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Ethanol blends in spark ignition engines: RON, octane-added value, cooling effect, compression ratio, and potential engine efficiency gain

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HIGHLIGHTS

- Literature about ethanol blends in SI engines is reviewed.
- Octane-added value is proposed for ethanol blends.
- Charge cooling effect of ethanol fuels is quantitatively assessed.
- Engine efficiency gain from ethanol blends is predicted.

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ABSTRACT

Identifying a sustainable, practical and low-emission energy supply for modern transportation has always been a challenge for energy and automotive researchers. While electrification of the vehicle powertrain is a promising long-term energy supply solution, bio-ethanol is currently playing an important role as a short- and mid-term solution for the popular spark ignition (SI) engine. The questions of how to use ethanol more effectively as an octane booster, how much potential engine thermal efficiency gain can be achieved by using ethanol blends and what their impacts on the vehicle mileage range are have become highly relevant. In this paper, a critical review and discussion regarding these questions is provided. Firstly, studies regarding octane rating and octane index of gasoline fuels, and K value (a scaling factor for calculating octane index) for various SI engines are reviewed. Then, a review of the research octane number (RON), motor octane number (MON) and octane sensitivity for ethanol blends is reported. Three established models for predicting RON of ethanol blends are reviewed and compared. In addition, a simple RON prediction model proposed by the authors of this paper is provided. Parameters such as octane value and octane-added index (OAI) are proposed to describe the effectiveness of using ethanol as an octane booster. It is found that there exits an optimised ethanol blend ratio that gives the maximum octane value; and this optimised blend ratio is insensitive to the octane rating of the base gasoline. Secondly, the charge cooling effect of ethanol blends and its corresponding equivalent octane number are discussed and reviewed. Thirdly, engine thermal efficiency improvement due to increased compression ratios, which results from the octane index gain achieved by using ethanol blends, is reviewed. Finally, a discussion about the impact of ethanol blends on the vehicle mileage range is presented. The lower heating value of ethanol is about 33% lower than that of typical gasoline, leading to a reduction in the mileage range of the vehicle, however, improved engine thermal efficiency achieved by using ethanol blends can partially, or even fully, offset the negative impact of the lower calorific value on the mileage range.

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Nomenclature

Acronyms and abbreviations
AFR air fuel ratio
AKI anti-knock index
BMEP brake mean effective pressure
CFR cooperative fuels research
CR compression ratio
DI direct injection
DISI direct injection spark ignition
DVPE dry vapour pressure equivalent
EGR exhaust gas recirculation
EOI effective octane index
FFVs flex fuel vehicles
HOV heat of vaporization
IMEP indicated mean effective pressure
KLSA knock limited spark advance
LHV low heating value
LMC linear molar-weighted model
LVC linear volumetric-weighted model
MON motor octane number
NA naturally aspirated
NMC non-linear molar-weighted model
NOI normalized octane improvement
NTC negative temperature coefficient
OAI octane-added index
OEMs original equipment manufacturers
ON octane number
ONCE octane number from cooling effect
$ONCE_{gasoline}$ ONCE of base gasoline
$ONCE_{ethanol}$ ONCE of ethanol
OI octane index
OI_{blend} octane index of ethanol blend
PCCE partially captured cooling effect
PFI port fuel injection
PRFs primary reference fuels
RON research octane number
RON_{blend} RON of ethanol blend
RON_{base} RON of base gasoline
$RON_{ethanol}$ RON of ethanol
SI spark ignition
TC turbo-charged
TOV total octane value
TOV_{a_blend} total octane value after blending
TOV_{b_blend} total octane value before blending
VOV volumetric octane value
VVT variable valve timing

Definitions
E′x′ x% volume-based ethanol in the blend; for example, E10 means 10 vol.% ethanol in the blend
K a scaling factor used in the calculation of octane index
Pg measured RON of 50%:50% molar ethanol-gasoline blend
S octane sensitivity, which equals to the difference between RON and MON
x_{mole} molar fraction of ethanol in the blend
x_{vol} volumetric fraction of ethanol in the blend
η engine Thermal efficiency

Octane related parameter
EOI chemical effect + sensitivity effect + cooling effect
Chemical effect RON minus PCCE (RON-PCCE)
Octane sensitivity effect $-K * S$
Cooling effect ONCE

Contents

1. Introduction

Sustained research and development has been conducted on bio-fuels in order to improve the sustainability of energy supplies and to reduce greenhouse gas emissions. The most widely used bio-fuel is bio-ethanol. First generation bio-ethanol is made from arable crops and sugar canes which can be easily converted using conventional technologies [1]. Second generation biofuels are made from lignocellulosic biomass or wood-based crops, agricultural residues or waste, which do not compete with human food consumption. In the following paper, 'bio-ethanol' will be simply referred to as 'ethanol'.

Ethanol has many favourable physiochemical properties which make it the preferred gasoline alternative. It can be used in its pure form in specially designed vehicles, or in blended forms with some vehicle modifications. Numerous studies have reported the effectiveness of ethanol blends in improving engine efficiency, in reducing emissions, such as particulates, carbon monoxide and

Table 1
Fuel properties.

	Unit	Gasoline[a]	Ethanol	Effect of Ethanol
Formula		C_4-C_{12}	C_2H_6O	No aromatic, olefin and sulphur content, less combustion chamber deposit, injector deposit and particle formation tendency [14–16,51]
RON		95+	107	Suppress engine knock [9]
RON		85+	89	
Oxygen content	wt.%	0	34.78	Low emissions, low energy density
Stoichiometric AFR		14.5	9	High fuel mass flow, fuel and wall interaction, and cooling effect
Density @ 15 °C	kg/m^3	720–775	790	High oil dilution [72]
Laminar flame speed[b]	m/s	0.44	0.55	High acceleration response and high engine efficiency
Lower heating value	MJ/kg	42	26.9	High fuel consumption [9]
Flash point	°C	−40	13	Bad cold start
Heat of vaporization @ λ = 1	kJ/kg_air	26	103	Bad cold start, good cooling effect
Initial boiling point	°C	Varies	78	Bad cold start
Reid vapour pressure	kPa	Varies (48–110)	15.5	Bad cold start
Water solubility	%	0	100	Bad cold start, risk of phase separation when ethanol is added into gasoline
Miles reduction relative to gasoline[c]	%	0	33	Potentially high fuel consumption

[a] Typical gasoline available in the EU market.
[b] Measurement condition: 1 bar initial pressure, 50°C initial temperature, and 1.1 equivalence ratio.
[c] Assume that engine thermal efficiencies for gasoline and ethanol are the same.

unburned hydrocarbons [2–13], and in reducing the fuel injection system's deposit formation [14–19]. Some of these benefits are briefly reviewed below.

Table 1 lists a summary of gasoline and ethanol fuel properties, along with direct/indirect impacts of ethanol on engine performance. The high octane rating of ethanol leads to a reduced engine knock tendency, thus ethanol can be used in high compression ratio (CR) SI engines, which results in improved engine thermal efficiency [20–22]. The high octane sensitivity of ethanol is also potentially beneficial for suppressing knock in SI engines when the K values of the engine at certain operating points (typically high load) are negative [19,23–25].

For ethanol, the measured research octane number (RON) and motor octane number (MON) in different publications varies. According to 16 publications from various research institutes and original equipment manufacturers (OEMs) [14,26–40], the reported RON and MON of ethanol generally fall in the range of 106–111 and 89–92, respectively. These variations are believed to be due to uncertainties in the RON and MON measurement methods and variations in the properties of ethanol itself. In the standard octane rating test, it is defined that the RON and MON of n-heptane and iso-octane, the primary reference fuels (PRFs), are 0 and 100, respectively. This definition makes the measurement of octane ratings higher than 100 difficult, relying on the extrapolated octane rating calibration curve, or on using high octane rating reference fuels that contain iso-octane and tetraethyl lead [40]. In addition, ethanol is highly soluble in water [41], which leads to potential water contamination in ethanol and therefore measurement errors. Due to the two above mentioned reasons, it is believed that different laboratories produced different measured octane ratings for ethanol. In this paper, for the purpose of consistency, the RON and MON of ethanol are considered as 107 and 89, respectively, which are the values recommended in John Heywood's widely used book [27].

Apart from the high octane rating, the faster laminar flame speed of ethanol helps to improve the engine's combustion process and reduce knock tendency [42,43]. Its oxygen content helps reduce hydrocarbon emissions, due to increased completeness of the main combustion and post oxidation processes [11], however, it must be noted that its lower heating value (LHV), resulting from its oxygen content, causes in high volumetric fuel consumption. In comparison to gasoline, the higher fuel/air ratio required for stoichiometric combustion and the lower LHV of ethanol leads to higher fuel mass flow and thus longer injection pulse width; as a consequence, there are higher cylinder wall and piston wetting tendencies. High ethanol blend ratios potentially lead to cold-start problems and high gaseous and particulate emissions, due to ethanol's relatively high boiling point, high heat of vaporization (HOV), and low Reid vapour pressure [44–50]. When blended into gasoline, ethanol dilutes its aromatic and olefin contents, therefore, ethanol blends potentially reduce particle emissions and deposit formation [14–16,51,52].

In the European and US gasoline fuel markets, gasoline usually contains 5–10 vol.% ethanol. Modern gasoline vehicles can be smoothly powered by these low percentage ethanol blends. Several studies have been done on engines powered by fuels containing up to 85 vol.% ethanol (E85) [53–55]. The vehicles that are capable of using E85 are typically called flexible fuel vehicles (FFVs). However, current FFVs do not utilize the full potential of high percentage ethanol blends because only engine calibration adaptations such as active ignition management are used; advanced engine hardware adaptations such as variable compression ratio can only be used to extract more benefits from ethanol blends once those technologies become more cost-effective [56,57].

The impacts of ethanol blends on octane rating, engine thermal efficiency and vehicle mileage range are important questions which need to be answered. In this paper, a critical review and discussion regarding the application of ethanol blends in spark ignition (SI) engines is provided. Firstly, studies of the octane rating for gasoline, as well as studies of the octane index and K value for SI engines are reviewed. Secondly, the octane rating of ethanol blends, and three established models for predicting the octane rating of ethanol blends are reviewed and compared. In addition, a simple RON prediction model proposed by the authors of this paper is provided. Thirdly, the charge cooling effect of ethanol blends and the corresponding equivalent octane number are discussed. Fourthly, the engine efficiency improvement due to increased compression ratios (CR), which result from the octane index gain achieved by using ethanol blends is reviewed. Finally, several ethanol blend case studies of engine thermal efficiency gains and vehicle mileage range reduction are discussed for naturally aspirated direct injection spark ignition (DISI) and turbocharged downsized DISI engines.

2. Octane rating

This section is comprised of three parts. Firstly, a literature review of the octane rating measurement history is conducted, and the newly proposed octane index by Kalghatgi, and the K value identification for SI engines are reviewed and discussed. Secondly, octane ratings of ethanol blends are reviewed, along with three established octane prediction models and a model proposed by the authors of this paper. Finally, the optimal ethanol blend ratio for maximizing the benefit of ethanol as an octane booster is discussed.

