This model was developed by Bai and Gosman in [34]. In the following, the droplet to wall-film transition criterion is presented, followed by the conservation equations governing the dynamic of the wall-film.

Transition criterion

Every droplet impacting on the wall obeys the criteria of the droplet-wall interaction model described in the previous section; if the droplet is remaining on the wall, then it is assumed to spread into a cylindrical form of diameter D_s . The droplet to wall-film transition criterion is based on the droplet surface coverage ratio γ_c , defined as

$$\gamma_c = \frac{\pi}{4A_c} \sum_i D_{s,i}^2 N_i, \quad (22)$$

where A_c is the area of the face of the cell in which the droplet resides, and N_i is the number of droplets in the i^{th} parcel. The coverage ratio is continuously updated until it reaches a threshold value that here is set to be 0.01. When the threshold value is reached, the mass of the droplets in the cell is converted into liquid film thickness and mass.

Wall-film conservation equations

The wall-film obeys conservation of mass, momentum and energy, written as

$$\frac{\partial \rho_l}{\partial t} + \frac{\partial \rho_l u_{l,i}}{\partial x_i} = \frac{\dot{m}_{imp}}{h}, \quad (23)$$

$$\frac{\partial \rho_l u_{l,i}}{\partial t} + \frac{\partial \rho_l u_{l,i} u_{l,j}}{\partial x_j} + \frac{\partial p_l}{\partial x_i} = \rho_l g_i + \frac{\partial \tau_{l,ij}}{\partial x_j} + S_{i,imp} \delta(\xi - h), \quad (24)$$

$$\frac{\partial \rho_l h_l}{\partial t} + \frac{\partial \rho_l u_{l,j} h_l}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\kappa_l \frac{\partial T_l}{\partial x_j} \right) + \frac{\dot{Q}_{imp}}{h}, \quad (25)$$

respectively. Subscript l refers to liquid film, with ρ_l denoting its corresponding density, $u_{l,i}$ is the velocity vector, $\dot{m}_{imp}$ is the mass source/sink per unit area due to droplet wall impingement, h is the film thickness, p_l is the film pressure, g_i is the gravity vector, $\tau_{l,ij}$ is the stress-tensor, $S_{i,imp}$ is the momentum source corresponding to the mass source, δ is the Dirac delta function, h_l is the enthalpy of the film, T_l is its temperature, κ_l the thermal conductivity, and $\dot{Q}_{imp}$ is the heat source corresponding to the mass and momentum source. Effects of evaporation and condensation were accounted for by activating the corresponding sub-models available in Star-CD. For further details see [35] and [36].

While no novel sub-models or extensions have been proposed here, this study aims carrying out an assessment of the existing framework by using the high fidelity measurements from an optical engine as presented in the following.

Experiment

The following section gives a brief overview of the fluorescence technique employed to image liquid-film thickness. Further details on the experimental technique and some results can be found in [37, 38, 39].

In Laser-Induced Fluorescence (LIF), laser light is absorbed by atoms or molecules, exciting them to higher electronic states. While returning to the ground state, energy can be released through radiative or non-radiative processes [40]. One possibility is the emission of fluorescence light, which is mostly red-shifted with respect to the irradiating laser light. This facilitates the separation between irradiation and measurement signal by an optical filter. LIF has been previously applied in engines, e.g. to detect liquid fuel inside the intake manifold of engines with port fuel injection [41, 42] or within the combustion chamber of engines with GDI [37, 43, 44, 45, 38]. Fuel-film thickness measurements by LIF were also done in constant volume chambers at engine-like conditions [46, 47, 48, 49]. However, quantification of the measured film thickness has proven challenging. To measure fuel-film thicknesses by LIF, it is necessary to use a mixture of a non-fluorescing surrogate fuel (here, iso-octane) and a fluorescing additive, the so-called tracer (here, 1 % toluene). To excite toluene fluorescence, short laser pulses at 266 nm were used. Since many factors influence the fluorescence signal, in particular, temperature, the tracer has to be well-characterized with respect to its photo-physical properties. For the case of toluene, details can be found in [37, 38, 39].

