

Article

Optimising the Particulate Emission Characteristics of a Dual-Fuel Spark Ignition Engine by Changing the Gasoline Direct Injection Strategy

Xiang Li ^{1,*} , Siyue Liu ², Wanzhong Li ^{3,*}, Yiqiang Pei ^{3,*}, Xuwen Zhang ¹, Peiyong Ni ¹, Zhijun Peng ⁴ and Chenxi Wang ⁵ 

¹ School of Mechanical Engineering, Nantong University, Nantong 226019, China; zhangxuwen@ntu.edu.cn (X.Z.); nipeiyong@ntu.edu.cn (P.N.)

² BYD Co., Ltd., Shenzhen 518038, China; liu.siyue@byd.com

³ State Key Laboratory of Engines, Tianjin University, Tianjin 300072, China

⁴ School of Engineering and Physical Sciences, University of Lincoln, Lincoln LN6 7TS, UK; jpeng@lincoln.ac.uk

⁵ Institute for Transport Studies, University of Leeds, Leeds LS2 9JT, UK; c.wang2@leeds.ac.uk

* Correspondence: xiang.li@ntu.edu.cn (X.L.); lwzhong@tju.edu.cn (W.L.); peiyq@tju.edu.cn (Y.P.)

Abstract: In the current global scenario, it is essential to find more effective and practical solutions to mitigate the problem of particulate emissions from vehicles. In this research, particulate emission characteristics with changing GDI pressure or applying a split GDI strategy with different second injection timings were initially explored in a Dual-Fuel Spark Ignition (DFSI) engine, which employs Ethanol Port Injection (EPI) plus Gasoline Direct Injection (GDI). The experimental results show that by increasing GDI pressure (P_{GDI}) from 5.5 MPa to 18 MPa, ignition delay (θ_F) shows a small decrease of 0.68 degrees. The parameters, such as maximum in-cylinder temperature (T_{MI}) and exhaust gas temperature (T_{EG}), each increase by 53.75 K and 13.84 K. An apparent reduction of 59.5% and 36.26% was achieved for the concentrations of particulate number (N_p) and particulate mass (M_p), respectively. Particulate emissions are effectively reduced by a split GDI strategy with an appropriate range of second injection timing (t_{GDI2}). Under $t_{GDI2} = -260^\circ\text{CA}$, N_p and M_p concentrations exhibit a relatively lower level. However, by delaying t_{GDI2} from -260°CA to -140°CA , there is an increase of more than 60% in N_p concentration. The research findings help offer new and valuable insights into optimising particulate number and mass emissions from DFSI engines. Moreover, the findings could contribute novel and valuable insights into the optimisation of particulate emission characteristics in DFSI engines.

Keywords: dual-fuel spark ignition (DFSI) engine; particulate emissions; ethanol port injection (EPI); gasoline direct injection (GDI); injection pressure; split GDI strategy



Citation: Li, X.; Liu, S.; Li, W.; Pei, Y.; Zhang, X.; Ni, P.; Peng, Z.; Wang, C. Optimising the Particulate Emission Characteristics of a Dual-Fuel Spark Ignition Engine by Changing the Gasoline Direct Injection Strategy. *Sustainability* **2024**, *16*, 8713. <https://doi.org/10.3390/su16198713>

Academic Editor: Francesco Nocera

Received: 3 September 2024

Revised: 4 October 2024

Accepted: 8 October 2024

Published: 9 October 2024



Copyright: © 2024 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution (CC BY) license (<https://creativecommons.org/licenses/by/4.0/>).

1. Introduction

Particulates emitted from vehicles have been a severe problem that affects air quality, climate change, human health, and environmental sustainability [1–4]. In order to mitigate particulate emissions, few advanced solutions have been explored in the existing vehicle power train.

Pure battery electric is a promising technology that has gradually gained an increasing market share. Nonetheless, popularising battery electric vehicles is generally limited by the anxiety of insufficient charging infrastructure in rural areas, cruising range, and long charging time [5,6]. Hydrogen Internal Combustion Engines (ICEs) and ammonia ICEs are newly developed solutions in recent years. However, these two solutions still face challenges related to abnormal combustion, safety, and toxicity, which need to be resolved to improve the stability and duration of the engines [7–9].

The vehicles powered by typical ICEs, such as ICE-only vehicles and hybrid vehicles, are currently the most common types of passenger transport. Moreover, new approaches are consistently implemented for ICE-only and hybrid vehicles to improve engine combustion efficiency and reduce exhaust emissions. Among them, Dual-Fuel Spark Ignition (DFSI) engines fuelled with gasoline and ethanol are one of the most attractive approaches, and can be classified into two modes. This approach can be classified into two modes, such as Gasoline Port Injection (GPI) plus Ethanol Direct Injection (EDI), and Ethanol Port Injection (EPI) plus Gasoline Direct Injection (GDI), as shown in Figure 1.

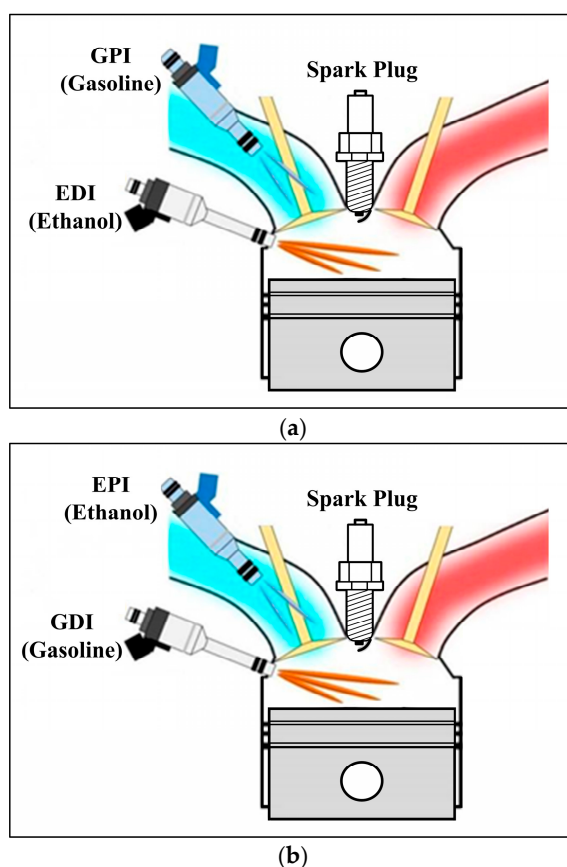


Figure 1. Schematic diagram of DFSI engine fuelled with gasoline and ethanol. (a) GPI plus EDI mode, (b) EPI plus GDI mode.

Regarding the recent important research on DFSI engines with GPI plus EDI mode, Zhuang et al. [10] examined the emission performance of carbon monoxide (CO), nitric oxide (NO) and hydrocarbon (HC) by changing the EDI energy ratio. Zhao et al. [11] found that both gaseous and particulate emissions can be mitigated by the combined effects of appropriate exhaust gas recirculation rates and the air–fuel equivalence ratio (λ). Li et al. [12] systematically explored the potential benefits of optimising EDI timing and pressure on reducing particulate emissions.

