Exploring Alternative Combustion Control Strategies for Low Load Exhaust Gas Temperature Management of a Heavy-Duty Diesel Engine

Wei Guan[1], Hua Zhao[1], Zhibo Ban[2], Tiejian Lin[2]

([1]Brunel University London, UK; [2]Guangxi Yuchai Machinery Company, China)

Abstract

The employment of aftertreatment systems in modern diesel engines has become indispensable in order to meet the stringent emissions regulations. However, a minimum exhaust gas temperature (EGT) of approximately 200°C must be reached to initiate the emissions control operations. Low load engine operations usually result in relatively low EGT, which lead to reduced or no exhaust emissions conversion. In this context, this study investigated the use of different combustion control strategies to explore the trade-off between EGT, fuel efficiency and exhaust emissions.

The experiments were carried out on a single cylinder common rail heavy-duty diesel engine at a light load of 2.2 bar indicated mean effective pressure. Strategies include the late intake valve closure (LIVC) timing, intake throttling, late injection timing (T_{inj}), lower injection pressure (P_{inj}), and internal exhaust gas recirculation (iEGR) as well as external EGR (eEGR) were investigated. Results showed that the use of eEGR and lower P_{inj} were not effective in increasing EGT. Although the use of late T_{inj} could result in a higher EGT, the delayed combustion phase led to the highest fuel efficiency penalty. Intake throttling and iEGR allowed for an increase in EGT by 42°C and 52°C at the expense of 7.2% and 17% fuel consumption penalties, respectively. In comparison, LIVC strategy achieved the best trade-off between EGT and ISFC, increasing the EGT by 52°C and the fuel consumption penalty by 5.3% while reducing NOx and soot emissions simultaneously. When the IVC timing was delayed to after -107 CAD ATDC, however, the combustion efficiency deteriorated, and hence very high HC and CO emissions. This could be overcome by combining iEGR with LIVC to increase the in-cylinder combustion temperature for a more complete combustion. The results demonstrated that the “LIVC + iEGR” strategy can be the most effective means, increasing the EGT by 62°C with small penalty in the fuel consumption of 4.6% and soot emission reduction by 85%. Meanwhile, maintaining high combustion efficiency as well as low HC and CO emissions of diesel engines.

Keywords

Heavy-duty diesel engine, variable valve actuation, late intake valve closing, internal EGR, exhaust temperature, exhaust emissions

Introduction

Increasingly stringent emissions regulations and fuel prices are driving the development of more efficient internal combustion engines. Diffusion combustion enables conventional diesel engines to operate at a high compression ratio and high thermal efficiency and produce high torque, making diesel engine the dominant powertrain in heavy duty vehicles. However, diesel engines face great challenge when it comes to emissions. The mixing controlled diesel combustion produces a large amount of particulate matter (PM) in the fuel rich burning region and high concentration of nitrogen oxide (NOx) in the high temperature zones [1]. In order to meet the Euro VI or equivalent emission regulations of 0.4 g/kW·h NOx and 0.010 g/kW·h PM

[2], significant reduction of these pollutants by in-cylinder combustion technologies and emission control aftertreatment systems are required.

Advanced combustion modes such as Homogeneous Charge Compression Ignition (HCCI), Pre-mixed Charge Compression Ignition (PCCI), and Low Temperature Combustion (LTC) are able to achieve a reduction in NOx and soot simultaneously, through reducing the peak in-cylinder combustion temperatures and the local equivalence ratios. However, it is still challenging to meet the emission regulations at all operating conditions by using these combustion modes and strategies without employing aftertreatment systems. In addition, low temperature combustion modes generally lead to an significant increase in Carbon Monoxide (CO) and Hydrocarbon (HC) emissions [3–5].

Therefore, it is important to couple complex aftertreatment systems with the diesel engine, in an effort to meet the tailpipe emission standards. Typical aftertreatment systems including Selective Catalytic Reduction (SCR), Diesel Particulate Filter (DPF), and Diesel Oxidation Catalyst (DOC) allow for NOx, PM, and CO and HC emissions reduction accordingly. The conversion efficiency of these aftertreatment systems is strongly dependent on the exhaust gas temperature (EGT). A minimum exhaust gas temperature of approximately 200 °C is required for catalyst light-off and initiate the emissions control [6,7]. This is extremely challenge at low load conditions when the exhaust gas temperature is too low to provide sufficient emissions reduction [8,9]. Therefore, a suitable control strategies to raise exhaust gas temperature while maintaining high engine efficiency at lower loads is significantly important.

In modern diesel engines, advanced technologies, such as high injection pressure, two-stage turbocharger, and multiple fuel injection strategy, have been developed to increase the engine combustion efficiency which is typically accompanied with a decrease in exhaust gas temperature [10]. For this reason, a number of recent studies have explored various techniques to increase exhaust gas temperature. These strategies can be basically divided into three main categories.

The first strategy is by means of air management, which controls the conditions of charge air to increase EGTs, such as intake temperature, pressure and its composition. As the low exhaust temperature at low loads is caused by the higher excess of air, the intake air throttling can be applied to reduce the amount of air for exhaust gas management. Mayer et al. [11] showed that a significant increase in exhaust gas temperature was achieved by throttling the air intake in a turbocharged diesel engine, but with higher NOx emission and combustion noise due to the higher combustion temperature and the rate of heat release, respectively. Honardar et al. [12] also analysed the impact of intake throttling on exhaust gas temperature at 2 bar BMEP under both warm and cold engine operating conditions. An increase in EGTs could be obtained at the expense of higher fuel consumption as well as HC and CO emissions due to a worse combustion efficiency.

The second strategy for the effective exhaust thermal management is controlling the mixture formation by adapting fuel injection parameters, such as main injection timing, post injection, and injection pressure. Honardar et al. [12] experimentally examined the impacts of retarded main injection timing and post injection on exhaust gas temperatures at low load conditions. Both strategies were capable of achieving higher exhaust gas temperatures at the cost of higher fuel consumption due to incomplete combustion and lower expansion work. Different fuel injection strategies were explored to raise the exhaust gas temperature for active regeneration of DPF by Parks et al. [13]. This experiment was performed at a 4-cylinder diesel engine with two different starting DPF temperatures of 150 and 300 °C. Cavina et al. [14] analysed the combined effect of air and fuel paths for thermal management by varying injection timings and VGT position. It was found that a significant increase in exhaust gas temperature was achieved by retarding injection timing while the resultant fuel penalty could be compensated by changing the VGT position.

The last strategy for increasing the exhaust gas temperature is by using external heating measures. Dosing liquid fuel or HC directly in the exhaust line upstream of the DOC device [15][16] and using Electrically Heated Catalyst [17,18] have been used for exhaust thermal management. However, the cost and complexity of the electrically heated catalyst limited their use in large volumes.

Variable valve actuation (VVA) offers an alternative approach to increasing the exhaust gas temperature as well as a number of other benefits. Ratzberger et al. [19] carried out an experimental and numerical study to evaluate the ability of an early exhaust valve opening (EEVO) and a late intake valve closing (LIVC) for exhaust thermal management. An increase in exhaust gas temperature was realized by LIVC, however, the dropped pressure and temperature at the start of combustion led to poor combustion stability and problematic unburnt HC emission. Results also showed that EEVO enabled a distinct increase of exhaust enthalpy at the expense of higher fuel consumption.

Garg et al. [20] investigated the influence of cylinder throttling with early and late intake valve closing timings. Results showed that both delaying and advancing IVC reduced the volumetric efficiency, resulting in a reduction in in-cylinder air mass flow. This contributed to an increase in EGT. The reduction in piston-motion-induced compression resulted in a lower in-cylinder gas temperature and higher degree of premixed combustion, which simultaneously curb NOx and soot formations. Ding et al. [21] evaluated cylinder deactivation during both loaded and lightly loaded idle conditions and concluded that cylinder deactivation improved exhaust thermal management. The higher exhaust gas temperature and lower unburnt HC and CO were attained at low load conditions by introducing iEGR with higher exhaust back pressures [22–24]. Other research into the use of a variable valve actuation to improve exhaust thermal management can be found in [25–27].

