

Highlights

Experimental and LES investigation of ignition of a spinning combustion technology combustor under relevant operating conditions

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- First presentation of the Radius chamber ignition test rig and measurements.
- Large-Eddy Simulations with semi-detailed chemistry describing the first instants of ignition and flame propagation; validation by comparison with experiments.
- Impact of two formulations (static and dynamic) for the subgrid closure of chemistry-turbulence interaction on the flame-propagation.

Experimental and LES investigation of ignition of a spinning combustion technology combustor under relevant operating conditions

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Abstract

SAFRAN Helicopter Engines has developed the spinning combustion technology in which the burnt gases from one injector travel tangentially along the combustor annulus towards the neighboring injectors. Compared to a conventional design, this arrangement modifies the ignition process, which is a critical phase for aeroengines. In order to understand the ignition process in this technology, experiments and Large-Eddy Simulation (LES) have been performed in a cylindrical combustion chamber where the flow is injected tangentially (named Radius chamber). Three cases are considered with different strain and turbulence levels representative of real combustor flows. Micro calorimetry and the Background-Oriented Schlieren technique allows for detailed temporal measurements of energy deposited in the flame kernel. Pressure measurement and Schlieren imaging are used to study the flame propagation. LES are performed with a 19-species and 184-reactions analytically-reduced chemistry together with the thickened flame approach allowing the description of the first instants of ignition in a quasi-DNS mode and ensuing flame propagation. Both a static and dynamic formulations of the wrinkling factor to describe sub-grid scale chemistry-turbulence interaction are used. Results show that LES is able to capture the flame kernel formation and trajectory as well as the time to reach maximum pressure within an error of 10% when using a dynamic formulation. On the other hand, the static formulation of the wrinkling factor predicts the time for maximum pressure within a maximum error of 20%.

Keywords: Ignition, Analytically Reduced Chemistry, LES, Spinning combustion technology

1. Introduction

A great challenge in aeronautical combustors consists in ensuring ignition at all flight conditions, in particular at high altitude where low temperature and pressure make ignition more difficult. Safran HE has recently developed the spinning combustion technology (SCT) depicted in Fig. 1. It consists in injecting the airflow and the fuel tangentially into the combustor. In this way, the hot gases produced by the flame attached to one injector are directed towards the next injector along the combustor annulus, eventually creating a unique flame ring which expands across the full combustor.

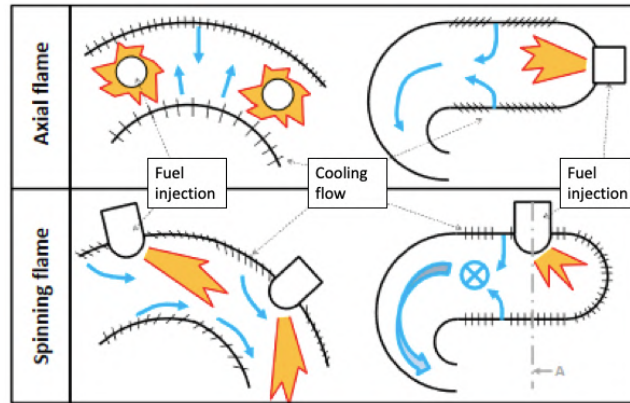


Figure 1: Spinning combustion technology (SCT) adapted from [1]

This technology has several advantages, among which it offers better igniting capability, which can be beneficial for future hybrid-electric systems, greater temperature homogeneity at the combustor exit, and reduced mass and cost. Due to the recent introduction of the Spinning Combustion Technology, according to the authors, there is not a substantial amount of literature directly related to it. SCT resembles some established combustion configurations such as MILD combustion [2, 3] by the fact that combustion products from one injector have some degree of mixing with the fresh gases from the other injectors. However, both the degree of mixing and the fuels

used are different, so definite conclusions cannot be obtained from these past experiences.

On the other hand, despite recent insights gained into ignition in conventional aeroengine designs [4, 5, 6], specific work is required for SCT, which introduces new mechanisms. Recent research [7, 8] indicate that the ignition kernel evolution is independent of the first instants just after sparking, thus reducing the need for a detailed description until times after spark $O(10\mu s)$. After the kernel is formed, ignition success depends on the chemical heat release rate overcoming the rate of heat diffusion. Only large enough kernels can sustain these conditions. The success in this phase is usually studied by the concept of minimum ignition energy (MIE) which is the value of energy that allows for 50% of probability of successful ignition [9].

The MIE depends on several factors [5]:

- Igniter parameters: The electrode geometry and the temporal power supply distribution control the spatial distribution and intensity of the deposited energy.
- Gas-phase parameters: The mean and fluctuating flows have a direct impact through mixing while local conditions may favor or disadvantage chemical activity. This effect is illustrated in Fig. 2
- Liquid-fuel parameters: Volatility and spray characteristics such as droplet density and size distribution are of primary importance for the vapor distribution.

The stochastic character of ignition mainly originates from turbulence inducing mixture fraction fluctuations [10], and velocity fluctuations at the spark location [11]. Recent research [12] contains a data-driven study performed over 450 realizations of relight in a combustor and two different fuels to shed light on the probabilistic behavior of ignition. This study was able to categorize diverse modes of failure which are different depending on the fuel type.

On the other hand, the effect of turbulence on the early flame structure depends on the relative size of the eddies to the kernel [13, 14], which may be evaluated with the ratios u'/S_L and L/d_k where u' is the velocity fluctuation, S_L the laminar flame speed, L a turbulent scale and d_k the kernel size. If the kernel is small compared to the turbulent scales, it is only convected by eddies. When the kernel grows, vortices start to interact with the flame [15].

Fast chemistry will allow the kernel to resist and be only wrinkled by the eddies. Conversely, if chemistry is too slow, the kernel may locally quench and break into smaller parts. These effects are summarized in Fig. 3.

Measurements of quenching distance (d_q), defined as the minimum radius for a flame kernel to lead to successful ignition, are available for different pressures, turbulence intensities, turbulence scales and equivalence ratios. The following dependencies were found for gaseous mixtures at several turbulence levels [16] with λ , ρ , c_p being respectively the heat conductivity, density and specific heat at constant pressure:

$$d_q = \frac{\frac{10\lambda}{c_p\rho}}{S_L - 0.16u'} \quad \text{for } u' \leq 2S_L$$

$$d_q = \frac{\frac{10\lambda}{c_p\rho}}{S_L - 0.63u'} \quad \text{for } u' > 2S_L$$
(1)

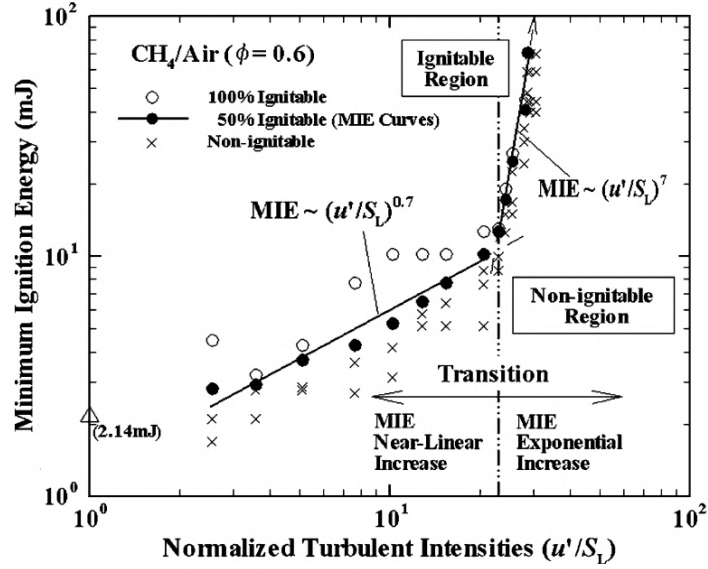


Figure 2: MIE of a uniform methane-air mixture in isotropic turbulence for different turbulent levels showing a change of behavior at $\frac{u'}{S_L} \approx 20$ [17]

The presence of the liquid phase further increases the MIE with respect to gaseous fuels due to the necessity to evaporate droplets [18, 19, 20, 16, 21].

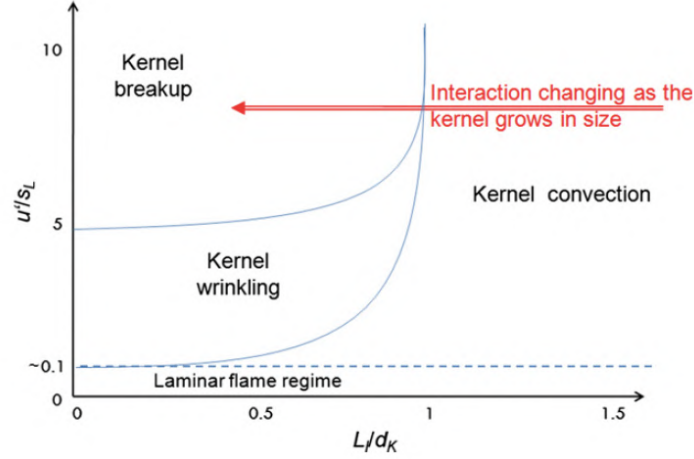


Figure 3: Turbulence regime map [13]

After the kernel has grown to the size of the integral length scale, its evolution will depend on the conditions encountered during the flame propagation [22, 23]. Homogeneous conditions lead to the flame evolving as a premixed or stratified flame. On the other hand, large mixture fraction fluctuations may lead to local quenching and edge flame occurrence [4]. Some authors use the Karlovitz number to determine success of this stage [24] despite not being the only decisive parameter [4]. Flame stretch is another important parameter. Failure in this phase is related to the second mode or “long” failure mode discussed in [4].

If successful, flame propagation results in the flame stabilizing on an injector. Experiments [25, 26, 27] show that the flame must be trapped by the recirculation zone, have enough time to grow, and also be able to ignite the region near the anchoring point.

The final phase consists in the flame propagation between injectors until all injectors hold stable flames. The ratio of inter-injector space to swirler diameter affects the speed and the mode of flame propagation. In this phase, the azimuthal velocity and the combustor liner cooling flows are expected to play an important role in the SCT.

The present study intends to give greater insight into the effect of turbulence and stretch levels on ignition in SCT under realistic engine conditions and strong swirling flow. For that purpose, an experimental configuration

consisting of closed chamber with a single injector in which ignition occurs in a cross-flow has been developed and is described in section 2 together with available measurements. In section 3, the numerical modelling is described focusing on the modelling of chemistry kinetics and turbulent combustion. Finally, results from a static and a dynamic formulation of the turbulent combustion model are presented and discussed against experimental data. While the present study does not include an analysis of the influence of combustion products on flames located downstream, an effect which will be present in SCT in the latest instants of the light-around process, this study provides insight into the first instants of the light around in which a flame kernel is formed near the igniter and grows in a swirling flow.

