Investigation of Dual Injection of Ethanol Fuel in Downsized Spark Ignition Engine

By

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A thesis in fulfillment of the requirements for the degree of

Doctor of Philosophy

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Certificate of Original Authorship

I certify that the work in this thesis has not previously been submitted for a degree nor has it been submitted as part of requirements for a degree except as fully acknowledged within the text.

I also certify that the thesis has been written by me. Any help that I have received in my research work and the preparation of the thesis itself has been acknowledged. In addition, I certify that all information sources and literature used are indicated in the thesis.

This research is supported by the Australian Government Research Training Program.

Nizar F. O. Al-Muhsen
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List of Publications

Journal Paper:


Conference Proceedings:


Abstract

Ethanol fuel, as a bioproduct has become a common option to address the issue of energy sustainability. However, the current method of blending ethanol with gasoline does not take the full advantages of ethanol fuel such as its high octane number and great latent heat which potentially allow the increase of the compression ratio and improvement of engine efficiency. Dual injection of ethanol fuel is currently in development and has aimed to make more effective and efficient use of ethanol fuel in SI engines. Experiments were performed on a small single-cylinder four-stroke SI engine equipped with two dual fuel injection systems to investigate both dual injection of ethanol fuel (DualEI) and ethanol port injection plus gasoline direct injection (EPI+GDI). The effect of EPI+GDI on knock mitigation was also investigated.

In the investigation of DualEI, the effects of the ratio of the directly injected (DI) ethanol fuel, spark timing, and DI timing on engine performance, combustion and emissions were analysed. The results demonstrated that the indicated mean effective pressure (IMEP) was improved over all the DI ratios in DualEI engine compared to the original engine with gasoline port injection (GPI) only. This improvement was mainly due to the enhanced combustion quality. However, at higher DI percentages, the over-cooling effect and poor mixture quality adversely affected the combustion performance. The indicated specific nitric oxide emission (ISNO) was reduced by the cooling effect enhanced by ethanol fuel and the DI strategy, but the indicated specific hydrocarbon emission (ISHC) and the indicated specific carbon monoxide emission (ISCO) were raised with the increased DI percentage. As shown by the results for the effect of spark timing, the greatest IMEP and thermal efficiency occurred at spark timing around 30 CAD bTDC at the light load and 23 CAD bTDC at the medium load, which was identified to be the MBT spark timing. The IMEP was increased and the combustion duration was shortened when the spark timing was advanced from 15 CAD bTDC to the MBT timings.

The effect of DI timing associated with spark timing was also investigated. Results showed that the early DI timing enhanced the DualEI engine performance. The variation of IMEP with DI timing was not significant either with early DI timing or in most of the tested conditions with late DI timing. However, the results showed different effects of early and late DI timings associated with the spark timing on engine emissions. With late
DI timing, the engine emissions of ISCO and ISNO increased with the advance of late DI timing and spark timing. With early DI timing, the engine emissions increased with the advance of spark timing. However, the variation of engine emissions with early DI timing was greater than that with late DI timing, showing more unstable combustion.

In the investigation of EPI+GDI, the IMEP did not increase obviously with the increased ratio of EPI. However, the indicated thermal efficiency increased with the increased ratio of EPI because the total heating value of the fuels reduced with the increase of EPI. This was mainly attributed to the enhanced combustion process as the initial and major combustion durations were shortened with the increased ratio of EPI. This also explained why the coefficient of variation of the IMEP reduced with the increased ratio of EPI. As a consequence of improved combustion, the ISCO and ISHC emissions decreased with the increased ratio of EPI. However, the ISNO was increased possibly due to the average combustion temperature increased with and the oxygen added by the increased ratio of EPI.

The EPI+GDI effectively mitigated the engine knock and permitted more advanced spark timing. Results showed that every 10% increment (by volume) of EPI permitted about 2.0 CAD advance of knock limit spark timing. When the EPI ratio was 30% and over, the engine knock was entirely suppressed. The knock intensity was decreased with the increased ratio of EPI until the engine knock was completely suppressed when EPI was increased to 30%.
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### Nomenclature and Abbreviation:

**Acronyms**

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>aTDC</td>
<td>after top dead centre</td>
</tr>
<tr>
<td>bTDC</td>
<td>before top dead centre</td>
</tr>
<tr>
<td>CAD</td>
<td>crank angle degree</td>
</tr>
<tr>
<td>CR</td>
<td>engine compression ratio</td>
</tr>
<tr>
<td>DFI</td>
<td>direct fuel injection</td>
</tr>
<tr>
<td>DI</td>
<td>direct injection</td>
</tr>
<tr>
<td>DualEI</td>
<td>ethanol dual-injection</td>
</tr>
<tr>
<td>EDI+GPI</td>
<td>ethanol direct injection plus Gasoline port injection</td>
</tr>
<tr>
<td>EPI</td>
<td>ethanol port injection</td>
</tr>
<tr>
<td>EVO</td>
<td>exhaust valve opened</td>
</tr>
<tr>
<td>EVC</td>
<td>exhaust valve closed</td>
</tr>
<tr>
<td>GDI</td>
<td>gasoline direct injection</td>
</tr>
<tr>
<td>GDI+EPI</td>
<td>gasoline direct injection plus ethanol port injection</td>
</tr>
<tr>
<td>H/C</td>
<td>hydrogen to carbon ratio</td>
</tr>
<tr>
<td>HRR</td>
<td>heat release rate ($J$/CAD)</td>
</tr>
<tr>
<td>IC engines</td>
<td>internal combustion engines</td>
</tr>
<tr>
<td>IMEP</td>
<td>indicated mean effective pressure</td>
</tr>
<tr>
<td>ISCO</td>
<td>indicated specific carbon monoxide</td>
</tr>
<tr>
<td>ISFC</td>
<td>indicated specific fuel consumption</td>
</tr>
<tr>
<td>ISHC</td>
<td>indicated specific hydrocarbon</td>
</tr>
<tr>
<td>ISNO</td>
<td>indicated specific nitric oxide</td>
</tr>
<tr>
<td>IVC</td>
<td>intake valve closed</td>
</tr>
<tr>
<td>IVO</td>
<td>intake valve opened</td>
</tr>
<tr>
<td>KI</td>
<td>knock intensity</td>
</tr>
<tr>
<td>KLST</td>
<td>knock limit spark timing</td>
</tr>
<tr>
<td>MBT</td>
<td>the spark timing for maximum brake torque</td>
</tr>
<tr>
<td>MFB</td>
<td>mass fraction burnt</td>
</tr>
<tr>
<td>MPFI</td>
<td>multipoint port fuel injection</td>
</tr>
<tr>
<td>O/C</td>
<td>oxygen to carbon ratio</td>
</tr>
<tr>
<td>PFI</td>
<td>port fuel injection</td>
</tr>
<tr>
<td>PI</td>
<td>port injection</td>
</tr>
</tbody>
</table>
RON research octane number
SI spark ignition
TDC top dead centre

Symbols

CA10-90% the major combustion duration (CAD)
CA0-10% the minor combustion duration (CAD)
CA50 the combustion phase when 50% of the fuel is burnt (CAD aTDC)
COV_{IMEP} the coefficient of variation of IMEP
θ Instantaneous crank angle degree (CAD)
θ_{pmax} the phase of peak pressure (CAD aTDC)
γ specific heat capacity
λ stoichiometric air/fuel ratio
σ_{IMEP} the standard deviation in IMEP
t time in msec
DIT’YY’ direct injection timing of YY CAD bTDC
E’XX’ XX% ethanol by volume e.g. E39 is 39% ethanol via port injection plus 61% gasoline via gasoline direct injection
IP Indicated power (W, kW)
m Mass (kg)
\dot{m}_{air} and \dot{m}_{fuel} air and fuel mass flow rates (kg/sec)
\mu_m Micrometer
n polytropic index
\eta_{ind.} indicated thermal efficiency (%)
\eta_{Vol.} engine volumetric efficiency (%)
ρ_{air} air density (kg/m³)
P pressure (kPa, bar)
P₀ the instantaneous pressure at (CAD)
P_{max} maximum cylinder pressure (kPa, bar)
\( Q_{HV,i} \) higher heating value of species \((i)\) (MJ/kg)

ST’XX’ spark timing of XX (CAD bTDC)

T Temperature (°C, K)

\( T_w \) cylinder wall temperature (°C, K)

V cylinder volume (m³)

\( V_d \) displacement volume (m³)

\( V_c \) clearance volume (m³)

\( W_i \) indicated work (J, kJ)

\( \chi_i \) the mole fraction of species
Chapter One

1 Introduction

Nowadays, the global industrial and architectural development has led to a steady increase in the demand for energy resources, especially on the crude oil reserves. Energy resources and environmental protection are the most important issues which require to be addressed urgently by the human society. Increasing the concern for the energy sustainability and environmental protection has led to the search for alternative fuels that are sustainable and environmentally friendly. Among various biofuels, ethanol fuel has been used more and more widely as an alternative fuel to gasoline fuel in spark ignition (SI) engines to address the issue of sustainability [1]. Ethanol is a renewable fuel produced from a variety of bio-resources such as sugarcane and corn. Some of its chemical properties can play a significant role in the internal combustion engine development. Compared to gasoline, ethanol has a greater laminar flame speed, research octane number (RON), and latent heat of vaporisation [2]. Furthermore, the lower adiabatic flame temperature and greater cooling effect could result in a lower combustion temperature and thus less heat losses through the combustion chamber walls [3]. However, ethanol has a lower heating value and smaller stoichiometric air-fuel ratio compared to that of gasoline. Slow evaporation rate is also reported for ethanol fuel at low ambient temperature, so cold starting can be an issue for the SI engines fuelled by ethanol [4].
Promoting ethanol fuel was a key part of the global response to climate change, as announced by the Australian Government in APEC in November 2006. However, E10 in the current market is 10% (by volume) ethanol fuel blended with 90% gasoline fuel prior to fuelling in an SI engine. As the engine uses a constant ratio of ethanol to gasoline, no matter how engine operating condition changes, ethanol merits such as its stronger ability to suppress engine knock than the gasoline fuel has no chance to show, but its disadvantages such as low heating value and low flashing temperature may degrade the engine performance. This had led to a new idea of separately fuelling gasoline and ethanol [5-7]. Inspired by this new idea and advanced technology of dual fuel injection, this project was aimed to investigate and contribute to the development of dual injection of ethanol fuel (DualEI) and gasoline direct injection plus ethanol port injection (GDI+EPI).

The ever increasingly tightened regulations have been enforced to reduce CO₂ emissions produced by passenger cars. The European Commission (EC) published the second set of CO₂ emissions regulation on past 2014. Under the EC regulation, the average new-vehicle CO₂ emissions level have to fall by around 27% (from 130 g/km to 95 g/km) by 2021 [8]. As CO₂ emission is the product of complete combustion, the only way is to reduce fuel consumption. Numerous technologies have been developed to reduce the fuel consumption and CO₂ including engine downsizing. Gasoline direct injection (GDI) system has been implemented in turbocharged and downsized SI engines [1-3]. The GDI technology aimed to reduce the engine size to increase the specific engine output power that could lead to reducing fuel consumption and emissions. To meet the requirement of engine power or maintain the peak power equivalent to that of current engines, downsized engines need increased compression ratio and/or turbocharging. If so, the peak combustion pressure will increase, and then the end-gas may reach the auto-ignition temperature and knock occurs. Auto igniting the end-gas mixture may result in a second
wave pressure causing engine knock. The engine knock can be at a light, medium or heavy level. If the end-gas auto-ignition starts before ignition timing (spark timing), the engine knock phenomenon can develop to a pre-ignition phenomenon resulting in a heavy engine knock and boosted combustion peak pressure to up to 250 bar [9, 10]. In this thesis, the only normal knock is defined and experimentally investigated.

Recently, more attention has been paid to the ethanol fuel in the automotive industry to use the ethanol fuel as an alternative fuel or enhancer agent to gasoline fuel in SI engines. Extensive research has been conducted to investigate the effect of blended fuel (ethanol and gasoline) on the SI engine performance with port fuel injection systems [11-14]. Results showed that a small percentage of ethanol fuel could slightly improve the engine performance. Higher ethanol percentages were commonly used in Brazil and the USA mainly for reducing the consumption of hydrocarbon fuels [15-17]. The effect of E85 (15% gasoline plus 85% ethanol) on the turbocharged GDI SI engine performance, combustion and emission characteristics was experimentally investigated [18]. The E85 fuel was directly injected into the cylinder of the SI engine to achieve higher thermal efficiency and less specific CO$_2$ emission. Their results showed that a greater compression ratio (CR) and higher turbocharged level could be achieved when a greater ethanol ratio was used.

Dual fuel injection technique has been applied in the naturally aspirated and turbocharged engines fuelled with gasoline only or blended fuel (gasoline plus ethanol) [19-21]. Recently, a new technique of ethanol direct injection (EDI) plus gasoline port injection (GPI) has been in development, aiming to use ethanol fuel more efficiently in SI engines with GPI [15, 16]. However, injecting ethanol fuel directly into the combustion chamber in a GPI engine requires two independent fuel-feeding systems. Moreover, the high
percentage of ethanol fuel directly injected into the combustion chamber may cause wall wetting and consequently deterioration in the combustion and emissions [15, 17]. Therefore, to maximize the benefits of using ethanol fuel to the naturally aspirated SI engine, multisets of experiments were conducted at a wide range of engine operating conditions. In the present study, dual ethanol injection strategy (DualEI), GDI plus ethanol port injection (EPI) strategy and engine knock testing were experimentally investigated. This thesis aims to report the investigation of DualEI and GDI+EPI in a small SI engine, including the effect of ethanol fuel on engine knock mitigation.

1.1 Research significance and objectives

As introduced above, ethanol fuel has been used as an alternative fuel in SI engines to improve the engine thermal efficiency and address the issue of sustainability. However, the majority of the existing methods such as blending gasoline and ethanol and pure ethanol (Flex-Fuel engines) do not fully utilize the ethanol potentials in improving SI engine performance. The ultimate goal of this project was to develop techniques that can make the use of ethanol fuel more efficient than the current methods in SI engines to contribute to addressing both issues of energy resources and CO₂ emission. The special aims are,

1. To experimentally investigate the effect of dual injection strategies (PI/DI ratio) on SI engine performance
2. To understand the effect of direct injection and spark timings on DualEI engine performance, and to identify the optimal spark timing for the maximum output power and minimum emissions.
3. To examine the effect of the ethanol port injection on the engine performance, combustion and emissions characteristics with GDI+EPI applied.
4. To assess the potential of the ethanol fuel in knock mitigation and engine downsizing with the application of GDI+EPI.

1.2 Thesis outline

The contents of the subsequent chapters of this thesis are listed and briefly described as follows,

- **Chapter Two** includes a literature review of the relevant works about the dual fuel engines. This review comprises the published papers on dual fuel injection systems in SI engines. The potential benefits of dual fuel injection systems on mixture formation, engine performance are summarised. Past works about the pure ethanol engines and GDI SI engines are reviewed and included in this chapter.

- **Chapter Three** introduces the setup of the research engine and experimental methodology. It presents the details of the experimental apparatus used, including the specifications of the research engine, data acquisition systems, engine control system, dual-injection system, and exhaust gas analyser.

- **Chapter Four** presents and discusses the experimental results of the engine performance equipped with DualEI system. The variations of DualEI engine performance, combustion and emissions characteristics with spark timing are investigated. Furthermore, the associated effect of the spark and direct injection timings on engine performance are experimentally investigated. Deep analysis of the combustion and emissions characteristics are presented to understand the mechanism behind the engine performance variation.
• **Chapter Five** investigates the effect of EPI/GDI ratio on the engine performance. A comprehensive analysis of the combustion and emissions characteristics are carried out at various ratios of EPI/GDI under different conditions including three engine speeds and two loads.

• **Chapter Six** presents and discusses the results of the effect of EPI/GDI ratio on the engine knock limit spark timing. A basic thermodynamic analysis is performed to evaluate the increase of compression ratio that can be achieved due to extending the knock limits.

• **Chapter Seven** concludes the thesis by summarising the contents of each chapter and recommending future work.
Chapter Two

2 Literature review

Fossil fuel depletion and global warming have been identified as worldwide issues and need to be addressed urgently [22]. Consequently, more strict legislations such as Euro 6 and China Stage 6 have been set to limit pollutant emissions including CO$_2$ produced by internal combustion (IC) engines [23, 24]. Exploring an alternative bio-fuel such as ethanol has become a crucial topic for researchers and the automotive industry recently. Dual fuel injection, PFI+DFI, and the engine downsizing have emerged as new concepts in automotive engineering. Research is required to investigate and develop these new ideas.

2.1 Properties of ethanol fuel

Improving the use of fossil fuels in the current markets or adopting an alternative or renewable fuel is the research focus in the IC engines area. Ethanol fuel has been a promising alternative to gasoline fuel [14, 25]. Adopting ethanol fuel as an alternative fuel to gasoline fuel can make a sustainable reduction of emissions (CO$_2$) to the environment [1]. Recently, ethanol as a biofuel has been considered as an attractive option that can be added to gasoline fuel or even replace it in IC engines [26, 27]. It was found that ethanol fuel could contribute to improving the combustion quality and reducing the pollutant emissions due to its properties such as the oxygen content and high flame speed [28-30]. Consequently, the engine thermal efficiency was enhanced, and the exhaust emissions were reduced [6, 20, 31]. Table 2.1 shows the properties of unleaded gasoline and ethanol fuel.
Table 2.1 Properties of gasoline and ethanol fuels.

