Numerical investigation of the effect of injection strategy on a high-pressure isobaric combustion engine

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Abstract

High-pressure isobaric combustion adopted in the double compression expansion engine (DCEE) has the prospect to achieve higher thermal efficiency compared to conventional diesel combustion. This work numerically explored the effects of various injection strategies on the combustion and emission characteristics of isobaric combustion. The study developed a mathematical model to predict the injection rate profile. After validations, extensive simulations were conducted with a peak pressure of up to 300 bar - mimicking the high-pressure unit of DCEE. Several major engine design parameters such as the exhaust recirculation gas (EGR) rate, [engine speed, injection strategy, and intake pressure](#) were varied and evaluated. The results demonstrated that a higher EGR rate resulted in a higher exhaust loss but a lower heat transfer loss owing to the lower combustion temperature, so the thermal efficiency exhibited a firstly growing and then declining trend. Besides, a higher engine speed generated a higher thermal efficiency due to the shorter combustion duration and thus lower heat transfer loss. Consequently, a peak thermal efficiency of 47.5% was achieved at EGR =50% and 1800 rpm. [The high-](#)

pressure cylinder performance can also be improved with an appropriate introduction of the isochoric combustion, but its impact on the whole DCEE setup needs further investigation.

Keywords: Isobaric combustion; Injection strategy; Double compression expansion engine; Thermal efficiency; Compression ignition

Nomenclature

BTE	Brake thermal efficiency	IMEPg	Gross indicated mean effective pressure
CDC	Conventional diesel combustion	LP	Low-pressure
CFD	Computational fluid dynamics	NOx	Nitric oxides
CI	Compression ignition	Phi, ϕ	Equivalence ratio
CO ₂	Carbon dioxide	ROI	Rate of injection
DCEE	Double compression expansion engine	SOI	Start of injection
EGR	Exhaust gas recirculation	TDC	Top dead center
HP	High-pressure	1D	One-dimensional
HRR	Heat release rate	3D	Three-dimensional
ICE	Internal combustion engine		

1. Introduction

Due to concerns about global warming, many countries have enacted plans to limit carbon dioxide (CO₂) emissions. Compared to sectors like power, industry, and construction, transportation is a major contributor to CO₂ emission, accounting for about 60% of total liquid fossil fuel consumption and 29% of total greenhouse gas emissions around the world [1, 2]. Currently, most on-road vehicles are powered by internal combustion engines (ICEs). Although there are plans to replace them with alternative technologies such as battery- and fuel-cell-driven power systems, most projections show that ICEs will still play a significant role in human society in the next few decades [3]. Especially, for areas like heavy-duty trucks, ships, and off-road machines, it is still challenging for other competing technologies to gain an advantage over ICEs [3].

It is anticipated that there will be a 45% growth in heavy-duty vehicles by 2040 [4]; meanwhile, the CO₂ emission standard is becoming more stringent [5]. To fulfill the ever more strict CO₂ reduction target, the direct solution is to employ eco-friendly and low-carbon fuels, such as hydrogen and methanol [6, 7]. Simultaneously, continuing efforts are needed to improve engine efficiencies to make an immediate impact on the environment. In the past few decades, numerous efforts have been made by global researchers, and various advanced engine combustion concepts were proposed, such as the Homogeneous Charge Compression Ignition [8], Partially Premixed Compression Ignition [9, 10], and Reactivity Controlled Compression Ignition [11-13]. However, the improvement from these combustion strategies is still restricted and it might not be sufficient enough to meet the more stringent CO₂ emission regulations. As a consequence, there is a continuing demand for new engine concepts that are more efficient than conventional engines [14].

Compared to the conventional four-stroke Otto and Diesel engine cycles, the split-cycle engine concept has the advantage of more freedom for optimization [15]. Based on the theory, researchers have proposed several engine setups, including the series XIV engine [16], Scuderi engine [17], CryoPower engine [18], and Double Compression Expansion Engine (DCEE) [14]. Of these newly proposed engine concepts, the DCEE concept is one of the most promising technologies that can potentially solve the efficiency issue faced by the convention engines [14, 19-22]. Figure 1 illustrates a schematic for the DCEE concept with a 4-4 stroke setup [14]. It includes a low-pressure (LP) cylinder and a high-pressure (HP) cylinder. Note that the LP cylinder has a large bore and a high compression ratio; it acts as a compressor to generate high-pressure compressed air, and as an expander to make use of the exhaust gas energy from the HP cylinder. Moreover, the LP cylinder enhances the mechanical efficiency due to the low peak cylinder pressure. Correspondingly, the HP cylinder has a small bore and a low

compression ratio; it serves as the primary work provider; the heat transfer loss is significantly reduced owing to the small wall surface area. Due to the additional compression-expansion process, and lower heat transfer and friction losses, the DCEE concept is able to obtain a higher engine efficiency than conventional ICEs. Besides, since the LP cylinder can supply high-pressure compressed air for the HP cylinder, the engine load can be significantly extended.