For weak incident light intensity (and other simplifying assumptions), the fluorescence intensity I_f can be described by equation (26), which directly relates the fluorescence intensity at each pixel with coordinates x, y to the local fuel film thickness $d(x, y)$

$$I_f(x, y) = \eta(x, y) \cdot I_0 \cdot \phi(T, \dots) \cdot (1 - \exp(-\varepsilon^* c d(x, y))). \quad (26)$$

Here, $\eta(x, y)$ is the efficiency of the detection system, $I_0(x, y)$ is the irradiating laser energy, ϕ fluorescence quantum yield, c the absorber concentration, and ε^* the extinction coefficient. Both $\eta(x, y)$ and $I_0(x, y)$ are non-uniform across the field of view (FOV). To take this into account, each LIF image is normalized by the measured fluorescence intensity distribution of uniform film (here, a piece of fluorescing glass with uniform thickness), yielding a “flat-fiel” image (FF). Fluctuations in the transverse laser profile were not taken into account, while temporal fluctuations of the total pulse energy were considered by measuring the energy of each pulse. With the spatial dependencies thus called out, equation (26) can be simplified:

$$I_f(x, y) = a(\phi(T, \dots)) \cdot (1 - \exp(-\varepsilon^* c d(x, y))). \quad (27)$$

Here, a is a global calibration factor that depends on ϕ , describing the ratio of emitted to absorbed photons. The calibration factor was determined by placing a thin-film cuvette onto the piston top in the standing engine. In this cuvette, films between 10 and 100 μm thickness were established at room temperature. By fitting the measurement data to equation (27), the required correlation between $I_f(x, y)$ and $d(x, y)$ can be determined. ϕ and thus a are a function of environmental conditions. In particular, ϕ decreases with increasing temperature [37, 38, 39]. In our experiments, we estimate this to cause a systematic error of up to 35% in magnitude, such that the measurement always tends to underestimate the real film thickness.

There are approaches for future simultaneous measurements of the film temperature [39].

In the engine experiments, at each crank angle LIF images from 40 cycles were averaged. From the film thickness distribution, the total film mass m_{film} in the FOV can be calculated:

$$m_{film} = \sum_{x=1}^o \sum_{y=1}^p P^2 \cdot m^2 \cdot d(x, y) \cdot \rho_F.$$

Here P is the magnification between image and object size, and ρ_F is the density of the fuel (iso-octane at ambient temperature: 690 kg/m³).

Engine and operating conditions

Simulations were performed in a geometry reproducing the optical engine. **A typical central mounted multi-hole-injector (Bosch HDEV 5) with six spray holes were employed.** The technical data of the engine are reported in Table 1. The engine was run motored with air as the bulk gas.

Table 1: Optical engine data.

Displacement volume	[cc]	465
Bore	[mm]	82
Stroke	[mm]	85
Connecting rod length	[mm]	146.5
Valves	[-]	4
Compression ratio	[-]	10.5:1

The domain is discretized with a total number of 2,160,000 cells at BDC, with a characteristic cell size on the order of 0.7 mm. The number of parcels injected per second is 2.5e7. Tuning of the spray for this grid resolution was carried out in a separate set-up, a constant volume vessel, where the spray was calibrated and validated over many operating conditions (not shown).

Table 2: Tuning parameters employed for the different models and sub-models of the spray and spray-wall interaction.