<table>
<thead>
<tr>
<th>Property</th>
<th>Gasoline</th>
<th>Ethanol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity (kg/m.s)</td>
<td>0.0004549 [32]</td>
<td>0.001007 [32]</td>
</tr>
<tr>
<td>O/C (Atom ratio)</td>
<td>0.0 [2]</td>
<td>0.5 [2]</td>
</tr>
<tr>
<td>Research octane number (-)</td>
<td>91</td>
<td>106 [5]</td>
</tr>
<tr>
<td>Lower heating value (MJ/kg)</td>
<td>42.9 [5]</td>
<td>26.9 [5]</td>
</tr>
<tr>
<td>Laminar flame speed @ 100kPa, 100°C, λ=1 (m/s)</td>
<td>~0.49 [33]</td>
<td>~0.62 [33]</td>
</tr>
</tbody>
</table>

Laminar flame speed plays an important role in engine combustion [34, 35]. Figure 2.1 (a) shows the variation of the laminar flame speed of the 5-dimethylfuran (DMF), gasoline and ethanol fuels with the equivalence ratio [36]. As shown in Figure 2.1, the ethanol flame speed is greater than that of gasoline over a certain range of lambda, which should result in a faster combustion process. Consequently, the heat losses between the burned gases and the combustion chamber walls may be reduced by the shortened combustion duration, improving the engine thermal efficiency and combustion stability [37, 38].
Figure 2.1 Laminar flame speed of DMF, gasoline and ethanol fuels at various equivalence ratios [36].

Ethanol fuel has a stronger cooling effect than that of gasoline because it possesses a greater latent heat of vaporization than that of gasoline when the ambient temperature is higher than 375 K. The cooling effect of fuel also depends on the injection strategy. Injecting ethanol at an early stage of the intake stroke gives a sufficient time for ethanol to evaporate and extract as much as heat from the fresh charge [39]. Longer evaporation process may lead to a greater cooling effect, reducing the temperature and increasing the density of the intake charge. This might result in the engine volumetric efficiency improvement as well. The relationship between the ethanol injection timing and charge cooling effect was observed in the volumetric efficiency vs engine speed [40, 41]. In fact, the engine thermal efficiency might improve with the enhanced cooling effect due to the reduced heat losses between the in-cylinder gases and combustion chamber walls [3].

The combined effect of ethanol latent heat of vaporization with the greater octane number and higher auto-ignition temperature may suppress the engine knock, permitting the
increase of compression ratio and then thermal efficiency [7, 42]. The cooling effect of fuel is led by the vaporization ability of fuel in various ambient temperatures [43, 44]. The fuel evaporation speed is defined by the distillation characteristics. The distillation characteristics of blending ethanol/gasoline fuels are shown in Figure 2.2. As shown in Figure 2.2, the initial boiling point of ethanol is lower than that of gasoline. This was attributed to the single component of ethanol that has a defined boiling point of 78°C (351.15 K) [45].

![Figure 2.2 Distillation curves for blending ethanol/gasoline fuels (E0 for gasoline only) [45].](image)

It is well known that the saturation vapor pressure of liquid fuels can strongly indicate the quality of the evaporation process that influences the combustion performance and the consequent emissions characteristics in SI engines [46]. Figure 2.3 shows the variation of the saturation vapor pressure with temperature for gasoline, Iso-octane, and ethanol fuels [32, 47]. As shown in Figure 2.3, the evaporation process of ethanol fuel becomes faster than that of gasoline when the temperature exceeds 375 K. It was found that the calculated
bulk gas temperature from the unfired in-cylinder pressure was over 600 K at 40 CAD bTDC [44].

Figure 2.3 Vaporisation curves for Gasoline, Iso-octane and Ethanol [32, 47].

The previously mentioned studies demonstrated that some properties of the ethanol fuel could be taken as advantages to improve the thermal efficiency of the IC engines. However, ethanol fuel has certain disadvantages such as slow evaporation rate at low ambient temperature, and low heating value as reported to be issues in engine cold starting conditions [4, 48]. To solve the problem of cold starting in an ethanol fuel engine, flex-fuel systems were adopted to aid the SI engine cold starting with gasoline port injection [40]. Flex fuel systems have been largely used in the US and South America such as Brazil to reduce the CO₂ emission by transferring the SI engines from gasoline-fuelled to be 100% ethanol engines [29, 49, 50]. However, the flex-fuel vehicles (FFVs) were equipped with conventional port fuel injection systems resulting in moderate engine performance and relatively poor mileage fuel economy.
2.2 Injection strategies in SI engine development

2.2.1 Blended fuels for SI engines

Extensive research has been conducted to investigate the effect of blended fuel (ethanol plus gasoline) on conventional SI engines. Bielaczyc et al. [13] and Koç M. et al. [51] tested an SI engine under a wide range of ethanol-gasoline blended fuel with ratios from 0% to 85% (E0% to E85%). The reported results showed that the engine performance was improved and emissions (HC, CO and NO) were reduced when the ethanol ratio was increased from E0% to E85%. Turner et al. [31] tested the effect of the ethanol-gasoline blended ratio on combustion and the emissions characteristics. Their results demonstrated that the combustion quality improved and the exhaust emissions were decreased with the increased ethanol percentage in the blended fuel. This improvement was attributed to the greater combustion speed of ethanol fuel and the oxygen content. Ramadhas et al. [11] investigated the combustion and emissions characteristics of different ethanol-gasoline blending ratios from 5% to 20% of ethanol, using a multipoint port fuel injection (MPFI) system on a small car engine. They concluded that using the MPFI system to the ethanol-gasoline blended fuel could improve the combustion quality and reduce the CO emissions by 15%, HC emissions by 20% and NO emissions by 45% at various load and speed conditions.

The ethanol greater octane number and latent heat of vaporisation allowed a higher compression ratio and more advanced level of turbocharged engines [52, 53]. Additionally, the cooling effect of the ethanol fuel could be further enhanced if it was injected directly into the cylinder [54]. This might reduce the combustion temperature and thus the nitric oxide emission (NO). Jiang C. et al. [55] experimentally investigated the combustion characteristics for three blended fuels with gasoline, which were ethanol,
DMF and methanol, on a 4-stroke SI engine equipped with an optical diagnostic facility. Compared to gasoline and the other blended fuels, ethanol blended fuel and the 100% ethanol fuel showed the fastest combustion speed and the shortest combustion duration. Moreover, the combustion speed of ethanol increased by 10% to 15% when the engine load was increased from 3bar to 5bar IMEP. Hydrous ethanol (6.8% water content in ethanol) vs. ethanol-gasoline blended fuel were experimentally investigated, and the combustion and emissions characteristics were compared [14, 56]. The results of hydrous ethanol showed better engine performance than that of ethanol-gasoline blended fuel. The specific fuel consumption was higher with hydrous ethanol than that with ethanol-gasoline blended fuel due to the lower heating value and the water content. However, the engine thermal efficiency with hydrous ethanol fuel was greater, and the CO and HC were lower than that with blended fuel, although the NO emission was higher.

### 2.2.2 Direct injection and dual injection in SI engines

The number of gasoline direct injection (GDI) engines in vehicles continues to increase globally [57]. However, GDI engines have come with drawbacks such as increased particulate matter (PM) and hydrocarbon (HC) emissions due to its resulting fuel impingement and the non-homogenous combustion [58, 59]. Wang et al. [58] investigated the effect of fuel type on emissions of a GDI engine and concluded that using high direct injection (DI) pressure plus ethanol fuel could potentially reduce the HC and PM emissions because the high injection pressure resulted in fine fuel droplets and reduced the fuel impingement.

To exploit the potentials of ethanol fuel to SI engines, Kasseris E. and Heywood J. [6, 7] developed a methodology to quantitatively evaluate the effect of ethanol charge cooling and auto-ignition resistance on the engine knock limit when dual injection of blended
gasoline and ethanol fuels E85 was applied. Bielaczyk et al. [13], Turner et al. [31] and Koç M. et al. [51] tested an SI engine under a wide range of ethanol-gasoline blended fuel ratios. Their results showed that the engine performance improved, and emissions reduced when the ethanol ratio was increased. The effect of ethanol port injection (PI) on GDI engine performance was widely investigated. Kim et al. [60] investigated the EPI with GDI to improve the engine thermal efficiency by increasing the compression ratio and extending the engine knock limit. Liu et al. [19] observed an improvement in the engine performance occurred using EPI in the GDI engine. Wang et al. [61] concluded from their experimental investigation that the engine knock could be effectively mitigated when the DI or PI of ethanol was used.

The EDI+GPI strategy in the SI engine has been in development in recent years [5]. The experimental results in [5, 26] showed that the EDI+GPI strategy improved the engine thermal efficiency and reduced the indicated specific nitric oxide (ISNO) emission in a certain range of EDI ratios. However, the indicated specific hydrocarbon (ISHC) and carbon monoxide (ISCO) emissions increased significantly. Huang et al. numerically simulated and analysed the processes of fuel injection, mixing and combustion to understand the causes of the above experimental results [4, 54]. Their results showed that the engine performance improved when the ratio of ethanol fuel increased by up to 60% because the higher flame speed of ethanol fuel contributed to the greater pressure rise rate and maximum cylinder pressure. Further increase in the EDI ratio led to poor mixture quality and combustion due to the ethanol impingement and over-cooling effect. As a consequence, the IMEP and thermal efficiency began to decrease with the increase of EDI when the EDI ratio was increased from 60% to 100% [54]. Due to the same causes, the ISCO and ISHC emissions increased with the increased ratio of EDI [4].
Although ethanol is a renewable fuel, adopting ethanol fuel as an alternative fuel to gasoline could come with considerable challenges [62, 63]. Nakata et al. and Taniguchi et al. [29, 63] from Toyota Motor Company investigated the feasibility of running the SI engine with 100% ethanol. They concluded that ethanol fuel could improve the engine thermal efficiency due to its greater octane number and cooling effect than that of gasoline. It was reported that 100% of ethanol DI could improve the engine performance and reduce the NO emission [64, 65]. This was mainly attributed to the ethanol oxygen content, fast flame speed, and adiabatic flame temperature. However, the authors’ results showed that the over-cooling effect associated with the severe fuel impingement caused by the high percentage of EDI was represented as an issue. Therefore, controlling the mixture formation processes inside the combustion chamber can be one of the challenges. DualEI system is a new technology that could control the amount of ethanol fuel port injected (injection durations) to the amount of fuel directly injected based on the engine needs and requires investigation [64, 66].

2.2.3 Effect of spark and direct injection timings on combustion

The importance of the DI and spark timings lies in its effect on the mixture quality and combustion efficiency. Advanced DI timing from TDC gives more time for the fuel spray to be atomized, evaporated and then mixed well with air. However, as shown in Figure 2.4, fuel impingement can become severe in the combustion chamber of a small engine, resulting in uncompleted combustion and large HC and CO emission [26]. Retarding the DI timing toward TDC may reduce the time available to form a homogeneous air-fuel mixture. However, the fuel impingement may decrease due to the piston being moved further from the fuel injector when the DI timing is retarded, as shown in Figure 2.4. Davy et al. [67] experimentally investigated the effect of DI timing on the fuel spray in a GDI
prototype engine. As shown in Figure 2.4, the fuel impingement, especially to the piston surface reduces with the retarded DI timing. However, the NO emission might increase due to the combustion temperature being increased by the increased heat release rate in combustion [2, 43].

![DI timing 390 CAD](image)

![DI timing 480 CAD](image)

**Figure 2.4 Spatial and Temporal Spray Distribution for Two Direct Injection Timings, at 390 and 480 CAD [67].**

The effect of port injection timing at different blending ratios of gasoline with ethanol fuels at a wide range of engine operating conditions was investigated experimentally [68]. Their results showed that the combustion efficiency was affected by the timing of injection due to its effect on the mixture quality. Early injection timing provided more time for the air-fuel mixture to be homogenously mixed, but later injection timing reduced the time, reducing the combustion efficiency due to the mixture quality deterioration.

The combustion can be characterised by the combustion phase (CA50) [69, 70]. CA50 is defined by the crank angle degree (CAD) at which fifty percent of the fuel mass has been
burnt [70, 71]. The importance of the CA50 comes from its influence on the peak in-cylinder pressure and its position from the TDC. CA50 can be adjusted to its optimal position by changing the factors affecting combustion including the operation parameters like spark timing. Spark timing is a key parameter that controls the combustion phase and optimises the output power [69, 71]. Ayala et al. [71] examined the correlation between the CA50 and the IMEP at a range of equivalence and compression ratios. Their results of the major combustion duration (CA10-90%) and initial combustion duration (CA0-10%) were analysed and correlated with the engine thermal efficiency. The results showed that the best thermal efficiency occurred when the CA10-90% was within 30 CAD and CA0-10% was within 40 CAD.

2.3 Engine downsizing and knock mitigation

The U.S. Department of Transportation and the Environmental Protection Agency (EPA) have put ambitious objectives on the fuel economy and rigorous limits to the GHG emissions regarding the light-duty vehicles [72]. Recently, engine downsizing has been identified as one of the strategies to reduce fuel consumption and CO₂ emission [73]. However, this idea has brought challenges, and the engine knock is at the top of them [74]. The maximum power of the downsized engine can be maintained by increasing the engine compression ratio and/or boosting the pressure in the intake manifold using supercharging or turbocharging [75]. This increases the chances for engine knock to occur [3, 76, 77]. As shown in Figure 2.5, the auto-ignition of the end-gas in front of the flame front that is propagating from the spark plug is the main cause behind the engine knock phenomenon [78] which is also commonly known as a normal knock as studied in this thesis.
The way to avoid the engine knock includes using fuels with high octane number [79, 80], reducing the peak combustion temperature and pressure and keeping the end-gas temperature below the auto-ignition temperature of the used fuel [26, 81]. Many researchers have investigated the potential of ethanol fuel to resist the SI engine knock by adopting different technologies such as dual fuel injection technology [4, 6, 7, 26, 52, 82]. Dual fuel injection applied to SI engines was widely investigated in recent years [18, 83]. Ford Motor Company investigated the effect of dual injection strategy on an ‘Ecoboost’ engine performance that was a turbocharged GDI added to a gasoline port injection SI engine [18]. Their results showed that the E85 direct injection significantly
mitigated the engine knock and thus improved the indicated thermal efficiency. As a result, higher levels of compression ratio and intake manifold pressure could be performed. A summary of the main key findings of the literature review is attached to the thesis appendix. This summary is divided into three main sections, which are ethanol combustion and emissions characteristics, ethanol dual injection strategy and spark and injection timings.

**Problem Definition and Research Uniqueness**

From the above review, extensive research has been conducted on renewable fuels for IC engines. The majority of the published papers was focused on the effect of the dual-fuel injection on the knock mitigation, performance, and emissions of the SI engine fuelled with different types of renewable fuel such as ethanol, methanol, and 2,5-dimethylfuran. However, the modern technologies such as the advancement in the engine control systems have opened a narrow but promising window to conduct such in-depth research. These technologies have only become available in the last few years.

The dual-injection of ethanol fuel (DualEI) in SI engines has not been investigated yet. Besides, the effects of engine control parameters such as the ratio of the direct fuel injection to the port fuel injection, direct injection timing and spark timing on the SI engine performance including the combustion and emissions characteristics were seldom reported in these publications. In this thesis, the effect of the DI/PI ratio on a DualEI engine performance will be first tested (Section 4.1). Secondly, the results of the effect of spark timing at different DI ratio on engine performance will be also reported (Section 4.2). Finally, the associated effect of the DI timing and spark timing will be investigated (Section 4.3). In each section, the combustion and emission characteristics will be comprehensively discussed and analysed.
As reported previously, different technologies had been investigated in development to exploit the benefits of ethanol as an alternative fuel to the SI engines. The dual injection of ethanol fuel plus gasoline fuel (GDI+EPI) to the SI engines has been in development. In this thesis, in addition to the DualEI strategy, GDI+EPI will be used to provide a better method to fully utilize the merits of ethanol fuel in improving the performance of SI engines. The effect of EPI ratio on the GDI engine performance will be experimentally investigated (Chapter 5). The results of the combustion and characteristics of the dual fuel (ethanol plus gasoline) dual injection (GDI+EPI) will be discussed and analysed. Moreover, the effect of EPI ratio on the knock limit spark timing of the GDI engine will be also examined (Chapter 6).
Chapter Three

3 Experimental setup and methods

3.1 Research engine with dual fuel injection

Experiments were performed on a Yamaha YBR250 motorcycle SI engine modified to meet the research needs. As shown in Figure 3.1, the research engine is a single cylinder four-stroke air-cooled naturally aspirated engine. Table 3.1 shows the major specifications of this research engine. The engine was modified to be equipped with a dual injection system, port plus direct injection, by Hents Technology. The engine modification included the installation of a DI injection system and an electronic control unit (ECU) which was used to control the throttle position, spark timing, and DI timing and fuel pressure. The quantity of the fuel injected per cycle and the ratio of the fuel port or directly injected can be adjusted by changing the injection pulse width and fuel pressure as input to the ECU. Figure 3.3 shows a photo of the research engine test rig.