Regarding the other mode, DFSI engines with EPI plus GDI, one major advantage in practical application is that the ability of engine transient response can be enhanced, particularly during low-load and cold-start conditions, owing to the heating value and reduced latent heat of vaporisation [12]. Several key research advancements in the combustion and emission characteristics of DFSI engines with EPI plus GDI mode have also been achieved, as follows. Kim et al. [13] explored the impacts of varying compression ratios and EPI timings on engine performance under a Wide Open Throttle (WOT) condition. It was found that compared to the GDI-only mode, the EPI plus GDI mode could benefit both the combustion performance improvement and particulate emissions reduction. The

maximum reduction in the number and mass of particulate emissions are 87% and 98.9%, respectively. Liu et al. [14] demonstrated that an optimal mass fraction of ethanol injection could help minimise particulate emissions while maintaining thermal efficiency and power output under WOT conditions. In a single-cylinder DFSI engine including EPI plus GDI mode, Kang et al. [15] investigated the effects of spark timing on Brake Mean Effective Pressure (BMEP), brake thermal efficiency, and combustion phase under WOT conditions. Yu et al. [16] examined the characteristics of combustion, gaseous, and particulate emissions by changing the GDI timing, GDI ratio, and lambda. It indicated that when lambda is 1.0 and 1.2, the total particle number shows a general rising trend by increasing the gasoline addition ratio from 0% to 40%. By further investigating the effects of GDI timing on a DFSI engine, Wang et al. [17] explored the characteristics of deposited fuel film using the laser-induced fluorescence method and analysed the engine particulate emission characteristics under different GDI timings. It was found that increasing the injection duration from 1.2 ms to 3.0 ms could enhance the effects of impingement distance on the deposited film mass ratio.

As stated above, DFSI engines fuelled with gasoline and ethanol have been proven to have advantages in optimising the characteristics of combustion and exhaust emissions. However, concerning DFSI engines with EPI plus GDI mode, the existing research has mainly concentrated on the effects of EPI timing, GDI timing, and mass fraction of EPI or GDI on combustion and particulate emission characteristics. Almost no findings have been reported about the variations and optimisation of particulate emission characteristics by changing GDI pressure or applying a split GDI strategy in DFSI engines with EPI plus GDI mode. Moreover, improved direct injection pressure or a split injection strategy has been widely demonstrated to effectively enhance air–fuel mixture quality, control exhaust emissions, and improve combustion performance in gasoline engines or diesel engines of common types [18–21].

Hence, in this research, particulate emission characteristics were quantitatively measured and analysed in an advanced DFSI engine with EPI plus GDI mode. Particularly, particulate emission characteristics were initially investigated and optimised by changing GDI strategy, such as changing GDI pressure or using a split GDI strategy with different second injection timings in the injection mode of DFSI engines. The novel findings could significantly enhance our understanding of the correlation between particulate emission characteristics and GDI pressure or a split GDI strategy in DFSI engines. Furthermore, new information and knowledge about the theoretical basis and practical approach for mitigating the particulate emissions of vehicles can be explored and gained through this research, thereby making a significant contribution to environmental sustainability.

2. Experimental Methodology

2.1. Experimental Setup

Figure 2 presents the schematic diagram of the engine testbed used in this research. An electrical dynamometer and a programmable Electronic Control Unit (ECU) controlled by the calibration software ETAS-INCA were utilised to monitor the engine operating conditions of speed and torque accurately. The DFSI engine is set to the mode of EPI plus GDI, the specifications of which are shown in Table 1. The displacement and compression ratio of the engine are 2.0 L and 9.6 each. The physicochemical properties of the fuels utilised in this research can be found in Table 2 [22].

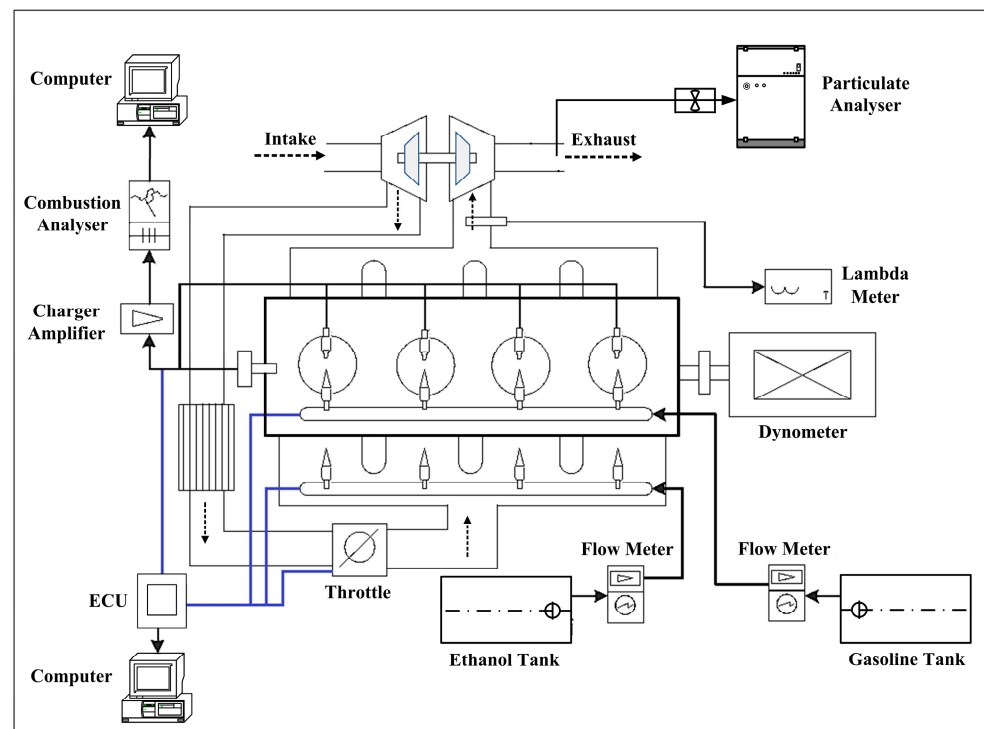


Figure 2. Schematic diagram of the DFSI engine testbed.

Table 1. Engine specifications.

Items	Content
Engine type	4-cylinder, 4-stroke
Bore × Stroke (mm)	82.5 × 92
Displacement (L)	2.0
Injection mode	EPI plus GDI
Intake mode	Turbocharged
Compression ratio	9.6:1
Rated speed (rpm)	5500
Rated power (kW)	160
Maximum torque (N·m)	320

Table 2. Fuel properties [22].

Fuel Type	Ethanol	Gasoline
Chemical formula	C ₂ H ₅ OH	C ₅ -C ₁₂
Relative molecular mass	46	95–120
Gravimetric oxygen content (%)	34.78	<1
Research octane number	107	95
Density (293 K) (kg/L)	0.789	0.73
Dynamic viscosity (293 K) (mPa·s)	1.2	0.52
Kinematic viscosity (293 K) (mm ² /s)	1.52	0.71
Surface tension coefficient (293 K) (mN/m)	21.97	22
Boiling range (K)	351	303–473
Lower heating value (kJ/kg)	26,900	44,300
Latent heat of vaporisation (kJ/kg)	840	370
Laminar flame speed (293 K) (m/s)	0.5	0.33
Stoichiometric air–fuel ratio	8.95	14.7

By connecting to the location in the engine exhaust main manifold before the three-way catalytic converter, a particulate analyser (Cambustion-DMS 500) was used to measure the

emissions of particulate within a diameter of 5 nm to 1000 nm. Based on the threshold of 30 nm diameter, particulates can be classified into nucleation mode (which is smaller than 30 nm) and accumulation mode (which is between 30 nm and 1000 nm) [23]. Additionally, an accurate understanding of combustion characteristics would be critically important because of a strong correlation between engine combustion and emissions. In this research, cylinder pressure signals were measured and recorded continuously in real time using pressure transducers (AVL-GH13Z), charge amplifiers (Kistler-5018A), a crank sensor (Kistler 2614CK1), and a combustion analyser (AVL-641). The lambda was measured instantaneously by a lambda meter (ETAS-LA4).

2.2. Experimental Procedure

During the experiment, the DFIG engine ran at 2000 revolutions per minute (rpm) and 2 bar BMEP, which is a typical urban operating condition of low load. Maximum Brake Torque (MBT) spark timing was applied for each experimental case. To simplify the research process and make it more efficient, the fuel injection mass ratio of EPI to GDI was maintained at 1:1 during the whole research.

