Abstract

Minimum ignition energy (MIE) of lean methane–air mixtures is quantitatively measured using a high-power pulse generator which can vary ignition energies of a spark-electrode in the central position of a large fan-stirred cruciform burner. The burner equipped with a pair of counter-rotating fans and perforated plates can be used to generate isotropic turbulence having a very wide range of turbulent intensities \(u'/S_L\) up to 8 m/s with negligible mean velocities. Observations of ignition, flame kernel development, and subsequent flame propagation in the central uniform region of the burner are recorded by a CMOS high-speed camera (5000 frames/s), showing distributed-like flames of very dispersive and fragmental structures with filiform edges for the first time. A complete MIE data set of lean methane–air mixtures at the equivalence ratio \(\phi = 0.6\) as a function of \(u'/S_L\) is obtained, where \(S_L\) is the laminar burning velocity. It is found that there is a transition on values of MIE due to different modes of combustion. Before the transition, MIE only increases gradually with \(u'/S_L\). Across the transition when \(u'/S_L > 24\) corresponding to the commonly defined turbulent Karlovitz number \(Ka = (u'/S_L)^2(Re_T)^{-0.5} > 8\), MIE increases abruptly, where \(Re_T\) is the turbulent Reynolds number based on the integral length scale of turbulence. This transitional value of \(Ka\) is much greater than the Klimov–Williams criterion \((Ka = 1)\). Since values of MIE under different levels of turbulence should be relevant to the size of the reaction zone at least in the beginning of turbulent combustion, MIE \(\sim \delta^3\) based on an order-of-magnitude criterion where \(\delta\) is the reaction zone thickness. It is thus concluded that this new experimental finding proves the existence of both thin and broken reaction zones regimes proposed by Peters for a new regime diagram of premixed turbulent combustion.

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Keywords: Minimum ignition energy; Lean premixed turbulent methane combustion; Transition; Flamelet and distributed regimes; Thin and broken reaction zones

1. Introduction

Turbulent combustion is of fundamental and practical importance. For example, lean premixed turbulent combustion in conjunction with increased compression ratios has great potential for increased fuel economy and reduced NOx...
emissions in spark–ignition engines [1,2] and gas turbines [3]. Though much has been learned through experimentation, modeling, and computation [4], this important and extensively studied subject still has many unresolved fundamental issues, such as the existence and properties of turbulent premixed combustion in the so-called distributed regime.

It has long been anticipated that there are different modes of turbulent combustion which influence all key features of premixed turbulent flames [4–6]. Focus is on “flamelet” (Ka < 1) and “distributed” (Ka > 1) regimes, traditionally separated by a Ka = 1 line, of which Ka is the turbulent Karlovitz number defined as Ka = (u′/S_L)(Re_T)^{-0.5} and Re_T = u′L_T/v, where u′ and L_T, v, and S_L are characteristic turbulent intensity and integral length scale, the kinematic viscosity, and the laminar burning velocity, respectively. Based on a scaling description of the size of the reaction zone, Peters [4] argued that flame broadening by turbulence in the thin reaction zone regime occurs only in the preheated zone without influencing the reaction rate and this thin reaction zone regime before changing to the broken reaction zone regime could sustain for Ka ≫ 1. However, how to prove experimentally the existence of the distributed-combustion regime or the thin/broken reaction zones is still an open issue. This motivates the present work that aims to investigate experimentally different combustion modes based on measurements of minimum ignition energy (MIE) in a large fan-stirred cruciform burner.

MIE is an extremely important property for safety standards as well as for the fundamental understanding of the ignition process of combustible mixtures [7–9]. Furthermore, accurate MIE data are crucial for optimization of ignition systems especially when ignition of lean premixed mixtures under turbulent combustion is considered [10,11]. It will be shown that measurements of MIE and observations of the flame kernel formation and its subsequent flame propagation in intense isotropic turbulence without mean velocities can provide an excellent opportunity to scrutinize different combustion modes and thus evaluate the new regime diagram of premixed turbulent combustion proposed by Peters [4]. This is the goal of this paper.