2.1. History of octane rating and K value identification for SI engines

Historically, the anti-knock characteristics of gasoline-type fuels are described by two parameters, RON and MON, which are measured in standardized single-cylinder naturally aspirated (NA) carburettor SI engines, first designed in 1929 [58–60]. These engines are widely known as cooperative fuels research (CFR) engines. Table 2 lists the summary of standard RON and MON test conditions. Details of the RON and MON test procedures are defined in the ASTM standards D2699-08 and D2700-08, respectively [61,62].

Octane rating is one of the most important parameters for gasoline-type fuels. OEMs choose the engine CR mostly based on the gasoline that is most widely available in the market (so called regular gasoline). In the US, anti-knock index (AKI), which is the average value of RON and MON, is used to regulate the octane quality (AKI $\geqslant$ 87). In European countries, the EN228 regulation sets the minimum limits for RON and MON at 95 and 85, respectively [63]. However, after nearly 90 years of development, modern SI engines tend to operate at relatively low temperatures and high pressure conditions, resulting from the use of advanced technologies such as DI, charge intercooling and cooled exhaust gas recirculation (EGR) [64,65]. Additionally, the physiochemical properties of the PRFs (iso-octane and n-heptane) used in CFR engines differ from gasolines available in the market, which consist of thousands of various hydrocarbons. The most important difference is that the PRFs have stronger negative temperature coefficients (NTC) than gasoline [66]. Therefore, due to significant deviations in hardware and the fuels used, between the CFR and modern engines, the relevance of RON and MON to modern SI engines needs to be re-assessed.

The significance of RON to modern gasoline engines has been proven by many investigations [9,67–71], however, the relevance of MON has been challenged by more and more research data, especially in the last 10 years [19,23–25]. It was found that for some engine types and at some operating conditions, a low MON could be beneficial to reduce engine knock tendency [23,24,72–74]. To compensate for the possible disagreement between CFR and modern engines results, the octane index (OI) was proposed by Kalghatgi [23]:

$$\mathrm{OI} = \mathrm{RON} - \mathrm{K} \times (\mathrm{RON} - \mathrm{MON}) = \mathrm{RON} - \mathrm{K} \times \mathrm{S} \tag{1}$$

where K is a scaling factor depending solely on the in-cylinder temperature and pressure history experienced by the end-gas prior to the onset of auto-ignition and S (the difference between RON and MON) is the octane sensitivity.

Table 2
Standard RON and MON test conditions [61,62].

Parameter	Unit	RON	MON
Inlet air temperature	°C	52	38
Inlet mixture temperature	°C	–	149
Intake air pressure	bar	Atmospheric	
Coolant temperature	°C	100	100
Engine speed	RPM	600	900
Spark timing	°bTDC	13	14–26
Compression ratio	–	4–18	4–18
Engine displacement	L	0.619	0.619
Stroke/Bore	mm	114.3/83.1	
Intake valve closing	°bTDC	146	
Exhaust valve opening	°aTDC	140	

The engine K value can be determined by either experimental or modelling data. The experimental method relies on the correlation of an engine/vehicle performance parameter relating to a fuels' auto-ignition property with a fuel matrix where RON and MON are decorrelated. The typical engine/vehicle performance parameters are knock limited spark advance (KLSA) and acceleration time. Details regarding the experimental method can be found in [23,24,73,75–77]. For the modelling method, in-cylinder pressure data is required as an input, which can be used to calculate the in-cylinder temperature. The crank angle of auto-ignition for a matrix of PRFs and Toluene/n-Heptane mixtures (gasoline surrogate) using the Livengood-Wu integral is calculated, and then the OI of PRFs and Toluene/n-Heptane fuel mixtures is determined through the PRF calibration curve. Based on the OI, MON and RON, the K value can be calculated. In the modelling method, the reason that toluene is added into n-heptane to produce the gasoline surrogate is because the mixture has closer physiochemical properties (especially ignition delay characteristics) to gasoline than PRFs. Details about the modelling method can be found in [64,74].

The OI is such that when the K value is positive, high MON is beneficial to suppress engine knock, and vice versa [76]. Many studies have suggested that the K value is not always positive [19,23–25]. Mittal and Heywood [78] tested fuels with various combinations of RON and MON in a single-cylinder port fuel injection (PFI) SI engine with 1 bar intake manifold pressure. The experimental results showed that the K value under the studied operating condition was negative. Remmert et al. [74] studied the octane appetite and K value of a 4-cylinder DISI engine. Seven RON and MON decorrelated fuels were tested at various high load conditions with different external EGR levels, boost pressures, back pressures and air/fuel ratios. They found that under high load conditions (approximately 20–30 bar BMEP) the engine's K value was in the range of −0.26 to −1.14. Davies et al. [64] investigated the K value of several engines under high boost and EGR conditions. They found that the K value was in the range of −0.86 to 0.5. Kalghatgi [75] reported that the averaged engine K value at full throttle conditions was −0.38 for 37 SI engines ranging from NA to turbocharged (TC) and 1.2–2.4 L.

Fig. 1 shows a summary of the engine K value study results. The engine K value is dependent on engine design and technology, engine speed, engine load and intake charge pressure. Fig. 1(a) shows that the K value distributions of historic SI engines, current NA SI engines and current TC DISI engines were different under various engine operating conditions; the average K values were 0.2, 0 and −0.3, respectively. As reported in Fig. 1(b), Mittal and Heywood found that K values of vehicles manufactured between 1951 and 1991 were declining and even became negative, due to the use of advanced cooling and breathing techniques and the replacement of carburettors with fuel injectors [71]. The engine K value consistently increases with engine speed (Fig. 1(c)), and this trend is consistent with different car model years. As reported in Fig. 1(d), Kassai et al. [79] found that K values decreased with the engine intake manifold pressure (or engine load) at both 1200 and 2000 rpm engine speeds; similar results were found in [66,77,80]. Bourhis et al. [80] studied the K value of a TC 4-cylinder 1.6 L DISI engine and found that the K value was positive at part load, and negative at the high engine load. Caroline et al. [77] conducted an octane sensitivity study on a 2007 Pontiac Solstice. They found that there was a clear negative correlation between the K value and intake charge pressure.

Due to variations of the engine K value for different engine design/technologies and engine operating conditions, two K values of 0 and −0.3 will be considered to represent the average engine K value for current NA DISI engines and current TC DISI engines, respectively.

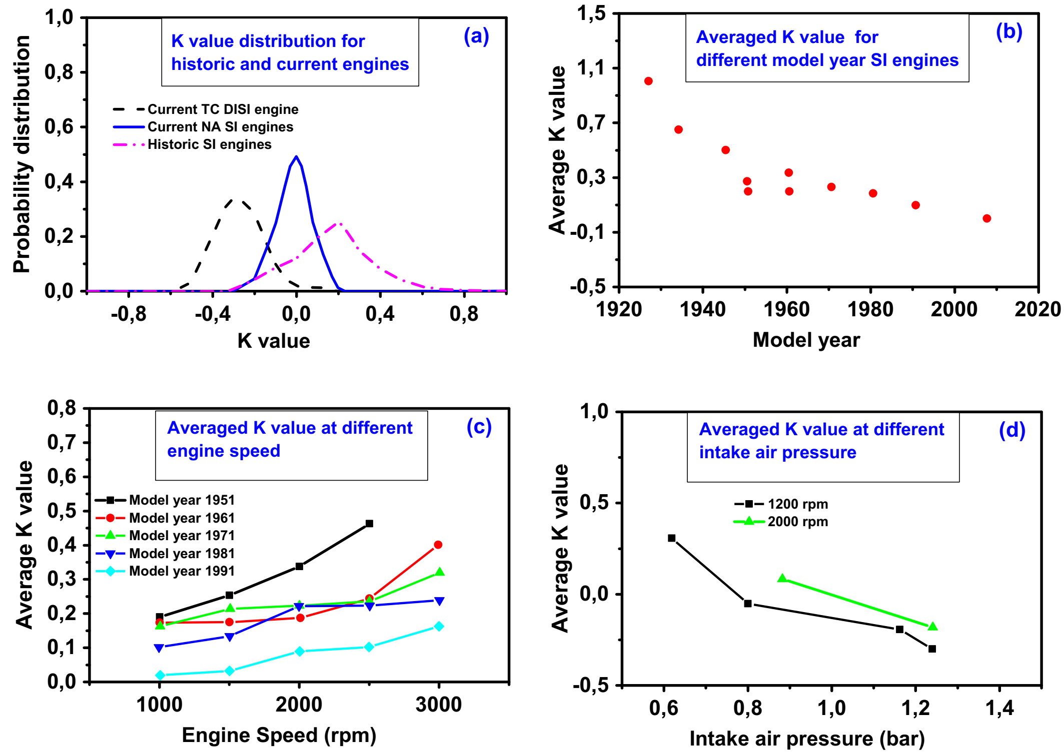

Fig. 1. Summary of K value study: (a) K value distribution for historic and current engines under various operating conditions [71]; (b) averaged K value for different model year SI engines [71]; (c) averaged K value at different engine speed [71]; (d) averaged K value at different intake air pressure [66].

Overall, this section briefly reviewed the history of octane rating and K value identification in SI engines. In the next section, octane rating prediction of ethanol blends will be discussed.

2.2. Octane prediction models for ethanol blends

Due to limited access to CFR engines, measuring the octane rating is not always possible; therefore, in some studies, no octane ratings of ethanol blends were given [7,10,81,82]. In this section, three reported RON and MON prediction models for ethanol blends [26,40,83,84] are discussed and compared, which are: (1) linear molar-weighted model (LMC); (2) linear volumetric-weighted model (LVC); and (3) non-linear molar-weighted model (NMC). Additionally, the authors of this paper have proposed a new and simple model which is discussed later in this section.

The LMC model is described as:

$$\mathrm{RON_{blend}} = (1 - x_{mole}) \times \mathrm{RON_{base}} + x_{mole} \times \mathrm{RON_{ethanol}} \tag{2}$$

The LVC model is described as:

$$\mathrm{RON_{blend}} = (1 - x_{vol}) \times \mathrm{RON_{base}} + x_{vol} \times \mathrm{RON_{ethanol}} \tag{3}$$

where $\mathrm{RON_{blend}}$, $\mathrm{RON_{base}}$ and $\mathrm{RON_{ethanol}}$ are the RON of ethanol blend, base gasoline and ethanol, respectively; x_{vol} and x_{mole} are the volumetric and molar fractions of ethanol in the ethanol-gasoline blend, respectively. For the prediction of MON, the same equations can be used.