2. The Radius test rig and available measurement

The experiment was carried out at PPRIME Institute in a constant volume vessel composed of a cylindrical chamber and equipped with a dynamic pressure gauge and optical access allowing pressure evolution measurements and high frequency visualization. Initially, the mixture of methane and air is stored at 6 bars in a tank, then it is tangentially injected through a gas injector into the initially empty chamber generating a swirling motion of the flow. The overall equivalence ratio is set to 0.7. The chamber is empty and it is filled by the mixture up to a pressure of 0.5 or 1 bar by varying injection duration. Ignition of the mixture is ensured by a high energy ignition system used by Safran HE. The time delay between the end of injection and the ignition time is varied to obtain different strain rate and turbulence intensity levels. Fig. 4 shows a sketch of the experimental setup.

To characterize and to control as much as possible the boundary conditions of the ignition process, the temporal evolution of the electrical power delivered to the igniter electrodes is measured as well as the transmission efficiency in transforming the electrical energy into thermal energy injected into the kernel. In addition, temporal evolution of pressure during the combustion process is recorded with a piezo-resistive gauge (Kistler 4005BA with an uncertainty of $\pm 0.2\%$), as well as Schlieren visualizations of all the reactive processes using a classical bench coupling mercury vapour lamp and a fast camera Photron SAZ at 48 KHz.

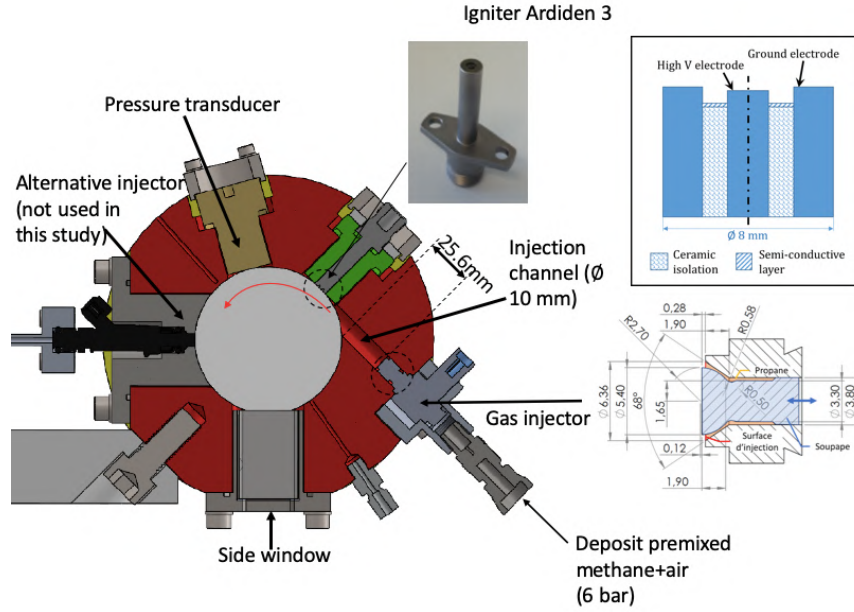


Figure 4: Sectional view of the chamber showing injection from the channel on the right, igniter at the top of the cylinder and in anticlockwise order: pressure sensor, liquid injector (not used in this study), window.

2.1. Electrical Characterization of the Ardiden igniter

Usually, a coaxial cable connects directly the igniter to the electric controller, which prevents the measurement of current and voltage characteristics. Therefore, an adaptive part was placed between the igniter and the coaxial cable to measure current and voltage temporal evolutions: the current probe (Pearson probe model 101) is placed around a copper rod, isolated by a POM-C case. This rod connects the high voltage electrode to its coaxial cable. A ground cable connects the ground electrode to the coaxial cable corresponding part. The voltage signal applied to the Ardiden 3 igniter is measured with a Tektronix P6015A probe.

During electrical discharges, voltage and current variations occur in the order of tens of ns (in particular, hot plasma discharge breakdown voltage drops in tens of nanoseconds), so wideband electrical probes are required. The Tektronix P6015A is chosen as high voltage probe, with a 75 MHz bandwidth and measuring voltage peaks up to 40 kV. Concerning the current, the Pearson current monitor model 101 is used as high currents are expected with

such igniter system, and it measures up to 50 kA with a 4 MHz bandwidth. Uncertainty for voltage and current are estimated to 3% and 1% respectively.



Figure 5: Adaptation to measure temporal current and voltage evolution

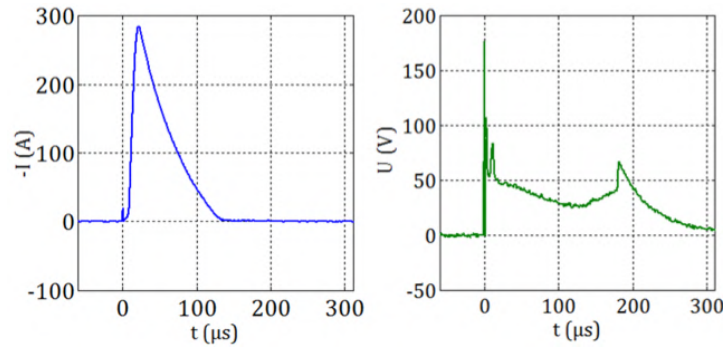


Figure 6: Typical electrical signals: Current (inversed, left) and Voltage (right).

Finally, the temporal evolution of current and voltage (cf. a typical signal in Fig. 6) allows to determine the electrical power and the total electrical energy (see Fig. 7) delivered to the electrodes of the Ardiden 3 igniter. For this igniter, the overall electrical energy is around 625 mJ per discharge, with a maximum power peak of 13.5 kW.

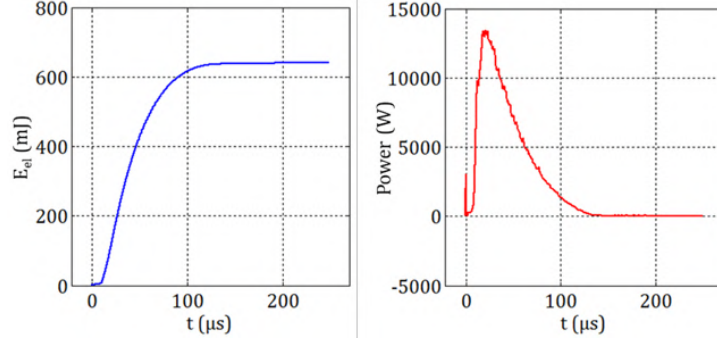


Figure 7: Typical electrical energy E_{el} (left) and electrical power (right).

2.2. Thermal Energy Released by the Ardiden 3 Igniter

The electrical energy determined by the current-voltage time evolution does not correspond to the thermal energy deposited into the fluid, which is at the origin of the mixture ignition. For this objective, a calorimetry methodology is followed to characterize the energy release to the fluid, and thus determining the transmission efficiency of the ignition process.

The chosen calorimetry methodology consists in measuring the pressure increase induced by an electrical discharge in a closed vessel filled by inert gas at a given pressure. Considering the small volume of hot kernel compared to the chamber size, a single zone model can be used. As the pressure increase is small, a constant gas density is assumed, as well as an ideal gas behaviour. The thermal energy remaining into the gas after the discharge (E_{th}) is obtained from the pressure increase in the given chamber volume as:

$$E_{th} = \int_V \rho C_V \Delta T dV = \frac{1}{\gamma - 1} \Delta P V \quad (2)$$

where ΔT and ΔP are the temperature and pressure increase in the test volume. Heating capacities C_V , heat capacity ratio γ and the chamber volume V are assumed to be constant.

The closed volume is a cylindrical chamber. It is made of POM-C, to reduce heat losses through the walls, and there are 25 mm diameter silica windows with anti-reflection coated surfaces (W2-PW1-2506M-UV-1064-0) at the cylinder bases. This chamber ensemble is shown in Fig. 8. The resulting volume of the chamber is $V = 19.6 \pm 0.2$ ml and it was chosen in order to facilitate the pressure rise measurement. The volume was verified

by filling it with water with a volumetric syringe. Because the transient increase pressure produced by an electrical discharge of few hundred mJ is of a few mbar in such a volume, a piezoresistive differential pressure transducer Meggitt 8510B-1 (0-1 psi range) is used. It is connected to a Meggitt DC Amplifier model 136, which also adapts and filters the pressure signal, with an integrated 10 kHz Butterworth physical filter (cf. red curve in Fig. 10). Signals are recorded by a LeCroy waverunner 104Xi high speed sampling oscilloscope. The transducer is located at least 1 cm from the electrodes tips and is protected by a porous metallic disc (around 50% porosity) to reduce the direct impact from the shockwave at breakdown. In this chamber, the maximum uncertainty is estimated at 6% for a discharge at $E_{th} = 100$ mJ. A pressurized system (a parallelepipedal counter pressure chamber) is used for experiments involving different gas pressures. It consists of an outer metallic chamber which contains the calorimetry chamber test (see Fig. 8). The counter pressure chamber is made of stainless steel, measuring $120 \times 76 \times 70$ mm³, with a 0.64 dm³ inner volume. It serves as a reference pressure for the piezo-resistive differential pressure transducer. Such configuration allows keeping a good accuracy on the pressure measurement in the calorimetry chamber test. A gas inlet at the base, closed by a valve, allows filling and emptying the volume with gas at different pressures. Two opposite sides of the chamber hold 125x75 mm² windows for visualization diagnostics.

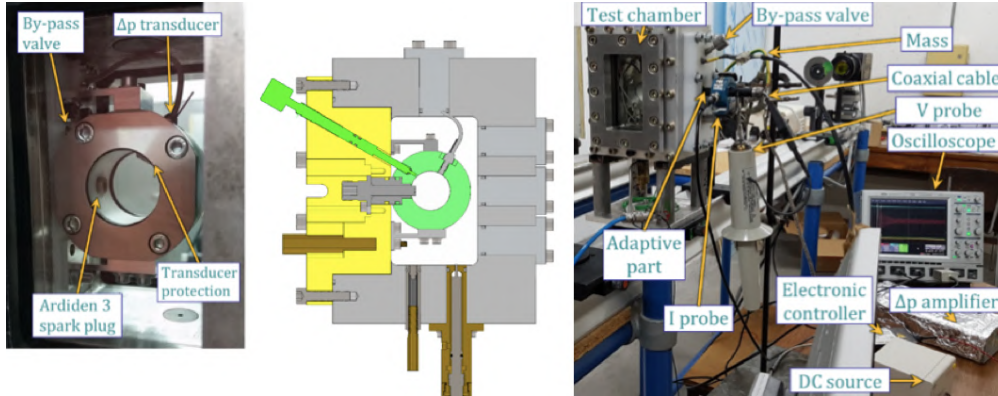


Figure 8: Calorimetry setup: detail of the 20 ml cylinder (left) and sketch and overview of the full calorimetry setup with the counter pressure chamber (center and right): 20 ml inner volume chamber (light green), by-pass valve (green), pressure transducer and spark plug (grey) and POM-C wall (yellow).

