Influence of Hydrodynamic Conditions on the Development of a Premixed Flame in a Closed Vessel

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ABSTRACT

During the cooperative Research Action "Modeling of combustion phenomena in piston engines" organized by the C.N.R.S., three identical variable hydrodynamic condition (vortex, turbulence) constant volume combustion chambers have been constructed. Different diagnostics : pressure history analysis and schlieren visualization, tomographic visualization and flame structure analysis, laser velocimeter analysis have been undertaken in three laboratories (Poitiers, Rouen, Marseille).

Capability of the device to reproduce the phenomena encountered in real engines is first tested. Influence of an initial turbulence, generated in the combustion chamber by the compression of a propane-air mixture through different perforated blocks, on flame propagation parameters such as flame speed, burning velocity are then studied in detail.

The arising problem is the estimation of the fluctuation intensity as it has also been pointed out in engines by several authors. It is shown that consideration of ensemble and cycle resolved standard-deviation defined with a cut off time parameter, leads to different values of turbulence intensity and characterization of an initial level of turbulence is relied on the combustion processes that are influenced by it. The first attempt to study spatial and temporal scales of turbulence is in course by means of a 2 points LDV method and provisional results are presented.

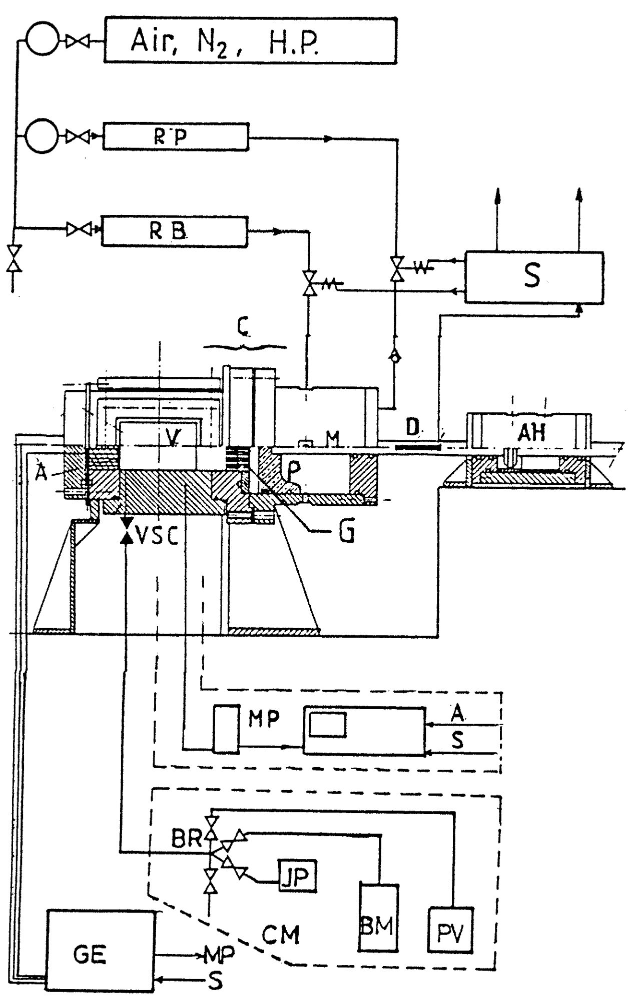
INTRODUCTION

Interactions between hydrodynamic flow field (mean flow and turbulence) and combustion has been studied for long years, both in continuous flow (1), (2) and in transient flow (3), (4). However, in the piston engine combustion domain, in spite of recent progress, mainly due to the use of large scale and high speed computers, there are not yet satisfactory numerical simulation codes. In our point of view, one of the reasons for this situation is the lack in comprehension of physical phenomena. In the French Cooperative Research Action (A.R.C.) on the "Modelling of the combustion phenomena in piston engines", important experimental works are, threrefore, included to provide data for model conception and validation. A new type of constant volume combustion chamber was developed to realize, with a good reproductivity, varous hydrodynamic conditions which are encountered in real engines (5). The purposes of this paper are : 1) to show how this chamber works and 2) to discuss the results obtained in the case of a flame propagating in a turbulent field in which the initial mean velocity is practically zero.