Concerning the ignition process, considerable information is available on MIE of quiescent and flow mixtures. For instances, Kono and his co-workers [11–13] investigated the effects of spark duration, spark path behavior, electrode size and material, discharge type and more on the minimum ignition energy. Ziegler et al. [14] have carried out experimental studies on spark–ignition of lean methane–air mixtures using high pressure glow and arc discharges. These results have demonstrated that the ignition energy of spark discharge depends upon gas density and species, electrode material, size and geometry, current, gap width, and type of discharge. For simplicity, the present work uses a commercial high-power pulse generator (Velonex Model 360) along with a V-1918 plug-output and a V-2428 pulse transformer which can control and vary discharge energies of a stainless-steel spark-electrode with very sharp needle ends on anode and cathode separated by a fixed gap width of 2.6 mm (close to the methane quenching distance). This spark-electrode is placed in the central position of a large fan-stirred cruciform burner which can produce intense isotropic turbulence without mean velocities. Hence, quantitative measurements of MIE for lean premixed methane–air combustion over a very wide range of u′/S_L and/or Ka covering different turbulent combustion modes can be achieved.

Using the cruciform burner (Fig. 1), the effect of turbulent straining on MIE is investigated for the first time in intense isotropic turbulence with negligible mean velocities. This novel experimental apparatus has been used to obtain qualitative understanding and quantitative analysis of various aspects of turbulent flame propagation, stretching, and global quenching with and/or without consideration of radiation heat losses [15–19]. The following section reviews experimental methods used in this work, including new modifications of the burner, the ignition system, the determination of MIE using high voltage and current probes, the imaging acquisition system, and the experimental procedure. A description on dynamics of turbulent flames from flame kernel formation to flame propagation is included and is followed by a complete set of measured MIE data. Both are then applied to investigate fundamental characteristics of lean methane–air mixtures in different regimes of turbulent combustion. Finally, a transition on MIE due to different modes of combustion is presented, the conclusion is offered, and the area for future studies is identified.

2. Experimental

Figure 1 shows schematic diagrams of the set-up, including a symmetric cruciform burner, a high-power spark–ignition system, and a high-speed imaging acquisition system. The burner is equipped with a pair of counter-rotating fans driven by two 10-HP electric motors and synchronized to the same speed. The maximum fan frequency is 172 Hz when frequency converters and a water cooling system to both motor shafts are applied, at which the corresponding u′ up to 8 m/s with negligible mean velocities can be achieved. A large volume up to 15 × 15 × 15 cm^3 of intense near-isotropic turbulence, having energy spectra with

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−5/3 slopes, can be generated in the central uniform region of the burner [15,20] where the spark-electrode is located. Due to the symmetry of pairs of counter-rotating fans and perforated plates, the presence of the thin electrodes has little influence on turbulent properties within experimental uncertainties, as confirmed by velocity measurements using high-speed particle image velocimetry.

The ignition system consists of a Velonex high-power pulse generator/transformer, the spark-electrode, high voltage and current probes, various resistances, and a Tektronix oscilloscope, which can control and vary discharge energies of the spark-electrode (Fig. 1). The present electrodes made of stainless steel have sharp ends (0.1 mm) with a fixed spark gap of 2.6 mm roughly equal to the methane quenching distance. This ignition system is capable of discharging high voltages up to 20 kV with selectable spark durations varying from 0.05 µs to 3 ms. The high-power pulse generator offers four discharge modes, including one shot, internal pulse repetition frequency, external drive, and external trigger. In this study, only one shot mode with fixed spark duration of 100 µs is applied, and various resistances from 0 to 50 kΩ are used to alter the currents and thus discharge energies of the spark-electrode. It is very difficult to measure the ignition energy quantitatively and accurately. After many tests, we found that direct measurements of voltage and current on the anode and the cathode of the spark-electrode using a Tektronix high-voltage probe and a Pearson current monitor, respectively, can give us repeated and accurate data of ignition energies.