SMD	[μm]	55
q	[-]	7
d_{inj}	[μm]	170
θ	[°]	16
B_0	[-]	0.61
B_1	[-]	40
C_3	[-]	0.3
C_τ	[-]	1
Bai_1	[-]	2.5e-7
Bai_2	[-]	0.005
Bai_3	[-]	300

The tuning parameters that were modified from their default values for the different models for spray injection, spray break-up, droplet-wall interaction and liquid film are reported in Table 2 and are briefly

described in the following. In our Lagrangian spray model approach, the SMD and a parameter q have to be defined, together with the effective nozzle diameter d_{inj} and the cone angle of the spray θ . For the KHRT model, model constants B_0 , B_1 , C_3 and C_7 need to be specified. Together with a grid sensitivity study, tuning of these spray constants has been carried out in a separate set-up, a spray bomb, where the spray was calibrated and validated over many operating conditions (not shown). For the Bai model for droplet/wall interaction the Bai_1 , Bai_2 and Bai_3 were the modified parameters calibrated on the present engine experiment.

Eight operating conditions were studied here: starting from a reference point (#1, with $SOI=-330^\circ$ CA, $P_{inj}=20$ MPa, $P_{int} = 1$ bar, $n = 1,200$ RPM), SOI , injection pressure P_{inj} , engine speed n and intake pressure (load) P_{int} were varied. Next, the engine speed n and the load P_{int} are varied. Fuel injection temperature and inlet temperature were fixed to $T_{inj}=323$ K and $T_{amb}=293$ K respectively, while the wall temperature was fixed to $T_w=368$ K, following a sensitivity study. The wall temperature, in fact, was arbitrarily fixed by keeping into account the coolant temperature, the effect of combustion in heating the engine walls and in order to tune the liquid film evaporation in the simulations for operating condition #1. The operating points are summarized in Table 3 (the crank-angle convention in this paper assigns 0° CA to compression TDC, that is, crank angles during intake and compression are negative with descending magnitude).

Table 3: Operating conditions for each operating point. Engine speed n ; Start Of Injection SOI ; injection pressure P_{inj} ; intake pressure P_{int} ; injection temperature T_{inj} ; ambient temperature T_{amb} ; total injected mass m_{tot} ; injection duration t_{inj} . Reference operating condition #1 in bold, hatched conditions denote changes from reference.

	#1	#2	#3	#4	#5	#6	#7	#8
RPM [rpm]	1,200	1,200	1,200	1,200	1,200	600	2,000	2,000
SOI [$^\circ$ CA]	-330	-360	-300	-330	-330	-330	-330	-330
P_{inj} [MPa]	20	20	20	10	35	20	20	20
P_{int} [-]	1	1	1	1	1	1	1	0.5
m_{tot} [mg]	24	24	24	24	24	24	27.4	13.7
t_{inj} [$^\circ$ CA]	14.43	14.43	14.43	19.94	10.89	6.54	24.95	11.57

For each condition, 40 experimental images were acquired phase-locked at 10° CA intervals after SOI , providing insights into the wall-film dynamics in terms of spatial extent, thickness and temperature. In the following, these results will be employed to determine liquid film mass, area and median thickness and will be used as validation for the simulations. Cycle-to-cycle variability in LES simulations was tested separately by running three cycles for operating condition #1, and it was found that differences from cycle-to-cycle were very little. Therefore, for each operating point, only the results from the second cycle are shown.

Results

Here, the simulation results obtained for liquid film mass, area and median thickness are reported for the operating conditions discussed above, and are compared with the experimental results obtained by LIF measurements. First, a validation for the reference point (#1 in Table 3) is carried out. The influence of the fuel path is subsequently studied by a variation of the SOI and a variation of the injection pressure. Concerning the influence of the operating point of the engine on the liquid-film dynamics, results for variations of the engine speed and load are finally presented.

Reference point validation

The reference point, OP #1 is validated with respect to the measurements first by a visual comparison, and next by the evolutions of mass, area and median thickness in time. Median thickness was calculated by standard means, sampling the 2D piston surface by discrete cells and thus computing the median for liquid film thicknesses higher than $3 \mu m$. In Figure 1, the evolution of the liquid film thickness over the engine piston is shown by seven snapshots at different times in both the simulations and the experiments (left and right respectively).