EXPERIMENTAL SETUP

There are different methods to study turbulent flame propagating in closed vessel. BRADLEY and his coworkers (3) have developed combustion chamber with fans which generate homogeneous and isotropic turbulence field. They study the influence of the turbulence on burning velocity during the pre-compression period. Recently ADOMEIT and his coworkers (4) developed a chamber with a moving grid and a moving piston. With this chamber, which can create compressed homogeneous turbulent field, they study the flame development in mixture with turbulence but, initially, without mean velocity under evolutive pressure conditions. In their chamber, cavity effects on combustion are avoided by careful fitting between the grid holes and the embossing on piston head.

As figure 1 shows, in the device developed for this study, various flow fields are generated by means of mixture injection from a variable volume tank T with a moving piston P (diameter : 120 mm, stroke : 50 mm) to the combustion chamber V (section : 60 mm x 60 mm, length : 150 mm) through an interchangeable flow generator G (grid, slit, channel) inserted between two parts. The tank and the chamber are initially filled by a same mixture with or without oil droplets. The piston is then activated and locked on the inlet surface of the flow



C: COMBUSTION CHAMBER
V: Visualization section
A: Multi-point ignitor
G: Flow generator
M: MOVING SECTION
T: Variable volume tank
P: Piston
AH:Hydraulic scock absorber
S: ELECTRONICAL SEQUENCER
RP, RB: DRIVING AND LOCKING AIR TANKS
GE: MULTI-SPARK GENERATOR (18mJ x 25)
MP: PRESSURE MEASURING SYSTEM
CM: MIXTURE FILLING SYSTEM

Fig.1: Schematic of the variable hydrodynamic costant volume combustion chamber.

generator by means of an electronically controlled (S) two stage compressed air supply (RP, RB) on its rear face. A multipoint parallel ignition system (A on Fig. 1 : 25 points, 18 mJ for each spark) is fitted at the other end of the chamber (V). 60 mm x 60 mm x 100 mm optical accesses are available on three surfaces, permitting to realize schlieren and tomography visualizations and Laser velocimetries from the ignition surface towards the flow generator.

The exposure duration, for schlieren pictures on ASA 400 film is less than 5 microseconds, while the tomographic ones are recorded in about 100 microseconds on a CCD camera equipped with an image-intersifyer and associated to a 20 W Argon-ion continous laser.

Velocity measurements are realized by a classical dual beam TSI LDV system. A two point, two probe-volume, LDV system, similar to one, investigated initially by Shafer (6), has been developed to measure integral and spatio-temporal scale under non reacting and reacting conditions.

Pressure measurement is done by means of a Kistler 601 piezo-electrical transducer located on the fourth surface. Central single-point ignition have exclusively been used in this work.

CHAMBER CHARACTERIZATION EXPERIMENTS

In spark ignited engines, combustion of premixed gas takes place in hydrodynamic fields with intense mean fluid motion (swirl and tumble) and local motion (turbulence). Such fluid motions enable sufficiently rapid combustions to ensure stable and economic running of engines. Consequently, three preliminary tests :

a) flame propagating in an initially quiescient mixture,

b) flame propagating in a distributed turbulent field without initial mean flow,

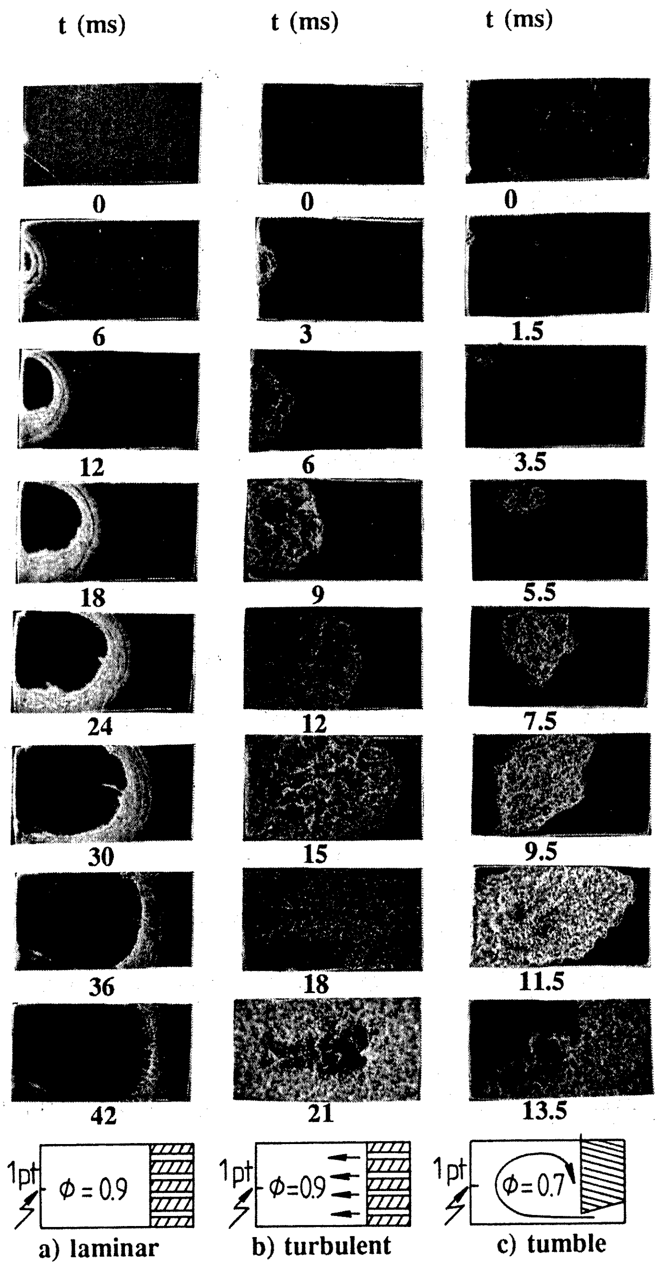
c) flame propagating in a turbulent field with initial mean flow,