The NMC model is described as:

$$\begin{aligned}\mathrm{RON_{blend}} &= (1 - x_{mole}) \times \mathrm{RON_{base}} + x_{mole} \times \mathrm{RON_{ethanol}} \\ &+ \mathrm{P_g} \times x_{mole} \times (1 - x_{mole}) \times (\mathrm{RON_{ethanol}} - \mathrm{RON_{base}})\end{aligned} \tag{4}$$

Compared to the LMC model, the NMC model has an extra non-linear term. $\mathrm{P_g}$ is a scaling factor for the non-linear term, which is determined for each base gasoline by minimizing the sum of squared errors of measured and modelled octane ratings of one or a few ethanol blends. Therefore, the RON/MON of one or a few ethanol blends needs to be measured in order to estimate $\mathrm{P_g}$. According to [40], for RON estimation, the $\mathrm{P_g}$ values were between 0.45 and 0.48 for the four base gasolines used in the study. For the prediction of MON, the same equation can be used with $\mathrm{P_g}$ values in the range of 0.94 and 1.21 [40].

In this paper, after analysing a large amount of data from the literature, a simpler model is proposed with inputs of ethanol volumetric content, RON of the ethanol and RON of the base gasoline. One parameter, normalized octane improvement (NOI), is used as an intermediate parameter for the RON prediction, and it is defined as:

$$\mathrm{NOI} = \frac{\mathrm{RON_{blend}} - \mathrm{RON_{base}}}{\mathrm{RON_{ethanol}} - \mathrm{RON_{base}}} \times 100 \tag{5}$$

Fig. 2(a) shows measured RON versus ethanol content, and those pieces of data were directly extracted from several publications [26,31,85,86]. It can be seen that the octane rating of ethanol blends increases non-linearly with increase of the ethanol volumetric content [26,31,40,85–87], and a higher marginal octane improvement is achieved with low and medium amounts of ethanol addition. This is a result of synergistic effect of ethanol interacting with alkanes in suppressing low temperature heat release. Alkanes have low-temperature oxidation chemistry involving peroxy radicals and hydroperoxyalkyl radicals, which plays a crucial role in their early ignition kinetics and produce the characteristic negative temperature coefficient region and low temperature heat release [88]. Ethanol shows a strong suppression of NTC [89]. The interaction of ethanol with alkanes is the possible reason for the non-linear octane relationship for ethanol blends.

Additionally, the high charge cooling effect of the ethanol may have been partially reflected in the measured RON with the CFR engine. Between ethanol contents of 50 and 75 vol.%, limited octane improvement was observed by adding extra ethanol into

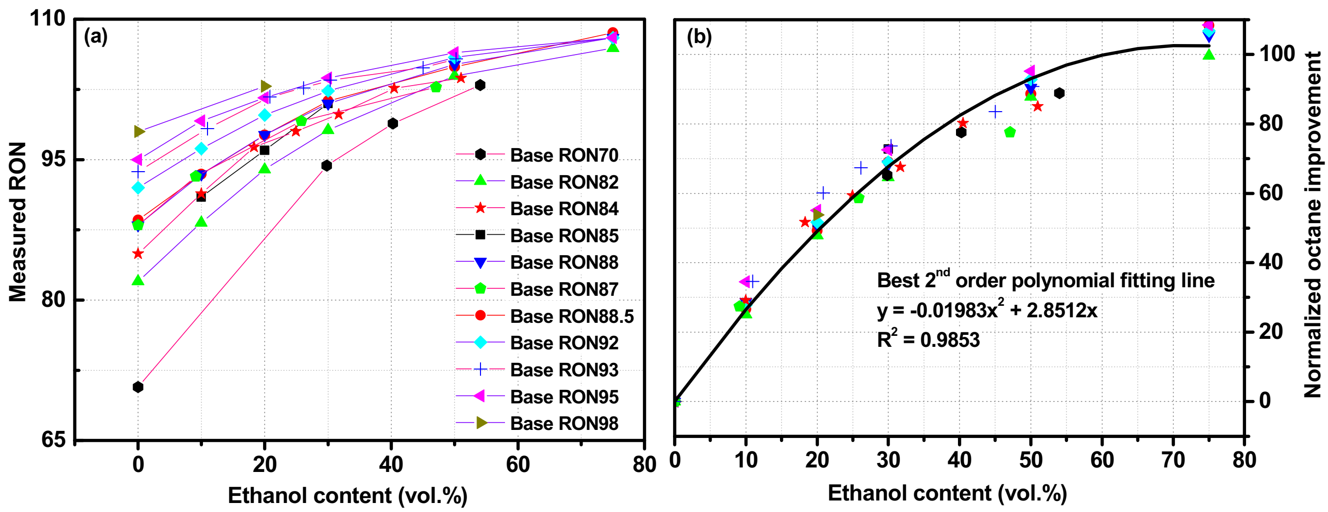

Fig. 2. Measured RON and NOI for ethanol blends in various base fuels: (a) measured RON; (b) normalized octane improvement (data extracted from Refs. [26,31,85,86]).

the blend. Fig. 2(b) shows the NOI of ethanol blends (0–75 vol.%). All pieces of data in Fig. 2(b) were converted from the data in Fig. 2(a) using Eq. (5). In Fig. 2(b), the values of a few NOI data points exceed 100. This is because the RON of ethanol in the original literature was higher than 107, however, in the calculation of NOI, the RON of ethanol is set as 107 (the value used by the authors of this paper).

A 2^{nd} order polynomial fitting line for ethanol content ranging from 0 to 75 vol.% is plotted in Fig. 2(b). The R^2 of the fitting line is 0.9853, therefore, it provides a good estimation. Eq. (6) shows the mathematic expression of this 2^{nd} order polynomial fitting line.

$$\mathrm{NOI} = -0.01983 \times x_{vol}^2 + 2.8512 \times x_{vol} \tag{6}$$

Once the RON of the base gasoline and ethanol, as well as the volumetric ethanol content are known, it is possible to calculate the RON of ethanol blends.

$$\mathrm{RON_{blend}} = \frac{\mathrm{NOI} \times (\mathrm{RON_{ethanol}} - \mathrm{RON_{base}})}{100} + \mathrm{RON_{base}} \tag{7}$$

A summary of three reported models and the model proposed by authors of this paper for RON prediction of ethanol blends is presented in Fig. 3(a), along with the measured octane rating. It is proposed that the accuracy ranking of these models is: authors' model ≈ NMC > LMC > LVC. The authors' model is simpler than, but as accurate as the NMC model because: (1) the NMC relies on estimation of P_g by measuring the RON of at least one ethanol blends, whilst the authors' model does not, and (2) the authors' model does not need the molar calculation, which can be problematic in itself, as the molecular formula of the gasoline is not exactly known.

Both the LMC and LVC models gave conservative estimations of the octane rating of ethanol blends, and the underestimation was up to 2.5 and 6 units of RON for the LMC and LVC models, respectively. The reason that LMC model gives a higher estimation of octane rating is because the molecular fraction of vaporized ethanol blends is made equal to their molar fraction in the model. The molecular weight of ethanol (46 g/mole) is much less than that of typical hydrocarbons in gasoline (95–115 g/mole), therefore, the mole fraction of ethanol in a blend is higher than its liquid volume fraction [83], thus, the LMC model is inaccurate in this sense. The NMC model gives a good prediction of octane rating of ethanol blends by adding an extra non-linear term for correcting the underestimation in the LMC model. The error of the NMC was less than 0.3, which is as good as the authors' model. In Fig. 3, only one gasoline base fuel with RON and MON of 88 and 82 was used. A more widely distributed octane rating for gasoline base fuels was

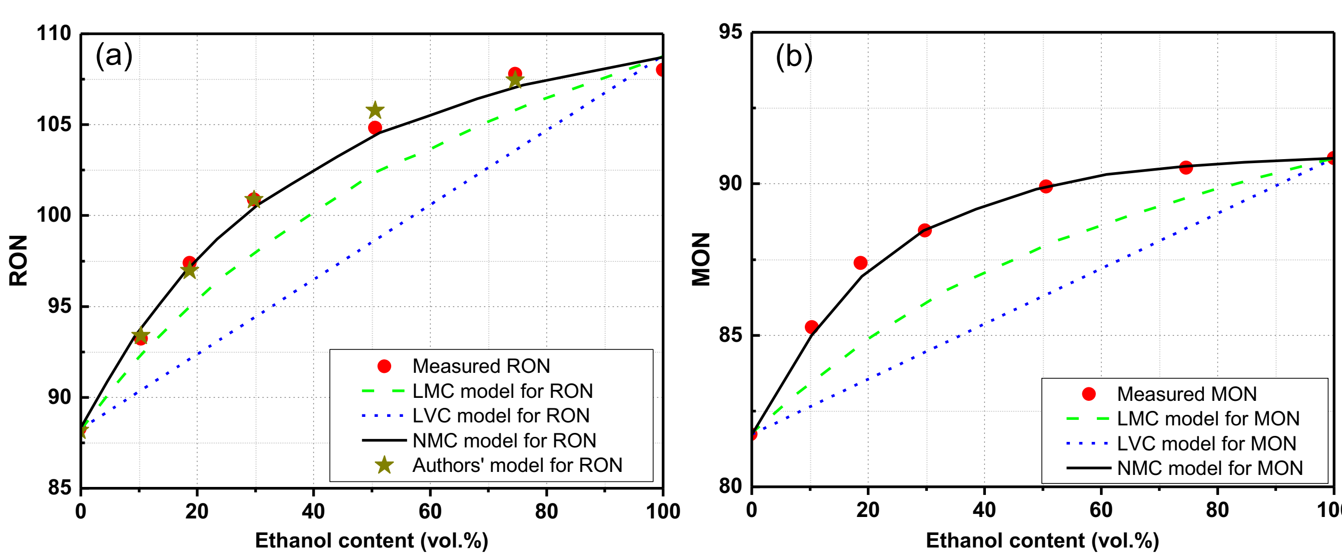

Fig. 3. Models for octane rating prediction: (a) RON; (b) MON (data extracted from Refs. [26,40,83]).

