Dual injection: An effective and efficient technology to use renewable fuels in spark ignition engines

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Please cite this article as:

Y. Huang, N.C. Surawski, Y. Zhuang, J.L. Zhou, G. Hong. Dual injection: An effective and efficient technology to use renewable fuels in spark ignition engines. ***Renewable and Sustainable Energy Reviews*** 2021; 143: 110921. https://doi.org/10.1016/j.rser.2021.110921

Abstract

Modern spark ignition engines mostly use one injection system to deliver gasoline into the combustion chamber, using either direct injection or port fuel injection. Both technologies have their respective advantages. To integrate their advantages and to promote the use of renewable fuels, dual injection engines are in development in recent years. Dual injection represents an advanced combustion system and is a novel technology to address the urgent issues of sustainability and environmental protection. This study reviews the state-of-the-art research on dual injection spark ignition engines with a focus on renewable fuels, their advantages and engine performance. The main advantages of dual injection include greater control flexibility, enhanced cooling effect, knock mitigation, engine downsizing, extended lean-burn limits, higher thermal efficiency and reductions of several emission species. The most promising renewable fuels for dual injection are ethanol, methanol and hydrogen. Each renewable fuel is aimed at different advantages of dual injection. Alcohol-gasoline dual injection provides great anti-knock ability by taking advantage of alcohols' large enthalpies of vaporisation and high octane numbers, while hydrogen-gasoline dual injection provides extended lean-burn limits by taking advantage of hydrogen's low ignition energy, wide flammability limit and high flame speed. Direct injection of renewable fuels is the optimal injection strategy because it effectively utilises the strong cooling effect of alcohols or avoids the volumetric efficiency reduction and pre-ignition of hydrogen. Dual injection generally demonstrates higher thermal efficiency than single injection. In addition, dual injection effectively reduces particulate emissions while there are usually trade-offs among gaseous emissions.

Keywords: Dual injection; Renewable fuels; Spark ignition engines; Thermal efficiency; Combustion; Emissions

Highlights

- Dual injection provides great control flexibility by integrating advantages of DI and PFI.
- Alcohols and hydrogen enhance knock mitigation and lean-burn limits respectively.
- Renewables DI plus gasoline PFI is the optimal strategy for both alcohols and hydrogen.
- Dual injection increases thermal efficiency and reduces certain emission products.
- Dual injection is an effective and efficient technology to use renewable fuels in SI engines.

Abbreviations and symbols

ABE	Acetone-butanol-ethanol blend
AFR	Air fuel ratio
BTDC	Before top dead centre
CDC	Conventional diesel combustion
CFD	Computational fluid dynamics
CI	Compression ignition
DDFS	Direct dual fuel stratification
DFSC	Dual fuel sequential combustion
DI	Direct injection
DME	Dimethyl ether
DMF	2, 5-dimethylfuran
EGR	Exhaust gas recirculation
GDI/EDI/MDI/HDI	Gasoline/ethanol/methanol/hydrogen direct injection
GPI/EPI/MPI/HPI	Gasoline/ethanol/methanol/hydrogen port injection
ICCI	Intelligent charge combustion ignition
IMEP	Indicated mean effective pressure
KLST	Knock limited spark timing
LHV	Lower heating value
LTC	Low temperature combustion
MBT	Minimum spark advance for best torque
NEDC	New European Driving Cycle
NMHC	Non-methane hydrocarbons
PEMS	Portable emission measurement system
PFI	Port fuel injection
PM	Particulate matter
PN	Particle number
RCCI	Reactivity controlled compression ignition
RDE	Real driving emissions
RON	Research octane number
RVP	Reid vapour pressure
SI	Spark ignition
THC	Total hydrocarbons
TWC	Three-way catalyst
WLTP	Worldwide harmonised Light vehicle Test Procedure
WOT	Wide open throttle
λ	Excess air ratio

1. Introduction

Modern economies rely heavily on the transport of goods and people, which are and will be largely powered by internal combustion engines in the next few decades [1-3]. The global numbers of cars and trucks were around 1.1 billion and 377 million in 2015, which are projected to reach 2.0 billion and 790 million in 2040, respectively [4]. Passenger cars are mostly (>80%) powered by spark ignition (SI) engines worldwide, except for the European Union, India and South Korea markets where compression ignition (CI) engines have a significant share (39%-52%) [5, 6]. Rapid increases in the number of vehicles consume significant amounts of fossil fuels and emit a large percentage of total greenhouse gas emissions [7]. The International Energy Agency estimated that global energy consumption more than doubled during 1971 to 2015, and the percentage of energy use by the transport sector increased noticeably from 23% to 29% during the same period while other sectors mostly did not change [8]. Vehicle engines are also a major contributor to urban air pollution, posing a serious health hazard to the public [9, 10]. A recent study estimated that vehicle tailpipe emissions caused 385000 premature deaths and US$1 trillion of health damage worldwide in 2015 [11].

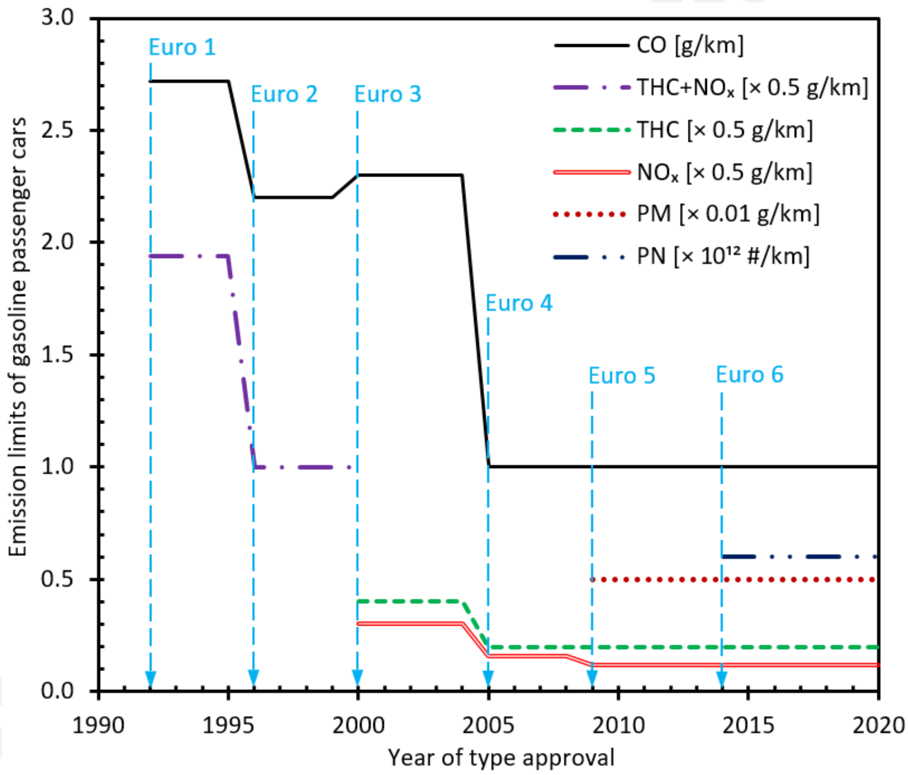

Fig. 1. Advancement of the European automotive emission standards for gasoline passenger cars. A non-methane hydrocarbons (NMHC) limit of 68 mg/km is introduced in addition to the total hydrocarbons (THC) limit since Euro 5 [12]. The particulate matter (PM) limit only applies to vehicles with DI engines and a limit of 4.5 mg/km is applied when using a revised measurement procedure (i.e. PMP) [12]. The particle number (PN) limit only applied to vehicles with DI engines and a limit of 6×10^{12} #/km was used in the first three years of Euro 6 [13].