Figure 2 presents typical recording traces of voltage and current generated by one shot of the spark-electrode. As can be seen in Fig. 2, the initial growth of the voltage trace is due to the onset of high voltage output to the circuit. Then, the discharge occurs abruptly producing the concurrent voltage \( V_d \) and current \( I_d \) waveforms, at which the ignition energy of such a spark discharge is calculated from the time \( t_1 \) to \( t_2 \), as indicated on Fig. 2. The voltage after \( t_2 \) is just the
residual voltage. It should be noted that these current and voltage traces obtained in air are essentially the same as those obtained in lean methane–air mixtures. In this study, the discharge energies across the electrodes are varied from 0 to about 71 mJ. The advantages of using such a commercial ignition system are to avoid possible uncertainties that may occur in different homemade ignition devices, and most importantly, the experiment can be confirmed and repeated by the same Velonex high-power pulse generator. Moreover, our goal is to find experimental evidences for the existence of distributed or broken reaction zones regime, so that the ignition conditions are kept as simple as possible and thus only one shot discharge mode is considered.

Using a CMOS high-speed camera capable of operating at 5000 frames/s with 512 × 512 pixels, instantaneous images of flame kernel development and its subsequent flame propagation can be obtained. Figure 1 also presents typical sequent images of ignition and flame propagation for methane–air mixtures at the equivalence ratio \( \phi = 0.7 \) in both laminar \( (f = 0 \text{ Hz}; u'/S_L = 0) \) and turbulent \( (f = 7 \text{ Hz}; u'/S_L \approx 2) \) cases. As can be seen, turbulent flames propagate much faster than laminar flames, more than 1.5 times higher on the average flame outward propagation speed (Fig. 1).

Before a run, methane–air mixtures at \( \phi = 0.6 \) are well mixed in a new separate mixing chamber and then injected into the evacuated cruciform burner to 1 atm. A run begins by igniting these mixtures either in a quiescent condition without fan-stirring or under various turbulent conditions with different fan-stirred intensities. More than four hundreds ignition experiments are carried out to obtain the ignition probability at given values of \( u'/S_L \). Specifically, there are three different ignition probabilities, respectively, 100%, 50%, and 0% of which we are interested, at a given value of \( u'/S_L \). For a point of interest at a fixed \( u'/S_L \), data of 100% and 0% ignitability can be clearly determined, and thus these experiments are first carried out in order to identify a smaller range for the same mixtures of 50% ignitability. Within this range, the spark discharges are repeated at least 30 times to determine the point of 50% ignitability, and this point is the MIE using the same definition as previous studies [10–14].

3. Results and discussion

3.1. Flame initiation, development, and propagation in different regimes

A key facet of turbulent premixed combustion is the mode of combustion, also known as Borghi diagram or regime diagram, which may affect all important properties of premixed flames [4]. Ignition is the beginning of combustion. By comparing time sequent images of ignition, flame kernel development, and its subsequent flame propagation in different regimes of premixed turbulent combustion, a basic understanding on differences or similarities of combustion characteristics in different regimes may be obtained, as shown on Fig. 3. These instantaneous time sequent photographs of \( \text{CH}_4 \)-air mixtures at \( \phi = 0.6 \) reveal three different regimes: (a) laminar \( (Ka = 0) \), (b) turbulent-flamelet \( (Ka \approx 1) \), and (c) distributed-like \( (Ka \approx 9) \), respectively.

In a quiescent condition (Fig. 3a), the initiation and subsequent propagation of flame fronts are symmetric with respect to the electrodes having more or less spherical geometries. It is found that for the turbulent-flamelet case (Fig. 3b), the initial stages of flame kernel development up to 10 ms are essentially the same as those in quiescent conditions. Similar results are also reported by Ishii et al. [11]. The time required for a laminar flame kernel transforming to a turbulent flame kernel is shortened as values of \( u'/S_L \) and/or \( Ka \) increase. Non-spherical geometries of these turbulent-flamelet fronts are observed in Fig. 3b due to turbulence and buoyancy effects. These flamelet fronts upon propagation are wrinkled by isotropic turbulence \( (u'/S_L \approx 5; \text{Fig. 3b}) \), where cellular structures can be observed because of lean methane-air flames having a Lewis number less than unity.

