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Experimental Investigation on Spray, Evaporation and Combustion Characteristics of Ethanol-Diesel, Water-Emulsified Diesel and Neat Diesel Fuels

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HIGHLIGHTS

- Micro-explosion of multi-component diesel sprays was investigated.
- Micro-explosion of ethanol and water expanded spray volume significantly in 900K evaporating spray.
- Micro-explosion of W10 occurred later and led to faster downstream combustion rate than E10.
- Blending ethanol and water in diesel reduced soot emissions by 21% and 39% respectively.

Abstract

This paper explored the spray and combustion characteristics of ethanol-diesel (E10), water-emulsified diesel (W10) and neat diesel (D100), especially micro-explosion of E10 and W10. The experiments were conducted in a constant volume combustion chamber under cold (383K, 0% O₂), evaporating (900K, 0% O₂) and combustion (900K, 21% O₂) conditions. Results showed that the spray expansion capacities of E10 and W10 under cold condition were much weaker than that of D100 due to the larger viscosity of emulsified diesels. Under evaporating condition, the spray volume of E10, W10 and D100 increased by 59%, 34% and 21% respectively comparing with cold spray volume. The higher increasing rates of E10 and W10 were mainly due to the micro-explosion effects of ethanol and water contents. Under combustion condition, the integrated natural flame luminosity (INFL) demonstrated that the ethanol content could accelerate the oxidation of soot, while the water content could prohibit soot generation. Therefore, both ethanol- and water-emulsified diesels could inhibit the soot emission, causing lower final residual soot emission of E10 and W10 than that of D100 by 21% and 39% respectively. Moreover, the flame lift-off length (LOL) and flame spread velocity showed that the effects of micro-explosion of E10 and W10 are different. The micro-explosion of ethanol occurred earlier, which enhanced the reaction rate in upstream flame and reduced the LOL. However, the micro-explosion of W10 occurred later, which enhanced the combustion rate in downstream flame.

Keywords: Ethanol-diesel; Water-emulsified diesel; Cold spray; Evaporating spray; Spray combustion; Visualization.

1. Introduction

The research on alternative fuels in internal combustion engines (ICE) has become one of the hot spots due to

1 the increasing serious environmental and energy problems [1, 2]. Using alternative fuels can not only partially
2 replace the traditional fossil fuels and alleviate the dependence on them [3, 4], but also reduce the pollutant
3 emissions from ICEs [5, 6], making it one of the most promising methods to save energy and reduce emissions.
4 Multi-component alternative fuel with different boiling point components could lead to a unique micro-explosion
5 phenomenon in the spray breakup and evaporation processes [7]. Specifically, the component with lower boiling
6 point could be superheated and experience a drastic evaporation or explosion during the fuel spray and combustion
7 [8, 9].

8 Ethanol gains the attention of many researchers because it is a readily available alternative fuel and has many
9 merits such as renewability, low-pollution and nontoxicity [10-12]. However, it is difficult to use ethanol as a neat
10 fuel directly in diesel engines because of its low cetane number and high latent heat [13, 14]. So far, there are three
11 methods to utilize ethanol fuel in diesel engines: (1) ethanol fumigation or ethanol injection into the intake port
12 [15-17], (2) dual injection of ethanol and diesel into the cylinder, and (3) injection of ethanol diesel emulsified
13 blends into the cylinder [18, 19]. Methods 1 and 2 can use higher ethanol ratio while method 3 can only achieve an
14 ethanol ratio lower than 30% due to the immiscibility. The engine performance has been intensively investigated
15 and the results showed that all the three ethanol utilization methods could reduce soot emissions [20-23]. However,
16 the results on brake specific fuel consumption (BSFC), NO_x, CO and HC were inconsistent among publications
17 [24].

18 To understand the underlying mechanisms, the effect of ethanol on the spray, atomization and combustion
19 processes inside the engine should be investigated [25]. However, comparing to the existed many engine studies,
20 research on the fundamental influence of ethanol on the spray, evaporation and combustion of ethanol-diesel blends
21 is scarce and incomplete. Particularly, the effect of micro-explosion on spray and combustion processes of
22 ethanol-diesel blends has rarely been explored deeply.

23 Diesel-water blend has become an active research topic since Sheng et al.'s study in 1995 [26]. It was reported
24 as a potential alternative fuel to reduce harmful emissions and improve combustion efficiency simultaneously [27,
25 28]. Diesel-water blends could be achieved by three approaches [29], including intake port water injection (WI),
26 direct in-cylinder WI and water-emulsified diesel [30]. Water-emulsified diesel is a kind of emulsion by diesel and
27 water [31, 32]. Previous studies [33-38] showed that diesel-water blends could effectively reduce soot and NO_x
28 emissions simultaneously in diesel engines. There are three main underlying reasons. Firstly, water has large
29 specific heat and high latent heat of vaporization, which can reduce the peak combustion temperature and thus
30 reduce the generation of soot and NO_x [39]. Secondly, the water-gas reactions can accelerate the oxidation of soot
31 [40]. Thirdly, the micro-explosion of water could enhance fuel droplet break-up and fuel-air mixing, and
32 consequently decrease the local equivalence ratio and reduce soot generation [41].

33 Comparing with intake port WI and direct in-cylinder WI, water-emulsified diesel can be used in the
34 conventional diesel engines directly without major modifications. Iwai et al. [42] experimentally investigated the

1 emission performance of neat diesel, intake port WI and emulsified diesels with 15% and 30% water. The results
2 showed that NO_x and soot emissions of water emulsified diesels were much lower than that of neat diesel.
3 Comparing with intake port WI, emulsified diesel with the same proportion of water also had better fuel economy
4 and emission performance. Adopting numerical and experimental study, Samec et al. [43] found a significant
5 emission reduction with no increase in BSFC by using water emulsified diesel. Chen [44] reported that the soot
6 emission of water emulsified diesel was nearly 35% less than that of neat diesel.

7 However, there were many inconsistent results on the BSFC, brake thermal efficiency (BTE), CO and HC
8 emissions regarding the use of water-emulsified diesel. Some researchers [45, 46] reported clear decreasing trends
9 in BSFC with the increase of water concentration in water-emulsified diesel, while some [30, 47-49] showed that
10 BSFC increased significantly by using water-emulsified diesel. Some researchers [50-57] reported that CO and HC
11 emissions increased by water-emulsified diesel comparing to neat diesel due to the lower combustion temperature
12 [50, 51] and more OH radicals dissociated from water [52], while others showed that CO and HC emissions
13 decreased slightly or nearly unchanged by water-emulsified diesel [36, 58, 59] because micro-explosion improved
14 the fuel combustion process [36, 59]. However, the existence of micro-explosion in spray is still in argument [60,
15 61]. The spatial scale of engine spray droplets is very small (10-30 μ m) while the spatial scale of experimental
16 droplets for micro-explosion is much larger (larger than 200 μ m). In addition, so far, no micro-explosion in spray
17 has been directly observed.

18 All these inconsistency and arguments are caused by the complicated combustion behaviors of emulsified
19 diesel fuels, including micro-explosion (or puffing), physical and chemical path of fuel emulsion, which need a
20 deeper study. To prove the existence of micro-explosion (or puffing) and to explore the fundamental effect of
21 micro-explosion (or puffing) on spray combustion of emulsified diesel fuels, a joint study on single droplet
22 micro-explosion and spray combustion should be conducted. Our team has begun this joint research [62-65] and
23 this paper is one study in our series research program.