Regulations are becoming increasingly stringent to reduce both the air pollutant and greenhouse gas emissions. **Fig. 1** demonstrates the advancement of emission limits for gasoline passenger cars from Euro 1 to 6. The limit values have been reduced substantially and more pollutant species have been regulated during the past three decades. For instance, the NO_x limit has been reduced by 60% from 0.15 g/km in Euro 3 [14] to 0.06 g/km in Euro 5 and 6 [12, 13]. Although the limit values are unchanged after Euro 5, the emission testing methods have become more stringent in Euro 6. Firstly, a Worldwide harmonised Light vehicles Test Procedure (WLTP) is introduced for type approval of new vehicles using chassis dynamometers, which is more representative of real-world driving than the outdated New European Driving Cycle (NEDC) [15, 16]. In addition to the laboratory based WLTP, a Real Driving Emissions (RDE) test procedure has been introduced to measure vehicle emissions in the real world using a Portable Emission Measurement System (PEMS) [15]. The RDE test aims to reduce the significant discrepancy between the laboratory and real-world performance of emissions and fuel consumption [17, 18]. An initial conformity factor of 2.1 was used in 2017, with the aim of reducing it to 1.0 as soon as possible and at the latest by 2023 [19]. Regarding regulations on greenhouse gas emissions, the European Union has tightened its fleet-wide average emission target from 130 g CO_2/km in 2015 to 95 g CO_2/km in 2021 which corresponds to a fuel consumption of 4.1 L/100 km for gasoline cars [20].

To meet the ever stricter regulations on emissions and fuel economy, significant efforts have been taken to improve engine combustion system and to search for renewable fuels. For SI engines, it is critical for the fuel to mix with the intake air and form a suitable mixture before the electrical discharge from the spark plug is initiated. There are mainly three fuel injection technologies, namely carburettor, port fuel injection (PFI) and direct injection (DI) [21, 22]. PFI replaced carburettors in the 1980s due to its advantages in fuel saving via more precise control of fuel injection and emission reductions via exhaust after-treatment using a three-way catalyst (TWC). DI was developed in the 1990s and offers further advantages in fuel saving when compared with PFI. In spite of this, DI has not fully replaced PFI in modern SI engines. **Table 1** shows the fuel injection technologies of the top 20 most popular car models in 2019 in Australia. As shown in **Table 1**, PFI and DI have similar market shares (14 vs 16 engine models) although DI is considered more advanced than PFI. This is because either of these two fuel injection technologies has its respective advantages and limitations, which will be discussed in detail in **Section 3**. This leads to a novel idea of using DI and PFI simultaneously in one engine (i.e. dual injection), which has the potential to integrate their advantages while avoiding their drawbacks. Dual injection offers greater flexibility in controlling mixture formation and combustion processes and is a promising technology to help achieve the ever stricter emissions and fuel efficiency standards. Dual injection has been in development in recent years and a few mass production cars have already adopted this concept, such as the Toyota RAV4, Toyota Camry and Volkswagen Golf. These vehicle models have shown advantages in engine downsizing (such as high compression ratios and turbocharging) and fuel economy compared with their competitors. In these dual injection engines, the same fuel (i.e. gasoline) is used for both PFI and DI.

Table 1. Engine specifications of the top 20 most popular gasoline cars in Australia in 2019.

No.	Car model	Sales*	Engine specifications†					
			Engine model code	Displacement (L)	Air intake system	Injection system	Compression ratio	Fuel consumption (L/100km)
1	Toyota Hilux	47649	2TR-FE	2.7	Aspirated	PFI	10.2:1	10.4-11.1
2	Ford Ranger	40690	No petrol model is available					
3	Toyota Corolla	30468	2ZR-FE	1.8	Aspirated	PFI	10.0:1	6.4-6.8
			2ZR-FXE	1.8	Aspirated	PFI	13.0:1	3.5-4.2 (hybrid)
			M20A-FKS	2.0	Aspirated	DI	13.0:1	6.0-6.5
4	Hyundai i30	28378	Gamma	1.6	Turbocharged	DI	9.5:1	7.1-7.5
			Nu	2.0	Aspirated	DI	11.5:1	7.3-7.4
			G4KH	2.0	Turbocharged	DI	9.5:1	8.0
5	Mitsubishi Triton	25819	4G64	2.4	Aspirated	PFI	9.0:1	11.4
6	Mazda CX-5	25539	PE-VPS	2.0	Aspirated	DI	13.0:1	6.9
			PY-VPR	2.5	Aspirated	DI	13.0:1	7.4
			PY-VPR	2.5	Turbocharged	DI	10.5:1	8.2
7	Mazda 3	24939	PE-VPS	2.0	Aspirated	DI	13.0:1	5.7-6.4
			PY-VPS	2.5	Aspirated	DI	13.0:1	6.0-6.6
8	Toyota RAV4	24260	M20A-FKS	2.0	Aspirated	DI+PFI	13.0:1	6.5-6.8
			A25A-FKS	2.5	Aspirated	DI+PFI	13.0:1	7.3
			A25A-FXS	2.5	Aspirated	DI+PFI	14.0:1	4.7-4.8 (hybrid)
9	Kia Cerato	21757	G4NA	2.0	Aspirated	PFI	10.3:1	7.4-7.6
			G4FJ	1.6	Turbocharged	DI	10.0:1	6.8
10	Mitsubishi ASX	20806	4B11	2.0	Aspirated	PFI	10.0:1	7.6-7.7
			4B12	2.4	Aspirated	PFI	10.5:1	7.9
11	Nissan X-Trail	19726	MR20DD	2.0	Aspirated	DI	11.2:1	8.2
			QR25DE	2.5	Aspirated	PFI	10.0:1	7.9-8.3
12	Toyota Landcruiser	18335	1UR-FE	4.6	Aspirated	PFI	10.2:1	13.4
13	Hyundai Tucson	18251	T-GDI	1.6	Turbocharged	DI	10.1:1	7.7
			2.0 GDI	2.0	Aspirated	DI	11.5:1	7.8-7.9
14	Mitsubishi Outlander	17514	4J11 MIVEC	2.0	Aspirated	PFI	10.5:1	7.0
			4B11 MIVEC	2.0	Aspirated	PFI	-	1.7 (plug-in hybrid)
			4J12 MIVEC	2.4	Aspirated	PFI	10.5:1	7.2
			4B12 MIVEC	2.4	Aspirated	PFI	12.0:1	1.9 (plug-in hybrid)
15	Holden Colorado	17472	No petrol model is available					
16	Isuzu D-Max	16892	No petrol model is available					
17	Toyota Camry	16768	2AR-FE	2.5	Aspirated	PFI	10.4:1	7.8-8.3
			A25A-FXS	2.5	Aspirated	DI+PFI	14.0:1	4.2-4.5 (hybrid)
			2GR-FKS	3.5	Aspirated	DI+PFI	11.8:1	8.7-8.9
18	Subaru Forester	15096	e-Boxer	2.0	Aspirated	DI	12.5:1	6.7 (hybrid)
			FB25	2.5	Aspirated	DI	12.0:1	7.4
19	Mazda CX-3	14813	SKYACTIV-G	2.0	Aspirated	DI	13.0:1	6.3-6.7
20	Volkswagen Golf	14355	CZDA	1.4	Turbocharged	DI	10.0:1	5.4-5.7
			CJSB	1.8	Turbocharged	DI+PFI	9.6:1	6.8
			DJHB	2.0	Turbocharged	DI+PFI	9.3:1	7.2-7.3
			CHHA	2.0	Turbocharged	DI+PFI	9.6:1	6.5

** The 2019 new car sales were from the Federal Chamber of Automotive Industries at https://www.caradvice.com.au/817278/vfacts-2019-new-car-sales-results <accessed 23.04.2020>*

† The engine specifications of 2019 car models were from https://www.redbook.com.au/ <accessed 23.04.2020>

The potential of dual injection in fuel saving and emissions reduction can be further enhanced when combined with renewable fuels by taking advantage of their fuel properties (**Table 2**), such as high octane number, increased cooling effect and wide flammability limit which will be discussed in **Section 4**. Renewable fuels are becoming increasingly important in combating global warming and fossil fuel depletion, among which ethanol is the most widely used alternative fuel for SI engines [23-25]. Currently, it is usually used by blending with gasoline (e.g. E10) to partially substitute fossil fuel due to its limited supply and compatibility with existing engines. However, blending renewable fuels with gasoline at fixed ratios would not achieve an optimal performance over the wide engine operating conditions. Thus the in-cylinder blending of gasoline and renewable fuels in the dual injection concept provides the flexibility to utilise renewable fuels more effectively and efficiently than pre-blending. Dual injection has great potential in improving the combustion performance of SI engines by changing the blending ratio and injection strategy according to the operating condition.