The DI injection system consisted of a returnless high-pressure mechanical fuel pump and a six-hole high-pressure injector [84]. The high-pressure injector specifications are listed in Table 3.2. The injector was side-mounted between the intake valve seat and the spark plug, as shown in Figure 3.2. The angles of the injector position are 15° from the horizontal surface of the cylinder head to the axis of the injector and 12° from the vertical surface of the cylinder head [85]. Two super-micro oval fuel flow meters with ±5 ml accuracy were used to measure the fuel flow rates. An eddy current dynamometer was connected to the engine to set the engine speed and torque. A Kistler 6115B spark plug pressure transducer and a Kistler 5011 charge amplifier were used to measure the in-
cylinder pressure. K-type thermocouples were used to measure the cylinder head and exhaust gas temperatures with a resolution of 0.1 °C and uncertainty of 0.35%. A MEXA-584L Horiba exhaust gas analyser was used to measure the exhaust gas emissions of CO, CO\textsubscript{2}, HC, NO and lambda (λ). Before the experiment started, the H/C and O/C ratios were manually set in the exhaust gas analyser based on the volume ratio of the ethanol and gasoline fuels. The H/C and O/C ratios were changed from 3.0 and 0.5 respectively for ethanol fuel only, and to 2.25 and 0.0 respectively for gasoline only testing. The intake airflow was stabilised in an 80L intake buffer tank, and the intake air flow rate was measured using a ToCeil20N thermal air-mass flow meter. Both fan and a radiator were used to reach the designated temperature of the engine body, cylinder head, and lubricating oil. As shown in Figure 3.1, two fuel tanks were used to supply the engine with gasoline and ethanol separately.
Chapter Three

Experimental setup and methods

Figure 3.1 Schematic Diagram of the Dual Injection SI Engine.

Figure 3.2 The relative position of the HP injector and spark plug in engine cylinder head [85].
Figure 3.3 Research Engine Test Rig.
Table 3.1 Research Engine Specification.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Type</td>
<td>Single Cylinder, 4-stroke, SOHC</td>
</tr>
<tr>
<td>Displacement</td>
<td>249.0 cc</td>
</tr>
<tr>
<td>Bore</td>
<td>74.0 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>58.0 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>9.8:1</td>
</tr>
<tr>
<td>Fuel Injection System</td>
<td>Dual Injection (direct and port)</td>
</tr>
<tr>
<td>Intake Valve Open (IVO)</td>
<td>382.2 CAD bTDC</td>
</tr>
<tr>
<td>Intake Valve Close (IVC)</td>
<td>126.2 CAD bTDC</td>
</tr>
<tr>
<td>Exhaust Valve Open (EVO)</td>
<td>594.6 CAD bTDC</td>
</tr>
<tr>
<td>Exhaust Valve Close (EVC)</td>
<td>340.7 CAD bTDC</td>
</tr>
</tbody>
</table>

Table 3.2 Specification of the Direct Fuel Injector [48, 86]

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manufacturer</td>
<td>Bosch Engineering</td>
</tr>
<tr>
<td>Operating pressure</td>
<td>Up to 500 bar</td>
</tr>
<tr>
<td>Number of holes</td>
<td>6 holes</td>
</tr>
<tr>
<td>Hole diameter</td>
<td>110 µm</td>
</tr>
<tr>
<td>Flow rate @ 100 bar</td>
<td>Up to 1640 g/min</td>
</tr>
<tr>
<td>Spray angle single beam</td>
<td>17°</td>
</tr>
<tr>
<td>Operating temperature range</td>
<td>-31 to 130 °C</td>
</tr>
</tbody>
</table>

3.2 Experimental control and data acquisition systems

3.2.1 Engine control systems

The ECU was provided by Hents Technology to adjust the experimental conditions of the modified engine to meet the research needs. The ECU was used to set up electronically the throttle position, fuel volume per cycle and the ratio of fuels directly and port injected, DI pressure, DI and spark timings. The quantities of the fuel directly and port injected were set by adjusting the injection widths of both injectors. Moreover, the ECU was used to monitor the engine cylinder head temperature, the exhaust gases temperature, the
air/fuel equivalence ratio, and the engine speed. The engine torque was provided by the eddy current dynamometer, which was coupled with the research engine.

3.2.2 Data acquisition systems

An open loop Kistler pressure transducer and charge amplifier via a HORIZON combustion analyser circuit were used to measure the in-cylinder pressure, as shown in Figure 3.4. A Kistler-2614b4 pulse multiplier was connected between the crank angle encoder and combustion analyser, as shown in Figure 3.4. It is capable of giving four CAD resolutions for the in-cylinder pressure recording, from 0.1 to 2.0 CADs. A 0.5 CAD resolution was used for most of the experiments, and a 0.1 CAD resolution was used for detecting the engine knock. The in-cylinder pressure data was recorded with 100 consecutive cycles in each sample. The ensemble average of the in-cylinder pressure data was used in calculations of IMEP and the combustion analysing. For each testing condition, five samples were recorded at a sample rate of 1.0 Hz, and the average values were used in the calculations and analyses. To ensure the consistency of lambda, the lambda values were measured using exhaust gas analyser and the Bosch lambda sensor simultaneously.
3.3 Combustion and emissions data analysis methods

**Indicated Mean Effective Pressure (IMEP)**

IMEP is a key parameter to evaluate the engine volume based power output. It can be calculated by cyclic integration of the in-cylinder pressure with respect of the cylinder volume as mathematically described in Equation 3.1

\[
IMEP = \frac{W_i}{V_d} = \frac{1}{V_d} \int PdV \tag{3.1}
\]

**The Coefficient of Variation of the IMEP (COV\_IMEP)**

The coefficient of variation of the indicated mean effective pressure (COV\_IMEP) is a parameter to measure the combustion stability, which is derived from the combustion
pressure [69]. As shown in Equation 3.2, it is the standard deviation of IMEP (\( \sigma_{IMEP} \)) over the mean IMEP.

\[
COV_{IMEP}(\%) = \frac{\sigma_{IMEP}}{IMEP}
\]

Equation 3.2

**Indicated Thermal Efficiency**

\[
\eta_{Indicated} = \frac{IP}{\dot{m}_{fuel} \cdot Q_{HV,fuel}}
\]

Equation 3.3

where the \( Q_{HV,fuel} \) is the fuel heating value, and the mass flow rate of the fuel is represented by \( \dot{m}_{fuel} \). The indicated thermal efficiency (\( \eta_{Indicated} \)) was defined as the indicated power (IP) per cycle over the rate of input heating value per cycle, as shown in Equation 3.3 [87]. The IP was calculated based on the average pressure value inside the combustion chamber. The ensemble average pressure of 100 consecutive cycles was used in the calculations and analyses. The average value of five recordings was used to ensure the accuracy of the experimental result at each targeted condition.

**Mass Fraction Burn and Combustion Duration**

A method was developed by Rassweiler-Withrow to analyse the released heat by combustion [88]. This method uses the recorded in-cylinder pressure trace to calculate the heat release rate during the combustion process. The total pressure inside the engine cylinder was assumed equal to the pressure rise that is caused by fuel burning and the piston movement. The method assumed that the pressure traces of the compression and expansion strokes are polytropic processes based on Equation 3.4.
\[ PV^n = \text{Const.} \]  

Equation 3.4

The mass fraction burn (MFB) can be calculated from Equation 3.5, which is denoted as \( X_\theta \) in the Equation 3.5.

\[ MFB = X_\theta = \frac{P_2 - P_1}{P_3 - P_1} \]  

Equation 3.5

\[ P_2 = P_\theta \cdot \left( \frac{V}{V_c} \right)^n \]  

Equation 3.6

where, \( P_1 \) and \( P_2 \) are the intersection points of the compression trace line and the expansion trace line respectively with the minimum volume line, as shown in Figure 3.5 (a). \( P_2 \) represents the projection of \( P_\theta \) on the minimum volume line as shown in Equation 3.6. \( P_\theta \) is the instantaneous pressure at CAD, which starts from 0.0% combustion (mixture only) to 100% combustion. \( V \) is the instantaneous volume, \( V_c \) is the minimum volume (clearance volume), and \( \theta \) is the crank angle degree. The polytropic index (\( n \)) was calculated for each testing condition as the slope of the function of \( \log(P) \) and \( \log(V) \), as shown in Figure 3.5 and Equation 3.7.

\[ n = \frac{\Delta \log(P)}{\Delta \log(V)} \]  

Equation 3.7

The combustion phases were determined by using the MFB curve which was calculated based on the logarithmic pressure-volume graph, as shown in Figure 3.5 (a) [88]. As shown in Figure 3.5 (b), the initial combustion duration (CA0-10%) was defined in CADs, which was calculated from the spark timing to the timing when 10% of the fuel
mass is burnt [69]. The combustion stability significantly depends on the initial combustion duration, and it is directly related to the mixture temperature and its quality at the outset time of ignition [26]. The major combustion duration (CA10-90%) was defined in CADs, which was calculated from 10% to 90% of the fuel mass fraction burnt. Optimising the combustion phase is a vital parameter that potentially affects the engine performance and the consequent emissions [69]. The CA50 defines the crank angle degree (CAD) at which 50% of the fuel mass has burnt, as shown in Figure 3.5 (b) [70, 71].

Figure 3.5 Demonstration of Rassweiler and Withrow Algorithm and Combustion Phases Calculation Method [88].
Heat Release Rate (HRR)

The gross HRR was written based on the first law of thermodynamics [69]. The method was based on the assumption that the pressure rise inside the engine cylinder is directly related to the chemical energy released by combustion and the cylinder volume change due to the piston movement. The gross HRR is defined in Equation 3.8.

\[
\frac{dQ_{ch}}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}
\]

\[+ V_{cr} \left\{ \frac{T'}{T_w} + \frac{T}{T_w(\gamma - 1)} + \frac{1}{bT_w} \ln \left( \frac{\gamma - 1}{\gamma' - 1} \right) \right\} \frac{dP}{d\theta} \quad \text{Equation 3.8}\]

\[+ \frac{dQ_{ht}}{d\theta}\]

In this thesis, the effect of the crevice and the convective heat transfer to the combustion chamber walls on the net HRR were disregarded, so the HRR can be defined in Equation 3.9 [89].

\[HRR = \frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} \quad \text{Equation 3.9}\]

where \(\gamma\) is the specific heat capacity ratio of the in-cylinder gas, and \(\theta\) is the instantaneous crank angle degree. The \(\gamma\) value was chosen as 1.3 for the expansion process and 1.37 for the compression process [90].

Indicated Specific Emissions

Four main engine exhaust gas emissions were experimentally measured during the experiments, which included the nitric oxide (NO), carbon monoxide (CO), carbon dioxide (CO\(_2\)) and hydrocarbons (HC). The specific emission is one of the most common
methods of measuring the pollutants in the engine exhaust gases [87]. The indicated specific emission \( \frac{g_{\text{Emission}}}{kW \cdot hr} \) can be defined in Equation 3.10.

\[
\text{Indicated Specific Emission (ISE)} \left( \frac{g_{\text{Emission}}}{kW \cdot hr} \right) = \frac{\dot{m}_{\text{Emission}}}{IP} \tag{Equation 3.10}
\]

where \( \dot{m}_{\text{Emission}} \) is the flow rate of the emission in \( g_{\text{Emission}}/hr \), and the \( IP \) is the indicated power (kW).

**Engine volumetric efficiency**

\[
\eta_v = \frac{\dot{m}_a}{\rho_a V_d N} \tag{Equation 3.11}
\]

where \( \dot{m}_a \) is the intake mass flow rate of air, \( \rho_a \) and \( V_d \) are the air density (at the ambient pressure and temperature) and engine displacement volume, respectively. \( N \) is the engine speed (RPM), and \( n \) is the number of revolutions per cycle.

**Calibration of fuel flow meters and Verification of Lambda measurements**

Two separate fuel tanks were set up to supply the ethanol fuel and gasoline fuel separately to the research engine. A low-pressure fuel pump operated at a fuel pressure of 0.25 MPa for supplying ethanol fuel, and 0.3MPa for gasoline fuel. The high-pressure pump was powered by an electrical motor at 0.55 kW and 1400-RPM. To assure the measurement precision of the mass of fuel injected per cycle, a super-micro oval fuel flow meter with \( \pm 5 \) ml accuracy was used to measure and calibrate the ethanol and gasoline fuels flow rates. The fuel flow meters were calibrated. Figure 3.6 shows the calibrations between the computer input, which is a function of the injection pulse width in \( \mu \text{sec} \) and the mass of the fuel injected. As shown in Figure 3.6, the calibration is linear with correlation coefficients greater than 0.98.
Figure 3.6 Port and Direct Fuel Injection Calibrations.
Chapter Four

4 Dual ethanol injection (DualEI) engine performance

In order to maximize the benefits of using ethanol fuel in SI engines, dual fuel injection with ethanol direct injection (EDI) plus ethanol port injection (EPI) (DualEI) was experimentally investigated. In this chapter, the results of the engine performance including the combustion and emissions characteristics will be presented and analysed. The effect of EDI ratio (volume based) on the DualEI engine performance will be first examined. Secondly, the effects of the spark timings on combustion and emissions characteristics will be presented and discussed in 4.3. Finally, in 4.4, the integrated effects of the DI and spark timings on the DualEI engine performance will be presented and analysed.

In the experiments, the engine was started and warmed up using GPI only until the cylinder head temperature became stable at around 200 °C (±5). After that, the GPI was switched to EPI, and the air/fuel equivalence ratio was kept at around the stoichiometric condition (λ≈1). To ensure the equivalence ratio correctly measured, Horiba MEXA-584L gas analyser [91] and Bosh Wide-band lambda sensor were used independently to verify the measured value of lambda. The lambda was calculated upon the measurement of HC, CO, CO₂, and O₂ using Equation 4.1 [91].

\[
\lambda = \frac{[\text{CO}_2] + \frac{[\text{CO}]}{2} + [\text{O}_2]}{1 + \frac{\text{H}_4}{4} - \frac{\text{O}_2}{2}} \times \left( \frac{\text{H}_4}{4} \times \frac{3.5}{3.5 + \frac{\text{CO}}{2}} - \frac{\text{O}_2}{2} \right) \times \left( \frac{[\text{CO}_2] + [\text{CO}]}{[\text{CO}_2] + [\text{CO}]} + (K_1 \times [\text{HC}]) \right)
\]

Equation 4.1
where [ ] is the volume based concentration in percentage for CO₂, CO and O₂, and ppm for HC. K₁ is the conversion factor for HC in n-hexane (C₆H₁₄) equivalent, 0.0006 in Equation 4.1. For ethanol (C₃H₆O), the H/C value was 3.0 and the O/C value was 0.5.

### 4.1 Effect of ethanol DI ratio on DualEI engine performance

To investigate the effect of ethanol DI ratio on DualEI SI engine performance, experiments were conducted at a light engine load with 20% throttle opening and a medium engine load with 33% throttle opening, as shown in Table 4.1. The volume ratio of ethanol fuel directly injected was increased from 0.0% (EPI only) to 100% (EDI only), by increasing the direct injection duration and decreasing the port injection duration. However, the total energy inputs of fuel per cycle were kept unchanged (around 540 J/cycle in medium load and around 400 J/cycle in light load). The DI timing was set at 300 CAD bTDC and PI timing at 410 CAD bTDC. The DI pressure was set at 4.0 MPa and the PI pressure at 0.25 MPa.

<table>
<thead>
<tr>
<th>Light load (20% Throttle opening)</th>
<th>Medium load (33% Throttle opening)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed (RPM)</td>
<td>3500</td>
</tr>
<tr>
<td>Spark Timing (CAD bTDC)</td>
<td>MBT</td>
</tr>
<tr>
<td>DI% (volumetric percentage)</td>
<td>Light load: DI0 (PI only), DI32(32%), DI46, DI55, DI66, DI80 and DI100</td>
</tr>
<tr>
<td>Port Fuel Injection Timing</td>
<td>Medium load: DI32, DI46, DI55, DI66, DI80 and DI100</td>
</tr>
<tr>
<td>Direct Fuel Injection Timing</td>
<td>410 CAD bTDC (Original PI timing)</td>
</tr>
<tr>
<td></td>
<td>300 CAD bTDC</td>
</tr>
</tbody>
</table>

Pre-experiments were conducted to find the MBT spark timing at a fixed engine speed of 3500 RPM. MBT timing is defined as the spark timing for the maximum brake torque. At
fixed engine speed conditions, the MBT timing is also the spark timing for maximum BMEP. Because BMEP is proportional to IMEP, the spark timing for the maximum BMEP is also that for the maximum IMEP. Figure 4.1 shows the variation of IMEP with spark timing when the ratio of EDI was varied from 0% to 100%. The spark timing was swept from 15 to 42 CAD bTDC at the light load and from 15 to 32 CAD bTDC at the medium load.