24 As reviewed above, both ethanol and water additions have promoting effects on the spray, combustion and
25 emissions processes of diesel engines. However, the suspended particle size and physical characteristics of
26 dispersed phase in these two emulsified diesels are very different and thus make the onset and strength of
27 micro-explosion (or puffing) vary significantly [66-69]. Moreover, direct evidence of micro-explosion (or puffing)
28 in the fuel spray under real engine conditions has rarely been observed in experiments and there is still much
29 controversy in this field. Therefore, the effect of micro-explosion (or puffing) on spray and combustion should be
30 analyzed by comprehensive spray and combustion measurements. To better understand the underlying mechanisms
31 and thus facilitate their utilization, in the present study, the spray and combustion characteristics of ethanol-diesel,
32 water-emulsified diesel and neat diesel are optically investigated in a constant volume combustion chamber.
33 Particularly, the possibility of micro-explosion (or puffing) in emulsified diesels with different boiling point
34 additives is inferred and compared. The effects of micro-explosion on spray and combustion processes with equal

1 mass ratio of ethanol and water addition in emulsified diesel are investigated under cold, evaporating and
 2 combustion conditions. The reported work sheds light on possible macroscopic effects of micro-explosion in spray,
 3 combustion and emission. The effect of micro-explosion in multi-component fuels on the spray, combustion and
 4 emission characteristics in this paper improves our understanding of underlying mechanisms on why emulsified
 5 diesels improve the spray atomization and reduce the soot emission. The reported work also can to some extent fill
 6 the gap on the inconsistent results on the BSFC, brake thermal efficiency (BTE), CO and HC emissions regarding
 7 the use of water-emulsified diesel. All these can guide the application of ethanol-diesel and water-emulsified diesel
 8 fuels in diesel engines in practice.

9 **2. Preparation of Ethanol-Diesel and Water-Emulsified Diesel**

10 Ethanol and water are immiscible with diesel due to their polar molecular structures. Therefore, suitable
 11 emulsifier is needed to prepare stable ethanol-diesel and water-diesel emulsions [60, 70]. In this paper, 0# diesel,
 12 99.8% purity ethanol, distilled water and two emulsifiers, Span-80 and Op-10, were chosen to prepare the
 13 emulsified diesels. For better comparison on the effects of ethanol and water additions, same mass ratios of ethanol
 14 and water additions were adopted. Hereafter, E10 means the ethanol-diesel emulsion with 10% ethanol by mass,
 15 W10 means the water-emulsified diesel with 10% water by mass and D100 means neat diesel.

16 The Hydrophilic Lypophilic Balance (HLB) value is the guiding index of an emulsifying solution. Higher
 17 HLB value indicates that the emulsifier is easier to dissolve in water, while lower HLB value means the emulsifier
 18 is easier to dissolve in diesel. The stability of various compositions was tested and Table 1 shows parameters of the
 19 stable emulsifying solutions for ethanol-diesel and water-emulsified diesel. The JP300G ultrasonic emulsifier was
 20 used to prepare the E10 and W10. The solutions in Table 1 can be kept over 15 days without delaminating, which
 21 meet the requirements in this study. The dynamic viscosity measured by a NDJ-8s rotational viscometer is also
 22 shown in Table 1. E10 and W10 have greater viscosities because the viscosity of emulsifier is much higher than
 23 neat diesel.

24 **Table 1.** Parameters of E10, W10 and D100.

Fuel	E10	W10	D100
Diesel (g)	1800	1800	2000
Ethanol (g)	200	0	0
Water (g)	0	200	0
Span-80 (g)	56	57.5	0
Op-10 (g)	1.8	4.75	0
HLB	4.6	5.0	0
Density (kg/m ³ at 20°C)	812.5	832.0	815.0
Dynamic viscosity (mPa·s at 20°C)	6.16	6.8	2.26
Lower heating value (MJ/kg)	40.0	37.4	42.8

25

3. Experimental setup and procedures

3.1. Constant volume combustion chamber system

Fig. 1 shows the schematic diagram of the pre-burning heated Constant Volume Combustion Chamber (CVCC) system. The CVCC system consists of an optically accessible chamber, an ambient gas supply device, an ignition system, a combustion pressure measurement system, a high-speed imaging system, and a high-pressure injection system. The chamber is a cube with edge length of 136 mm. The chamber body is sealed by fluorine rubber rings and gland to achieve the high pressure and temperature conditions (e.g., 15 MPa and 1500 K). During the experiment, the CVCC body was kept at 383K by heating sheets to prevent water vapor condensing on quartz windows. The high-speed imaging system consists of a schlieren optical component, a high-speed digital charge-coupled device (CCD) camera and a data acquisition module. Two 50 mm thickness JGS3 far-infrared optical quartz glasses are installed in the light pathway. The parallel light generated by a single collimated source is distorted by the uneven density gradients in the spray zone, forming bright and shade textures in the schlieren images [71]. Therefore, the instantaneous density distribution of the spray field can be captured by the images, from which the spray and combustion characteristics are measured. The high-speed camera used is the Motion Pro Y4-S1 and the lens is Tokina 100mm (f/2.8). In the present experiments, the image resolution is 640×280 pixels and the camera speed is 20000 fps. More details about the apparatus and test approach can be found in refs. [63, 72, 73].