In order to remove the resonance frequency of the chamber, the recorded pressure signal is processed by suppressing frequencies over 4 kHz in the Fourier domain before performing the inverse transform (blue curve in Fig. 9). The thermal energy released to the gas is then obtained from the filtered unsteady pressure evolution a few milliseconds after the discharge, averaged over 1 ms (green curve in Fig. 9). A set of 45 tests, performed at ambient pressure in air, indicates that the electrical to thermal efficiency of this igniter is on average 13.7%, with an average electrical energy used of 625 mJ (see Fig. 10) with a total time of energy deposition of 150 μ s. The corresponding standard deviations are 49 mJ, 7 mJ and 0.4% for the electrical energy, thermal energy and efficiency respectively. This scattering results from the stochastic nature of the ignition process. Among the different causes, one may evoke microscopic differences in the surface of the electrodes between consecutive discharges and the scattering in the formation process of the plasma channel. Tests performed at different pressures have shown a very weak influence of gas pressure on the global features of the discharges of this igniter. The efficiency ranges from 12 to 14% when increasing gas pressure from 0.25 to 2 bar (in particular: 12.5% for 0.5 bar and 13.7% for 1 bar). This electrical to thermal efficiency is smaller than the case of other electrodes (e.g. pin) as mentioned in [28].

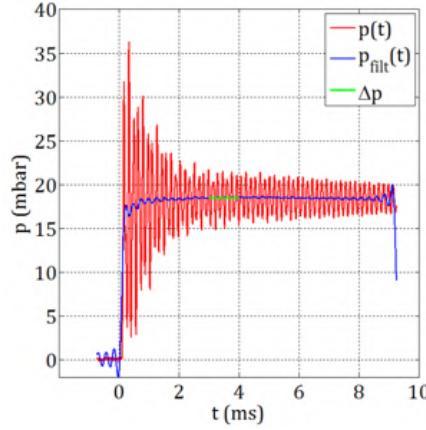


Figure 9: Typical temporal relative pressure evolution measured and filtered in the micro-calometric chamber.

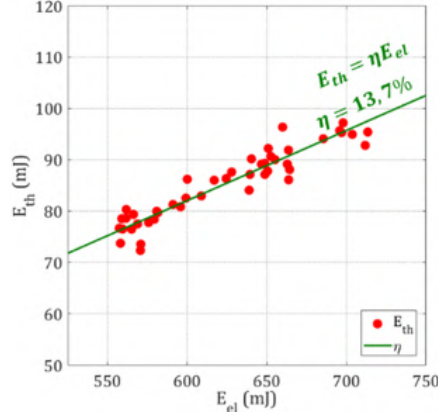


Figure 10: Electrical (E_{el} , abscissa) to thermal (E_{th} , ordinates) energy correlation for the Ardiden 3 igniter at ambient pressure

2.3. Ignition Kernel Spatial Characteristics

To determine the kernel size evolution and its projection from the igniter, high magnification Speckle Background Oriented Schlieren (SBOS) visualization developed in a previous study [29] has been used. Such technique highlights the density gradient, which is related to the refraction index gradient through the Gladstone-Dale relation.

The light source is a diode-pumped continuous solid-state laser (MxL-F, $\lambda = 532$ nm, 3 W). The emission power is stabilized for steady conditions at 1%. The beam is expanded to a 45 mm diameter parallel beam by a collimator impacting a 1 mm wide ground glass that produces the speckle by scattering effects. In order to record sufficiently sharp images of the kernel and of the generated shock wave, the exposure time is reduced down to 1 μ s using a Princeton CCD intensified camera (PiMax 1k GenII RB-SG). A 532 nm interferometric filter is used to reduce the light emission from the hot kernel. The deviations of light rays due to refractive index variations induce displacements of the speckle patterns that are determined by PIV-like multipass cross correlation processing [29]. Time evolution of the kernel is then reconstructed by test repetition, shifting the delay between the spark trigger and the image timing. The resulting displacement fields are reported in Fig. 11 for ambient pressure showing the expansion of the kernel with time as well as the shock wave propagation.

The kernel penetrations, defined as the distance between igniter surface

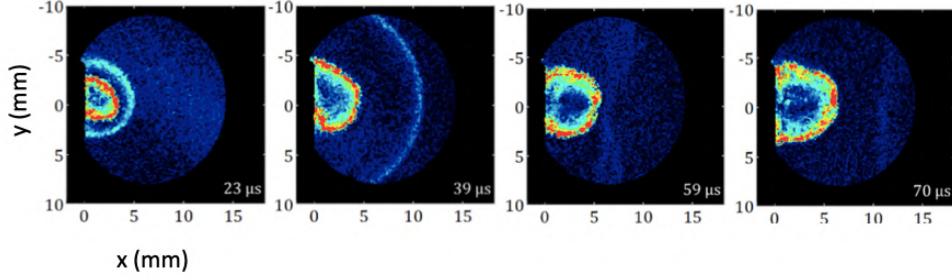


Figure 11: Ardiden 3 igniter discharge kernel evolution for $P=1$ bar and $T=293$ K. Visualisation for 23, 39, 59 and 70 μs after start of the spark. Positions are reported in mm from the igniter surface, along the symmetry axis.

and tip of the kernel and the kernel volume obtained assuming symmetry around the igniter axis, are determined from these visualizations for two pressure values, as reported in Table 1 at a given delay after the spark.

Table 1: Geometrical characteristics of the hot kernel obtained from Schlieren images				
Pressure (bar)	Kernel thermal E. (mJ)	Penetration (mm)	Kernel Vol. (mm^3)	Delay (μs)
0.5	72	6.2	≈ 330	57
1	85	5.3	≈ 280	48

3. Numerical modelling

The three-dimensional compressible LES code AVBP from CERFACS (www.cerfacs.fr/avbp7x) has been used. It solves the LES filtered compressible Navier-Stokes equations (mass, momentum, energy and $N_s - 1$ species) which read:

$$\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial(\bar{\rho} \tilde{u}_i)}{\partial x_i} = 0 \quad (3)$$

$$\frac{\partial \bar{\rho} \tilde{u}_i}{\partial t} + \frac{\partial(\bar{\rho} \tilde{u}_i \tilde{u}_j)}{\partial x_j} = - \frac{\partial(\bar{P} \delta_{ij} - \bar{\tau}_{ij} - \bar{\tau}_{ij}^t)}{\partial x_j} \quad (4)$$

$$\frac{\partial \bar{\rho} \tilde{E}}{\partial t} + \frac{\partial(\bar{\rho} \tilde{E} \tilde{u}_j)}{\partial x_j} = - \frac{\partial[\bar{u}_i(\bar{P} \delta_{ij} - \bar{\tau}_{ij}) + \bar{q}_j + \bar{q}_j^t]}{\partial x_j} + \frac{\Xi_\Delta}{F} \bar{\omega}_T + \bar{Q} \quad (5)$$

$$\frac{\partial \bar{\rho} \tilde{Y}_k}{\partial t} + \frac{\partial (\bar{\rho} \tilde{Y}_k \tilde{u}_j)}{\partial x_j} = - \frac{\partial [\overline{J_{j,k}} + \overline{J_{j,k}^t}]}{\partial x_j} + \frac{\Xi_\Delta}{F} \bar{\omega}_k \quad (6)$$

where the overline indicates the filtered elements and tilde the Favre-filtered terms. $\bar{\omega}_k$ stands for the chemical source terms, while ω_T accounts for the energy change due to chemical reactions, \dot{Q} indicates energy sources such as the one provided by the ignition system. Ξ_Δ and F are respectively the wrinkling factor (efficiency) and thickening of the thickened flame model, which will be detailed later. The perfect gas law is used.

The filtered viscous flux, heat flux and species diffusion terms are modelled as:

$$\bar{\tau}_{ij} \approx \bar{\mu} \left(\frac{\partial \tilde{u}_j}{\partial x_i} + \frac{\partial \tilde{u}_i}{\partial x_j} - \frac{2}{3} \delta_{ij} \frac{\partial \tilde{u}_k}{\partial x_k} \right) \quad (7)$$

$$\bar{q}_i \approx \bar{\lambda} \left(\frac{\partial \bar{T}}{\partial x_i} + \sum_{k=1}^N \overline{J_{i,k} h_{s,k}} \right) \quad (8)$$

$$\overline{J_{i,k}} \approx -\bar{\rho} (\bar{D}_k \Xi_\Delta F \frac{W_k}{W} \frac{\partial \tilde{X}_k}{\partial x_i} - \tilde{Y}_k \tilde{V}_i^c) \quad (9)$$

where $h_{s,k}$ is the sensible enthalpy for species "k", D_k is the mass diffusivity of species "k", W and W_k the molecular weight of the mixture and species "k" respectively, and v^c stands for the correction velocity used to ensure mass conservation, which follows the Hirschfelder formulation [30]. For the closure of the species and energy equation, a constant Schmidt and Prandtl numbers are considered equal to 0.6. More details about the modelling of each term can be found in [31]. The respective subgrid components are denoted by the "t" superscript in Eqs. 4 to 6. For the momentum equation, the SIGMA turbulent subgrid model [32] is used. The interaction of the flame with turbulence is modelled using the thickened flame model. In this study, the flame sensor recently proposed in [33] is used which automatically identifies the heat release zones and applies thickening according to a single user-specified number of cells within the flame front (in this study 7 cells). This sensor has the advantage of being independent of flow conditions (equivalence ratio, pressure, etc.) and to adapt automatically to the mesh resolution.

Since the flame grows in the whole domain and interacts with the chamber walls, there is the need for a high-fidelity description of the turbulence-chemistry interaction in the whole domain. Modelling approaches such as Detached-Eddy Simulation would require additional modelling assumptions,

in particular when the flame approaches the walls, so a LES approach was selected. In order to describe the non-resolved flame wrinkling due to subgrid-scale turbulence, the thickened flame model [34] uses the wrinkling factor Ξ_Δ , also known as the efficiency function. In this work, both the static and the dynamic formulations of the efficiency developed in [35] are tested and compared.