Figure 1: Snapshots of the liquid film thickness (LFTH) evolution on the piston surface at -320° CA, -300° CA, -280° CA, -240° CA, -200° CA, -160° CA, and -80° CA, expressed in mm. Left column: simulations. Right column: experiments.

The qualitative evolutions of both simulation and experiment are very similar: During film formation, between -320° CA and -300° CA, the experiment shows considerably higher values. This is an artifact from the presence of the spray and residual droplets, which the LIF technique cannot distinguish from the actual film adhering to the surface. After this first phase, evaporation of the liquid film can be clearly observed, in particular in the zones where the liquid film is thin, such as in the part of the liquid film fingerprint toward the center of the piston. This process continues until all the fuel has evaporated. The

snapshots further indicate that the simulations exhibit a slightly longer evaporation time. In fact at -80° CA the major part of the liquid film observed in the experiment has disappeared, while the simulation still predicts small quantities. Apart from this overestimation of the evaporation time, the predicted liquid films show very good agreement with the experiment in terms of shape, size and approximate thickness.

In order to better quantify the liquid film formation and evaporation, the results of liquid film thickness were processed to obtain liquid film mass, area and median thickness. The evolution of these three quantities for both the simulation and the experiment is shown in Figure 2.

Figure 2: Comparison of the evolution of liquid film mass, area and median thickness in simulation and experiment for OP #1.

Shortly after the start of injection at -360° CA, the liquid film starts to form, reaching its maximum mass and area a few degrees after the end of injection. The timing of the peak in the LES simulation is in good agreement with the experiment. The difference in magnitude stems from the experimental artifacts discussed above. After reaching its maximum, the liquid film mass and area decrease as a consequence of fuel evaporation. The simulation shows a similar evaporation rate as the experiment (especially from 240° CA onwards). While the liquid film median thickness is initially very similar in simulation and experiment, evaporation for LES is completed roughly 100° CA later than in the experiment. However, this “end of evaporation” is strongly dependent on the threshold chosen for calculating the median. In all the simulations and experiments reported in this work the minimal liquid film thickness threshold was set to $3\ \mu\text{m}$ to avoid background noise present in the experiments. Below this thickness the liquid film is not taken into account in the statistics.

In general, the comparison of the liquid film computed by the simulation shows reasonably good agreement with the experiment. This allows us to use these calibrated parameter settings as a reference in order to proceed with a parametric variation in the air path and fuel injection. In the next sections, the model constants remain fixed to assess the predictive capability of the model.

Start of injection

The first parametric analysis is a variation of the SOI. Its impact on liquid film dynamics is studied comparing simulations to experiments in terms of mass, Figure 3, area, Figure 4 and median thickness, Figure 5.

Figure 3: Influence of SOI on liquid film mass evolution over time. Left: simulations. Right: experiments.

The simulations show that the highest peak in liquid film mass is reached with SOI at TDC, and that the peak mass decreases by advancing SOI, eventually reaching no liquid film formation at even later injections. Comparing the results with the experiments, the latter show a very similar trend, in particular concerning the complete evaporation of the liquid film: the later the start of injection, the lower the peak of maximum mass, and the faster the complete evaporation of liquid film. This is simply explained by the fact that a later injection means that the piston is further from the injection nozzle, and the spray takes longer to reach it. A very similar dynamic is observed for the liquid film area, with the difference that the peak is observed at slightly later stages with respect to TDC, in our case at around -330° CA. This is because the spray has an approximately conical shape, and thus the area evolution is a compromise between the projection of this conical shape on the piston head, versus the droplets’ speed and size as a function of the SOI. This trend can also be recognized in the three

experimental measurements, indicating the good capability of the simulations to capture such non-monotonic trends arising from the complex spray and droplets/wall interaction dynamics.