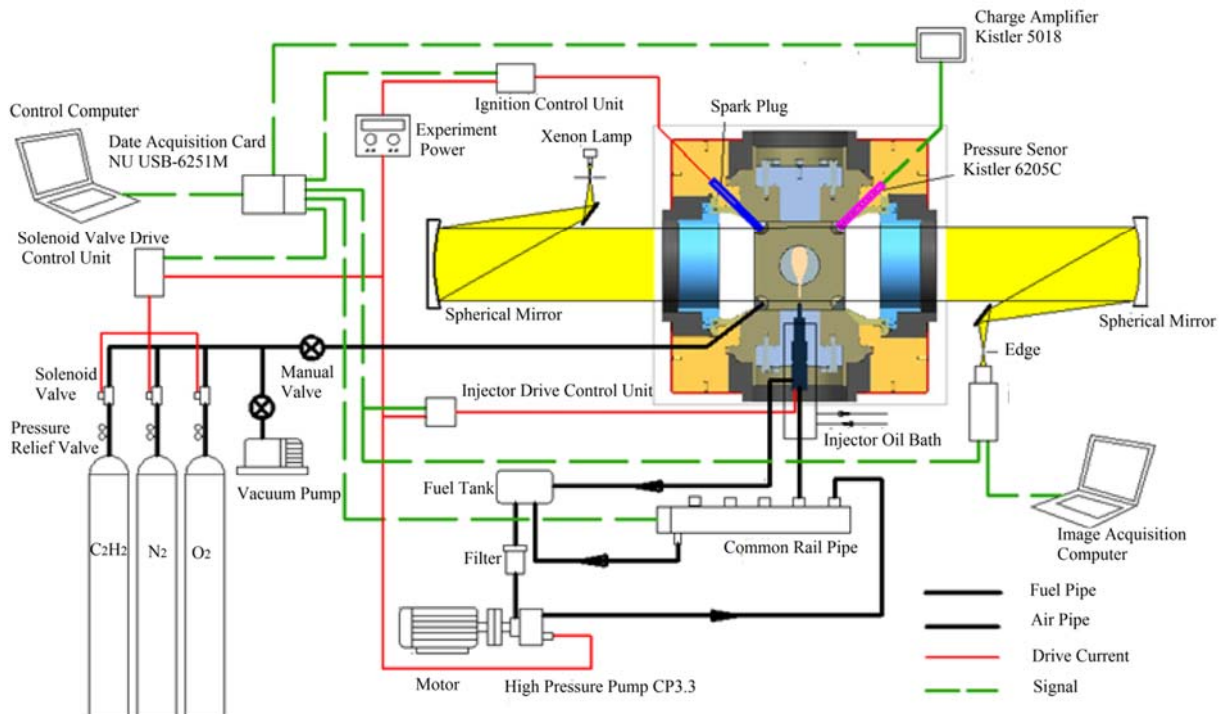


Fig. 1. Schematic of the CVCC system.

1 3.2. Experimental procedures

2 Table 2 gives the experimental conditions investigated in the present study. The experiments were carried out
3 in cold, evaporating and combustion conditions using E10, W10 and D100 fuels. In the cold spray condition, N₂
4 was used to generate an ambient environment of 15 kg/m³ density and 383K temperature (same as the temperature
5 of chamber body). This temperature was much lower than the boiling range of diesel. Therefore the spray could be
6 considered as non-evaporating. In the evaporating and combustion conditions, the ambient temperature was set at
7 900 K. Pre-combustion of C₂H₂, N₂ and O₂ mixture ignited by a spark was used to generate a high-temperature and
8 high-pressure engine-like condition. The injection system and high-speed camera were triggered when the ambient
9 temperature dropped to 900K. By changing the initial composition of pre-combustion mixture, the oxygen
10 concentration at the start of injection (SOI) was 0% in the evaporating condition to prevent fuel combustion but was
11 21% in the combustion condition. For all three experimental conditions, the injection pressure, injection duration,
12 fuel temperature and ambient density were kept fixed as shown in Table 2.

13 The spray tip penetration, spray area, spray volume, integrated natural flame luminosity (INFL), flame area,
14 flame lift-off lengths (LOL), flame tip spread distance (FS) and flame tip spread velocity (FSV) are defined
15 according to the Engine Combustion Network [74]. The images were processed by a MATLAB program.
16 Thresholding method is critical for image segmentation of spray and combustion images [72, 75]. In terms of spray
17 images, an automatic threshold determination algorithm was used to segment the boundary of cold and evaporating
18 spray from the schlieren background. The detailed image processing method can be found in references [76, 77].
19 For combustion images, a flame luminosity classification method was used to clarify the luminosity gradient and
20 flame luminosity distribution [78]. The raw and treated images are shown in Fig.2. The combustion flame is
21 classified into six luminosity ranges according to the grayscale value of the pixel. Specifically, grayscale range of
22 0-4 is defined as background (BG), ranges of 5-63, 64-127, 128-191 and 192-254 are defined as R1, R2, R3 and R4
23 respectively, and the overexposed area with a grayscale value of 255 is defined as SA.

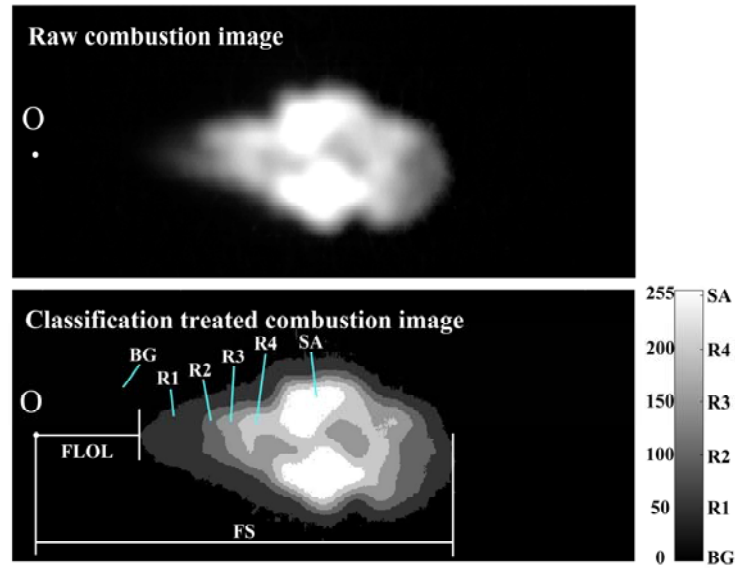


Fig. 2. Raw and classification images of combustion flame.

Table 2. Experimental conditions.

Test fuels	E10, W10, D100
Ambient temperature	383 K (cold condition) 900 K (evaporating and combustion conditions)
Ambient oxygen concentration	0% (cold and evaporating conditions) 21% (combustion condition)
Ambient density	15 kg·m ⁻³
Injection pressure	150 MPa
Injection duration	2.5 ms
Injector temperature	383 K
Injector type	Bosch CRIN 2
Nozzle type	single hole SAC
Nozzle Diameter	0.234 mm

To test the repeatability and accuracy of experimental results, each experimental condition was repeated three times in this study. The averaged values of the three experimental results are adopted to analyze the spray and combustion characteristics. The error bars were used to show the experimental uncertainties.