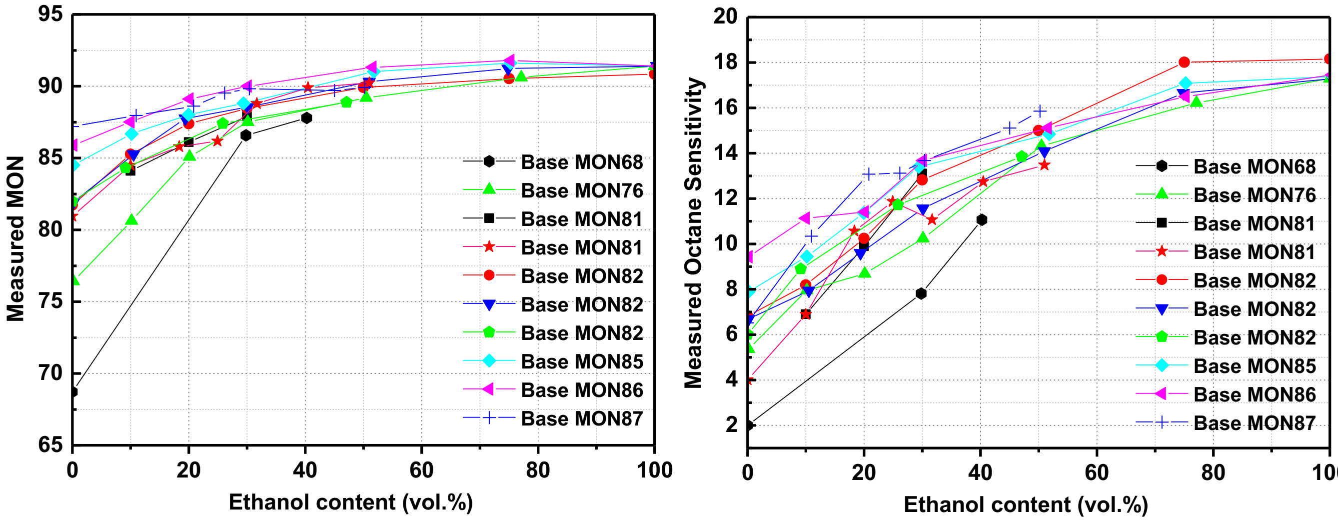

Fig. 4. Measured MON and octane sensitivity (data extracted from Refs. [26,40,85,86]).

also tested, and the authors' model and the NMC model showed a good agreement (less then 0.5 deviation from measured values) with experimental octane ratings measured in CFR engines. A summary of three reported models for MON prediction of ethanol blends is presented in Fig. 3(b), along with the measured octane rating. The accuracy ranking of these models is: NMC > LMC > LVC.

A similar literature review regarding MON and octane sensitivity is conducted and presented in and Fig. 4. It can be seen that MON is also not linear; higher MON improvement is observed with low ethanol content. Octane sensitivity, however, is almost linear to ethanol content. The OI of ethanol blends can be expressed as:

$$OI_{blend} = \frac{NOI \times (RON_{ethanol} - RON_{base})}{100} + RON_{base} + K \times S_{blend} \quad (8)$$

where OI_{blend} and S_{blend} are the OI and the octane sensitivity of ethanol blend, respectively. The S_{blend} can be linearly estimated by the sensitivities of base gasoline and ethanol.

2.3. Optimal ethanol blend ratio

Ethanol can be used as an octane booster, as demonstrated in the above sections. Even though adding more ethanol into gasoline always leads to higher octane rating, the marginal octane improvement decreases as the ethanol content is increased. Therefore, there exists an optimal ethanol blend ratio that maximizes ethanol's octane boost effect without reducing the lower heating value of the blend unnecessarily. The question of what the optimal ethanol blending ratio, is in terms of maximizing the overall octane rating, is highly relevant in the application of ethanol blends in SI engines. The following discussion in this section focuses on finding the answer to this question.

To assess octane value changes after blending, several parameters are defined. Volumetric octane value (VOV) is defined as a parameter that has a value equal to the RON of a fuel but in a unit of #/L. Total octane value (TOV) is defined as VOV multiplied by the fuel volume. Total octane value before blending (TOV_{b_blend}), and the total octane value after blending (TOV_{a_blend}) can be calculated from Eqs. (9) and (10), assuming that there is 1 L ethanol available for blending.

$$TOV_{b\_blend} = RON_{ethanol} + \left(\frac{100}{x_{vol}} - 1\right) \times RON_{base} \quad (9)$$

$$TOV_{a\_blend} = RON_{blend} \times \frac{100}{x_{vol}} \quad (10)$$

The octane-added index (OAI) is defined as a normalized octane value improvement after ethanol blending. This parameter is used to quantitatively describing the effectiveness of using ethanol as an octane booster.

$$OAI = \frac{TOV_{a\_blend} - TOV_{b\_blend}}{TOV_{b\_blend}} \quad (11)$$

Fig. 5 shows the OAI of ethanol blends in various base gasolines. The data in Fig. 5(a) was calculated using some of the measured data presented in Fig. 2(a). The data in Fig. 5(b) were calculated from Eqs. (7) and (11). For every base gasoline there exists an optimised blend ratio for maximizing the OAI. This is because the octane improvement of adding ethanol into gasoline is not linear to the ethanol content. As reported in Fig. 2, low percentage ethanol blends lead to more significant improvements than the medium and high percentage ethanol blends. It can be seen from Fig. 5 that the optimised ethanol content is approximately 40 vol.%; the RON of the base fuel showed a limited impact on this optimised blend ratio.

Another method to calculate the optimal ethanol ratio for boosting the octane rating of gasoline is to calculate the RON difference between the measured points and the RON estimated from linear regression [90]. Data from Rankovic et al. [90] suggested that 40 vol.% of ethanol addition into base gasoline gave the maximum RON difference, which matches with the results in Fig. 5.

3. Cooling effect and the effect of octane index for ethanol blends

There are two parts in this section. First, the fuel's cooling effect in DISI engines is reviewed and quantified. Second, the partially captured cooling effect (PCCE) from the standard RON test in the CFR engine is discussed, along with the effective octane index (EOI), which accounts for the original OI and cooling effect.

3.1. Quantification of cooling effect for ethanol blends

In DISI engines, apart from the fuel's octane rating, the charge cooling effect of the fuel is another important contributor in suppressing knock. The charge cooling effect is related to the fuel's heat of vaporization. When fuel is injected directly into the cylinder, the fuel spray/droplet is vaporized by the heat from the warm in-cylinder gases, during which in-cylinder gas temperature drops.

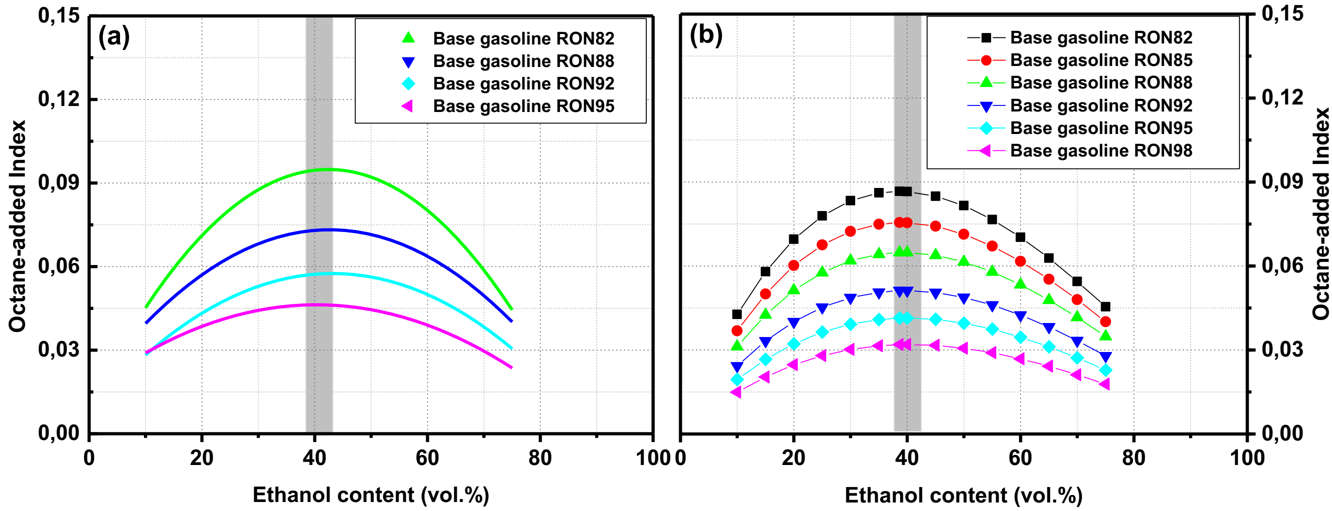

Fig. 5. Octane-added index of ethanol blends in various base fuels: (a) data extracted from [31]; (b) data calculated from Eqs. (7) and (11).

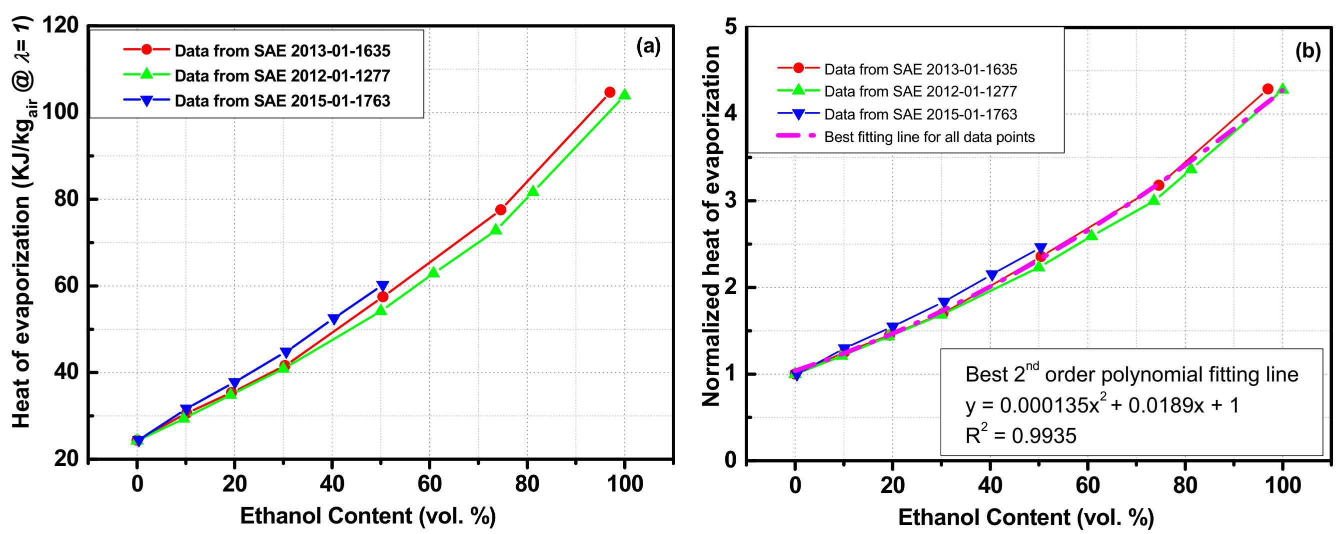

Fig. 6. Heat of vaporization for ethanol blends: (a) heat of vaporization; (b) normalized heat of vaporization (data extracted from Refs. [26,31,86]).

Fig. 6 presents the heat of vaporization (HOV) of ethanol blends. The data from Fig. 6(a) was extracted from the literature and if necessary, the data converted into kJ/kg_air at the stoichiometric air fuel ratio (AFR). The HOVs of ethanol and conventional gasoline are approximately 103 and 26 kJ per kilogram of air at stoichiometric AFR, respectively. Therefore, compared to gasoline, ethanol has a cooling effect approximately 3 times more significant.

The data from Fig. 6(b) are normalized data from Fig. 6(a), obtained by dividing the HOVs of the ethanol blends by the HOV of the base gasoline. A 2nd order polynomial fitting line for all data was plotted in Fig. 6(b), which has an R^2 value of 0.9935. Eq. (12) shows the mathematical expression of this 2nd order polynomial fitting line:

$$\text{Normalized HOV} = 0.000135 \times x_{vol}^2 + 0.0189 \times x_{vol} + 1 \tag{12}$$

In this paper, an assumption is made that the HOV of the fuel has a linear relationship to the cooling effect in direct injection engines. This assumption is supported by the data reported in Ref. [91], where the RON and cooling effect were quantitatively separated. More detailed verification of this assumption will be presented in the second part of this section.