The first part of this research is to optimise engine particulate emission characteristics by changing GDI pressure from 5.5 MPa to 18 MPa. Meantime, the injection timing of GDI was fixed at -300°CA , and the fuel injection mass ratio of EPI to GDI was maintained at 1:1.

The second part of this research was to optimise engine particulate emission characteristics using a split GDI strategy with different second injection timings. In this part, the injection pressure of GDI was fixed at 5.5 MPa, and the fuel injection mass ratio of EPI to GDI was still maintained at 1:1. Two-stage injections of GDI were performed, each accounting for 50% of the total GDI mass. The timing of the first injection was still fixed at -300°CA , while the injection timing of the second stage was selected at -260°CA , -220°CA , -180°CA , and -140°CA under different cases.

In order to obtain more precise results, emissions and combustion characteristics were conducted after the engine stabilised for two minutes under each case. Moreover, cylinder pressure was averaged over two hundred consecutive cycles, and the emission results were taken from the average of three repeated measurements. The lambda was maintained at 1 ± 0.01 , with the intercooler outlet temperature and coolant temperature stabilised at $298 \pm 2\text{ K}$ and $360 \pm 2\text{ K}$, respectively. As listed in Table 3, the uncertainties of measured parameters were calculated based on Holman's root mean square method [24].

Table 3. Uncertainties of measured parameters.

Measured Parameters	Uncertainty (%)
Engine speed	± 0.1
BMEP	± 0.1
Fuel consumption rate	± 0.2
Pressure	± 0.1
Crank angle	± 0.1
Lambda	± 0.2
Coolant temperature	± 0.4
Intercooler output temperature	± 0.4
Particulate emissions	± 1.5

In this research, D_p (nm), N_p (n/cm^3), and M_p ($\mu\text{g}/\text{m}^3$) are introduced to represent the diameter, number density, and mass density of particulate measured by the particulate analyser (Cambustion-DMS 500), respectively. The conversion from D_p to M_D (a particulate with a size of D_p) can be seen in Equation (1) [25].

$$M_D(\mu\text{g}) = 1.72 \times 10^{-15} \times D_p^{2.65}(\text{nm}) \quad (1)$$

Further, P_{GDI} represents the GDI pressure. t_{GDI2} represents the injection timing of the second stage in the split GDI strategy. The following combustion parameters can be obtained or calculated by the combustion analyser (AVL-641) and relevant software (AVL-Indicom). φ_{CA10} represents the Crank Angle (CA), where 10% of the cumulative heat has been released. θ_F represents the ignition delay, which is the CA interval from the spark timing to φ_{CA10} . T_{MI} and T_{EG} each represent the maximum in-cylinder temperature and exhaust gas temperature.

3. Results and Discussion

3.1. Optimising Particulate Emission Characteristics by Changing GDI Pressure

In this section, the characteristics of particulate emission were thoroughly examined under P_{GDI} values of 5.5 MPa, 10 MPa, 14 MPa, and 18 MPa.

Figure 3 presents the N_p size distribution with varying P_{GDI} . The N_p size distribution is represented by a size spectral density in $dN/d\log D_p$, which was obtained and presented by the particulate analyser. The advantage of this approach is that integration across different size ranges can be straightforwardly facilitated to determine the particle concentration.

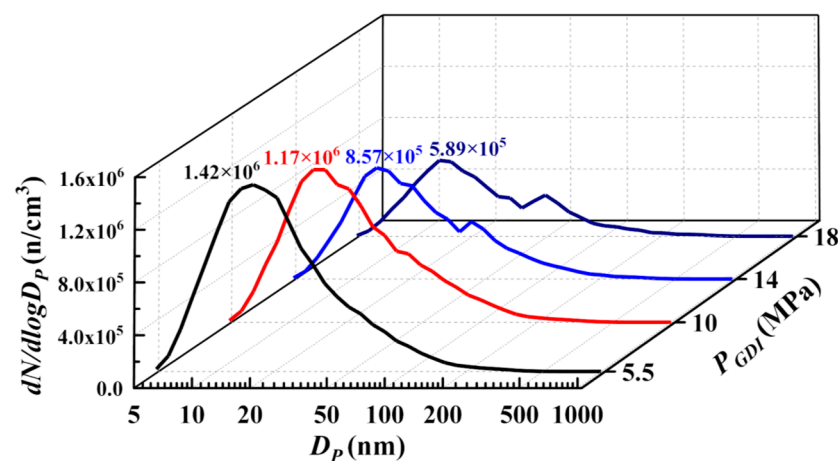


Figure 3. N_p size distribution with varying P_{GDI} .

The most notable feature in Figure 3 is that as P_{GDI} increases from 5.5 MPa to 18 MPa, the peak value of the N_p size distribution curve shows a steady downward trend, which reduces from 1.42×10^6 to 5.89×10^5 . Further, some changes can be observed in the shape of the curve. Under the conditions of $P_{GDI} = 5.5$ MPa and $P_{GDI} = 10$ MPa, the curve displays an approximately unimodal shape, with a peak corresponding to D_p between 10 and 20 nm. Meanwhile, under higher P_{GDI} conditions, the curve changes to be a bimodal shape, including a second peak from the accumulation mode. The magnitudes of the second peak under $P_{GDI} = 14$ MPa and $P_{GDI} = 18$ MPa are each 4.42×10^5 and 3.19×10^5 , which are still lower than the corresponding values under lower P_{GDI} .

The variation in the N_p concentration subject to P_{GDI} can be found in Figure 4. The overall perspective indicates that with the increase in P_{GDI} , a clear benefit to reducing N_p can be provided. In quantitative terms, the N_p concentration of total particulates is up to 9.89×10^5 under $P_{GDI} = 5.5$ MPa. Then, it would progressively reduce to 8.09×10^5 , 5.87×10^5 and 4.01×10^5 when P_{GDI} rises to 10 MPa, 14 MPa, and 18 MPa, respectively. Consequently, the greatest reduction in percentage can be 59.5%. Simultaneously, the lowest values in the N_p concentration of nucleation and accumulation are each 2.87×10^5 and 1.14×10^5 under $P_{GDI} = 18$ MPa, which are 62.39% and 49.6% lower compared to the corresponding values under $P_{GDI} = 5.5$ MPa.