The dual injection concept is not new and has been investigated extensively in CI engines. Two dual injection configurations have been widely explored in CI engines, namely DI+PFI and DI+DI. Depending on the proportion of premixed fuel, dual injection CI engines can work in low temperature combustion (LTC) or conventional diesel combustion (CDC) mode, where LTC behaves more like a premixed flame while CDC behaves more like a diffusion flame. DI+PFI is usually studied in LTC mode such as reactivity controlled compression ignition (RCCI) [26-31] and dual fuel sequential combustion (DFSC) [32, 33], although it also works in CDC mode such as natural gas/hydrogen/alcohols PFI + diesel DI [34-36]. Meanwhile, DI+DI is studied in both LTC (e.g. intelligent charge combustion ignition (ICCI) [37, 38] and direct dual fuel stratification (DDFS) [39, 40]) and CDC (e.g. hydrogen-diesel [41] and methanol-diesel [42] dual DI) modes. When it comes to SI engines, however, dual injection is a relatively new concept and has attracted great attention in recent years. So far there is a lack of critical review in this area.

This paper aims to review the recent progress in dual injection of gasoline and renewable fuels in SI engines. The dual injection concept here specifically refers to the combination of DI and PFI which has attracted the most research attention, although there are a few studies on dual PFI [43-49] and dual DI [50, 51]. This paper is organised as follows. Firstly, the properties, benefits and challenges of the key suitable renewable fuels for dual injection SI engines are introduced. Then the mechanisms, advantages and disadvantages of DI and PFI are examined. Following that, the main advantages of dual injection concept in integrating DI and PFI's advantages and promoting the use of renewable fuels in SI engines are discussed. Finally, the combustion and emissions performance of various dual injection SI engines are reviewed.

2. Renewable fuels for SI engines

Properties of renewable fuels that are compatible with the dual injection concept in SI engines are compared with gasoline in **Table 2**. The alternative fuels proposed can be broadly broken into gaseous fuels, alcohols, oxygenates and furans. All of these alternative fuels have been thoroughly investigated in automotive applications, with the exception of furans which have more recently attracted research attention due to the development of catalytic processes that may lead to scalable production capacity from biomass [52, 53]. Although the production of several of these fuels could be achieved with fossil fuels such as coal and oil, in this work we wish to emphasise the opportunity that exists with producing all of these alternative fuels renewably. In **Table 2**, we also overview a range of parameters that will determine the performance of each fuel in an SI engine application. This includes both chemical and physical properties that will dictate the charge cooling effect, volatility, spray, ignition and combustion dynamics.

Out of the many fuel properties that dictate SI engine performance, arguably, the most important two are the research octane number (RON) and volatility characteristics. RON captures the knock resistance of a fuel, with higher values enabling the engine to run with a higher compression ratio which improves power output and thermal efficiency. As shown in **Table 2**, all renewable fuels exhibit a larger RON than gasoline. Out of these fuels, hydrogen has the highest RON by a considerable margin. The driveability of a vehicle is influenced by a fuel's volatility which is partially captured by the Reid vapour pressure (RVP). The renewable fuels overviewed in **Table 2** vary widely in RVP, with higher alcohols having the lowest RVP and gaseous fuels showing the highest RVP. Apart from the central importance of RON and RVP, several other fuel properties are essential for good combustion performance.

From an emissions perspective, one of the key benefits of fuels containing oxygen (such as alcohols, ethers and furans) is their demonstrated role in reducing CO and HC emissions. Despite this, oxygenated fuels have reduced lower heating values (LHV) which can be considered a disadvantage. On the other hand, furans have energy densities that are more comparable with gasoline and generally have fuel properties that are quite similar, making them a fuel type with great "drop in" potential in SI engines. Oxygenated fuels such as alcohols also have higher enthalpies of vaporisation which can increase the charge cooling effect and intake air density. The charge cooling effect also plays a role in reducing the flame temperature which leads to NO_x reduction. On the contrary, the lower RVP and higher enthalpy of vaporisation of alcohols can lead to engine cold start issues.

In terms of fuel chemical composition, the only carbon free fuel is hydrogen. Benefits of using hydrogen as a transport fuel include its increased LHV, wide flammability limit, and its considerable RON. Owing to its wide flammability limit, hydrogen mixtures can be combusted in an overly lean manner for improved thermal efficiency compared to the stoichiometric mixtures usually applied in SI engines. In spite of these advantages, there are safety matters to consider such as invisible flames during the combustion process and the possibility of flashback into the intake port due to hydrogen's high laminar flame velocity.

Table 2. Physico-chemical properties of renewable fuels and gasoline that are compatible with the dual injection concept in SI engines.

Properties	Gasoline	Hydrogen	Methanol	Ethanol	Iso-Butanol	n-Butanol	Methane	MTBE	ETBE	MF	DMF
Chemical formula	~ C_4-C_{12}	H_2	CH_3OH	C_2H_5OH	C_4H_9OH	C_4H_9OH	CH_4	$C_5H_{12}O$	$C_6H_{14}O$	C_5H_6O	C_6H_8O
Molar mass (g/mol)	58-170	2	32	46	74	74	16	88	102	82	96
H/C ratio	1.7-1.9	Carbon free	4	3	2.5	2.5	4	2.4	2.33	1.2	1.33
Oxygen content (%w)	0	0	50	34.8	21.6	21.6	0	18.2	15.7	19.5	16.7
Density @ 20 °C (g/cm^3)	0.72-0.78	0.0013*[54]	0.796	0.798	0.801	0.810	0.72[55]	0.735	0.742	0.913	0.890
Dynamic viscosity (mPa·s)	0.37-0.44	0.009[54]	0.6	1.5	8.3	3.6	0.01[54]	0.31	0.53	0.4	0.53
Boiling point (°C)	25-210	-253[54]	65	78	108	118	-162[54]	55	73	65	92
Auto-ignition temperature (°C)	257	572[56]	385	363	415	343	540[54]	374	375	450	286[57]
Ignition energy (mJ)	0.25[56]	0.018[58]	0.14[58]	0.23[59]	NA	0.6[60]	0.28[58]	NA	NA	0.225^[60]	NA
RVP @ 37 °C (kPa)	54-103	NA	32	16	3.3	2.2	NA	32	30	18.5[61]	13.4[61]
LHV (MJ/kg)	41-44	120[54]	19.7	26.8	33.1	33.2	50[55]	38.2	36	31.2	32.9
Enthalpy of vaporisation (kJ/kg)	373	448[54]	1110	912	566	584	510[54]	340	323	358	332
RON	88-98	120-140[55]	109	109	105	98	120[55]	117	118	103	101
Flammability limits (%)	1.4-7.6[54]	4-75[54]	7.3-36[54]	4.3-19[54]	1.2-10.9[62]	1.4-11.2[62]	5-15[54]	1.5-8.5[63]	1.2-7.7[64]	1.9-14[65]	NA
Stoichiometric AFR	14.7	34.3[54]	6.4	9	11.1	11.1	17.2[55]	11.7	12.2	10.1	10.7
Laminar flame speed (m/s)	0.37-0.43[66]	1.85[66]	0.56[67]	0.39[66]	0.45[68]	0.48[68]	0.38[66]	0.35[69]	0.30[70]	0.47[71]	0.38[72]
Adiabatic flame temperature @ 1 bar, 20 °C, λ=1	2346	2377[73]	2216	2310	2372	2388	2222[73]	2623	2399	2535	2509