As shown in Figure 4.1, the IMEP increases with the advanced spark timing from 15 CAD bTDC to around 30 CAD bTDC at light load and to around 23 CAD bTDC at medium load. When the spark timing is further advanced, the IMEP decreases with the proceeded spark timing. It was concluded that the best IMEP could be achieved when the majority of the combustion takes place near the TDC [69]. The effect of spark timing on IMEP will be discussed in section 4.1.1. The improvement of IMEP with advanced spark timing may be attributed to the combustion quality enhancement and the right phase (near TDC) at which the largest portion of combustion takes place [92]. On the other hand, the reduction in the IMEP with further advance of spark timing can be attributed to the negative work generated due to the early spark timing. Based on the results shown in Figure 4.1, the MBT spark timing is set to be 30 CAD BTDC for a light load and 23 CAD BTDC for a medium load. Regarding the effect of DI ratio, it can be noticed that the peak power was produced at around 30% ethanol DI. This was attributed to the earlier combustion phasing and higher peak pressure (as shown in Figure 4.4) which indicates faster flame propagation speed (as shown in Figure 4.3). This is also consistent with the ISNO emission trend as shown in Figure 4.5.
4.1.1 Engine performance and combustion characteristics

Experimental results of DualEI strategy are compared with those of GPI only testing conditions to examine the effect of DualEI strategy on the SI engine performance. The IMEP results of the DualEI strategy and GPI only are shown in Figure 4.2. It can be seen that the IMEP increment continues over the full range of the DI percentages, starting from 0% (EPI only) to the 100% EDI. Compared to GPI only, the IMEP increases by 3.48% at 46% DI ratio in the light load condition. At medium load, the IMEP increases by 4.35% at 66% DI ratio. This improvement might be attributed to the enhanced mixture quality due to using DualEI strategy combined with some ethanol fuel properties [12]. However, at both engine loads, the effect of EDI percentages on the IMEP is essentially negligible at the full set of the DualEI engine testing conditions. The independent behavior of IMEP from DI ratio can be attributed to the fixed quantity and energy of the total ethanol fuel injected per cycle throughout the test.
To further understand the mechanism behind the increase of IMEP, the combustion characteristics of corresponding results will be presented and discussed. Figure 4.3 shows the major combustion duration (CA10-90%) of DualEI strategy compared with that in GPI only. CA10-90% is the main combustion duration defined by the time from 10% of the fuel burnt to 90% burnt [69]. At light engine load, Figure 4.3 shows that CA10-90% decreases directly after the EPI is started, and the CA10-90% continues to decrease with the increased percentage of ethanol directly injected reaching the minimum value at 32% of DI ratio. In medium load condition, CA10-90% decreases in the range of DI percentage from 32% to 66%, and then it increases when DI ratio further increases to 100%. The shortest CA10-90% occurs at DI32% in light load conditions, and at DI66% in medium load conditions. These results are highly consistent with the peak in-cylinder pressure tendency, as shown in Figure 4.4. This performance might be attributed to two main reasons. Firstly, in addition to the oxygen content, the greater the flame speed of ethanol fuel compared to that of gasoline may lead to combustion quality improvement. Secondly, injecting the right portion of ethanol fuel directly into the combustion chamber probably...
enhances the homogeneity of the air-fuel mixture and thus reduces the combustion duration. However, the CA10-90% increases with the increase of ethanol DI ratio when DI ratio exceeds 66% in light load and 80% in medium load. This can be attributed to two main reasons. Firstly, at high DI percentages, fuel impingement might become significant and lead to wall wetting, which could adversely affect the mixture quality. Secondly, the over-cooling effect of DI associated with ethanol fuel may reduce the combustion temperature to be too low and thus reduce the combustion speed leading to a longer combustion duration. This is clearly reflected in the peak pressure trend, which started to decrease when the DI was greater than 66% and 88% in the light and medium load respectively.

Figure 4.3 Variation of the Major Combustion Duration with the EDI Percentage.
4.1.2 Emissions characteristics

Figure 4.5 shows the variation of the indicated specific nitric oxide (ISNO) at the light and medium engine loads with DualEI strategy. The ISNO emission is strongly related to the combustion temperature [2, 11, 93]. When the DualEI strategy is used, the ISNO significantly lessens with the increased percentage of DI. As shown in Figure 4.5, at medium load, ISNO significantly reduces once the DualEI strategy started at DI32%, compared with that of GPI only. This may be due to the cooling effect of ethanol fuel that may be enhanced by DI strategy [54]. Compared with GPI only, 55.1% and 58.46% are ISNO reduction percentages in the light and medium engine loads respectively when the engine was operated at 100% EDI. Concerning the ethanol fuel properties, the high ethanol latent heat of vaporization can significantly contribute to the ISNO reduction. This possibly explains the ISNO emission reduction when the EPI only is used at light engine load. Furthermore, the DualEI strategy may play an important role in fully utilizing the ethanol fuel properties such as the latent heat of vaporization compared to EPI only, resulting in a further decrease in the combustion temperature [6, 12, 54].
The effect of DualEI strategy on the indicated specific hydrocarbon emission (ISHC) is shown in Figure 4.6. The ISHC increases when the DualEI strategy is started at DI percentage of 32% for both engine loads. The ISHC continues to increase along with the percentage of the ethanol fuel directly injected, reaching its maximum value at DI100% of ethanol. The incomplete combustion could be the main reason for this unfavorable tendency of the ISHC which might be caused by two main reasons. Firstly, a poor mixture quality which could be caused by fuel films formed on the wall due to fuel impingement, resulting in the nonhomogeneous distribution (local rich mixture regions) of ethanol fuel which is directly injected into the combustion chamber [54]. Secondly, the low combustion temperature and the consequent incomplete combustion could be caused by not only the ethanol fuel’s vaporization but also the cooling effect enhanced by the direct injection[5, 94].

![Figure 4.5 Variation of the ISNO with the EDI Percentage.](image)

Figure 4.5 Variation of the ISNO with the EDI Percentage.
Figure 4.6 Variation of the ISHC with the EDI Percentage.

Figure 4.7 shows the effect of DualEI strategy on the indicated specific carbon monoxide (ISCO) emission at the two defined engine loads. The lack of oxygen and incomplete combustion are the two main reasons behind the incomplete combustion, resulting in ISCO formation [69]. As shown in Figure 4.7, the ISCO emission increases with the increased ratio of ethanol fuel directly injected, which is highly consistent with the ISHC emission results as shown in Figure 4.6. At light engine load, the ISCO is significantly reduced in the range of EDI of 0% to 46%, compared with the ISCO in the GPI only. This tendency may be attributed to the ethanol oxygen content and flame speed combined with mixture quality improved by direct injection of a relatively small amount of ethanol in the combustion chamber [31]. However, when the DI percentage goes over 46% at light load and 32% at medium load, the over-cooling effect might reduce the combustion temperature to be too low, resulting in incomplete combustion and thus a greater quantity of the ISCO emission [69]. This can also be linked with the ISNO results which are decreased with the increased ratio of ethanol fuel directly injected as shown in Figure 4.5. When the DI ratio increases, the fuel impingement may increase and result in less even
air-fuel mixture distribution and higher cooling effect decelerating the oxidation process of the fuel used per cycle. This low temperature explains the increase of the HC and CO and decrease of NO emissions[95]. Furthermore, the poor mixture quality (local rich mixture regions) due to the high-level fuel impingement of ethanol fuel onto the combustion chamber walls and the lower vapor pressure at high EDI percentages for both engine loads might adversely affect the combustion quality, which leads to the increase of the ISCO and the ISHC as well.

![Figure 4.7 Variation of the ISCO with the EDI Percentage.](image)

4.2 Effect of spark timing on DualEI engine performance

In the experiment, the engine was started and warmed up using GPI only until the cylinder head temperature became stable at around 200 °C. After the engine temperature became stable, the GPI was switched to EPI, and the mixture was kept at the stoichiometric condition \(\lambda=1\). The volumetric ratio of the ethanol DI to EPI was changed from DI0\% (PI only) to DI100\% (direct injection only). At each DI ratio, the spark timing was changed from 15 to 42 CAD bTDC at light engine load, and from 15 to 32 CAD bTDC
at medium load. Table 4.2 lists the engine operating conditions for the experiments. The spark timing of 15 CAD bTDC (ST15) was set as a baseline because it was the original spark timing before the engine was modified. The DI timing was set at 300 CAD bTDC and port injection at 410 CAD bTDC, aimed to give sufficient time for fuel evaporation and mixing processes, based on previous experimental investigation [96, 97].

Table 4.2 Experimental operating conditions for subsection 4.3.

<table>
<thead>
<tr>
<th>Engine Loads</th>
<th>Light load (20% Throttle opening)</th>
<th>Medium load (33% Throttle opening)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed (RPM)</td>
<td></td>
<td>3500</td>
</tr>
<tr>
<td>Spark Timing (CAD bTDC)</td>
<td>Light load: 15, 20, 25, 30, 32, 34, 36, 38, 40 and 42</td>
<td>Medium Load: 15, 18, 20, 22, 23, 24, 26, 28, 30 and 32</td>
</tr>
<tr>
<td>DI% (volumetric percentage)</td>
<td>Light load: DI0 (PI only), DI35, DI56, DI60, DI80 and DI100</td>
<td>Medium load: DI30, DI50, DI70, DI100</td>
</tr>
<tr>
<td>Port Fuel Injection Timing</td>
<td>410 CAD bTDC</td>
<td></td>
</tr>
<tr>
<td>Direct Fuel Injection Timing</td>
<td>300 CAD bTDC</td>
<td></td>
</tr>
</tbody>
</table>

The experimental results will be presented and discussed in the following sub-sections. The effect of spark timing on the IMEP and indicated thermal efficiency will be reported in the first subsection 4.2.1. The effect of spark timing on the combustion and emissions characteristics will be presented and discussed in subsections 4.2.2 and 4.2.3.

### 4.2.1 Engine performance

Figures 4.8 and 4.9 show the effect of spark timing (ST) on the IMEP at different DI ratios and engine loads. At light load and all the DI ratios, IMEP significantly increases when the ST is advanced to up to ST30. It then decreases with further advance of spark
timing. As shown in Figure 4.8, at DI100%, the IMEP increases from 0.40 MPa to 0.46 MPa when the ST is advanced from ST15 to ST30 at light load. Likewise, at DI100% and medium load condition, the IMEP increases from 0.63 MPa to 0.672 MPa when the ST is advanced from ST15 to ST23, as shown in Figure 4.9. Results shown in Figure 4.8 and Figure 4.9 are consistent with those of any internal combustion engine. The maximum IMEP was achieved when the advanced ST lead to increased in-cylinder pressure in the compression stroke, and consequently, the bulk of combustion was completed near the TDC and thus more net-work was produced. This is made possible by advancing the spark timing to the point at which the CA50 occurs at certain degrees aTDC [31]. Here, CA50 is defined as the combustion phase (CAD) when 50% of the fuel is burnt. Based on the results in Figure 4.8 and Figure 4.9, ST30 is identified to be the MBT spark timing at the light load, and ST23 the MBT spark timing at the medium load. It should be noticed that, as shown in Figure 4.8 and Figure 4.9, both advancing spark timing and DualEI strategy play a significant role in the increase of the IMEP. In general, the IMEP increased with the increased percentage of DI, which may be attributed to two main reasons. Firstly, the charge cooling effect due to ethanol having a high latent heat of vaporisation associated with DI injection strategy might increase the fresh charge density hence allowing more air to be charged during the intake stroke [40]. Secondly, the oxygen content of ethanol fuel and the high combustion speed might help improve the combustion quality.
Figure 4.8 Effect of spark timing on IMEP at different DI ratios (a-light load).

Figure 4.9 Effect of spark timing on IMEP at different DI ratios (b-medium load).

Figure 4.10 and Figure 4.11 show that the indicated thermal efficiency increases with the advanced spark timing until ST30 at light load and ST23 at medium load. At light load and DI80%, the maximum increase of the indicated thermal efficiency is 10.68%, when the spark timing is advanced from ST15 to ST30, as shown in Figure 4.10. Similarly, at medium load and DI70%, the maximum improvement of the indicated thermal efficiency
is 5.42%, when the spark timing is advanced from ST15 to ST23, as shown in Figure 4.11. Further advancing the spark timing results in a reduction of thermal efficiency at both engine loads. The results shown in Figure 4.10 and Figure 4.11 may be explained by three main causes. Firstly, the position of the start of combustion (spark timing) plays an important role in the engine performance [69, 71]. A reduced combustion chamber volume at CA50 leads to increased combustion pressure and temperature. The increased cylinder temperature may enhance the evaporation rate of ethanol fuel and thus improve the mixture and combustion quality. Secondly, further advancing the spark timing from the MBT timing decreases the thermal efficiency due to the early combustion. Moreover, the greater combustion temperature may promote the convective heat losses, which can partly explain the thermal efficiency reduction [3]. On the other hand, retarding the spark timing from MBT timing reduces the combustion pressure and temperature [52]. Consequently, the over-cooling effect caused by DI strategy may deteriorate the combustion performance, which can partly explain the reduction of engine thermal efficiency. Thirdly, early combustion means a greater area of the cylinder wall that can lead to an increase in the heat loss through the combustion chamber walls. Furthermore, the air-fuel mixture has relatively less time to be homogeneously mixed before the combustion starts. This may result in a poor mixture and combustion quality.
4.2.2 Combustion and emissions characteristics

Combustion characteristics such as CA50, maximum in-cylinder pressure (denoted as $P_{\text{max}}$) and its phase (denoted as $\theta_{P_{\text{max}}}$), the major combustion duration (CA10-90%) and
the heat release rate (HRR) will be examined, aiming to understand the mechanism behind the experimental results.

Figure 4.12 and Figure 4.13 show the effect of the spark timing on \( P_{\text{max}} \) and \( \theta_{P_{\text{max}}} \) at a selected DI ratio (DI56% in light load, DI50% in medium load). It is well known that the engine power can be maximised when the \( P_{\text{max}} \) occurs within few degrees aTDC [69]. At DI56% and light load as shown in Figure 4.12, the \( \theta_{P_{\text{max}}} \) moves by 12 degrees toward the TDC, and the \( P_{\text{max}} \) increases by about 1.7 MPa when the ST is advanced from ST15 to ST42. As shown in Figure 4.13, at DI50% and medium load, the \( \theta_{P_{\text{max}}} \) moves by about 13.5 degrees toward TDC, and the \( P_{\text{max}} \) rises by about 1.87 MPa with spark timing advanced from ST15 to ST23. These results are consistent with the IMEP and indicated thermal efficiency results, as shown in Figures 4.8 - 4.11. The maximum IMEP and indicated thermal efficiency are recorded at ST30 while the \( \theta_{P_{\text{max}}} \) is located at about 10 CAD aTDC at the light load. At medium load, the maximum IMEP and indicated thermal efficiency are noticed at ST23 while the \( \theta_{P_{\text{max}}} \) is located at around 12 CAD aTDC.

Figure 4.12 Effect of spark timing on the maximum pressure and its phase with DI56% at light load.
Figure 4.13 Effect of spark timing on the maximum pressure and its phase with DI50% at medium load.

Figure 4.14 shows the effect of the spark timing by comparing the in-cylinder pressure, HRR and MBF at ST15 and ST30 at light load and DI56%. All the results were calculated according to the recorded in-cylinder pressure experimentally. As shown in Figure 4.14, the $P_{\text{max}}$ rises significantly and the $\theta_{P_{\text{max}}}$ advances toward TDC by about 5.0 CAD when the ST is advanced from ST15 to ST30. Correspondingly, the combustion performance, represented by the HRR and the MBF, enhances considerably. At ST30, the peak HRR increases by about 4.0%, and the major combustion duration (CA10-90%) is 41.8% less when the ST is advanced from ST15 to ST30, as shown in Figure 4.14 and Figure 4.15. Although the CA10-90% at ST32 is slightly shorter than that at ST30, the IMEP is maximum at ST30 which is the MBT timing [69]. The improved combustion performance can be attributed to the enhanced mixture quality and the reduced heat loss when the ST is advanced. Moreover, the relatively smaller combustion chamber volume where the CA50 occurs can result in higher combustion pressure and temperature [98]. This may improve the combustion quality, and increase the heat released rate, and then reduce the combustion duration. However, greater combustion pressure and temperature may
increase the heat loss from the combustion chamber walls caused by the high-temperature difference between the inside and outside of the cylinder wall [87]. These results partly explain why the IMEP and indicated thermal efficiency decrease with further advance of spark timing from the MBT timing. The greater heat losses can be associated with negative work caused by the extreme advance to the spark timing.

![Graph showing cylinder pressure, MBF, and HRR at ST30, DI56% and light load.](image)

Figure 4.14 Cylinder pressure, MBF, and HRR at ST30, DI56% and light load.
Figure 4.15 Variation of the major combustion duration (CA10-90%) with DI ratio at light load.

Figure 4.16 shows the effect of spark timing on CA50 at different DI ratios in light engine load. As shown in Figure 4.16, the CA50 consistently advances toward TDC due to the combined effect of advancing spark timing and DualEI strategy. For instance, at DI56%, the CA50 advances by 10 CAD when the ST is advanced from ST20 to ST30. On the other hand, the DualEI strategy also affects the phase of the combustion. Figure 4.16 also shows that CA50 advances toward TDC with the increased DI ratio at the same spark timing. At ST25, the CA50 moves toward TDC by 3.0 CAD at DI35% compared with DI0%. However, further increase of DI ratio from DI35% leads to CA50 retarding from the TDC caused by the combustion quality deterioration.
Figure 4.16 Effect of spark timing on CA50 at different DI ratios and light load.