The focus of this work is the exploration and direct comparison of different strategies for exhaust gas temperature management and the trade-off with fuel efficiency and emissions. Conventional strategies regarding air and fuel paths such as retarded injection timings, lower diesel injection pressures, intake throttling as well as external EGR will be investigated here as the base line results and for comparison with VVA strategies including LIVC and iEGR. The experimental study was carried out in a single cylinder heavy-duty (HD) common rail fuel injection diesel engine equipped with an intake VVA system. The investigation was conducted at a speed of 1150 rpm and 2.2 bar net indicated mean effective pressure (IMEP) within the area of World Harmonized Stationary Cycle (WHSC) test cycle [28]. The influence of various strategies on the engine combustion, performance, and emissions were analysed and compared.

Experimental setup and Methodology

Experimental setup

The work was conducted on a single-cylinder four-stroke HD diesel research engine equipped with a common rail injection system and coupled to an eddy current dynamometer. The specifications of the engine are given in Table 1 and the schematic of the experimental setup is illustrated in Figure 1.

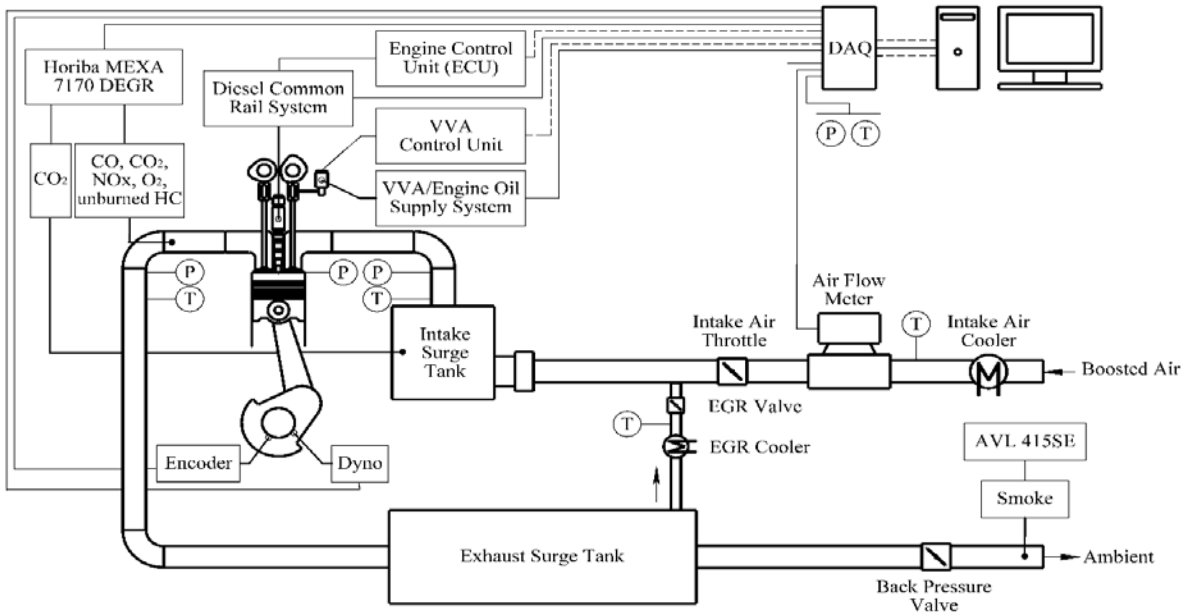

Figure 1. Layout of the engine experimental setup.

Table 1. Specifications of test engine.

Displaced Volume	2026 cm^3
Stroke	155 mm
Bore	129 mm
Connecting Rod Length	256 mm
Geometric Compression Ratio	16.8:1
Number of Valves	4
Piston Type	Re-entrant bowl
Diesel Injection System	Bosch common rail
Nozzle design	8 holes, included spray angle of 150°
Maximum fuel injection pressure	2200 bar
Maximum in-cylinder pressure	180 bar

Intake and Exhaust System

Overhead camshafts were installed on the cylinder head. The intake camshaft of the engine was equipped with a hydraulic lost motion VVA system, in which a hydraulic tappet on the valve side of the rocker arm is incorporated to realize the required valve events such as late intake valve closing and intake valve re-opening. Two large damping chambers were installed in the intake and exhaust systems to damp out the strong pressure fluctuations in the intake and exhaust manifold resulted from the gas exchange dynamics of the engine. The compressed air flow was supplied by an external supercharger with closed loop control. The intake mass flow rate was measured by a thermal mass flow meter. The intake manifold pressure was fine adjusted by means of an electronic intake throttle valve. Two piezo-resistive pressure transducers were installed to measure the instantaneous intake and exhaust ports pressures. The exhaust back pressure was independently controlled through a butterfly valve located downstream of the exhaust surge tank. External EGR flow was achieved via an electronic EGR valve. When the desired EGR rate could not be reached through using the electronic EGR valve alone, higher exhaust back pressure was used to increase the EGR rate. Engine coolant and lubrication oil were supplied externally and their temperature controlled via heaters and coolers.

Fuel Delivery system

In the test bench the diesel fuel was supplied by an electric motor driven high pressure pump to a common rail system at pressure up to 2200 bar, and injected by a solenoid injector. The fuel consumption was measured by two Coriolis flow meters. One was used for measuring the total fuel supplied while the other one was used for measuring the fuel return from the high pressure pump and injector. A bespoke electronic control unit (ECU) was used to control fuel injection parameters such as the injection pressure, injection timing, and the number of injection pulses.

Exhaust Measurement

The gaseous exhaust emissions such as NOx, CO, CO2, and HC were measured by an emission analyzer (Horiba MEXA-7170 DEGR). With the use of the analyser system, gaseous including CO and CO2 were measured through a Non-Dispersive Infrared Absorption (NDIR) analyser, HC was measured by a Flame Ionization Detector (FID), and NOx was measured by a

Chemiluminescence Detector (CLD). To allow for high pressure sampling and avoid condensation, a high pressure sampling module and a heated line were used between the exhaust sampling point and the emission analyzer. The smoke concentration was measured downstream of the back pressure valve by an AVL 415SE Smoke Meter, and thereafter converted from FSN to mg/m3 [29]. All the exhaust gas components were converted to net indicated specific gas emissions according to [2].

In this study, the definition of the EGR rate was defined as the ratio of the measured CO_2 concentration in the intake surge tank to the CO_2 concentration in the exhaust manifold as

$$EGR\ rate\ = \frac{(CO_2\%)_{intake}}{(CO_2\%)_{exhaust}} * 100\% \quad (1)$$

where the $(CO_2\%)_{intake}$ and $(CO_2\%)_{exhaust}$ are the CO_2 concentration in the intake and exhaust manifolds, respectively.

Experiment Data Processing

A piezo-electric pressure transducer with the sampling revolution of 0.5 deg CAD was mounted in the cylinder for measuring the instantaneous in-cylinder pressure recorded by an AVL Amplifier. The in-cylinder pressure data was averaged over 200 engine cycles and used to calculate the apparent Heat Release Rate (HRR) and combustion characteristics. According to [30], the apparent HRR was calculated as

$$HRR = \frac{\gamma}{(\gamma-1)}\, p\, \frac{dV}{dt} + \frac{1}{(\gamma-1)}\, V\, \frac{dp}{dt} \qquad (2)$$

where γ is defined as the ratio of specific heats; V and p are the in-cylinder volume and pressure, respectively; t is the time.

In this study, the CA10 and CA50 (Combustion Phasing) were defined as the crank angle when the mass fraction burned reached 10% and 50%, respectively. Ignition delay was defined as the crank angle between the start of injection and the start of combustion. The combustion stability was measured by the coefficient of variation of the net IMEP (COV_IMEP) over the sampled cycles. A displacement sensor mounted on the top of the intake valve spring retainer was used to record the intake valve lift continuously.

Methodology

The study was carried out at a light load of 2.2 bar IMEP, which represents one of typical low exhaust gas temperatures operating conditions of a HD drive cycle and is located within the area of the WHSC test cycle, as shown in Figure 2.

