A review of water injection application on spark-ignition engines

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Abstract

The increasingly stringent emission and fuel economy regulations force the advancement of internal combustion engines towards higher power density and lower emissions. Knock and thermal limits are bottlenecks that prevent improvement of the spark ignition (SI) engines' thermal efficiency. Due to its significant knock mitigation and cooling abilities, water injection technology has great potentials in enhancing thermal efficiency of SI engines and thus regains research attention in recent years. This paper aims to present a comprehensive review of water injection applications on SI engines. Various methods for the implementation of water injection in SI engines are introduced and compared. Methods to maximize the working efficiency of water injection and its detailed physical and chemical effects on combustion and emissions are discussed. It was found that, with different purposes, water injection can improve the combustion and performance of SI engines. Typically, better combustion phasing and thus higher thermal efficiency can be achieved especially under high load conditions. Emissions such as nitrogen oxides and soot can be largely suppressed. However, crucial issues such as water supply and wall wetting still restrict wide application of water injection technology. The detailed kinetic mechanisms of chemical effects and coupling of physical and chemical effects are needed to be investigated.

Keywords: SI engine; water injection; fuel efficiency; emission; chemical effect; vaporization

Abbreviations

AFR	Air fuel ratio	ISFC	Indicated specific fuel consumption
AKI	Anti-knock index	IVO	Inlet valve open
ATDC	After top dead center	IWI	Indirect water injection
BMEP	Brake mean effective pressure	KLSA	Knock limited spark advance
BTDC	Before top dead center	LBV	Laminar burning velocity
CAD	Crank angle degree	LPI	Low pressure injection
CAI	Controlled auto ignition	LSPI	Low speed pre ignition
CFD	Computational fluid dynamics	MBT	Maximum brake torque
CFR	Cooperative fuel research	MFB	Mass fraction burned
CI	Compression ignition	NO	Nitric oxide
CO	Carbon monoxide	NO_x	Nitrogen oxides
CO_2	Carbon dioxide	NIMEP	Net indicated mean effective pressure
COV	Coefficient of variation	NEDC	New European driving cycle
CR	Compression ratio	PFI	Port fuel injection
CWI	Central water injection	PM	Particulate matter
DISI	Direct injection spark ignition	PWI	Port water injection
DWI	Direct water injection	RDE	Real driving emissions
EGR	Exhaust gas recirculation	SCR	Selective catalytic reduction
EOI	End of injection	SI	Spark ignition
EWI	Exhaust water injection	SMD	Sauter mean diameter
GDI	Gasoline direct injection	TSCI	Thermally stratified compression ignition
GPI	Gasoline port injection	TWC	Three way catalyst
HC	Hydrocarbon	VCR	Variable compression ratio
HCCI	Homogenous Charge Compression Ignition	VVA	Variable valve actuation
HPI	High pressure injection	W/F	Water to fuel ratio
ICE	Internal combustion engine	WLTP	Worldwide harmonized light vehicle test procedure
IMEP	Indicated mean effective pressure		

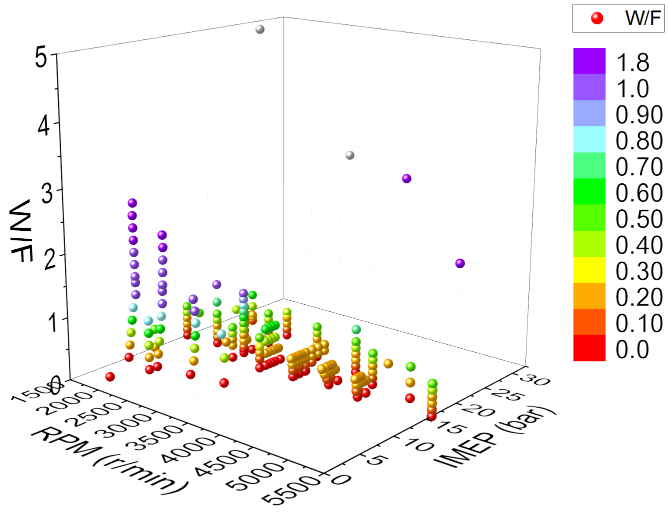
1. Introduction

Admittedly, there is a global focus and enthusiasm on electric vehicles. However, internal combustion engines (ICEs) will still continue to be the dominant source of propulsion power for road transport in the near future [1]. Specifically, spark-ignition (SI) engines are the main power source for light transportation sector worldwide except for some parts of Europe [2].

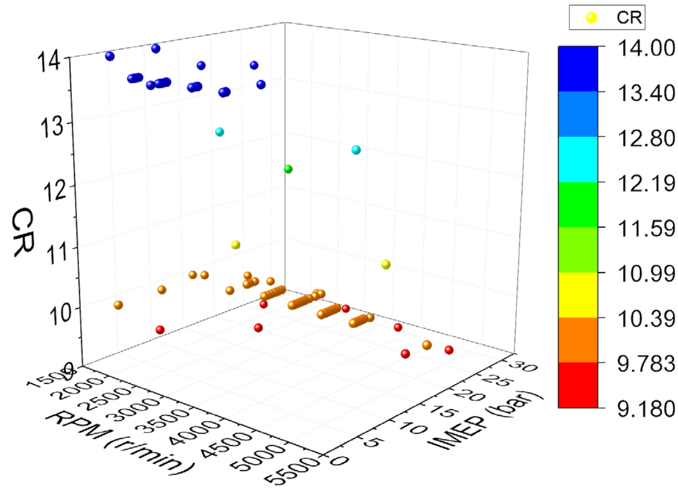
The increasingly stringent emission legislations combined with the consumers' requirements for power performance and drivability put forward stricter demands for SI engines to achieve higher thermal efficiency. The current development trend of SI engines towards higher power density and better fuel economy is mainly realized by high-boost and downsizing technologies, which are seriously constrained by knock and the thermal limits of components. Various solutions have been hence proposed, such as exhaust gas recirculation (EGR), Miller cycle, variable compression ratio (VCR), water injection, duel-fuel injection, hybridization, and inspection and maintenance [3, 4].

Among them, water injection shows significant ability in knock suppression and cooling. The earliest history of water addition in ICEs can be traced back to the early 20^{th} century [5]. Afterwards, as an effective means for knock suppression, water injection was widely adopted in aircrafts and racing cars to obtain temporary power enhancement [6-8]. With the emergence of intercooler (also called charge air cooler), people's enthusiasm for water injection technology gradually declined. Recently water injection has been applied on the mass produced cars by BMW, which achieved substantial power boost and fuel economy improvement [9, 10].

As discussed above, knock and thermal limits are the primary obstacles to further enhance the thermal efficiency of SI engines. Since water is an effective cooling and anti-knock agent, water injection technology attracts attention again. Due to the impending needs for enhancing the engine performance, substantial studies have been conducted to investigate the potential of water injection with the foci on achieving higher compression ratio (CR), improving the working efficiency and extending the anti-knock area of SI engines. **Fig. 1** shows the statistical analysis results of the application of water injection in SI engines based on published papers, which covered different CR, engine speed, water/fuel (W/F) ratio, indicated mean effective pressure (IMEP) and injection methods [11-26]. It can be seen from **Fig.** 1-(a) that most studies adopted W/F ratios lower than 1. Only a few researches attempted to raise water injection quantity and adopted up to 5 W/F ratios. This indicated that proper water injection quantity is lower than a certain value. On one hand, the vaporization of water is limited. On the other hand, the water supply and engine proper working life should be taken into account. It can be seen from **Fig.** 1-(b) that water injection allows CR to be increased up to 14. However, most researches adopted CRs ranging from 10 to 11. **Fig.** 1-(c) indicated that for port fuel injection (PFI) engines, direct water injection (DWI) shows poor working efficiency, achieving IMEPs below 10 bar. However, the range of IMEP can be significantly increased up to 22 bar by using port water injection (PWI). For gasoline direct injection (GDI) engines, it is better to use DWI at low and medium engine speeds to achieve higher IMEP. When at high engine speeds, PWI shows better effects in raising IMEP. This could be related to the vaporization process of water. On the whole, GDI engines showed higher power increase potential with the aid of water injection compared with PFI engines.



(a)



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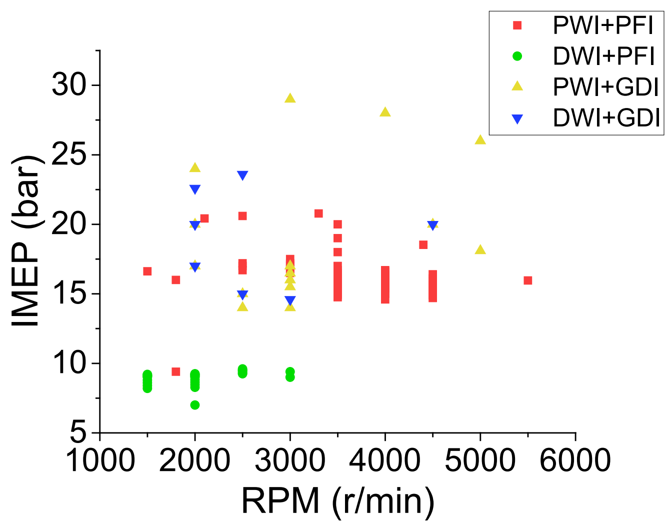

(c)

Fig. 1. Statistics of water injection applications on SI engine: Effect of (a) W/F ratio, (b) CR and (c) water injection methods on IMEP under various engine speeds.

It is proved that water injection could effectively inhibit knock combustion [11, 15, 16, 18, 19, 21, 22, 24, 27-29]. Therefore, in high knock propensity areas (commonly low speed, high load conditions or high speed, high/full load conditions), water injection allows more advance of spark timing to achieve optimal combustion phasing and thus higher thermal efficiency [25]. Additionally, as an effective cooling agent, water helps eliminate the need for fuel enrichment under high load conditions, which is used to thermally protect pistons and catalytic converters [10]. These merits all contribute to the improvements in power performance, fuel economy and emission control, as summarized in **Fig. 2**. Fig. 2 shows that the most important contribution of water injection technology is its charge cooling effect, which reduces the density of the intake air, allowing more fuel to be burned to obtain more power. Water injection has been widely applied on aero-engines before the advent of intercoolers, which are well documented [6, 30]. Currently, the increasingly stringent emission laws make highly downsized, boosted SI engines popular in transportation sector. The use of higher CR is one of the most important means to further enhance the thermal efficiency of SI engines. However, it is seriously restricted by knock combustion onset [31]. Water injection technology helps to cool the air fuel mixture, thus significantly reducing the knock propensity. Lanzafame and Brusca [27, 32] proved that water injection led to the increase of octane number on a single cylinder Cooperative fuel research (CFR) engine using the same fuel. Therefore, the use of higher CR enabled by water injection is one solution to further enhance the thermal efficiency of SI engines without the need for adopting higher octane number fuel than the-state-of-art. Another way to improve the thermal efficiency by using water injection's anti-knock effect is to apply larger spark advance. It is worth mentioning that, with constant spark timing, the cooling and dilution effect of water injection leads to delayed combustion phasing and elongated combustion. The deteriorated combustion hence induces lower thermal efficiency and thus poor fuel economy and/or power performance. In terms of emissions, NO_x emissions are commonly reduced while HC and CO emissions are worsened. Therefore, in order to effectively utilize the anti-knock advantage of water injection technology, advancing of spark timing is required when CR is kept unchanged. In fact, the enhancing effect of water injection on combustion can be realized only if the engine is running in the high knock propensity area. Under high risk of knock combustion conditions, spark advance is commonly reduced to suppress detonation onset, which leads to poor combustion phasing and hence lower thermal efficiency. With the aid of water injection, much larger spark advance can be achieved to realize optimal combustion phasing, leading to better fuel economy and/or power performance. In addition, under full load and high speed conditions, knock combustion is not the only critical issue to be concerned. Another important problem is the high thermal stress on pistons and catalytic converters. Normally, rich combustion is adopted to lower the exhaust gas temperature under high load conditions. The excessively injected fuel is used to cool the end gas. Being a high efficient cooling agent, water has the potential to eliminate the need for fuel enrichment operation and obtain stoichiometric combustion at broader engine map. This leads to the improvement of fuel economy, power performance and even emission control. Additionally, higher boost pressure is possible for further increase of the engine power density. **Fig. 3** shows the engine efficiency improvements of water injection from literatures [10, 12, 16, 18, 19, 21, 24, 25, 33-39]. It can be seen that both PWI and DWI have promotion effects on engine efficiency. Moreover, the effect of PWI is more prominent than DWI. Recently, more scholars tend to investigate PWI. Another reason can be attributed to the lower costs and control complexity compared with DWI.