Figure 4.17 and Figure 4.18 show the effect of spark timing on the ISCO and ISHC emissions at different DI ratios and light load. The ISCO and ISHC emissions increase with advanced ST and increased DI ratio. This emissions performance may be because of incomplete combustion, which can be caused by the insufficient time for the ethanol fuel to evaporate and mix homogeneously with air when the spark timing is advanced. Fuel impingement due to direct injection can be another cause and would result in an overcooling and over-rich local regions inside the combustion chamber [99]. However, this is not a problem when ethanol fuel is port injected. As shown in Figure 4.17 and Figure 4.18, the ISCO and ISHC are significantly reduced when the DI ratio is reduced from DI35% to DI0%. When the DI ratio is reduced, more fuel can be well evaporated in the port before it is mixed with air. This should reduce the fuel impingement and improve the mixture quality and thus the combustion performance [54]. Figure 4.17 shows that the ISCO emission increases with the advanced spark timing at different DI ratios. However, the effect of spark timing on ISCO is not obvious at DI0% but becomes more and more
significant when the DI ratio increases. At DI56%, the ISCO emission increases by up to 71.84% when the spark timing is advanced from ST15 to ST42. This apparently results in incomplete combustion caused by the reduced available time for fuel evaporation, mixture formation and then combustion when the spark timing is advanced. Regarding the effect of DI ratio, in general, the ISCO increases with the increased DI ratio. As shown in Figure 4.17, at ST30, the ISCO increases by up to 302.84% when the ratio of the DI increases from DI0% to DI100%. The causes may include the over-cooling effect and the high fuel impingement enhanced by the increased DI ratio.

![Figure 4.17 Effect of spark timing on ISCO at different DI ratios.](image)

Figure 4.18 shows that the ISHC emission sharply increases by 109.64% at DI35% and by 134% at DI100% when the spark timing is advanced from ST15 to ST42. As reported in [4], the high DI ratio and the slow vaporisation of the ethanol fuel at low temperature can result in an ethanol film being formed on the combustion chamber walls which resulted in over-cooling effect when the fuel is evaporated. The combination of the over-cooling effect and the high fuel impingement can deteriorate the mixture quality and thus the combustion performance in the range of DI ratio of DI56%-DI100%. Besides, the
advance of spark timing raises the peak in-cylinder pressure as shown in Figures 4.13 and 4.14, which leads to more unburned hydrocarbons being trapped in the crevice volumes and a corresponding increase in HC emissions. Consequently, the ISHC increases consistently with the advance of spark timing.

Figure 4.18 Effect of spark timing on ISHC Variation at different DI ratios.

Figure 4.19 shows the effect of the spark timing on the ISNO emission at different DI ratios. Fundamentally, the ISNO is strongly related to the combustion temperature and mixture quality [2, 69]. As shown in Figure 4.19, the ISNO increases with advanced spark timing from ST15 to ST42 possibly due to the increase of the peak in-cylinder pressure (Figures 4.13 and 4.14) and temperature when the spark timing is advanced. These results are consistent with those presented in Figure 4.14, the value of the combustion pressure and HRR increase with advanced spark timing leading to high combustion temperature and thus increased ISNO. On the other hand, the ISNO decreases with the increase of DI ratio. As shown in Figure 4.19, at ST30, the ISNO emission significantly decreases by 37.53% when DI ratio increases from DI0% to DI56% and 67.39% when DI ratio increases from DI0% to DI100%. This is apparently attributed to the DI strategy and
ethanol high latent heat, which reduce the combustion temperature \[6\], and consequently decrease the ISNO emission.

4.3 Integrated effect of DI and spark timings on DualEI engine performance

In the experiments reported in this section, the volumetric ratio (DI\%) of the ethanol directly injected to ethanol port injected was kept at 50\% at light load and 56\% at medium load because they were the optimal DI ratios for achieving effective cooling with minimal fuel impingement \[54, 66\]. The fuel heating value was fixed at around 420 J/cycle at light load and 610 J/cycle at medium load. The spark timing was swept from 19 to 25 CAD bTDC (denoted by ST’XX’ hereafter) at light load and from ST28 to ST34 at medium load, as they were the spark timings to produce the maximum output power \[66\]. At each spark timing, DI timing was varied from 330 CAD bTDC to 240 CAD bTDC (denoted by DIT’XX’ hereafter) before the intake valve was closed (early DI timing), and from DIT120 to DIT60 after the intake valve closed (late DI timing). The range of the late DI timing was chosen to eliminate the effect of the hot residuals, hot combustion chamber

![Figure 4.19 ISNO Variation with spark timing at different DI ratios.](image-url)
walls and intake flow velocity and then to address the effect of DI timing more independently. The range of early DI timing was chosen to include the impact of the previously mentioned factors and the time on engine performance. As shown in Figure 4.20, the early injection was swept with 30 CAD increment, and the late injection was swept with 20 CAD increment. The DI timing between DIT240 and DIT120 was avoided due to the pressure fluctuation caused by flow when the intake valve was closing. The throttle position was set at 23\% for light engine load and 34\% for a medium load. The air/fuel equivalence ratio (\( \lambda \)) was kept at around the stoichiometric condition. Further details about the engine operating conditions are listed in Table 4.3.

Figure 4.20 Valve, Spark and Injection Timings set in the experiments.
Table 4.3 Shows Section 4.4 Experimental Operating Conditions.

<table>
<thead>
<tr>
<th>Engine loads</th>
<th>Light load (23% throttle opening)</th>
<th>Medium load (34% throttle opening)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed (RPM)</td>
<td>3500</td>
<td></td>
</tr>
<tr>
<td>Spark timing (CAD bTDC)</td>
<td>Light load: 28, 30, 32 and 34 Medium Load: 20, 24, 26 and 28</td>
<td></td>
</tr>
<tr>
<td>DI ratio</td>
<td>Light load: 50% Medium load: 56%</td>
<td></td>
</tr>
<tr>
<td>PI timing</td>
<td>410 CAD bTDC</td>
<td></td>
</tr>
<tr>
<td>DI timing (CAD bTDC)</td>
<td>Early DI timing: 240, 270, 300 and 330 Late DI timing: 60, 80, 100 and 120</td>
<td></td>
</tr>
</tbody>
</table>

The experimental results will be presented and discussed as follows. Section 4.3.1 reports the effect of early DI timing on the IMEP and indicated the thermal efficiency of the DualEI engine. The combustion characteristics will be analysed and discussed to understand the mechanism behind the change in the engine performance. In section 4.3.2, the effect of early DI timing on emissions will be discussed and analysed. Sections 4.3.3 and 4.3.4 will discuss the effect of late DI timing on engine performance, combustion and emissions characteristics.

### 4.3.1 Effect of Early DI timing on Engine Performance

The effect of early DI timing on IMEP and thermal efficiency of DualEI engine at different spark timing is shown in Figure 4.21. The coefficient of variation of IMEP (COV\textsubscript{IMEP}) at various DI and spark timings are shown in Figure 4.22. COV\textsubscript{IMEP} is derived from in-cylinder pressure and measures the combustion stability (cyclic variability) [69]. As shown in Figure 4.22, the COV\textsubscript{IMEP} is almost independent of DI and spark timings’ effects at both engine loads. At light load, COV\textsubscript{IMEP} remains below its maximum of 4.46% at DIT300 and ST34. At medium load, the COV\textsubscript{IMEP} approaches its minimum value of
Chapter Four Dual ethanol injection (DualEI) engine performance

1.99% at DIT330 and ST25. The results of COV_{IMEP} shown in Figure 4.22 exhibit acceptable combustion stability (less than 5%) in the full range of DI and spark timings.

As shown in Figure 4.21, the IMEP and thermal efficiency slightly decrease when DI timing is advanced from DIT240 to DIT300 at different spark timings, except for ST19 and ST28 at medium and light loads respectively. Further advancing of DI timing from DIT300 to DIT330 slightly increases IMEP and thermal efficiency of DualEI engine. The reduction in IMEP and thermal efficiency might be attributed to rich air/fuel regions formed due to the impingement of ethanol fuel injected onto the combustion chamber walls [54] when the piston is moving upward to reduce cylinder volume, as shown in Figure 4.20. The increment of IMEP and thermal efficiency could be due to the longer time available at DIT330 compared with DIT300. This could potentially improve the mixing process of air with ethanol fuel before spark timing. Moreover, hot residuals might heat ethanol fuel that is directly injected at a few degrees after the exhaust valve was closed. This could increase the evaporation rate of ethanol inside the cylinder and thus improve the quality of mixture and combustion. At ST19 and ST28, IMEP and thermal efficiency increase with advanced DI timing from DIT240 to DIT330. Figure 4.21 shows the greatest IMEP and efficiency occur at the latest spark timing, ST19 for medium load and ST28 for a light load. The maximum increment of the IMEP occurs 1.67% at medium load and ST19, and 1.51% at light load and ST28 when the DI timing is within the range from DIT240 to DIT330.
Figure 4.21 Variation of IMEP (a) and indicated thermal efficiency (b) with an early DI and spark timings.
To understand the combustion characteristics in the DualEI engine, the results of the initial (CA0-10%) and major (CA10-90%) combustion durations at different DI and spark timings conditions are presented and analysed. CA0-10% is the initial combustion duration defined by the crank angle degrees starting from the spark timing to the time with 10% heat release. Early flame development and propagation have a strong effect on combustion stability [69]. As shown in Figure 4.23 (a), at both engine loads, CA0-10% significantly decreases with advanced early DI timing from DIT240 to DIT300, and then it slightly increases when DI timing is further advanced to DIT330. Spark timing shows a weak effect on CA0-10% at medium load condition. However, in the light load, CA0-10% significantly decreases with retarded spark timing from ST34 to ST28. The CA0-10% reduction could partially explain the improved combustion stability which is shown by the COV_{IMEP} in the light load condition in Figure 4.22. As shown in Figure 4.22, the COV_{IMEP} is in the range of 2%-5% when the early DI timing is in the range of 240-330 CAD bTDC. It decreases from 3.5% to about 2% when the spark timing is retarded from...
34 to 28 CAD bTDC at DIT330. CA10-90\% is the main combustion duration defined by the time from 10\% of the fuel burnt to 90\% burnt. Figure 4.23 (b) does not show a close link between the CA10-90\% and the IMEP and the indicated thermal efficiency. However, CA10-90\% significantly decreases when the DI timing is advanced from DIT240 to DIT330 in the light load condition. Figure 4.24 shows the combustion phase (CA50) related to the results shown in Figure 4.21. CA50 is defined as the crank angle degree when 50\% of the fuel is burnt. As shown in Figure 4.24, CA50 reduces when DI timing is advanced from DIT240 to DIT300 in the full set of spark timing, and at the two engine loads. Then, CA50 slightly increases when DI timing is advanced further, from DIT300 to DIT330. The advanced CA50 (near or even before the engine top dead centre (TDC)) means a negative indicated power could be generated. Therefore, IMEP and indicated thermal efficiency slightly increase as shown in Figure 4.21 when CA50 decreases with advanced DI timing from 300 to 330 CAD bTDC, as shown in Figure 4.24.
Figure 4.23 CA0-10% (a) and CA10-90% (b) vs. spark timing at DIT330.
Figure 4.24 Variation of CA50 with an early DI and spark timings.

4.3.2 Effect of Early DI Timing on Emissions Characteristics

Figure 4.25 (a) shows the effect of early DI timing on ISCO at different spark timing. The lack of oxygen in combustion and rich fuel mixture introduces a larger concentration of CO emission [43, 100]. At both engine loads, the spark timing does not show a clear effect on the ISCO emission when DI timing is in the range from 240 to 330 CAD bTDC. As shown in Figure 4.25 (a), the spark timing does not show a clear effect on ISCO emission when DI timing is in the range from 240 to 330 CAD bTDC. On the other hand, as shown in Figure 4.25 (a), at both engine loads, the ISCO increases when the DI timing is advanced from DIT240 to DIT300. However, the ISCO significantly reduces when the early DI timing is further advanced to DIT330. The increased CO emission could be attributed to the rich mixture regions caused by fuel impingement due to smaller combustion chamber volume with advanced DI timing, as shown in Figure 4.20. Severe fuel impingement could form a thick film, and slow down the fuel vaporization [101].
The decreased ISCO might be because the hot residuals heated the directly injected fuel since about half of fuel was injected at 10.7 degrees after the exhaust valve closed. This could accelerate the evaporation rate of ethanol directly injected into the cylinder [101-103]. Compared to DIT240, ISCO is increased by 85.9% at DI timings of DIT300 and 2.3% at DIT330, at ST23 and medium load. At light load condition and ST32, ISCO increases by 41% when DI timing is advanced from DIT240 to DIT300, but it decreases by about 43% with advanced DI timing to 330 CAD bTDC, as shown in Figure 4.25 (a). The ISCO results could partly explain the results of the indicated thermal efficiency as shown in Figure 4.21 (b).

Figure 4.25 (b) shows the effect of early DI timing on HC emission at different spark timing. HC emission contains unburned fuel from incomplete combustion due to local flame extinction that occurs when temperatures are lower and reaction times may become larger than mixing times [43, 100]. As shown in Figure 4.25 (b), at light load, ISHC increases when the early DI timing is advanced from DIT240 to DIT300, and then slightly decreases when DI timing is further advanced to DIT330. In medium condition, ISHC increases with advanced DI timing from 240 to 330 CAD bTDC. Spark timing exhibits a clear effect on ISHC emission that increases with advanced spark timing from ST28 to ST34 in light load, and from ST19 to ST25 in medium load. The effect of DI and spark timing is stronger in medium load than that in low load condition. Since the engine speed is fixed, the available time for the mixing process is constant for both engine load conditions. Nevertheless, the mass of fuel injected per cycle is about 45% greater (from 420 J/cycle for the light load to 610 J/cycle for the medium load) when the load increases from light to medium. This probably causes insufficient time available for forming a homogenous mixture at medium load compared to that in light load resulting in greater HC emission at the medium load condition.
Figure 4.25 Variation of ISCO (a) and ISHC (b) with an early DI and spark timings.
NO emission is formed from the N\textsubscript{2} and O\textsubscript{2} dissociation at the high combustion temperature. Figure 4.26 shows that the ISNO increases with advanced spark and DI timings in the light and medium load conditions. As shown in Figure 4.26, at ST21, the ISNO increases by 28\% when DI timing is advanced from 240 to 330 CAD bTDC in medium load. At light load and ST30, it increases by 39\% when DI timing is advanced. At medium load and DIT330, the ISNO increase by about 64\% when the spark timing is advanced from ST19 to ST25. Likewise, at light load condition but with less significance, the ISNO increases when the spark timing is advanced 28 to 34 CAD bTDC.

4.3.3 Effect of Late DI timing on Engine Performance

Figure 4.27 shows the effect of late DI timing on the IMEP and indicates the thermal efficiency of the DualEI engine at different spark timing. As shown in Figure 4.27, the IMEP and indicated thermal efficiency are almost independent of the spark timing after the intake valve is closed. This may be because the mixture quality and condition are
similar before the spark timing at each DI timing. As shown in Figure 4.27, the effect of DI timing on IMEP and indicated thermal efficiency is not strong in the light engine load condition. At medium load, the effect of DI timing is not so obvious until the DI timing is later than 80 CAD bTDC. The IMEP slowly decreases from 0.62 MPa to 0.60 MPa and indicated thermal efficiency from 26.6% to 25.9% when the DI timing is retarded from 120 to 80 CAD bTDC at ST19. However, the IMEP decreases from 0.60 MPa to about 0.50 MPa and the indicated thermal efficiency from 25.9% to about 21% when the DI timing is retarded from 80 CAD bTDC to 60 CAD bTDC in the medium load condition. As the engine speed is fixed at 3500 rpm, the time for mixture formation is equal in light and medium load conditions. In light engine conditions, 50% of the fuel is injected directly into the cylinder. The time for forming the mixture of fuel and air looks sufficient, as the IMEP and thermal efficiency are quite stable in the full range of DI timing, as shown in Figure 4.27. However, when the engine load is increased from light to medium condition, the ratio (56%) of ethanol fuel directly injected into the cylinder and the fuel flow rate are increased. When the DI timing is retarded from 80 to 60 CAD bTDC, the time for the fuel to evaporate and to mix with the fuel port injected and air may become insufficient at medium load, resulting in a significant decrease in IMEP and indicated thermal efficiency. This shows that the DI and spark timings interactively work together when the DualEI strategy is used. As shown in Figure 4.20, late DI timing is located just before the timing of the spark ignition and closing of the intake valve, which might adversely affect the mixture quality at the time of ignition.
Figure 4.27 Variation of IMEP (a) and indicated thermal efficiency (b) with the late DI and spark timings.
To further understand the effect of late DI timing on the engine performance, the combustion characteristics will be analysed and discussed. Figure 4.29 shows the CA0-10% and CA10-90% corresponding to the results shown in Figure 4.27. Early flame development and propagation have a strong effect on combustion stability [69]. As shown in Figure 4.29 (a), the effect of late DI timing on CA0-10% is not significant in the light load condition. However, in medium load condition, the CA0-10% is reduced with the DI timing advanced from 60 to 100 CAD bTDC and then is slightly increased with DI timing further advanced to 120 CAD bTDC. It reduces more quickly when the DI timing is advanced from 60 to 80 CAD bTDC. The reduced CA0-10% means increased flame speed and improved stability. This explains the strong effect of DI timing on the IMEP and indicated thermal efficiency when the DI timing is advanced from 60 to 80 CAD bTDC and then the weak effect of DI timing with further advance of DI timing, as shown in Figure 4.27. The improved combustion stability associated with reduced CA0-10% is also shown by the COV$_{\text{IMEP}}$ in medium load condition in Figure 4.28. As shown in Figure
4.28, the COV\textsubscript{IMEP} is in the range of 5\%-20\% when the DI timing is 60 CAD bTDC. It is decreased from 17.7\% to 8.2\% when the DI timing is advanced from 60 to 80 CAD bTDC at ST21. Figure 4.29 (b) does not show a close link between the CA10-90\% and the IMEP and the indicated thermal efficiency, but there is a significant increase of CA10-90\% when the DI timing is advanced from 80 to 120 CAD bTDC in the light load condition.
4.3.4 Effect of Late DI Timing on Emissions Characteristics

The variation of the ISNO, ISCO, and ISHC with the DI and spark timings are presented in Figure 4.30. Fundamentally, the elevated temperature of combustion promotes NO formation. As shown in Figure 4.30 (a), the ISNO decreases noticeably when the late DI timing is retarded from 120 to 60 CAD bTDC. It is also decreased when spark timing is retarded from ST25 to ST19 in the medium load and from ST34 to ST28 in the light load condition. The cooling effect of DI strategy could play a significant role in the reduction of NO emission in the late DI timing condition. Advanced DI timing could reduce the mixture temperature at an earlier time, but this also increases the convective heat transfer from combustion chamber walls to the mixture. This could enhance the mixture quality and improve the engine performance, as shown in Figure 4.27 and Figure 4.29. When the DI timing is retarded, the cooling effect due to fuel evaporation associated with DI strategy could be well preserved. This could decrease the combustion temperature and
then the NO emission, as shown in Figure 4.30 (a). Figure 4.30 (a) shows that the ISNO decreases with retarded spark timing in the light and medium load conditions. This could be attributed to the low gas temperature due to the phase change of combustion when spark timing is retarded [66].

