

Effect of Pressure and Dilution on Flame Front Displacement in Boosted Spark-Ignition Engine Combustion

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ABSTRACT

The development of new types of engines (*such as, the 'downsizing' engine*) needs new characterizations and understandings of the combustion process inside the combustion chamber. Indeed, the usual turbulent premixed combustion models to represent as well as possible the combustion processes occurring in spark ignition (S.I) engine are based on the "flamelet" theory. But the operation mode for Boosting S.I engines are not usual for flame development: high pressure level and high dilution rate. Therefore, this paper investigates experimentally the local flame propagation of iso-octane-air-diluents stoichiometric mixtures in a boosted S.I engine. The Mie scattering laser tomography technique is used to obtain the local flame front displacement in typical operating conditions of a 'downsizing' S.I. engine (*i.e. high pressure level and high dilution rate*). The turbulence and the flame front displacement characterizations are performed for the same burnt fraction angle (~5%) over the intake pressure and Exhaust Gas Recirculation (EGR) ranges: pressure = 0.70 to 1.50 bars and EGR = 10 to 35% by volume. Thank to the mean flame front displacement velocity (S_d) and the mean aerodynamic flow field (U_{Gas}) determined from the laser tomography images and Particle Image Velocimetry (PIV), turbulent burning velocity (S_T) is estimated. Thus, the behaviour of the turbulent burning velocity should be analyzed in order to provide a more comprehension of the combustion in "downsizing" conditions and a database for testing various models of premixed turbulent combustion. A number of various empirical correlations for the combustion rate (S_T/S_L^0), available in the literature, are then tested and compared with our experimental trends.

The results show an increase in S_T by pressure despite the decrease in the laminar burning velocity (S_L^0) and a decrease in S_T by EGR. In spite of some discrepancies and that this empirical correlations are based on the "flamelet" theory, theory still liable to discussion in

"downsizing" conditions, good agreement between correlation and experimental results are obtained.

INTRODUCTION

A large proportion of engine operation is spent at part-load and this is where the largest fuel economy savings may be achieved. So, one way to reduce fuel consumption and so the greenhouse gas CO₂ emissions would be to increase the internal efficiency. Nowadays, "downsizing" (*i.e. reduction of engine size and increase of intake pressure by turbo-charging*) represents a major way of improving engine efficiency thanks to pumping and friction losses reduction [1, 2, 3, 4]. On the other side, it is difficult to provide the quintessence of downsizing due to the risk of knocking. One solution to avoid these troubles is to lower the compression ratio of the engine. However, this would result in a reduction of combustion thermal efficiency, thus partly destroying the pursued target. So Exhaust Gas Recirculation (EGR) presents an attractive potential to reduce knocking and NO_x emissions by diluting the air-fuel mixture [5].

At the present time, the usual turbulent premixed combustion models to represent as well as possible the combustion processes occurring in S.I. engine are based on the "flamelet" theory. The most important assumption flamelet modeling is related to the topology of the flow and the fact that fresh and burned gases are separated by thin continuous front where chemical reactions take place [6]. According to the concept of most conventional turbulent combustion theory, the ratio of the turbulent burning velocity to the local burning velocity is proportional to the increase of the ratio of the flame surface area caused by turbulence to the laminar flame surface area. The area ratio is well approximated to be proportional to the ratio of the turbulent intensity to the local burning velocity, where the local burning velocity is usually taken to be equal to the laminar burning velocity [7, 8]. Several works [9, 10] have demonstrated that the laminar burning velocities change

with pressure and temperature and that the chemistry and thermo-diffusion properties of hydrocarbons have an important effect on the combustion process. In this study, laminar burning velocities of isooctane/air mixture are estimated, in the engine conditions, using the PREMIX code of CHEMKIN package with dilution effect and the Hasse et al. [11] reduced chemical kinetic mechanism (48 reactions and 29 species).

The objective of this study is to investigate experimentally the effects of high pressure level and high dilution rate on the flame behavior. One of the most important parameter to characterize a premixed turbulent combustion is the turbulent velocity (S_T). Driscoll [12] reviews three unambiguous and not equal common definitions of the S_T . To some researchers, S_T is an indicator of the mass per second of reactants consumed. Thus, they define S_T as a global or local consumption speed. To others, S_T indicates how rapidly the flame front will traverse a certain distance. Thus, they define S_T as a displacement speed. In this study, it is desired to determine the distance covered by the front flame for a certain period of time, and so the displacement speed definition is chosen. This last definition should be defined as:

$$S_T = (S_d - U_{Gas}) \cdot n \quad (\text{Eq. 1})$$

Where S_d is the velocity of the flame front, U_{Gas} the velocity of the gas into which the flame front is propagating and n the normal to the flame front.