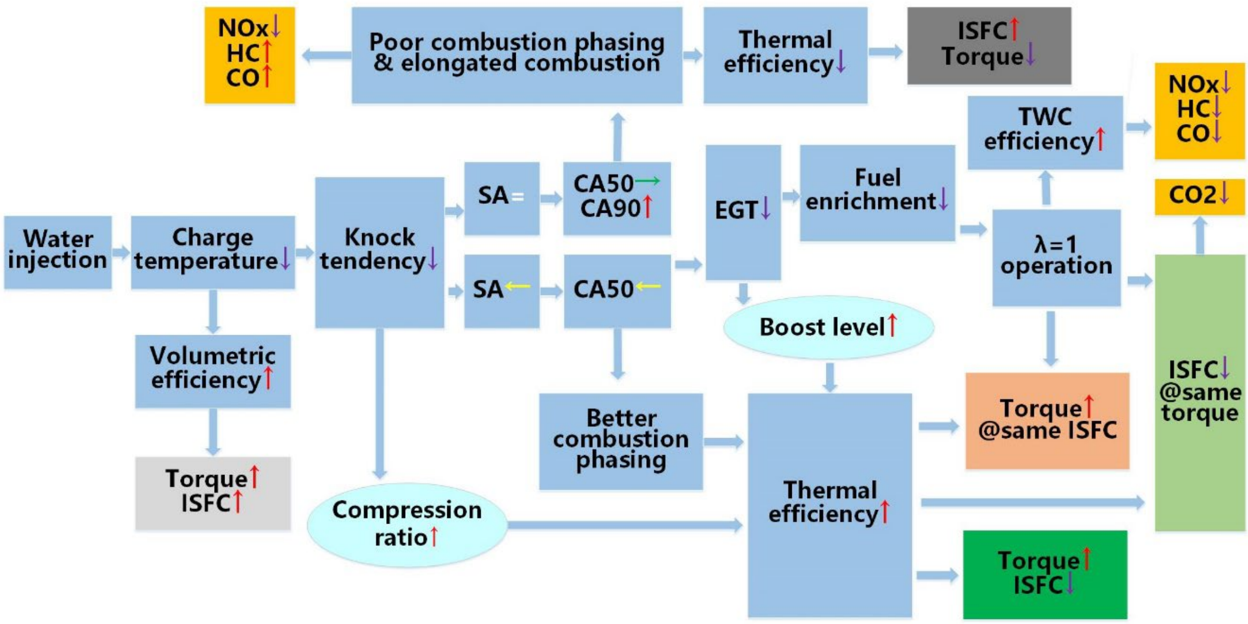

Fig. 2. Effects of water injection on engine performance

(*** Symbols:** ↑, increase; ↓, decrease; ←, advanced; →, delayed; ＝, unchanged).

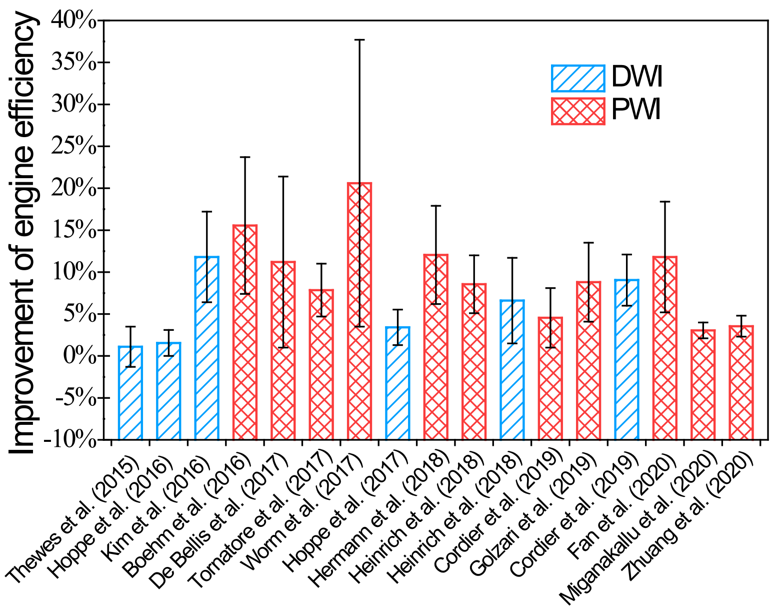

Fig. 3. Water injection effects on improvements of engine efficiency.

There are a few reviews on the water injection application in ICEs [40-42]. However, the mechanisms of water injection especially the vaporization process and chemical effects of water on combustion are rarely discussed. Understanding the kinetic mechanism of water on SI engine combustion process is crucial for further exploring its potential in improving engine performance. This article reviews the recent studies on the mechanisms of water injection on SI engine performance. The remaining of this review is organized as follows. Section 2 introduces and compares different methods for the implementation of water injection in SI engines. Then effect of water injection on mixture formation is introduced. Section 3 introduces thermal and chemical effects of water injection on SI engines. Next, sections 4 and 5 introduce the effects of water injection on combustion and emissions, respectively. Finally, challenges and future research directions are discussed in section 6 and conclusions are made in section 7.

2. Effect of water injection on mixture formation

Due to its high enthalpy of vaporization, water is proved to be a good cooling and anti-knock agent in historical usage. It was found that the significant cooling effect of water is the primary contributor to the knock mitigation of water injection technology. When water is added into the engine, the over fueling could be eliminated to achieve better fuel economy. In addition, more spark advances can be applied to achieve better combustion phasing, which leads to improvement of thermal efficiency. However, these benefits could be realized only if water is well vaporized to fulfill its large cooling effect. Therefore, this section reviews water injection methods, spray characteristics and vaporization.

2.1 Water injection methods

As shown in **Fig. 4**, water can be added into the engine in various ways, which would affect the vaporization process and thus the engine performance as well as the cost. Generally, there are two ways to introduce water into the engine, namely emulsion and dual injection. The first method injects fuel and water blends into the engine using one injection system, while the second method injects fuel and water separately in two independent injection systems. Due to the instability of water-gasoline emulsion, expensive chemical surfactant and the fixed blending ratio, emulsion method is less attractive than the dual injection method, which is more widely and deeply investigated. Therefore, this review focuses on the dual injection method.

Water injection can be divided into two types, namely DWI and indirect water injection (IWI). DWI is also called chamber injection or in-cylinder injection. Similar to GDI, DWI injects water directly injected into the combustion chamber via an independent injection system. The conventional GDI injectors can be directly adopted for DWI. It is worth mentioning that direct adoption of gasoline injector may lead to poor water atomization effect, due to the different physical properties of water and gasoline, e.g. the surface tension and viscosity of water are much larger than gasoline. This will be discussed in detail in section 2.2. The properties of isooctane, alcohol and water are summarized in **Table 1**.

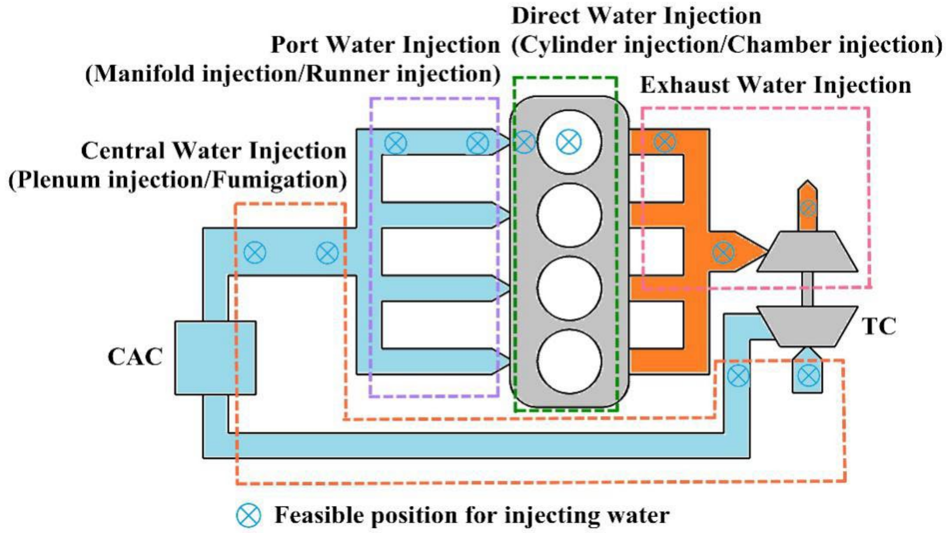

Fig. 4. Water injection methods.

Table 1. Properties of isooctane, alcohol and water.

Property	**Water**	**Isooctane**	**Methanol**	**Ethanol**
Chemical Formula	H_2O	C_8H_{18}	CH_3OH	C_2H_5OH
Density at 25℃ (kg/m3)	997	649	786	786
Lower Heating Value (MJ/kg)	-	44.3	19.9	26.8
Stoichiometric AFR	-	15.1	6.5	9
Research Octane Number	-	100	106	109
Motor Octane Number	-	100	92	98
Latent Heat of Vaporization (kJ/kg) at 1bar	2257	307	1170	930
Oxygen Content by weight (%)	88.9	0	50	34.8
Surface Tension σ (mN/m) at 20℃	72.71	18.32	20.14	22.27
Vapor Pressure (kpa) at 20℃	2.34	5.10	12.3	5.8
Reid vapor pressure (at 37.8 °C) [psi]		1.86	4.60	2.30
Viscosity (mpa·s) at 20℃	0.89	0.47	0.54	1.07
Boiling Point (℃)	100	99.2	64.7	78
Mole ratio of products to reactants	-	1.058	1.065	1.061
Adiabatic Flame Temperature [K]	-	2276	2143	2193
flammability limits in air [λ]	-	0.26~1.51	0.23~1.81	0.28~1.91
Minimum ignition energy in Air [mJ]	-	1.20	0.14	0.23
Quenching distance [mm]	-	3.50	1.85	1.65

IWI can be further divided into central water injection (CWI) and PWI. CWI is also called single point injection or plenum water injection [10]. CWI has many in common with PWI. The main difference lies in multi-cylinder engines, in which CWI does not adjust water injection according to combustion phasing of different cylinders but uses a simple continuous water injection into the intake air. Thus it is also called air humidification or fumigation in some literatures [43]. In spite of its simple configuration and low cost, CWI receives little research attention due to the small effect on engine combustion performance and the low control level. PWI is also called intake manifold injection or runner injection and the number of water injectors usually equals the number of cylinders. In PWI, a low-pressure injection system injects water into the intake manifold. Heat from the intake air, inlet ports, and valves are absorbed by a mist of water. It then arrives at the engine cylinder in a state that is partially vapor and liquid. It is a simple and cost-effective method to reduce combustion temperature.

When the engine is at induction stroke, water is injected into the intake charge to cool down the air, thus volumetric efficiency can be improved. The advantage of this method is its simplicity since the gasoline PFI system can be directly used for water injection with few modifications [18]. In addition, negative impacts such as corrosion and freezing issues caused by water are relatively small and easy to solve.

Another possible location for water injection is in the upstream of exhaust pipe or downstream of the turbocharger. The aim of exhaust water injection (EWI) lies in getting lower exhaust gas temperature. However, water injection in the exhaust has no influence on the combustion process. Therefore fuel enrichment is still necessary to suppress knock combustion and thus water injection in the exhaust line seems to be ineffective and superfluous [12, 44]. Only a handful of such studies have been reported.

There is also a classification method by injection pressures, albeit not so popular. IWI is commonly classified as low pressure injection (LPI) (3-20 bar) which usually uses typical PFI system for water injection while DWI is seen as high pressure injection (HPI) (20-200 bar) for its use of GDI system.

Different injection pressures and injector locations have different degrees of impact on the combustion process. Therefore, many studies have been performed to exposit the distinction of various water injection systems, such as Hermann et al. [10], Hunger et al. [45] and Bertolini et al. [46]. **Table 2** summarizes their results. Compared with PWI, DWI provides higher cooling efficiency, which could be attributed to the better water atomization and vaporization processes. This could further explain why DWI has lower water consumption and larger knock mitigation ability than PWI. In addition, DWI shows higher control flexibility and water can be precisely distributed into the cylinder. However, DWI is more expensive and much integration effort should be paid on highly downsized SI engines, especially on GDI engines. PWI is more cost effective and easy to be installed on either PFI or GDI engines.