The effect of DI and spark timings on ISCO and ISHC are shown in Figure 4.30 (b) and (c). It can be seen that at all the tested spark timings, ISCO increases when the DI timing is advanced from DIT60 to DIT120. Although the DIT120 provides more time for fuel evaporating compared to DIT60, the in-cylinder pressure and then the temperature is lower than that of DIT60, as shown in Figure 4.20. Greater compression temperature may promote a faster evaporation rate of ethanol. Furthermore, the air/fuel equivalence ratio (λ) is slightly decreased when the DI timing is advanced from DIT60 to DIT120, as shown in Figure 4.31. Consequently, less amount of oxygen is available to the fixed amount of injected fuel per cycle, which results in greater ISCO and ISHC emission. The ISCO and ISHC results do not show a clear link with IMEP or the results of the combustion characteristics, which are shown in Figure 4.27 and Figure 4.29.
Chapter Four                                     Dual ethanol injection (DualEI) engine performance

(a)

(b)
Figure 4.30 Variation of ISNO (a), ISCO (b) and ISHC (c) with the late DI and spark timings.

Figure 4.31 Variation of Equivalence Air/Fuel Ratio ($\lambda$) with DI and spark timings in light load condition.
4.4 Possible mechanisms behind engine performance changes

According to the aforementioned results for the IMEP and emissions at different engine testing conditions, it can be seen that significant changes occurred in the engine performance when the testing conditions were changed, as shown in sections 4.1, 4.2 and 4.3. It was reported that the mixture quality (air-fuel mixing process) could strongly govern the SI engine performance, including the combustion efficiency and the consequent emissions [4, 104]. The homogeneity of the air-fuel mixture could be potentially controlled by several parameters, including the fuel injection strategy (DI plus PI), conditions inside the cylinder particularly at the point of injection and ignition, and the spray characteristics of the injected fuel, especially that of the direct fuel injector. The effects of these factors on engine performance will be presented and discussed in this section.

Dual fuel injection strategy

Dual fuel injection strategy (DI plus PI) has the potential to improve the engine performance. This concept was originally transferred from CI engines to be used in SI engines. Shuai et al. [105] emphasised that Ikoma et al. [106] initially proposed the dual fuel injection concept to be utilised in gasoline engines. The main aim of using such technology was to increase the degree of freedom of running the gasoline engines under the MBT condition by extending the engine knock limits. Recently, ethanol has become more and more popular as an alternative or octane-enhancer for gasoline fuel in SI engines. To utilise ethanol fuel benefits more flexibly and efficiently, ethanol DI strategy for a conventional gasoline PI engine was proposed by [5]. The mixture formation of dual fuel injection strategy was numerically investigated [4]. Their results showed that the dual
fuel injection strategy could potentially improve thermal efficiency, and decrease the NO emissions. The partially premixed combustion could potentially improve the mixture quality, and provide full utilisation of ethanol fuel properties. The slow evaporation process of ethanol could be overtaken when part of the used fuel is early injected in the intake port. In this research, the timing of the port fuel injection is about 110 CAD earlier than that of the DI timing. Moreover, the port injected fuel could fully benefit the intake flow rate in addition to the longer time available for the fuel to be completely mixed with air. This can be demonstrated by the combustion quality enhancement when the DI ratio is reduced between 30% and 55% at both engine loads, as shown in Figure 4.3 and Figure 4.4. The fuel film formation and the over-cooling effect inside the combustion chamber could be partially avoided when the dual injection strategy is used, as shown in Figure 4.6 and Figure 4.7 in section 4.1.

**In-cylinder conditions (in-cylinder gases history)**

The results presented in this chapter showed that the combustion performance is strongly linked with the timing of direct fuel injection and spark onset. The heat losses through the combustion chamber walls could strongly affect the engine performance [52, 107]. The DI timing could manage how much heat (of hot residuals and combustion chamber walls) can be utilised by the fuel/air mixture before the recycled heat is lost to the surroundings through the cylinder wall. Advanced injection timing could strongly accelerate the evaporation rate of ethanol fuel directly injected as it is located a few degrees after the exhaust valve is closed. For instance, as shown in Figure 4.23 and Figure 4.24, the CA0-10% and CA10-90% significantly decreases with advanced DI timing from DIT240 to DIT330. Similarly for CA50 that advances toward the TDC when the DI timing is advanced, as shown in Figure 4.24. This combustion performance is clearly reflected in
the CO and NO emissions, as shown in Figure 4.25 and Figure 4.26, where the ISCO decreases, and ISNO increases with advanced DI timing. These results are highly correlated with temperatures of the cylinder head and exhaust gases that were recorded during the experiments, as shown in Figure 4.32.

Figure 4.32 shows the relationship between the DI timing and recorded temperature of the cylinder head and the exhaust gases at various spark timings and medium engine load. In general, it can be observed that the cylinder head temperature increases, and exhaust gas temperature decreases when DI timing is advanced from 240 to 330 CAD bTDC. More heat can be utilised by the air-fuel mixture (with advanced DI timing) which means a faster evaporation rate and better mixture quality. This could lead to improving the combustion quality and increasing the combustion temperature, resulting in increasing the engine body temperature represented by the cylinder head. On the other hand, the decrease in the exhaust gas temperature could be attributed to the shortened CA10-90% with advanced DI timing. This means that faster flame speed might complete the combustion earlier. This might give more time for the burnt gases to release more heat through the combustion chamber walls, resulting in greater engine body temperature and lower exhaust gases temperature.
4.5 Chapter Summary

1. When the total fuel heating energy was kept the same, the IMEP at the full set of DualEI testing was greater than that of GPI only. However, the DI ratio did not affect the IMEP results during the full range of DualEI testing conditions. In addition, at both engine loads and DualEI strategy testing, the CA10-90% was smaller than that of GPI only when the DI ratio of ethanol fuel was in the range of 32% to 66%. However, because of the over-cooling effect and mixture quality deterioration, the CA10-90% increased when the DI percentage was over 80%.

2. Compared with GPI only, ISNO emission was reduced with the increase DI ratio of ethanol fuel caused by the enhanced cooling effect. However, because of incomplete combustion and a non-homogeneous air-fuel mixture when the percentage of DI was greater than 32%, the ISHC increased with the increased ratio of DI. In addition, compared with GPI only, the ISCO was generally increased when the DualEI strategy
was used. The ISCO continued to increase with the increase ratio of DI, which was mainly attributed to the over-cooling effect and the poor mixture quality caused by the severe fuel impingement on the combustion chamber walls.

3. At both loads and DualEI testing conditions, the IMEP and indicated thermal efficiency increased with the advanced spark timing. This effect on the indicated thermal efficiency was stronger at the light load than that at the medium load. MBT was identified in terms of the best IMEP and thermal efficiency. The MBT spark timing was 30 CAD bTDC at light load and 23 CAD bTDC at medium load. At the MBT timing, the IMEP and engine thermal efficiency increased with the increased DI ratio.

4. At the MBT spark timing, $\Theta_{P_{\text{max}}}$ was located at about 10 CAD aTDC at light load and about 12 CAD aTDC at medium load. The CA10-90% reduced when the DI ratio was increased when the spark timing was advanced from ST15 to ST30 at light load. Similarly, the CA50 was advanced toward the TDC when the DI ratio was increased and the spark timing was advanced. However, further increase of the DI ratio from DI35% to DI100% increased the CA10-90% and retarded the CA50 from TDC due to the fuel impingement and over-cooling effect.

5. All the emissions were increased with advanced spark timing. However, the DualEI strategy had different effects on different emissions. The ISHC and ISCO were increased when the ratio of the DI was increased. This might be a result of incomplete combustion due to fuel impingement, mixture quality deterioration and over-cooling effect [94]. ISNO was reduced with the increased DI ratio because of the charge cooling effect enhanced by direct injection and consequently the reduced combustion temperature.
6. In early DI timing conditions (before the intake valve is closed), the effect of DI timing on IMEP and indicated thermal efficiency was not strong, as was the effect of the spark timing. At medium engine load, the IMEP and indicated thermal efficiency slightly decreased when the DI timing was advanced from 240 to 300 CAD bTDC with small differences caused by the spark timing. When the DI timing was further advanced from 300 to 330 CAD bTDC, the IMEP and indicated thermal efficiency increased slightly or keep constant. At light engine load, the IMEP and indicated thermal efficiency increased slightly with the advanced spark timing but independently with the DI timing. As the intake valve is still opened in early DI timing, the air and ethanol fuel port injected are mixing with the gas and fuel directly injected at low cylinder pressure. The mixing process and the quality of the mixture are not significantly affected by the DI timing.

7. In late DI timing conditions (after the intake valve is closed), IMEP and the indicated thermal efficiency increased with the advanced DI timing. However, the effect of DI timing was not very strong when the DI timing was earlier than 80 CAD bTDC. When the DI timing was retarded from 80 to 60 CAD bTDC, the IMEP was decreased from about 0.60 MPa to 0.50 MPa and indicated thermal efficiency from about 26% to 22%. This was mainly because the DI timing of 60 CAD bTDC is too late and leaves insufficient time for the ethanol fuel directly injected to evaporate and mix with the gases in the cylinder before it is ignited. This is evident with the longer initial combustion duration with DI timing of 60 CAD bTDC than that with more advanced DI timing. The specific emissions of NO and CO increased with the advanced DI timing as well as spark timing.

8. This DualEI engine performed better with early DI timing than with late DI timing, in terms of IMEP and indicated thermal efficiency. With early DI timing, the IMEP was
in a range of 0.60-0.68 MPa and indicated the thermal efficiency of 30-32% at medium load and 0.47-0.50 MPa and 29-31% at light load. With late DI timing, the IMEP was in a range of 0.47-0.62 MPa and indicated efficiency 21-27% at medium load and 0.41-0.44 MPa and 22-24% at light load. As identified, the DI timing should not be later than 80 CAD bTDC at medium load. The spark timing should not be later than 19 CAD bTDC at medium load and 28 CAD bTDC at light load.
Chapter Five

5 Ethanol port injection plus gasoline direct injection

5.1 Introduction

As reviewed in Chapter 1, gasoline direct injection (GDI) is an approach transferred from Diesel engines to improve the engine performance and reduce the pollutant emissions of SI engines. However, the challenges in GDI include reduction of particulate matter (PM) and hydrocarbon (HC) emissions due to the impingement of fuel directly injected and the non-homogenous mixture [57-59]. Dual fuel injection can help reduce the fuel impingement by injecting part of the fuel into the intake port or manifold. The overall objective of the present study is to make the use of ethanol fuel more effective and efficient. This chapter will be focused on dual fuel injection with a port injection of ethanol fuel which is an additive to the gasoline fuel directly injected. This study is aimed to contribute to addressing not only the energy resources by using ethanol as an additive fuel but also the challenges in applications of direct injection to SI engines.

To reach the above aims, the experiments were conducted at RPM of 3500, 4000 and 4500 in each of the light and medium engine load conditions with throttle opening of 20% and 35% respectively. The volumetric ratio of ethanol fuel port injected to gasoline fuel directly injected was varied from E0 (GDI only) to E50 (50% EPI plus 50% GDI) by an increment of about 10% of ethanol fuel port injected. The timing of DI was fixed at 330 CAD bTDC and for PI at 410 CAD bTDC. The fuel pressure for DI and PI were set at 4.0 MPa and 0.25 MPa respectively. The air/fuel ratio was kept at about the stoichiometric ratio ($\lambda \approx 1.0$). The engine was started with GDI only and warmed up to 200±5 °C
(cylinder head temperature) which was the designated engine operating temperature. Then, the percentage of EPI gradually increased from 0.0% to 50% with an approximately 10% increment (i.e. E0, E10, E20, E29, E39 and E50). Table 5.1 lists the engine operating conditions for this chapter. Experimental results will be presented and discussed in terms of the effects of ethanol fuel ratio port injected on engine performance in Section 5.2, on combustion characteristics in Section 5.3 and on engine emissions in Section 5.4.

Table 5.1 Engine testing conditions.

<table>
<thead>
<tr>
<th>Engine load</th>
<th>Light load (20% throttle opening)</th>
<th>Medium Load (35% throttle opening)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speeds (RPM)</td>
<td>3500, 4000, 4500 (only in light load)</td>
<td>Light load MBT (30 CAD bTDC)</td>
</tr>
<tr>
<td>Spark timing</td>
<td>Medium load MBT (23 CAD bTDC)</td>
<td></td>
</tr>
<tr>
<td>Ethanol ratio by volume</td>
<td>E0 (GDI only), E10, E20, E29, E39, E50</td>
<td></td>
</tr>
<tr>
<td>PI timing</td>
<td>410 CAD bTDC</td>
<td></td>
</tr>
<tr>
<td>PI pressure</td>
<td>0.25 MPa</td>
<td></td>
</tr>
<tr>
<td>DI timing</td>
<td>330 CAD bTDC</td>
<td></td>
</tr>
<tr>
<td>DI pressure</td>
<td>4.0 MPa</td>
<td></td>
</tr>
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</table>

5.2 Effect of EPI ratio on engine performance

Figure 5.1 shows the effects of EPI ratio on the IMEP of the three engine speeds 3500, 4000 and 4500 RPM and two engine loads. As shown in Figure 5.1, the IMEP increment is not significant until the ethanol ratio reaches 39%. This is because the variation of IMEP is less than the COVIMEP which is shown in Figure 5.3. When the EPI ratio is further increased from 39% to 50% in the light load, the IMEP increment increases by 6.44% at
3500 RPM, 7.44% at 4000 RPM and by 8.54% at 4500 RPM, compared with that of GDI only (reference condition). Likewise, in the medium load and 4000 RPM engine speed, the IMEP increases by about 10.6% at E50 compared to that of GDI only. However, the IMEP increase is insignificant at 3500 RPM engine speed, as shown in Figure 5.1.

The increase of the IMEP could be attributed to two main factors. Firstly, the air/fuel mixture quality may be improved with the increased ratio of EPI. This is made by the dual fuel injection system when the ratio of ethanol fuel is increased. As a result, the fuel impingement to the cylinder wall caused by direct injection is decreased with the decreased ratio of gasoline fuel directly injected, leading to reducing the rich mixture regions that might be formed near the cylinder liner or in the piston crevice [108, 109]. This could also be associated with the PI timing (120 degrees earlier) earlier than that of DI timing, which provides more time for ethanol fuel to evaporate before the combustion starts. Furthermore, with the increase in ethanol ratio, the ethanol fuel port injected may get full advantage of the high flow velocities that are generated during the intake process [69]. This could strongly help more ethanol fuel to be evaporated, and homogeneously mixed with air before the combustion process starts. Secondly, since ethanol fuel has smaller stoichiometric air/fuel ratio than the gasoline, more mass of ethanol fuel is required to maintain the mixture at the stoichiometric value. Consequently, a greater cooling effect in the intake port enhanced by the increased ethanol fuel port injected may increase the intake flow density. This could explain the increase of the engine volumetric efficiency as shown in Figure 5.2. The increment of the airflow rate equates to the increase of fuel energy flux in a stoichiometric mixture, and the heating value of a stoichiometric mixture increases with the increased ratio of ethanol fuel [5, 81]. This means the increased mass of ethanol fuel and consequently, the increased heating value could partly explain
the increase of the IMEP with the increased ratio of ethanol fuel port injected at medium load and 4000 RPM and at light load and when the ratio is increased from 39% to 50%.

![Figure 5.1 IMEP variation with ethanol PI ratio.](image)

The engine volumetric efficiency is an important parameter to evaluate the effect of using ethanol fuel in SI engines. As shown in Figure 5.2, with the increased ratio of EPI, the engine volumetric efficiency is generally increased in both engine load conditions. At the light load, the volumetric efficiency increases by 6.2%, 2.8% and 2.6% at 3500, 4000 and 4500 RPM respectively when the ratio of EPI is increased from E0 to E50. Similarly, at
the medium load, the volumetric efficiency increases by 4.4% and 6.7% at 3500 and 4000 RPM respectively. As discussed above for results shown in Figure 5.1, this could be attributed to the charge cooling effect enhanced by the increased ratio of ethanol fuel injected into the port. When the ethanol fuel is injected into the intake port, the heat taken by the injected ethanol fuel enhances the charge cooling effect. In addition, the injection timing of EPI is much earlier than that of direct injection (GDI), providing more time for the mixing process of ethanol fuel with air. Consequently, the cooling effect of ethanol fuel may affect the charge that is inside the engine cylinder in addition to the charge in the intake port region. All of these factors help increase volumetric efficiency.