The wrinkling factor is defined as:

$$\Xi_\Delta = \frac{S_{T\Delta}}{S_l^0} = \frac{A_{sgs}}{\Delta^2} \quad (10)$$

where Δ is the length scale associated to the LES filtering, $S_{T\Delta}$ is the turbulent flame speed at the subgrid scale level, S_l^0 is the laminar unstretched flame speed and A_{sgs} the subgrid scale flame surface area. Both static and dynamic formulations assume equilibrium between the turbulent motion and the flame wrinkling. This assumption is valid as long as the time scales associated to subgrid-scale flame dynamics are much smaller than the other time scales of the flow field [35].

The static formulation postulates the following expression for efficiency:

$$\Xi_\Delta = \left(1 + \min \left[\left(\frac{\Delta}{\delta_l^0} - 1 \right), \Gamma_\Delta \left(\frac{\Delta}{\delta_l^0}, \frac{u'_\Delta}{S_l^0}, Re_\Delta \right) \frac{u'_\Delta}{S_l^0} \right] \right)^\beta \quad (11)$$

where Re_Δ and u'_Δ are the Reynolds number at subgrid-scale and the corresponding velocity fluctuation, δ_l^0 is the laminar unstretched flame thickness and Γ_Δ is a function that accounts for the straining effects of the vortices smaller than Δ on the flame, obtained in [35] from DNS calculations. In this study, following the formulation in [34], the velocity fluctuation is calculated from the rotational part of the resolved velocity field as indicated in Eq. 12 where c_2 is a model constant, obtained from isotropic homogeneous turbulence test cases, and equal to 2.

$$u'_\Delta = c_2 \Delta^3 |\nabla \times (\nabla^2(\tilde{u}))| \quad (12)$$

In the static formulation, a constant value of $\beta = 0.5$ is used as suggested in [35]. The dynamic formulation introduces a non-constant β coefficient, which is dynamically computed [36, 37]. The dynamic computation is performed with a double filtering operation, equating the flame surfaces calculated at a filtered and test-filtered scales and employing a ‘‘Germano-like’’ equation shown in Eq. 13.

$$\left\langle \overbrace{\Xi_{\Delta} \frac{W_{\Delta}(\hat{q})}{\Delta}} \right\rangle = \left\langle \Xi_{\gamma\Delta} \frac{W_{\gamma\Delta}(\hat{q})}{\gamma\Delta} \right\rangle \quad (13)$$

where γ is the ratio of the combination of test and LES filter scale to the LES filter scale. The overbrace indicates filter at $\gamma\Delta$ length, while the top hat denotes Favre-filtering at scale $\gamma\Delta$. Lastly, the right and left brackets denote spatial averaging. W_{Δ} is the resolved reaction rate, expressed as function of a temperature based progress variable $q = (T - T_u)/(T_b - T_u)$ where T_u and T_b are the fresh and burnt gas temperature. In this study, γ is set to 1.5. Previous works [38, 36] suggest that this parameter does not have a strong impact on the simulations when $1.5 < \gamma < 7$ in a similar configuration. Applying Eq. 10 to express the wrinkling factors gives an equation for β . In this work, the local formulation is used, in which the previously mentioned equality is enforced over a control volume denoted by $\langle \cdot \rangle$. Further details of how the β parameter is calculated can be found in [37].

The second-order explicit Lax-Wendroff scheme was used with a time step constrained by both the CFL condition and chemical activity and was of order $O(10ns)$.

3.1. Initial and Boundary conditions

For the laminar cases, all surfaces were treated as non-adiabatic walls, using heat transfer resistances obtained with the electric analogy and equal to $1.43e-2 \text{ m}^2K/W$ for the quartz boundaries (the cylinder faces) and $2.18e-2 \text{ m}^2K/W$ for the steel boundaries (rest of boundaries) and assuming the surrounding air at ambient conditions. The laminar cases were initialized with a quiescent premixed mixture of methane and air at the corresponding conditions.

For the turbulent cases, in the experiments, the filling of the chamber starting from vacuum to 1 bar lasts 135 ms (67.5 ms for 0.5 bar). This procedure was reproduced numerically. Due to the pressure ratio between the tank (6 bar) and the chamber ($0 < p < 1$ bar), the injector nozzle is choked throughout the filling process. Following the measured linear increase in pressure with time during filling, a constant uniform mass flow rate boundary at a constant temperature of 387 K was imposed at the inlet. This approach does not take into account the true shape of the inlet velocity profile, but the lack of measurements at the inlet did not allow to do better. The solid walls are treated with wall-functions. Both pressure and density at the end

of injection were verified to be within 1% of the experiment, which gives confidence to the numerical setup during the filling phase. Once the desired pressure is reached, injection stops and the inlet boundary is modelled as a wall.

3.2. Chemical scheme

The analytically reduced chemistry for methane/air employed is described in [39] and consists of 19 species and 184 reactions (Lu19). Several past studies [5, 15] have shown that detailed chemistry description is needed for ignition studies and that two-step chemical schemes are not generally able to recover the right response to strain and auto-ignition times. The chosen chemistry scheme has been thoroughly validated for auto-ignition (for $1000 < T < 8000$ K), 1D flames, perfectly stirred reactors for $1 < p < 30$ atm and has been recently used in the ignition study in [40]. Within the context of this study, this scheme has been compared to the skeletal mechanism GRI 3.0 and an in-house 22 species methane scheme (S22R195, derived from the detailed scheme from the CRECK group at PoliMi [41]) in a counterflow premixed flame at different operating conditions as detailed in [42]. Results show a maximum difference of 15% in consumption speed and 10% in species profiles for strain levels ranging from 0 to 20000 s^{-1} . A summary of these results for inlet conditions of $p=1$ bar, 320K and $\phi = 0.83$ is shown in Fig. 12 and Fig. 13. These conditions are representative of operating conditions under study. Note that the source term for species H_2O_2 and HO_2 was reformulated in a semi-implicit integration following the methodology described in [31] to avoid numerical oscillations due to stiffness.

3.3. Ignition modelling

The energy supplied by the igniter was modelled as a source term in the energy equation following the model in [43]. The temporal evolution of power of this igniter measured in the experiment was introduced in the model. This power profile was shown in Fig. 7 (right). Note that the amount of energy is higher and the deposition time lower than in conventional ignition systems such as pin-pin electrodes [7]. Two electrical to thermal efficiency values were used during the energy deposition. For the breakdown ($t < 20ns$), $\eta = 0.95$, similarly to [7], while the rest of the process used a constant efficiency measured in the experiment equal to 0.137. To avoid any bias due to the igniter model parameters, the volume where energy is deposited is a sphere whose radius and position have been adjusted so as to match the

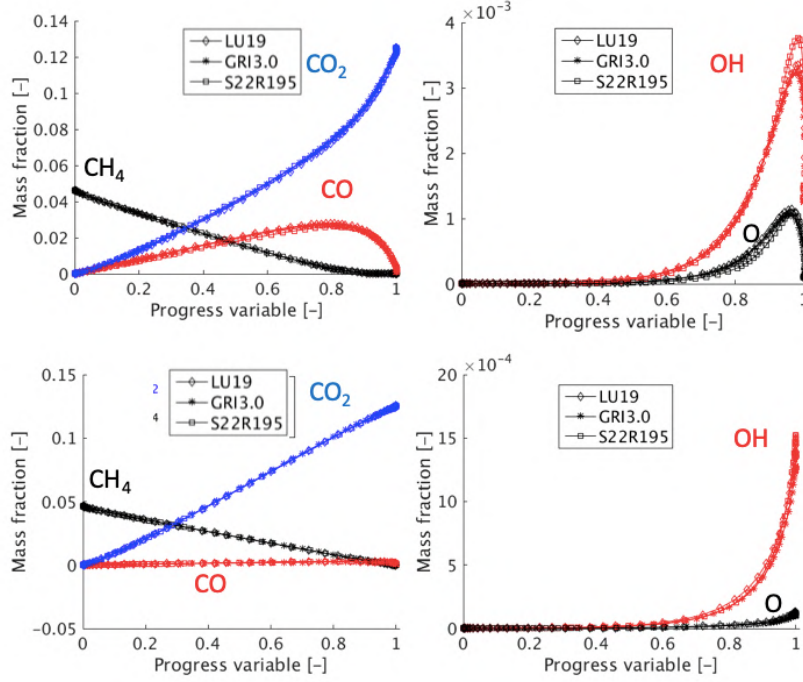


Figure 12: Methane-air combustion: comparison of species profiles in a lean premixed flame for inlet conditions of $p=1$ bar, 320K and $\phi = 0.83$ at strain rate equal to 2000 s^{-1} (top) and 20000 s^{-1} (bottom) or Lu19, GRI3.0 and S22R195. Progress variable based on CH_4 .

kernel penetration depth observed at several pressure conditions and times (as in Table 1). The deposited energy spatial profile has a hyperbolic tangent shape so as to provide a smooth transition to zero outside the ignition zone. The maximum temperature attained at the kernel location is approximately 10000 K . All thermodynamic properties were obtained from the reference NASA database [44] up to 5000 K and extrapolated to higher values assuming constant C_p and γ equal to those at $T=5000\text{ K}$.

Three different tetrahedral meshes were used during this study. During the filling phase of each case, a coarse 18-million cell mesh was used with a mesh size of 0.2 mm in the injection channel and 0.33 mm for the rest of the chamber. During the time of energy deposition, this coarse mesh is refined in a spherical region around the igniter of 10 mm of radius with a resolution varying linearly between $75\text{ }\mu\text{m}$ and 0.33 mm at the external radius in order

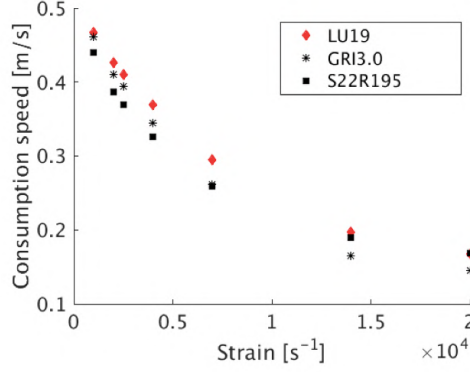


Figure 13: Consumption speed versus strain rate for a strained premixed flame at inlet conditions of $p=1$ bar, 320 K and $\phi = 0.83$ for Lu19, GRI3.0 and S22R195

to resolve the strong temperature gradients as shown in Fig. 14. Most importantly, this refinement avoids the use of the thickened flame model during the first instants of ignition when the kernel does not have a propagating flame structure [45]. One millisecond after energy deposition finishes it is posited that the flame has a propagating flame structure, which is in accordance with the Schlieren images in the experiment. At this instant, the solution is interpolated into the third and final mesh, which is characterized by a uniform cell size of 0.2 mm everywhere (44 million cells). Note that y^+ remains below 30 in all the wall regions.