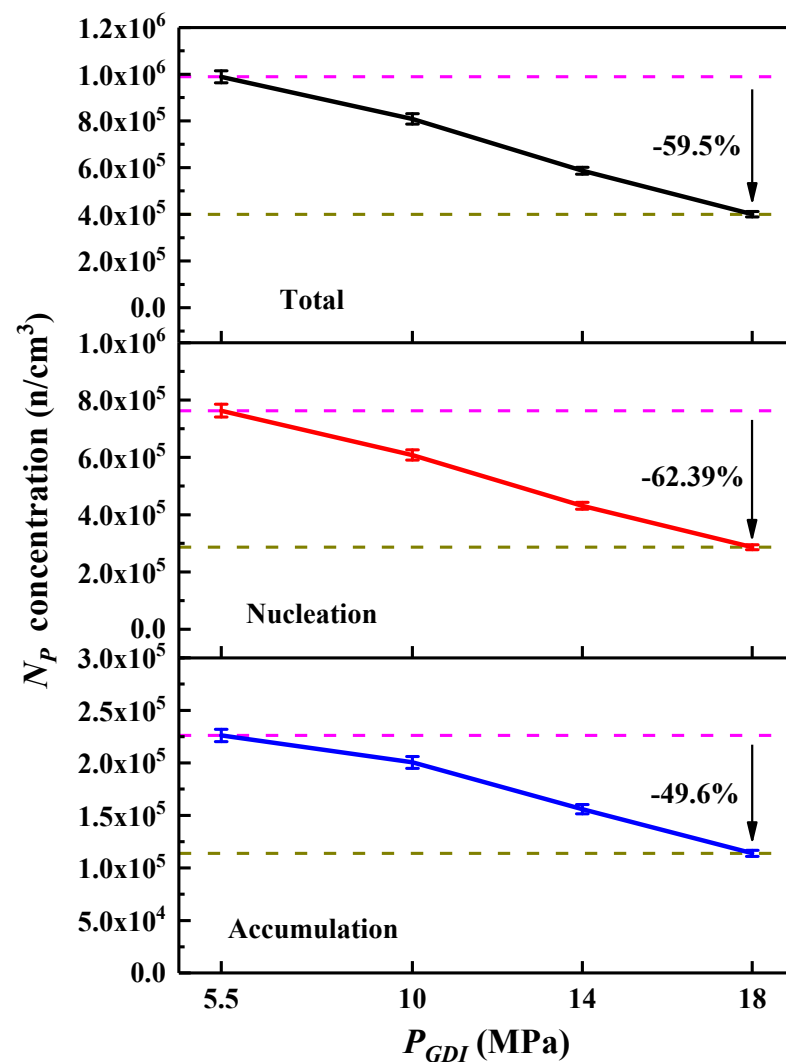


Figure 4. N_p concentration of nucleation, accumulation, and total particulates with varying P_{GDI} .

Figure 5 presents the M_p concentration of nucleation, accumulation, and total particulates with varying P_{GDI} . On the whole, it indicates that changing P_{GDI} could positively affect the M_p concentration of the DFSI engine under low load. As P_{GDI} rises from 5.5 MPa to 18 MPa, the M_p concentration of total particulates decreases from 109.92 to 70.06, which is 36.26% in percentage. Meantime, for the nucleation and accumulation modes of particulates, the maximum reductions in the M_p concentration can be 66% and 35.15%, respectively.

The main reason for the reduction in the concentration of N_p and M_p is that the process of spray atomisation would be quickly completed under the conditions of higher P_{GDI} . The proportion of larger spray droplets would be greatly decreased by increasing P_{GDI} , contributing to forming a more homogenous in-cylinder air–fuel mixture, thereby lowering the possibility of particulate formation [18,26]. Further, it is noticeable that for total particulates, the percentage reduction in M_p is significantly less than that of N_p . This is closely related to the impacts of combustion characteristics with varying P_{GDI} , which is explained in the discussion and analysis of Figures 6–8.

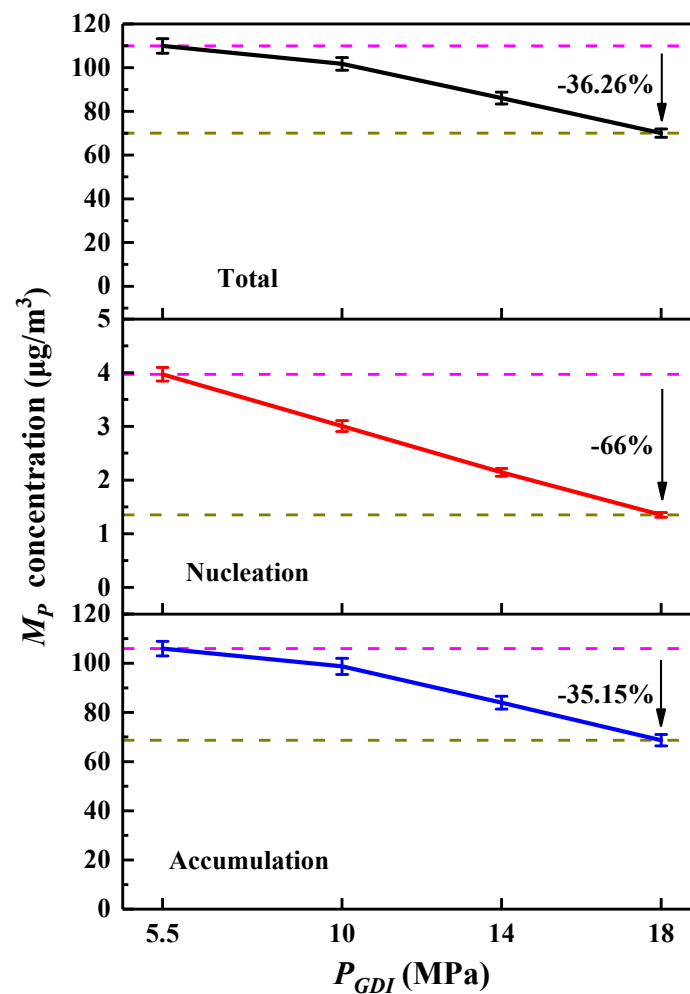


Figure 5. M_P concentration of nucleation, accumulation, and total particulates with varying P_{GDI} .

The exploration of combustion characteristics can be conducted with θ_F , T_{MI} , and T_{EG} . Figure 6 illustrates the effects of P_{GDI} on θ_F . It is observed that under $P_{GDI} = 5.5$ MPa, θ_F is 28.03 degrees, while it reduces to 27.66 degrees, 27.46 degrees, and 27.35 degrees as P_{GDI} increases. From the view of normalisation, it shows that under $P_{GDI} = 18$ MPa, θ_F is shortened by 2.43% compared to the condition of $P_{GDI} = 5.5$ MPa. Although the shortening of θ_F is not very apparent, it still means the combustion process is accelerated, which would be helpful in promoting the improvement in in-cylinder temperature.

The changes in T_{MI} and T_{EG} with varying P_{GDI} can be seen in Figures 7 and 8, respectively. It shows that by increasing P_{GDI} from 5.5 MPa to 18 MPa, T_{MI} rises from 2518.84 K to 2572.59 K, showing a growth of 53.75 K or 2.13%. In the meantime, T_{EG} exhibits an increase of 1.51%, rising from 916.74 K to 930.58 K. As a result, higher T_{MI} and T_{EG} could effectively facilitate more complete combustion and increase the oxidation reaction rate, reducing the probability of in-cylinder particulate formation. Further, the effects of high-temperature pyrolysis would be enhanced under a higher T_{MI} , which would promote the production of particulate precursors [27,28]. Hence, it would partially offset the advantages in particulate reduction by more complete combustion and enhanced oxidation, weakening the decrease in accumulation mode particulates. Consequently, for total particulates, the percentage reduction in M_P is relatively mild compared to that in N_P .

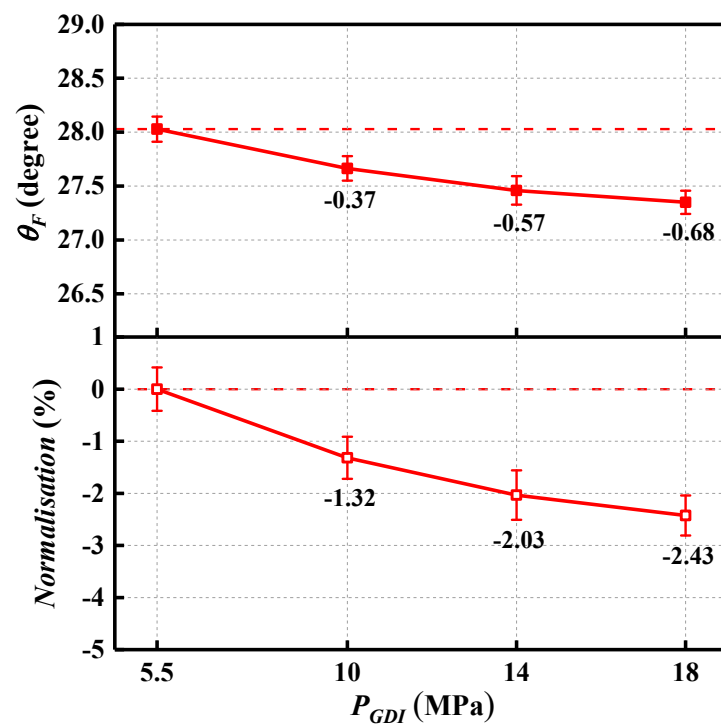


Figure 6. Effects of P_{GDI} on θ_F .