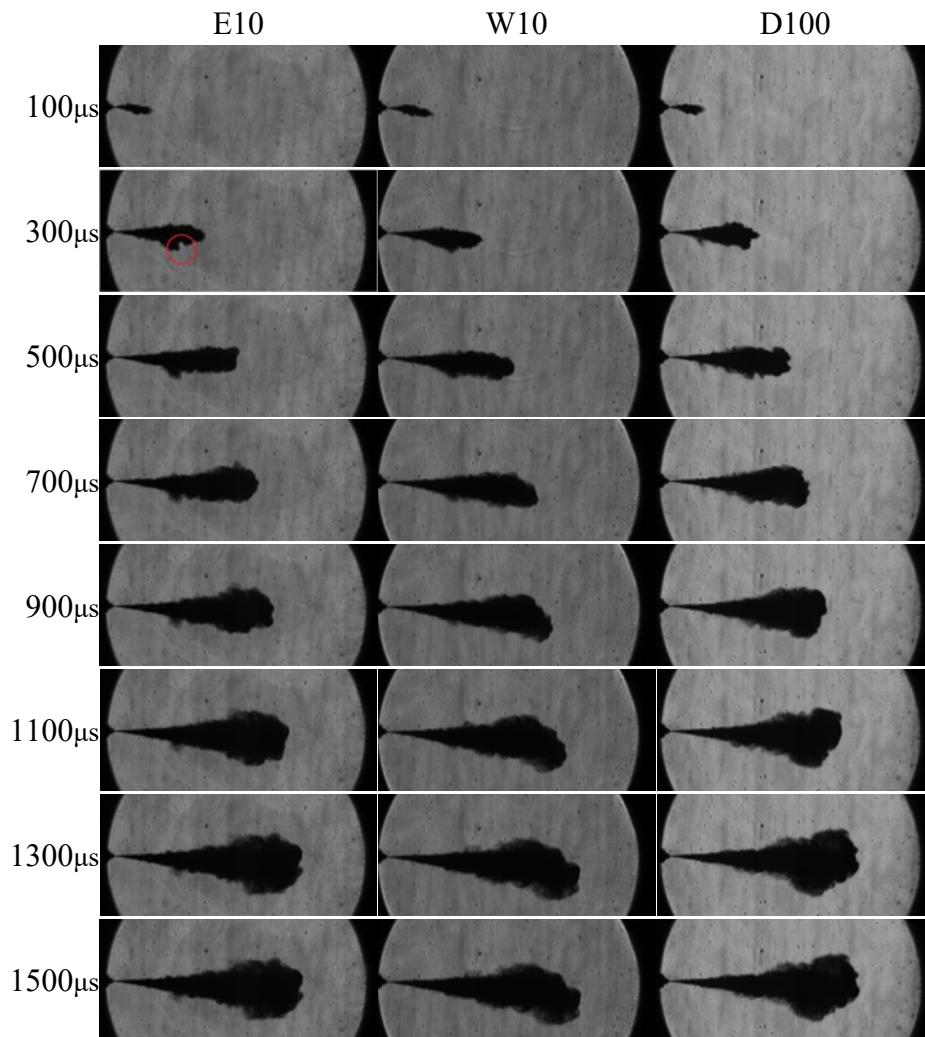
4. Results and Discussion

The experimental results will be presented and discussed as follows. Sections 4.1 and 4.2 report the spray characteristics of E10, W10 and D100 fuels in the cold (383K and 0% O₂) and evaporating (900K and 0% O₂) conditions, respectively. In Section 4.3, the spray, combustion and emissions characteristics of the three fuels will be investigated in the combustion condition (900K and 21% O₂).

4.1. Spray characteristics in cold condition

Fig. 3 shows the development of cold sprays of E10, W10 and D100 fuels. As shown in Fig. 3, E10 and W10

1 sprays are slightly thinner than that of D100 during 100~900 μ s. That is mainly because the viscosities of E10 and
2 W10 are much larger than that of D100 (Table 1). This means that the liquid columns of E10 and W10 have greater
3 capability to resist the deformation caused by gas-liquid interaction and in-nozzle cavitation. Besides, E10 spray
4 shows a clear split at 300 μ s, which is distinctive from W10 and D100. The possible reason is that ethanol
5 evaporates under ambient temperature of 383K and the ethanol vapor ruptures the spray liquid column and causes
6 the spray split. The occurrence of split in E10 spray shown in Fig. 3 is frequent, but not in every spray event.
7 However, the onset timing and location of this split is random. Therefore, further investigation would be carried out
8 to get deeper insight into this unique phenomenon for ethanol diesel blend spray.



9
10 **Fig. 3.** Images of E10, W10 and D100 in cold spray condition.
11

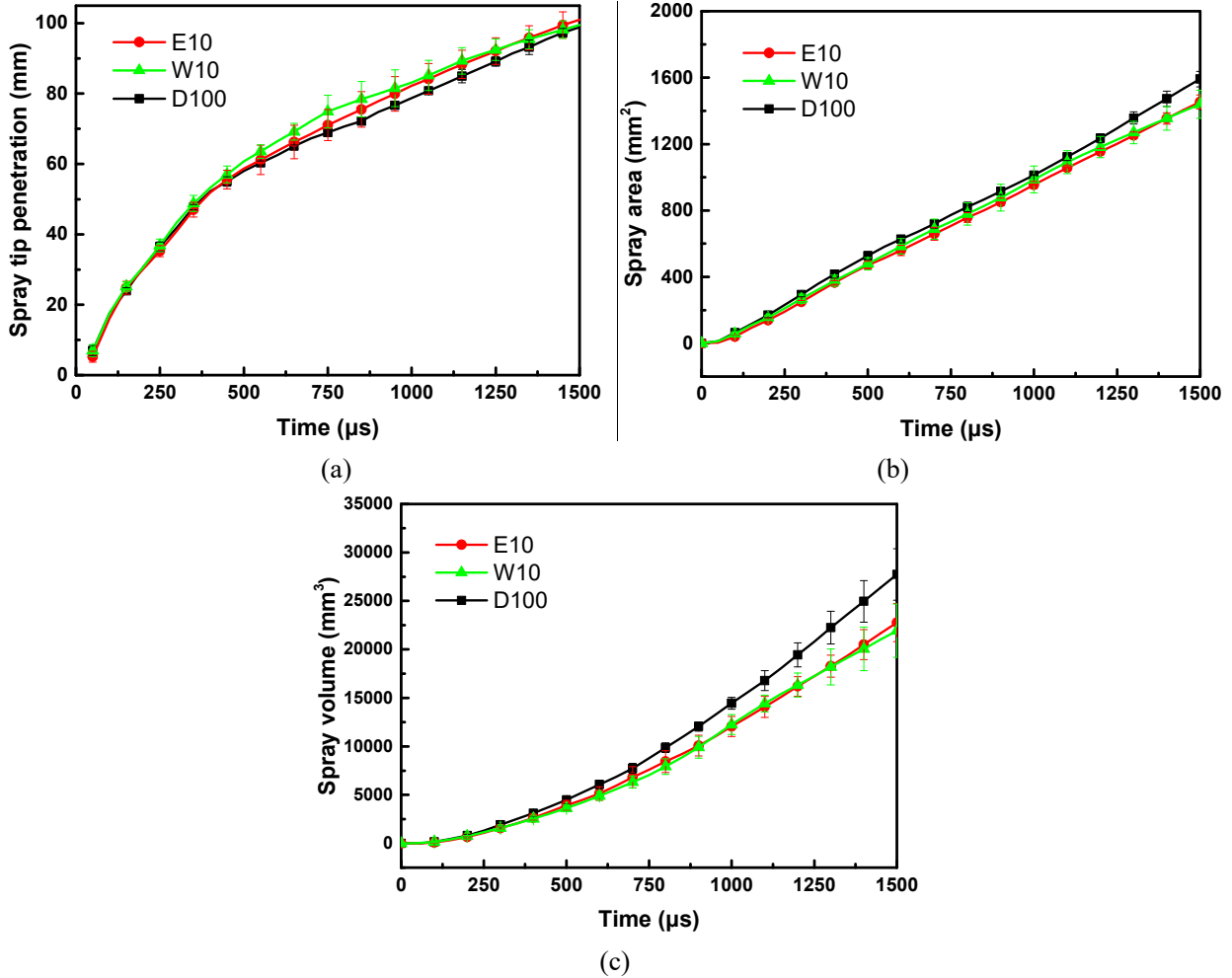


Fig. 4. Cold spray characteristics of E10, W10 and D100: spray tip penetration (a), spray area (b) and spray volume (c).