3.4. Computational Cost

Each calculation with the static version of efficiency necessitated on average 483,000 processor hours to be completed. On the other hand, the use of the dynamic formulation required an average of 1,161,000 processor hours due to the additional filtering operations to calculate the parameter β . However, a recent study [38] has reduced this additional cost to an overload of 15% by introducing a modification in which the additional operations for the dynamic formulation are only performed in the regions of the domain where thickness is greater than one. The study in [38] included a section which verified that the PDF β remained identical to the case where the filtering operations were applied to the whole domain. In addition to that, [38] limited the frequency of calculating β to once every 750 iterations. These modifica-

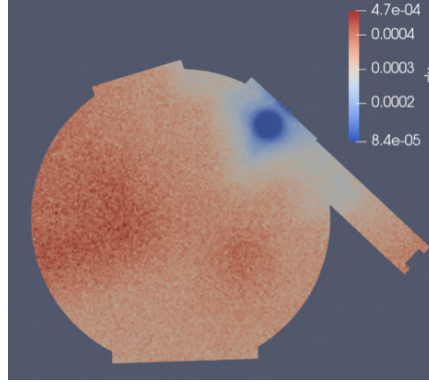


Figure 14: Overview of the mesh with mesh sizes in meters used during the energy deposition instants showing mesh refinement near the igniter

tions could have a similar impact in the radius chamber case, but due to the publication of [38] after the computations were finished, the operations for calculating β are made every iteration in the whole domain.

4. Results

4.1. Laminar case

In order to validate the modelling of the wall heat losses, a laminar case with $p_{ini} = 1\text{bar}$, $T_{ini} = 300\text{K}$ and $\phi = 0.7$ was performed. Fig. 15 shows the Schlieren and density gradient images at $t=38\text{ ms}$ and Fig. 16 shows the temporal evolution of pressure up to 45 ms . The same propagation speed was recovered as in the experiments. This offers evidence that heat losses are well modelled for this configuration.

4.2. Turbulent cases: non-reacting filling phase

Three operating conditions are discussed in detail and are presented in Table 2 and are representative of the operating conditions of an actual Safran SCT engine. All were performed with a $\phi = 0.7$ mixture. Turbulent case A is characterized by injection at 1 bar (lowest injection velocity, note that inlet is a constant mass flow rate boundary, thus velocity diminishes as the density increases as the chamber fills up), and a short delay between end of injection and sparking (5ms), which results in high turbulence intensity and, therefore, a high Karlovitz number (Ka). Turbulent case B ignition starts at

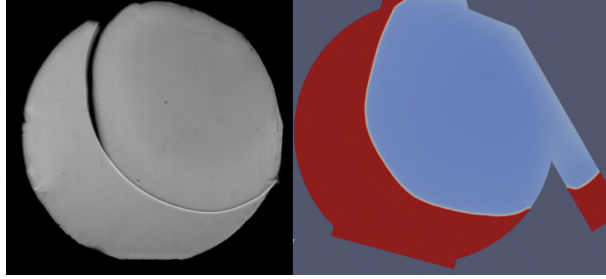


Figure 15: Flame visualization at $t=38$ ms after sparking. Experiment Schlieren (Left). Computation: normalized density field (Right)

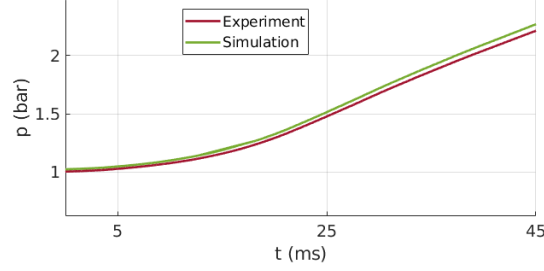


Figure 16: Temporal pressure evolution for a case with $p_{ini}=1\text{bar}$, $T_{ini}=300$ K, $\phi=0.7$

0.5 bar (highest injection velocity) and has the same delay of 5 ms, thus also having a strong turbulent intensity. However, Ka has changed due to faster diffusion. Finally, turbulent case C also starts at 0.5 bar (higher injection velocity), but its longer delay (30 ms) attenuates the turbulence intensity with respect to cases A and B. The three cases also feature different strain rate at the igniter location, highest for case B.

Table 2: Cases under study. Sparking delay refers to the time between the end of injection and sparking. The turbulence and strain levels are obtained from the tangential velocity at a position 1 mm above the igniter and are averaged values over $\pm 1\text{ms}$ around the time of sparking

Case name	P at end of injection	Sparking Delay	Strain level at igniter start	Ka	Re
Case A	1 bar	5 ms	3600 s^{-1}	63.2	31200
Case B	0.5 bar	5 ms	5500 s^{-1}	3.4	22200
Case C	0.5 bar	30 ms	3300 s^{-1}	0.6	13900

The turbulent field is well resolved except for small regions where recirculation occurs (injector exit and corners), as can be seen in Fig. 17 and Fig. 18 showing fields of ν_{turb}/ν_{lam} . Fig. 19 shows the probability density function (PDF) of tangential and radial velocities around the igniter observed in cases at the sparking instant for cases A, B and C. The wider distributions in A and B (short delays) with respect to C (long delay) give evidence of higher turbulence levels. On the other hand, the mean value of the tangential velocity in B corroborates the higher injection velocity compared to cases A and C.

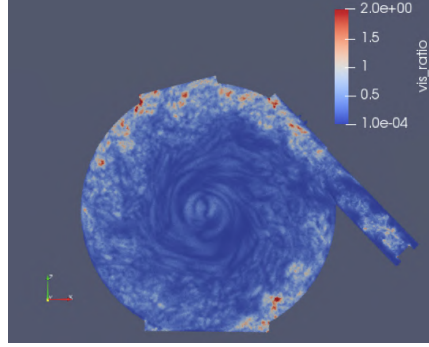


Figure 17: Field of ν_{turb}/ν_{lam} after filling up to $p=1$ bar for case A, before spark

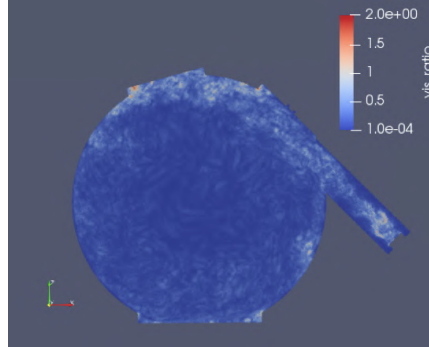


Figure 18: Field of ν_{turb}/ν_{lam} after filling up to $p=0.5$ bar for case B, before spark

4.3. Turbulent cases: reacting phase using the static efficiency formulation

Figure 20 contains the experimental Schlieren images for cases A, B and C. Figure 21 shows a comparison of the temporal evolution of pressure

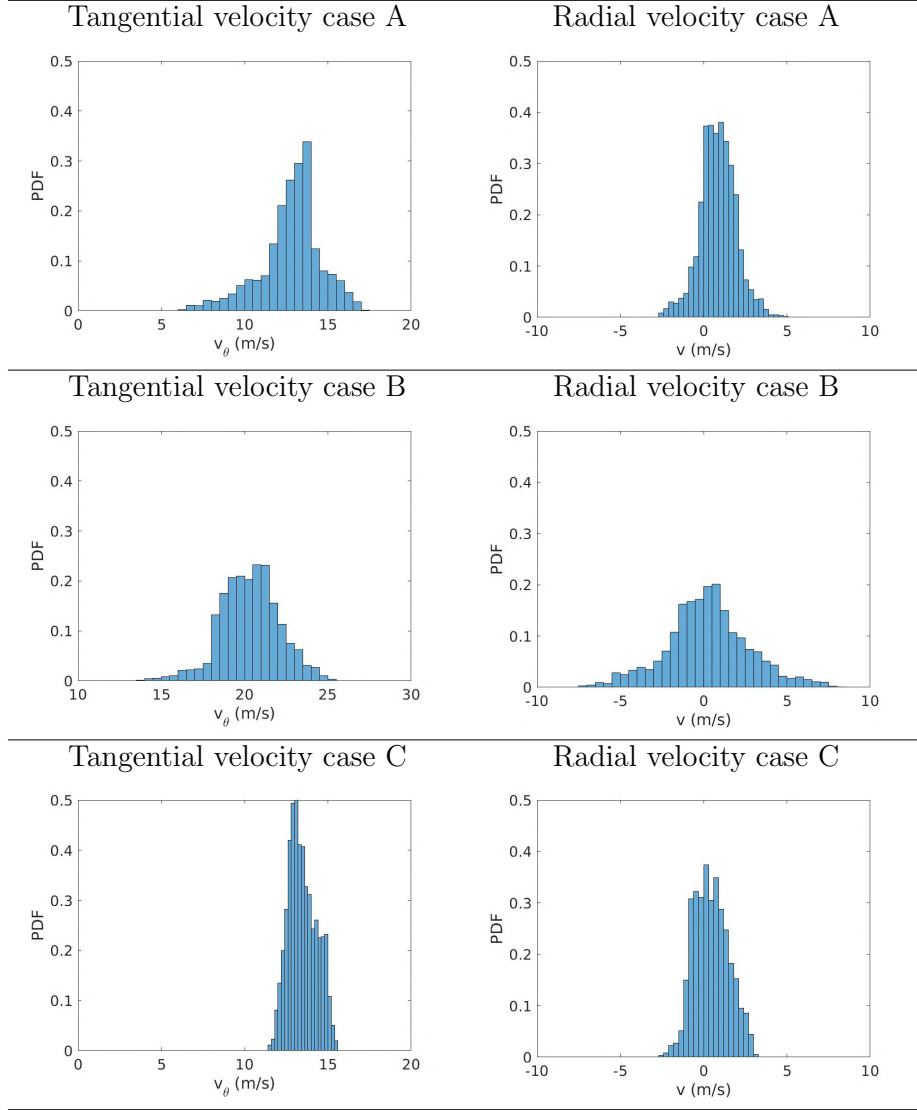


Figure 19: Comparison of PDFs of tangential and radial velocities in a sphere of $R=3.5\text{cm}$ around the igniter location at the time of spark for the three cases

between experiments and LES for the three cases. The pressure signals have been postprocessed with a moving average window of period 0.2 ms. The time for maximum pressure in case A is smaller than cases B and C due to

the higher Ka (more flame surface wrinkling). This is manifested by a higher efficiency value for case A, as it will be shown later in this section. A complete comparison of Schlieren images from experiments and LES density-gradient line-of-sight integration (LOS) can be found in the Supplementary Material. The pressure curve and image series demonstrate an overall good agreement between LES and experiments for all cases. Although the temporal evolution of pressure curves of LES in cases A, B and C are different, they can be decomposed into similar phases, which will be explained.