Table 2. Comparison of various water injection methods ("+" means positive effect, number of "+" means extent)

Water injection	Indirect water injection (IWI)			Direct water injection (DWI)	
	Air intake pipe (CWI)	Intake manifold (PWI-far from inlet valve)	Intake port (PWI-close to inlet valve)	Combustion chamber	Water fuel emulsion
Cooling of the working gas	+	++	+++	+++++	+++++
Transient operation (changing W/F ratio)	+++++	+++++	+++++	+++++	+
Costs	+++++	++++	++++	++	+
Equal distribution for	+	++	+++	+++++	++++++

cylinders					
Droplet size	+	+	+	+++++	++++++
break up	+	+	++	+++++	++++++
Packaging/integration effort	+++++	++++	+++	+	++
Water consumption for stoichiometric combustion	+	+	++	+++++	+++++
Pump (Pinj)	+++++	+++++	+++++	+	+
Knocking mitigation	+	++	+++	+++++	+++++
Homogenization	+	+	++	++++	+++++
Evaporation	+	+	++	++++	+++++
Thermal efficiency	+	+	++	++++	+++++
Calibration easy	+++++	++++	+++	+	++
Few wall wetting in cylinder	+++++	+++++	++++	+	++

2.2 Water sprays and vaporization

For SI engines, combustible mixture formation is mainly influenced by the fuel atomization process, which can be divided into primary and second breakup. Due to the high volatility of gasoline fuel, the mixing and vaporization processes are well accomplished to fulfill an almost homogeneous fuel distribution inside the combustion chamber. However, water properties are quite different from gasoline fuel, meaning that the same considerations suited for gasoline cannot be applied on water. Specifically, water has poorer vaporization process compared with gasoline. When water is injected, mixture formation would be affected by the atomization and vaporization of water. On the other hand, the time available for the vaporization can be assumed less than 250 crank angle degrees (CAD) [47] and the vaporization time reduces linearly as the engine speed increases.

To better understand and predict the water vaporization process in a limited time window, the water behaviors during and after injection (atomization, mixing and evaporation) should be analyzed in detail.

2.2.1 Atomization

Being the first phase of an injection event, a good atomization provides favorable conditions for liquid to vaporize. Apparently, the smaller the droplets size is, the faster is the heat transfer and evaporation processes will be [47]. The Reynolds (Re) and Ohnesorge (Oh) numbers can be used to classify the primary breakup. The interpretation of the Reynolds number (see equation (1)) corresponds to the quotient of the inertia and the viscous forces. The Oh number (see equation (3)) is the quotient of the square root of the Weber (We) number (see equation (2)) over Reynolds number. Accordingly, dividing the viscous forces by the square root of both the inertia forces and the surface tension produces the Oh number. The terminology of the symbols used in equations (1) – (3) are given in the Nomenclature table.

$$Re = \frac{\rho \cdot v \cdot d}{\eta} \quad (1)$$

$$We = \frac{\rho \cdot v^2 \cdot d}{\sigma} \quad (2)$$

$$Oh = \frac{\sqrt{We}}{Re} = \frac{\eta}{\sqrt{d \cdot \rho \cdot \sigma}} \quad (3)$$

Table 3. Comparison of physical properties of water and gasoline.

T_{ref}=300K	Water	Gasoline	Water/Gasoline
Saturation pressure, P_{sat}(Pa)	3543	53951	≈ 1/15
Surface tension, $\sigma(N/m)$	0.072	0.025	≈3
Density, ρ(kg/m^3)	1000	745	1.3
Dynamic viscosity, μ(Pa·s)	0.00087	0.00045	≈2

The main properties determining spray break-up are shown in **Table 3**. It could be seen that water has higher density, viscosity, surface tension but lower saturation vapor pressure than gasoline. This indicates that water has lower Re and Oh numbers than gasoline [25], thus indicating a poorer atomization level of water. To reach the same liquid break up level as gasoline, water needs a much higher injection pressure.

To be specific, as shown in **Fig. 5**, an increment of injection the pressure from 50 to 100 bar results in a remarkably improved primary breakup. However, only slight improvements were recorded when the injection pressure exceeded 200 bar. When the maximum injection pressure was assumed to be 500 bar, the primary breakup of water remained inferior to that of isooctane at 200 bar injection pressure. It could be concluded that that water needs more than 2 times injection pressure (more than 500 bar) to reach the same atomization level of isooctane (200 bar) [25]. Therefore, water injection pressure is vital for ensuring good atomization. This could well explain the difference between PWI and DWI. Due to the different injection pressures of PWI (5-25 bar) and DWI (50-200 bar), a much better atomization performance could be foreseen for DWI for its remarkably higher injection pressure. It has been proved that DWI shows greater ability in cooling the charge [10-12, 15].

It's worth noting that, even with the same droplet size, water droplets probably never fully vaporize before combustion like gasoline, since water has a lower saturated vapor pressure. The temperature reduction by the vaporization of water would lower the vapor pressure, thus imposing negative effect on mixture formation and vaporization process. Therefore, mixture formation and vaporization processes should be analyzed.

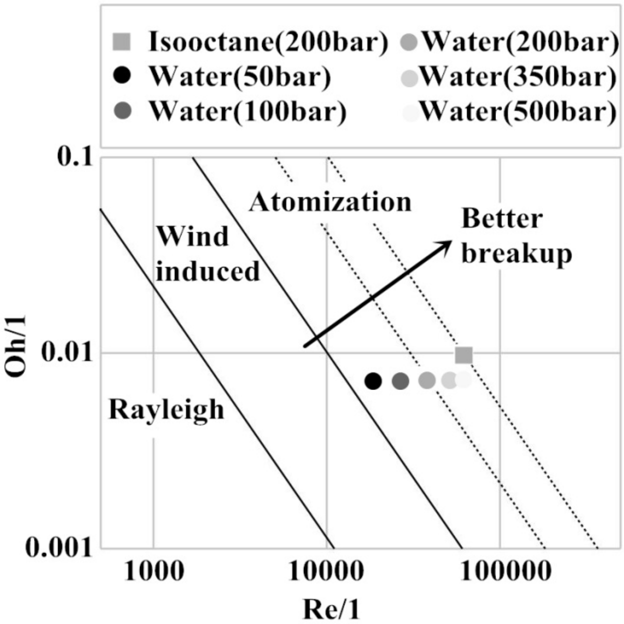

Fig. 5. Ohnesorge–Reynolds diagram of primary breakup regions for isooctane and water under different injection pressures [25].

2.2.2 Mixing and distribution

According to the numerical analysis on a DWI system by Raut et al. [48], water takes at least 15-20 CAD to start vaporizing in all water injection arrangements. It indicates that, for the first 20 CAD after start of injection, water droplets is basically collecting within the combustion chamber and mixing with the air-fuel mixture. Therefore water's spatial distribution within the cylinder is of great significance for a better cooling effect. Raut et al. [48] suggested that a decent spatial distribution of water should adhere to two rules. Firstly, a high water vapor concentration is needed at the cylinder wall and lower around the spark plug. Secondly, there needs to be a uniform distribution of water vapor near the cylinder wall. Berni et al. [28] reported that an improved fuel stratification resulted in leaner end gases (to raise the knock resistance) and somewhat richer λ close to the spark plug (to promote flame kernel development). Battistoni et al. [23] reported that by improving the atomization level, a more homogeneous water distribution is fulfilled to achieve better cooling effect. Otherwise, water only cools a small area and the decreased temperature inhibits further vaporization of water.

2.2.3 Vaporization

Due to its high saturation pressure, gasoline is highly volatile which shows a good evaporation performance even at environment temperatures. Water has much lower saturation pressure and thus a poor vaporization behavior could be foreseen at environment temperatures. According to the numerical analysis by Vacca et al. [11] , vaporization of water is much slower even when water is injected earlier (390 CAD BTDC) than gasoline (320 CAD BTDC). At ignition timing, only 16% of total injected water is vaporized, while gasoline was almost fully vaporized (98% of the total injected quantity).

The atomization level also affects the vaporization process. Numerical analysis by Falfari et al. [47] indicated that finer droplets induced remarkable increase of vaporization rate. They adopted different injection pressures (10, 50 and 200 bar) to produce different sizes of droplets (Sauter mean diameters (SMD) of 30, 20 and 10 μm). Their results indicated that 10 μm case achieved about 8 times higher evaporation rate compared with 30 μm case at 450 K. When it comes to a higher temperature such as 650 K, the effect was even more profound. It is worth to mention that when the ambient temperature remains at a relatively low level (300 K), the effect of droplet size becomes less important. This means proper thermodynamic condition is a precondition for good vaporization. When the ambient temperature is not suitable for water vaporization, finer atomization no longer contributes to better vaporization anymore.

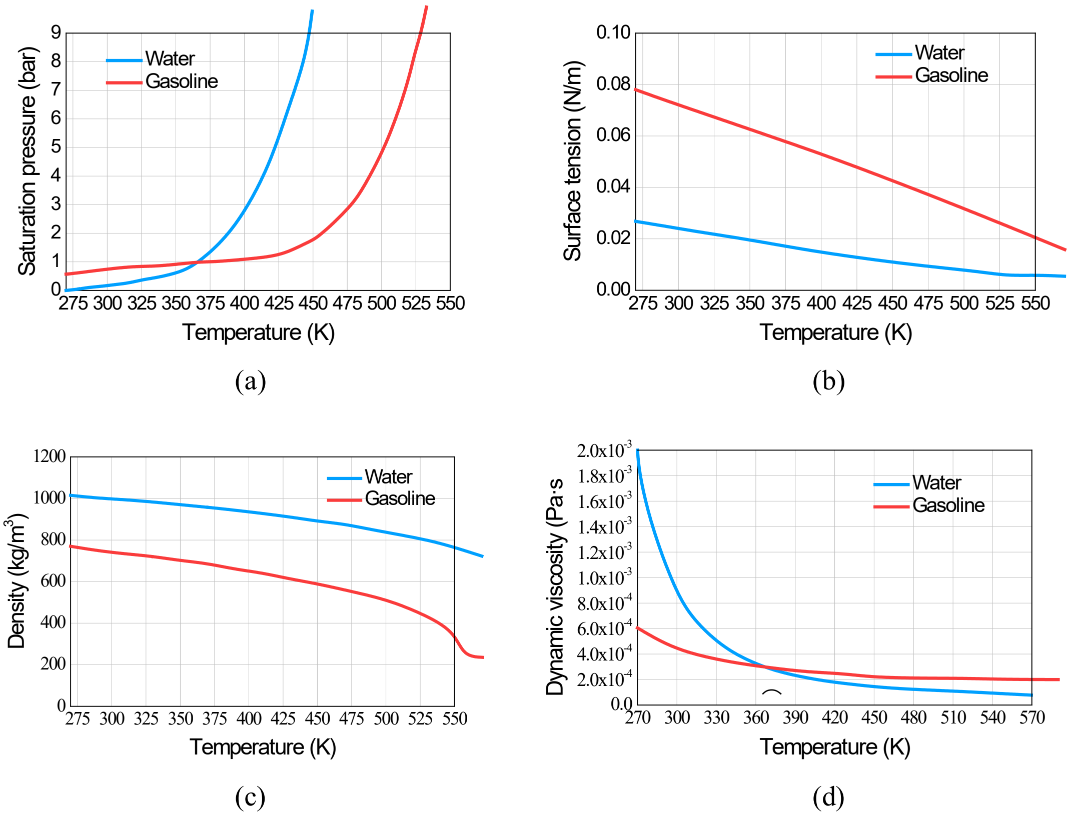

Fig. 6. Comparison of water and gasoline properties: (a) saturation pressure, (b) surface tension, (c) density, (d) dynamic viscosity.

According to **Fig. 6**, the saturation pressure of water is lower than gasoline until the temperature reaches 370 K, suggesting that higher ambient temperature is needed for water to vaporize to the same extent of gasoline. In fact, Falfari et al. [47] suggested that temperature should be higher than 450 K, which shows a prominent improvement of vaporization even with a low injection pressure under 10 bar. They stressed that the higher ambient temperature is a crucial precondition for good vaporization.

Actually, the evaporation rate and saturation limits rely on the location of the water injector or, more precisely, the position where liquid water evaporation begins. IWI mainly makes water vaporize in the intake pipe/runners and DWI makes water vaporize directly in the combustion chamber, the thermodynamic condition of which is of great difference. Investigations [11, 12] have proven that DWI shows greater influence on the combustion, indicating a better vaporization performance compared with IWI, mainly due to the higher ambient temperature inside the cylinder and the better atomization level. It is also suggested that for IWI, putting the water injector as close to the inlet valve as possible makes a quasi-direct water injection effect [11, 23], since the ambient temperature is higher for water to vaporize. According to Boehm et al. [34], a 4% water injection rate through the plenum valve could cool down the intake air by approximately 14 K at a relative humidity of 70% in the intake plenum. This configuration does not allow more injected water to vaporize in the intake air. An increment in the amount of water injection causes water to condense as a wall film, or the droplets merge to create larger drops that are harder to vaporize. Currently, two popular configurations are close to or far from the intake valves. Also there are some other measures to improve the vaporization process. Due to the shortened vaporization period with the increase of engine speed, water vaporization becomes harder and may impose strong negative effect on combustion. According to investigation by Breda et al.[49], methanol plus water addition enhances the vaporization due to better evaporation characteristic of methanol.