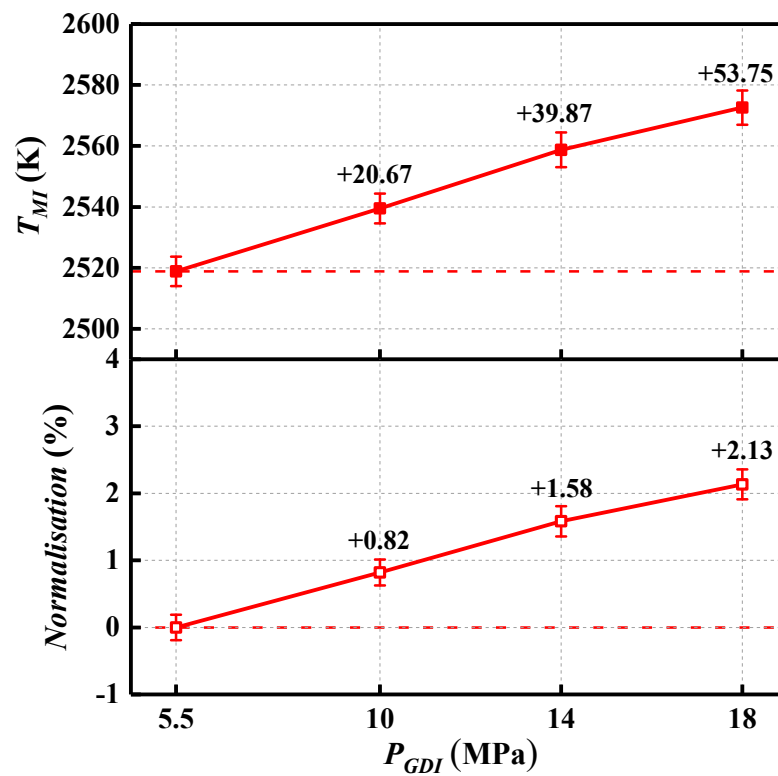


Figure 7. Effects of P_{GDI} on T_{ML} .

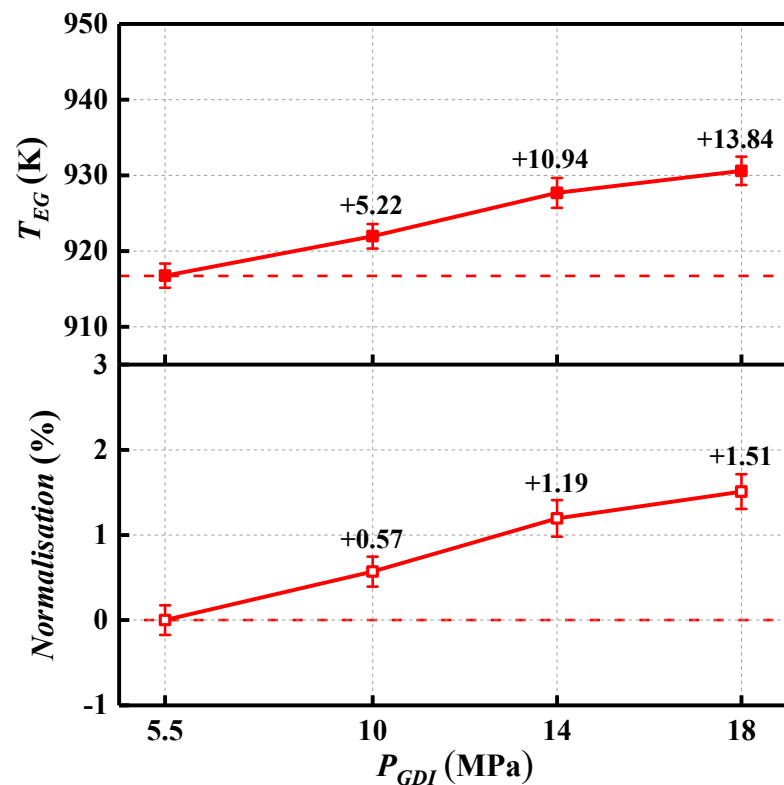


Figure 8. Effects of P_{GDI} on T_{EG} .

3.2. Optimising Particulate Emission Characteristics by Using the Split GDI Strategy with Different Second Injection Timings

In this section, we attempted to optimise the particulate emission characteristics by using the split GDI strategy with different second injection timings.

Figure 9 shows the N_p size distribution with varying t_{GDI2} . It can be found that under the conditions of $t_{GDI2} = -260$ °CA and $t_{GDI2} = -220$ °CA, the peak values of the N_p size distribution curve are each 1.01×10^6 and 1.28×10^6 . These values are relatively lower than the curve's peak of the single injection conditions under $P_{GDI} = 5.5$ MPa, which is 1.42×10^6 , as shown in Figure 3. With the delay in t_{GDI2} , the curve's peak presents a growing tendency, rising to 1.43×10^6 and 1.62×10^6 under t_{GDI2} of -180 °CA and -140 °CA, respectively.

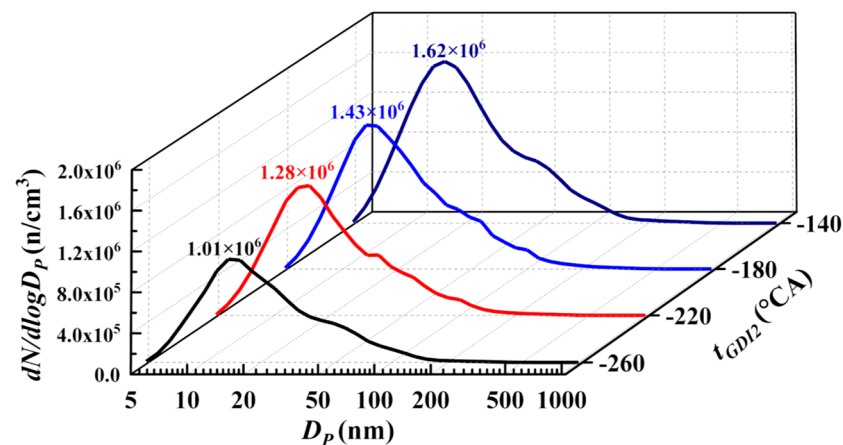


Figure 9. N_p size distribution with varying t_{GDI2} .

Figures 10 and 11 present the effects of t_{GDI2} on N_p and M_p concentrations of nucleation, accumulation, and total particulates. Based on the presented information, two key features can be summarised as follows. First, under $t_{GDI2} = -260$ °CA, the N_p concentration and M_p concentration of total particulates are each 6.66×10^5 and 56.79, which are much lower than the corresponding values (9.89×10^5 and 109.92) of the single injection conditions under $P_{GDI} = 5.5$ MPa, as shown in Figures 4 and 5. Second, by delaying t_{GDI2} from -260 °CA to -140 °CA, the N_p concentration of nucleation, accumulation, and total particulates shows an increase of 65.17%, 65.88%, and 62.89%, respectively. Simultaneously, a deterioration can also be observed in the M_p concentration, which has a maximum growth of more than 60%.