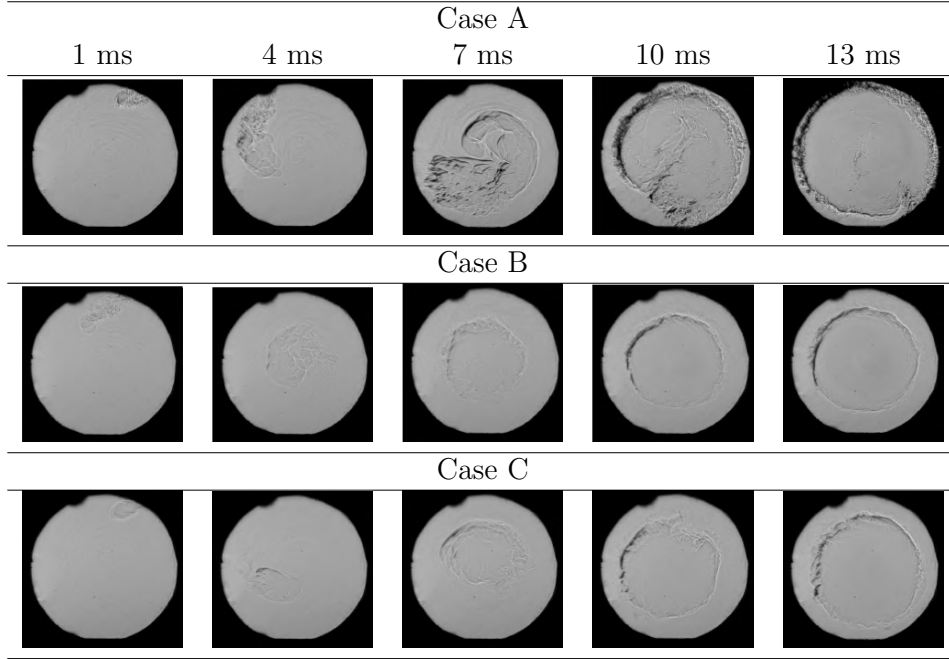


Figure 20: Experimental Schlieren images for cases A, B, C. The first image corresponds to 1 ms after sparking and each successive image is taken at an interval of 3 ms

In case A, both experiment and LES show a small pressure rise during the first 5 ms after spark (phase I), where the flame kernel propagation is governed mainly by the flow straining effects and is only weakly affected by turbulence. After the kernel has reached the chamber center at $t \approx 5$ ms, it starts to expand radially (phase II). This behavior is also shown graphically in the LOS images in the Supplementary Material. LES displays a slightly delayed kernel rotation. This effect may be attributed to the uniform mass

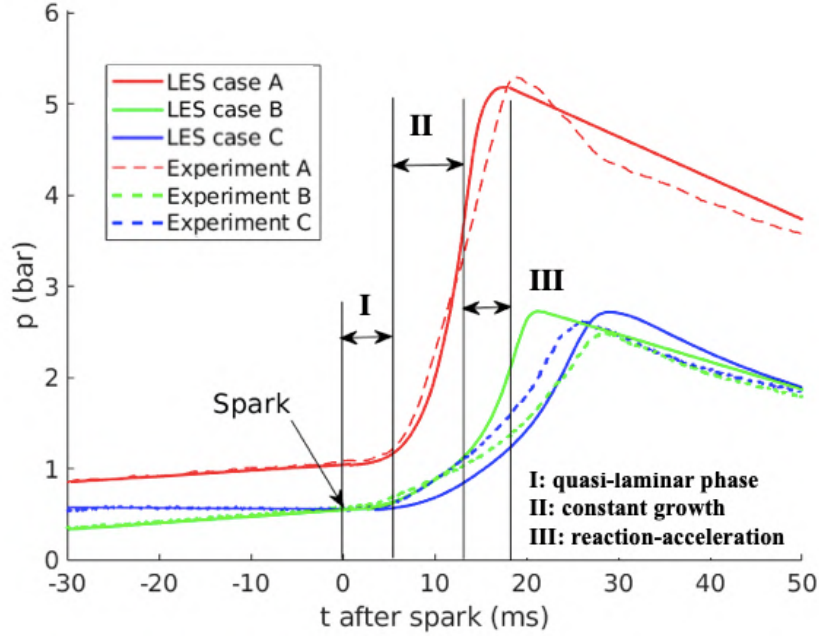


Figure 21: Temporal evolution of pressure for cases A, B and C. The evolution in all cases can be divided in three phases: a) weak turbulence interaction b) constant flame growth c) burning rate acceleration

flow rate boundary which does not take into account the actual shape of the inlet velocity profile. Around $t \approx 10$ ms the flame boundary is close to the walls and there is an acceleration of the burning rate (phase III). Complete combustion is reached at $t \approx 20$ ms.

In case B, identically to case A, both the experiment and LES show a weak pressure rise while the flame kernel is being strained and travels towards the center of the chamber (phase I). It is followed by a radial expansion (phase II), where the flame kernel expands radially at a constant rate. This behavior is well represented in LES. After $t \approx 20$ ms, the third phase (phase III) is much more visible. As for case A, in this phase, the consumption speed in LES case B is faster than in experiments. This reaction acceleration when the flame comes closer to the walls is linked to the interaction of the flame with the vorticity generated near the walls.

Case C has an overall similar behavior to cases A and B, but the pressure

rise is slower due to the attenuation of turbulence during the delay between end of ignition and sparking. Similarly to case A, the comparison of the Schlieren and LOS images during the first instants ($t < 10$ ms) indicates a slight delay of kernel rotation in computations, which is attributed to the aforementioned uncertainty in the initial inlet velocity profile. The flame anchors at the center of the chamber at $t=7$ ms in experiments and $t=10$ ms in the LES and the constant growth phase (phase II) starts. Phase II ends at $t \approx 20$ ms, and acceleration occurs then to reach complete combustion at $t \approx 30$ ms in the LES. The pressure rises when the flame comes closer to the walls, which results in a narrowing of the delay between LES and the experiment. Similarly to cases A and B, this behavior can also be seen in the LOS images in the Supplementary Material.

Figure 22 displays the temporal evolution of the change of pressure with time. The three distinct phases are clearly visible for case B and C. There is a small pressure rise in phase I, an almost constant value for case II, and an acceleration for phase III.

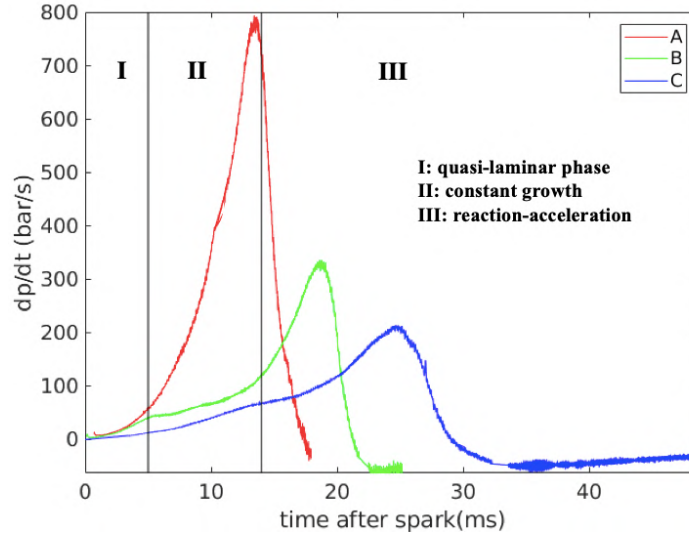


Figure 22: Temporal evolution of the change of pressure with time with lines indicating the phase boundaries for cases B and C. For case A, phase II takes place between $5 < t < 10$ ms

Cases B and C, both at an initial pressure of 0.5 bar, permit a direct comparison of the effect of turbulence and strain. In experiments, realizations

of case B have a faster pressure rise than case C for $0 < t < 10$ ms after spark. During this time, the pressure rise is mainly governed by the straining effects in the flame kernel, and the higher strain in case B is represented by a faster pressure increase. At $t=10$ ms the pressure curves of B and C intersect and for $t > 10$ ms the pressure growth for both cases are similar, with case C having a slightly faster growth rate. Eventually, case C attains its pressure peak 1 ms before case B. The behavior in the experiments for $t > 10$ ms can be explained by the fact that the pressure rise during this phase is mainly governed by the effect of turbulence. In case B, at $t=10$ ms, the viscosity has increased due to the earlier growth of the flame kernel during the straining phase. In addition to that, a higher decay rate of turbulence level in case B may rend the turbulence intensity closer to case C for $t \geq 10$ ms. By the combination of these effects, the turbulence levels in B and C may attain similar levels for $t > 10$ ms and, therefore, experience a similar growth rate. The LES calculations replicate the same trend as experiments: during $t < 10$ ms case B rises faster than C, and for $t > 10$ ms the growth rate between B and C is similar. However, LES computations fail to replicate the intersection of pressure curves. The main reason for this is the fact that, for case C, LES predicts that the flame kernel is not wrinkled by turbulence until $t > 10$ ms, while Schlieren images of experiment C indicate a substantial level of wrinkling for $t \geq 7$ ms. For that reason, LES display a delay in pressure rise which originates at $t < 10$ ms which is maintained constant for $10 < t < 20$ ms. For $20 < t < 30$ ms the acceleration in reaction rate predicted by the LES serves to decrease this delay and both LES and experiments attain their peak within 1 ms.

The reason for the delay of LES case C in the effect of turbulence on pressure rise may be found in the formulation of u'_Δ in [35], which is indicated in Eq. 12. This formulation, which has been originally developed in homogeneous isotropic turbulence flows, predicts low values of u'_Δ/S_l near the chamber center in this configuration (which is close to solid rotation conditions) as shown in Fig. 23. Consequently, the values of wrinkling factor in case C while the flame surface is located near the center of the chamber ($7 < t < 10$ ms) is low and the pressure rise is slow. The effect of small values of u'_Δ/S_l near the chamber center is not felt as strongly in cases A and B because in these cases the flame grows faster during the phase governed by straining. As a result of that, in cases A and B, the flame surface is located inside the region of high u'_Δ near the wall boundaries by the time the pressure rise becomes governed by turbulence.