Fig. 4(a) compares the spray tip penetrations of E10, W10 and D100 in cold condition. It clearly shows that the penetrations of E10 and W10 are longer than that of D100. This is because the viscosities of E10 and W10 are much greater than D100. As a result, E10 and W10 sprays can resist stronger gas-liquid relative movement without deformation and break-up. Therefore, their axial spray penetrations become longer. Figs. 4(b) and 4(c) show the spray areas and volumes of E10, W10 and D100 in cold condition. The spray areas and volumes of E10 and W10 are roughly the same, but they are less than that of D100. Specifically, at 1500μs, the spray volumes of E10 and W10 are 22764 and 21926 mm³ respectively, which are 18% and 21% smaller than that of D100 (27725mm³). This indicates that the higher viscosities of E10 and W10 result in weaker spray expansion capacities of E10 and W10 sprays, which is consistent with the thinner spray structure of E10 and W10 shown in Fig. 3.

4.2. Spray characteristics in evaporating condition

Fig. 5 shows the spray processes of E10, W10 and D100 in evaporating condition. There are mainly two regions in the evaporating spray zone, namely the inner liquid core region with dark black color and the outer sheath-like liquid-gas transition region with brighter color [79]. In the beginning (0-400 μs), only the dark liquid

1 region can be observed because of the limited evaporation rate before spray break-up. As the spray penetrates
 2 further (400-500 μs), the heating effect of the entrained hot gas becomes strong and the fuel starts to evaporate in
 3 the spray tip zone. Therefore, a sheath-like gas-liquid mixed region is formed in the spray tip zone. After 700 μs ,
 4 the length of the liquid-core region becomes stable for all the three fuels, indicating that the fuel evaporation rate
 5 equals to its injection rate. After the end of injection at 2500 μs , the heating of ambient gas gradually vaporizes the
 6 liquid-core region which disappears after 3000 μs . Comparing Figs. 5 and 3, it is obvious that the boundary of
 7 evaporating spray is much smoother than that of cold spray. This suggests that the tiny burrs, which are supposed to
 8 occur in the evaporating spray boundary, are rapidly evaporated by the hot gas entrained, forming a gas phase
 9 region at the spray periphery that is blurred but is still recognizable in motion images.

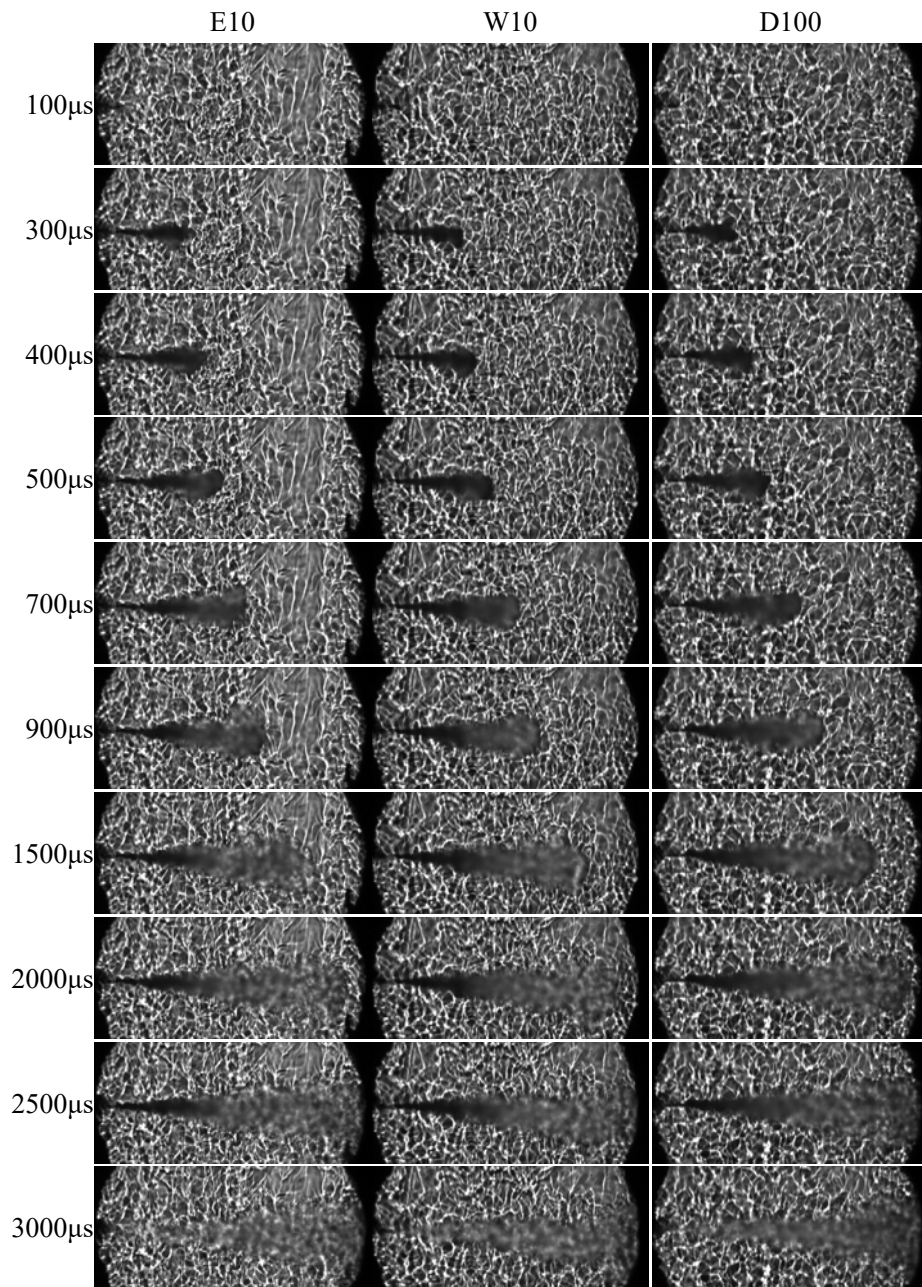
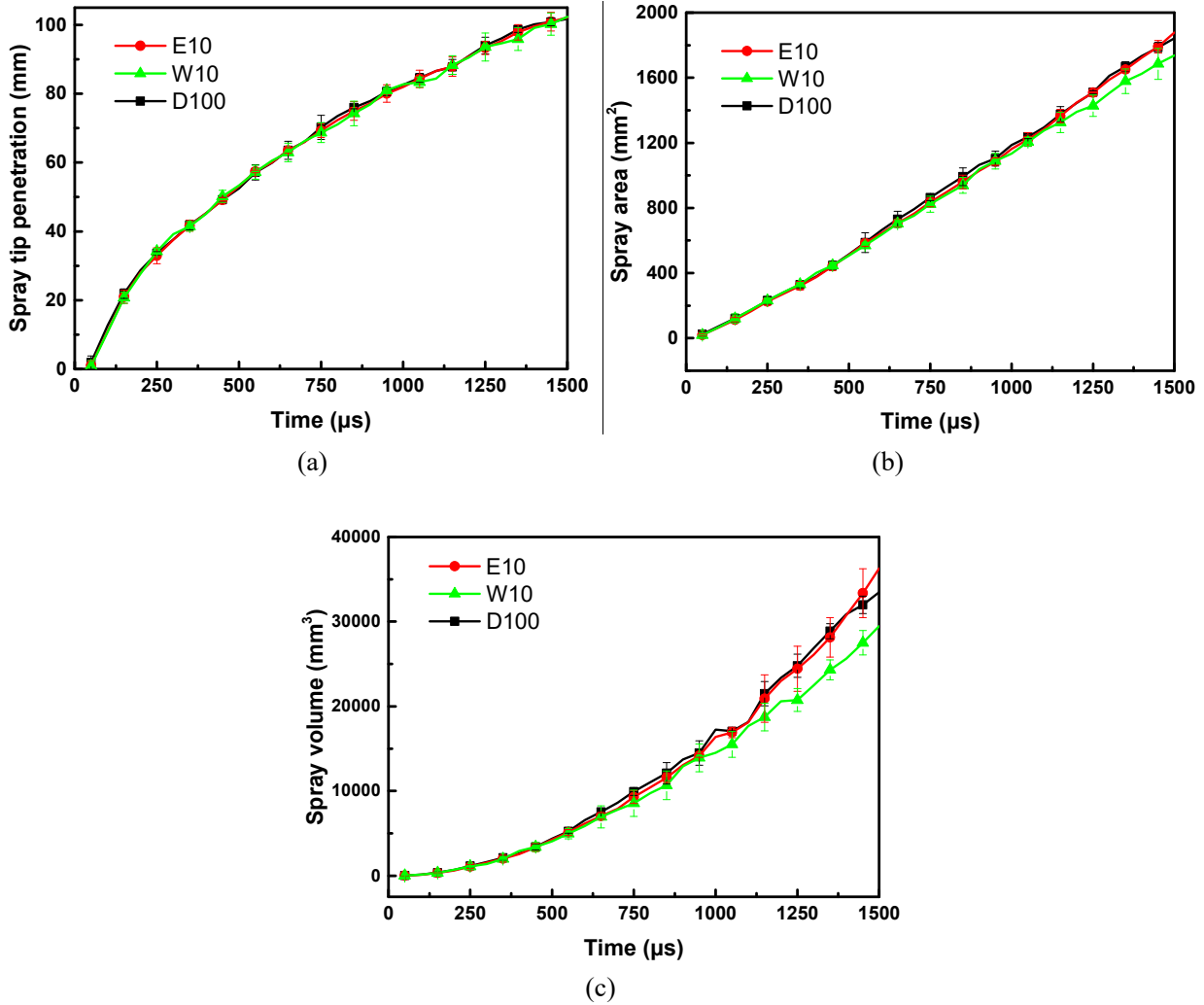