During the test, the coolant and oil temperatures were kept within at 80 ± 2 °C and oil pressure was maintained within 4.0 ± 0.1 bar. The average pressure rise rate and COV_IMEP were limited to below 20 bar/CAD and 5%, respectively. The intake valve opening (IVO) and closing (IVC) timings of baseline case were set at 367 and 545 CAD after top dead centre (ATDC) while the exhaust valve timings were fixed over the experiments. The VVA system also enables a second opening event of intake valve (2IVO) during the exhaust stroke to trap residual gas. The maximum lifts of intake valve and 2IVO event were 14mm and 2mm, respectively. Figure 3 shows the intake and exhaust valve profiles for the baseline as well as the late IVC strategy and 2IVO event. All valve opening and closing events in this study were considered at 1mm valve lift. Additionally, the pressure based effective compression ratio was defined as

$$ECR = \frac{V_{ivc\_eff}}{V_{tdc}} \qquad (3)$$

where V_{tdc} is the in-cylinder volume at TDC position, and V_{ivc_eff} is the effective in-cylinder volume where the in-cylinder gas pressure is equivalent to the intake manifold pressure, rather than the in-cylinder volume at IVC [31].

In addition, the mean in-cylinder gas temperature (T_m) was calculated using the ideal gas state equation

$$PV = mRT_m \qquad (4)$$

where P and V are the in-cylinder pressure and volume, respectively; m is the mass of charge and R is the specific gas constant.

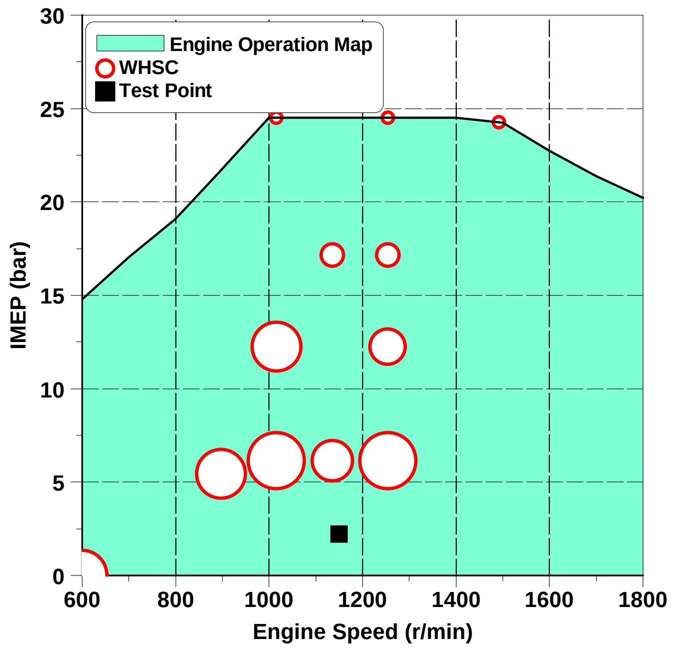

Figure 2. Test point and WHSC operation conditions.

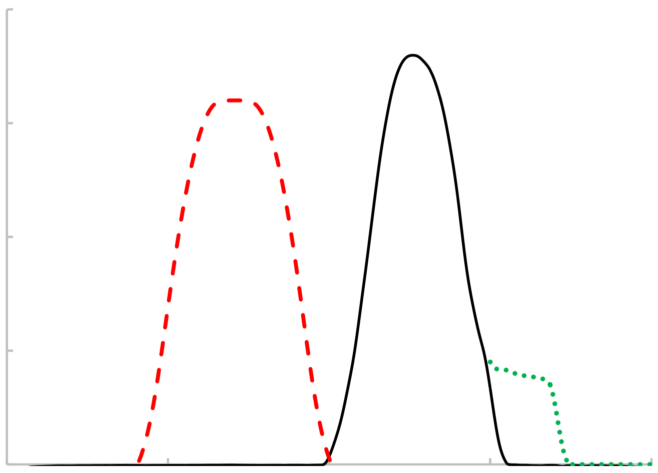

Figure 3. Fixed exhaust camshaft timing and variable intake valve lift profiles with VVA.

Results and discussion

The experimental results were divided into three subsections. In the first section, the impact of different control strategies to increasing exhaust gas temperatures, such as LIVC, intake throttling, late injection timing (T_{inj}), lower injection pressure (Pinj), iEGR, and eEGR are presented and analysed. This is followed by an analysis and comparison of the most effective strategies from the first section. Finally, the combined LIVC with iEGR strategy is investigated for its potential to overcoming the negative effect of LIVC strategy on CO and HC emissions.

Evaluation of the effectiveness of various strategies for increasing EGT

In this part, different strategies are explored with respect to their capability to increase EGTs and impact on the relative fuel efficiency penalty, from which the efficient EGT control strategies are selected for further analysis. Table 2 summarises the engine operating conditions and the matrix of test cases of all strategies.

Table 2. Main operation conditions of various control strategies.

Testing modes	Case number	Speed r/min	Load bar	T_{inj} CAD ATDC	P_{inj} bar	eEGR %	Int_pressure bar	Exh_pressure bar	2IVC	IVC CAD ATDC	ECR
Baseline	1	1150	2.2	-5.8	515	0	1.15	1.20	off	-178	16.8
LIVC	2			-5.8	515	0	1.15	1.20	off	**-136**	15.3
	3			-5.8	515	0	1.15	1.20	off	**-118**	13.5
	4			-5.8	515	0	1.15	1.20	off	**-107**	12.3
	5			-5.8	515	0	1.15	1.20	off	**-100**	11.4
Intake throttling	6			-5.8	515	0	**1.08**	1.20	off	-172	16.8
	7			-5.8	515	0	**1.03**	1.20	off	-172	16.8
	8			-5.8	515	0	**0.98**	1.20	off	-172	16.8
iEGR	9			-5.8	515	0	1.15	**1.20**	**on**	-172	16.8
	10			-5.8	515	0	1.15	**1.27**	**on**	-172	16.8
	11			-5.8	515	0	1.15	**1.33**	**on**	-172	16.8
	12			-5.8	515	0	1.15	**1.42**	**on**	-172	16.8
Late T_{inj}	13			**-2**	515	0	1.15	1.20	off	-178	16.8
	14			**1**	515	0	1.15	1.20	off	-178	16.8
	15			**2**	515	0	1.15	1.20	off	-178	16.8
Low P_{inj}	16			-5.8	**360**	0	1.15	1.20	off	-178	16.8
	17			-5.8	**300**	0	1.15	1.20	off	-178	16.8
eEGR	18			-5.8	515	**16**	1.15	1.20	off	-178	16.8
	19			-5.8	515	**31**	1.15	1.20	off	-178	16.8

Figure 4 shows an overview of the variations in fuel consumption versus exhaust gas temperatures when different control strategies were applied. The variation value in both ISFC and EGT is defined as the difference between the other test cases and the baseline operation depicted in Table 2. It illustrated that exhaust gas temperatures can be effectively increased by LIVC, intake throttling, or iEGR but much less affected by the injection pressure and eEGR rate. By delaying combustion phase through the late injection timings, higher EGT was observed but with highest fuel efficiency penalty. These behaviours were supported by Bai et al. [9] carrying out an experimental study at medium and low loads to investigating the effects of injection advance angle and injection pressure on exhaust thermal management. Therefore, strategies including late injection timings, lower injection pressures, and increased eEGR rates have been excluded from further analysis in the next subsection.