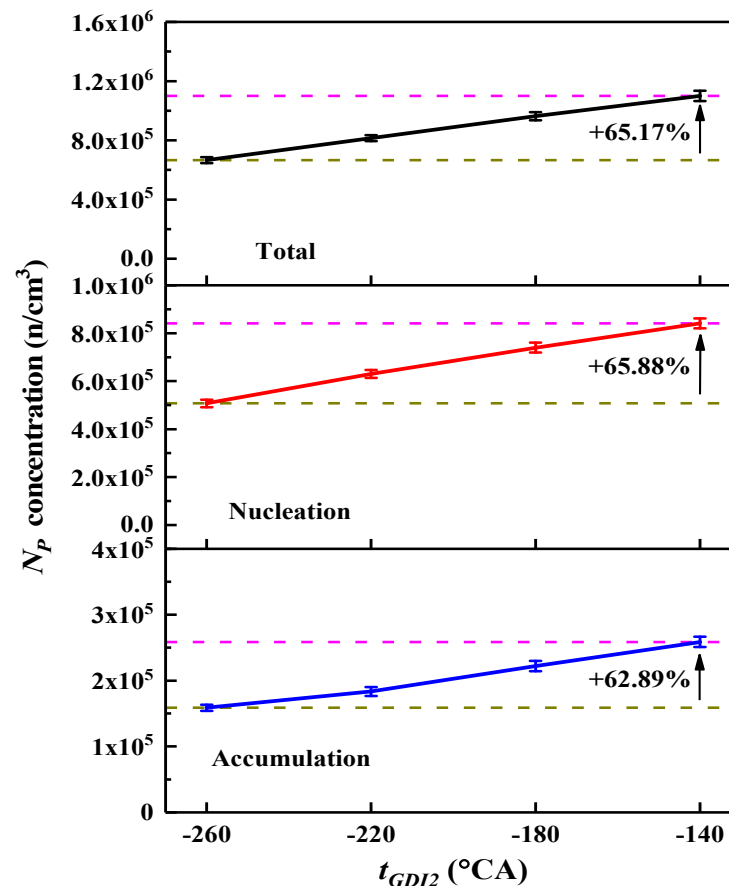


Figure 10. N_p concentration of nucleation, accumulation, and total particulates with varying t_{GDI2} .

This means that within an appropriate range of t_{GDI2} , the split GDI strategy could effectively help reduce particulate emissions from engines. This is mainly because when applying the split GDI strategy, the mass of injected spray each time is around half of only a single injection. Thus, the occurrence probability of spray impingement can be significantly decreased and almost minimised due to the reduced jet penetration length. As a result, this would contribute to improving the homogeneity of the air–fuel mixture, which is good for mitigating PM emissions. However, due to the shorter duration of air–fuel mixture, a delayed t_{GDI2} could result in a relatively poor-quality air–fuel mixture, raising the probability of particulate formation.

The variation in particulate emissions from changing the second injection timing of the split GDI strategy can be further explained by the combustion performance characterised by θ_F , T_{MI} , and T_{EG} , as shown in Figures 12–14.

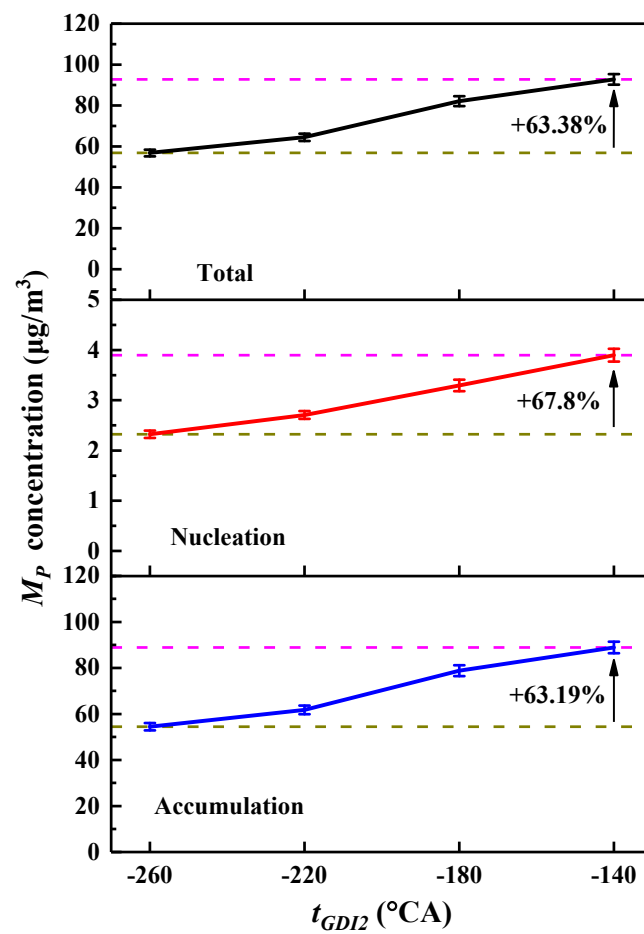


Figure 11. M_P concentration of nucleation, accumulation, and total particulates with varying t_{GDI2} .

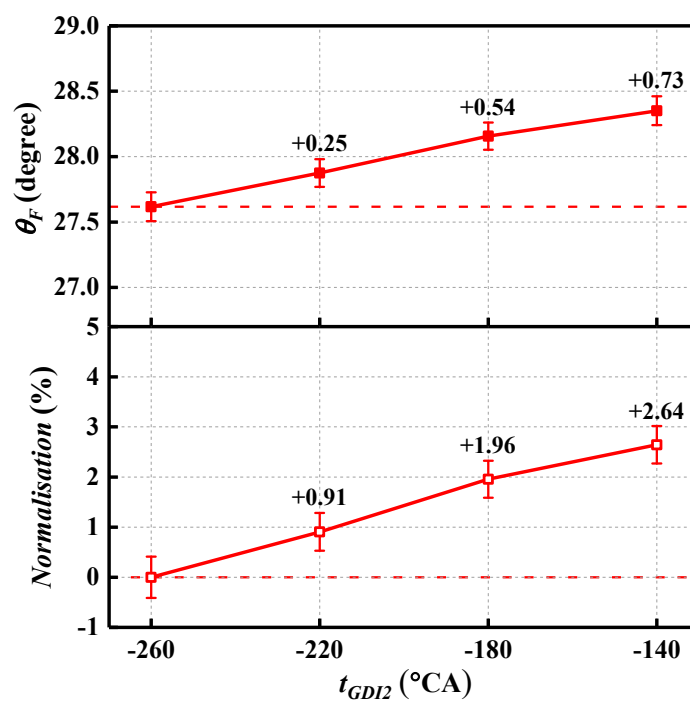


Figure 12. Effects of t_{GDI2} on θ_F .

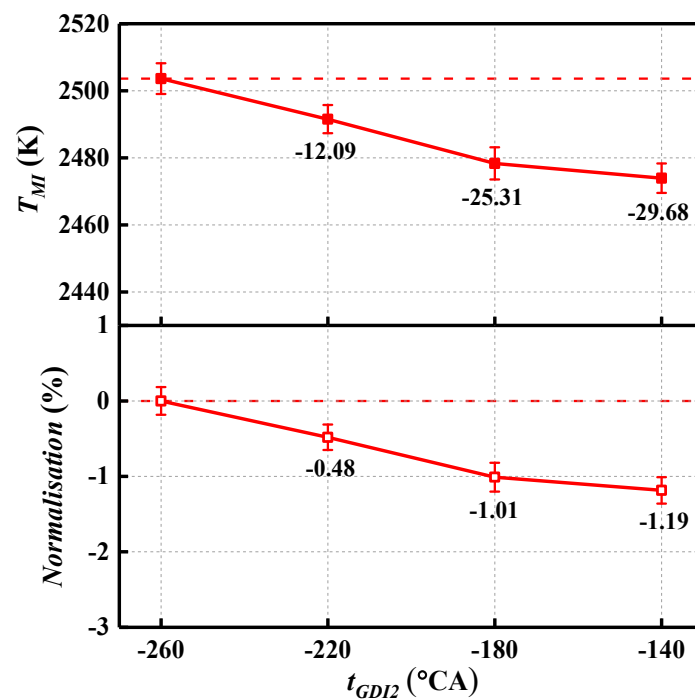


Figure 13. Effects of t_{GDI2} on T_{MI} .