Fig. 5. Images of E10, W10 and D100 in evaporating spray condition.

1 Schlieren imaging technique measures the spray by the density gradient between the fuel and ambient air. The
 2 greater the density gradient is, the more clearly the spray boundary is. The molecular weight of diesel (170,
 3 represented by n-dodecane) is significantly larger than those of water (18), air (29) and ethanol (46). It can be
 4 deduced that the spray boundary between diesel vapor and air would be readily to be distinguished, while the
 5 boundary between ethanol (or water) vapor and air would be less clear. Therefore, it can be roughly determined that
 6 all the evaporation zone detected by the schlieren system in this paper should be the diesel vapor region.

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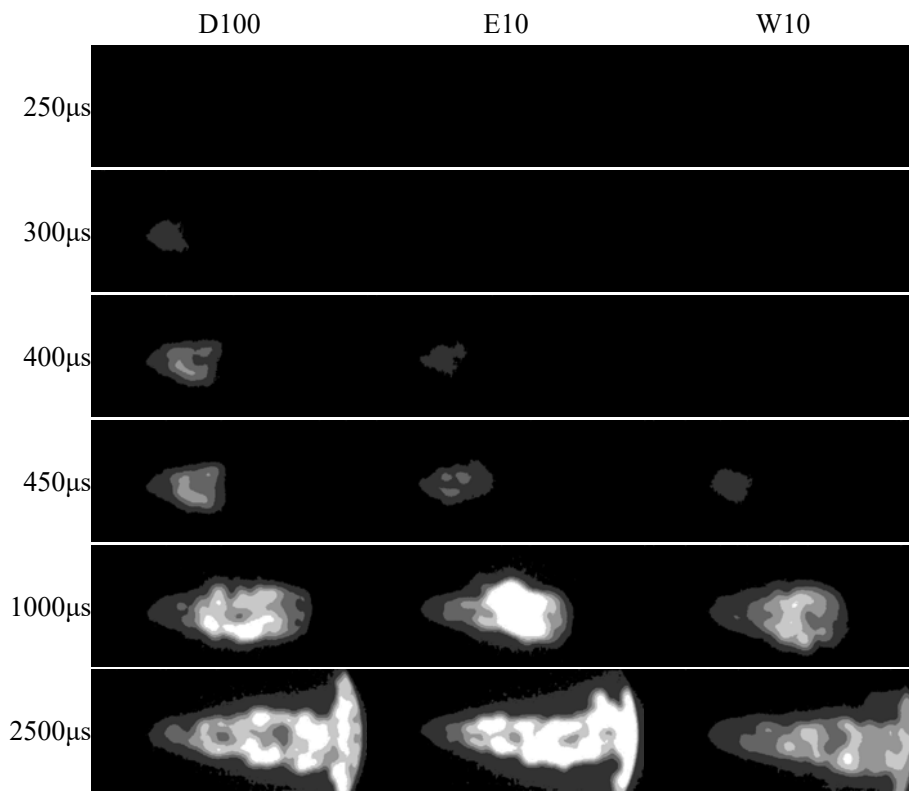
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13 **Fig. 6.** Evaporating spray characteristics of E10, W10 and D100: spray tip penetration (a), spray area (b) and spray volume (c).

14 Fig. 6(a) compares spray tip penetrations of E10, W10 and D100 in evaporating condition. The penetrations of
 15 the three fuels are close to each other, which indicate that the effect of evaporation mitigates the influence of
 16 viscosity on spray penetration under high temperature condition. Figs. 6(b) and 6(c) show the spray areas and spray
 17 volumes of E10, W10 and D100. The spray areas and volumes of E10 and D100 are nearly identical, but they are
 18 clearly bigger than those of W10. At 1500 μs, the evaporation volumes of E10, D100 and W10 are 36283, 33468
 19 and 29476 mm³ respectively. Compared with cold condition, the spray volumes of E10, W10 and D100 increase
 20 respectively by 59%, 34% and 21%. This demonstrates that the evaporation expansions of E10 and W10 are

1 significantly greater than D100 under high temperature condition. In the view of that schlieren evaporation images
 2 only visualizes diesel vapor (as described above), it can be inferred that there are micro-explosions in E10 and W10
 3 which will eject diesel droplets, thereby promoting the expansion of the spray volume. However, the existence of
 4 puffing cannot be verified. The figures also show that the strength of E10 micro-explosion is greater than that of
 5 W10. Fig. 6(c) also shows that the spray volume of E10 almost overlaps with D100, while the spray volume of
 6 W10 is 12% lower than D100 at the ending point. It means the enhancing effect of ethanol micro-explosion can
 7 completely overcome the poor expansion caused by higher viscosity of E10, while the micro-explosion of water
 8 cannot overcome the effect of higher viscosity of W10.