Fig. 1. Schematic of the DCEE concept with a 4-4 stroke setup [14].

By performing one-dimensional (1D) simulations, Lam et al. [14, 20] reported the potential to reach a brake thermal efficiency (BTE) of 56% for the DCEE concept by using a lambda of 3.0. Besides, since the combustion pressure is extremely high in the HP unit, the employment of isobaric combustion mode instead of conventional diesel combustion (CDC) mode is necessary. In comparison to CDC, isobaric combustion reduced the heat transfer loss but increased the exhaust loss from the HP unit [14]. The high exhaust loss cannot be perceived as a limitation in the DCEE, since it has an additional expansion stage to utilize exhaust enthalpy. However, isobaric combustion using a single injector suffers from high soot emission because fuel jets continuously enter into oxygen-deprived regions [23]. To further enhance isobaric combustion performance, Okamoto and Uchida [19, 24, 25] proposed to employ a multi-injector setup, including one central injector and two side injectors; by adjusting the injection strategy, desired heat release rate (HRR) profile and high thermal efficiency were obtained. Recently, Nyrenstedt et al. [26] conducted three-dimensional (3D) computational fluid dynamics (CFD)

simulations to compare the single- and multi-injector setups under the CDC mode; they confirmed that compared to the single-injector setup, the multi-injector setup resulted in less heat transfer loss but higher exhaust loss, which is beneficial for the DCEE concept.

Based on Lam et al.'s works [14], a peak combustion pressure of 300 bar for the HP unit led to the highest engine efficiency. However, such a high peak combustion pressure is still a challenge in practical experiments due to the mechanical limit of the metal engine body. To guide the practical engine design, it is better to conduct 3D simulations beforehand, which helps to expedite the optimization of engine parts and control parameters [27, 28]. Additionally, due to the excessive pressure rise rate under the CDC mode, it is desirable to adopt an isobaric combustion strategy [21]; **however**, this brings about more complexity of injection control strategy since multiple injection events are needed to achieve isobaric combustion [29].

This work intends to numerically investigate the effects of various injection strategies on a high-pressure isobaric combustion engine. A mathematical model to predict the rate of injection (ROI) profile for a Delphi injector was developed and validated against the recent experimental measurements. An extensive parametric study was performed to identify the optimal injection strategy that led to the most efficient and clean combustion process. This paper is organized as follows. The first part describes details of the mathematical ROI model. Then experimental data and computational models are introduced. The next section presents the simulation results. The conclusions are reported in the last section.

2. Mathematical model of injection rate

2.1. Model description

Desantes et al [30] first proposed a mathematical model to predict the ROI profiles for the boot-

type injections. Recently, based on their work, Xu et al. [31] developed a modified model to predict the ROI profiles for Bosch common-rail injectors; their predictions demonstrated very good agreement with the experimental data. However, this model failed to reproduce the measured data for Delphi injectors, owing to the different internal flow features compared to Bosch injectors. Since multiple injections are required for the isobaric combustion and accurate prediction of the spray development process is significant in reproducing the complicated in-cylinder spray-combustion behavior, it is necessary to develop a new robust injection model.

The new model generally follows the same rule as proposed by Xu et al.'s work [31]. Figure 2 illustrates the schematic of this injection model. The basic theory of this model is to split the injection profile into five continuous stages, where each stage is described with a polynomial equation, which represents the physical motion of the needle valve. As demonstrated in Fig. 2, t_1 represents the rapid valve opening; t_2 represents the slow valve opening; t_3 represents the full valve opening; t_4 represents the slow valve closing, and t_5 represents the rapid valve closing. Note that the injection rate will change with the variations of injection pressure and energizing time; therefore, t_2 and t_3 may not exist when the energizing time is too short.

Fig. 2. Schematic of the mathematical injection model.