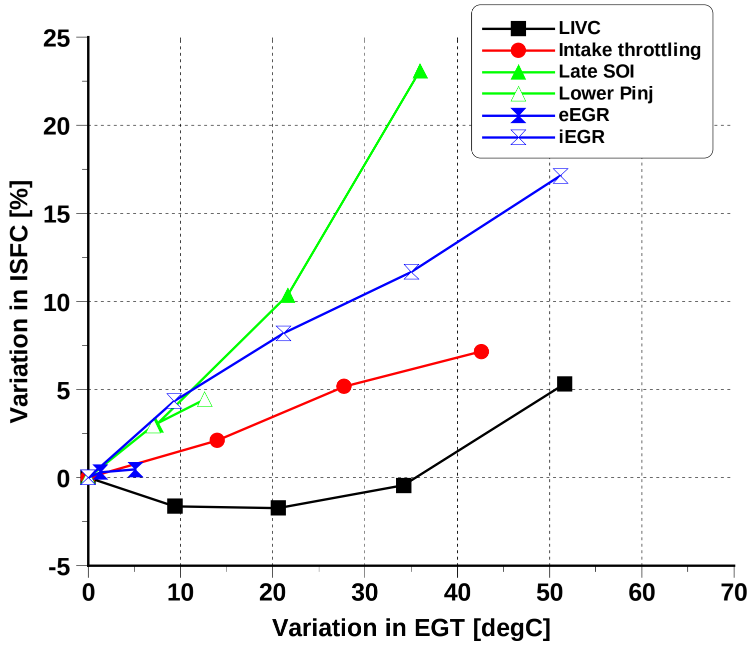

Figure 4. Potential of different strategies in increasing EGT.

Comparison of the LIVC, intake throttling, and iEGR strategies

In this section, the three effective EGT control strategies of LIVC, intake throttling, and iEGR are further analysed and compared to the baseline case.

As shown in Figure 3 and Table 2, the original valve lift profile was used in a baseline case and the corresponding IVC timing was -178 CAD ATDC. The intake manifold air pressure and exhaust back pressure were set to 1.15 bar and 1.20 bar, respectively. The LIVC strategy was run with four different intake valve closure timings from -178 CAD to -100 CAD ATDC. For the intake throttling strategy, the intake air pressure was gradually reduced from 1.15 bar to 0.98 bar while keeping the exhaust pressure constant at 1.20 bar. In the case of iEGR strategy, the intake air pressure was set at 1.15 bar while the exhaust back pressures of 1.27bar, 1.33 bar, and 1.42 bar were performed by using the exhaust back pressure valve in order to trap higher fraction of the residual exhaust gas.

The Effect of LIVC, intake throttling, and iEGR strategies on combustion process

Figure 5 shows the in-cylinder pressures of the baseline, Cases 2 and 5 of the LIVC, Cases 6 and 8 of the intake throttling, and Cases 9 and 12 of the iEGR strategies. It can be seen that both LIVC and intake throttling strategies reduced in-cylinder pressures due to the lower effective compression ratio and reduced intake air pressure respectively, especially at the Case 5 of LIVC strategy and Case 8 of intake throttling strategy. The use of iEGR strategy via 2IVO event showed less impact on the in-cylinder pressure, even with higher exhaust back pressure of 1.42 bar in Case 12. This behaviour was the result of the hot residuals trapped, which increased the in-cylinder gas temperature and hence the compression pressure and temperature. The hot residuals also accelerated the fuel evaporation and combustion process, resulting in a similar in-cylinder pressure curve to that of the baseline case.

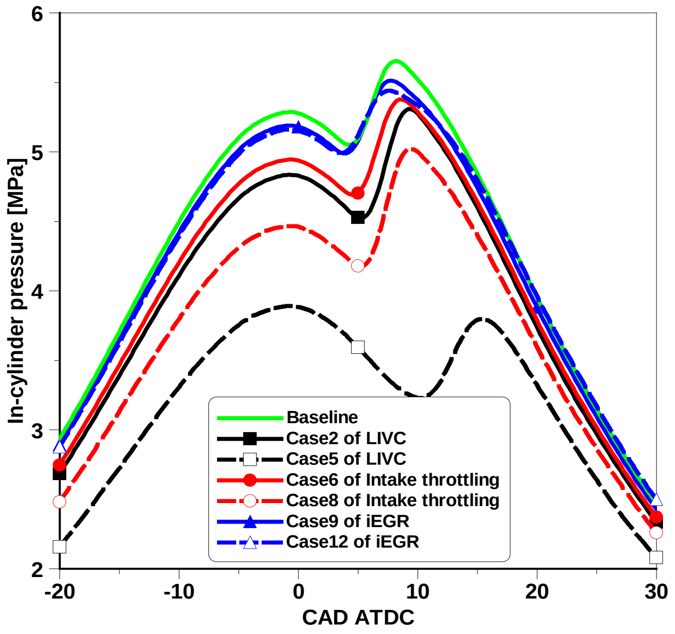

Figure 5. In-cylinder pressures for various strategies.

Figure 6 shows the heat release rate (HRR) of the baseline case, Cases 2 and 5 of the LIVC, Cases 6 and 8 of the intake throttling, and Cases 9 and 12 of the iEGR strategies. The LIVC strategy was characterised by longer ignition delay and higher degree of premixed combustion. The highest peak heat release rate was obtained in Case 2 of the LIVC strategy, but it declined to the same level of the baseline case when delaying IVC timing to Case 5. This was because the lower in-cylinder temperature and pressure allowed more time for mixture preparation before autoignition, and hence higher degree of premixed combustion. The longer ignition delay in Case 5 of the LIVC strategy shifted the combustion process further away from TDC, resulting in lower burned gas temperature and peak HRR. In comparison, the peak HRR increased greatly as the intake throttling strategy was employed from Case 6 to Case 8. This was a result of the reduced in-cylinder charge density caused by the lower intake air pressure, resulting in a higher peak mean in-cylinder gas temperature, as shown in the next section. Thus, the combustion rate was accelerated with hotter combustion process [24].

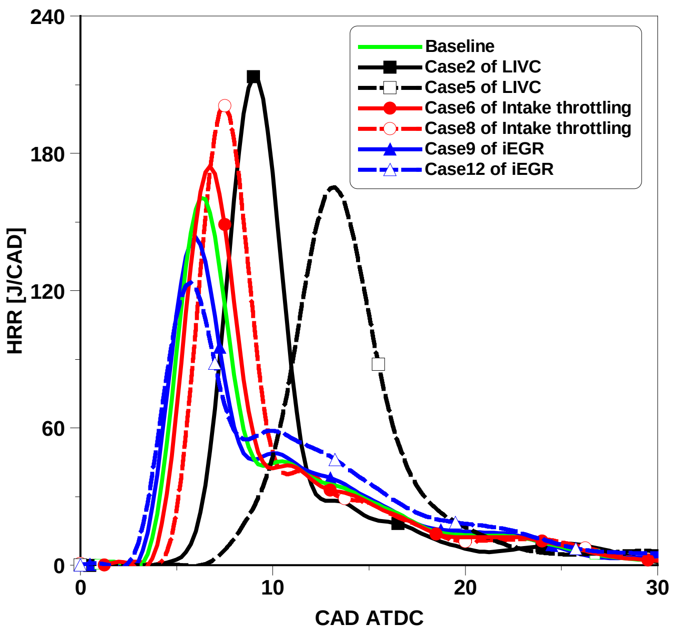

Figure 6. Heat release rate for various strategies.

In contrast to the LIVC and intake throttling strategies, the use of iEGR strategy decreased the peak HRR, in particular in Case 12 with higher fraction of the residuals gas. The reason could be explained by the higher in-cylinder charge temperature resulted from the hot residuals. This reduced the ignition delay and caused the combustion happened earlier than a baseline case. Figure 7 shows the in-cylinder charge temperatures (T_m) calculated by the ideal gas state equation. It can be seen that the compression temperature was reduced by using LIVC strategy due to lower effective compression ratio. However, it increased with iEGR strategy due to the hot residuals. The highest T_m was achieved in Case 12 of the iEGR strategy due to higher fraction of residuals gas.