have been undertaken to verify whether it is possible to reproduce, in this chamber, flow fields encountered in engines. The flow generator consisting of a plexiglass block of 60 mm x 60 mm section and 50 mm length, on which 25 equidistanced 3 mm diameter channels are drilled, is used for tests a) and b), while, a generator with a 60 mm wide and 3 mm height slit has been used to produce a tumble flow in the test c). The filling pressure of mixture was 1 bar. In the case a) the piston has been pushed slowly and locked. Ignition has occured after several minutes with an initial pressure of about 2.5 bars. In case b) and c) the piston travels the first 45 mm stroke in 50 ms, then slows down on the last 5 mm by hydraulic damper and then locks 90 ms after the beginning of the movement. Ignitions occurs at t = 95 ms.

Propane-air mixtures have been utilized for this study. Equivalence ratio of 0.9 is employed for the first two cases and that of 0.7 for the last case.

Laminar flame

Laminar flame has been studied under static conditions. Fig. 2-a shows the flame aspect evolution during flame propagation. From ignition to t = 18 ms; flame front develops hemispherically, then elongates until 24 ms with smooth flame surface. At about 25 ms, the flame surface decreases by touching the surrounding walls. From 42 ms, slight waving is observed on the flame surface indicating the interaction between flame and acoustic wave. Onset of the phenomenon has been emphasized, in this configuration, due to the presence of the channels of the flow generator.



Mixture: C_3H_8 - Air

Pi = 1 bar (2.5 bars for laminar flame)

Ti = 293 K

Pp = 6 bars rel, Pb = 20 bars rel.

t_v = 61 ms

Fig.2: Influence of the flow field on a cofined premixed flame.

Flame propagating in a distributed turbulence field

As can be seen on figure 2-b, the flame front is perturbed from the ignition to the end of combustion. An elongated hemispherical flame, the surface of which is very perturbed, develops from the very instance of ignition. Flame front does not touch the walls till t=18 ms (uniform black region on the ignitor side). Combustion continues with flame front of which the scale of perturbation seems to become smaller and smaller. Overall combustion duration decreases by more than 50% with regard to laminar case.

Flame propagating in a flow field with mean flow and turbulence

As figure 2-c shows, under tumble and turbulent condition, the flame core, the surface of which is already very perturbed at the ignition, is convected by the mean flow, and it seems to grow up rapidly around the center of tumblig mouvement. The overall combustion duration becomes less than 15 ms even with a mixture of 0.7 equivalence ratio.

As we have seen above, the combustion chamber developed for this study reproduces well phenomena encountered in spark ignited piston engine.

FLAME-FLOW INTERACTION IN A DISTRIBUTED TURBULENCE FIELD

A detailed study has been undertaken first on the case of flame propagation in mixture with turbulence but without initial mean flow, using channel type flow generator (Fig. 3). The level of the turbulence energy at the ignition instant has been ajusted by changing either channel diameter (2.6, 4.2, 6, 8.5 mm) with a fixed ignition delay (5 ms after piston locking : 95 ms after the beginning of the piston movement) or ignition delay with a fixed channel diameter (2.6 mm).