Five polynomial equations are employed to model the injection profiles, as defined by

$$Q_1(t) = a_1 t^3 + b_1 t^2 + c_1 t \quad (1)$$

$$Q_2(t) = a_2 t^2 + b_2 t + c_2 \quad (2)$$

$$Q_3(t) = Q_{max} \quad (3)$$

$$Q_4(t) = a_4 (t - t_{open})^2 + Q_{t_{open}} \quad (4)$$

$$Q_5(t) = a_5 (t - t_{open} - t_4) + c_5 \quad (5)$$

In these equations, units of mass, time, and pressure are mg, ms, and bar, respectively. From the experimental data provided by Aljohani et al. [32], t_1 and t_2 are found to be 0.15 and 0.75 ms, respectively. c_1 , a_2 , and b_2 are obtained by fitting the measured ROI profile and c_2 is calculated by solving $Q_2(t_2) = Q_{max}$, where the peak mass flow rate (Q_{max}) is defined as,

$$Q_{max} = n C_d \rho_f A_0 u_{max} \quad (6)$$

$$u_{max} = \sqrt{\frac{2\Delta P}{\rho_f}} \quad (7)$$

$$C_d = -0.3128 \left(\frac{10000}{Re} \right)^2 + 0.1867 \left(\frac{10000}{Re} \right) + 0.8057 \quad (8)$$

In equations (6-8), n is the nozzle number, ρ_f is the fuel density, A_0 is the nozzle area, ΔP is the difference between injection and ambient pressures, and Re is the Reynolds number. Because $Q_2(t)$ is already known, a_1 and b_1 are calculated by solving $Q_1(t_1) = Q_2(t_1)$ and $Q_1'(t_1) = Q_2'(t_1)$. [Note that \$t_{open}\$ represents the needle opening duration and needs to be solved by an iterative method.](#)

Since the needle valve cannot be closed immediately owing to the water hammer effect, there is a gradual valve closing period (t_4) followed by a rapid valve closing period (t_5), which are defined as,

$$t_4 = t_{peak} (0.1 t_{peak}^2 - 0.32 t_{peak} + 0.46) \quad (9)$$

$$t_5 = t_{peak} (-0.1 t_{peak} + 0.26) \quad (10)$$

$$t_{peak} = \min(t_2, t_{open}) \quad (11)$$

In equations (4-5), a_4 , a_5 , and c_5 are calculated by solving $Q_4(t_{open} + t_4) = Q_5(t_{open} + t_4)$,

$$Q_4'(t_{open} + t_4) = Q_5'(t_{open} + t_4), \text{ and } Q_5(t_{open} + t_4 + t_5) = 0.$$

2.2. Model validation

The measured injection rate data from Aljohani et al. [32] is used for the model validation. The experiment was conducted on a modified single-cylinder optical engine [33]. A spray momentum method [34, 35] was adopted to measure the injection rate. A six-nozzle Delphi injector that is capable of reaching an injection pressure up to 2700 bar was employed. The major specification of the injector is summarized in Table 1. Figure 3 shows the schematic of the injection rate test rig [32]. The optical components were removed to install the spray momentum rig. A piezoelectric pressure sensor was mounted in front of an injector nozzle. The force signal was acquired with a resolution of 200 kHz (5 μ s). A low-sulfur diesel was employed as the fuel and a cylindrical container was used to collect the injected fuel. More details of the experiment can be referred to [32].

Table 1. Specification of the injector.

Manufacturer	Delphi
Type	F2, Common rail solenoid
Nozzle number	6
Nozzle diameter (μ m)	240
Spray angle ($^{\circ}$)	147
Static flow rate at 100 bar (L/min)	2.5

Fig. 3. Schematic of the injection rate test rig [32].

The mathematical model is developed in the MATLAB environment. Seven parameters are needed to calculate the injection profile, including the nozzle number and diameter, total injected mass, injection and ambient pressures, fuel density, and fuel viscosity. Figure 4 compares the experimental and predicted injection rate profiles at various injection pressures and energizing times. The model can reasonably well reproduce the experimental data. With the increase of energizing time the peak injection rate firstly grows and then reaches a steady value, indicating that the injector needle valve is fully open. Owing to the higher injection velocity, an earlier fully open time and a higher peak injection rate are observed with the increase of injection pressure. Note that the real injection duration is slightly shorter than the energizing time owing to the hydraulic delay. Besides, the nozzle was fully open after 0.95 ms, which resulted in an earlier ending. Moreover, the data are depicted with an interval of 0.2 ms after 1.05 ms; therefore, the intervals look longer than the 0.1 ms ones.