It is worthy noting that this turbulent-flamelet regime can sustain at much higher level of turbulence \( (Ka \approx 8) \) than the Klimov–Williams criterion \( (Ka = 1) \) which is commonly used to separate flamelet and distributed regimes in the regime diagram. By comparing these sequent images on Figs. 3b and a, turbulent-flamelet fronts propagate radially at an average speed much faster than laminar flames can achieve. Due to the space limit, we will present the effect of turbulent straining on flame propagation speeds elsewhere.

Under very intense turbulence conditions where \( u'/S_L \approx 26 \) and \( Ka \approx 9 \) (Fig. 3c), much
higher ignition energies must be required to ignite the same methane–air mixtures at $\phi = 0.6$, the formation of turbulent flame kernel is fast within 1 ms after the spark discharge, and the kernel may occur at random positions slightly outside the electrodes. As can be seen from images at 4 ms up to 10 ms in Fig. 3c, the distributed-like flame kernel looks irregular and dispersive. This differs drastically with flame kernels at 10 ms in Figs. 3a and b that are regular and nearly spherical. Unlike turbulent-flamelet fronts which have clear-cut boundaries from its initiation to subsequent propagation, distributed flame fronts have disrupted and fragmental structures with filiform edges. Pockets and islands of reactants are observed in the later stage of distributed flame propagation (after 24 ms), appearing as small dark regions randomly scattered inside the largely intertwined and dispersed flame body (Fig. 3c). It should be noted that burning of these turbulent-distributed flames can go on much beyond 50 ms and even sustain longer than that of turbulent-flamelet flames at higher values of $u'/S_L$, indicating that the burning rate of fresh gases does not be faster in the distributed combustion regime than in the turbulent-flamelet combustion regime. This is because collisions and annihilations of these turbulent filiform flame brushes inhibit flame burning rates, as can be seen from motion pictures taken by the high-speed camera.

### 3.2. Minimum ignition energy in quiescent condition

It is logical to first measure MIE data in the quiescent condition without fan-stirring, so that comparison with previous laminar MIE data can be made. Figure 4 shows MIE data, as indicated by black circles (50% ignitability), as a function of $\phi$ for lean methane–air mixtures ($u'/S_L = 0$), indicating that the burning rate of fresh gases does not be faster in the distributed combustion regime than in the turbulent-flamelet combustion regime. This is because collisions and annihilations of these turbulent filiform flame brushes inhibit flame burning rates, as can be seen from motion pictures taken by the high-speed camera.

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be very close to previous data reported by Ziegler et al. [14] and Lewis and von Elbe [8] with no more than 7% difference using the same methane–air mixtures and spark discharges. As $\phi$ decreases toward the lean flammability limit, values of MIE increase.

### 3.3. Minimum ignition energy in flamelet and distributed regimes

Figure 5 shows turbulent MIE (black circles) as a function of $u'/S_L$ for lean methane–air mixtures at $\phi = 0.6$. Again, the regions of white circles (100% ignitability) and cross symbols (0% ignitability) are marked as ignitable and non-ignitable regions on Fig. 5. Also plotted is the value of MIE at quiescent condition ($u'/S_L = 0$) marked as an open triangular symbol on the ordinate. It is found that a transition on MIE of lean premixed methane–air combustion exists due to different modes of combustion. Below the transition, values of MIE increase gradually only from 2.14 to 10.15 mJ when values of $u'/S_L$ increase from 0 to as high as 23 for which MIE $\sim (u'/S_L)^{0.7}$. The transition occurs at $Ka > 8$ when relatively large values of $u'/S_L$ above 24 are achieved. Across the transition, the slope of MIE as a function of $u'/S_L$ changes abruptly from 0.7 to 7, corresponding to different regimes varying from turbulent-flamelet or possibly thin reaction zone regimes (small to large values of $Ka < 8$) to distributed or possibly broken reaction zone regimes (very large values of $Ka > 8$). The maximum ignition energy used in this study is 70.8 mJ that cannot ignite ultra-lean methane–air mixtures at $\phi = 0.6$ when $u'/S_L > 29$ corresponding to $Ka > 10.3$, as indicated by cross symbols on the right end of Fig. 5.