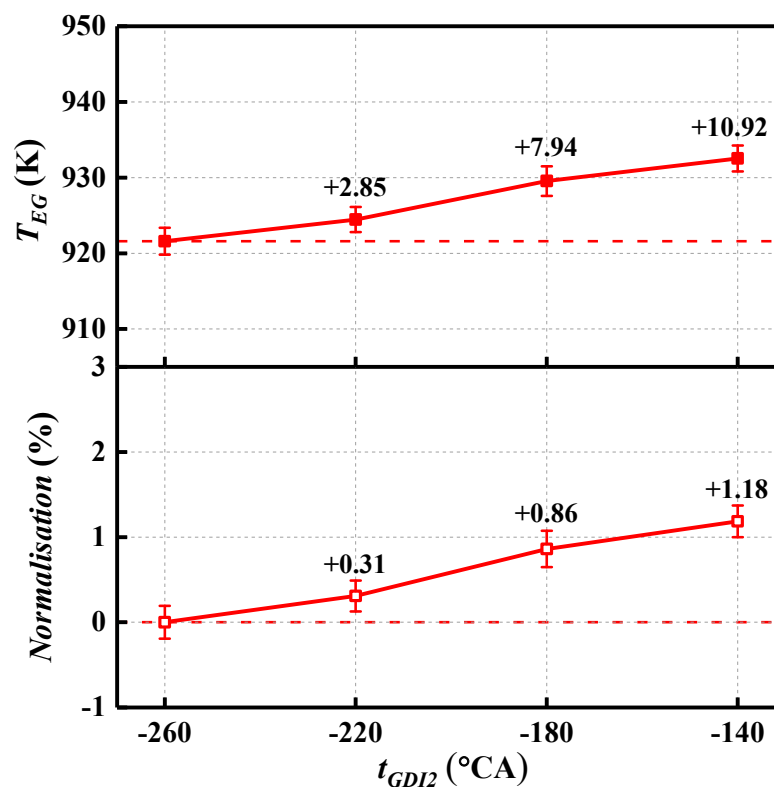


Figure 14. Effects of t_{GDI2} on T_{EG} .

Figure 12 shows that with the delay of t_{GDI2} from -260 °CA to -220 °CA, θ_F has a slight increase of 0.25 degrees, which is 0.91% in percentage. Then, a maximum increase of 0.73 degrees or 2.64% can be found by further delaying t_{GDI2} to -140 °CA. Simultaneously, a maximum reduction in T_{MI} can also be observed, which is 29.68 K or 1.19%. Thus, it indicates that combustion speed can be negatively impacted, leading to a more gradual and

mild combustion process, potentially expanding the unburned area, and then increasing the possibility of particulate formation. On the other side, T_{EG} shows a steady rise with the delay of t_{GDI2} , as depicted in Figure 14. Under $t_{GDI2} = -260$ °CA, T_{EG} is 921.61 K, while it has a growth to 932.53 K when t_{GDI2} postpones to -140 °CA. Although a higher T_{EG} would help promote the oxidation–reduction rate of particulate, its positive effects on emissions would be quite modest compared to the major disadvantages of the deterioration of air–fuel mixture and combustion qualities.

4. Conclusions

Experimental research was conducted to explore and analyse the particulate emission characteristics in a DFIG engine with EPI plus GDI mode by changing the GDI pressure or applying a split GDI strategy with different second injection timings. The research findings help offer new and valuable insights into the optimisation of particulate number and mass emissions from DFIG engines. The main conclusions can be seen as follows.

- (1) By increasing P_{GDI} from 5.5 MPa to 18 MPa, θ_F has a moderate reduction of 0.68 degrees, while T_{MI} and T_{EG} show a growth of 53.75 K and 13.84 K, respectively.
- (2) The peak value of the N_P size distribution curve decreases from 1.42×10^6 to 5.89×10^5 with the increases in P_{GDI} from 5.5 MPa to 18 MPa.
- (3) Positive impacts on the concentrations of N_P and M_P can be provided by increasing P_{GDI} . Compared to the condition of $P_{GDI} = 5.5$ MPa, the reduction in N_P and M_P for total particulates can be 59.5% and 36.26% under $P_{GDI} = 18$ MPa.
- (4) By delaying t_{GDI2} from -260 °CA to -140 °CA, θ_F increases by 0.73 degrees. Further, T_{EG} shows a steady rise from 921.61 K to 932.53 K, while a maximum reduction of 29.68 K in T_{MI} can be observed.
- (5) A split GDI strategy can be an effective method for reducing the particulate emissions of DFIG engines by setting t_{GDI2} within an appropriate range. Under $t_{GDI2} = -260$ °CA, the N_P and M_P concentrations of total particulates are much lower than the corresponding values of the single injection conditions under the same P_{GDI} of 5.5 MPa.
- (6) With the delay in t_{GDI2} , the magnitude of the N_P size distribution curve shows a growing tendency. N_P concentrations of nucleation, accumulation, and total particulates present an increase of 65.17%, 65.88%, and 62.89%, respectively.

Author Contributions: X.L.: conceptualization, methodology, formal analysis, investigation, visualization, writing—original draft. S.L.: visualization. W.L.: formal analysis, visualization. Y.P.: methodology, project administration, funding acquisition. X.Z.: visualization. P.N.: formal analysis. Z.P.: conceptualization, writing—reviewing and editing. C.W.: Formal analysis. All authors have read and agreed to the published version of the manuscript.

Funding: This work was supported by the National Key Technology R&D Program of China (No. 2014BAG10B01), and the National Engineering Laboratory for Mobile Source Emission Control Technology (No. NELMS2017C01).

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Data will be made available on request.

Conflicts of Interest: Siyue Liu was employed by the BYD Co., Ltd. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

Nomenclature

BMEP	Brake Mean Effective Pressure
CA	Crank Angle
CO	Carbon monoxide

DFSI	Dual-Fuel Spark Ignition
ECU	Electronic Control Unit
EDI	Ethanol Direct Injection
EPI	Ethanol Port Injection
GDI	Gasoline Direct Injection
GPI	Gasoline Port Injection
HC	Hydrocarbon
ICE	Internal Combustion Engine
MBT	Maximum Brake Torque
NO	Nitric Oxide
WOT	Wide Open Throttle
rpm	revolutions per minute
D_p	particulate diameter
M_p	particulate mass
N_p	particulate number
P_{GDI}	Gasoline Port Injection Pressure
T_{EG}	exhaust gas temperature
T_{MI}	maximum in-cylinder temperature
t_{GDI2}	injection timing of the second stage in the split GDI strategy
θ_F	ignition delay
φ_{CA10}	the CA where 10% of cumulative heat has been released