2.3 Parameters affecting water injection efficiency

Water injection efficiency was defined as the water fraction effectively involved in the working gas cooling process [12]. Existing studies have found that, water affects engine performance mainly through its thermal effects [12]. Inappropriate injection parameters would lead to undesirable results, such as wall wetting, high water consumption, and poor engine stability [45].

To achieve the maximum water injection efficiency, relevant parameters should be discussed separately. It is worth noting that although each parameter matters, the final effect of water injection depends highly on the synergy of injection position, timing, pressure and quantity.

Concerning the injection position, in section 2.2.3 water injection efficiency has been discussed in detail. To summarize, DWI has higher water injection efficiency than PWI. For PWI, putting water injector as close to the inlet valve as possible would improve the water injection efficiency. With regard to the influence of injection timing, Berni et al. [28] found that for PFI engine equipped with PWI system, the optimal start of water injection lies in the exhaust stroke at 470 CAD BTDC, about 100 CAD before inlet valve open (IVO). At this timing, the maximum vaporized water within the chamber allows the charge to reduce to the lowest temperature at the end of the compression stroke. It suggests that water needs time to fully mix with air-fuel mixture to vaporize. Too early injection would cause water film on the intake runner and too late injection would induce insufficient atomization and vaporization time. Given the fairly low water injection pressure and the high-speed working setting, water should be injected before opening of the intake valve. As for the injection pressure, water injection efficiency has been fully discussed in section 2.2.1 and 2.2.2. Higher injection pressure is always wanted since this could improve atomization, mixing and finally the vaporization of water. In regard to water quantity, more water brings higher cooling effect. However, the over injected water may deteriorate combustion and cause serious issue to the engine, which will be discussed in section 6.

3. Thermal and chemical effects of water injection on SI engines

3.1 Thermodynamic effect of water injection

3.1.1 Cooling effect

With reference to **Table 2**, the main thermodynamic properties of water are compared with that of gasoline [15]. Water presents six times higher heat of vaporization compared to gasoline fuel. Therefore, water acts as a cooling agent to effectively reduce the charge temperature in SI engines, which could inhibit detonation and eliminate fuel enrichment operation. The charge cooling effect of water injection is well documented in investigations [19, 23, 28, 49, 50]. Berni et al. [28] reported that, 9.2mg per cycle PWI reduces the charge temperature before ignition timing (710 CAD) from 760K to 738K. Netzer et al. [15] investigated the charge cooling effect of PWI and DWI systems. They found that 0.8 W/F ratio PWI reduced the charge temperature in the end of compression stroke from 736K to 720K. DWI showed significantly greater ability in charge cooling, which lowered the charge temperature in the end of compression stroke from 736K to 668K. Similar results could also be found in [23, 25, 26, 45, 49].

Although water injection shows significant charge cooling effect in real engine applications, the theoretical cooling ability of the liquid water should be even larger than it actually appears to be. According to numerical analysis by Battistoni et al. [23], only 20% of the total cooling ability of water was exploited. Hunger et al. [45] reported that, the working gas temperature in the end of compression stroke calculated on the basis of thermodynamic properties of water injection is approximately 20K lower than the measured temperature, suggesting that theoretical cooling effect of water is not entirely fulfilled under real-world conditions. The incomplete utilization of water injection's charge cooling effect could be attributed to several factors. As discussed in section 2.2, due to the poorer atomization level of liquid water compared with gasoline fuel, it is difficult for water to fully diffuse and vaporize before ignition timing, especially when the injected water quantity is large and/or the ambient thermal condition is not ideal. Therefore, a part of liquid water is not fully vaporized until the main combustion process proceeds, which would make little contribution to the cooling of unburned gas. On the other hand, due to the poor atomization of liquid water and/or in-cylinder charge motion and/or improper water injection timing, the in-cylinder distribution of the liquid water may be inhomogeneous, which further induces wall wetting phenomenon. The charge cooling effect is weakened, when wall wetting occurs, since part of the heat extraction effect is from the engine components instead of from the working gas. This is verified in the simulation by Mingrui et al. [50]. Their simulation results indicated that when water is injected by 0.25 W/F mass ratio, the temperature could be decreased by more than 50K, 9K and 6K for the piston crown, exhaust valves and combustion chamber, respectively. Additionally, the cylinder movement would impose effects on the change of water properties. Bertolini [46] found that, physical properties of water vary with in-cylinder temperature, as shown in **Fig. 7**. This change may also impose negative impacts on atomization. Lastly, the in-cylinder pressure increases during the compression stroke, which negatively affects the phase change of water. Numerical investigation by Rohit et al. [41] revealed that the enthalpy of vaporization of water varies with pressure, as shown in **Fig. 7**. This would lead to decrease of the enthalpy of vaporization of water, finally inducing bias from theoretical computation.

To summarize, in order to fully exploit the charge cooling effect of water injection, it is imperative to obtain optimal water vaporization and to avoid wall wetting. Great efforts have been made in improving the charge cooling effect of water injection, such as selection of better injection timing [28, 45, 51], choice of water injection methods [15], raising the injection pressure [23] and reasonable parameters (location, orientation, spray patterns, etc) of the injector [28, 52] .

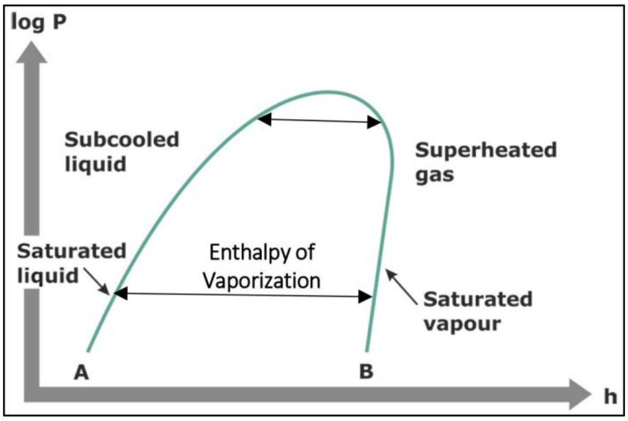

Fig. 7. Pressure-Enthalpy curve for water showing the decreasing latent heat of vaporization with increasing pressure [41].

3.1.2 Specific heat

Besides its large enthalpy of vaporization, water has high specific thermal capacity, which also helps control the in-cylinder temperature. When water is added into the combustion chamber, the overall thermal capacity is increased since water has larger thermal capacity than both the intake air and injected gasoline fuel. Therefore, the in-cylinder temperature increment will be decreased under the same heat absorption condition. Additionally, the cooling effect enabled by high specific thermal capacity is not limited by the phase change of water. This means both liquid and vapor water have temperature control ability, which is an advantage compared with the cooling effect enabled by high enthalpy of vaporization.

However, the effect of specific heat capacity is remarkably lower than it appears to be, especially when compared with enthalpy of vaporization. According to theoretical calculation of Hoppe et al. [25], under the W/F ratio 0.5 condition, the cooling effect of vaporization is four times more than that of the specific heat capacity. This is because the injected water mass is relatively small compared to the mixture mass. Hunger et al. [45] reported that, when the W/F ratio is 0.3, the charge's overall thermal capacity is only increased by 2% to 4%, which is almost negligible. The same conclusion is also made by Kim et al. [24] that tiny quantity of water injection poses comparatively low effect on the charge's overall thermal capacity. It is worth mentioning that effect of dilution and specific heat capacity could be seen when steam is injected into the engine. To investigate the water vapor effect on engine combustion, Cesur et al. [53] conducted both numerical and experimental research on a twin-cylinder SI engine. Their results indicated that the peak in-cylinder temperature was reduced by 67K from 2279 to 2212 K at 2400rpm and reduced by 78K from 2382 to 2304 K at 3600rpm. The influence of latent heat of vaporization has little effect since the water injected is in vapor form. The cooling effect could be attributed to the specific heat and dilution effect of water steam. Specifically, the dilution effect dominates, which will be discussed in detail in section 3.1.3.

3.1.3 Dilution effect

Dilution is a physical process which reduces the concentration of the existing solution by adding more solvent. Usually the solution and solvent are in liquid or vapor form. Since dilution effect enables reducing concentration of certain component, it imposes effects on combustion phenomenon. As a common method for compression ignition (CI) engines to control NOx emissions, EGR recycles part of the exhaust gas into the combustion chamber to dilute the concentration of oxygen in the charge. Similarly, the mixture is diluted when liquid water is added into the combustion chamber and vaporized into steam. Of course, directly injecting the water vapor into the engine also has dilution effect, however the great thermal effect of water is not utilized. It has been widely proved that the dilution effect of water injection imposes prominent impacts on combustion process of SI engines. Specifically, the dilution effect of water injection leads to the reduction of laminar burning velocity [54-56], elongated ignition delay time [36, 48] and combustion duration [12, 23]. Combustion phasing is also delayed due to the dilution effect of water injection [12, 24].

Dilution effect of water injection on engine combustion could be primarily attributed to two aspects. Firstly, the vaporized water increases the total mass of charge in the cylinder, which effectively dilutes the relative concentration of oxygen in the charge. According to a study by Hyundai Motor, the oxygen concentration in the charge could be reduced significantly from 21% to 16% by dilution effect of water injection [24]. In addition, the injected water leads to increase of trapped cylinder mass and raise of the overall heat capacity, just as EGR does. This helps control the in-cylinder temperature. However, since the total dilution rate of water injection (commonly lower than 10%) is remarkably lower than EGR (commonly 20-30%), the heat capacity effect is negligible, as discussed in section 3.1.2. The total dilution rate could be expressed as equation (4).

$$Dil_{m,total}=\frac{m_{dil}}{m_{dil}+m_{air,\,AFR=1}} \qquad (4)$$

The dilution rate is only approximately 7% when W/F ratio is as high as 1 [12], which is considerably lower than that of EGR. Falfari et al. [57] indicated that when the W/F ratio varies in the range 0.1-0.5, the mass of water is only about 1.2% to 3.1% of the total trapped mass inside the cylinder. Cordier et al. [12] also indicated that when the dilution rates are kept the same, water injection shows much larger influencing ability on the combustion phasing compared with EGR. The primary reason may be the temperature difference of dilution gas. The cooling effect of water liquid is also a contributor. Similar results were also found by Cazzoli et al. [55] that under the same dilution rate, water injection leads to lower laminar burning speed compared with EGR. Therefore, the dilution effect of water injection mainly works through its oxygen dilution effect.

3.2 Chemical effect of water injection

When water is injected into the manifold or directly into the cylinder, the cooling and dilution effects would reduce the combustion speed and thus prolong the combustion duration. Most researchers take water as inert material, which does not consider the chemical reaction. However, the chemical effect of injected water on combustion should not be ignored [40]. Harrington tried to separate the physical and chemical effects of water on a single cylinder engine by injecting liquid water and water vapor into the same engine [58]. It's proved that water did impose chemical effects on the combustion, albeit pretty small compared with dilution and thermal effects. Some may argue that since water is the product of hydrocarbon combustion, water is already present in the combustion. However, water is not produced until relatively late in the combustion process, which has little potential in affecting the combustion process. Nowadays, new technologies such as spectroscopic and chemiluminescence measurements have made possible the observation of some crucial intermediate species, which would remarkably contribute to the better understanding of the elementary reactions and chemical mechanisms in combustion [59] . Also, the advancement of computer simulation provides us opportunities to analyze the chemical effect of water on combustion.

Water addition was found to affect chemical kinetic reactions and thus imposed chemical effect on combustion. Since gasoline fuel is a compound of several hydrocarbons, it is difficult to understand the detailed chemical kinetic reactions during combustion process. Attempts have been made to reveal the chemical effects of water on combustion mechanisms based on the simple hydrocarbon combustion.