The transitional value of $Ka (\sim 10)$ is found to be much greater than the Klimov–Williams criterion ($Ka = 1$) that separates both flamelet and distributed regimes traditionally. Since values of MIE under different levels of turbulence should be relevant to the size of the reaction zone at least in the beginning of turbulent combustion, the
present finding proves the existence of a thin reaction zone regime proposed by Peters [4]. He argued on a scaling base that in the thin reaction zone regime, turbulence can only influence the preheated zone and the reaction zone remains thin even for $Ka \gg 1$. However, the difference is on the upper limit of the thin reaction zone regime: $Ka \sim 10$ (the present work at $\phi = 0.6$) vs. $Ka \sim 100$ (Peters [4]). We anticipate that the upper limit of $Ka$ should be increased if heat losses due to the effect of the cold boundary could be eliminated or when values of $\phi$ move toward $\phi = 1$ (stoichiometry).

Damköhler [21] may be the first to describe a disrupted flame structure by intense turbulence which should be very similar to that shown on Fig. 3c. Damköhler argued that strong turbulence influenced turbulent burning rates mainly through the increase of diffusive transport inside the distributed (broadened) flame front without distinguishing the difference in thickness between the general flame front and its reaction zone (see [22]). Furthermore, Sánchez, Liñán and co-workers [23] stated that MIE is proportional to $\delta^3$ based on an order-of-magnitude criterion, where $\delta$ is the reaction zone thickness. Thus, for flamelet and thin reaction zone regimes, $\delta_T$ is only slightly larger than $\delta_L$, so that values of $MIE$ only increase slowly with increasing values of $u'/S_L$ up to $24 (Ka \approx 8)$, where the subscripts represent turbulent and laminar cases. For distributed-like or broken reaction zone regimes, values of $MIE$ increase drastically because $\delta_T$ is much greater than $\delta_L$.

The present experiment shows that not only small-scale turbulence can reside in the broadened flame front, but also when turbulence is sufficiently intense small-scale nibbling can occur to disrupt the flame structure (the reaction zone) through turbulent stretching, collisions and annihilations of these distributed filiform flame brushes. It is believed that this small-scale nibbling mode of turbulent combustion is one of the reasons for causing the strong bending effect on turbulent burning rates ($S_T/S_L$) in premixed turbulent combustion for which further increasing $u'/S_L$ to a sufficient high level of turbulence can actually result in a decrease of $S_T/S_L$ as previously reported [15].

4. Conclusions

A complete data set of MIE for lean premixed turbulent methane combustion at $\phi = 0.6$ at high Reynolds number covering different regimes of turbulent combustion is presented, which reveals the following.

1. There is a transition on values of MIE due to different modes of combustion. Before the transition, values of MIE only increase gradually with $u'/S_L$ at which $MIE \sim (u'/S_L)^{0.7}$. Across the transition when $u'/S_L > 24$ and/or $Ka > 8$, values of MIE increase abruptly at which $MIE \sim (u'/S_L)^{3/2}$.

2. The transition occurs at $Ka \sim 10$ which is much greater than the Klimov–Williams criterion ($Ka = 1$). Moreover, values of MIE are relevant to the size of the reaction zone for which $MIE \sim \delta^3$. This is why a drastic increase of MIE is found in distributed-like or broken reaction zone regimes. Thus, this finding proves for the first time the existence of both thin and broken reaction zones regimes proposed by Peters for a new regime diagram of premixed turbulent combustion.

3. Unlike turbulent-flamelet fronts with clear-cut boundaries from the initiation to subsequent propagation, turbulent-distributed flame fronts are very dispersive and fragmental with filiform edges. Pockets and islands are observed.

4. Burning of distributed-like flames in the broken reaction zone regime can last longer than that of turbulent-flamelet flames for a given mixture in the burner due to collisions and annihilations of these turbulent disrupted flames with filiform edges that may inhibit flame burning rates.

In the near future, we shall test whether the transition of MIE which occurs at $Ka \sim 10$ for $\phi = 0.6$ would remain the same or require even greater values of $Ka$ when higher values of $\phi$ ranging from 0.7 to 1 are considered. Moreover, flame propagation speeds in different regimes, from laminar to flamelet and from thin reaction zone to broken reaction zone, will be analyzed and reported elsewhere.

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