9 4.3. Spray, combustion and emission characteristics in combustion condition

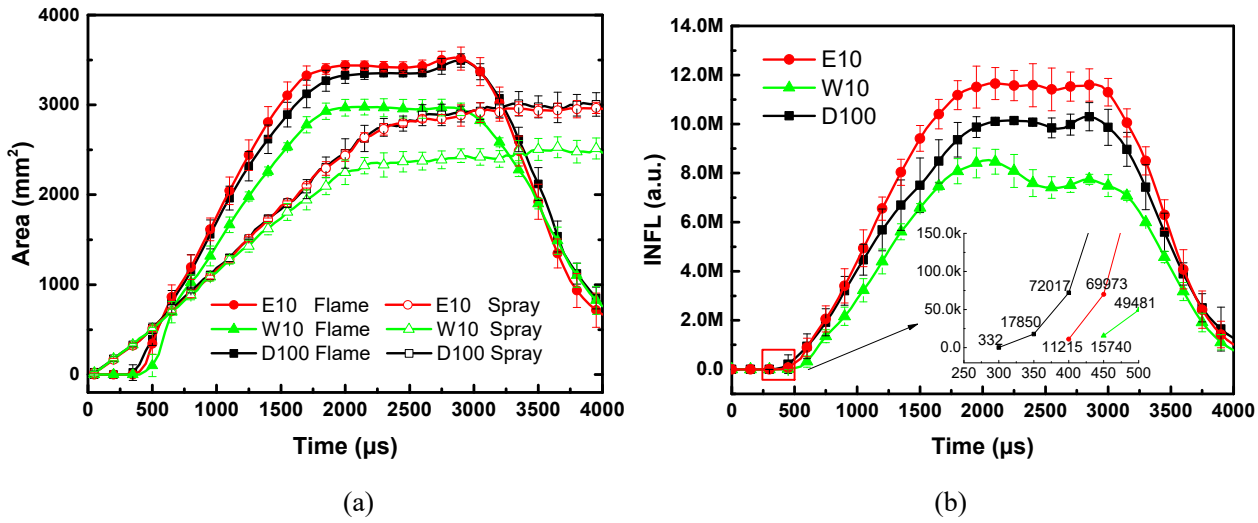


10 **Fig. 7.** Images of E10, W10 and D100 in spray combustion condition.

11 Fig. 7 shows the images of E10, W10 and D100 in the combustion condition. Based on the image luminosity,
 12 the ignition delays of D100, E10 and W10 are 300, 400 and 450 μ s, respectively. Although the intensity of flame
 13 luminosity is strongly dependent on the optical configuration, the contrast of the luminosity-based ignition delay in
 14 different test conditions is still meaningful. Results in Fig. 7 demonstrates that the ethanol and water contents can
 15 significantly slow down the ignition of the blended diesel fuels. The reason for longer ignition delay of E10 could
 16 be deduced as follows. As discussed in Fig. 6(c), the mass of entrained hot air in the evaporating spray is
 17 approximately the same for E10 and D100. However, since the latent heat of ethanol is larger, the overall
 18 temperature of E10 spray should be lower than D100 before ignition, leading to a longer ignition delay of E10 than
 19 D100. The reason for the longer ignition delay of W10 could be interpreted from three aspects. Firstly, the mass of

1 entrained hot gas in W10 spray is less than that in E10 and D100 sprays. Secondly, the higher specific heat and
 2 latent heat of water further reduce the local temperature. Thirdly, water contains no energy and therefore dilutes the
 3 energy density of fuel-air mixture.

4 Fig. 8(a) shows the flame areas and evaporation spray areas of E10, W10 and D100. The flame data covers a
 5 time duration of 4000 μ s, corresponding to about 45 crank angle degrees after TDC in engine condition at 1900
 6 r/min. The combustion process finishes after this crank angle in a real engine equipped with the same injector.
 7 Therefore, flame features at 4000 μ s are used to represent emission performance of three fuels in this paper. Fig. 8(a)
 8 shows that flame areas of E10 and D100 are nearly the same at the beginning of combustion, and both are greater
 9 than that of W10. As combustion progresses, flame area of E10 becomes larger than that of D100 gradually.
 10 Between 1700~3000 μ s, flame areas of all three fuels step into a stable stage, indicating that the combustion reaches
 11 a quasi-steady state. Compared with evaporation spray areas, the flame areas of E10, W10 and D100 increased
 12 respectively by 14.1%, 16.6% and 8.4%. This means that the spray area further expands under the combustion
 13 conditions and the flame area expansions of E10 and W10 are greater than that of D100. So do the flame volumes.
 14 It can be inferred that micro-explosion occurs in the combustion of E10 and W10, which expands the spray volume
 15 further and enlarges the flame area.



18 **Fig. 8.** Flame and evaporation spray areas (a) and Integrated Nature Flame Luminosity (b) of E10, W10 and D100

19 In comparison, it should be noted that the expansion ratios of E10, W10 and D100 from cold to evaporating
 20 sprays are 59%, 34% and 21%, respectively. While, the expansion ratios of E10, W10 and D100 from evaporating
 21 to combustion sprays are 14.1%, 16.6% and 8.4%, respectively. This indicates that the strength of E10
 22 micro-explosion is greater than that of W10 under evaporating condition, while W10 becomes greater under
 23 combustion condition. It means, E10 micro-explosion occurs at a lower temperature and it mainly affects the
 24 evaporation characteristics, while W10 micro-explosion occurs at higher temperature and it mainly affects the
 25 combustion characteristics.

26 Integrated natural flame luminosity (INFL) is defined as the sum of flame luminosity values of all pixels in the

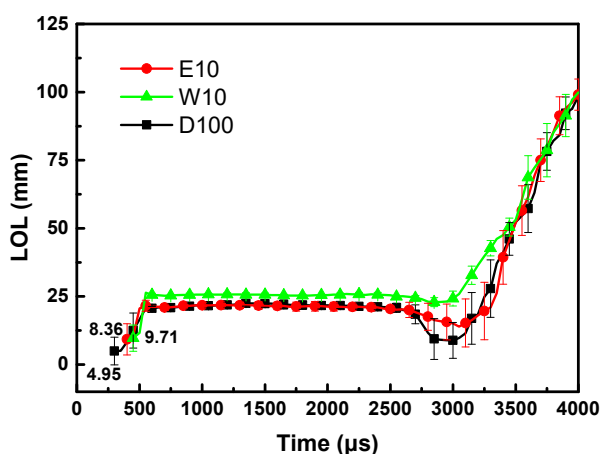
1 combustion zone. It qualitatively reveals the soot concentration to some extent during the combustion process [80].
2 Since the fuel heating value has a subtle impact on INFL [81], INFLs of E10, W10 and D100 were corrected by
3 their equivalent heating values in order to quantitatively analyze the combustion characteristics. The INFLs under
4 the same heating value are compared in the following analysis. Fig. 8(b) shows the corrected INFLs of E10, W10
5 and D100. The peak INFLs of E10, D100 and W10 are 11.65M, 10.30M and 8.53M, respectively. According to the
6 similarity principle, they show that the mass of soot generated in E10 and W10 combustion are 13.11% more and
7 17.18% less than that of D100, respectively. It is noteworthy that the peak value and increasing rate of E10 INFL is
8 much greater than that of D100, which contradicts with the common sense that ethanol-diesel should have lower
9 INFL due to the following two factors. Firstly, E10 has higher latent heat than D100, which should reduce the local
10 gas temperature. Secondly, the micro-explosion of ethanol should enhance the fuel-air mixing and reduce the
11 equivalence ratio. The possible cause for this contradiction could be the greater combustion rate of E10 resulted
12 from the better fuel-air mixing. The greater combustion rate may lead to higher local temperature and stronger soot
13 radiation, as addressed by [82-84]. In addition, the flame area of E10 is larger than D100 (as shown in Fig. 8(a)).
14 Consequently, a greater INFL increasing rate is observed in E10. On the other hand, the peak value and increasing
15 rate of W10's INFL are much lower than that of D100. It should be noted that the gas entrainment capacity of W10
16 is weaker than that of E10 and D100. Together with the high specific heat and latent heat of water, the local
17 temperature rising rate in W10 is supposed to be much lower than D100. This could decrease the combustion rate,
18 combustion temperature, soot generation and INFL.