Through measuring and simulating the laminar flame speeds of $CO/H_2/O_2/H_2O$ mixtures by Bunsen burner method, Meng et al. [60] investigated the influence of water addition on CO/H_2 combustion characteristics. They found that chemical effects of water on combustion were also of great importance especially under large water addition conditions, where reaction paths were prominently changed. According to their research, the chemical effects of water on combustion could be attributed to direct reaction effect and indirect effect, between which exist competing effects. The direct effect dominates when small quantity of water is added into the flame, which promotes the reaction rate of reaction R1 and thus yields more OH radicals. The increased OH radicals are beneficial to the reaction R2, which is the main CO oxidation reaction. Therefore the direct chemical effect of water enhances the combustion process and the promotion effect is more obvious at high CO/H_2 ratios due to the enhancement of reaction R2. However, when the injected quantity is large, the indirect effect of water becomes strong and starts to change the reaction paths. The reactions R4 and R5 are hence intensified, producing more HO_2. Consequently reactions R6 and R7 are strengthened, leading to the reduction of the concentrations of H, O and OH radicals in the flame, thus bringing the chemical reaction to the end. A series research of them [61] indicated that water addition accelerated reaction R1 and promoted combustion under CO content >42% conditions, but it inhibited combustion by reducing the reaction rate of R3 under CO < 42% conditions.

$$H_2O+O \rightleftharpoons OH+OH \quad (R1)$$
$$CO+OH \rightleftharpoons CO_2+H \quad (R2)$$
$$O+H_2 \rightleftharpoons H+OH \quad (R3)$$
$$H_2O(+H_2O) \rightleftharpoons H+OH(+H_2O) \quad (R4)$$
$$H+O_2(+H_2O) \rightleftharpoons HO_2(+H_2O) \quad (R5)$$
$$HO_2+O \rightleftharpoons O_2+OH \quad (R6)$$
$$HO_2+OH \rightleftharpoons O_2+H_2O \quad (R7)$$
$$H+O_2 \rightleftharpoons O+OH \quad (R8)$$

Park et al. [62] numerically investigated the chemical effects of steam on methane-hydrogen-air diffusion flames. They found that the chemical effects of water vapor inhibited the radicals of H and O, but augmented OH radical. They stressed that the dominant H_2O-related reaction step is reaction R1, which was also verified in [63-65]. Specifically, water addition enhances the reaction R1 and suppresses the principal chain branching reaction R8, which is an indicator of overall reaction rate. It is worth mentioning that when small quantity of water vapor (mole fraction of 0.1) is added into the flame, the maximum flame temperature is slightly increased, which could be attributed to the increase of OH radicals. However, when the mole fraction is increased, chemical kinetic pathways could be varied and finally induce the suppression of the combustion process.

According to the above review, it could be seen that water does impose chemical effects on combustion. However, due to the complex composition of gasoline, there are rare investigations regarding influence of water injection on the combustion mechanisms of gasoline fuel in SI engines. Therefore, in the following sections, chemical effects of water on simple hydrocarbon fuel combustion would be reviewed according to the different effects on engine performance, especially emission control and knocking onset.

4. Effects of water injection on combustion

Water affects combustion mainly through its thermal, dilution and chemical effects. Some parameters are remarkably critical to SI engine combustion, such as the minimum ignition energy, auto ignition delay time, and burning velocities.

Combustion in SI engine is premixed combustion, which means its combustion velocity is determined by flame propagation speed. According to knocking theory, raising the flame propagation speed helps reduce knock propensity. For the SI engines, burning velocity is commonly characterized by the laminar burning velocity and turbulence burning velocity, both of which determine the heat release rate [66]. Combustion phases such as flame development (CA10) and main combustion duration (CA10-90) are also important parameters to characterize the combustion process.

4.1 Laminar burning velocity

As a fundamental property of combustion in SI engines, laminar burning velocity (LBV) characterize the reactivity, diffusivity and exothermicity of oxidizer-fuel mixture [67]. The LBV of a mixture exclusively depends on the fuel composition, the mixture quality expressed by equivalence ratio and dilution, and thermodynamic state in terms of temperature and pressure. In SI engines, combustion process typically lies in the flamelet regime, therefore the LBV is an important parameter to calculate the combustion development in turbulent conditions [67].

Firstly, it is worthwhile to recall this parameter's crucialness for engine combustion. In SI engines, the preliminary stage of combustion is the quasi-laminar flame development of the original ignition kernel [68]. Therefore, the duration of this phase (practically represented by the crank angle duration taken to burn the first 2, 5, or 10% of the cylinder charge) is to the first order inversely proportional to the LBV. Previous studies have demonstrated that this phase has a significant influence on the stability of the combustion, which affects cyclic variation. It is demonstrated by the manner in which the dilution tolerance rises with LBV [68]. The primary combustion phase is governed by turbulent flame propagation, LBV also influences this phase, even though it is a smaller level [68]. A proper LBV is of great significance for SI engines. Lower LBV usually means not ideal combustion and may cause problems such as low speed pre ignition (LSPI), knock, misfire, and cyclic variation. Increasing heat transfer from wall to coolant and deviation of the combustion from ideal phasing are also the main problems of low combustion speed. Higher LBV indicates faster flame propagation, which contributes to a higher knock resistance by shortening the time for end-gas to reach auto ignition condition. Shorter combustion duration brought by higher LBV also increases combustion efficiency. Higher LBV also brings a higher tolerance for dilution, which means leaner operation or higher EGR ratio is allowed. However, too high LBV is also undesired, since this would induce crude operation due to too high pressure increase rate and may be harmful to engine components.

The current literature commonly shows empirical measurements of LBV. However, in all experiments, the initial pressure and temperature are regulated at low values because of the test conditions. In general, the pressure for detecting LBV is less than 25 bar, and the temperature is seldom over 550 K. The development of highly accurate chemical kinetic models combined with the significant advances in computer performance facilitates numerical predictions of the laminar flame speed over a different conditions for a series of fuel mixtures. Cazzoli et al. [67] numerically investigated LBV under engine-like conditions and reported that LBV was highly sensible to air fuel ratio (AFR). Therefore, dilution of the mixture either by EGR or water injection would induce a remarkable reduction of LBV, which has been reported [11, 15, 55, 57, 69, 70].

Water injection imposes negative effects on LBV, which could be attributed to its dilution and cooling effects. Water is usually compared with EGR, since they both slow combustion down through dilution effect. Falfari et al. [57] performed an assessment by comparing the LBV of normal air-fuel mixture, diluted mixture with EGR and diluted mixture with water. They found that water vapor had greater ability in slowing down the combustion than EGR, which was also observed by Paltrinieri et al. [69] and Cazzoli et al. [55]. They indicated that this may be caused by the remarkably higher temperature of exhaust gas than water vapor.

When water is injected into the combustion system, the cooling and dilution effects would significantly reduce the LBV, thus affecting the combustion. When other parameters are kept unchanged, the dilution of the air-fuel mixture results in lower LBV, thus increasing the combustion duration and the exhaust gas temperature more than using fuel enrichment. However, since the water injection technology greatly reduces the possibility of detonation onset, further advanced spark timing is allowed, which even compensates the negative effect of LBV reduction on combustion.

The investigation of chemical effect of water on gasoline combustion is rare, especially on LBV. However, a few researchers have studied the effect of adding water into simple hydrocarbon fuels, such as ethylene. According to [38], the existence of water inside the fresh charge slows the oxidation process within the flame thickness, leading to lower laminar flame speed. Liu et al. [71] numerically investigated the effects of adding water vapor to the air stream on flame characteristics in a laminar co-flow ethylene/air diffusion flame. Moreover, it was also found that there were small straightforward chemical effects of water vapor on the laminar burning velocity. Furthermore, the authors also indicated that, in the laminar diffusion flame studied, the order of importance of the four mechanisms of adding water vapor to the oxidizer stream included dilution, chemical, thermal, and radiative. Adding water vapor to the oxidizer stream provides an effective way to diminish radiation heat loss from the flame.

It is remarkably difficult to measure the LBV in real combustion. Combustion duration, usually characterized by mass fraction burned (MFB) 10-50, is commonly used to represent combustion speed. Combustion duration is inversely proportional to LBV, meaning that higher LBV would induce shorter combustion duration. LBV is decreased as water involves into the combustion. Moreover, the injection method also imposes effect on LBV. Vacca [11] found that, DWI achieved higher flame speed than IWI or PWI.

4.2 Combustion phasing

Combustion phasing, commonly characterized by combustion center (CA50 or MFB50), is of great importance in SI engine combustion. An appropriate combustion phasing would enhance the engine combustion efficiency. Generally, too late combustion phasing leads to lower combustion efficiency, which induces poor engine power and fuel economy performance. In addition, higher exhaust gas temperature will also be a critical issue, which imposes severe thermal stress on turbocharger and three way catalyst (TWC), consequently causing permanent damage of engine pipe-out components and deterioration of emission performance. Too early combustion phasing is also not wanted, since this would cause severe knocking combustion. Normally, CA50 between 8~10CAD ATDC is regarded optimal for SI engines to achieve the best thermal efficiency. However, optimal CA50 is not achievable in some parts of the whole engine map, primarily due to the detonation onset.

Combustion center of SI engines is predominantly determined by spark timing [72]. An optimal combustion center would be achieved by advancing the spark timing. However, this is usually limited by knock onset, which is called knock limited spark advance (KLSA). Thanks to the anti-detonation effect, water injection helps extend the KLSA range and accomplishes optimal combustion phasing. Teodosio et al. [13] reported that, water injection had the greatest potential in obtaining the best combustion phasing compared with variable valve actuation (VVA), VCR and EGR techniques.

On the other hand, water injection reduces LBV and thus induces elongated combustion as discussed in section 4.1. Thanks to the significant knock mitigation ability of water, larger spark advance could be applied to ensure the optimal combustion phasing. Compared with combustion speed, combustion phasing contributes much more to the indicated efficiency (approximately 3 times larger influence than combustion speed) [12]. Therefore, the merit of optimized combustion phasing outweighs the demerit of lower LBV brought by water addition. Several studies have pointed out that water injection combined with KLSA benefit improve combustion phasing and thus enhance engine performance.

To assess the water injection effect on engine combustion enhancement, some studies investigated the relationship between combustion phasing and fuel consumption performance. Cordier et al. [12] observed an improvement both of the combustion phasing and fuel economy with the increase of W/F ratio on an engine equipped with PWI system at working point 2000rpm and 17bar IMEP. Water injection advanced the CA50 from 32 to 23 CAD ATDC (9CAD increment) combined with 8.8% indicated specific fuel consumption (ISFC) improvement when W/F ratio was increased to 1. It's noteworthy that the advance of CA50 and improvement of ISFC become smaller when W/F ratio was over 0.5, which could be attributed to the fact that vaporization of water becomes harder when the water injection quantity is increased. Similar results were also found by Hunger et al. [45] on a single-cylinder engine equipped with DWI system. Their results indicated that ISFC was improved by 10.5% through advancing the combustion center with up to 0.5 W/F ratio water injection at 2500 rpm and IMEP = 20 bar. Additionally, they suggested an almost linear correlation between injected water quantity and CA50 advancement. Approximately 0.05 W/F advanced combustion center by 1 CAD. Numerical simulation helps reduce the experimental efforts and gives broader views on the water injection effect on engine combustion phasing at various load points. De Bellis et al. [21] numerically investigated the water injection effect on combustion phasing with PWI system under 8-18bar brake mean effective pressure (BMEP) and 3500rpm conditions. The maximum break torque (MBT) operation (i.e. ideal spark advance) was achieved from low or medium loads (8-13 bar BMEP) by utilizing the increased knock resistance and improved combustion phasing of water injection. When it comes to higher loads (14 to 18 bar BMEP), MBT operation could no longer be maintained with water injection due to the limitation of in-cylinder peak pressure (80 bar). However, water injection still enabled 10 to 15 CAD advance of combustion center compared with no water injection operation, achieving remarkable brake specific fuel consumption (BSFC) improvements (approximately 20%). The BSFC reduction brought by water injection could be attributed to better combustion phasing and stoichiometric operation, which will be discussed in detail in the section 4.3. On the other hand, the combustion phasing benefits on the engine performance could be reflected from the maximum power output. Iacobacci et al. [20] indicated that 0.3 W/F ratio PWI induced 0.7 CAD advance of combustion center at 3500rpm, achieving 7.3% IMEP increase. At 4000rpm and 4500rpm, 2.4 and 2.9 CAD advance of CA50 was achieved, resulting in 3.2% and 2.6% IMEP gain, respectively. It could be seen that the enhancement of IMEP is weakened at higher engine speeds, which could be attributed to the less favorable combustion phasing. Therefore for higher speeds condition, larger spark advance should be applied with increased water quantity to ensure a better combustion phasing. However, this operation would cause higher peak in-cylinder pressure, which is limited at 85 bar as suggested by the manufacturer. It is worth mentioning that further advancing the spark timing is also restricted by other issues such as the peak in-cylinder pressure (commonly 80 bar) [20, 26],cycle variation and/or wall wetting.

The water injection effect on combustion phasing is affected by injection methods, injection pressure and injection timing. With regard to the injection methods, DWI has been proved to impose greater influence on combustion phasing than IWI does. Cordier et al. [12] indicated that DWI allowed further 5 CAD advance of combustion center than IWI with the same W/F ratio, indicating its greater potential in advancing combustion phasing and, thus achieving further approximately 5% improvement of fuel consumption compared with PWI. The gas temperature at 30 CAD BTDC showed the greater ability of DWI in cooling the air fuel mixture before combustion, which provides perspective on the higher cooling efficiency along with the further extension of KLSA by DWI with regard to PWI. The higher potential of DWI compared with IWI in combustion phasing was also verified by Vacca et al. [11] and Hunger et al. [45]. In addition, Vacca et al. [11] indicated that IWI shows greater potential in exhaust gas temperature reduction.