Notes: *MTBE, methyl tert-butyl ether; ETBE, ethyl tert-butyl ether; DMF: 2,5-dimethylfuran; MF, 2-methylfuran; NA, not applicable/available; * @ 15 °C; ^ Furan; All remaining data are from [74].*

3. Fuel injection technologies of SI engines

Fuel injection systems of modern SI engines are dominated by DI and PFI, as shown in **Table 1**. In a PFI system (**Fig. 2a**), fuel is metered by a nozzle and sprayed into the intake manifold, which is carried into cylinders by the intake air [75]. There are two types of PFI, namely single- and multi-point injection. Single-point injection (e.g. throttle body injection) injects fuel at the main intake manifold which serves all the cylinders, while multi-point injection injects fuel at the back of intake valve(s) of each cylinder. Multi-point injection distributes fuel more evenly and precisely among cylinders than single-point injection, and thus is the dominant type in modern PFI engines. In a DI system (**Fig. 2b**), fuel is sprayed into the engine combustion chamber directly via a high pressure injector. DI engines are designed in two combustion modes, namely homogeneous stoichiometric combustion and stratified lean combustion [76]. The homogeneous mode injects fuel during early intake stroke to form a homogenous stoichiometric mixture by the time of spark discharge. The stratified mode injects fuel later to form an ignitable mixture in the regions around the spark plug but a lean mixture in other regions, which can be realised by either a wall-, air- or spray-guided mixing process. Although the stratified mode has higher fuel efficiency, the conversion efficiency of the TWC is low. As a result, current DI engines mostly adopt the homogenous stoichiometric combustion mode.

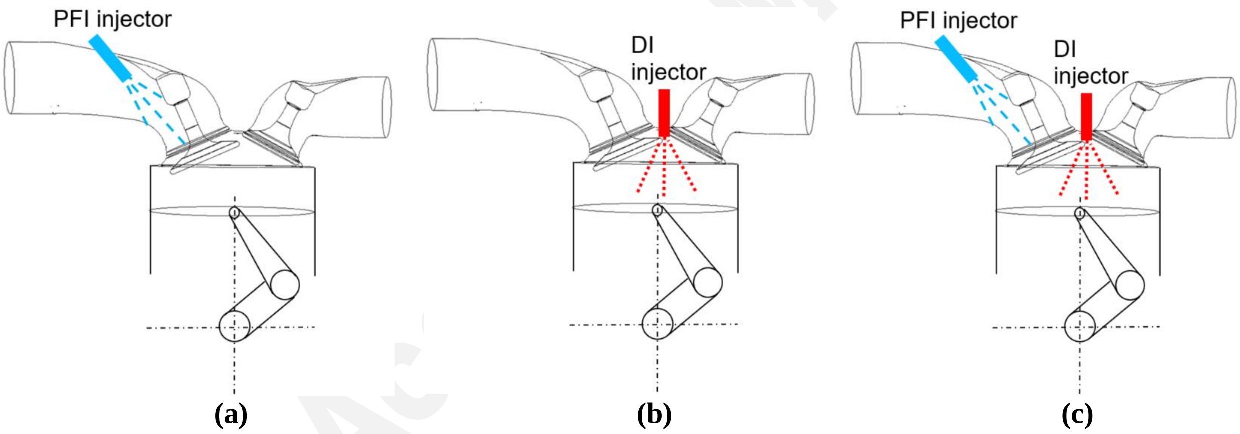

Fig. 2. Schematics of PFI (a), DI (b) and dual injection (c) systems.

Table 3 compares the advantages and disadvantages of PFI and DI systems [22, 76, 77]. PFI usually starts injection before the intake valves open and fuel films are formed on the surfaces of intake ports and valves. The liquid fuel absorbs heat from the hot surfaces and evaporates partially before being entrained into the engine combustion chamber by the intake air. Therefore, PFI engines have sufficient time for fuel evaporation and mixing, and low pressure injection (of a few bar) is applied. The homogenous stoichiometric combustion of PFI engines ensures a high TWC conversion efficiency. In addition, the good evaporation and mixing processes reduce the formation of particulate emissions. On the other hand, port wall wetting causes a time lag between the injection and fuel delivery into cylinders, leading to metering errors and slow transient responses. Particularly, over-fuelling is required for cold start when intake port is cold, which results in worsened fuel economy and increased HC emissions.

DI engines inject fuel directly into each cylinder, therefore time lag is eliminated and transient response is improved. Fuel evaporation inside the combustion chamber cools the air, which increases volumetric efficiency and reduces knock propensity. Thus a higher compression ratio can be achieved to increase the thermal efficiency. DI technology also offers an extended exhaust gas recirculation (EGR) tolerance limit and enhanced potential for system optimisation [22]. However, all these benefits come at a cost. Since the time allowed for fuel evaporation and mixing is reduced, high pressure injection (up to 20 MPa) is required to generate fine spray droplets. High pressure fuel injection and mixture formation have higher control complexity over a wide engine operating range, and fuel impingement may occur on the cylinder and piston walls. Unevaporated liquid fuel droplets by spark timing will participate in diffusion burning and thus produce high particulate emissions like diesel engines. Saliba et al. [78] investigated the gaseous and particulate emissions performance of 15 DI and 67 PFI light-duty petrol vehicles. The results demonstrated insignificant difference of regulated gaseous emissions between PFI and DI vehicles. However, the particulate emissions of DI vehicles were higher by a factor of two than PFI vehicles. Higher particulate emissions of DI engines were also reported in [79-83] which tested smaller sample sizes. The latest emission regulations such as Euro 6 and China 6 have enforced PM and PN limits, and thus DI engines will need to adopt gasoline particulate filters to meet the new regulations [84, 85]. Finally, injector deposits and ignition fouling are also challenges in DI engines. Injector deposits are formed due to the exposure to high-temperature combustion environment, which can reduce injector fuel flow rates and alter the designed spray characteristics [86]. Ignition fouling is caused by fuel impingement on the electrodes of spark plug due to the close spacing between injector and spark plug [22].

Table 3. Comparison of advantages and disadvantages between PFI and DI technologies.

	PFI	DI
Advantages	Long time for fuel evaporation and mixing Low cost of low pressure injection system Low gaseous emissions with TWC Low PM and PN emissions	Improved transient response High compression ratio and fuel efficiency Low HC during cold start Extended EGR tolerance limit Enhanced potential for system optimisation
Disadvantages	Fuel delivery delay due to port wall wetting Low compression ratio and fuel efficiency Fuel evaporation in intake port reduces volumetric efficiency Over fuelling and high HC during cold start	Short time for fuel evaporation and mixing High cost of high pressure injection system Reduced TWC efficiency under lean mode High PM and PN emissions In-cylinder fuel impingement Injector deposits and ignition fouling

4. Advantages of dual injection

As discussed above, both DI and PFI have their respective advantages. DI+PFI dual injection (**Fig. 2c**) is a novel concept to integrate their advantages. Dual injection concept offers greater flexibility in the control of mixture formation and combustion processes. It is also a more effective and efficient method to use the limited supply of renewable fuels than blending them with gasoline at fixed ratios. In addition, the use of renewable fuels can further enhance the benefits of dual injection by taking advantage of their fuel properties such as higher enthalpy of vaporisation, greater octane number and wider flammability limit. Specifically, dual injection provides the following advantages when compared with DI or PFI.