References

- Pan, M.; Huang, R.; Liao, J.; Ouyang, T.; Zheng, Z.; Lv, D.; Huang, H. Effect of EGR dilution on combustion, performance and emission characteristics of a diesel engine fueled with n-pentanol and 2-ethylhexyl nitrate additive. *Energy Convers. Manag.* **2018**, *176*, 246–255. [\[CrossRef\]](#)
- Huang, Y.; Lee CK, C.; Yam, Y.S.; Mok, W.C.; Zhou, J.L.; Zhuang, Y.; Surawski, N.C.; Organ, B.; Chan, E.F.C. Rapid detection of high-emitting vehicles by on-road remote sensing technology improves urban air quality. *Sci. Adv.* **2022**, *8*, eabl7575. [\[CrossRef\]](#)
- Pfau, S.A.; La Rocca, A.; Haffner-Staton, E.; Fay, M.W.; Cairns, A. Linking operating conditions of a GDI engine to the nature and nanostructure of ultrafine soot particles. *Combust. Flame* **2022**, *245*, 112315. [\[CrossRef\]](#)
- Huang, H.; Chen, R.; Tao, S.; Wang, Y.; Zhang, L. Study of soot dynamic behavior and catalytic regeneration in diesel particulate filters. *Chem. Eng. J.* **2024**, *489*, 151498. [\[CrossRef\]](#)
- Li, L.; Guo, S.; Cai, H.; Wang, J.; Zhang, J.; Ni, Y. Can China's BEV market sustain without government subsidies?: An explanation using cues utilization theory. *J. Clean. Prod.* **2020**, *272*, 122589. [\[CrossRef\]](#)
- İnci, M.; Büyüç, M.; Demir, M.H.; İlbel, G. A review and research on fuel cell electric vehicles: Topologies, power electronic converters, energy management methods, technical challenges, marketing and future aspects. *Renew. Sustain. Energy Rev.* **2021**, *137*, 110648. [\[CrossRef\]](#)
- Chai, W.S.; Bao, Y.; Jin, P.; Tang, G.; Zhou, L. A review on ammonia, ammonia-hydrogen and ammonia-methane fuels. *Renew. Sustain. Energy Rev.* **2021**, *147*, 111254. [\[CrossRef\]](#)
- Gao, J.; Wang, X.; Song, P.; Tian, G.; Ma, C. Review of the backfire occurrences and control strategies for port hydrogen injection internal combustion engines. *Fuel* **2022**, *307*, 121553. [\[CrossRef\]](#)
- Duan, X.; Xu, L.; Xu, L.; Jiang, P.; Gan, T.; Liu, H.; Ye, S.; Sun, Z. Performance analysis and comparison of the spark ignition engine fuelled with industrial by-product hydrogen and gasoline. *J. Clean. Prod.* **2023**, *424*, 138899. [\[CrossRef\]](#)
- Zhuang, Y.; Zhu, G.; Gong, Z.; Wang, C.; Huang, Y. Experimental and numerical investigation of performance of an ethanol-gasoline dual-injection engine. *Energy* **2019**, *186*, 115835. [\[CrossRef\]](#)
- Zhao, Z.; Yu, X.; Huang, Y.; Shi, W.; Guo, Z.; Li, Z.; Du, Y.; Jin, Z.; Li, D.; Wang, T.; et al. Experimental study on combustion and emission of an SI engine with ethanol/gasoline combined injection and EGR. *J. Clean. Prod.* **2022**, *331*, 129903. [\[CrossRef\]](#)
- Li, X.; Li, D.; Liu, J.; Ajmal, T.; Aitouche, A.; Mobasheri, R.; Rybdylova, O.; Pei, Y.; Peng, Z. Exploring the potential benefits of Ethanol Direct Injection (EDI) timing and pressure on particulate emission characteristics in a Dual-Fuel Spark Ignition (DFSI) engine. *J. Clean. Prod.* **2022**, *357*, 131938. [\[CrossRef\]](#)
- Kim, N.; Cho, S.; Min, K. A study on the combustion and emission characteristics of an SI engine under full load conditions with ethanol port injection and gasoline direct injection. *Fuel* **2015**, *158*, 725–732. [\[CrossRef\]](#)
- Liu, H.; Wang, Z.; Long, Y.; Xiang, S.; Wang, J.; Fatouraie, M. Comparative study on alcohol–gasoline and gasoline–alcohol Dual-Fuel Spark Ignition (DFSI) combustion for engine particle number (PN) reduction. *Fuel* **2015**, *159*, 250–258. [\[CrossRef\]](#)
- Kang, R.; Zhou, L.; Hua, J.; Feng, D.; Wei, H.; Chen, R. Experimental investigation on combustion characteristics in dual-fuel dual-injection engine. *Energy Convers. Manag.* **2019**, *181*, 15–25. [\[CrossRef\]](#)
- Yu, X.; Guo, Z.; Sun, P.; Wang, S.; Li, A.; Yang, H.; Li, Z.; Liu, Z.; Li, J.; Shang, Z. Investigation of combustion and emissions of an SI engine with ethanol port injection and gasoline direct injection under lean burn conditions. *Energy* **2019**, *189*, 116231. [\[CrossRef\]](#)

17. Wang, C.; Pei, Y.; Peng, Z.; Li, X. Insights into the effects of direct injection timing on the characteristics of deposited fuel film and Particulate Matter (PM) emissions from a Dual-Fuel Spark Ignition (DFSI) engine. *Fuel Process. Technol.* **2022**, *238*, 107515. [\[CrossRef\]](#)
18. Lee, Z.; Park, S. Particulate and gaseous emissions from a direct-injection spark ignition engine fueled with bioethanol and gasoline blends at ultra-high injection pressure. *Renew. Energy* **2020**, *149*, 80–90. [\[CrossRef\]](#)
19. Chang, F.; Luo, H.; Zhan, C.; Nishida, K.; Ogata, Y. Droplets velocity and diameter variations of wall impinging spray created by slicer. *Fuel* **2021**, *299*, 120894. [\[CrossRef\]](#)
20. Ramalingam, S.; Babu, D.; Santhoshkumar, A.; Deepakkumar, R.; Ravikanth, D. Impact of exhaust gas recirculation and split injection strategy combustion behavior on premixed charge compression ignition engine fuelled with moringa oleifera methyl ester. *Fuel* **2022**, *319*, 123702. [\[CrossRef\]](#)
21. Kumar, T.S.; Ashok, B.; Kumar, M.S.; Vignesh, R.; Saiteja, P.; Hire, K.R.B.; Tote, M.H.; Pandey, R.; Jadhav, A.; Gupta, A.; et al. Biofuel powered engine characteristics improvement through split injection parameter multivariate optimization with titanium based nano-particle additives. *Fuel* **2022**, *322*, 124178. [\[CrossRef\]](#)
22. Li, X.; Pei, Y.; Ajmal, T.; Rana, K.-J.; Aitouche, A.; Mobasheri, R.; Peng, Z. Numerical investigation on implementing Oxy-Fuel Combustion (OFC) in an ethanol-gasoline Dual-Fuel Spark Ignition (DFSI) engine. *Fuel* **2021**, *302*, 121162. [\[CrossRef\]](#)
23. Liu, H.; Li, Z.; Xu, H.; Ma, X.; Shuai, S. Nucleation mode particle evolution in a gasoline direct injection engine with/without a three-way catalyst converter. *Appl. Energy* **2020**, *259*, 114211. [\[CrossRef\]](#)
24. Holman, J.P. *Experimental Methods for Engineers*, 8th ed.; McGraw-Hill: New York, NY, USA, 2021.
25. Chen, L.; Liang, Z.; Zhang, X.; Shuai, S. Characterizing particulate matter emissions from GDI and PFI vehicles under transient and cold start conditions. *Fuel* **2017**, *189*, 131–140. [\[CrossRef\]](#)
26. Li, X.; Zhang, X.; Ni, P.; Weerasinghe, R.; Pei, Y.; Peng, Z. Insights into the spray impingement process from a gasoline direct injection fuel system fuelled with gasoline and ethanol. *J. Energy Inst.* **2023**, *110*, 101331. [\[CrossRef\]](#)
27. Richter, H.; Howard, J.B. Formation of polycyclic aromatic hydrocarbons and their growth to soot—A review of chemical reaction pathways. *Prog. Energy Combust. Sci.* **2000**, *26*, 565–608. [\[CrossRef\]](#)
28. Jin, H.; Yuan, W.; Li, W.; Yang, J.; Zhou, Z.; Zhao, L.; Li, Y.; Qi, F. Combustion chemistry of aromatic hydrocarbons. *Prog. Energy Combust. Sci.* **2023**, *96*, 101076. [\[CrossRef\]](#)

Disclaimer/Publisher’s Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.