As Fig. 3, obtained without combustion, shows, in this configuration, piston movement generates, at first, a turbulent flow in the chamber, but after the piston locking, mean flow becomes negligible and only turbulent movement, decaying with time, remains. Ignition is initiated in this phase.

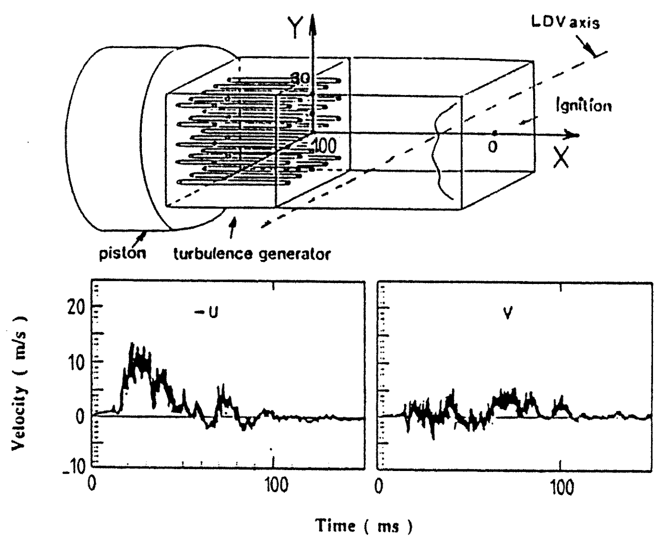


Fig.3: Raw L.D.V. data on an individual experiment without combustion.

Turbulence definition problem

Estimation of the fluctuation intensity that must represent turbulence intensity is a problem in the case considered here, as has been also pointed out in an engine by several authors (7, 8, 9). Figure 4 gives, at the center of the chamber for the turbulence generator with holes diameter of 2.6 mm, a comparative display of turbulence intensity in the case where a simple ensemble averaging is considered and in the cycle-resolved analyse case. When the cutoff time parameter Δt is changed from 1 ms to 5 ms the turbulence intensity increases by a factor of 2 (4 in energy) and it increases by a factor of 3 with ensemble averaging.

The criteria for the selection of the filter then become the basic question. Bracco (5) has adopted the criteria that, in an engine, what is turbulence and what is bulk motion is relative to the process that is influenced by the flow : the combustion in the present study. More specifically, combustion process responds differently to flow fluctuations depending on whether the time and length scales of fluctuations are much larger or of the same order, or much smaller than those of the process. Thus, the characteristic time and length scales of this process becomes the cut off time and length scale to be used in the analysis of the records. So the time and length scales of the flow must be resolved in each cycle and the results of first attempt are presented at the end of this paper.

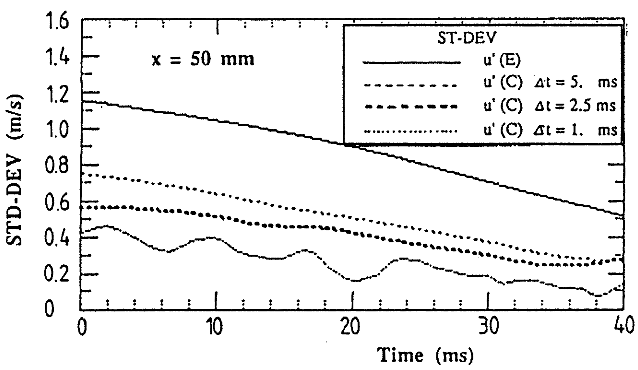


Fig.4: Effect of cutoff frequency on the deduced tubulence intensity

Combustion regime

The level of the initial turbulent energy, determined with a cutoff time of 5 ms, is comprised between 0 to 0.5 m^2/s^2. The analysis on a turbulent Reynolds number - Damköler Number diagram (Fig. 4) shows that the combustion of a propane-air mixture with equivalence ratio of 0.9 will realize in wrinkled flame sheet regime. Furthermore it is seen that the combustion regime realized in this chamber corresponds well to that realized in real engines.