The start of injection of PWI is found to affect the final effect of water injection, especially for quasi-direct water injection. As discussed in section 2.3, proper start of injection is crucial for achieving high water injection efficiency [28] and better combustion phasing. To assess the effects of water injection timing on combustion phasing, Hoppe et al. [25] and Hunger et al. [45] experimentally and numerically investigated start of injection effect of DWI on combustion. They indicated that start of injection during compression stroke (approximately 110 CAD BTDC) achieved the best combustion center at constant water injection quantity. Simulation results revealed that working gas temperature variation was well correlated with MFB50 variation, indicating that maximizing the cooling effect of water on the air-fuel mixture helps mitigate detonation onset and thus allows for an optimal combustion phasing. Even the early injected water started to vaporize at approximately 120 CAD BTDC due to the poor vaporization before compression stroke. The early injected water is inevitably impinged on the combustion chamber wall or cylinder liner, extracting heat from engine components instead of directly from the charge. When it comes to the compression stroke, too late start of injection also lowers the influence on combustion phasing due to insufficient time for water injection and vaporization.

To assess the effect of water injection pressure on combustion phasing, Hunger et al. [45] conducted experiments on a single-cylinder test engine. Their results indicated that higher injection pressure was beneficial to better combustion phasing. Raising the pressure from 40 bar to 100 bar obtained around 4 CAD advancement of combustion center and further raising to 140 bar made little difference. This suggests that 100bar is reasonable value for DWI.

Based on detailed chemistry, Netzer et al. [15] modelled the impact of different parameters on KLSA.

According to the results, laminar flame speed, heat of vaporization, chemical equilibrium, water vapor heat capacity, third body efficiency and, ignition delay time were significant factors with a downward order of significance.

Water is usually compared with EGR, since they both slow combustion down by dilution effect. However water injection has far greater influence on combustion phasing. Cordier et al. [12] found that, the ability of water to delay combustion was much weaker although the combustion duration was extended as the same with EGR. Moreover, regardless of the injection methods (i.e. DWI vs IWI), water injection is far superior to EGR in terms of the ability of advancing combustion center and improving fuel economy. Through numerical investigation, Bozza et al. [26] compared the water injection and cooled EGR effect on combustion phasing. Their results indicated that water injection has greater ability than cooled EGR in achieving better combustion phasing because vaporization heat of water brings much higher cooling effect. Thus better BSFC performance is achieved due to more advanced CA50. However, water injection quantity is limited especially at higher engine speed by peak in-cylinder pressure and exhaust gas temperature. Moreover, EGR requires increase in the boost pressure to recover the cylinder filling penalties. However, water injection does not need any boost level increase, and even could be decreased. Similar results are also found in [17] that water has greater influence on combustion center than EGR.

4.3 Combustion duration

SI engine is premixed combustion, which includes mixing of fuel and air, spark ignition, and flame propagation. The shorter the combustion process is, the closer the SI engine to the ideal Otto cycle is. In SI engines, the flame propagation speed is usually evaluated by combustion duration, which is commonly characterized by CA90 or MFB10-90. Apparently, the shorter the combustion duration is, the faster the flame propagation process is. Therefore, shorter combustion duration would enhance the engine combustion efficiency.

As discussed in sections 4.1 and 4.2, water injection reduces the LBV when spark advance is kept constant. This would lead to elongated combustion duration and poor thermal efficiency. However, due to the significant knock mitigation ability of water injection, earlier start of combustion could be achieved through advance of spark timing [11, 24]. With recalibration of spark advance, combustion duration is also changed. Vacca et al. [11] reported shortened the combustion duration by using up to 0.5 W/F ratio water injection plus earlier spark advance. Although the overall combustion duration with water injection was always shorter than baseline, their experiments indicated that the combustion duration optimization effect was weakened with the increase of injected water quantity due to the poor vaporization of water. Similar results could also be found in [24, 26]. These researches indicated that water injection plus larger spark advance operation induces optimized flame propagation process. However, bias would exist due to the difference in experimental instruments and working condition or the error of simulation. Some researchers reported that water injection lengthened the CA90 with advanced spark timing [11, 73]. The combustion duration was lengthened with the load increase [11, 21, 24, 73, 74]. In general, little influence is imposed on the combustion duration through water injection plus larger KLSA operation. On the other hand, the benefit of optimized combustion phasing dominates, finally contributing to the enhancement of engine efficiency.

There are several factors which could affect the water injection efficiency on the combustion duration [12]. As discussed in section 4.2, DWI has greater potential in achieving better CA50 than IWI. As for the CA90, it presents a two-stage performance between DWI and PWI according to the injected water quantity. When the injection quantity is comparatively low (W/F ratio = 0.1-0.2), DWI imposes little effect on CA90 however PWI lengthens the combustion duration. When the water quantity is increased (W/F ratio = 0.3-1), PWI keeps CA90 at a constant level while DWI starts to slow down the combustion and finally exceeds PWI. It is worth mentioning that DWI always has larger CA50 and better fuel economy compared with PWI. Small amount of water could be fully vaporized and PWI provides a more homogeneous mixture than DWI, therefore dilution effect of PWI turns out to be the contributor to combustion slow down. While large quantity of water could not be fully vaporized for both DWI and PWI, the dilution effect of DWI exceeds the PWI in lengthening the combustion duration. Hoppe et al. [25] experimentally investigated the effect of injection timing and pressure on combustion duration. 120 CAD BTDC start of injection achieved the shortest CA90, which could be primarily attributed to the influence of CA50. Up to 100 bar pressure led to decrease of combustion duration, which should be mainly attributed to the forward of CA50. However, when the pressure was 150 bar, the CA50 was no longer further advanced and thus more homogeneous mixture caused by higher injection pressure helps enhancing the dilution effect, which would increase the combustion duration.

Importantly, it should be stressed that when too much water was injected, the combustion duration ultimately increased when the dilution's negative effect exceeded the advantage gained from advanced spark timing. Due to the aberration from the perfect theoretical constant volume combustion process, higher combustion duration was not beneficial for both IMEP and ISFC.

4.4 Ignition delay

Ignition delay, commonly characterized by CA10, is used to express the flame initiation process. Ignition delay is influenced by several factors, e.g. fuel properties, air-fuel ratio, in-cylinder temperature and pressure, CR, gas flow, spark energy, and residual gas composition. According to the combustion theory, the ignition delay in SI engines should be as short as possible so that the flame is quickly initiated to achieve better control ability of combustion process. Ignition delay in SI engines could be shorted by advancing the spark timing. Therefore, spark advance or CA50 normally attracts more attention than CA10.

However, several engine parameters are tightly connected with ignition delay time. The increase of ignition delay could result in an exponential increase in coefficient of variation (COV) of IMEP if a critical value is achieved, which has previously reported in the existing literature [75]. Also, misfire happens when ignition delay is seriously elongated. Additionally, longer ignition delay inhibits the flame initiation process, which finally increases the knocking propensity especially at high load condition where thermal stress is extremely high. This has been verified by Mehl et al. [76] that the anti-knock index (AKI) decreased with the increase of ignition delay time. Therefore, the impact of water injection on ignition delay should be reviewed in detail.

Water acts as cooling agent and inert gas when added into the engine. This leads to the decrease of the mixture temperature and reduces the chemical reactivity, and consequently increases the ignition delay. Netzer et al. [15] indicated that the ignition delay is increased with increase of water injection quantity. In addition, W/F ratio of 1 led to the increase of the ignition delay by approximately 14% in the temperature range from 800K to 900K. The authors also stressed that the sensitivity to the presence of water is highest at 750 K. The increase in ignition delay time for a W/F ratio of 1 parallels to 16 CAD for 2500 rpm and 26 CAD for 4000 rpm. However, the sensitivity reduces as the temperature increases. The ignition delay time is insensitive to the addition of water at or above 950 K. Li et al. [77] also indicated that the ignition delay time increased with the increase of water injection quantity at 5500rpm with constant spark timing of 21 CAD BTDC and constant equivalence ratio of 1.1. Results indicated that 0.25 W/F ratio led to the increase of ignition delay from 13 to 14.5 CAD due to the reduction of average temperature and pressure in the cylinder. Paltrinieri et al. [69] found that, W/F ratio 1 water vapor injection increased the ignition delay by 7%. It is noteworthy that W/F ratio 1 is relatively high (commonly in the range was below 0.5). Evidently, the ignition delay time under engine relevant conditions is not largely affected by a small quantity of water addition. Although ignition delay was reported remarkable increase (around 5CAD) with increase of water quantity (from W/F ratio 0 to 0.8) in [10]. The reason should be primarily attributed to the large A/F ratio variation.

Similar to combustion phasing and combustion duration, the little elongated ignition delay could be recovered by applying advancement of spark timing. In some studies [17, 24], a prominent decrease of CA10 was reported. Experiments by Kim et al. [24] on a PFI engine with DWI indicated that W/F ratio from 0 to 1 monotonically decreased the CA10 from 15 to -1 CAD ATDC.

Numerical investigation on a 2 stroke gasoline engine indicated that from 0 to 0.4 W/F ratio monotonically decreased the CA10 from 16 to 5 CAD ATDC [17]. There exists a same point between the two cases, namely extremely large CA50 forward. The former is from 27 to 10 CAD ATDC while latter is from 31 to 17 CAD ATDC. The extreme forward of combustion center overcomes the dilution effect by water to induce a large decrease of ignition delay. Interestingly, a consistent monotonic decrease trend of CA90 and CA10 with combustion center forward is found in both the results in [17, 24]. This suggests that CA50 has some influence on both CA10 and CA90. According to [75, 78, 79], a clear association does not exist between the 0-10% burn duration variability (standard deviation), and the standard deviation of the 10-90% burn duration (or otherwise the COV of IMEP). However, the authors reported that the first combustion event, 0-10% burn duration, is the reason for identical behavior of the second combustion event, 10-90% burn duration, which is beyond this article’s scope.

On the other hand, some researchers reported the increase of CA10, albeit to an extremely low degree. D’Adamo et al. [73] observed that the ignition delay is only 0.8 CAD increase by PWI. Due to the competing effect of the above-mentioned diminished effect on laminar flame speed at ignition and the increment in spark advance. In their simulation, the IMEP was kept the same, so the spark advance may not be forwarded enough. Further advance of spark timing would cause earlier CA50, which induces change of ignition delay. Hunger et al. [45] indicated that when keeping the same IMEP, water injection induced ignition delay decrease by 0.7 CAD at the same IMEP.

In SI engines, this happens in the vicinity of spark plug since the ignition delay involves the initial flame propagation process. Therefore the mixture condition has great influence on the ignition delay. The inconsistence of current investigation results could be partially attributed to the inhomogeneous mixture or the turbulence difference near the spark position. Although this has not been verified be experiments, computational fluid dynamics (CFD) analysis could be helpful to reveal the water injection effect of turbulence and mixture distribution. Initial attempts are made [28]. According to numerical investigation by Berni et al. [28], the flame kernel development could be improved through proper fuel injection modification. Owing to the decrease of fuel enrichment enabled by water injection, fuel could be injected later to maintain the same end of the injection (EOI) time as original to realize improved fuel stratification, leading to leaner end gases (to increase the knock resistance) and a more affluent λ close to the spark plug (to encourage flame kernel advancement).

Finally, it should be stressed that water injection with spark advance retuning operation is proved to impose remarkably weaker effect on CA10 than effect on CA50 and CA90 [45]. **Table 4** compares the combustion characteristics of water injection effects on SI engines.

Table 4 Comparison of combustion characteristics of water injection effects on SI engines.