19 In the later stage of combustion (3000-4000 μ s), E10's INFL reduces faster than that of D100, suggesting that
20 higher combustion temperature of E10 flame promotes the oxidation of soot. Additionally, ethanol is an oxygenated
21 fuel. The blending of 10% ethanol is equivalent to 3% oxygen being added into the fuel, which further promotes the
22 oxidation of soot. Although more soot is generated during the combustion, the oxidation rate of soot in E10 is faster
23 than that of D100, leading to less soot emission in E10 than D100 at the end of combustion. On the other hand, the
24 declining rate of W10 INFL is roughly the same as that of D100 during 3600-4000 μ s. This suggests that the soot
25 oxidation rates of W10 and D100 are approximately the same although the combustion temperature of W10 is lower.
26 The possible explanation is that water-gas reactions occur in W10 combustion, which promotes the oxidation of
27 soot. Consequently, at 4000 μ s, INFLs of E10, W10 and D100 are 1.008M, 0.78M and 1.27M respectively. This
28 implies that the final residual amount of soot in E10 and W10 combustion are 21% and 39% less than D100.

29 As shown in Fig.8, the combustion duration of W10 is clearly shorter than D100, which indicates that the
30 overall combustion rate of W10 is higher than D100 although the combustion temperature of W10 is lower than
31 D100. It can be inferred again that the micro-explosion occurs in the combustion of W10, which enhances the
32 fuel-air mixing and accelerates the diffusion combustion rate.

33 Fig. 9 shows the flame lift-off lengths (LOL) of E10, W10 and D100. LOL is defined as the distance between
34 the nozzle outlet and the flame root. Fig. 9 shows that LOLs of three fuels become constant shortly after the

1 ignition. Higgins and Siebers [85, 86] found that the preparation speed of the combustible fuel-air mixture
 2 approximately equaled to the reaction speed at the stabilized LOL position. Therefore, LOL can be used to evaluate
 3 the mixing speed of the spray [74]. During the quasi-steady state, the LOL of E10 is nearly the same as that of
 4 D100, and both are smaller than W10. Figs. 6(b) and 6(c) show that the gas entrainment capacity of E10 equals to
 5 D100 and both are stronger than W10. However, considering that the latent heat of E10 is larger than that of D100
 6 which means the local temperature of E10 is lower, it can be inferred that the oxygenated feature of ethanol
 7 promotes the ignition process and compensates the negative effect of lower temperature. The temperature rising
 8 rate of W10 is even lower than that of E10 due to W10's weaker gas entrainment capacity and higher specific heat
 9 and latent heat. Unfortunately, the negative impact of lower temperature can no longer be compensated by water's
 10 oxygenated feature because it's incombustible.



11
 12 **Fig. 9.** Flame lift-off length of E10, W10 and D100.

13 Fig.10 shows the flame tip spread velocity (FSV) of E10, W10 and D100. FSV demonstrates the burning rate
 14 in downstream. As shown in Fig. 10, although FSV of W10 was lower than E10 and D100 at first, it surpassed E10
 15 and D100, and reached the highest peak value latter. Possible reason for this phenomenon could be interpreted as
 16 follows. In downstream, though the combustion temperature in W10 is lowest, the spray liquid cores have been
 17 pre-heated by the surrounding flame and the strength of micro-explosion is promoted which can overcome the
 18 negative effects of higher endothermic properties and consequently accelerate the burning of W10, making the
 19 fastest FSV in W10. The results further prove that the onsets of micro-explosion in W10 occur later and the strength
 20 of micro-explosion in W10 is greatest near the spray tip which leads to the greatest downstream burning rate of
 21 W10.

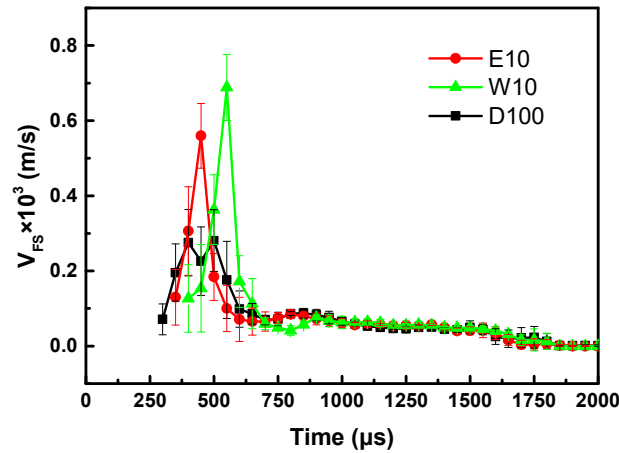


Fig. 10. Flame tip spread characteristics of E10, W10 and D100.

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Conclusions

In the present study, the spray, combustion and emissions characteristics of ethanol-diesel (E10), water-emulsified diesel (W10) and neat diesel (D100) were investigated experimentally in a constant volume combustion chamber. The experiments were performed under cold (383K and 0% O₂), evaporating (900K and 0% O₂) and combustion (900K and 21% O₂) conditions. By means of subtle evolution analysis on image area and estimated volume of fuel sprays from 383K to 900K and combustion conditions, the reported work explores the existence of micro-explosion in E10 and W10 under real engine conditions and the effect of micro-explosion on spray and combustion of multi-component fuels. The major findings of are summarized as followings.

(1) In cold condition, the spray volumes of E10 and W10 were much smaller than that of D100 due to their higher viscosities. Compared with cold sprays, the evaporating spray volume of E10, W10 and D100 increased by 59%, 34% and 21%, respectively. Compared with evaporating sprays, the flame volumes of E10, W10 and D100 increased by 14.1%, 16.6% and 8.4%, respectively. The enhancing effects in spray and flame volume of E10 and W10 were clearly stronger than that of D100 from cold to evaporating and then combustion conditions. This proves the existence of micro-explosion in E10 and W10.

(2) E10 micro-explosion started at a lower temperature (reached before ignition) and mainly affected the evaporation characteristics, while W10 micro-explosion started at a higher temperature (reached after ignition) and mainly affected the combustion characteristics.

(3) LOL tendency showed that the micro-explosion and oxygenated feature of ethanol could counterweigh the negative impact of higher latent heat of E10. However, the micro-explosion of water was less strong and could not overcome the negative impacts in flame upstream. FSV tendency demonstrated that the micro-explosion strength of water was promoted greatly by pre-heat and led to higher peak FSV of W10, which was reached later in downstream than those of D100, and E10.

(4) The final soot emissions of E10 and W10 were 21% and 39% lower than that of D100.

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