Figure 5 shows flame structure (laser sheet visualization) and pressure history evolutions in function of the ignition delay τ_d. As turbulence intensity decays with time, longer the ignition delay is, lower is the turbulence intensity. It is observed that :

1. the flame structure is effectively of wrinkled flame sheet type,

2. the combustion duration, defined here by the maximum pressure peak, decreases when the turbulence intensity increases : 70 ms for laminar flame and 22 ms for τ_d = 10 ms which correspondes to a turbulence level of 0.5 m^2/s^2 at the ignition instant,

3. with the initial turbulence, the value of the first peak of dP/dt, corresponding to the instant when the flame front surface touches chamber walls, increases (250 bars/s for laminar case and 600 bars/s for τ_d = 10 ms) and the time to attain to this peak decreases (24 ms and 16 ms). It is clear that the turbulence acts on the flame from the beginning of its formation,

4. influence of turbulence on the second peak of dP/dt is more important than that on the first one. It increases from only 170 bars/s for laminar case to about 2500 bar/s for τ_d = 10 ms,

5. the hollow between two peaks, corresponding to the minimum effective flame surface area after the contact of the flame with chamber walls, decreases when turbulence increases and disappeares practically for τ_d = 10 ms. This fact shows that the increase in flame surface due to the turbulence compensates the decrease of it due to its contact with chamber wall.

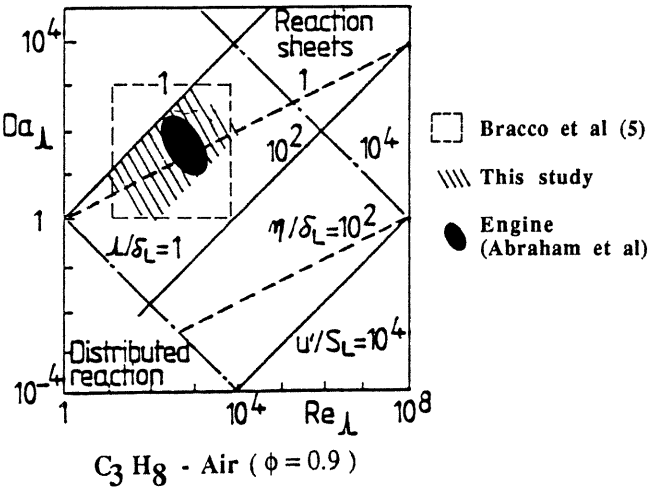


Fig.5: Combustion regime evaluation on a Re - Da diagram.

Influence of the turbulence on the flame

Figure 7 shows longitudinal velocity of the unburned mixture flow, measured by LDV method associated with oil droplets seeding, from the ignition instant at the position X = 70 mm and Y = 0 mm in the chamber. LDV signal disappears at