Fig. 4. Comparison between the experimental [38] and predicted injection rate profiles at various injection pressures and energizing times. Solid lines: experiment; dotted lines: simulation.

3. Computational models and validation

3.1. Computational models

The CONVERGE/3.0 [36] package is employed for the 3D CFD simulations. The turbulence is simulated by the renormalization group k- ϵ model [37]. The liquid jet dynamics are modeled by the Lagrangian-parcel approach [38]. The spray breakup is simulated by the Kelvin-Helmholtz Rayleigh-Taylor model [39]. The droplet collision and evaporation are simulated by the no-time-counter algorithm [40] and the Frossling correlation approach [41], respectively. The combustion is modeled by the detailed chemistry solver SAGE [42] coupled with a reduced combustion mechanism of n-heptane [43]. The heat transfer is modeled by the method proposed by O'Rourke and Amsden [44]. Details of these modules are described in [45]. All of the simulation cases start from the intake valve closing timing to the exhaust valve opening timing. The initial in-cylinder mixture is considered homogeneous. The initial pressure and temperature values are taken from the experiment. The exhaust gas recirculation (EGR) rate is defined as the ratio of intake CO_2 mass divided by exhaust CO_2 mass.

Owing to the axisymmetric piston geometry and 7-nozzle injector, a 51.4° -sector mesh is adopted to save computational expenses. Figure 5 depicts the computational mesh. This work employs a base

mesh of 2.0 mm. A fixed embedding region is imposed near the nozzle tip. Besides, the adaptive mesh refinement model is activated and a refined scale of 3 is used, which generates a minimum mesh size of 0.25 mm. Based on our previous mesh-sensitivity studies [46-48], this mesh setup can obtain convergent results.

Fig. 5. Illustration of the computational mesh having a 51.4°-sector with a base mesh of 2.0 mm.

3.2. Computational validation

The experimental data from the work of Babayev et al. [29] is used for the computational validation. Note that this experiment also adopted the same type of Delphi injector as in [30] but the nozzle number and diameter were slightly different. Since the internal nozzle structures are similar for both injectors, the injection model is still able to capture the injection rate. Table 2 shows specifications for the engine and injector. Further details about the experiment are available in [29]. In brief, the engine was operating at 1200 rpm. A high intake pressure of about 3.1 bar was employed to achieve a peak combustion pressure of 150 bar. The intake temperature was kept at 353 K and no EGR was adopted. Various engine loads were achieved using a multi-injection method. Injection details for each case are summarized in Table 3. The injection profiles are calculated by the mathematical model that was developed herein.

Table 2. Engine and injector specifications.

Engine type	Volvo, 4-stroke
Bore (mm)	131
Stroke (mm)	158
Connecting rod length (mm)	255
Displacement volume (L)	2.13

Geometric compression ratio	17:1
Injector type	Delphi F2
Nozzle number	7
Nozzle diameter (μm)	~ 225
Spray angle ($^\circ$)	150
Injection pressure (bar)	2300

Table 3. Details of the injection strategy.

	3 inj.	4 inj.	5 inj.
Exp. IMEPg (bar)	5.96	11.1	19.6
Injected mass (mg/cycle)	64.7	117.6	222.0
Injection pressure (bar)	2300	2300	2300
SOI 1 st inj. ($^\circ$)	-3.0	-3.0	-3.0
SOI 2 nd inj. ($^\circ$)	0.5	0.5	0.5
SOI 3 rd inj. ($^\circ$)	4.5	4.5	4.5
SOI 4 th inj. ($^\circ$)	-	9.0	9.0
SOI 5 th inj. ($^\circ$)	-	-	12.7

Figure 6 compares the experimental and predicted pressure and HRR profiles, indicated thermal efficiency, and NOx and soot emissions. The simulation work can reasonably predict the experimental data at various engine loads. Owing to the isobaric combustion feature, more injection events are needed to achieve a higher engine load. However, too many injection events lead to a prolonged combustion process and lower engine efficiency. Both the NOx and soot emissions demonstrate a declining trend with a higher engine load. Note that the unit is g/kW-h, so the absolute NOx emission is still higher with the growth of engine load due to the longer combustion period and higher combustion temperature.

Fig. 6. Comparison between the experimental and predicted (a) pressure and HRR profiles, (b) engine efficiency, and (c) NOx and (d) soot emissions.