Figure 4: Influence of SOI on liquid film area evolution over time. Left: simulations. Right: experiments.

The liquid film thickness behavior in Figure 5, has two major features: first, earlier injection produces a slightly thicker liquid film than later injection, second, complete evaporation of the fuel is reached earlier with later injection, because less liquid film is deposited on the piston surface. The same is observed for both simulations and experiments, although in the experiments complete evaporation is reached earlier.

Figure 5: Influence of SOI on liquid film median thickness evolution over time. Left: simulations. Right: experiments.

Non-monotonicity of the median thickness can be attributed to the employment of the minimum thickness threshold discussed above, which renders it very sensitive.

While experiments are limited to imaging films on the piston top, the simulations also includes liquid films forming on the cylinder liner. In Figure 6, the liquid film mass and area evolution are presented for both the piston and the liner together. The two plots indicate that, unlike the liquid film on the piston head, delaying the injection timing increases the liquid film mass of the liner. At -360° CA no liquid film is formed on the liner, and for -330° CA only a small amount is formed, but for -300° CA, a similar amount of liquid film is formed on piston and liner. However, summing the two locations up still yields an overall smaller amount of liquid film than for earlier SOI. This indicates that in general for these injection timings early in the intake stroke a later injection time produces less liquid film.

Figure 6: Influence of SOI on liquid film distribution on piston head and liner respectively. Left: mass. Right: area.

Injection pressure

The second parameter that we want to analyze here is the influence of the injection pressure P_{inj} . In Figure 7 to Figure 9, the evolution of mass, area and median thickness are shown for P_{inj} of 35 MPa, 20 MPa and 10 MPa respectively.

In general, both simulations and experiments show that the injection pressure plays a secondary role in the liquid film dynamics, in particular with respect to the injection timings discussed in the previous section. This holds true for all quantities reported: although slight differences can be observed, the mass, the area and the median thickness follow the same trend, both with respect to the timing of the peaks of mass and area, as well as for the evaporation rate, which lead more or less to the same wall-film formation. Although with a much less pronounced variation, increasing injection pressures shows a similar trend as anticipating SOI. By decreasing the injection pressure in fact, the peak mass decreases, while the peak area increases until -330° CA and then decreases again. The lower peak mass with decreasing injection pressure is mainly due to the fact that the mass flow rate is lower for lower injection pressure, but the injection is longer. This means that more of the fuel is injected when the piston is already farther away from the injector.

As in the SOI variation, also in the pressure variation, for the area of the liquid film the conical shape of the spray plays a role, resulting in a nearly non-monotonic behavior with an increasing peak area which decreases from approximately -330° CA on. As for the SOI variation, however, the discrepancy between simulations and experiments is quite notable when looking at the moment of complete evaporation, clearly visible in the plot of the liquid film median thickness (Figure 9). The end of evaporation for the simulations is delayed by about 50-100° CA, although this criterion, as discussed above, is somewhat arbitrary due to the sensitivity to thresholds employed in the post-processing.

Figure 7: Influence of injection pressure on liquid film mass evolution over time. Left: simulations. Right: experiments.

Figure 10: Influence of engine speed on liquid film mass evolution over time. Left: simulations. Right: experiments.

Figure 8: Influence of injection pressure on liquid film area evolution over time. Left: simulations. Right: experiments.

Figure 11: Influence of engine speed on liquid film area evolution over time. Left: simulations. Right: experiments.

Figure 9: Influence of injection pressure on liquid film median thickness evolution over time. Left: simulations. Right: experiments.

Figure 12: Influence of engine speed on liquid film median thickness evolution over time. Left: simulations. Right: experiments.

Engine speed variation

The influence of the engine speed n on the liquid film evolution is evaluated in Figure 10 to Figure 12. Overall, the simulations show the expected trend: with increasing engine speed, the liquid-film peak for mass and area are lower, and later the liquid film is consumed faster. Again, this occurs because for higher speeds more of the injection takes place with the piston already at lower positions.