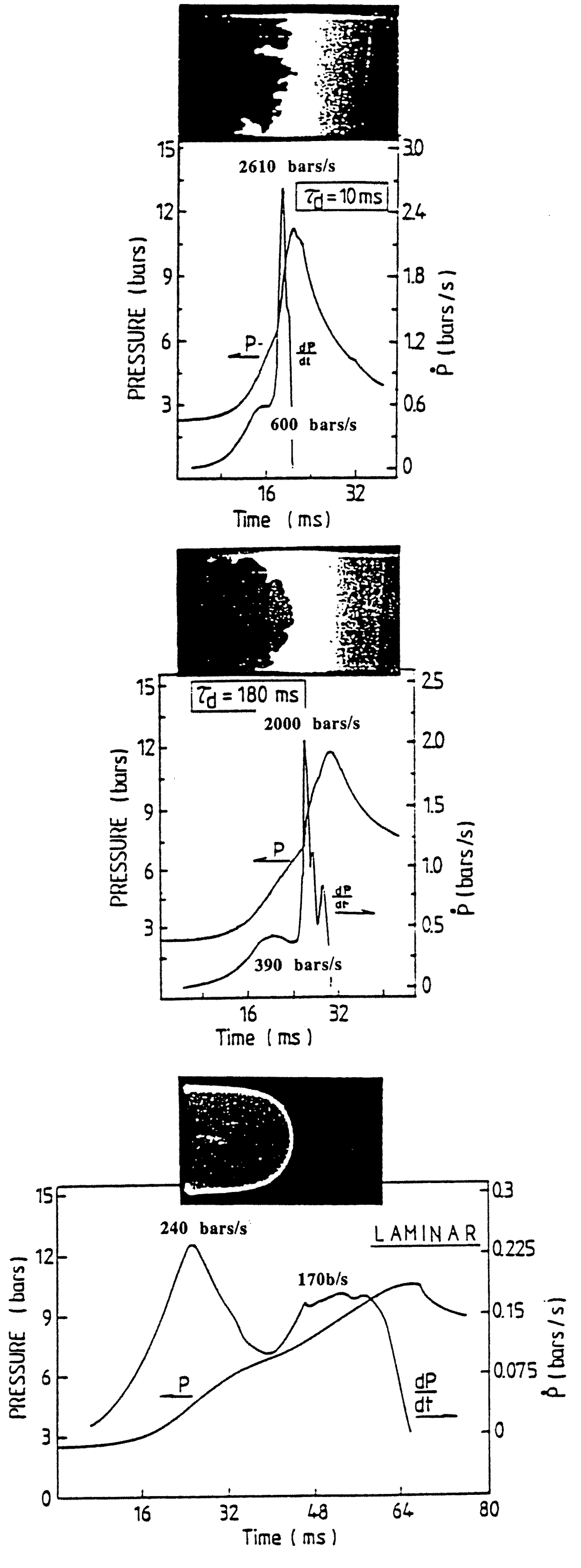


Fig.6: Evolution of pressure, pressure increase rate and flame structure in function of time for different ignition delays after piston locking. (C3H8-Air Ø=0.9)

the moment of flame passage. Up to down are plotted raw LDV data of 25 cycles raw data for each of different hole diameters. It is seen that classically flame arrival time decreases as the turbulence intensity increases. Furthermore cycle fluctuations increase with the initial turbulence intensity.

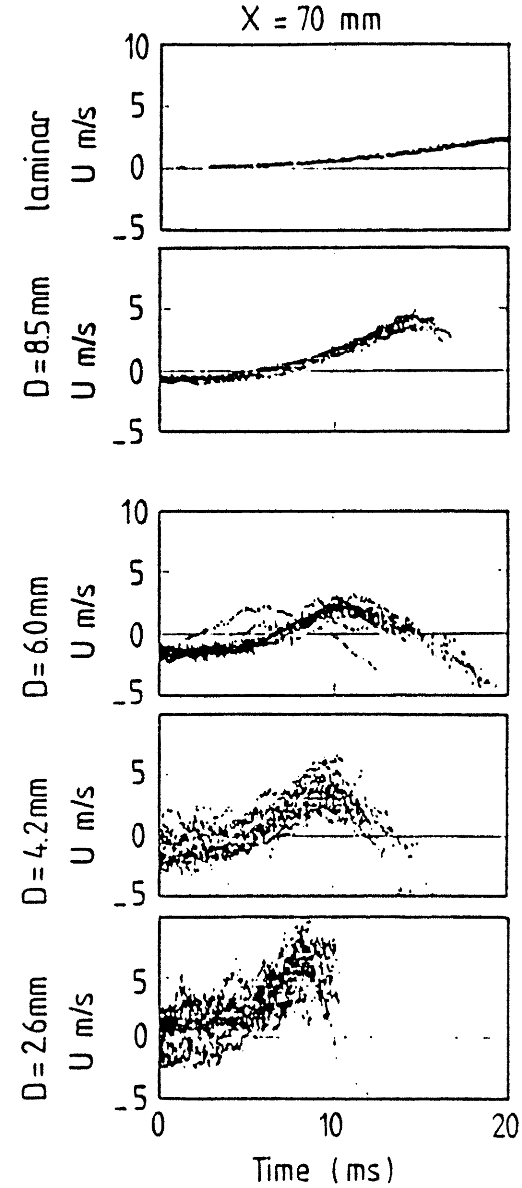


Fig.7: Evolution of unburned gas longitudinal velocity at x = 70 mm and y = 0 mm for different level of initial turbulence intencity (C3H8-Air Ø = 0.9)

Figure 8 shows initial turbulence intensity (8-a), flame speed (8-b) and burning velocity (8-c) in function of the longitudinal position in the chamber and of the channel diameter. Mean unburned gas flow x-direction velocity component U_g is determined as the average of the values measured at a same point in individual cycles, at the signal disappearance instants. The mean flame speed U_f can be deduced from the knowledge of flame arrival time at different positions. Burning velocity S_u is then determined as $S_U = U_F - U_g$. It is observed that :

1. for the most turbulent case, flame speed is roughly of a factor 4 greater than laminar flame speed.