4.1. Control flexibility

Dual injection enables an engine to switch between PFI, DI and dual injection modes to achieve optimal performance under various operating conditions. For example, DI can be used for engine cold start to avoid over-fuelling of PFI and thus to improve the fuel efficiency and HC emissions. Under low load conditions, the engine can work in PFI mode to avoid fuel impingement and poor evaporation of DI due to the low-pressure and low-temperature in-cylinder environment. Dual injection has also been proposed to address the issue of high particulate emissions under cold-start and transient conditions since a large fraction of particulate emissions are generated at these times in a test cycle [87, 88]. As engine load increases, the percentage of DI fuel can be increased to cool the combustion temperature and thus to suppress engine knock. Moreover, the anti-knock ability can be greatly enhanced by using fuels with high RON and/or large enthalpy of vaporisation (such as ethanol and premium unleaded petrol). Such a strategy is often referred to as octane-on-demand [89-92] and several patents have been granted using this idea [93-97].

Dual injection can also change the renewable fuel and gasoline blending ratio in real time according to the operating condition. Experiments showed that different biofuel-gasoline blending ratios were needed to achieve an optimal engine performance under different operating conditions [43, 98-102]. The supply of renewables is limited and can only partially substitute gasoline fuel. Thus dual injection enables an on-demand control of in-cylinder blending ratio, which is a more effective and efficient use of renewable fuels than pre-blending them with gasoline at a fixed ratio. Daniel et al. [103, 104] compared the combustion performance of gasoline-biofuel dual injection with DI of gasoline-biofuel blends. The results showed advantages of dual injection over DI, including a shorter combustion duration, greater in-cylinder pressure and higher thermal efficiency, and reduced CO and mean PM diameter.

4.2. Enhanced cooling effect

DI has stronger cooling effect than PFI because of fuel evaporation inside the engine combustion chamber. Such cooling effect brings various benefits to SI engines, including higher volumetric efficiency, lower NO_x emissions, reduced knock propensity, larger compression ratio, greater turbocharging and higher thermal efficiency. Moreover, the cooling effect of DI could be further strengthened by using fuels with larger enthalpies of vaporisation. **Table 2** shows that the enthalpies of vaporisation of alcohol fuels, in particular ethanol and methanol, are much larger than gasoline. Therefore, existing dual injection studies mostly used gasoline for PFI and alcohols for DI.

The cooling effect of DI can be quantified in various ways. The simplest way is to directly measure the air temperature inside the cylinders. Cold-wire resistance thermometers were applied to measure the in-cylinder air temperatures of PFI and DI engines [105, 106], which required fast-response and thin sensors due to the highly transient nature of in-cylinder flows. As a result, such experiments were only conducted under non-firing conditions to protect the fragile sensors [105, 106]. Attar et al. [107] developed a tracer-based PLIF method to measure the in-cylinder gas temperature of a DI engine under both motoring and firing conditions. So far, the majority of experimental studies quantified the cooling effect indirectly by measuring parameters that linked to charge cooling, such as in-cylinder pressures [108], volumetric efficiencies [109] and anti-knock effects [110, 111]. Wu et al. [112] measured the volumetric air flow rate of a gasoline PFI plus gasoline/DMF/ethanol DI engine. They found that air flow rate increased with the DI ratio and the increase was much bigger for ethanol than DMF and gasoline. Zhuang and Hong [113, 114] found that the volumetric efficiency of a gasoline PFI plus ethanol DI engine increased with the DI ratio only when the injection was during intake valve open. However, volumetric efficiency or intake flow rate could only reflect a proportion of the charge cooling effect that occurred before the intake valves are closed. In a DI SI engine, fuel evaporation would continue during the compression stoke or even the combustion process. Therefore, knock onset was considered a better parameter to quantify the cooling effect. It is worth mentioning that the anti-knock ability of DI can come from two parts, namely the thermal benefit (i.e. cooling effect) and chemical benefit (i.e. higher RON). It was found that the benefit of ethanol's cooling effect was comparable to its higher RON [111, 115]. To quantitatively compare ethanol's thermal and chemical benefits, an increase of 2-8 kJ/kg in the mixture cooling power is equivalent to one-point increase in the RON [116], or adding 10% ethanol into gasoline increases the RON by five points [117].

Meanwhile, numerical simulation is an economic and powerful tool to overcome the challenges of experimental methods and has been adopted to quantify the cooling effect of DI and dual injection systems. Wyszynski et al. [109] used a 0-D model to estimate the theoretical increases in volumetric efficiency of DI over PFI. Kasseris and Heywood [111] used an 1-D model to evaluate the anti-knock benefits of a DI engine fuelled with ethanol-gasoline blends [111]. However, 0-D and 1-D models were for specific aims and thus the output information was usually very limited. Huang et al. [102] used a 3-D computational fluid dynamics (CFD) model to investigate the cooling effect of a gasoline PFI plus ethanol DI dual injection engine. They found that the overall cooling effect increased with DI ratio within 0%-58% but not with higher DI ratios, due to ethanol's low evaporation rate and in-cylinder wall wetting. Ethanol evaporates relatively slowly when compared with gasoline under low-temperature conditions (such as naturally-aspirated engines) [118, 119], which limits the amount of the realised cooling effect from ethanol. This can be improved by increasing the air temperature. Kasseris and Heywood [110] explored the effect of intake air temperature on cooling effect by 3-D CFD modelling. They found that all the theoretical cooling effect could be realized under a high intake air temperature condition of 120 ℃.

4.3. Knock mitigation

The anti-knock ability of a fuel is described by RON in SI engines. **Table 2** shows that renewable fuels all have higher RON than gasoline. In addition, dual injection of gasoline and alcohol fuels can further enhance the anti-knock ability by taking advantage of the cooling effect of DI and the higher enthalpies of vaporisation of alcohols. As discussed above, the thermal and chemical benefits have the same level of importance in knock mitigation [111, 115]. The advantages of dual injection in knock mitigation have been extensively investigated by engine experiments. Liu et al. [120] investigated the knock characteristics of a methanol PFI plus gasoline DI engine and reported that dual injection could effectively suppress engine knock, extend high-load limit and improve fuel economy. Experiments on an ethanol PFI plus gasoline DI engine showed simultaneous reductions of knock propensity and emissions when ethanol was injected during intake valve open [121]. Meanwhile, Zhuang et al. [122] evaluated the knock mitigation ability of a gasoline PFI plus ethanol DI engine. They found that dual injection effectively reduced knock by increasing the ethanol DI ratio, and permitted a more advanced spark timing and higher intake air pressure when compared with gasoline PFI or DI. However, experiments on a gasoline PFI plus n-butanol DI engine showed that dual injection of 20% and 50% n-butanol had a higher knock propensity and intensity than gasoline PFI, although dual injection demonstrated a higher indicated mean effective pressure (IMEP) [123]. A comparison between the results in [123] and [120-122] clearly demonstrates that biofuels (e.g. ethanol and methanol) with higher enthalpies of vaporisation and larger RON will have greater anti-knock ability.