The same is reflected in the liquid film median thickness, which is greater for lower engine speed, and with a lower evaporation rate, while we have the opposite for higher speed, the median thickness being overall lower and the full evaporation of the fuel occurring earlier. In the case of the experiments the trend is the same only for the liquid film median thickness, while for the mass and the area we have a considerably lower peak for the case of 600 rpm with respect to the other. This may be due to the fact that the evaporation time scale is very similar in the three cases, but we are plotting the evolution in the crank angle domain. This means that for the case of 600 rpm, residual droplets that may increase the apparent liquid film have more time to evaporate (more than 3 times more than the case with 2000 rpm), thus reducing the measured liquid film down to more accurate values.

Load

The last comparison is focused on the influence of the load of the engine on the liquid film: the intake pressure is decreased from 1 bar to 0.5 bar. As a direct consequence, in order to ensure the same air/fuel ratio, the injection time is halved; this allows maintaining the mass flow rate nearly the same in the two cases, keeping the spray characteristics similar in terms of droplets velocity and size. The evolution of mass, area and median thickness is depicted in Figure 13, Figure 14 and Figure 15, respectively.

Figure 13: Influence of the load on liquid film mass evolution over time. Left: simulations. Right: experiments.

Figure 14: Influence of the load on liquid film area evolution over time. Left: simulations. Right: experiments.

Figure 15: Influence of the load on liquid film median thickness evolution over time. Left: simulations. Right: experiments.

As expected, a decrease of load, and subsequently of the injected mass, leads to a decrease in the liquid film formation, as can be clearly seen in the liquid film mass and area plots. What is less trivial, is that the peak for the case of half-load is not half of the one at full load: this may be traced back to the fact that the air density is lower in the case with lower load, and thus the droplets velocity is in general higher than the case with nominal load. A higher velocity leads, in this specific case, to a slightly more liquid film formation. Another reason may be the fact that the majority of the liquid film is formed when the spray impinges on the dry wall, while when the spray impinges on the wetted wall many of the droplets rebound away from the wall. This means that after a certain initial delay when the film is formed, an addition of droplets coming from a longer injection does not produce a proportional additional liquid film mass. This is further confirmed in the liquid film area evolution, which is nearly the same for the two loads. The trend suggested by the simulations is further confirmed by the experimental results for mass and area. For what concerns the liquid film median thickness, both simulations and experiments suggest that a lower load, and thus a lower injected mass, leads to a faster complete evaporation of the wall film fuel. However, while the simulations show a comparable liquid film thickness, the experiments show a higher liquid film thickness for the full-load case. This may be due to the higher number of residual droplets related to the spray, which may increase the apparent liquid film thickness with respect to the one that is actually appearing in the engine.

Conclusions

A joint numerical and experimental study has been carried out to explore the dynamics of formation and evolution of liquid films from spray/wall interaction in a typical gasoline engine. For the simulations, LES is used for the gaseous phase, coupled with a Lagrangian method for the liquid phase and the droplet/wall interaction model. Additionally, reduced equations are employed to simulate the dynamics of the liquid film thickness resulting from the impact of the droplets with the wall. Experimentally, an LIF technique employing toluene as fuel tracer is employed.

Variations in parameters include injection timing and pressure, engine speed and load. Calibration of the model parameters was carried out for a reference operating condition by comparing with the experimental evolutions of the liquid film fingerprint on the piston, film mass, area and median thickness. The calibrated model is subsequently applied in the remaining seven operating conditions with unchanged model constants and yields good agreement with respect to the trends observed experimentally, indicating that the dynamics of liquid film as a function of the parametric variations can be predicted fairly precisely.