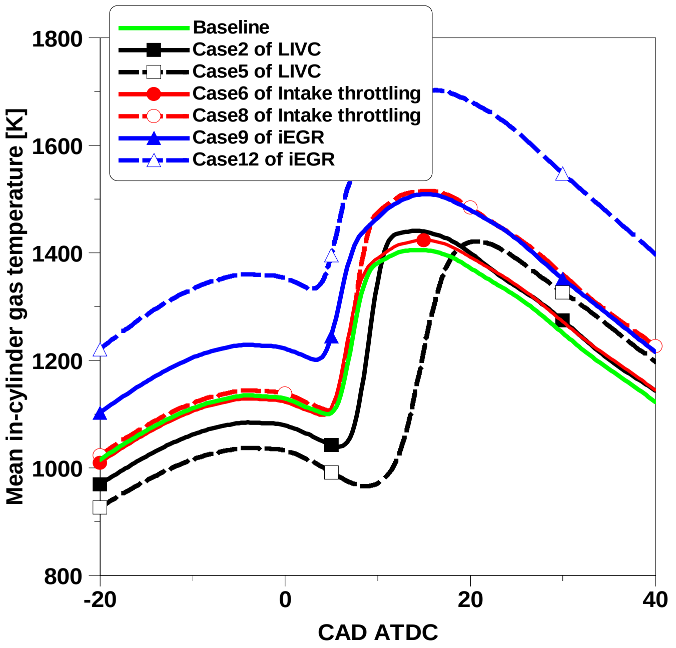

Figure 7. Mean In-cylinder gas temperatures for various strategies.

In the case of the intake throttling, the peak T_m increased as the intake air pressure was reduced via intake throttling, although the compression temperature is similar to that of the baseline case. For the LIVC strategy, however, the peak T_m increased initially with Case 2 of the LIVC strategy, but dropped as IVC timings were further retarded to Case 5 due to the aggressively reduced compression temperature and pressure and later combustion process.

The effect of LIVC, intake throttling, and iEGR strategies on combustion characteristics

Figure 8 shows log P-V diagrams of the baseline case, Case 5 of the LIVC strategy, Case 8 of the intake throttling strategy, and Case 12 of the iEGR strategy at the same injection timing and injection pressure. There was a very small difference in the pumping loop area between the baseline case and LIVC strategy as the intake and exhaust pressures were maintained. However, the iEGR and intake throttling operations were characterised with significantly increased pumping loop areas due to the higher exhaust back pressure and lower intake manifold pressure, respectively.

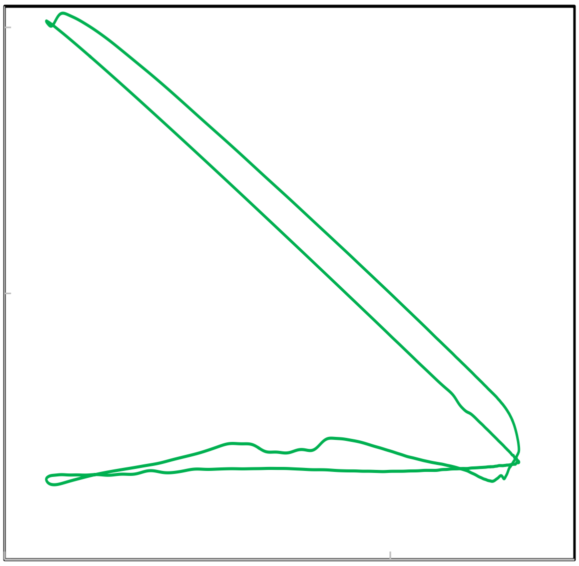

Figure 8. Log P-V diagrams of the baseline case, Case 5 of the LIVC strategy, Case 8 of the intake throttling strategy, and Case 12 of the iEGR strategy.

Figure 9 shows the combustion characteristics of different strategies versus the variation in exhaust gas temperatures. For the LIVC strategy, the reduced effective compression ratio increased the ignition delay and lead to the shortest combustion duration. The iEGR strategy was characterised with the shortest ignition delay due to the charge heating effect of the hot residual gas but slowed down the initial heat release rate and longer combustion period because of the dilution effect of residual gas. Finally, as a result of reduced air mass, the intake throttling strategy led to fastest initial combustion as measured by CA10-CA50 and slightly shorter combustion duration than the iEGR operation.

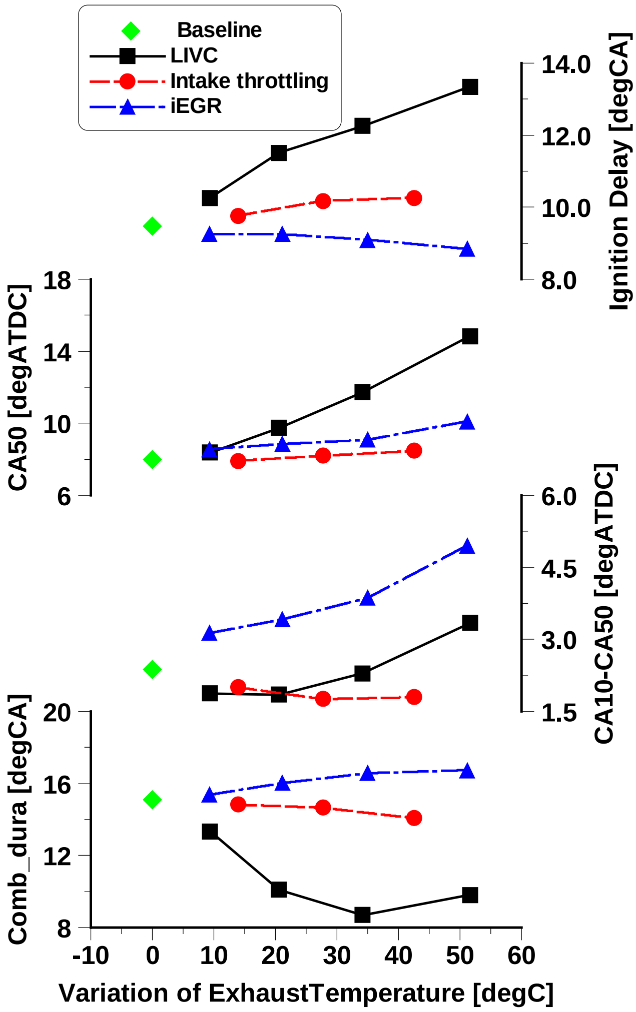

Figure 9. Comparison of the combustion characteristics at different strategies.

The effect of LIVC, intake throttling, and iEGR strategies on engine performance and emissions

It is known that the exhaust gas temperature is mainly determined by the total in-cylinder charge which can be ascertained by the measured excess air ratio (lambda) in the exhaust. Figure 10 shows the lambda of different strategies versus the variation of exhaust gas temperatures for the four cases. It can be seen that the EGT increased linearly with a decrease in lambda as reported by Garg et al. [9] irrespective of the strategy used. The slight deviation of the iEGR operation in the lambda value can be explained by the dilution effect of residual gas.

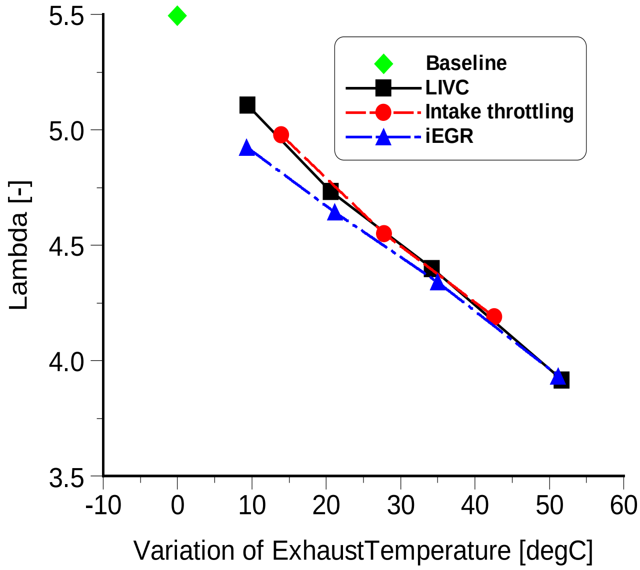

Figure 10. Comparison of the lambda at different strategies.

Figure 11 shows the combustion efficiency, pumping mean effective pressure (PMEP), and variation of fuel consumption of different strategies. The combustion efficiency dropped significantly as the IVC timing was delayed from -178 to -100 CAD ATDC due to the lower compression pressure and the retarded combustion timing, causing the work loss during the combustion stroke. Higher PMEP was found in the strategies of intake throttling and iEGR, which can be explained by the larger pumping loop area in the log P-V diagram as shown in Figure 8, and hence higher fuel consumption. The changes in fuel consumption of LIVC strategy was different from the others. The fuel consumption increased slightly with the retarded LIVC due to the higher degree of premixed combustion and shorter combustion duration. But as the IVC timing was further retarded, the fuel consumption went up more rapidly because of the poor combustion efficiency. In order to increase the exhaust gas temperature by 52 °C, the LIVC and iEGR strategies were accompanied by 5.3% and 17% penalties in ISFC, respectively.