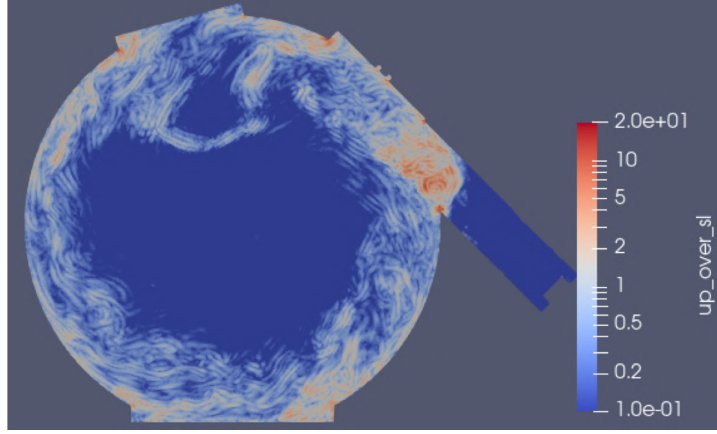


Figure 23: Contour fields of u'_Δ/S_l for case C at $t=2$ ms after spark showing low values near the chamber center

In summary, the overall agreement is fairly good, and the pressure rise and kernel shape are well modelled with small time differences. In case A, the LES calculation predicts the pressure peak at $t=17$ ms, 2 ms before the experiment. In case B, the pressure rise during phases I and II follows the experiments. However, the reaction acceleration predicted by the static formulation of the wrinkling factor (phase III) is the origin for the anticipation of the pressure peak instant in LES ($t_{peak} = 21$ ms) with respect to the experiments ($t_{peak} = 28$ ms). Finally, in case C, the time to reach the pressure peak in the calculation is 29 ms, which is 1 ms longer than in the experiment. To better understand the turbulent flame behavior, additional analysis of the LES is now performed.

Figure 24 shows the numerical temporal evolution of the resolved surface-to-volume ratio of the flame kernel S/V using an isovolume of progress variable based on temperature at the value $q = 0.65 \pm 0.15$. For all cases, S/V grows with time as the flame becomes more wrinkled due to turbulence. Note that the evolution of S/V shows a linear growth in time that does not replicate the pressure rise acceleration when the flame approaches the walls ($t \approx 10$ ms for cases A and B). The impact of turbulence on the flame speed is the combination of both the flame wrinkling and the local modification of the flame front, which is characterized by its local burning intensity. The flame wrinkling can be estimated with S/V : as Fig. 24 shows, it does not increase as fast as the pressure does. This leads to the conclusion that the

local flame burning intensity has changed.

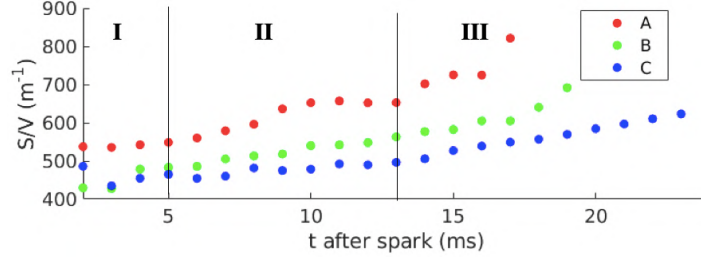


Figure 24: Temporal evolution from LES of the surface to volume ratio of the kernel S/V for the three cases. Phases I, II, III indicated for case A as for Fig. 21

To further investigate this question, Fig. 25 shows the temporal behavior of the wrinkling factor averaged over the region $q = 0.65 \pm 0.15$ for the three cases. Higher efficiency values are accompanied by higher consumption speeds. The evolution of the three cases is very similar to the pressure curves in Fig. 21. For $t < 5$ ms (phase I) efficiency decreases and reaches a minimum. Efficiency starts to grow again when the flame stabilizes around the center of the chamber (phase II). Finally, there is a noticeable increase of efficiency when the flame comes closer to the walls (phase III) (case A; $t \approx 10$ ms, and case B; $t \approx 20$ ms).

The impact of the walls is confirmed in Fig. 26, showing only the regions where wall-originated vorticity interacts with the flame have an efficiency increasing noticeably over 1.

Because the interaction between the turbulence at the walls and the flame as well as the compression caused by the flame expansion may not be well modelled with the static formulation of efficiency, the dynamic formulation of efficiency is now studied.

4.4. Turbulent cases: using the dynamic efficiency formulation

In the static efficiency formulation, the coefficient β is a constant set to 0.5 based on standard academic turbulent flows. It has been however demonstrated that β may vary significantly in transient cases or complex turbulent flows. As the discrepancies between LES and experiment are here attributed to the incorrect wrinkling factor near the walls when β is fixed to 0.5, a more accurate formulation is attempted with the dynamic formulation presented in Section 3.

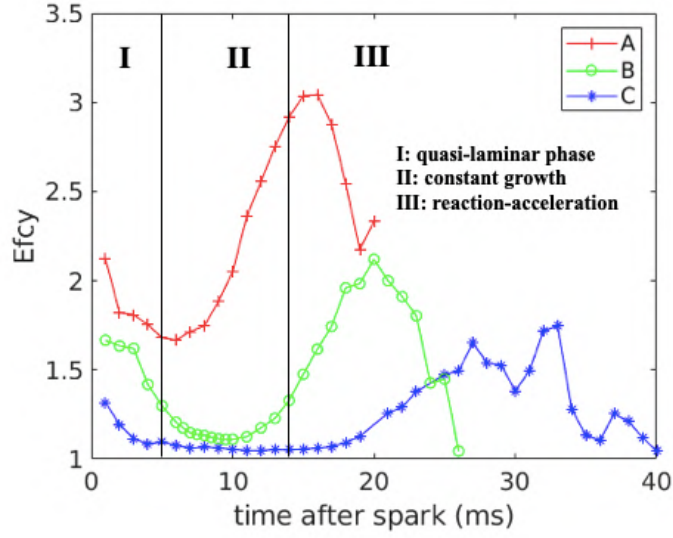


Figure 25: Temporal evolution of the average value of efficiency in $q = 0.65 \pm 0.15$ for the three cases

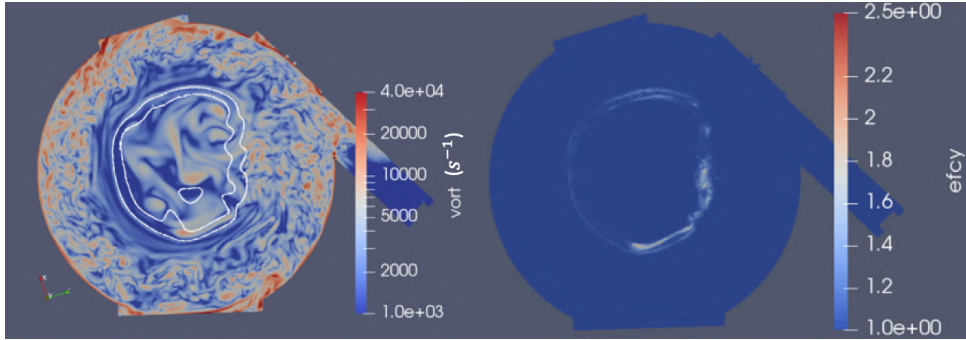


Figure 26: Left: Vorticity field with a white contour indicating flame position for case B at $t=6\text{ms}$ after spark. Right: Efficiency field showing high values only where wall-originated vorticity is interacting with the flame.

Figure 27 shows the pressure evolution obtained with both formulations of efficiency. From spark to the constant radial growth phase (phases I and II), both static and dynamic efficiency formulations predict the same rate of pressure increase. As expected, the difference becomes visible when the flame approaches the walls, where the dynamic modelling predicts a lower

consumption speed, hence slower pressure rise, than the static model for all cases: case A attains its peak at $t=19$ ms (+2 ms compared to the static efficiency formulation), case B at $t=27$ ms (+6 ms compared to the static formulation) and C at $t=31$ ms (+2 ms compared to the static formulation). To be compared to the experimental peaks reached at $t=19$, 25 and 28 ms respectively. Interestingly in [38], a study of ignition in the MICCA-spray setup, found a 15% to 20% reduction in the consumption speed when employing the dynamic wrinkling factor with respect to the static version. This reduction, which was determined to be linked to a reduction in the subgrid-scale wrinkling, caused a corresponding increase in ignition delay time of the same magnitude as in the present study. It is to be noted that, for case C, both wrinkling factor formulations indicate a delayed initial pressure rise with respect to experiments which has been explained by the low value of u'_Δ/S_L predicted near the chamber center. While this time difference is maintained approximately constant in the dynamic formulation, the overprediction of reaction rate by the static formulation during phase III renders reduces this time difference, but due to unphysical effects.

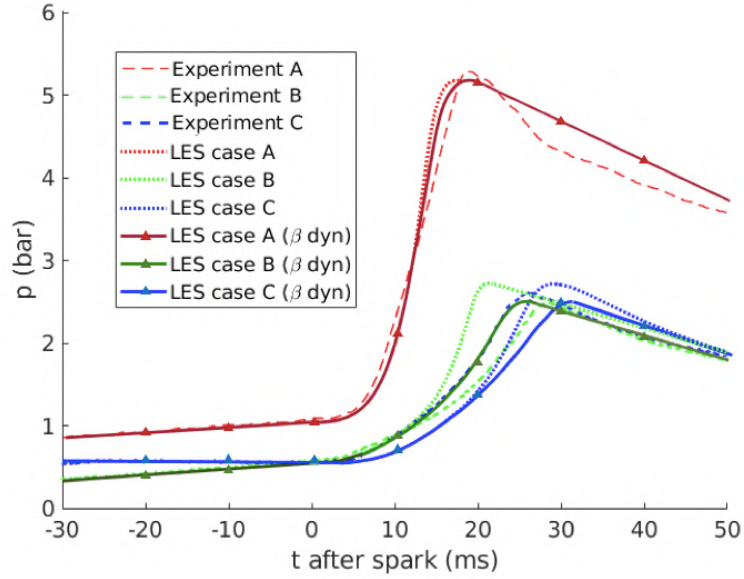


Figure 27: Left: Pressure time evolution for the three cases obtained with the static and the dynamic efficiency formulations and compared to experiments

The figures in Fig. 28 clearly indicate that the lower consumption speed predicted with the dynamic formulation is directly linked to lower values of efficiency at times $t > 7$ ms.