4. Results and discussions

Lam et al. [14, 20, 22] reported that for the DCEE concept the isobaric combustion mode performed better compared to the CDC mode since it could generate higher exhaust energy that could be recovered at the expansion stage. However, studies on the isobaric combustion mode are still very limited and most of them [27, 28, 49] adopted a peak combustion pressure no higher than 150 bar. With the help of CFD tools, this work focuses on exploring the isobaric combustion mode with a significantly high peak pressure of about 300 bar. A high engine load with a cyclic injection mass of 222 mg is investigated, corresponding to a gross indicated mean effective pressure (IMEPg) at about 19.6 bar [29]. Sustaining such a high combustion pressure would pose a significant challenge for the engine control strategies. As a result, to provide more guidance for the practical engine design, it is worthwhile investigating the

effects of major engine design parameters such as EGR rate, engine speed, and injection strategy on the engine combustion performance and emissions.

In the subsequent simulations the 5 inj. case is taken as the baseline case. Since the peak pressure is only 150 bar, pressure at the intake valve closing timing (P_{IVC}) should be elevated from 3.3 to 6.3 bar to achieve the peak pressure of 300 bar. Although it is a challenge for a conventional turbo- or electric-charger to achieve such a high peak pressure, it is possible for the DCEE concept by using the compression work of a piston through the LP cylinder.

4.1. Effect of EGR

Due to the high intake pressure, the global lambda will be significantly high, which promotes NO_x formation. As a result, EGR is required to inhibit NO_x emissions. Figure 7 shows the relationship between EGR rate, O₂ concentration, and global lambda. Both lambda and O₂ concentration show a near-linear relationship with the EGR rate when it is increased from 0% to 30%. However, when the EGR rate is further increased, O₂ concentration and lambda are reduced more rapidly, especially when the EGR rate is higher than 60%. The global lambda is still higher than 2 even at an EGR rate of 60%, indicating the potential of a clean combustion process.

Fig. 7. Relationship between EGR rate, O₂ concentration, and lambda.

Figures 8(a) and 8(b) show the predicted traces of pressure and HRR, and average temperature and

mass fraction of $\phi > 1.5$, respectively. Note that slightly higher intake pressures are employed with the introduction of EGR to maintain the same compression trace because EGR affects the initial species composition and adiabatic index. Besides, the ROI profiles were also optimized to achieve an isobaric combustion process. For instance, at an EGR rate of 0%, five injections are required, but with the increase of EGR rate fewer injections are adopted to maintain a high injection rate and an isobaric combustion process. Owing to the thermal and dilution effects of EGR which reduce the combustion temperature and postpone the heat release process, earlier injections and higher injection rates are demanded. Details of the P_{IVC} value and injection strategy at various EGR rates are listed in Table 4.

Fig. 8. Comparison of the predicted traces of (a) in-cylinder pressure and HRR and (b) average temperature and mass fraction of $\phi > 1.5$ at various EGR rates.

Table 4. P_{IVC} and injection strategy at various EGR rates.

EGR rate	0%	10%	30%	50%	60%
P_{IVC} (bar)	6.33	6.35	6.40	6.51	6.59

Injection pressure (bar)	2300	2300	2300	2300	2300
SOI/Mass 1 st inj. (°, mg/cyc)	-2.2/2.7	-2.2/2.7	-2.2/2.7	-0.5/20	-0.5/20
SOI/Mass 2 nd inj. (°, mg/cyc)	-0.6/18	-0.6/18	0.3/18	2.1/44	2.1/44
SOI/Mass 3 rd inj. (°, mg/cyc)	2.3/44	2.3/44	2.2/44	4.9/158	3.9/158
SOI/Mass 4 th inj. (°, mg/cyc)	5.1/53	5.1/53	5.0/53	-	-
SOI/Mass 5 th inj. (°, mg/cyc)	8.2/104	8.2/104	8.1/104	-	-

Figure 9(a) compares the predicted energy percentage at each EGR rate. The total work output shows a firstly growing and then declining trend with the increase of EGR rate. Heat transfer loss monotonically decreases due to the lower combustion temperature, as shown in Fig. 8(b). However, the exhaust loss increases since the lower combustion temperature and oxygen concentration slow down the heat release process, as shown in Fig. 8(a). Due to the competitive relationship between heat transfer loss and exhaust loss, a peak engine efficiency of 47.1% is obtained at an EGR rate of 50%. However, even at a peak EGR rate of 60%, the engine efficiency is still comparable to that at a zero-EGR condition. When deploying the DCEE concept, the case with 60% EGR may still be attractive as the exhaust losses can be translated into useful work through the additional expansion stage. Previous studies [50, 51] ran the engines at low peak combustion pressures so that low engine efficiency was obtained at a high EGR rate. In contrast, in the current work, the combustion efficiency is always high owing to the abundant oxygen concentration.