Numerous researchers have investigated turbulent flame speeds using different methods under various conditions and a large number of empirical burning rate correlations (S_T/S_L^0) have been proposed and tested [13 - 16]. The final aim of this paper is to test some of these correlations in a Boosted S.I. engine for different intake pressure and mixture conditions.

EXPERIMENTAL SETUP

The experiments were performed in a "transparent" boosted pent-roof single cylinder engine. Thus, flame front displacement measurements are allowed through the cylinder head accesses, a transparent top piston (ϕ 50mm) and mirror inclined at 45 degrees inserted in extended piston. This piston, equipped with graphite rings, slides along the transparent cylinder liner without oil lubrication. In this study, the engine speed was fixed at 1200 rpm and a standard plug was used to ignite the mixture. The spark timing was set to produce maximum indicated mean effective pressure where knocking did not occur. The intake air was seeded by small droplets of silicone oil for the Mie scattering and the seeding density was adjusted in accordance with operation conditions and optical technique to ensure acceptable quality of images. Moreover, the intake air pressure was recorded and regulated with a piezoelectric pressure transducer and the different flows (C_8H_{18} , N_2 , CO_2 and H_2O) were controlled, with a laboratory simulated EGR

system, in order to obtain the stoichiometric equivalence ratio. The EGR was simulated by a mixture of 73.5% N_2 , 12.5% CO_2 and 14.0% H_2O by volume: this composition represents burned gases issued to the complete combustion of an isooctane-air stoichiometric mixture, and was regulated at 50 °C with an electrical heated system.

The main specifications of this engine are summarized in Table 1. With all these operating conditions, the indicated mean effective pressure (IMEP) range was from 3.30 to 13.70 bars (annex A).

Item	Specifications
Type	4 valves/single cylinder
Bore	85 mm
Stroke	88 mm
Displacement volume	499 cm ³
Compression ratio	10.2
Fuel	Isooctane (C_8H_{18})
Engine speed	1200 rpm
Intake pressure	0.7 to 1.5 bars
Dilution	10 to 35%

Table 1: engine specifications and operating conditions

OPTICAL MEASUREMENTS TECHNIQUES

The silicon oil particles were illuminated with a thin light sheet (thickness around 0.5 mm) delivered from an Nd:Yag laser (Quantel, Twin) and a combination of spherical and cylindrical lenses. The output energy was 180 mJ for each pulse at 532 nm with pulse duration of 10 ns. The delay between the two pulses was setting using the software (Insight 3G) and varied according to the optical technique used: same apparatus for PIV and Mie scattering laser tomography technique are used. Images were obtained with a 2048 x 2048 pixels² TSI camera.

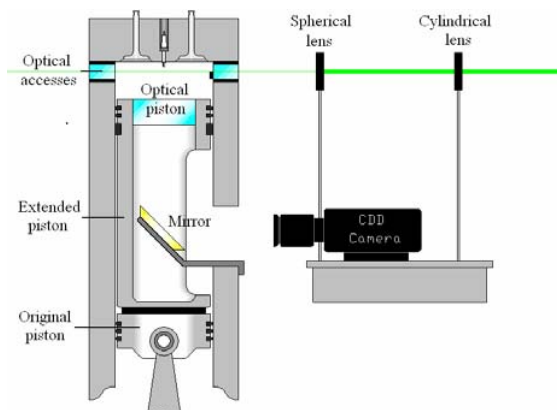


Figure 1: Experimental set-up for aerodynamic and flame front displacement characterization.

The experimental setup, described in the Figure 1, allows the characterisation of the internal aerodynamic,

the turbulence and the flame front displacement for the same burnt mass fraction angle ($\sim CA05$).

To estimate the burnt mass fraction angle ($CA05$), the conventional apparent heat release is taken: amounting effectively to a single zone model with a perfect gas assumption with no heat transfer and constant properties and composition during the closed valve period:

$$\delta Q_{Apparent} = \frac{\gamma}{\gamma-1} p.dV + \frac{1}{\gamma-1} V.dp \quad (Eq.2)$$

This apparent heat release can easily computed from the change in volume and measured pressure inside the cylinder. Thus, the visualization temperature is obtained by applying the ideal gas law.