2. turbulent burning velocity S_{UT} is faster than that of the laminar one S_{UL}, from the very beginning in the case of high turbulence intensity and it increases rapidly in the second part of the chamber with flame propagation. This fact is in good agreement with that observed by visualization and by pressure analysis. The increases of S_{UT} by a factor of 12 with regards to the S_{UL} , in the three most turbulent cases, is the consequence of the initial turbulence.

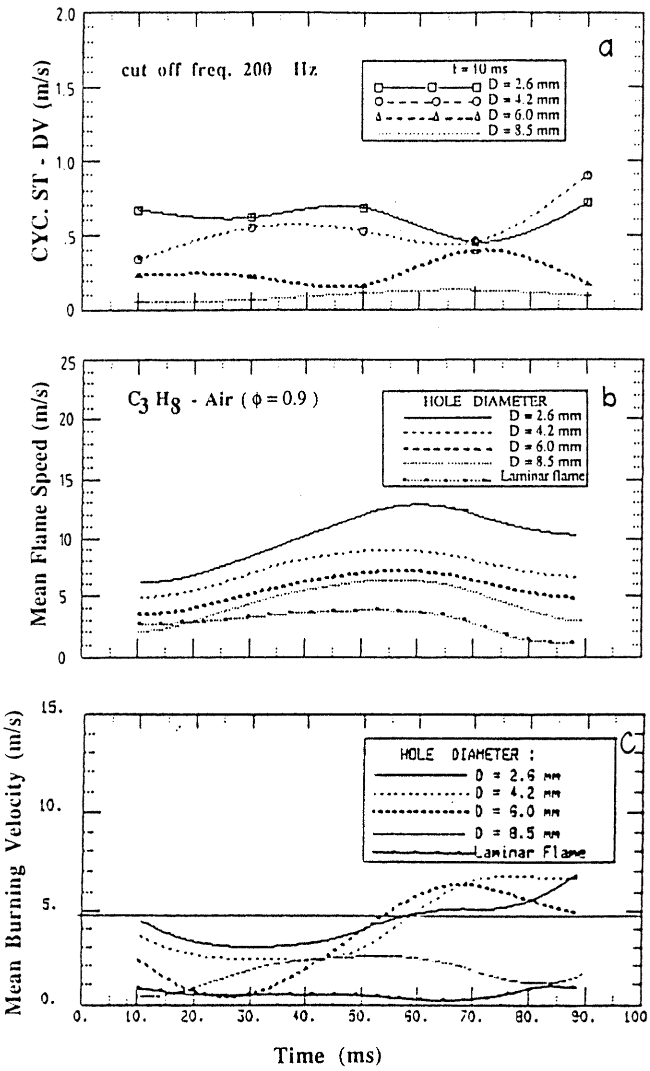


Fig.8: Initial turbulence level (a), mean flame speed (b) and mean burning velocity (c) in function of flame position for different hole diameter

<u>Spatial correlation</u>

Using a two point LDV method, the spatial correlation R(x, x+Δx, τ = 0) has been measured by using ensemble averaged and cycle -resolved technique during flame propagation and compared to the one without combustion (fig. 9). Results with combustion seem to present a periodic component. However spatial scales determined by means of the fitted curves lead to the same values than without

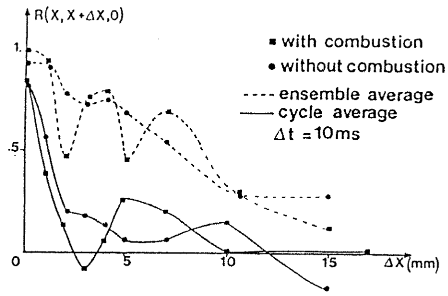


Fig.9: Comparison of the spatial correlation coefficient for two turbilence definitions (cutoff time parameter Δt =1 ms, hole diameter : 2.6 mm, x = 70 mm, y = 0 mm)

combustion. A more detailed analysis is undertaken (10).

CONCLUSION

Our objective to deliver experimental data for model validation has been realized for the case when a flame propagates in an initially homogeneous and isotropic turbulent field without mean flow. The device allows a detailed description of turbulent evolution <u>during flame propagation</u> while in an engine, environment constraints don't permit a full investigation of turbulent caracteristics.As far as the correlation between turbulent energy and turbulent burning velocity is concerned, it is necessary to study more thoroughly the turbulent energy definition in such a transient situation where cycle to cycle variation exists with continuous flame acceleration .

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