In order to quantitatively describe the cooling effect, a parameter, octane number from the cooling effect (ONCE), is defined. The abbreviations of $ONCE_{gasoline}$ and $ONCE_{ethanol}$ stand for the cooling effect of gasoline and ethanol, respectively.

Based on an study of the CR distribution of engines sold in the North American market in 2013 [92], it was found that DI engines have approximately 1 unit higher CR than PFI engines. The investigated engine models include 85 NA PFI engines (averaged CR = 10.4), 34 NA DI engines (averaged CR = 11.7), 16 boosted PFI engines (averaged CR = 9.2) and 32 boosted DI engines (averaged CR = 9.8). The increase of CR in DI engines is mainly due to the charge cooling effect. Therefore, $ONCE_{gasoline}$ leads to a 1 unit increase of CR. For gasoline engines, every 4 units increase of octane number (ON) allow a 1 unit increase of CR, which will be explained in detail in a later section. Therefore, it is assumed that ONCEgasoline is equal to 4 units of ON. Compared to base gasoline, the additional cooling effect from the ethanol blends equals to:

$$\begin{aligned}\Delta ONCE &= ONCE_{blend} - ONCE_{gasoline} \\ &= (0.000135 \times x_{vol}^2 + 0.0189 \times x_{vol} + 1 - 1) \times 4 \\ &= 0.00054 \times x_{vol}^2 + 0.0756 \times x_{vol}\end{aligned} \tag{13}$$

There is some data available for describing the ONCE of ethanol blends in the literature. Fig. 7 shows the cooling effect of ethanol blends published in [28,31]. The red line on the figure is plotted by using data calculated from Eq. (13). It can be seen that, compared to the literature data, the red line derived from HOVs gives conservative estimations of the cooling effect of ethanol blends. For example, the underestimation for E40 is approximately 2.5 units of octane. It was consequently decided that in this study, the literature data would be used to estimate the ΔONCE between ethanol blends and base gasoline, which is expressed as follow:

$$\Delta ONCE = ONCE_{blend} - ONCE_{base} = 0.1543 \times x_{vol} \tag{14}$$

3.2. Partially captured cooling effect in standard RON Test, and Effective OI

The cooling effect of ethanol blends is partially captured in the standard RON test. In the standard RON test in the CFR engine, the intake air is heated up to 52 °C (see Table 2). The temperature of the air and fuel mixture entering the engine varies, depending on the HOV of the fuel. Before the fuel enters the cylinder, it evaporates and mixes with the air by absorbing heat from the intake air, engine components and intake value. Gasoline can be vaporized prior to entry into the CFR engine [91]. However, for blends with a high ethanol content, the fuel is only partially vaporized and there exists a near-saturated and potentially two-phase air/fuel mixture during the induction process [91].

Foong et al. [91] designed a so-called 'modified RON' test by modifying the intake system in the CFR engine for the study of the effect of charge cooling on the RON measurement. In the modified intake system, the temperature of the air-fuel mixture entering the engine was maintained at 36 °C, which is the intake air-fuel mixture temperature that is measured during the standard RON test when only primary reference fuels (PRFs) are used. At this temperature, the air-fuel mixture was not saturated when entering the CFR engine, even for the air-ethanol mixture. They modelled the thermodynamics of the air-fuel mixture preparation in this modified RON test; and they concluded that ethanol blends were fully vaporized prior to compression and that the temperature during the compression stroke was similar to those in primary reference fuels. Thus, the modified RON tests only captured the auto-ignition chemistry effect from ethanol blends. By comparing the RON results from standard and modified tests, the partial cooling effect captured by the standard RON test in the CFR engine was quantified. Fig. 8 shows the results from standard and modified RON tests, along with the RON deviation from these two tests.

From Fig. 8, it can be seen that for ethanol content less than 20 vol.%, the rate of RON deviation versus ethanol content was much lower than that for ethanol content higher than 40 vol.%. This is because as ethanol addition exceeded approximately 20 vol.%, the ethanol vapour was fully saturated; adding more ethanol into the blends led to the cooling effect from the extra ethanol addition being fully captured in the RON test. The RON difference between the standard and modified tests had a strong linear relationship to the ethanol content, from 40 vol.% to 100 vol.%. Since the HOV of ethanol blends is almost linear to ethanol content, it is safe to assume that the cooling effect is linear to HOV in DISI engines, an assumption made in the first part of this section.

Based on Fig. 8 [91], the partially captured cooling effect (PCCE) in the standard RON test can be quantified as:

$$PCCE = 0.00028 \times x_{vol}^2 + 0.0200 \times x_{vol} \tag{15}$$

In DI engines, the EOI of ethanol blends, which accounts for the original OI and cooling effect, is defined as:

$$\begin{aligned} EOI &= OI + (ONCE - PCCE) \\ &= (RON - K \times S) + (ONCE - PCCE) \\ &= (\mathit{RON} - \mathit{PCCE}) - K \times S + ONCE \end{aligned} \tag{16}$$

In this paper, it is defined that "RON–PCCE"' represents the chemical effect on the anti-knock property of fuel, "$-K \times S$" represents the octane sensitivity effect, and ONCE, as defined earlier, represents the cooling effect.

Therefore, EOI = chemical effect + octane sensitivity effect + cooling effect.

The difference in EOI between base gasoline fuel and ethanol blends in DISI engines is given by:

$$\Delta EOI = RON_{blend} - RON_{base} - PCCE - K \times \Delta S + \Delta ONCE \tag{17}$$

Overall, this section analysed the HOV and cooling effect of ethanol blends, along with the PCCE during the standard RON test. Finally, the EOI of ethanol blends was discussed. The purpose of this section is to determine the EOI improvement of ethanol blends in comparison with base gasoline, for the purpose of estimating potential CR increase of using ethanol blends, which will be presented in the next section.

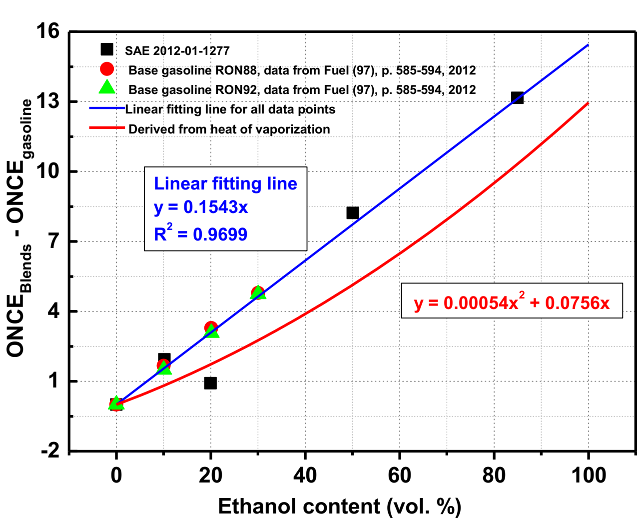

Fig. 7. ONCE improvement in ethanol blends (data extracted from Refs. [28,31]).

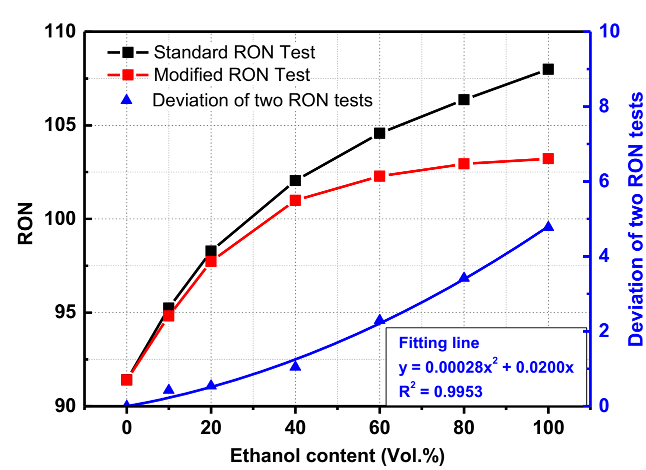

Fig. 8. Results from standard and modified RON test (data extracted from Ref. [91]).

Table 3
Summary of RON and CR study.

RON/CR	RON	CR	Reference
2,5	RON91-E10/RON96-E20sp	10:1/11,9:1	Ford and AVL [85]
2,4	RON95/RON 107-E100	10:1/15:1	Aachen University [94]
3	RON91-E10/RON99-E30sp	10:1/13:1	Oak Ridge National Lab [93]
3	RON89-E0/RON97-E30sp	NA	Ford [28]
3,3	RON91-E10/RON101-E30sp	10:1/13:1	Ford and AVL [85]
3,5	RON89-E0/RON92,6-E10sp	NA	Ford [28]
3,5	RON89-E0/RON95,8-E20sp	NA	Ford [28]
3,5	RON95/RON102	10.2:1/12.2:1	BP [95]
3,5	RON95/RON102-E20sp	10:1/12,2:1	Aachen University [94]
3,6	AKI87-E0/RON101-E30sp	9,2:1/12:1	Oak Ridge National Lab [93]
5	RON88/RON108-E75	10:1/14:1	AVL, BP and Ford [31]

4. Compression ratio and engine thermal efficiency

Many studies have focused on the application of ethanol blends in the SI engine with an adapted CR [28,31,85,93–95]. The two key questions are: (1) how much ON improvement is required to be able to increase one unit of CR and (2) how much engine thermal efficiency can be gained by increasing one unit of CR. This section discusses the answers to these two questions by comprehensively reviewing the relevant literature. The final answers to these two questions are empirical, since the combustion characteristics in the two engines with the base CR (base fuel) and the high CR (high octane fuel) should remain the same. However, this is challenging, or even impossible to achieve within the entire engine operating map. Therefore, the data published by OEMs are highly valuable, since they mostly choose the CR of the passenger vehicles based on the main-grade gasoline available in the market. Different markets for example US and EU, have different gasoline fuel standards, which forces OEMs to conduct a lot of research on CR adaptation and relevant engine calibration. Additionally, the large number of cases reviewed in this study increases the credibility of empirical answers.

4.1. Compression ratio

Table 3 summarises the CR improvements enabled by the use of high octane rated fuels. Fig. 9 shows the bar chart and statistical analysis of data listed in Table 3 and other references mentioned in Ref. [92]. In Fig. 9, reference codes were given in the x axis; and their corresponding reference numbers are provided in the caption. The statistical analysis shows that approximately every 3.9 units of ON enables a 1 unit increase of CR.