LAMINAR BURNING VELOCITY

As the laminar burning velocity is needed in the most of the burning rate correlation, this first part focuses on this fundamental parameter. The available data on isooctane - air laminar burning velocity (S_L^0) are limited, and it is still difficult to determine the correct value of S_L^0 experimentally or/and numerically. After some previous experimental validations [17], S_L^0 is estimated for the different engine conditions, using the PREMIX code of CHEMKIN package with diffusion effect and Hasse et al. [11] kinetic mechanism. The cylinder temperature and pressure at $CA05$ (T_{CA05} and P_{CA05}) were also calculated to determine S_L^0 . The results are summarized in Table 2 and 3.

	P_{CA05} (bar)	T_{CA05} (K)	S_L^0 (cm.s ⁻¹)
0.70 bar	14.00	760	56.14
1.00 bar	20.00	790	55.20
1.30 bar	22.40	800	54.95
1.50 bar	28.00	830	56.33

Table 2: Laminar burning velocities for stoichiometric isooctane-air mixture for engine conditions at 20% EGR

	P_{CA05} (bar)	T_{CA05} (K)	S_L^0 (cm.s ⁻¹)
10% EGR	22.60	750	65.39
15% EGR	22.50	830	73.70
20% EGR	22.40	800	54.95
25% EGR	21.78	775	41.51
28% EGR	26.00	860	46.70
30% EGR	26.40	870	44.12
32% EGR	24.70	800	31.78
33% EGR	23.00	760	24.68
35% EGR	24.00	800	26.70

Table 3: Laminar burning velocities for stoichiometric isooctane-air mixture for engine conditions at 1.30 bars

TURBULENT CHARACTERISTICS

CHARACTERIZATION OF THE FLOW FIELD - Measurements of aerodynamic and turbulence characteristics were carried out with *PIV* technique. The

delay between the two laser pulses was 5 μ s and a 32 x 32 pixels² interrogation window with an overlapping of 75% was used to determine the internal aerodynamic for different intake pressures. The picture presented in Figure 2 was obtained from an average of 240 instantaneous field of velocity. Two recirculation zones are located from each side (*down and up*) of the intake flow.

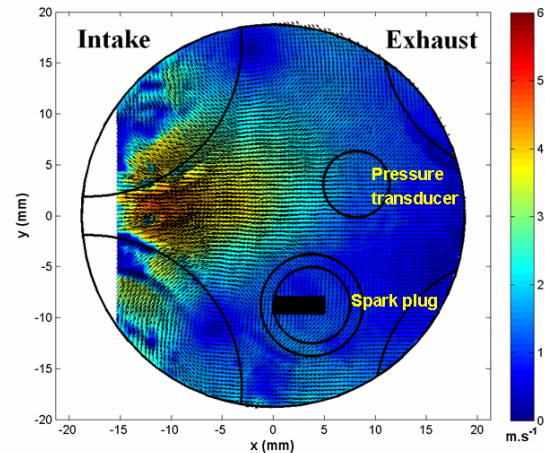


Figure 2: Mean flow field at 1.30 bar and 10 Crank Angle Degrees Before Top Dead Center

In order to have an adequate resolution for the turbulence characterization, the visualization region was reduced to a square of 24.0 x 24.0 mm², hence a magnification of 12 μ m/pixel. The main difficulty to characterize turbulence in engine is to identify the turbulent fluctuations and the cyclic variations from the aerodynamic flow. Thus, all the values, reported in this paper were extracted from instantaneous velocity fields by performing a Proper Orthogonal Decomposition (*POD*) analysis which is well adapted to this problem [18, 19]. Nevertheless, the choice of the order of truncation, which represents the limit between cyclic dispersions and turbulent fluctuations, is still not evident. There are several criteria of truncation: evolution of the proper modes structures or energy, the behaviour of the Eigen values, etc. In this present study, the evolution of proper mode energy has been chosen as criteria. From that, the integral length scale and the turbulent intensity have been determined. As illustrated in Figure 3, no clearly identified evolution can be drawn for the evolution of turbulence properties as function of the intake pressure.