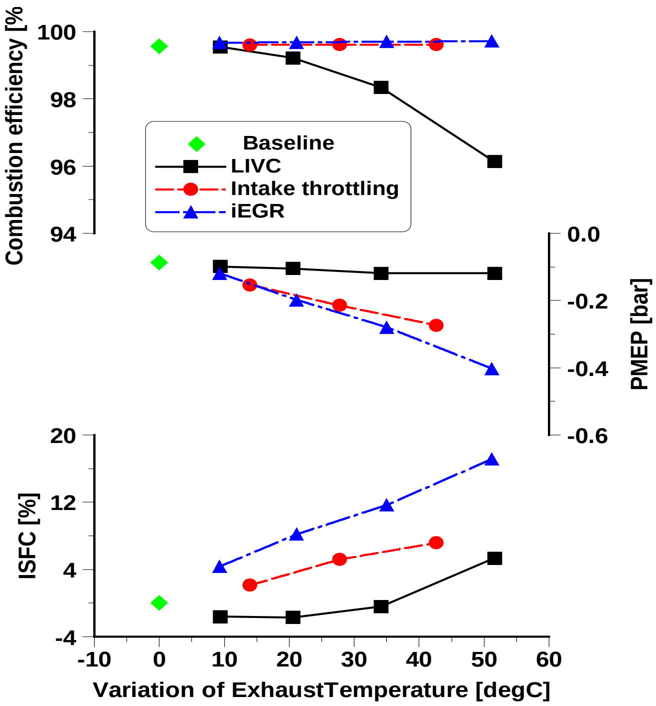

Figure 11. Comparison of the combustion efficiency, PMEP, and ISFC at different strategies.

Figure 12 shows the engine-out emissions versus the variation of exhaust gas temperatures. The use of intake throttling and iEGR strategies had less impact on CO and HC emissions, maintaining as low as the baseline case. However, significant increases in CO and HC emissions were observed in the operation of LIVC strategy when the combustion temperature was much lower because of the lower compression ratio. This was because the CO and HC emissions are mainly affected by the local oxygen availability during combustion and the combustion temperature. The increased combustion temperature by means of intake throttling and iEGR strategies contributed to the low levels of CO and HC emissions.

The change in soot emissions showed a strong correlation with the variation of the ignition delay. The soot emissions decreased in both LIVC and intake throttling strategies because of the prolonged ignition delay, and it increased by iEGR due to the shorten ignition delay. The NOx emissions demonstrated a different trend from that of soot emissions. With the intake throttling, NOx emission increased due to the higher combustion temperature but decreased by iEGR due to the higher dilution and heat capacity of residual gas.

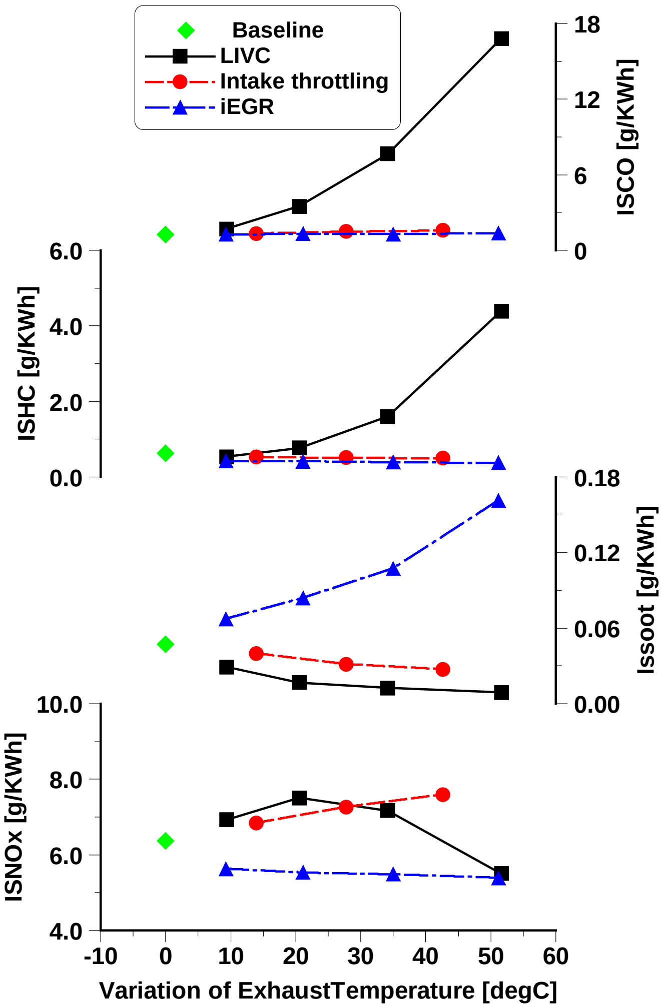

Figure 12. Comparison of the indicated specific emissions at different strategies.

Interestingly, the NOx emission did not change linearly with LIVC strategy. It increased initially due to higher combustion temperature caused by the reduced charge mass and then decreased when the IVC timing was delayed to beyond -118 CAD ATDC when the combustion took place later and experienced lower combustion and pressure.

Combined effects of LIVC and iEGR

According to the discussion and analysis in the above subsection, the LIVC strategy has been demonstrated as an enabling technology for efficient increase in exhaust gas temperatures while maintaining reasonable fuel consumption penalty compared to others. However, a significant increase in CO and HC emissions limit the potential of LIVC strategy. Internal EGR, as analysed in former section, was an effective means in curbing CO and HC emissions. Therefore, the iEGR strategy was introduced when operating with LIVC strategy in order to offset the negative effects of LIVC on CO and HC emissions.

Figure 13 shows that the addition of iEGR to the LIVC operation advanced the combustion timing and increased the peak HRR because the in-cylinder gas temperature was increased by the presence of hot residual gas as shown in Figure 14. Both “LIVC-only” and “LIVC + iEGR” operations were characterised with much lower cylinder pressure than the baseline operation.

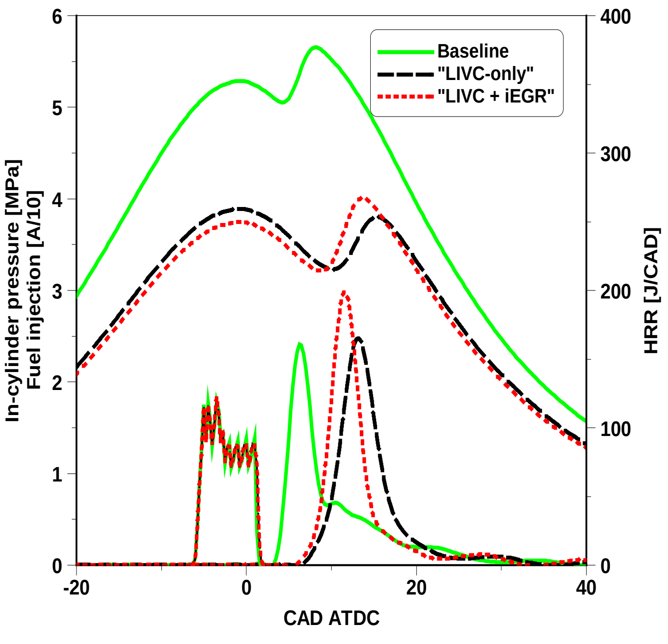

Figure 13. In-cylinder pressure, HRR, and injection signal of a baseline case, Case 5 of LIVC strategy, and Case 9 of “LIVC + iEGR” strategy.