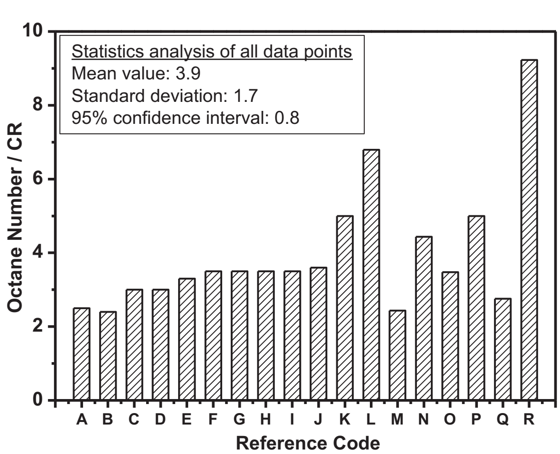

Fig. 9. Octane number required for 1 unit increase in CR (Reference Code: A = [85]; B = [94]; C = [93]; D = [28]; E = [85]; F = [28]; G = [28]; H = [95]; I = [94]; J = [93]; K = [31]; L = [97]; M = [98]; N = [98]; O = [99]; P = [100]; Q = [101]; R = [102]).

Splitter et al. [93] from Oak Ridge National Lab experimentally investigated spark-ignited combustion with E0 (RON 90 and MON 84) and E30 (RON 100 and MON 89). A single-cylinder research engine was used with a low and a high CR of 9.20:1 and 11.85:1, respectively. All fuels were operated at full-load conditions with $\lambda = 1$, using 0% or 15% external cooled EGR. Using E30 with the CR of 11.85:1 led to the same onset knock load as E0 with the CR of 9.20:1, representing 3.8 ON/CR. In Ref. [92], the authors from Ford and General Motors suggested that 3 units of ON was necessary for a 1 unit increase of CR. Heywood et al. [43] from MIT suggested that 4–5 units of ON was required for a 1 unit increase of CR.

The estimation in the literature varies, depending on engine technology, geometry, and operating conditions. As mentioned in the beginning of this section, it is hard to assume that the knocking behaviour in all configurations presented in the literature is the same, however, by summarising a wide arrange of literature, especially the data published by engine/vehicle OEMs, and by comparing the estimation made by other research institutes, the risk of underestimating or overestimating the unit of ON required to increase one unit of CR can be reduced.

Considering the literature data in Fig. 9, and the suggestion from Refs. [43,92], in this study, an empirical assumption, 4 ON/CR, is suggested for the prediction of CR improvement resulting from the use of high octane rated fuels.

4.2. Engine thermal efficiency improvement from increased CR and flame speed

This section covers the estimation of engine thermal efficiency gain by using ethanol blends due to the increase of CR and flame speed. Engine thermal efficiency depends on various factors: such as engine design, combustion chamber design, engine operating conditions (load and speed) and the fuel being used. However, when the engine operating parameters and engine design (apart from CR) are the same, the most influential factors on thermal efficiency are the CR and fuel property. In this study, the use of splash-blended ethanol blends enables a higher CR. In addition, ethanol has a higher flame speed, therefore, ethanol blends benefit from a faster burning rate. Faster burning is beneficial for SI engines because more fuel is burned during the most optimum part of the engine cycle. Thus, both high CR and a faster burning rate contribute to a higher engine thermal efficiency.

Eq. (18) shows the calculation of engine thermal efficiency, where the lower calorific value is used for the fuel energy input calculation.

$$\text{Engine theraml efficiency} = \frac{\text{Work exerted on the piston per cycle}}{\text{Fuel energy input per cycle}} \tag{18}$$

In this paper, the intention is not to comment on the specific engine thermal efficiency at certain operating conditions. The intention is to show empirical estimations of engine thermal efficiencies across the entire engine map using various ethanol blends.

Fig. 10(a) presents the relative thermal efficiency gain at various CRs. Fig. 10(b) shows the thermal efficiency gain rate per CR calculated from the data presented in Fig. 10(a). In both Fig. 10 (a) and (b), the reference codes are given in the legend or x axis, and their corresponding reference numbers are provided in the caption. For the reference code 'L', the thermal efficiency gain was calculated based on the theoretical ideal Otto cycle using the equation: $\eta = 1-1/(CR^{\gamma-1})$. $\gamma = C_p/C_v$, where C_p and C_v are the specific heat of gas at constant pressure and volume, respectively. In Fig. 10, only data with a CR in the range of 8:1–14:1, was chosen to be presented. This is based on the CR distribution of engines currently on the market. In modern gasoline engines, OEMs generally choose a CR in the range of 8:1–13:1. There are barely any gasoline engines with a CR higher than 13, with Mazda claiming that their SKYACTIV gasoline engine has the world's highest CR (14:1) in the marketplace of mass production gasoline engines. Normally, for CRs above 12:1, the Atkinson or Miller cycle is used with variable valve timing or a twin-cam mechanism.

When a linear best fitting line is plotted for any cases presented in Fig. 10(a), the fitting line has an R^2 of at least 0.9600. Therefore, even though it is true that engine thermal efficiency gain reduces with CR due to the increase in surface/volume ratio and other factors, within the CR range of 8:1–14:1 it is reasonable to assume that engine thermal efficiency improvement is linear to CR.

In Fig. 10(b), a statistical analysis of all data points is given. It shows that an average thermal efficiency gain per 1 unit of CR is 0.018 (1.8%). Paul Miles from the Combustion Research Facility in Sandia National Laboratories suggested that 1.6–2.0% is a good estimation for engine thermal efficiency gain per unit increase of CR [43]. The estimation of 1.8% falls in between 1.6% and 2%; therefore, the assumption of 1.8% thermal efficiency gain per CR is used in this study.

Apart from the fuel's octane rating and the CR, there are a few other factors that are important in regards to their effect on engine thermal efficiency, such as flame speed and engine downsizing. Flame speed is an important parameter for gasoline fuels. Engine knocking occurs when the end gas in the combustion chamber is auto-ignited before the flame ignites it, therefore, higher flame speed help to reduce the engine's knocking tendency and thus they increase the engine's thermal efficiency by allowing more advanced ignition timings. It is also well known that gasoline fuels with high flame speeds are more favourable for a lean burn combustion mode; and they are more tolerable to a higher EGR rate, and more fuel is burned during the most optimum part of the engine cycle.

According to the data published in Ref. [42], increasing laminar flame speed from 0.42 to 0.47 m/s enables a 1.47% gain of engine thermal efficiency. Typical laminar flame speeds for gasoline and ethanol at atmospheric pressure and temperature are 0.44 and 0.55 m/s, respectively [43]. Assuming that the laminar flame speed increases linearly with the ethanol content, which is also the case in Ref. [42], every 10 vol.% ethanol increase in fuel blend leads to a 0.32% increase in thermal efficiency. In Ref. [43], it is suggested that a 1% increase in thermal efficiency is achievable when the flame speed is increased from 0.43 to 0.46 m/s. In other words, every 10% ethanol increase in fuel blends leads to a 0.37% improvement in engine thermal efficiency due to the flame speed. In this paper, a conservative value (0.20%) will be used for the estimation of the increase of engine thermal efficiency for every 10 vol.% increase of ethanol content in blends.

Engine downsizing is a technology that increases engine thermal efficiency by forcing an engine to operate at more efficient high load regimes, instead of operating at low load regimes where pumping losses significantly reduce engine thermal efficiencies. In Ref. [92], it is suggested that the thermal efficiency increase multiplier from additional engine downsizing for TC DI engines is 1.1.

5. Case studies for ethanol blends in SI engines

In this section, case studies for ethanol blends in SI engines are presented. The selected SI engines for these case studies include NA DISI engines, and TC DISI engines, representing current and future technologies. K values, 0 and −0.3, will be considered to represent the average engine K value for NA DISI engines and TC DISI engines, respectively. The justification of selecting these K values is provided in the earlier section.

Two base fuels are selected for these case studies. The first base fuel has a RON and S of 92 and 10 (AKI = 87), respectively, and a LHV of 41 MJ/kg, representing the main-grade gasoline available in the US market. The justification is based on a fuel survey conducted for the US main-grade gasolines in 2013 [96]; about 459 winter and summer samples from 30 major metropolitan areas

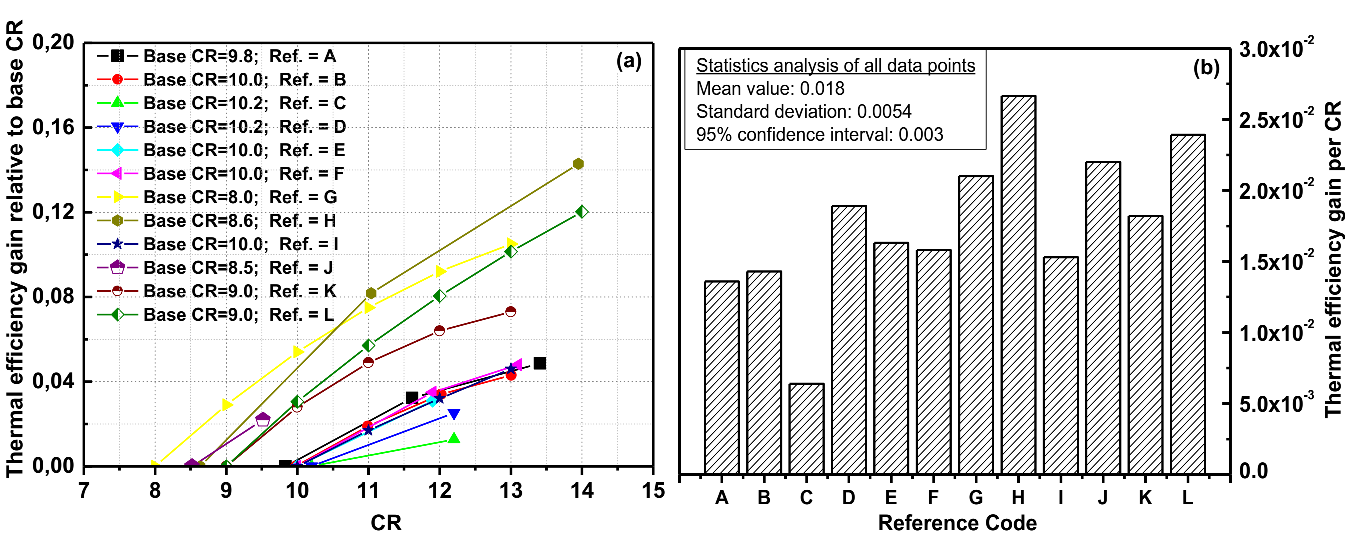

Fig. 10. Thermal efficiency at various compression ratios: (a) thermal efficiency gain relative to base CR; (b) thermal efficiency gain per CR (Reference Code: A = [103]; B = [104]; C = [105]; D = [105]; E = [85]; F = [85]; G = [106]; H = [107]; I = [108]; J = [109]; K = [110]; L = ideal Otto cycle).