Fig. 9. Comparison of the predicted (a) energy percentages and NOx and soot emissions at various EGR rates.

Figure 9(b) shows the predicted NOx and soot emissions at each EGR rate. In agreement with previous works [52, 53], a trade-off relationship is observed between NOx and soot emissions. NOx formation is significantly inhibited at a higher EGR rate owing to the lower combustion temperature and oxygen concentration, which, however, also significantly inhibits the soot oxidation. As observed from Fig. 9, to achieve a high engine efficiency but simultaneously low levels of NOx and soot emissions, an EGR rate of 50% is found to be the optimum value.

4.2. Effect of engine speed

Different from the conventional diesel combustion mode, which typically adopts only a single injection, the isobaric combustion mode is significantly more sensitive to various engine speeds because multiple injections are usually required; this poses a great challenge to the injection control strategy and the situation becomes more complicated when EGR is added. As a result, in this subsection, the effect of engine speed on the isobaric combustion at various EGR rates is investigated. The simulated cases at an engine speed of 1200 rpm in Section 4.1 are taken as the baseline cases, so each case has the same fuel energy input. Note that at a higher engine speed the injection strategy is modified to maintain the isobaric combustion process.

Figure 10 compares the predicted traces of (a) pressure and HRR, (b) average temperature and heat transfer rate, and (c) mass fraction of $\phi > 1.5$ at various engine speeds with an EGR rate of 30%. Table 5 lists the injection details. By varying the injection strategy isobaric combustion is achieved at various engine speeds. Fewer injection events are required at a higher engine speed to maintain the isobaric combustion process because at a higher engine speed there is less cyclic time (in seconds) for the development of spray and combustion processes. As a result, the employment of fewer injection events promotes the average injection rate and combustion heat release rate. As seen in these figures, with the increase of engine speed from 1200 to 1500 rpm, the modification of injection strategy from five injections to two consecutive injections is effective in achieving a similar isobaric combustion process. However, even with a single injection strategy, it is still challenging at 1800 rpm to achieve isobaric combustion after 10° due to the reduced residence time, which reduces the combustion temperature and a richer mixture distribution was generated thereafter.

Fig. 10. Comparison of the simulated traces of (a) in-cylinder pressure and HRR, (b) average temperature and heat transfer rate, and (c) mass fraction of $\phi > 1.5$ at various engine speeds with an EGR rate of 30%.

Table 5. Injection strategy at various engine speeds (EGR = 30%).

Engine speed (rpm)	1200	1500	1800
Injection pressure (bar)	2300	2300	2300
SOI/Mass 1 st inj. (°, mg/cyc)	-2.2/2.7	-1.0/21	-1/222
SOI/Mass 2 nd inj. (°, mg/cyc)	0.3/18	1.0/201	-
SOI/Mass 3 rd inj. (°, mg/cyc)	2.2/44	-	-
SOI/Mass 4 th inj. (°, mg/cyc)	5.0/53	-	-
SOI/Mass 5 th inj. (°, mg/cyc)	8.1/104	-	-

Figure 11 compares the predicted energy percentages at various engine speeds and EGR rates. In agreement with the previous section, for all engine speeds, there is always an initially growing and then declining trend for engine efficiency with an increase in the EGR rate, and a peak engine efficiency is obtained at an EGR rate of 50%; this is owing to the competitive relationship between the lower heat transfer loss and higher exhaust loss at a higher EGR rate. On the other hand, higher engine efficiency is obtained at a higher engine speed, primarily owing to the lower heat transfer loss. As seen from Fig. 10(b), the higher engine speed leads to a lower heat transfer rate. Note that here the heat transfer rate is in the unit of J/deg. By converting crank angles into seconds, their peaks for various cases would be

similar but their durations in seconds would be shorter at a higher engine speed, which consequently results in the lower total heat transfer loss.

Fig. 11. Comparison of the predicted energy percentages at various engine speeds and EGR rates.