4.4. Engine downsizing

Engine downsizing is considered a key technology to achieve future carbon reduction targets [124, 125]. The concept of engine downsizing is to use a smaller engine in a car to provide similar power performance to a larger engine by boosting with a turbocharger while keeping the compression ratio as high as possible to achieve the best thermal efficiency. The main advantages of engine downsizing include lower mechanical and thermal losses, reduced engine weight, and more operation time within the optimal performance zone of an engine [125]. Turner et al. [126] demonstrated that it was possible to reduce engine displacement by 60% and fuel consumption by 35%, while still achieve a comparable torque performance of a modern large (e.g. 5.0 L) naturally-aspirated engine. However, the key challenges of downsizing of SI engines are the increased knock propensity at high load and reduced fuel economy at part load [124]. The greater control flexibility and anti-knock ability of dual injection could help address these challenges. As shown in **Table 1**, by applying dual injection, the Toyota M20A-FKS/A25A-FKS/A25A-FXS naturally-aspirated engines have achieved high compression ratios of 14:1 in a hybrid configuration and 13:1 in a conventional configuration in mass production cars (i.e. Camry and RAV4 models). The Volkswagen CJSB/DJHB/CHHA turbocharged engines have also adopted dual injection systems. It should be pointed out that the same fuel (i.e. gasoline) is used for both PFI and DI in these dual injection engines. It is expected that more aggressive engine downsizing (e.g. higher turbocharging, larger compression ratio and more spark advance) could be adopted by using renewable fuels with the dual injection concept due to their greater RON and enthalpies of vaporisation.

4.5. Fast combustion speed

Renewable fuels mostly have higher flame speeds than gasoline. Therefore, adding renewable fuels into gasoline engines could enhance the burning rate. Most studies have observed faster combustion speeds of dual injection systems either indirectly via in-cylinder pressure related parameters or directly via visualisation of in-cylinder flows by experimental and numerical methods. For example, experiments on engines equipped with gasoline PFI plus biofuels DI [112, 113], alcohols PFI plus gasoline DI [127] and gasoline PFI plus hydrogen DI [128] dual injection systems all reported faster combustion speeds with shorter/earlier combustion durations/phases (e.g. 0-10%, 10%-90% and 50% of mass fraction burnt), higher/earlier phase of peak in-cylinder pressure, or higher/earlier phase of peak heat release rate. Jiang et al. [129] visualised the combustion process in a gasoline PFI plus ethanol/DMF DI optical engine and found that both gasoline-ethanol and gasoline-DMF dual injection systems had faster combustion speeds than gasoline PFI. CFD modelling results of a gasoline PFI plus ethanol DI engine also showed higher flame propagation speed of dual injection than gasoline PFI when the ethanol ratio was less than 76% [102, 130].

4.6. Extended lean burn limit

Lean burn technology is an effective strategy to reduce fuel consumption and NO_x emissions of SI engines due to lower pumping losses and combustion temperatures [131, 132]. However, the application of lean burn technology is limited by the issues of higher cyclic variation, lower combustion speed and higher ignition energy. Dual injection of renewable fuels offers the potential to extend the lean burn limit. In particular, hydrogen has significantly higher flame velocity, wider flammability limit, lower ignition energy and faster diffusion rate than gasoline [133], and thus has been extensively investigated for extending the lean burn limit of SI engines. Experiments on a gasoline PFI plus hydrogen DI engine showed that hydrogen addition increased thermal efficiency and reduced cyclic variation under lean conditions [128, 134, 135]. The lean burn limit increased with hydrogen DI ratio and could reach an excess air ratio (λ) of 2.65 with 10.5% of hydrogen [134]. Hydrogen addition could also extend the EGR limit under lean-burn conditions [136]. Gong et al. [137, 138] reported that a dual injection engine equipped with hydrogen PFI plus methanol DI could effectively extend the lean-burn limit from λ=1.6 without hydrogen to λ=2.2 with 3%–6% of hydrogen DI, as well as reduced cyclic variation.

Liquid renewable fuels could also improve the lean-burn performance of dual injection engines. Zhuang et al. [139] reported that the lean-burn limit of a gasoline PFI plus ethanol DI engine was increased by on average 20% when compared with gasoline PFI. In addition, the lean burn limit increased with the increase of the ethanol ratio and the advance of DI timing. Experiments on a dimethyl ether (DME) PFI plus gasoline DI engine under lean-burn conditions also observed increased thermal efficiency and reduced cyclic variation under dual injection mode [140].

4.7. Higher thermal efficiency and selected emissions reduction

The ultimate goal of the dual injection concept is to increase engine thermal efficiency and thus to save fossil fuels. This has been well achieved by the advantages discussed above, in particular greater control flexibility, engine downsizing, faster combustion speed and extended lean burn limit. In terms of pollutant emissions, most dual injection studies have reported reductions in particulate and specific gaseous emissions. The detailed effects of dual injection on thermal efficiency and pollutant emissions will be reviewed and discussed in **Section 5**.

5. Performance of dual injection engines

This section discusses the thermal efficiency and pollutant emissions performance of various dual injection engines. It should be noted that the dual injection concept can be applied relatively flexibly so that different fuels are used in dual injection with various combinations, such as dual injection of gasoline plus one renewable fuel (e.g. gasoline PFI + ethanol DI [141]), dual injection of a single fuel (e.g. ethanol PFI + ethanol DI [142]), or dual injection of two renewable fuels (e.g. acetone-butanol-ethanol blend (ABE) PFI + hydrogen DI [143] and hydrogen PFI + methanol DI [144, 145]). However, the supply of renewable fuels is still limited so that they can only partially substitute gasoline use in real world applications. Therefore, this section focuses on dual injection of renewable fuels with gasoline, which is considered as a more effective and efficient method to use renewable fuels in SI engines than pre-blending them with gasoline at fixed ratios. So far, the renewable fuels that have attracted the most research interests in such application are ethanol, methanol and hydrogen. For conciseness, gasoline/ethanol/methanol/hydrogen PFI and gasoline/ethanol/methanol/hydrogen DI are abbreviated as GPI/EPI/MPI/HPI and GDI/EDI/MDI/HDI in this section, respectively.

5.1. Ethanol-gasoline dual injection engines

Ethanol is the most popular alternative fuel to gasoline in SI engines and thus has attracted the most research attention for dual injection applications. Compared with gasoline, ethanol has several advantages including a larger RON, a greater enthalpy of vaporisation and a faster laminar flame speed. These advantages make ethanol an ideal anti-knock agent in dual injection engines to implement engine downsizing technologies. Cohn et al. [146] firstly proposed an ethanol boosted engine concept which had the potential to increase the gasoline engine efficiency by approximately 30%. They proposed to directly inject a small volume of ethanol into the combustion chamber as an anti-knock agent only when high torque output was required, whilst gasoline was delivered via PFI. This could effectively mitigate the engine knock due to ethanol's high RON, and supplemented by the stronger cooling effect due to DI and ethanol's greater enthalpy of vaporisation. The reduced knock propensity made it possible to adopt a high compression ratio and turbocharging in a downsized engine, and consequently to increase the fuel efficiency significantly. A series of patents have been granted based on this concept [93, 95-97]. Similarly, Ford Motor Company introduced an EcoBoost turbocharged DI engine in the 2010 Lincoln MKS [147], which used gasoline PFI for starting and light to medium load operation, and E85 DI was only applied to supress knock during high load operation. The EcoBoost engine demonstrated great leveraging effects of E85 in lowering gasoline consumption.

Following that, significant research has been conducted to investigate the performance of ethanol-gasoline dual injection engines. **Fig. 3** compares the thermal/fuel efficiency performance of various ethanol-gasoline dual injection engines. The corresponding engine operation conditions and emissions performance are given in **Table 4**. Since the most important advantage of ethanol fuel is knock mitigation, most existing studies explored engine performance under knock limited spark timing (KLST) or minimum spark advance for best torque (MBT) conditions while the air fuel ratio (AFR) was kept stoichiometric, as shown in **Table 4**. Although the original ethanol boosted engine concept proposed to use ethanol via DI, both DI and PFI were explored for ethanol utilisation in the research community, namely GPI+EDI and EPI+GDI. **Fig. 3** shows that both dual injection configurations could improve the engine thermal/fuel efficiency compared with single injection, mainly due to ethanol's faster combustion speed and more spark advance allowed by ethanol's anti-knock ability. Regarding the emissions performance, **Table 4** shows that both configurations could effectively reduce the PM and PN emissions when compared with GDI [104, 121, 148-151]. For gaseous emissions, the variation trends are highly dependent on engine operating conditions and there are usually trade-offs between emission species such as NO_x *vs* CO/HC due to their different or conflicting emission formation pathways [152].