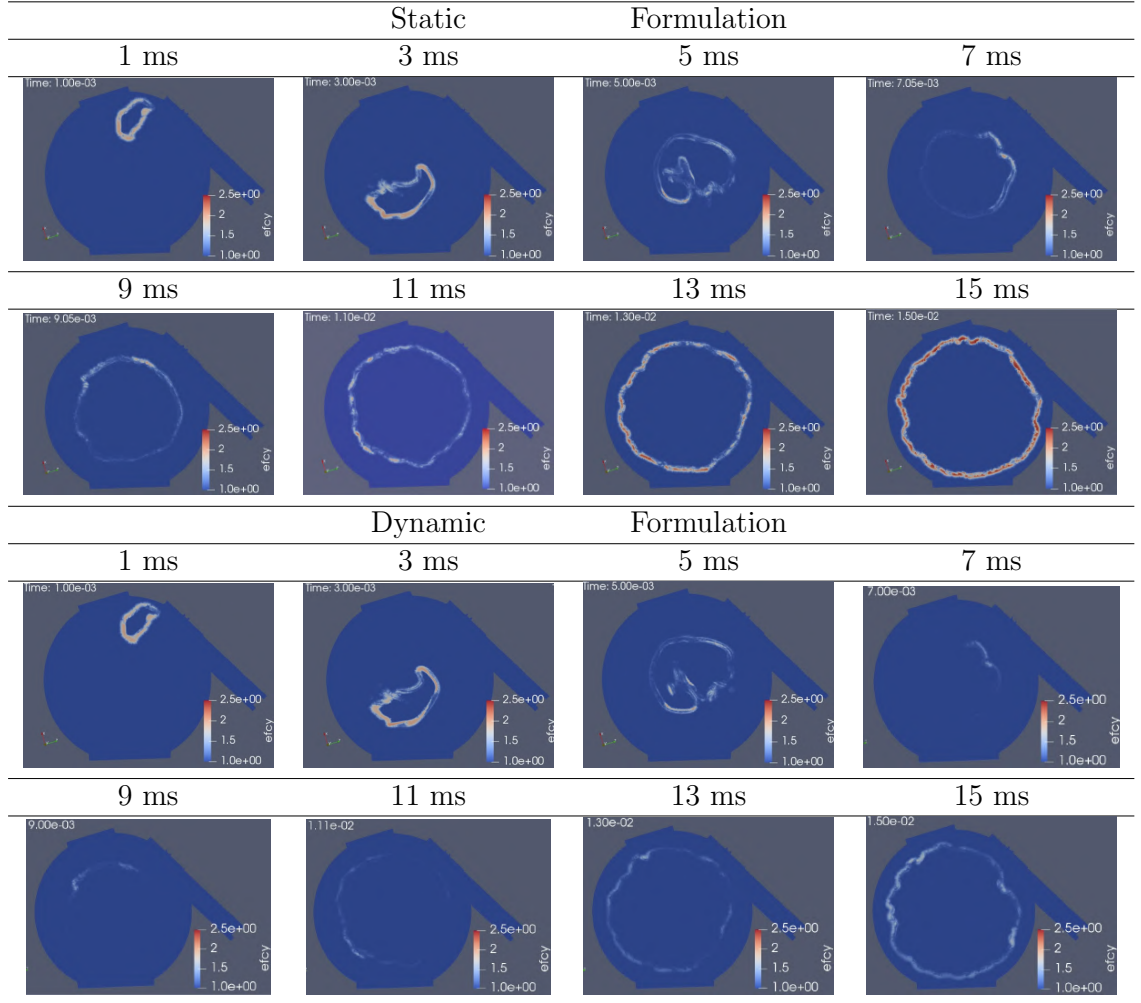


Figure 28: Comparison of static and dynamic efficiency values in the middle plane for case B showing lower efficiency values of the dynamic formulation at later instants. Top: static formulation, bottom: dynamic formulation. Time after spark

Figure 29 reveals that for these conditions, on average, the parameter β

is smaller than the fixed value assumed in the static formulation $\beta = 0.5$. In particular, β only nears 0.5 in isolated highly curved regions.

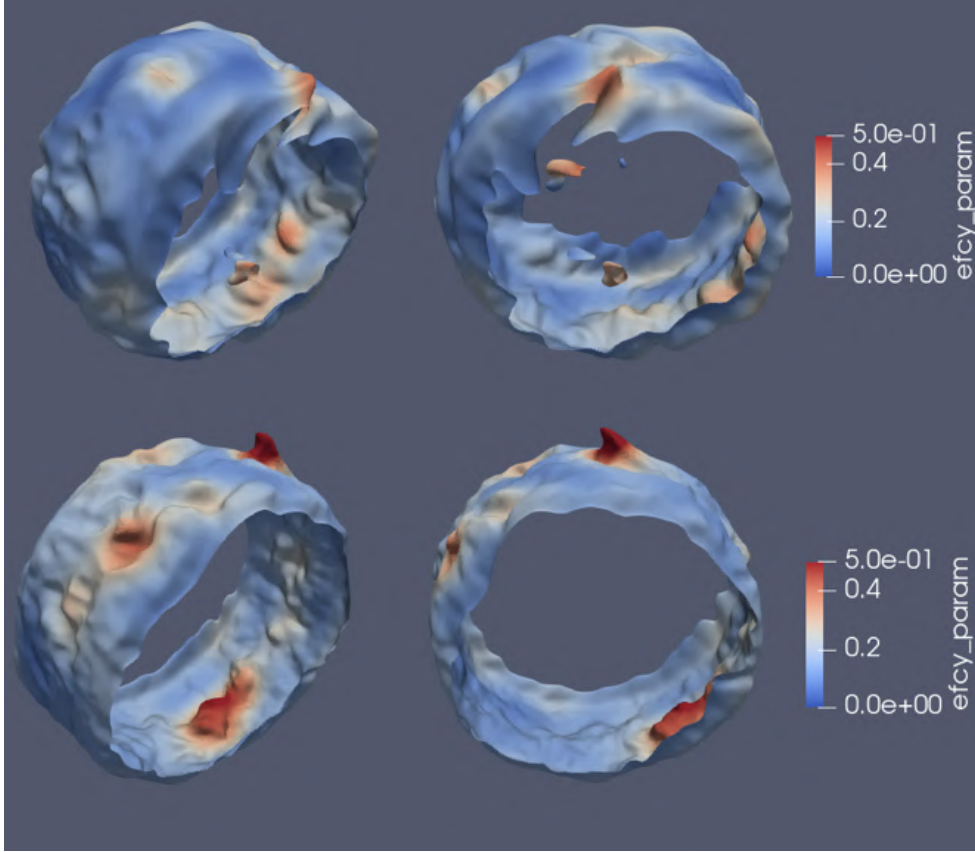


Figure 29: Case B lateral and frontal views of β for an isosurface of $q=0.5$ at $t=15$ ms (up) and $t=20$ ms after spark (bottom) showing that, on average, $\beta < 0.5$ for these conditions

Figure 30 displays the temporal evolution of the volume-averaged value of β over the isovolume of $q = 0.65 \pm 0.15$. For all cases, the average β increases from the start to the point of maximum pressure. During the first instants, the kernel develops far from the walls and the effect of turbulence on the kernel is weak. As the kernel expands radially, it interacts with the vorticity generated at the walls and this is translated into higher values of β . The case initially at atmospheric pressure (case A) shows the highest increase, starting at an average value of β equal to 0.2 and increasing to 0.5.

For cases B and C, the averaged value of β starts at 0.1 and increases to 0.2 approximately. Results in Fig. 30 are consistent with the high Karlovitz number in case A. The average efficiency value being closer to 0.5 is also consistent with the moderate impact of the dynamic efficiency for case A.

Probability density functions of β for cases A, B, C for an isosurface of $q = 0.5$ are shown in the Supplementary Material. They all show a mostly monomodal distribution with an average of 0.15 (cases B and C) and 0.35 (case A). As a comparison in [38], the ignition of the annular MICCA-spray chamber led to bimodal β distributions, the first peak centered around 0.15 and the second peak at 0.65 (the second peak associated to the flame region near the injectors). In the present configuration, the flame is subjected to comparable levels of wrinkling corresponding to the low values of β . Because there is no flame-anchoring mechanism, no high values of β are found. The PDF of β becomes narrower with time indicating the transition between an isothermal turbulent flow field to a growing hot gas expansion flow. The fact that β in the radius chamber experiment is different and lower than the average value found in [37] can be explained by two factors. First, it has to be noted that β is an increasing function of u'/s_L [37] and cases A, B, C feature $u'/s_L \approx 1$ while cases shown in [37] feature $u'/s_L \geq 10$. In addition to that, the parameter β is case-dependent and the radius chamber turbulent field is different from the decaying isotropic turbulence in [37].

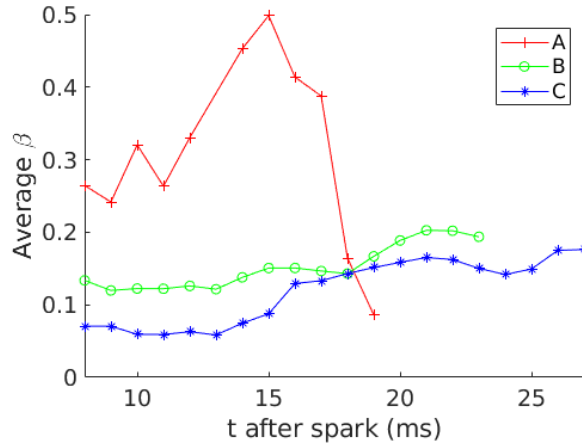


Figure 30: Evolution of the volume integral of the wrinkling parameter β over an isovolume of $q = 0.65 \pm 0.15$

5. Conclusions

Ignition in a lab-scale configuration reproducing conditions of a SCT chamber has been studied both experimentally and numerically by means of LES. The effect of the igniter on the fluid in LES has been modelled using a source of energy in the Navier-Stokes equations with the energy value and temporal variation experimentally measured with probes and using a recently developed Speckle Background Oriented Schlieren visualization technique.

Three different conditions, which are representative of actual engine operating conditions, have been carried out experimentally by varying the initial pressure in the chamber as well as the delay between the end of injection and sparking. The experiment shows that the pressure rise in all cases can be divided into an early phase when the strain effects are dominant, and a later phase when the turbulence governs the pressure rise. In order to perform the LES, a static formulation of the wrinkling factor has been used which, for some cases, indicates a low value of the reaction rate when the flame surface is far from the chamber walls, while it overpredicts the reaction rate when the flame surface is near to the chamber walls. A dynamic formulation of the wrinkling factor has been used which does not present the overprediction of reaction rate near to the walls. The main difference between formulations is the smaller value of wrinkling factor in the dynamic case when the flame interacts with the vorticity generated at the walls. This is due to the fact that the wrinkling factor parameter (β) for this configuration is generally lower than the value proposed in the static formulation (0.5).

Acknowledgements

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