References	Comparison baselines*	Engine conditions (CR, induction, speed, AFR, spark timing, load)	Combustion performance*		
			CA0-10	CA50	CA10-90
Hoppe et al. (2016) [25]	GDI+DWI *vs* GDI	13.5:1, aspirated, 2000rpm, λ=1, KLSA(maintain optimal CA50 at 7-8° CA ATDC), IMEP=10.5bar		←	↔
		13.5:1, aspirated, 3000rpm, λ=1, KLSA(maintain optimal CA50 at 7-8° CA ATDC), IMEP=14.6bar		←	=
		13.5:1, aspirated, 2000rpm, λ=1, KLSA(maintain optimal CA50 at 7-8° CA ATDC), IMEP=22.6bar		←	→
		13.5:1, aspirated, 3000rpm, λ=1.4, KLSA(maintain optimal CA50 at 7-8° CA ATDC), IMEP=14.6bar		←	=
Kim et al. (2016) [24]	GPI+DWI *vs* GPI	13.5:1, aspirated, 2000rpm, λ=1, KLSA, BMEP=7bar	←	←	=
		13.5:1, aspirated, 1500rpm, λ=1, KLSA, WOT			←
		13.5:1, aspirated, 2000rpm, λ=1, KLSA, WOT			←
		13.5:1, aspirated, 2500rpm, λ=1, KLSA, WOT			→
		13.5:1, aspirated, 3000rpm, λ=0.89-0.93, KLST, WOT			→
Iacobacci et al. (2017) [20]	GPI+PWI *vs* GPI	10:1, boosted, 3500rpm, λ=0.90, KLSA(SA 13-17CAD BTDC), WOT		↔	
		10:1, boosted, 4000rpm, λ=0.93, KLSA(SA 15-19CAD BTDC), WOT		↔	
		10:1, boosted, 4500rpm, λ=0.89, KLSA(SA 15-21CAD BTDC), WOT		↔	

Hermann et al. (2018) [10]	GDI+PWI *vs* GDI	Unknown, boosted, 3000rpm, λ≤1, KLSA, WOT	→	←	→
		Unknown, boosted, 4000rpm, λ≤1, KLSA, WOT	→	←	→
		Unknown, boosted, 5000rpm, λ≤1, KLSA, WOT	→	←	→
Hunger et al. (2018) [45]	GDI+DWI *vs* GDI	10:1, boosted, 2500rpm, λ=1, SA is kept to maintain the same IMEP with basis case, IMEP=20bar	←	←	→
		10:1, boosted, 2500rpm, λ=1, SA is kept to maintain the same CA50 with basis case, IMEP<20bar	→	=	→
Cordier et al. (2019) [12]	GDI+PWI *vs* GDI	12.5:1, aspirated, 2000rpm, λ=1, KLSA, IMEP=17bar		←	→
	GDI+DWI *vs* GDI			←	→
Golzari et al. (2019) [36]	GDI+PWI *vs* GDI	11.43:1, boosted, 1000rpm, λ=1, KLSA, medium load(NIMEP=8.83bar)	→	←	=
		11.43:1, boosted, 2000rpm, λ=1, KLSA, medium load(NIMEP=16.04bar)	→	←	→
		11.43:1, boosted, 2000rpm, λ=1, KLSA, high load(NIMEP=20bar)	→	←	→
		11.43:1, boosted, 3000rpm, λ=1, KLSA, medium load(NIMEP=16.04bar)	→	←	→
		11.43:1, boosted, 3000rpm, λ=1, KLSA, high load(NIMEP=20bar)	→	←	→
Fan et al. (2020) [37]	GDI+PWI *vs* GDI	10.5:1, boosted, 2000rpm, λ=1, KLSA, 170Nm		←	←
		10.5:1, boosted, 3500rpm, λ=1, KLSA, 170Nm		←	←
		10.5:1, boosted, 5000rpm, λ=0.88 gradually to λ=1, KLSA, 155Nm		←	→
Miganakallu et al. (2020) [38]	GDI+PWI *vs* GDI	10.93:1, aspirated, 1500 rpm, λ=1, KLSA, NIMEP=8bar	→	←	→
Zhuang et al. (2020) [80]	GDI+PWI *vs* GDI	11.7:1, boosted, 1500rpm, λ=1, KLSA, WOT		←	←
		11.7:1, boosted, 4850rpm, λ=1, KLSA, WOT		←	←

* **Symbols:** ←, advanced; →, delayed or elongated; ↔, depended on operation conditions; =, insignificant change; Nil, not reported; λ<1, rich combustion

5. Effect of water injection on emissions

To protect the environment and meet more stringent regulatory, tail-pipe emissions are always a critical issue. The regulated exhaust emissions comprise of carbon monoxide (CO), hydrocarbons (HC), nitrogen oxides (NOx), and particulate matter (PM) [81]. Existing measures to control these emissions include post-treatment measures, such as TWC, selective catalytic reduction (SCR) systems, and particulate filters.

The use of water as an emission control agent in combustion applications was proposed in the early 1960s

[82]. Water injection has shown great ability in suppressing NOx emission. However, higher HC emission is always a penalty for NO suppression due to their opposite formation mechanisms. Water injection enables stoichiometric combustion, thus the catalytic converter can operate inside its high conversion efficiency window. So HC, CO and NOx emissions could be reduced significantly.

5.1 NO_x emission

NO formation mechanisms have been well documented in literatures such as [72, 83-90]. SI engines commonly work around stoichiometric conditions and thermal NO is the main path of NO emission. Thermal NO formation chemistry is described well by extended Zeldovich mechanism which comprised the reactions (R9-R11).

$$O+N_2 \rightleftharpoons NO+N \qquad (R9)$$

$$N+O_2 \rightleftharpoons NO+O \qquad (R10)$$

$$N+OH \rightleftharpoons NO+H \qquad (R11)$$

The primary factor governing NO formation is the combustion temperature, to be more specific, the peak in-cylinder temperature. Thus any parameter inducing variation in the peak temperature of the burned gas will impose a significant impact on the NO formation. These factors are cooling, dilution, spark timing (or combustion phasing) and engine speed [91]. The second important contributor affecting NO emission is oxygen concentration. Water injection has long been proved to suppress NO formation due to its physical and chemical effects [82, 92-94].

With regard to physical effect, cooling effect of water injection directly lowers the mixture temperature, which effectively suppresses thermal NO formation. Besides direct cooling effect, water injection imposes significant effect on combustion phasing, which has been discussed in section 4.2. The final temperature of the burned gases is a function of the time when combustion occurs. Late combustion phasing induces lower peak in-cylinder pressure and temperature, with the penalty of increased exhaust gas temperature and poor combustion efficiency. In addition, water injection can enable higher AFR under high load conditions, because fuel enrichment to thermally protect pistons and catalysts would be attenuated. The change of equivalence ratio affects the NO emission. Among all the three factors above, direct cooling effect is expected to decrease NO emission and the influence on equivalence ratio is expected to increase NO emission. Combustion phasing effect could be either positive or negative for the NO emission performance, which is determined by the specific spark advance modification. The final effect of water injection on NO emission is the synthetic effects of above three factors.

With regard to chemical effect, water vapor could reduce the O radical concentration through scavenging reaction (R1) [95]. Since O radicals are crucial for initiating reaction (R9) and OH radicals do not attack N_2 efficiently, thermal NO is thus inhibited.

Generally, the final effect of water injection on NO emission is positive, which could be found in studies [12, 14, 20, 24, 25, 36, 37, 39, 45]. However, NO emission performance could be varied due to the competing effects.

In some cases, NO emission is increased [19, 37]. It is also found that even with the same water injection configuration, NO emissions are influenced by engine speed and/or load conditions [14, 20, 25], showing an varied emission performance. Therefore due to the difference of experimental design and engine specifications, different NO emission results are common to see.

5.2 HC emission

HC formation mechanism has been well documented in literature [96]. In a warmed-up engine, the engine-out HC emissions at part load are caused by: crevices (~38%), oil layers and deposits (~16% each), flame quenching (~5%), and in-cylinder liquid fuel effects (~20%), exhaust valve leakage (＜7%). HC emissions are incomplete combustion products and/or part of hydrocarbon fuels. Therefore high temperature, high oxygen and sufficient reaction time contribute to complete combustion and thus suppress HC emission. As discussed in section 5.1, these factors are negative for NO emission control. Consequently, it is very difficult to fulfil the simultaneous reduction of NO and HC emissions without compromises. HC and NO emissions usually showed opposite trends [12, 14, 19, 20, 24, 25, 36, 37, 45]. In general, NO is mostly decreased while HC is increased [12, 20, 24, 25, 36, 37, 45].

To be specific, water injection impose both physical and chemical effects on HC emission. Physically, the cooling effect leads to the temperature reduction of mixture and incomplete combustion, which finally increase HC emission. In addition, the cooled cylinder liner may increase the flame quenching and thus higher HC emission. Also, when earlier spark advance is applied, the raised IMEP could increase crevice HC emission. Chemically, OH radicals are produced through reaction (R1). Since OH radicals are good oxidizing agents and could effectively react with unburned hydrocarbon fuel, the HC emission is hence to be suppressed. The final effect of water on HC emission is the synthetic effect of physical and chemical effects.

The extent of the water injection influence on HC emissions usually depends on the specific design features of the engine. Additionally, different water injection methods and specific configurations also impose large effects on HC emissions. Therefore, there could be some different results in HC emission concerning water injection application. For example, advanced spark timing caused high peak in-cylinder pressure and temperature, which should lead to decrease of HC emission. In addition, the higher peak in-cylinder pressure may also induce more mixture trapped in crevice, which combines with the uneven temperature distribution finally may increase HC emissions. The flame propagation may be extinguished by the water or mist concentrated spots which also increased the HC emissions. On the other hand, the participation of water in the combustion process may provide additional OH radicals which decrease HC emissions. Additionally, although stoichiometric combustion would lead to higher HC emission than slight lean combustion, the gaseous emissions can be effectively converted by TWC due to the high working efficiency under lambda one operation.

5.3 CO emission

CO is the output of incomplete combustion and is affected by AFR, fuel type, combustion chamber design, engine load, and speed [97]. CO emission could be influenced by water injection due to several aspects.

Firstly, at full load condition where fuel enrichment is applied, the lack of oxygen under rich combustion is the main cause of CO. The injection of water reduces or eliminates the need for fuel enrichment, thus reducing CO emissions. Tornatore et al. [19] reported that, CO emission was largely decreased by more than 80% since stoichiometric combustion was achieved with W/F ratio of 0.2. Their research also showed that the effect of spark timing was almost negligible compared with the variation of AFR. Similar results could also be found in [10, 24] that CO emission is significantly decreased by moving AFR to the stoichiometric combustion. Furthermore, the dilution effect of injecting water eliminates the local areas abundant in fuel, such as the area within proximity of the injector tip and spark plug which has a lower local AFR, achieving a more homogeneous oxygen distribution and hence more complete combustion and reduced CO emission. However, research regarding this aspect is rare. The authors suggest that numerical simulation would be helpful to address this issue.

Since CO and HC are both the results of incomplete combustion, there exist connections with each other. Generally, CO emission showed high similarity to HC emission in SI engines when water was injected [12, 14, 19, 20, 34, 37].

5.4 Soot emission

When oxygen is absent at temperatures in excess of 673 K, hydrocarbon fuels exhibit a strong tendency to form soot [50]. Therefore, soot emission is severe for CI engines due to its working mode. Combustion in conventional PFI SI engines is premixed combustion, so the mixture is mostly homogeneous and therefore not prone to generate soot. Compared with conventional PFI SI engines, direct injection spark ignition (DISI) engines can achieve much better power performance and fuel economy [98]. The increasingly stringent legislations force the widespread of GDI engines. Additionally, downsized boosted SI engine is more attractive thanks to its higher thermal efficiency. However, boosted DISI engines generally have higher soot emissions than PFI engines [99], which is attributed to the incomplete fuel volatilization and mixing in DISI engines [100].

Water injection was proven effective in soot reduction in diesel engines [101, 102], which suggests an idea to adopt water injection on gasoline engines for soot suppression. Several researchers have investigated the water injection effect on soot reduction of gasoline engines. The soot reduction effect of water injection can be attributed to several reasons. Firstly, water injection eliminates the need for fuel enrichment operations under high load conditions. Therefore stoichiometric combustion could be implemented in wider operating areas, and thus soot is reduced [10]. Secondly, water injection could potentially improve the mixing process of air-fuel mixture inside the cylinder, which helps form a more homogeneous mixture and thus suppress soot formation. Moreover, water would be decomposed into O, H and OH radicals at high temperature during the combustion stroke. As discussed in section 3.2, the concentrations of OH and O radicals are increased while the H radical is decreased, both enhance soot reduction. Therefore, soot emissions and the concentrations of the polycyclic aromatic hydrocarbons experience a drastic reduction. Water injection effect on soot inhibition is also well documented in studies such as [34, 36, 50]. **Table 5** compares the emissions performance of water injection effects on SI engines.

Table 5 Comparison of emissions performance of water injection effects on SI engines.