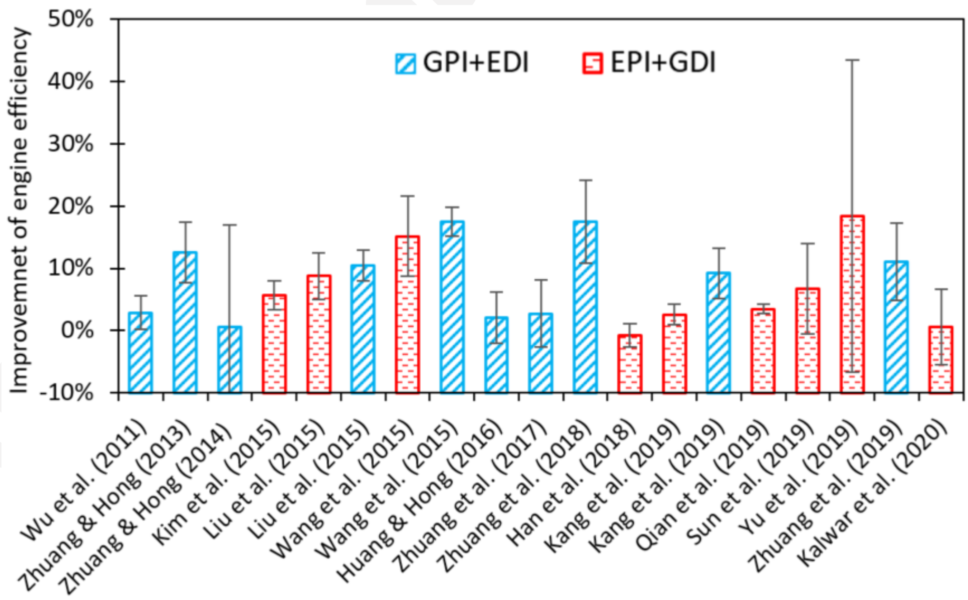

Fig. 3. Improvements in thermal/fuel efficiency of ethanol-gasoline dual injection engines. The results are shown as the mean and range of efficiency improvements observed.

Table 4. Comparison of emissions performance of ethanol-gasoline dual injection engines.

Experiments	**Engine conditions** (compression ratio, induction, speed, AFR, spark timing, load)	**Comparison baselines***	**Emissions performance***				
			CO	HC	NO_x	PM	PN
Wu et al. (2011) [112]	11.5:1, aspirated, 1500 rpm, λ=1, KLST, IMEP=4.5-8.5bar	GPI+EDI *vs* GPI		↓	↓		
Daniel et al. (2013) [104]	11.5:1, aspirated, 1500 rpm, λ=1, KLST, IMEP=5.5bar	GPI+E_xDI *vs* E_xDI					↑
Zhuang & Hong (2013) [113]	9.8:1, aspirated, 3500-5000 rpm, λ=1, 15° BTDC, light-medium	GPI+EDI *vs* GPI	↑	↑	↓		
Zhuang & Hong (2014) [153]	9.8:1, aspirated, 3500-4000 rpm, λ=1 & lean burn limit, 21-25° BTDC & KLST, light-medium	GPI+EDI *vs* GPI		⇅	⇅		
Kim et al. (2015) [121]	9.5/13.3:1, aspirated, 1000 rpm, λ=1, KLST, wide open throttle (WOT)	EPI+GDI *vs* GDI	↓	⇅	⇅	↓	↓
Liu et al. (2015) [148]	13:1, aspirated, 1600 rpm, λ=1, MBT, WOT	EPI+GDI *vs* GDI					↓
		GPI+EDI *vs* GDI					↓
Wang et al. (2015) [154]	13:1, aspirated, 1600 rpm, λ=1, MBT, WOT	EPI+GDI *vs* GDI					
		GPI+EDI *vs* GDI					
Huang & Hong (2016) [155]	9.8:1, aspirated, 3500-4000 rpm, λ=1, 15° BTDC & MBT, medium	GPI+EDI *vs* GPI	⇅	⇅	↓		
Zhuang et al. (2017) [122]	9.8:1, aspirated, 3500 rpm, λ=1, KLSA, IMEP=7.2-8.5bar	GPI+EDI *vs* GDI	↓	⇅	↑		
Zhuang et al. (2018) [139]	9.8:1, aspirated, 3500-4000 rpm, lean burn limit, 25° BTDC, IMEP=5.5-7.5bar	GPI+EDI *vs* GPI		⇅	↑		
Han et al. (2018) [149]	10.5:1, boosted, 1500 rpm, λ=1, KLST, IMEP=9-13bar	EPI+GDI *vs* E_xDI		↓	⇅		↓
Kang et al. (2019) [50]	12:1, aspirated, 1500 rpm, λ=1, MBT, WOT	EPI+GDI *vs* GDI					
		GPI+EDI *vs* GPI					
Qian et al. (2019) [91]	10:1, boosted, 2000 rpm, λ=1, KLST, BMEP=6bar	EPI+GDI *vs* GDI	↑	↓	↓		
Sun et al. (2019) [150]	9.6:1, aspirated, 1500-2100 rpm, λ=1, MBT, partial load	EPI+GDI *vs* GDI		↓			↓
Yu et al. (2019) [156]	9.6:1, aspirated, 1500 rpm, λ=1.0-1.4, MBT, IMEP=2-4bar	EPI+GDI *vs* EPI	⇅	⇅	⇅		⇅
Zhuang et al. (2019) [157]	9.8:1, aspirated, 3500 rpm, λ=1.0, 15° BTDC, light-medium	GPI+EDI *vs* GPI	⇅	⇅	⇅		
Kalwar et al. (2020) [151]	10.5:1, aspirated, 2000 rpm, λ=1.0, 24° BTDC, IMEP=5-7bar	EPI+GDI *vs* GDI	↓	↑	↓	↓	↓

Symbols: ↑, increased; ↓, decreased; ⇅, depended on operating conditions; Nil, not reported; E_x, ethanol-gasoline blends.

Studies have been conducted to compare engine performance between different dual injection configurations [50, 148, 154, 158]. They all reported that GPI+EDI offered better thermal efficiency than EPI+GDI because EDI could better utilise the cooling effect of ethanol. As a result, the majority of existing studies adopted the dual injection configuration of GPI+EDI. Particularly, Huang and Zhuang et al. have conducted significant works in this area by both experiments and CFD simulations. Firstly, they systematically evaluated the leveraging effect [113, 157], knock mitigation [122, 153] and lean burn performance [139, 153] of a GPI+EDI engine under various conditions. The experimental results showed that engine performance was improved by EDI, including higher thermal efficiency, lower NO_x, extended lean burn limits and more spark advance without knock issues. However, CO and HC were increased by EDI. To understand the underlying mechanisms, CFD simulations [102, 130, 159] and spray experiments [160, 161] were performed to investigate the spray evaporation and combustion processes of the GPI+EDI engine. The CFD results revealed that the flame propagated faster and combustion temperature was lower in GPI+EDI than GPI, leading to higher thermal efficiency and lower NO_x. However, ethanol evaporated slowly under the low-temperature condition before combustion started, causing incomplete combustion. Moreover, over-cooling effect and in-cylinder wall wetting occurred when ethanol DI ratio was over 58%, which worsened the incomplete combustion issue. This led to the increases of CO and HC observed in the engine experiments. To address these issues, EDI heating was proposed as an economic and effective method to generate fine and fast evaporating DI sprays in the GPI+EDI engine [155]. The experimental results showed that ethanol fuel heating was an effective method to solve the problems of ethanol's slow evaporation and over-cooling effect in the EDI+GPI engine in terms of minimising the emissions.