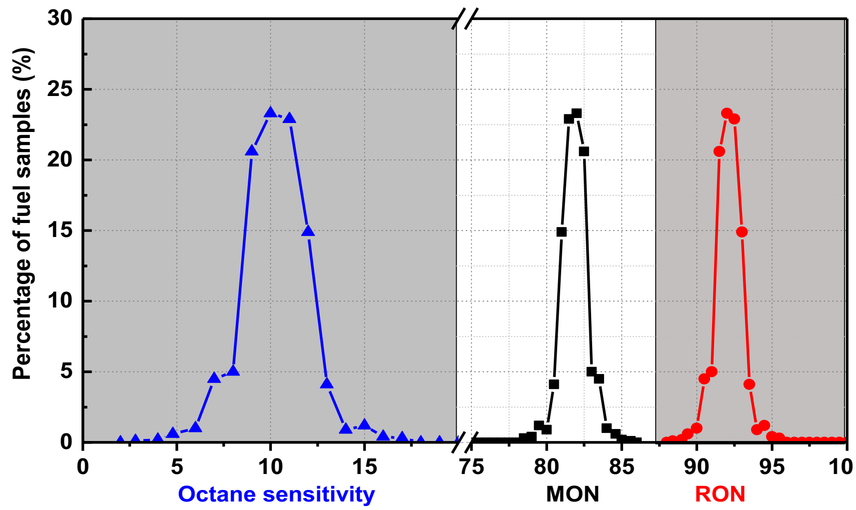

Fig. 11. Octane sensitivity, MON and RON distributions of fuel samples in the US market in 2013 (data source: Ref. [96]).

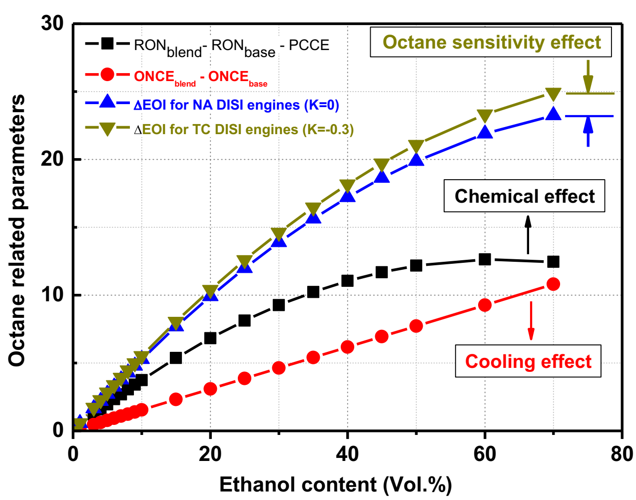

Fig. 12. Summary of octane related parameters for ethanol blends ($RON_{base} = 92$; S = 10; $LHV_{base} = 41$ MJ/kg; $LHV_{ethanol} = 26.7$ MJ/kg; K = 0 for the NA DISI engine; K = −0.3 for the TC DISI engine).

were surveyed. The summary of this survey is presented in Fig. 11. The second base fuel has a RON and MON of 84 and 74 (AKI = 80), respectively, and a LHV of 41 MJ/kg. The blend of 10 vol.% ethanol into the second base fuel leads to E10 that has a RON and S of 90 and 10.8 (AKI = 87), respectively, also representing the main-grade gasoline available in the US market.

Fig. 12 summaries the octane related parameters for ethanol blends in the range of E0–E70. As mentioned earlier, EOI = chemical effect + octane sensitivity effect + cooling effect, in which the chemical effect is equal to RON minus PCCE (RON-PCCE), the octane sensitivity effect is equal to $-K * S$, and the cooling effect is equal to ONCE. From Fig. 12, it is clear that the ranking of these factors is: chemical effect > cooling effect > octane sensitivity effect. As the ethanol content is increased, the increased rate of the cooling effect is higher than that of the chemical effect. For E70, the difference between the chemical effect and cooling effect is less than only two units. Since the K value for TC DISI engines is set as −0.3, and the octane sensitivity difference is only 5.6 for E70, the octane sensitivity effect for E70 is only 1.7 units higher than that of the base gasoline.

Fig. 13(a) shows the engine thermal efficiency gain and LHV reduction for ethanol blends in the range of E0–E70. The reduction of LHV is linear to the ethanol content, whilst the marginal thermal

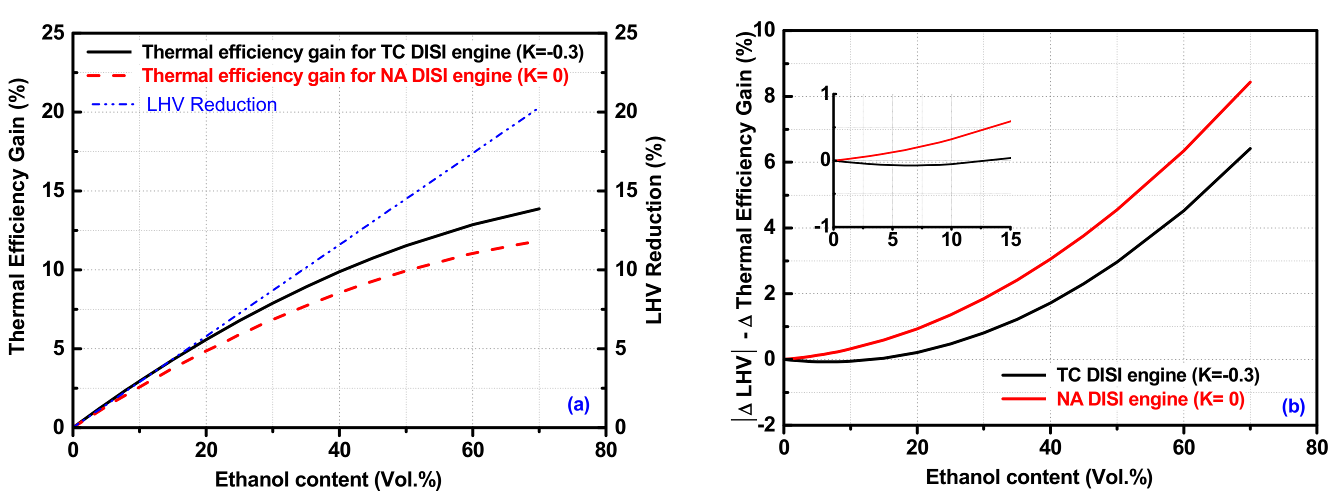

Fig. 13. Engine thermal efficiency gain and LHV reduction for ethanol blends ($RON_{base} = 92$; S = 10; $LHV_{base} = 41$ MJ/kg; $LHV_{ethanol} = 26.7$ MJ/kg; K = 0 for the NA DISI engine; K = −0.3 for the TC DISI engine).

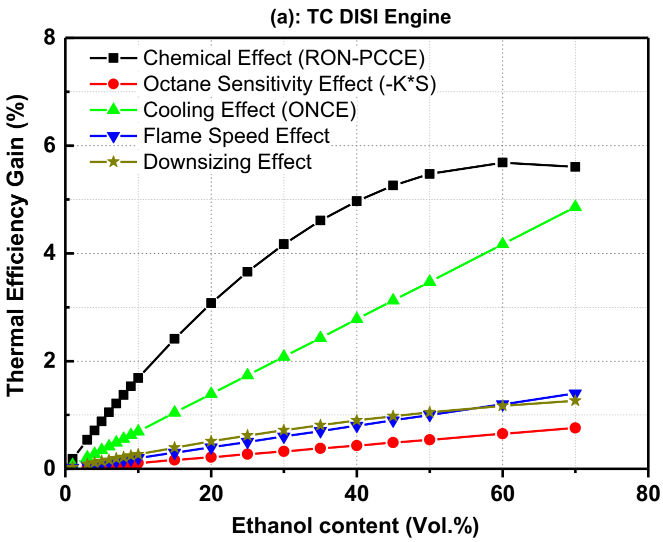

Fig. 14. Breakdown of relative engine thermal efficiency gain: (a) TC DISI engines; (b) NA DISI engines; (RON_{base} = 92; S = 10; LHV_{base} = 41 MJ/kg; $LHV_{ethanol}$ = 26.7 MJ/kg; K = 0 for the NA DISI engine; K = −0.3 for the TC DISI engine).

efficiency gain decreases with ethanol content. The vehicle's mileage range could be reduced due to the use of low energy density ethanol blends. However, the reduced LHV can be partially or even completely offset by improved engine thermal efficiency. For the NA DISI engine, the engine thermal efficiency gain is always less than the LHV reduction, whilst for the TC DISI engine, the thermal efficiency gain outweighs the LHV reduction for ethanol blends up to E14. Due to the extra octane sensitivity effect, there is more engine thermal efficiency gains for TC DISI than NA DISI.

In this paper, the vehicle mileage range reduction is simply calculated by subtracting the absolute value of LHV reduction by the engine thermal efficiency gain ($|\Delta LHV|-\Delta\eta$). The results are presented in Fig. 13(b). Overall, Fig. 13(b) demonstrates the possibility of using high ethanol blends without significantly deteriorating the vehicle mileage range. If a maximum of 2% mileage reduction is considered as acceptable, the maximum ethanol content allowed is approximately 30 vol.% and 43 vol.% for NA DISI and TC DISI engines, respectively. If a maximum of 5% mileage reduction is considered as acceptable, the maximum ethanol content allowed is approximately 52 vol.% and 63 vol.% for NA DISI and TC DISI engines, respectively. The LHV of base gasoline is assumed to be 41 MJ/kg, which is within the normal range of gasoline fuels. If the LHV of base gasoline is higher than 41 MJ/kg, the reduction of LHV for ethanol blends presented in Fig. 13(a) would be underestimated, and vice versa.

Fig. 14 shows the breakdown of engine thermal efficiency gain under various ethanol blends for NA DISI and TC DISI engines. The contributors to the improved engine thermal efficiency are the

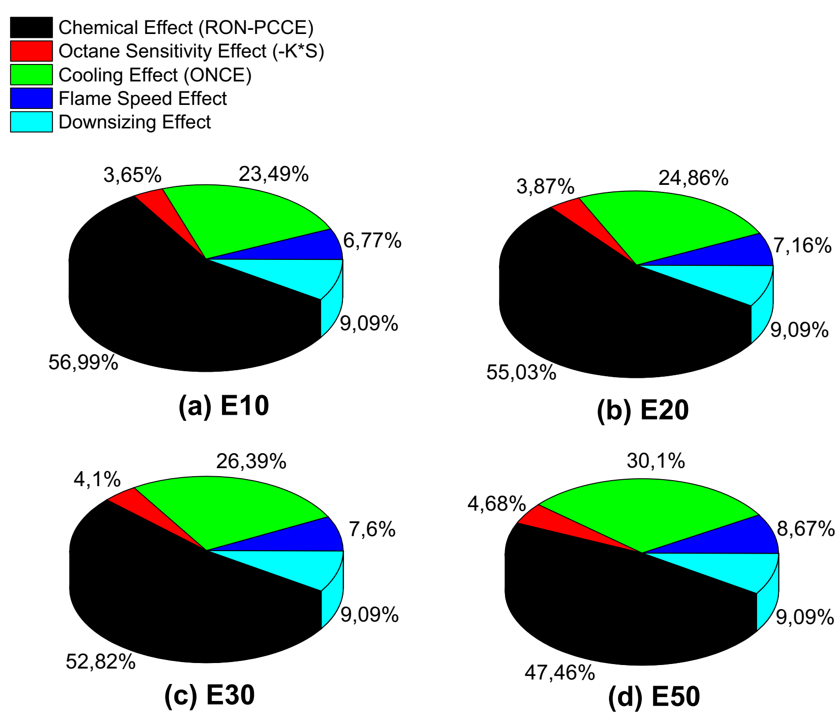

Fig. 15. Breakdown of Engine thermal efficiency gain for TC DISI engines: (a) E10; (b) E20; (c)E30; (d) E50 (RON_{base} = 92; S = 10; LHV_{base} = 41 MJ/kg; $LHV_{ethanol}$ = 26.7 MJ/kg; K = −0.3 for the TC DISI engine).