Figure 12 compares the predicted NOx and soot emissions at various engine speeds and EGR rates. Still, there is a trade-off relationship between NOx and soot emissions at a higher EGR rate. At each EGR rate, NOx emission demonstrates a declining trend but soot emission demonstrates a growing trend. The results can be explained by Fig. 13, which shows the predicted distributions of temperature, equivalence ratio (ϕ), and soot mass fraction at 16°. Since the lowest combustion temperature and richest mixture distribution are obtained at 1800 rpm, the lowest NOx emission but highest soot emission are generated. Additionally, compared to 1200 rpm, a similar combustion temperature is got at 1500 rpm; this leads to their similar NOx emissions. Although both the 1200 and 1500 rpm cases show a similar mixture distribution, the latter case has less time in seconds for the soot oxidation process. As a result, a higher soot emission is generated at 1500 rpm. In the context of the DCEE concept, the high soot emission with higher engine speeds will be reduced with the additional expansion stage, since the charge has a longer residual time for soot oxidation.

Fig. 12. Comparison of the predicted (a) NOx and (b) soot emissions at various engine speeds and EGR rates.

Fig. 13. Predicted distributions of temperature, equivalence ratio, and soot mass fraction at various engine speeds with EGR = 30%.

4.3. Effect of injection strategy

Various injection strategies can be employed to achieve isobaric combustion, so there should always exist an optimum solution to achieve the highest engine efficiency but lowest emissions. In this subsection, the effects of various injection strategies on engine combustion performance and emissions are explored. The baseline case is taken from Section 4.1 with an EGR rate of 0%.

Figures 14(a) and 14(b) show the predicted traces of in-cylinder pressure and HRR, and average temperature and mass fraction of $\phi > 1.5$, respectively, using various injection strategies. Four injection strategies from 1 injection to 4 injections are simulated. Injection details are listed in Table 6. Generally, the addition of an injection event leads to a smoother and more isobaric-like in-cylinder pressure trace. For example, limited by the peak pressure of 300 bar, a late injection timing is adopted for the 1 inj. case; this leads to an obvious bump of the in-cylinder profile. For the 2 and 3 inj. cases, earlier injections are added and the bump is eliminated. With 4 injections, a perfect isobaric combustion process is obtained. Besides, the 3 and 4 inj. cases have a leaner mixture distribution during combustion. After about 15°CA the 2 inj. case demonstrates a richer mixture distribution than the 1 inj. case, which is due to the late injection timing employed for the second injection event.

Fig. 14. Comparison of the predicted traces of (a) in-cylinder pressure and HRR, (b) average temperature and mass fraction of $\phi > 1.5$ using various injection strategies at EGR = 0%.

Table 6. Details of various injection strategies at EGR = 0%.

Case	1 inj.	2 inj.	3 inj.	4 inj.
SOI/Mass 1 st inj. (°, mg/cyc)	2/222	1.0/78	-2.2/22.2	0.5/22.2
SOI/Mass 2 nd inj. (°, mg/cyc)	-	6.1/144	0.3/111	2.7/44.4
SOI/Mass 3 rd inj. (°, mg/cyc)	-	-	2.2/88.8	5.6/111
SOI/Mass 4 th inj. (°, mg/cyc)	-	-	-	11.5/44.4

Figure 15 compares the predicted (a) energy percentages and (b) NOx and soot emissions using various injection strategies with an EGR rate of 0%. Various cases yield similar engine efficiencies for their similar combustion phase, which is primarily owing to the oxygen-abundant condition and thus the combustion duration is small. However, although the 2 inj. case yields a richer mixture distribution after 15°CA, as shown in Fig. 16, it only caused a slight efficiency decrease, which can be attributed to its lower combustion temperature and thus the lower heat transfer loss. For the same reason, it generates the lowest NOx emission but the highest soot emission consequently, as shown in Fig. 15(b).

Fig. 15. Comparison of the predicted (a) energy percentages and (b) NOx and soot emissions using various injection strategies at EGR = 0%.

Fig. 16. Predicted distributions of equivalence ratio using various injection strategies at 20°.

4.4. Effect of isobaric combustion ratio

The previous works [19, 21] revealed that the isobaric combustion ratio, defined as the fraction of input energy during the constant-pressure combustion phase, had a significant impact on the engine combustion performance, but these works were mostly based on 1D simulations or running at low peak combustion pressure conditions (maximum at 150 bar). Therefore, to further explore the effect of isobaric combustion ratio on the high-pressure engine combustion performance 3D simulations need to be conducted.