5.2. Methanol-gasoline dual injection engines

Methanol has a comparable RON and an even higher enthalpy of vaporisation than ethanol. Therefore, methanol is another effective anti-knock agent in dual injection engines to implement engine downsizing technologies. Liu and co-workers have conducted significant experimental works to explore the performance of methanol-gasoline dual injection engines. They found that MPI+GDI could effectively suppress engine knock, extend high-load limits and reduce PN emissions when compared with GDI [120, 162]. They further compared the performance between methanol and ethanol, and found that MPI+GDI had better fuel efficiency and knock mitigation ability than EPI+GDI [127]. Kalwar *et al.* [151] also observed a higher brake thermal efficiency for MPI+GDI than that of EPI+GDI. This could be attributed to methanol's slightly stronger cooling effect and higher laminar flame velocity than ethanol. Experiments were also conducted to compare the performance of different methanol-gasoline dual injection configurations. Same as for ethanol-gasoline dual injection, they found that GPI+MDI demonstrated higher fuel efficiency, greater anti-knock ability and larger PN reductions than MPI+GDI [148, 154]. The enthalpy of vaporisation of water is even much larger than that of methanol and thus has significant anti-knock potentials. Experiments on a GDI engine showed that KLST could be advanced by water-methanol blends PFI, with pure water PFI having the most advanced KLST [163].

5.3. Hydrogen-gasoline dual injection engines

Hydrogen is carbon free and is an attractive alternative fuel for SI engines. Compared with gasoline, the most important advantages of hydrogen are its significantly higher flame velocity, wider flammability limits and lower ignition energy. These advantages make hydrogen an excellent agent to extend the lean burn limit of SI engines, and thus to improve the fuel efficiency. Hydrogen can be introduced into the engine via either PFI or DI, and many studies were conducted to investigate their performance under lean burn conditions.

Unlike alcohol-gasoline dual injection engines which could vary the alcohol ratio in a wide range from 0% to 100%, hydrogen-gasoline dual injection engines usually used only a small hydrogen ratio (typically <15%) to partially substitute gasoline fuel.

Ji *et al.* [164] studied the performance of an HPI+GDI engine under a lean condition of λ=1.2. Comparing with GDI, HPI+GDI effectively increased the thermal efficiency and significantly reduced the PN, CO and HC emissions and cyclic variation, while NO_x emissions were increased by hydrogen addition. However, an inherent disadvantage of HPI is the reduced volumetric efficiency because hydrogen displaces air in the intake port [133]. This can be avoided well by HDI after intake valve closes. HDI also has the potential to avoid pre-ignition – another major challenge of HPI – by controlling the DI timing [133]. Therefore, GPI+HDI attracted more research attention. Yu and co-workers carried a series of works on the lean burn performance of GPI+HDI engines. Compared with GPI, GPI+HDI effectively improved the thermal efficiency, expanded the lean burn limit and reduced the cyclic variation [134, 135, 165]. In addition, GPI+HDI had better ignition reliability during cold start [166] and extended the EGR limit [136] than GPI. In terms of emissions performance, CO, HC and PN were greatly reduced while NO_x was increased by hydrogen addition [128, 167, 168]. The increased NO_x of GPI+HDI could be controlled by increasing the excess air ratio (e.g. λ=1.5) [128], by using EGR [136] and by split hydrogen injection [169, 170]. Yu *et al.* [171] further compared GPI+HDI with GPI+GDI, and reported that GPI+HDI demonstrated larger improvements in thermal and exergy efficiencies with the increase of DI ratio.

5.4. Other renewable fuel-gasoline dual injection engines

Although receiving less research attention, other renewable fuels have also been explored for dual injection engines, such as n-butanol, ABE, DMF and DME. Compared with gasoline, n-butanol has a comparable RON and moderately higher enthalpy of vaporisation. Therefore, n-butanol is a less effective anti-knock agent when compared with methanol and ethanol. Experiments on a GPI plus n-butanol DI engine showed that dual injection could achieve up to 7% lower energy consumption than GPI [123]. He et al. [172] and Wang et al. [173] examined the performance of different n-butanol-gasoline dual injection configurations. It was reported that GPI plus n-butanol DI had similar IMEP under a lean condition (λ=1.3) [172] but higher IMEP under stoichiometric and rich (λ=0.9) conditions [173] than n-butanol PFI plus GDI. Meanwhile, GPI plus n-butanol DI emitted higher PN but lower PM emissions than n-butanol PFI plus GDI [173]. ABE is a solution of acetone, butanol and ethanol, which reduces the separation and purification costs of bio-butanol production. Guo et al. [174-176] investigated the performance of ABE-gasoline dual injection engines. Their results showed that both ABE PFI plus GDI and GPI plus ABE DI could effectively improve the engine combustion and emissions performance compared with the gasoline only condition [174, 175], and GPI plus ABE DI performed better than ABE PFI plus GDI [176].

DMF has a higher RON but lower enthalpy of vaporisation than gasoline. Therefore, the anti-knock ability of DMF is much smaller than that of ethanol and methanol. Experiments on a GPI plus DMF DI engine demonstrated that thermal efficiency was increased at high load, was comparable at medium load but was lower at low load when compared with GPI [112]. Meanwhile, HC was decreased while NO_x was increased at all loads.

Finally, although DME is considered an alternative fuel to diesel, Shi et al. [140] studied the performance of a DME PFI plus GDI engine under split DI and lean-burn conditions. Compared with GDI, dual injection increased thermal efficiency, reduced HC and NO_x emissions, but increased CO emissions.

6. Summary and outlook

This study critically reviewed the dual injection concept for SI engines. It can be concluded that dual injection is an effective and efficient method to use renewable fuels in SI engines. Dual injection offers greater flexibility in the control of mixture formation and combustion processes by integrating the advantages of both DI and PFI technologies and is a promising technology to help achieve the increasingly stringent emission standards. Compared with the conventional single injection engines, the main advantages of dual injection engines include greater control flexibility, enhanced cooling effect, knock mitigation, engine downsizing, faster combustion speed, extended lean-burn limits, higher thermal efficiency, and reductions of some emission species. So far, the renewable fuels that have attracted the most research attention for dual injection SI engines are ethanol, methanol and hydrogen. Each of them is aimed at different advantages of the dual injection concept. Alcohol-gasoline dual injection engines provide great anti-knock ability by taking advantage of alcohols' large enthalpies of vaporisation and high octane numbers, while hydrogen-gasoline dual injection engines provide extended lean-burn limits by taking advantage of hydrogen's low ignition energy, wide flammability limits and fast flame velocity. The renewable fuels can be delivered into the dual injection engines via either PFI or DI. However, renewables DI plus gasoline PFI is the optimal strategy for both alcohol and hydrogen fuels. This is because alcohols DI can better utilise the cooling effect of alcohol fuels which has the same level of importance in knock mitigation as high octane rating of alcohols. Meanwhile, hydrogen DI can effectively avoid volumetric efficiency reductions and pre-ignition by adjusting the DI timing. Regarding the combustion and emissions performance, dual injection engines generally reported higher thermal efficiencies when compared with PFI or DI single injection engines. In addition, dual injection engines could effectively reduce particulate emissions but there were usually trade-offs among gaseous emissions.

Whilst most of the reviewed studies have revealed the benefits of dual injection over conventional single injection, it should be noted that all these studies were conducted under steady state conditions. Future studies are needed to investigate the performance of dual injection engines under transient conditions and to address the challenges of the optimal control of two injection systems. It is expected that increasing production of dual injection cars (but fuelled by one fuel) will be available in the automotive markets, with more mature engine design and global drive for renewable transport fuel. We advocate that future studies investigate practical applications of dual injection of two fuels on real vehicles with a focus on the most promising renewable fuels such as ethanol, methanol and hydrogen that are identified in this study.

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