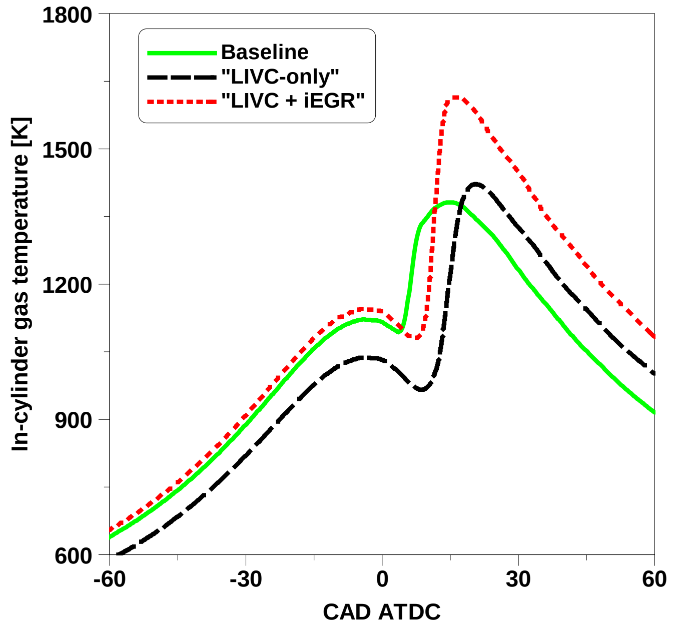

Figure 14. Mean in-cylinder gas temperature of a baseline case, Case 5 of LIVC strategy, and Case 9 of "LIVC + iEGR" strategy.

As shown in Figure 15, the ignition delay is slightly reduced when combining LIVC with iEGR due to the advanced combustion phasing. The combustion duration of "LIVC + iEGR" strategy was longer initially, and then reduced to a similar level of "LIVC-only" strategy when the IVC timing was further delayed.

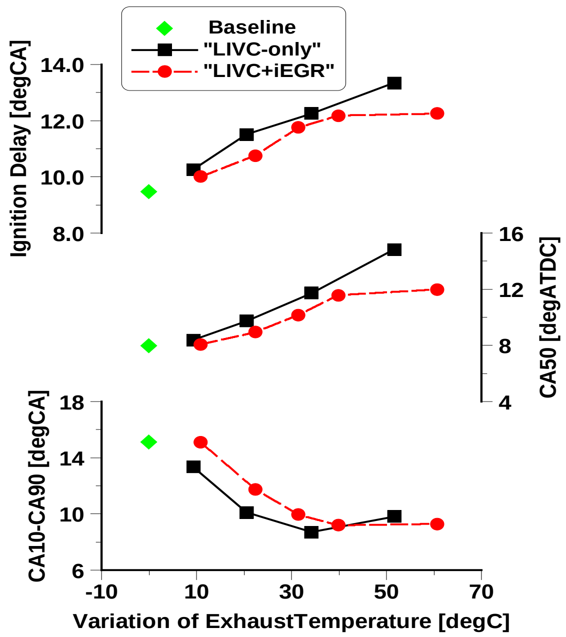

Figure 15. Combustion characteristics of a baseline case, Case 5 of LIVC strategy, and Case 9 of "LIVC + iEGR" strategy.

Figure 16 shows the relationship between lambda and EGT was the same for “LIVC-only” strategy and “LIVC + iEGR” strategy. The lambda was maintained at a similar level when achieving the same variation in EGT by using LIVC strategies with and without iEGR. Compare to “LIVC-only” strategy, the combustion efficiency was clearly improved in “LIVC + iEGR” strategy due to the increased combustion temperature. The higher fuel efficiency penalty of the “LIVC + iEGR” strategy with small EGT variation was due to the longer combustion duration. With further delayed IVC timings the combustion efficiency became higher and reduced the fuel consumption penalty. It is noted that an increase in EGT more than 62 °C was obtained with only 4.6% fuel penalty when operating LIVC with iEGR strategy.

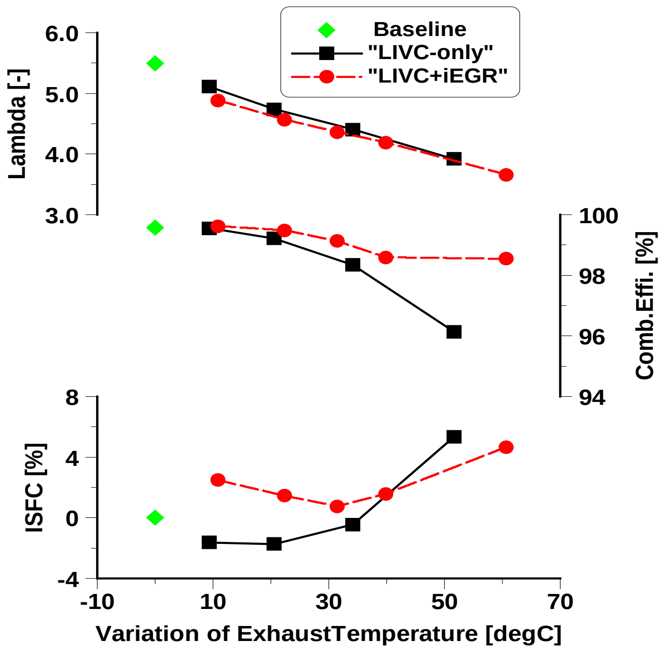

Figure 16. Engine performance of a baseline case, Case 5 of LIVC strategy, and Case 9 of “LIVC + iEGR” strategy.

As shown in Figure 17, the introduction of iEGR reduced both CO and HC emissions, especially when large EGT increase was needed. This was mainly due to the higher in-cylinder combustion temperature, which helped the oxidation of CO and HC emissions. As it did not change the ignition delay clearly, the iEGR had little impact on soot emission. The dilution and heat capacity effects of iEGR caused the lower NOx emissions of "LIVC + iEGR" than those of "LIVC-only" in most cases, other than the highest EGT increase operations during which combustion temperature and hence NOx formation were reduced due to the most retarded combustion after TDC.

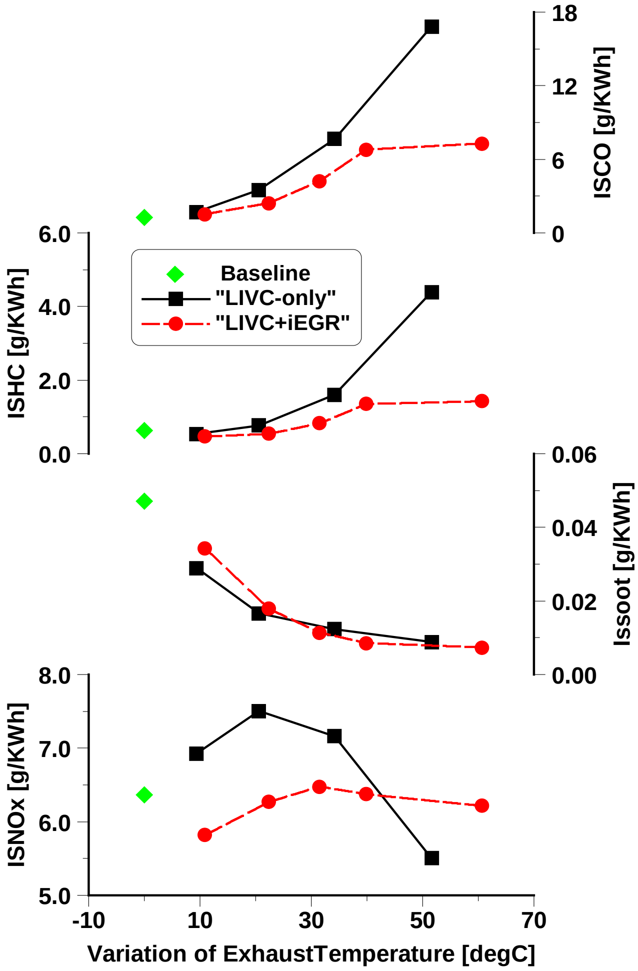

Figure 17. Indicated specific emissions of a baseline case, Case 5 of LIVC strategy, and Case 9 of "LIVC + iEGR" strategy.

Figure 18 provides an overall assessment of the potential of combined LIVC and iEGR strategy to achieve the best trade-off between the EGT, fuel consumption, and emissions. As shown in Figure 18, the results of the optimum "LIVC-only" and "LIVC + iEGR" operations and baseline operation were compared.