In this subsection, four cases with isobaric combustion ratios of 100%, 85%, 70%, and 55% are simulated. The 3 inj. case in Section 4.3.2 is taken as the baseline case and each case has the same fuel energy input. Besides, to achieve a peak combustion pressure of 300 bar, the intake pressures are adjusted because a smaller isobaric combustion ratio will result in a higher pressure rise. Table 7 shows the P_{IVC} and injection strategy adopted for various cases. Note that in this work, the isobaric combustion ratio is calculated as 100% subtracted by the mass fraction of the first injection, since the first injection was consumed during the isochoric combustion period to achieve the peak combustion pressure of 300 bar.

Table 7. P_{IVC} and injection strategy at various isobaric combustion ratios.

Isobaric combustion ratio (%)	100	85	70	55
P_{IVC} (bar)	6.5	6.2	5.9	5.6
SOI/Mass 1 st inj. (°, mg/cyc)	-0.5/21	-3.0/33	-3.0/71	-4.0/100
SOI/Mass 2 nd inj. (°, mg/cyc)	2.1/44	0.5/22	3.3/151	3.5/122
SOI/Mass 3 rd inj. (°, mg/cyc)	4.9/157	3.5/167	-	-

Figures 17 and 18 show the predicted traces of in-cylinder pressure and HRR and energy percentages using various isobaric combustion ratios, respectively. With the reduction of isobaric combustion ratio, a higher peak for the first-stage heat release rate is generated since more fuel is injected during the early combustion period near TDC; this leads to a higher engine efficiency due to an earlier combustion phase. Besides, the heat transfer loss demonstrates a growing trend but the exhaust loss demonstrates a declining trend. The results prove that an appropriate addition of isochoric combustion is beneficial for the improvement of the HP-cylinder efficiency. However, the impact on the DCEE system needs further investigations in the future, since the change of isobaric combustion ratio also affects the LP-cylinder performance and more optimizations are demanded.

Fig. 17. Comparison of the predicted traces of in-cylinder pressure and HRR at various isobaric combustion ratios.

Fig. 18. Comparison of the predicted energy percentages at various isobaric combustion ratios.

Figure 19 compares the predicted NOx and soot emissions at various isobaric combustion ratios. Both emissions demonstrate a declining trend with the reduction of the isobaric combustion ratio. The reduction of NOx emission is primarily owing to the lower oxygen concentration. However, soot formation is more complicated. To clarify it, Fig. 20 shows the predicted distributions of equivalence ratio and soot on the central plane at 12°, which crank angle represents the turning point when soot formation for each case becomes different. Note that despite the higher oxygen concentration, cases with a higher isobaric combustion ratio still generate the richer air-fuel mixture distribution, which is in favor of soot production. The major cause of it is the latter ending of injection timing. In addition, the late combustion phasing reduces the residual time [for soot oxidation](#).

Fig. 19. Comparison of the predicted (a) NOx and soot emissions and (b) soot evolution profiles at various isobaric combustion ratios.

Fig. 20. Comparison of the predicted distributions of (a) equivalence ratio and soot at 12° at various isobaric combustion ratios.

4.5. Summary

The results of this work indicate that the injection strategy should be modified at various engine speeds to achieve isobaric combustion. A slight introduction of the isochoric combustion is beneficial

for the improvement of fuel economy. Since this work is primarily focused on studying the injection strategy, there are still many other approaches to further improve the HP-cylinder performance, such as the implementation of a multi-injector setup and optimization of piston shape. Additionally, the effects of various parameters on the whole DCEE concept need more systematic studies in the future.

5. Conclusions

The purpose of this work was to numerically investigate the effects of various injection strategies on the combustion and emission characteristics of a high-pressure isobaric combustion engine. Primary engine design parameters such as [EGR and engine speed](#) were varied and evaluated. The major conclusions are summarized as follows.

(1) The modified mathematic model of injection rate is able to well capture the experimentally measured data.

(2) Due to the boosted intake pressure, combustion efficiency is still high even at $\text{EGR} = 60\%$. For the competitive relationship between the higher exhaust loss and lower heat transfer loss with the growth of EGR rate, a peak thermal efficiency of 47.1% is achieved at $\text{EGR} = 50\%$.

(3) The higher engine speed yields a higher thermal efficiency owing to the lower heat transfer loss. The root cause is the shorter cyclic time.

(4) An appropriate reduction of the isobaric combustion heat release is beneficial for the improvement of the HP-cylinder combustion performance. But its effect when integrated with the whole DCEE setup including the LP-cylinder needs further investigations in the future.

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