Table 4
Summary of results of the case study for ethanol blends in NA DISI engines using a base fuel with RON of 84 and S of 10.

Ethanol content Vol.%	ΔONCE	ΔRON - PCCE	ΔEOI	η gain from improved CR %	η gain from flame speed %	Total η gain %	LHV reduction %	LHV reduction - η %
0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
1	0.15	0.63	0.78	0.35	0.02	0.37	0.29	−0.08
3	0.46	1.86	2.32	1.05	0.06	1.11	0.87	−0.24
4	0.62	2.46	3.08	1.39	0.08	1.47	1.16	−0.31
5	0.77	3.05	3.83	1.72	0.10	1.82	1.45	−0.37
6	0.93	3.64	4.56	2.05	0.12	2.17	1.74	−0.43
7	1.08	4.21	5.29	2.38	0.14	2.52	2.03	−0.49
8	1.23	4.77	6.00	2.70	0.16	2.86	2.32	−0.54
9	1.39	5.32	6.71	3.02	0.18	3.20	2.61	−0.59
10	1.54	5.87	7.41	3.33	0.20	3.53	2.90	−0.63
15	2.31	8.43	10.75	4.84	0.30	5.14	4.35	−0.79
20	3.09	10.76	13.85	6.23	0.40	6.63	5.80	−0.83
25	3.86	12.84	16.70	7.51	0.50	8.01	7.25	−0.77
30	4.63	14.68	19.31	8.69	0.60	9.29	8.70	−0.59
35	5.40	16.28	21.68	9.75	0.70	10.45	10.15	−0.31
40	6.17	17.63	23.80	10.71	0.80	11.51	11.60	0.09
45	6.94	18.74	25.68	11.56	0.90	12.46	13.05	0.59
50	7.72	19.61	27.32	12.29	1.00	13.29	14.50	1.20
52	8.02	19.89	27.91	12.56	1.04	13.60	15.08	1.48
60	9.26	20.61	29.87	13.44	1.20	14.64	17.40	2.76
70	10.80	20.64	31.44	14.15	1.40	15.55	20.30	4.75

fuel's chemical effect, cooling effect and its octane sensitivity effect, as well as its improved laminar flame speed and the engine downsizing it enables. For the TC DISI engine, the contribution ranking is: chemical effect > cooling effect > downsizing effect ≈ flame speed effect > octane sensitivity effect. The contribution of the chemical effect to the engine thermal efficiency gain reduces with ethanol content. For the NA DISI engine, only the chemical effect, cooling effect and flame speed effect contribute to the gains of engine thermal efficiency. The absolute magnitudes of their contributions are the same as those for the TC DISI engines.

Fig. 15 shows a more detailed breakdown of engine thermal efficiency gains for TC DISI engines when using E10, E20, E30 and E50 as fuels. The majority of the thermal efficiency gains are achieved due to chemical effect. For E10, the contribution of the chemical effect is 57%, and it reduced to 48% for E50. The dominance of the chemical effect reduces with increased ethanol content, due to the non-linear improvement of the octane rating with ethanol content. The contribution of the cooling effect increased from 23% for E10 to 30% for E50. Since the downsizing multiplication factor for TC DISI engines is set at 1.1, the contribution of engine downsizing is fixed at 9.1%. The contribution of flame speed increases from 6.8% for E10 to 8.7% for E50. The contribution of octane sensitivity increases from 3.7% for E10 to 4.7% for E50.

The summary of results for these case studies for using a base fuel with a RON and S of 84 and 10 (AKI = 80), respectively, and LHV of 41 MJ/kg, are presented in Tables 4 and 5 for NA DISI and TC DISI engines, respectively. The blend of 10 vol.% ethanol into the second base fuel leads to a E10 that has a RON and S of 90 and 10.8 (AKI = 87), respectively, also representing the main-grade E10 available in the US market. The results presented in Tables 4 and 5 show a similar trend with the results presented in

Table 5
Summary of results of the case study for ethanol blends in TC DISI engines using a base fuel with RON of 84 and S of 10.

Ethanol content Vol.%	ΔONCE	ΔRON - PCCE	Octane sensitivity effect	ΔEOI	η gain from improved CR %	η gain from downsizing %	η gain from flame speed %	Total η gain %	LHV reduction %	LHV reduction - η %
0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
1	0.15	0.63	0.02	0.81	0.36	0.04	0.02	0.42	0.29	−0.13
3	0.46	1.86	0.07	2.40	1.08	0.11	0.06	1.25	0.87	−0.38
4	0.62	2.46	0.10	3.18	1.43	0.15	0.08	1.66	1.16	−0.50
5	0.77	3.05	0.12	3.95	1.78	0.19	0.10	2.06	1.45	−0.61
6	0.93	3.64	0.14	4.71	2.12	0.22	0.12	2.46	1.74	−0.72
7	1.08	4.21	0.17	5.46	2.46	0.26	0.14	2.85	2.03	−0.82
8	1.23	4.77	0.19	6.20	2.79	0.29	0.16	3.24	2.32	−0.92
9	1.39	5.32	0.22	6.93	3.12	0.33	0.18	3.63	2.61	−1.02
10	1.54	5.87	0.24	7.65	3.44	0.36	0.20	4.01	2.90	−1.11
13.5	2.08	7.69	0.32	10.10	4.54	0.48	0.27	5.29	3.91	−1.38
15	2.31	8.43	0.36	11.11	5.00	0.53	0.30	5.83	4.35	−1.48
20	3.09	10.76	0.48	14.33	6.45	0.68	0.40	7.53	5.80	−1.73
25	3.86	12.84	0.60	17.30	7.78	0.83	0.50	9.11	7.25	−1.86
30	4.63	14.68	0.72	20.03	9.01	0.96	0.60	10.57	8.70	−1.88
35	5.40	16.28	0.84	22.52	10.13	1.08	0.70	11.92	10.15	−1.77
40	6.17	17.63	0.96	24.76	11.14	1.19	0.80	13.14	11.60	−1.54
45	6.94	18.74	1.08	26.76	12.04	1.29	0.90	14.24	13.05	−1.19
50	7.72	19.61	1.20	28.52	12.83	1.38	1.00	15.22	14.50	−0.72
60	9.26	20.61	1.44	31.31	14.09	1.53	1.20	16.82	17.40	0.58
63	9.72	20.72	1.51	31.96	14.38	1.56	1.26	17.20	18.27	1.06
70	10.80	20.64	1.68	33.12	14.91	1.63	1.40	17.94	20.30	2.36

Figs. 12–15 where a base fuel with an AKI of 87 is used. Compared to the ethanol blends with a base fuel that has an AKI of 87, the ethanol blends with a base fuel that has an AKI of 80 showed a greater potential of engine thermal efficiency gains due to more improvement in the octane rating. For example, in a NA DISI engine, the breakeven ethanol content for a matched vehicle mileage range with the AKI80 base fuel is 40 vol.%, whilst when the AKI 87 base fuel is used, there is not an ethanol content that allows a breakeven vehicle mileage range. In a TC DISI engine, the breakeven ethanol content for a matched vehicle mileage range with the AKI80 base fuel is 55 vol.%, whilst it is 14 vol.% for the AKI87 base fuel.

6. Conclusions

In this study, a critical review and discussion regarding the application of ethanol blends in spark ignition engines is provided. The focuses of this study are on the octane rating, cooling effect, compression ratio increase, potential engine efficiency improvement and possible vehicle mileage range reduction. The following are the main conclusions:

1. The history of octane rating definition and measurement in CFR engines are reviewed. It has been proven that RON is a good indication of fuel's anti-knock characteristics, but MON showed a poor correlation. The octane index is proposed in the literature as a better indicator for characterizing the octane appetite of modern SI engines.
2. Three reported RON and MON prediction models for ethanol blends are discussed and compared, including linear molar-weighted model (LMC), linear volumetric-weighted model (LVC) and non-linear molar-weighted model (NMC). Additionally, the authors of this paper proposed a new and simple model. It is proposed that the accuracy ranking of these models is: authors' model ≈ NMC > LMC > LVC.
3. The octane-added index (OAI) was proposed to describe the effectiveness of using ethanol as an octane booster. There exists an optimised ethanol-blend ratio (40 vol.% ethanol in blends) that gives the maximum octane value. This optimised blend ratio is insensitive to the octane rating of the base gasoline.
4. The cooling effect of ethanol is quantitatively described as an equivalent octane number. The partially captured cooling effect during the standard RON test is also reviewed and quantified. It is proven that the cooling effect is a significant factor for suppressing engine knocking.
5. A prediction tool for engine thermal efficiency gain for ethanol blends was proposed. When adapted compression ratio and turbo-charging are used, the vehicle mileage reduction caused by the low energy density of ethanol can be partially or even fully offset by improved efficiency. When ethanol is blended with a base fuel that has a RON and MON of 92 and 82 (AKI = 87), respectively, and a LHV of 41 MJ/kg, if a maximum of 5% mileage reduction is considered as acceptable, the maximum ethanol content allowed is approximately 52 vol.% and 63 vol.% for NA DISI and TC DISI engines, respectively.
6. The contributors to the improved engine thermal efficiency are the fuel's chemical effect, cooling effect and its octane sensitivity effect, as well as its improved laminar flame speed and the engine downsizing it enables. The contribution ranking is: chemical effect > cooling effect > downsizing effect ≈ flame speed effect > octane sensitivity effect.

Limitation of this review: In this paper, many assumptions based on literature data are made. For example, the selection of required octane improvement for enabling one unit increase of CR is empirical. Ideally, the combustion characteristics of the two engines with the base CR (base fuel) and the high CR (high octane fuel) should be kept the same. However, this is challenging, or even impossible, to achieve across the entire engine operating map. The K values, 0 and −0.3, are considered to represent the average engine K value for NA DISI engines and TC DISI engines, respectively. In reality, K values vary across the entire engine operating map. All of the above assumptions may lead to errors of the final estimation of engine thermal efficiency gains and vehicle mileage range reductions. Nevertheless, this paper presents a methodology of analysing the impact of ethanol blends on modern SI engines in terms of potential engine thermal efficiency gains and vehicle mileage range reduction. More experimental data is welcomed to verify/improve the assumptions/conclusions made in this paper.

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