It can be seen that "LIVC-only" strategy could increase EGT by 52 °C (31%) at the expense of a 5.3% penalty in fuel consumption. By combining LIVC and iEGR, EGT could be increased by 62 °C (37%) with 4.6% fuel efficiency penalty and much lower HC and CO emissions than the "LIVC-only" operation. Compared to the baseline operation, the soot emission was reduced substantially by LIVC operation with or without the iEGR, accompanied with a small decrease in NOx emissions.

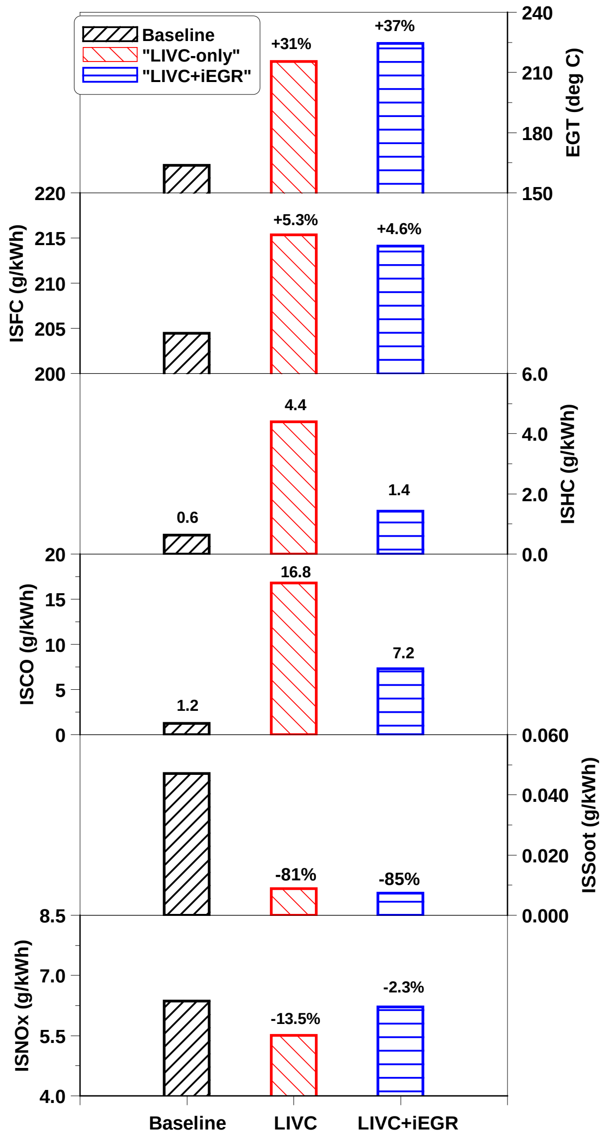

Figure 18. Comparison between experimental results for "Case 5 of LIVC" strategy and "Case 5 of LIVC + Case 9 of iEGR" strategy.

Therefore, the combination of LIVC and iEGR was identified as the most effective means amongst the various technologies examined in the current study to raising the exhaust gas temperature with lower engine-out emissions and minimum penalty in fuel consumption.

Conclusions

Experimental studies were carried out to explore different control strategies for increasing exhaust gas temperatures and analyse the impact on fuel consumption and emissions at a light load condition. The experiments were performed on a single cylinder heavy-duty diesel engine with a common rail fuel injection system. The engine was equipped with a VVA system on the intake camshaft for the application of the LIVC strategy and the introduction of iEGR. The main findings can be summarized as follows:

1. Sufficient high EGT could be obtained by means of the LIVC, intake throttling, and iEGR strategies. Reduced diesel injection pressure and external EGR were not effective in increasing EGT. Retarded diesel injection timing could be used to raise EGT but at the expense of high penalty in fuel consumption.
2. The use of iEGR kept the specific emissions of NOx, HC and CO as low as baseline case, but resulted in large increase in fuel consumption and more soot emission, because of higher pumping work and shorten ignition delay.
3. Throttling intake air flow was effective in increasing the exhaust gas temperature by 42°C due to reduced cylinder charge but accompanied with a 7.2% fuel consumption penalty and slightly higher NOx emission.
4. The application of LIVC enabled an increase in exhaust gas temperature by 52°C and lower NOx and soot emissions with small fuel consumption penalty of 5.3%. However,

the lower combustion temperature led to large increase in HC and CO emissions due to a lower combustion efficiency.

5. The combined use of LIVC and iEGR was identified as the most effective means amongst the various technologies examined in the current study, increasing the exhaust gas temperature by 62°C with lower engine-out emissions and minimum penalty of 4.6% in fuel consumption.

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Contact information

Wei Guan
Wei.guan@brunel.ac.uk
gwei916@163.com
Centre for Advanced Powertrain and Fuels Research
College of Engineering, Design and Physical Sciences
Brunel University London
Kingston Lane
Uxbridge
Middlesex UB8 3PH
United Kingdom

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Definitions/Abbreviations

ATDC	After Firing Top Dead Center.

CA10-CA90	Combustion Duration.
CAD	Crank Angle Degree.
CO	Carbon Monoxide.
CO_2	Carbon Dioxide.
DOC	Diesel Oxidation Catalyst.
DPF	Diesel Particulate Filter.
ECR	Effective Compression Ratio.
ECU	Electronic Control Unit.
EGR	Exhaust Gas Recirculation.
EGT	Exhaust Gas Temperature.
HRR	Heat Release Rate.
HC	Hydrocarbons.
IMEP	Indicated Mean Effective Pressure.
IVC	Intake Valve Closing.
IVO	Intake Valve Opening.
ISFC	Net Indicated Specific Fuel Consumption.
ISSoot	Net Indicated Specific Emissions of Soot.
ISNOx	Net Indicated Specific Emissions of NOx.
ISCO	Net Indicated Specific Emissions of CO.
ISHC	Net Indicated Specific Emissions of Unburned HC.
NOx	Nitrogen Oxides.
SCR	Selective Catalytic Reduction.
TDC	Firing Top Dead Centre.
VVA	Variable Valve Actuation
WHSC	World Harmonized Stationary Cycle.

Dear Organizers and Reviewers,

Thank you for your kind comments and suggestions to the manuscript. We have modified the manuscript accordingly, and detailed corrections are listed below point by point. The paragraphs in black are the reviewers' comments, while our responses are listed in blue. All the modifications are highlighted in red.

We look forward to hearing from you.

Sincerely,

Wei Guan

Brunel University London

Reviewer #1:

1) The biggest issue with this paper is that the experiments were completed with a single cylinder engine. As a result, the authors must provide assurances that the intake and exhaust manifold pressures reasonably approximate what would occur in a turbo-charged Diesel engine. They do not do so. As an example, the intake and exhaust pressures implemented (per table 2) are not justified.

Thanks for the kind suggestion. In this paper, the initial set values for intake and exhaust manifold pressures are taken from Yuchai YC-6K multi-cylinder diesel engine.

2) Motivation for the 2.2 bar, 1150 rpm operating point must be given.

The motivation for testing at 1150 rpm, 2.2 bar IMEP is mainly because the exhaust gas temperature of this test point is below 200 °C. This represents one of typical low exhaust gas temperatures operating conditions of a heavy-duty drive cycle. However, a minimum exhaust gas temperature of approximately 200°C must be reached to initiate the emissions control operations. In addition, this operating point is located within the area of the world harmonized stationary cycle (WHSC) test cycle.

3) The abstract should clearly state that the experimental results are from a single cylinder engine.

Thanks, this has been revised in the abstract on Page 1 as follows:
"The experiments were carried out on a single cylinder common rail heavy-duty diesel engine at a light load of 2.2 bar indicated mean effective pressure."

4) The abstract and conclusions should both include key quantitative findings.

The key quantitative findings have been added in the abstract and conclusions on the pages 1, 28 and 29 accordingly.

5) Reference(s) should be added for effective compression ratio.

Thanks for the kind suggestion. The reference "[31]" for effective compression ratio has been added on page 11.
"Estimation of effective compression ratio for engines utilizing flexible intake valve actuation."