As shown in Figure 5.2, the increased volumetric efficiency means that an additional amount of air should be available for the injected fuel per cycle, leading to a more stable and completed combustion process [97, 110]. These may also explain the improvement in the combustion stability represented by the coefficient of variation of IMEP (COV IMEP) which is reduced when the EPI ratio is increased from E0 to E50, as shown in Figure 5.3. The COV IMEP is derived from in-cylinder pressure and measures the cyclic variability (combustion stability) [69]. When the ratio of EPI is increased from E0 to E50 at light load, the COV IMEP considerably decreases by 35.1%, 41.6% and 43.4% at engine speeds of 3500, 4000 and 4500 RPM respectively. In the medium load condition, the COV IMEP reduces by about 45.4% and 54.0% at 3500 and 4000 RPM respectively. It can be noted that the results of COV IMEP, shown in Figure 5.3, exhibit an acceptable limit of cyclic variability, which is less than 5%, at all the tested conditions.
Figure 5.2 Volumetric efficiency variation with ethanol PI ratio.
5.3 Effect of EPI ratio on combustion performance

To understand the mechanism behind the results as shown in section 5.2, the combustion characteristic parameters, CA0-10%, CA50, and CA10-90% are presented and analysed. CA0-10% is the initiative combustion period defined by CADs from 0% to 10% heat release. CA50 is the CAD at which 50% of the fuel (in mass) has burnt. CA10-90% is the major combustion duration and defined by the time between the CAD at which 10% of
the fuel is burnt and the CAD at which 90% of the fuel is burnt. CA0-10% is influenced primarily by the mixture quality, composition, and motion in the vicinity of the spark plug [69]. Good air-fuel mixing process could reduce the leaning effect around the spark plug, facilitating the initiation of ignition, reducing the ignition delay period and initial combustion duration. This could be associated with the flame speed of fuel, accelerating the flame propagation across the combustion chamber, reducing both the initial and major combustion durations. CA0-10% is a short time period but critical for engines combustion as it can have a strong influence on the combustion stability and quality [111]. Generally, a shorter initial combustion duration means better combustion stability and quality [112, 113]. Figure 5.4 shows the variation of CA0-10% with EPI ratio at the three speeds and two engine loads. Except for the 4000 RPM operating condition, it can be seen that the CA0-10% decreases with the increase of EPI ratio. In the light load condition, CA0-10% decreases by 4.5% and 7.3% at 3500 and 4500 RPM respectively when the ratio of EPI increased from E0 to E50. In medium load condition and 3500 RPM engine speed, CA0-10% decreases by 6.8% at E50 compared to GDI only. In the case of 4000 RPM, the CA0-10% first slightly increases at E10 in the light load and E29 in the medium load before it reverts back to decrease by 2.5% and 5.2% respectively with the increased ratio of EPI to E50. As a result, more stable combustion (less COV_{IMEP}) is achieved with the increased EPI ratio, as shown in Figure 5.3. The shortened CA0-10% may be attributed to the improved mixture quality when the ratio of EPI is increased. Moreover, the greater flame speed and low ignition energy required of ethanol could permit an easier ignition and faster combustion of gasoline/ethanol mixture in the vicinity of the spark plug [36].
The optimum CA50 is a few degrees after the TDC to produce the maximum indicated power [114]. Figure 5.5 shows the CA50 variation with EPI ratio. Generally, CA50 decreases with the increased ratio of EPI in all engine speeds and loads. In light load, CA50 reduces by 2.2, 1.05 and 2.75 degrees at 3500, 4000 and 4500 RPM respectively when the ratio of ethanol is increased from E0 to E50. In the medium load, CA50 first advances (between E0 and E10 in 3500 RPM, and between E0 and E29 in 4000 RPM) before it retards slightly from TDC until the ethanol ratio reaches the 29%. After E29%
as shown in Figure 5.5, the CA50 advances toward the TDC with the increase ratio of EPI. Advancing CA50 to be closer to engine TDC means that the work produced by combustion could be increased, resulting in greater IMEP, as shown in Figure 5.1.

![Figure 5.5 Combustion phase (CA50) variation with ethanol PI ratio.](image)

Figure 5.6 and Figure 5.7 show the variation of the major combustion duration (CA10-90%) and indicated thermal efficiency with EPI ratio. Normally, the shorter combustion duration leads to an increase in the engine thermal efficiency. As the combustion duration is prolonged, the maximum in-cylinder pressure may happen later in the expansion stroke,
reducing the expansion work transfer from the cylinder gases to the piston, decreasing the IMEP [115]. As shown in Figure 5.6, the CA10-90% decreases with the increase in the ratio of EPI until it reaches its minimum in E50 except at 3500 RPM and medium load condition. In light load conditions, the major combustion duration decreases by 4.4%, 5.0% and 7.4% at 3500, 4000 and 4500 RPM respectively when the ethanol ratio is in the range from E10 to E50. At 4000 RPM in medium load conditions, Figure 5.6 shows that CA10-90% at ethanol ratio of 50% is 10% shorter than that of GDI only. On the other hand, at 3500 RPM in medium load, CA10-90% first decreases by about 2.8% at E10 compared to GDI only before it returns to increase by around 3.0% when the EPI ratio further increases to 50%. The reduced CA10-90% could be attributed to the improved air-fuel mixture quality (mixing process) caused by the dual fuel injection strategy, and to some of the remarkable ethanol fuel properties such as the flame speed and oxygen content.

The variation of indicated thermal efficiency with the ratio of EPI is shown in Figure 5.7. It can be noticed that the indicated thermal efficiency increases with the increase ratio of EPI. This performance may be partly explained by the combustion characteristics results shown in Figure 5.4, Figure 5.5 and Figure 5.6 [43, 69, 100]. The combustion performance, which is represented by CA0-10%, CA50 and CA10-90%, shows an approximately similar trend to the indicated thermal efficiency. The mixing time possibly plays an important role in the engine thermal efficiency variation. Increasing the EPI ratio means a larger portion of the used fuel per cycle can have a longer mixing time, improving the mixture quality and then reducing the combustion duration. Consequently, a reduction in the heat losses may be achieved due to the shorter combustion duration, increasing the indicated thermal efficiency. This is thought to be associated with the decreased gasoline fuel impingement on the combustion chamber walls, improving the mixture homogeneity.
and then the combustion quality. In addition, the fuel flame speed is another significant property that could strongly affect the engine performance [110, 116]. The shorter the combustion duration is the less heat may be lost through combustion chamber walls [43, 117]. The greater flame speed of ethanol compared to that of gasoline might shorten the CA10-90%, and increase the indicated thermal efficiency, as shown in Figure 5.6 and Figure 5.7. Nevertheless, when the ratio of ethanol port injected is over 39% at 3500 and 4000 RPM, and over 29% at 4500 RPM, the enhanced cooling effect of ethanol may prolong the CA10-90% which is strongly related with the mixture temperature [60]. On the other hand, this cooling effect may reduce the heat losses through the combustion chamber walls, improving the engine thermal efficiency.
Figure 5.6 Major combustion duration (CA10-90%) variation with ethanol PI ratio.
5.3.1 Effects of EPI on emissions characteristics

Figure 5.8 and Figure 5.9 show the variation of indicated specific carbon monoxide (ISCO) and hydrocarbon (ISHC) emissions with the ratio of ethanol fuel port injected. Compared with that of GDI only, as shown in Figure 5.8, the ISCO emission is reduced by 16.3–83.6% at 3500 RPM, 24.8–77.6% at 4000 RPM and 20.6–56.9% at 4500 RPM when the ethanol ratio is increased from E0 to E50. However, the ISHC emission first decreases when ethanol ratio increases from E0 to E20, and then slightly increases (but it
is still less than that of GDI only) when the ethanol ratio is further increased to E50. The maximum reduction of ISHC in the light load condition is 46.1% at 3500 RPM, 40.8% at 4000 RPM and 46.5% at 4500 RPM when the ethanol ratio is in a range of 20–39%. In the medium load condition, the reduction of ISCO and ISHC is more significant compared to that at the light load, as shown in Figure 5.8 and Figure 5.9. ISCO sharply decreases by 87.8% at 3500 RPM and 78.2% at 4000 RPM when EPI is increased from E0 to E50. Besides, the ISHC emission decreases by 50.9% and 27.4% at 3500 RPM and 4000 RPM respectively.

The ISCO and ISHC decrease could be mainly attributed to the improved air-fuel mixing quality and the homogenous air-fuel mixture distribution inside the cylinder that could be attributed to two main reasons. Firstly, a reduced fuel impingement (caused by GDI strategy) can be achieved with an increased ratio of ethanol fuel. This may reduce the unburned fuel (HC) concentration in emissions caused by incomplete combustion due to local flame extinction that occurs when temperatures are lower and reaction times may become larger than mixing times [43]. Secondly, the longer time available at PI timing (410 CAD bTDC) compared to that at GDI timing (330 CAD bTDC) could potentially enhance the mixing process and then the quality of combustion, which could result in less ISCO and ISHC as well. Moreover, dissociating the CO₂ can occur at the high combustion temperature increasing the CO concentration in the combustion products [100]. This means reducing the combustion temperature to a certain limit could fundamentally contribute to the CO emission reduction. Therefore, the ethanol latent heat of vaporization associated with the early injection timing could essentially reduce the air-fuel mixture temperature and then the CO emission. However, the over-cooling to the air-fuel mixture could slow down the oxidation process of the hydrocarbon fuel, which possibly can increase the CO emission. This analysis may partly explain the sharp CO reduction with
the increase EPI ratio from 0% to around 20% at light load and 40% at the medium load, and the moderate CO reduction when the EPI is further increased to 50% at both engine loads conditions.

![ISCO variation with ethanol PI ratio.](image)

In addition, at light engine load conditions, the ISHC emission slightly increases when the EPI ratio is larger than 20%. This could be related to the increased ratio of ethanol fuel port injected, which impinges the surfaces of the intake port and valve [118]. The evaporation of this fuel depends mainly on heat transfer from the hot surfaces of the intake
port and valve to the fuel film, which is unlike the directly injected gasoline fuel that vaporises mainly by absorbing the heat from the charge in the cylinder. Therefore, the evaporation of the ethanol fuel in EPI condition cools the charge less effectively than that in GDI conditions and leads to a higher wall wetting thus greater HC emissions. Besides, the reduced cooling effect of ethanol fuel (port injected) inside the combustion chamber plus the improved combustion quality (as shown in Figures 5.4 - 5.6) may explain the increase of the NO emission with the increase ratio of ethanol fuel.

Figure 5.9 ISHC variation with ethanol PI ratio.
Figure 5.10 shows the variation of the indicated specific nitric oxide emission (ISNO) with the ratio of EPI at different engine speeds and loads. As shown in Figure 5.10, in the light engine load, ISNO increases by 17.2%, 50.8% and 36.5% at 3500, 4000 and 4500 RPM respectively at E50 compared to that of GDI only. In the medium engine load, at both engine speeds, the ISNO emission increases by about 10.0% when the ethanol ratio is in range 10–50% compared to GDI only. It is well known that the NO emission is strongly related to the flame temperature and the oxygen percentage, which is formed from the N₂ and O₂ dissociation at that elevated temperature [119]. The increase of the NO emission may be attributed to the increased flame temperature [120] caused by enhanced combustion quality, as shown in Figure 5.6 and Figure 5.7. The oxygen content of ethanol fuel could be a second reason that can explain the increase of ISNO with the increased ratio of ethanol port injected [98]. It can be observed that the emissions results shown in Figure 5.8, Figure 5.9 and Figure 5.10 are related to the engine performance results in sections 5.2 and 5.3.

As shown in Figure 5.10, at both engine loads, the ISNO emission decreases when the engine speed is increased. Since the amount of fuel (ethanol plus gasoline) injected per cycle is constant at each testing condition, faster engine speed means less time available for fuel especially for ethanol fuel to be evaporated to absorb more heat from the intake charge. This may reduce the cooling effect of ethanol fuel inside the intake port when the engine speed is higher, but this may increase partly the cooling effect inside the cylinder because most of the evaporation process of ethanol could take place inside the cylinder rather than in the intake port. The reduction in the cooling effect of ethanol fuel when the engine speed is higher can be also seen in the volumetric efficiency performance shown in Figure 5.2.
Chapter Five

Ethanol port injection plus gasoline direct injection

Figure 5.10 ISNO variation with ethanol PI ratio.

5.4 Chapter Summary

The IMEP increased with the increased ratio of EPI. This improvement was mainly attributed to the enhanced quality and distribution of the air-fuel mixture inside the engine cylinder when the dual fuel injection strategy (EPI plus GDI) was used. In addition, the ethanol fuel properties might significantly contribute to this improvement. The improved volumetric efficiency could supply more air to the injected fuel per cycle, leading to the
improved combustion process. The COV_{IMEP} decreased with the increased ratio of EPI, producing a more stable combustion process compared to that of GDI only.

The general trend of the combustion performance was improved when the ratio of ethanol port injected was increased. CA0-10% was reduced by about 7.3% at light load and by 6.8% at the medium load. The CA50 was advanced toward the TDC by 2.7 CAD in the light load and by 2.8 CAD at the medium load when the EPI ratio was increased from E0 to E50. Furthermore, CA10-90% was also shortened with the increased ratio of EPI. When the EPI ratio was increased from E0 to E50, the CA10-90% was shortened by 7.4% in the light load and by 2.8% in the medium load condition.

The ISCO and ISHC decreased with the increased ratio of EPI at different engine speeds and loads. The ISCO reduction was more significant than that of ISHC. The ISCO was decreased by 84% and 88% in the light and medium load respectively. Besides, the ISHC was decreased by about 46% at the light load and by about 51% in the medium load condition. On the other hand, the ISNO increased with the increased ratio of EPI. The ISNO increased by about 51% and 10% in the light and medium load respectively. The reduction in the ISCO and ISHC emissions was attributed to the enhanced combustion quality. The ISNO emission increased because it is strongly related to the combustion temperature and oxygen percentage in the air-fuel mixture.
Chapter Six

6 Effect of EPI+GDI on engine knock

6.1 Introduction

Downsizing SI engines is required in order to minimize the fuel consumed and the CO₂ emitted by engines. However, the downsized engines are also required to maintain the thermal efficiency and maximum rated power. Therefore, technologies such as turbocharging and increased compression ratio are needed in the downsized SI engines with increased risk of engine knock. Engine knock is an abnormal combustion phenomenon that arises from the auto-ignition of the end-gas in front of the propagating flame [121]. This could be associated with high-pressure oscillations and sudden release to the fuel chemical energy of the end-gas [78]. Severe pressure oscillations could cause damage to engine parts when engine knock occurs. Two of the factors affecting engine knock are the fuel octane number and the maximum combustion temperature. Fuels with greater octane numbers allow a higher compression ratio. Lowering the combustion temperature could effectively mitigate the engine knock by keeping the end-gas temperature below the self-ignition degree of fuel [26, 81]. Ethanol fuel octane number is greater than gasoline. In general, direct injection results in a stronger cooling effect than port injection does. The work reported in this chapter is focused on the investigation of engine knock mitigated by the ethanol fuel port injected in a GDI engine.

Experiments aimed to investigate the engine knock mitigation were performed at 22±2 °C ambient temperature and 4000±10 RPM engine speed. The throttle opening percentage was gradually increased from the idle condition to 35% when the medium load condition
was reached. The stoichiometric air/fuel ratio was maintained by adjusting the injection duration. The engine was started with gasoline direct injection and warmed up until the engine body (cylinder head) temperature stabilised at 235±5 °C. Raising the engine temperature to this level was to increase the propensity of engine knock. This method was adopted because the tested engine had a relatively small compression ratio (9.8:1) and large valve overlapping (41.5 CAD), as shown in Figure 4.20. After the engine body temperature was stabilised, the spark timing was advanced from the MBT position (23 CAD bTDC) until the knock event was observed from the in-cylinder pressure curve. The ratio of EPI was gradually increased from 0% (GDI only) to 30% with a 10% (by volume) increment. The GDI timing was fixed at 330 and EPI at 410 CAD bTDC. The fuel pressure for GDI was set at 4.0 MPa and for EPI 0.25 MPa.

The engine knock analysis was performed based on the recorded real-time in-cylinder pressure. The knock was detected if more than 10% of the 100 consecutive cycles had noticeable pressure oscillations (peak-to-peak pressure value over 2.5 bar) [122]. The knock intensity (KI) was derived from the oscillating in-cylinder pressure to quantitatively evaluate the knock severity. In addition to KI, the knocking cycle can be identified by the high frequency of the in-cylinder pressure oscillations, which is within the range of 6 to 12 kHz. The normal cycle pressure frequencies are in the range of 15 to 100 Hz [123].