References	Comparison baselines*	Engine conditions (CR, induction, speed, AFR, spark timing, load)	Emissions performance*			
			CO	HC	NOx	PN
Boehm et al. (2016) [34]	GDI+DWI *vs* GDI	11:1, boosted, 5500rpm, λ=0.82 gradually to λ=1, KLSA, IMEP=22bar	↓	↓		↓
Hoppe et al. (2016) [25]	GDI+DWI *vs* GDI	13.5:1, aspirated, 2000rpm, λ=1, KLSA(maintain optimal CA50 at 7-8° CA ATDC), IMEP=10.5bar		↑↓	↓	
		13.5:1, aspirated, 3000rpm, λ=1, KLSA(maintain optimal CA50 at 7-8° CA ATDC), IMEP=14.6bar		↑	↑	
		13.5:1, aspirated, 2000rpm, λ=1, KLSA(maintain optimal CA50 at 7-8° CA ATDC), IMEP=22.6bar		↑	↓	
		13.5:1, aspirated, 3000rpm, λ=1.4, KLSA(maintain optimal CA50 at 7-8° CA ATDC), IMEP=14.6bar		↑	↑↓	
Kim et al. (2016) [24]	GPI+DWI *vs* GPI	13.5:1, aspirated, 1500rpm, λ=1, KLSA, WOT	↓	↑	↓	
		13.5:1, aspirated, 2000rpm, λ=1, KLSA, WOT	↑↓	↑	↓	
		13.5:1, aspirated, 2500rpm, λ=1, KLSA, WOT	↓	↑	↓	
		13.5:1, aspirated, 3000rpm, λ=0.89-0.93, KLST, WOT	↓	↑	↑↓	
Iacobacci et al. (2017) [20]	GPI+PWI *vs* GPI	10:1, boosted, 3500rpm, λ=0.90, KLSA(SA 13-17CAD BTDC), WOT	↑	↑	↑	
		10:1, boosted, 4000rpm, λ=0.93, KLSA(SA 15-19CAD BTDC), WOT	↑	↑	↓	
		10:1, boosted, 4500rpm, λ=0.89, KLSA(SA 15-21CAD BTDC), WOT	↑	↑↓	↓	
Tornatore et al. (2017) [19]	GPI+PWI *vs* GPI	10:1, boosted, 2500rpm, λ=0.89 without WI and λ=1 with WI, SA 8CAD and KLSA(9-11CAD BTDC), WOT	↓	↓	↑	
		10:1, boosted, 3000rpm, λ=0.92 without WI and λ=1 with WI, SA 12CAD and KLSA(13-18CAD BTDC), WOT	↓	↓	↑	
		10:1, boosted, 3500rpm, λ=0.89 without WI and λ=1 with WI, SA 14CAD and KLSA(15-20CAD BTDC), WOT	↓	↓	↑	
		10:1, boosted, 4000rpm, λ=0.91 without WI and λ=1 with WI, SA 15CAD and KLSA(16-20CAD BTDC), WOT	↓	↓	↑	
		10:1, boosted, 4500rpm, λ=0.89 without WI and λ=1 with WI, SA 14CAD and KLSA(17-19CAD BTDC), WOT	↓	↓	↑	
Hunger et al. (2018) [45]	GDI+DWI *vs* GDI	10:1, boosted, 2500rpm, λ=1, KLSA, IMEP=20bar		↑	↓	
Sun et al. (2018) [14]	GDI+PWI *vs* GDI	9.5:1, boosted, 3000rpm, λ=1, KLSA, high load(BMEP=14bar)	↑	↑	↓	
		9.5:1, boosted, 5000rpm, λ<1, KLSA, full load(BMEP=18.1bar)	↓	↓	↑	
Cordier et al.	GDI+PWI *vs*	12.5:1, aspirated, 2000rpm, λ=1, KLSA, IMEP=17bar	↑	↑	↓	

(2019) [12]	GDI					
	GDI+DWI *vs* GDI		↓	↑	↓	
Golzari et al. (2019) [36]	GDI+PWI *vs* GDI	11.43:1, boosted, 1000rpm, λ=1, KLSA, medium load(NIMEP=8.83bar)	↓	↑	↓	↓
		11.43:1, boosted, 2000rpm, λ=1, KLSA, medium load(NIMEP=16.04bar)	↓	↑	↓	↓
		11.43:1, boosted, 2000rpm, λ=1, KLSA, high load(NIMEP=20bar)	↓	↑	↓	
		11.43:1, boosted, 3000rpm, λ=1, KLSA, medium load(NIMEP=16.04bar)	↓	↑	↓	
		11.43:1, boosted, 3000rpm, λ=1, KLSA, high load(NIMEP=20bar)	↓	↑	↓	
Fan et al. (2020) [37]	GDI+PWI *vs* GDI	10.5:1, boosted, 2000rpm, λ=1, KLSA, 170Nm	↑	↑	↓	↓
		10.5:1, boosted, 3500rpm, λ=1, KLSA, 170Nm	↑	↑	↓	↓
		10.5:1, boosted, 5000rpm, λ=0.88 gradually to λ=1, KLSA, 155Nm	↓	↓	↑	↓
Zhuang et al. (2020) [39]	GDI+PWI *vs* GDI	11.7:1, boosted, 1500rpm, λ=1, KLSA, WOT(IMEP=50bar)	↓	⥮	↓	⥮

* **Symbols:** ↑, increased; ↓, decreased; ⥮, depended on operation conditions; Nil, not reported; λ<1, rich combustion

6. Challenges and future research directions

6.1 Challenges

Although water injection has shown great potential in enhancing the thermal efficiency and emission control of SI engines, it is still not a matured technique for mass production automobiles due to the following three challenges. Firstly, suitable water injection control and water supply systems should be developed for complex engine operating conditions. Attempts have been made to address the water supply issue [14, 34, 103-105]. There are currently four methods to maintain the water supply, as suggested by VW [103] and BMW [34], namely manual filling, surface water, condensation in the air conditioning system and collecting water from the exhaust gas, as shown in **Fig. 8**.

Moreover, water wall wetting or impingement is another concern to be addressed. When water could not be vaporized quickly especially under excessive injection quantity or higher engine speed, water condenses as wall film or the small droplets join up to form bigger ones, leading to serious wall wetting. Wall wetting issue was numerically investigated in [23, 45, 46, 57, 69, 106, 107]. Generally, wall wetting can be mitigated by adopting proper injection strategies to improve the vaporization of injected water. Even if some liquid water impinges the cylinder wall, the high temperature will accelerate the vaporization process. However, this leads to decreased water injection efficiency, since the heat is extracted from cylinder wall instead of from the charge. Lastly, long-term tests should be performed to identify the reliability and durability of the water injection system and the impacts of water injection on engine life. There are still many theoretical and engineering problems to be handled for the wide commercialization of the water injection system on SI engines.

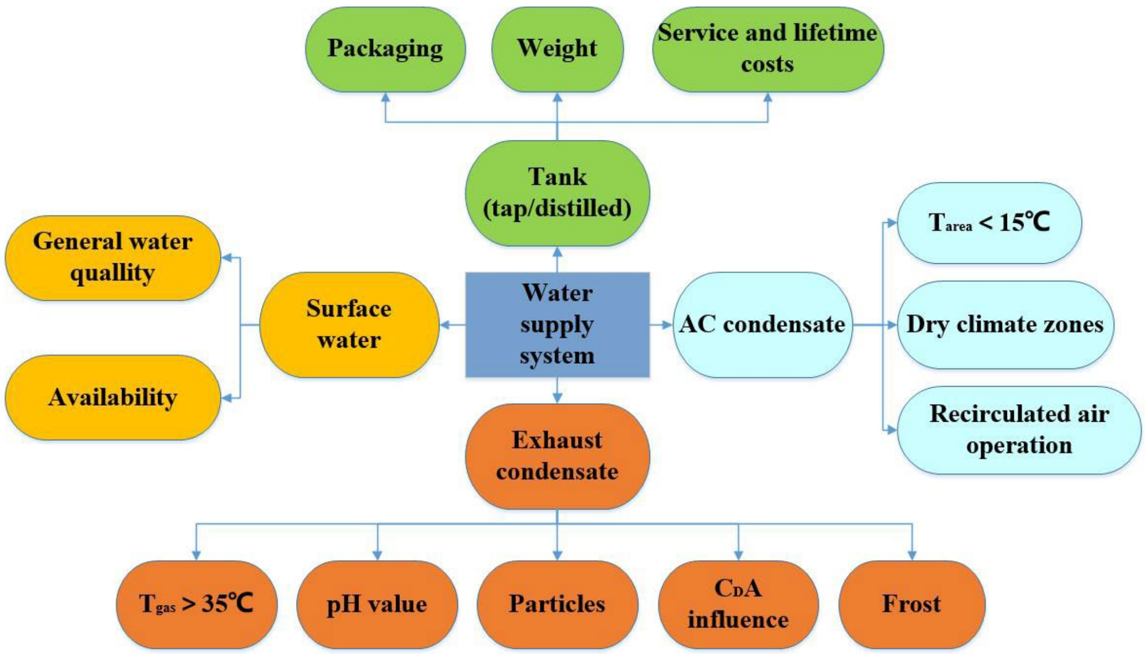

Fig. 8. Overview of water supply systems.

6.2 Future research directions

To further enhance the thermal efficiency of SI engines, water injection regains the research attention. More researchers are investigating this topic. Some research directions are very attractive and promising. For example, water injection enables super lean combustion mode in SI engines to achieve higher thermal efficiency. Experimental research by Nagasawa et al. [108] reported that water injection enables single cylinder research engine to achieve thermal efficiency of 52.63%, which is higher than conventional CI engines (35%-45%). Additionally, water injection has the potential to enable controlled auto ignition (CAI) [109] or called homogenous charge compression ignition (HCCI), which is a promising combustion mode with higher thermal efficiency than SI engines while lower NO_x and soot emissions than CI engines [110]. The investigations regarding this issue are well documented in [109-117]. Thermally stratified compression ignition (TSCI) can also be fulfilled by water injection technology, which can be found in literatures [107, 118]. Duel injection mode is also suitable for application of water injection system, such as water/methanol injection [38, 49, 119, 120] or water/hydrogen injection [121-125] to further improve the working efficiency of SI engines. Water injection can also be compared or combined with other technologies such as Miller cycle, VCR, VVA, EGR. Teodosio et al. [13] numerically assessed these technologies' effect on working efficiency of SI engines. Similar studies can be found in [25, 26, 54, 126, 127]. However, most of these studies were conducted on research engines or by numerical analysis. Combined effects of these technologies applied on mass production GDI engines should be investigated in detail. Also, a combination with other technologies on enhancing fuel economy and emissions should be investigated. The effect of these technologies on new test cycles such as real driving emissions (RDE) and worldwide harmonized light vehicle test procedure (WLTP) should be evaluated. To the best of the authors' knowledge, the investigation regarding this area is rare [35, 74].

7. Conclusions

This article systematically reviews the mechanisms of water injection on SI engine performance, including different water injection methods, water evaporation process, thermal and chemical effects of water injection, and water injection effects on combustion and emissions performance of SI engines. The main conclusions are drawn as follows:

1. Due to its remarkable cooling and anti-knock abilities, water injection technology reduces the heat stress and knock propensity of SI engines, leading to higher thermal efficiency. Water injection is highly effective especially in high knock propensity areas. It allows more advance of spark timing to achieve optimal combustion phasing, which is normally compromised to suppress detonation onset. On the other hand it helps eliminate the need for fuel enrichment, which is used to thermally protect pistons and catalytic converters.
2. Water can be added into SI engines in various ways, among which PWI and DWI are most widely investigated. Specifically, PWI is cost effective and easy to install on both PFI and GDI engines. However, due to the poor atomization and vaporization, PWI has lower water injection efficiency compared with DWI. In addition, PWI has lower control flexibility and precision. Currently, there is no definite conclusion on which method is better.
3. To maximize the cooling and anti-knock effects, the injected water should be fully vaporized. However, due to the large differences in physical properties, water exhibits poorer atomization and vaporization than gasoline. Therefore, higher injection pressure is needed to ensure high water injection efficiency. Additionally, the injection parameters (injection timing and quantity) and the injector design (location and spray number) are also of great importance. Otherwise, poor vaporization would increase water consumption. Also, negative impacts such as wall wetting may occur and cause engine performance deterioration.
4. The addition of water exerts both physical and chemical effects on engine combustion. Physical effects mainly include charge cooling and dilution. Chemical effects include direct chemical effect and three body effect. Physical effects are well investigated and proved to play the dominate role. Much less attention is paid on chemical effects. There may be complex coupling relationship between physical and chemical effects, which needs further investigation.
5. As a cooling agent and inert material, the injected water reduces the charge temperature and imposes negative impacts on engine combustion. Specifically, flame propagation is slowed down and combustion is worsened.

However, this could be compensated by applying spark advance. It has been proved that water injection plus advanced spark timing could effectively improve the combustion such as better combustion phasing and shorter combustion duration, leading to higher thermal efficiency.

6. In general, water injection helps inhibiting NO formation since thermal NO formation is suppressed by water's cooling effect. HC emission usually shows opposite trend with NO emission due to their quite opposite formation mechanisms. However, the degree of influence on HC emissions through water injection usually depends on the specific design features of the engine and the different water injection methods and parameters. Therefore there could be different results in HC emission concerning water injection application. Water has little effect on CO emission. Experimental researches showed that, the influence of water on engine emission is relatively complex, which is not only related to engine operating conditions, but also water injection methods and parameters.
7. Fundamental knowledge regarding both thermal and chemical effects of water injection on engine combustion and emissions is still lacking. More work is needed on different water injection methods to individual engine type. Suitable water supply and consumption control systems combined with injection strategies should also be developed for various engine operating conditions. Water injection has potentials to enable super-lean combustion mode and other new combustion modes (e.g. CAI/HCCI and TSCI), which are promising combustion modes and should be further investigated.

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