A variation of SOI (all after gas-exchange TDC) leads to a progressive decrease in the peak in the liquid film mass formation, and a subsequent faster complete evaporation of the liquid fuel. In practice, this is in contrast with the desire to inject as early as possible for improved homogenization of the fuel in the combustion chamber. The maximum film area is achieved slightly later than at -360° CA, showing thus a non-monotonicity in the peak of liquid film area formation. It is also shown that a later injection moves progressively the formation of liquid film from the piston to the liner, while no influence on the total fuel evaporation time is present; however, evaporation is in general faster for later injections. Changes in injection pressure show only a minor influence in particular for the peak mass formation. A lower injection pressure reduces the time

needed for complete evaporation since the time required to inject the same amount of fuel is longer and in this time the piston moves further from the injector. Higher engine speed produces less fuel film and thus leads to a faster complete evaporation of the fuel. Finally, the influence of the engine load was explored, by reducing the load by half. This leads to a slight decrease in the initial film mass and thus to a faster evaporation of the liquid-film. Like injection pressure, however, this plays a secondary role with respect to injection timing and engine speed.

Overall, the LES methodology developed and validated using the LIF experimental data in this study proved to be effective in reproducing all the experimentally observed trends and can thus be employed for engine optimization or to explore further the dynamic of liquid film under different relevant conditions. Future investigations will include in particular near-wall fuel stratification and its impact on combustion and soot formation.

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Acronyms

CA	Crank Angle
CD	Calibration Device
FOV	Field Of View
FF	Flat-Field
KHRT	Kelvin-Helmholtz/Rayleigh-Taylor
LES	Large Eddy Simulation
GDI	Gasoline Direct Injection
LIF	Laser Induced Fluorescence
RPM	Revolutions Per Minute
SMD	Sauter Mean Diameter
SOI	Start Of Injection
TDC	Top Dead Centre
WOT	Wide Open Throttle

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Appendix

Figure 16: Snapshots of the liquid film thickness evolution on the piston surface for run #1 at -320° CA, -300° CA, -280° CA, -240° CA, -200° CA, -160° CA, and -80° CA, expressed in mm. Left column: simulations. Right column: experiments.

Figure 17: Snapshots of the liquid film thickness evolution on the piston surface for run #2 at -350° CA, -330° CA, -310° CA, -270° CA, -230° CA, -190° CA, and -110° CA, expressed in mm. Left column: simulations. Right column: experiments.

-230° CA, -190° CA, and -80° CA, expressed in mm. Left column: simulations. Right column: experiments.

Figure 18: Snapshots of the liquid film thickness evolution on the piston surface for run #3 at -290° CA, -280° CA, -260° CA, -220° CA, -200° CA, -170° CA, and -140° CA, expressed in mm. Left column: simulations. Right column: experiments.

Figure 19: Snapshots of the liquid film thickness evolution on the piston surface for run #4 at -320° CA, -300° CA, -280° CA, -240° CA, -200° CA, -160° CA, and -80° CA, expressed in mm. Left column: simulations. Right column: experiments.

Figure 20: Snapshots of the liquid film thickness evolution on the piston surface for run #5 at -320° CA, -300° CA, -280° CA, -240° CA, -200° CA, -160° CA, and -80° CA, expressed in mm. Left column: simulations. Right column: experiments.

Figure 22: Snapshots of the liquid film thickness evolution on the piston surface for run #7 at -320° CA, -280° CA, -260° CA, -220° CA, -200° CA, -170° CA, and -140° CA, expressed in mm. Left column: simulations. Right column: experiments.

Figure 21: Snapshots of the liquid film thickness evolution on the piston surface for run #6 at -320° CA, -280° CA, -260° CA, -220° CA, -200° CA, -170° CA, and -140° CA, expressed in mm. Left column: simulations. Right column: experiments.

Figure 23: Snapshots of the liquid film thickness evolution on the piston surface for run #8 at -320° CA, -300° CA, -280° CA, -260° CA, -240° CA, -220° CA, and -210° CA, expressed in mm. Left column: simulations. Right column: experiments.