Figure 6.1 illustrates typical knocking and normal cycles against the crank angle position. The KI was defined as the peak-to-peak in-cylinder pressure value for the recognized knocking cycle [76, 78, 124]. When the knock condition was identified, the spark timing at this condition was defined as the knock limit spark timing (KLST). In each tested condition, the spark timing was first advanced to reach KLST at GDI only. Then the GDI
ratio was decreased and the EPI ratio was increased. Then the spark timing was advanced until the engine reached the new KLST at that EPI ratio. During the experiments, the EPI ratio was gradually increased by 10% (by volume) interval and the knock limit was progressively advanced.
6.2 Effect of EPI+GDI in engine knock

Figure 6.2 shows the relation between the volume ratio of the ethanol fuel port injected with the KLST and KI. As shown in Figure 6.2, the GDI only is less effective in mitigating the engine knock than the EPI+GDI is. In GDI condition, the KLST is around 37 CAD bTDC, and the KI severity greater than 7.0 bar is the worst. When EPI is switched on, the engine knock limit is significantly extended. The KLST is advanced by about 4.0 CAD when the EPI ratio increases from E0 (GDI only) to E20. Furthermore, the KI is reduced from over 7.0 bar in GDI condition to be around 1.0 bar in the E20 condition. When the EPI ratio is further increased to E30 and over, the engine knock is completely suppressed, and KI decreases to be 0.5 bar or less. In the experiments, the spark advance was stopped at 45 CAD bTDC due to the engine performance deterioration when the COV_{IMEP} became over 10% [69].
The knock limits extended with the increased ratio of ethanol fuel port injected could be explained as follows. Firstly, the improved combustion quality plays an important role in engine knock suppression. Ethanol fuel has faster flame speed than that of gasoline fuel, accelerating the flame front propagation, shortening the time available to the end-gas to be self-ignited. Moreover, the vapor pressure can be greater than that of gasoline fuel for the dual fuel (ethanol plus gasoline) at a ratio between E10 and E50 [125]. Consequently, at the ratios of E10-E30, the mixture might evaporate faster, providing a better mixture quality and then better combustion efficiency. Secondly, the octane number of the dual fuel (ethanol plus gasoline) increases with the increased ethanol ratio due to the greater ethanol research octane number [89]. Consequently, the dual fuel mixture could effectively resist the engine knock, pushing the engine knock limit to its maximum. Thirdly, reducing the end-gas temperature could reduce the engine knock propensity. The latent heat of vaporization of ethanol fuel is almost three times greater than that of gasoline.
gasoline, which could have a stronger cooling effect on reducing the combustion temperature. Furthermore, since the equivalence ratio was kept at the stoichiometric condition and the input heating value was fixed, the effect of the ethanol latent heat of vaporization could be amplified due to the smaller stoichiometric air/fuel ratio of ethanol (9.0:1.0) compared to gasoline (14.6:1.0).

6.3 Chapter Summary

The EPI+GDI was more effective on mitigating the engine knock than that of GDI only. In GDI conditions, the KLST was around 37 CAD bTDC, and the KI was greater than 7.0 bar. When the ethanol fuel via port injection was used, the engine knock limit was advanced. The KLST advanced by around 4.0 degrees at E20 compared to that of GDI only. Furthermore, the KI was reduced from over 7.0 bar at GDI only to be around 1.0 bar in the E20 condition. When the EPI ratio was further increased to E30 and over, the engine knock was entirely suppressed, and KI decreased to be less than 0.5 bar in the worst case.
Chapter Seven

7 Conclusions and future work

7.1 Conclusions

Experiments were conducted on a small SI engine to investigate two dual fuel injection strategies: DualEI and EPI+GDI, ultimately aiming to make the use of ethanol fuel more efficient than the current E10 and E85. IMEP, engine thermal efficiency, and emissions were analysed to evaluate the effects of the fuel injection strategies on engine performance. Combustion characteristic parameters were derived from the in-cylinder pressure data for understanding the effects of the fuel injection strategies on engine performance. Additionally, the effect of ethanol fuel on knock mitigation was also examined.

DualEI strategy

At the light and medium load conditions, the results demonstrated that the IMEP was improved over all the DI ratios in DualEI engine compared to GPI only. This improvement was mainly due to the enhanced combustion performance. Compared with GPI only, CA10-90% reduced when the DI ratio was less than 80%. This shows the positive effect of ethanol greater flame speed and the enhanced mixture quality when the DualEI strategy was applied with a certain percentage of the ethanol fuel injected directly into the combustion chamber. However, because of the over-cooling effect and mixture quality deterioration, CA10-90% became to increase when the DI percentage was over 80%.
The emission of NO decreased with the increased ratio of ethanol fuel directly injected. This was due to the cooling effect enhanced by both ethanol latent heat of vaporization and DI strategy. On the other hand, because of the high fuel impediment when the percentage of DI was greater than 32%, the HC emission was increased with the increased ratio of DI. Furthermore, at light engine load, the CO emission was reduced when a relatively small amount of DI percentage was applied. The CO emission reduction was attributed to the enhanced combustion quality. On the other hand, the CO emission was increased with the high percentage of DI which was caused by the over-cooling and poor mixture quality.

Regarding the effect of the spark timing, the IMEP and indicated thermal efficiency were increased with the advanced spark timing. This effect on the indicated thermal efficiency was stronger at the light load than that at the medium load. MBT was identified in terms of the best IMEP and thermal efficiency. The MBT spark timing was 30 CAD bTDC at light load and 23 CAD bTDC at medium load. At the MBT timings, the IMEP and engine thermal efficiency increased with the increased DI ratio, and the \( \Psi_{\text{max}} \) was located at about 10 CAD aTDC at light load and about 12 CAD aTDC at medium load. Furthermore, the CA10-90% was shortened with the increase of the DI ratio, and when the spark timing was advanced from ST15 to ST30 at light load. However, further increase of the DI ratio from 35% to 100% prolonged the CA10-90% due to the over-cooling effect and mixture quality deterioration.

All the emissions were increased with the advanced spark timing. However, the DI ratio had a different effect on different emissions. The HC and CO emissions were increased when the ratio of DI was increased. This might be a result of incomplete combustion due to fuel impingement, mixture quality deterioration, and over-cooling effect. NO was
reduced with the increased DI ratio due to the charge cooling effect enhanced by direct injection and consequently the reduced combustion temperature.

The results for the effect of early DI timing associated with spark timing showed that at medium engine load, the IMEP and indicated thermal efficiency slightly decreased when the DI timing was advanced from 240 to 300 CAD bTDC with small differences caused by the spark timing. When the DI timing was further advanced from 300 to 330 CAD bTDC, the IMEP and indicated thermal efficiency increased slightly. At light load, the IMEP and indicated thermal efficiency increased slightly with the advanced spark timing but independently with the DI timing. As the intake valve was still opened in early DI timing, the air and ethanol fuel port injected were mixing with the gas and fuel directly injected at low cylinder pressure. The mixing process and the quality of the mixture might not be significantly affected by the DI timing.

In the late DI timing, the results demonstrated that the IMEP and indicated thermal efficiency increased with the advanced DI timing. However, the effect of DI timing was not strong when the DI timing was earlier than 80 CAD bTDC. When the DI timing was retarded from 80 to 60 CAD bTDC, the IMEP was decreased from about 0.6 MPa to 0.5 MPa and indicated thermal efficiency from about 26% to 22%. This was mainly because the DI timing of 60 CAD bTDC was too late and left insufficient time for the ethanol fuel directly injected to evaporate and mix with the gases in the cylinder before ignition. This was evident with the longer initial combustion duration at DI timing of 60 CAD bTDC than that with more advanced DI timing. The specific emissions of NO and CO increased with the advanced DI timing as well as spark timing.

This DualEI engine performed better with early DI timing than with late DI timing, in terms of IMEP and indicated thermal efficiency. With early DI timing, the IMEP was in
a range of 0.60-0.68 MPa and indicated the thermal efficiency of 30-32% at medium load and 0.47-0.5 MPa and 29-31% at light load. With late DI timing, the IMEP was in a range of 0.47-0.62 MPa and indicated efficiency 21-27% at medium load and 0.41-0.44 MPa and 22-24% at light load. As identified, the DI timing should not be later than 80 CAD bTDC at medium load. The spark timing should not be later than 19 CAD bTDC at medium load and 28 CAD bTDC at light load.

**Ethanol port injection plus GDI**

In the investigation of EPI+GDI, the IMEP did not increase obviously with the increased ratio of EPI. However, the indicated thermal efficiency increased with the increased ratio of EPI because the total heating value of the fuels reduces with the increase of EPI. This improvement was mainly attributed to the enhanced air and fuel mixing process and distribution inside the engine cylinder when the EPI+GDI strategy was used. In addition, the ethanol fuel properties such as the flame speed and the oxygen content could strongly contribute to this improvement. Besides, the COV_{IMEP} was significantly decreased with the increased ratio of EPI, indicating more stable combustion than that of GDI only.

In terms of the combustion characteristics, when the EPI ratio was increased from E0 to E50, CA0-10% was reduced by about 7.3% at light load and by 6.8% at the medium load. The CA50 was advanced toward the TDC by 2.7 CAD in the light load and by 2.8 CAD at the medium load. Furthermore, the CA10-90% was shortened by 7.4% and 2.8% in the light load and medium load respectively.

The emissions of CO and HC decreased with the increased ratio of EPI at different engine speeds and loads. The reduction of CO emission was more significant than that of HC emission. The CO emission was decreased by 84% and 88% in the light and medium load
respectively. Besides, the HC emission was decreased by about 46% at the light load and by about 51% in the medium load condition. The reduction in the CO and HC emissions was attributed to the enhanced combustion quality. On the other hand, the NO emission increased with the increased ratio of EPI possibly due to the increase of the combustion temperature, and oxygen added by the increased ethanol fuel. The NO emission increased by about 51% and 10% in the light and medium load respectively.

**Engine knock mitigation**

In the GDI condition, the KLST was around 37 CAD bTDC, and the KI was greater than 7.0 bar. When the ethanol fuel via port injection was applied, the KLST was advanced. The KLST at E20 was four crank angle degrees more advanced than that at GDI only. Furthermore, the knock intensity was reduced from 7.0 bar and more at GDI only to be around 1.0 bar in the E20 condition. When the EPI ratio was further increased to E30, the engine knock was entirely suppressed, and the knock intensity decreased to be less than 0.5 bar.

**Contributions and significance**

The main contribution of this research associated with each of the above conclusions will be summarised in this section. The significance of the conducted experiments in this thesis will also be highlighted.

The first conclusion of this thesis was achieved by conducting the DualEI strategy to experimentally investigate the performance of the downsized SI engine. The research was motivated by the need to find a substitute for fuels derived from petroleum and/or an improvement to the blending of ethanol fuel with the gasoline. Although several studies used the ethanol fuel solely or to be blended with the gasoline, the majority of engine
researches used conventional port injection systems. In this thesis, special concern was
given to comprehensively analyse the combustion and the emissions characteristics of
ethanol fuel dually injected. The DI to PI ratio of ethanol fuel was changed from 0% to
100%. The effect of spark timing and the associated effect of the direct injection timing
with the spark timing on the performance of the DualEI engine were deeply investigated.
The results showed superior performance of the DualEI engine over the gasoline one at
most engine operating conditions. The best conditions of the DualEI engine performance
were also determined.

The second conclusion of this thesis was drawn from the experimental investigation of
the ethanol port injection to the GDI engine (EPI+GDI). The ratio of ethanol fuel was
changed from 0% to 50% (E0 to E50). The experiments showed that the ratio of ethanol
fuel could be increased greater than the common percentage (~10%) used in Australia.
The results demonstrated that a certain improvement in the engine performance could be
achieved with the increase in the ratio of ethanol fuel port injected. Moreover, the effect
of the ratio of EPI on the GDI engine knock suppression was also examined. The EPI
effectively mitigated the engine knock and permitted more advanced spark timing.
Results showed that every 10% increment (by volume) of EPI permitted around 2.0 CAD
advance of knock limit spark timing. When the EPI ratio was 30% and over, the engine
knock was entirely suppressed.

7.2 Future work

Firstly, experiments on the GDI engine demonstrated a strong potential for ethanol fuel
in the engine knock mitigation, which advanced the knock limit spark timing and reduced
the knock intensity (KI). The engine knock could develop to become a super-knock (pre-
mature ignition) when the SI engine is downsized to maintain the output power and reduce
the CO2 emissions. The super-knock phenomenon may happen due to the needs for a high level of turbocharging and a large compression ratio. The KI could be severely increased from around 5-10 bar in the normal knock to greater than 250 bar when the super-knock occurred as reported by many works in the literature. This could cause severe damage to the engine parts. Therefore, it is practically beneficial to investigate the mechanism behind the super-knock phenomenon to eliminate its effect on the SI engines development. In order to do a further investigation on the engine downsizing concept especially the super-knock phenomenon, a set of modifications are required. For instance, the large valve overlapping was a real obstacle in implementing intake flow boosting to simulate the turbocharger effect on the engine performance. Consequently, the overlapping needs to be reduced and an engine turbocharger needs to be installed. Because of the hardware limitation of the engine dynamometer, it needs to be replaced by a capable one that is able to control the engine at the full load conditions. Furthermore, it was realized from the experimental results that the injection was strongly related to the cooling effect of ethanol fuel plus the mixing quality of air-fuel mixture. Therefore, it would be useful if the ECU were further modified to be able to control some of the port injection parameters such as injection timing and pressure. This should help to conduct a deeper experimental investigation about the effect of the EPI on the GDI engine performance and its role in the engine knock mitigation.

Secondly, the EPI has shown great potential in suppressing the engine knock. The physiochemical properties of ethanol fuel such as the great latent heat of vaporization, high octane rating, low adiabatic flame temperature, and high flame speed have been considered for their capacity in contributing towards knock suppressing. However, which property has played the more dominant role and what is the order of the effectiveness for these factors? These questions should be answered via simulation works. The results of
the simulation work can provide helpful feedback for the future design work on the GDI+EPI engine. In such a way, the effectiveness of utilising the EPI to suppress the engine knock can, therefore, be optimized.
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gasoline and gasoline–alcohols dual-fuel spark ignition (DFSI) combustion for
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References


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References


### Appendix

**Literature review key findings**

<table>
<thead>
<tr>
<th>Reference</th>
<th>Experimental method</th>
<th>Key findings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Taniguchi et al. [63]</td>
<td>Ethanol PI (100% ethanol), SI engine.</td>
<td>The full load engine performance was improved especially at the higher compression ratio but the HC emission was increased. At partial load conditions, the thermal efficiency was improved, and the HC and NO emissions was reduced.</td>
</tr>
<tr>
<td>Bielaczyc et al. [13]</td>
<td>Blended fuel E0-E85 (gasoline plus ethanol), SI engine.</td>
<td>Adding ethanol to gasoline fuel increased the engine output power and fuel consumption and reduced the CO, NO and HC emissions. Ethanol–gasoline blends allowed increasing compression ratio without knock occurrence.</td>
</tr>
<tr>
<td>Turner et al. [31]</td>
<td>Blended fuel E0-E100 (gasoline plus ethanol), SI engine.</td>
<td>Adding ethanol to gasoline reduced combustion initiation duration, and higher in-cylinder pressure was achieved. The combustion efficiency and stability were improved. The NO emission was similar or reduced, and CO emission was decreased.</td>
</tr>
<tr>
<td>Ramadhas et al. [11]</td>
<td>Blended fuel E0-E20 (gasoline plus ethanol), SI engine.</td>
<td>The E20 blends reduced the CO emission by up to 15%; HC emissions by up to 20% and NO emissions up to 45% compared to that of E0. A longer combustion duration was noticed with ethanol blends (E5-E20). No variations in IMEP and the cumulative heat transfer values were observed.</td>
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<table>
<thead>
<tr>
<th>Ethanol Dual Injection Strategy</th>
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<tbody>
<tr>
<td>Yuan et al. [5]</td>
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<tr>
<td>Huang et al. [54]</td>
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</table>
over cooling effect occurred, and HC and CO emissions were increased.

<table>
<thead>
<tr>
<th>Authors</th>
<th>System Configuration</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kim et al. [126]</td>
<td>Dual fuel injection (EPI+GDI), two ethanol port injectors</td>
<td>The knock was suppressed when ethanol was added, and an increase in the maximum IMEP was observed consequently. The HC and particulate emissions were reduced while the NO emission level did not substantially change.</td>
</tr>
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</table>

### Spark and Injection Timings

<table>
<thead>
<tr>
<th>Authors</th>
<th>Configuration</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Davy et al. [67]</td>
<td>GDI SI engine, various DI timings</td>
<td>At the early injection timing, fuel sprays were controlled by the state of the intake and intake-generated gas flows at the start of injection, which was directly affecting the tip penetration, wall impingement, and air-fuel mixing. At the late direct injection timing, the development of fuel sprays was dominated by the increased in-cylinder pressure at the start of injection.</td>
</tr>
<tr>
<td>Yuan et al. [26]</td>
<td>Dual fuel injection (EDI+GPI), various EDI timings</td>
<td>LEDI was effective in suppressing knock and permitting more advanced spark timing but resulted in low engine efficiency and high emissions. In EEDI conditions, the volumetric efficiency was increased and thermal efficiency was improved.</td>
</tr>
<tr>
<td>Kim et al. [60]</td>
<td>Dual fuel injection (EPI+GDI), various port injection timing</td>
<td>The knock occurrence decreased as the ethanol injection timing was held while intake valves were open. Significant reductions in CO, HC, and particulate emissions were noticed under a compression ratio of 13.3 compare to that of 9.5.</td>
</tr>
</tbody>
</table>