

Late Intake Valve Closing as an Emissions Control Strategy at Tier 2 Bin 5 Engine-Out NO_x Level

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ABSTRACT

A fully flexible valve actuation (FFVA) system was developed for a single cylinder research engine to investigate high efficiency clean combustion (HECC) in a diesel engine. The main objectives of the study were to examine the emissions, performance, and combustion characteristics of the engine using late intake valve closing (LIVC) to determine the benefits and limitations of this strategy to meet Tier 2 Bin 5 NO_x requirements without after-treatment.

The most significant benefit of LIVC is a reduction in particulates due to the longer ignition delay time and a subsequent reduction in local fuel rich combustion zones. More than a 95% reduction in particulates was observed at some operating conditions. Combustion noise was also reduced at low and medium loads due to slower heat release. Although it is difficult to assess the fuel economy benefits of LIVC using a single cylinder engine, LIVC shows the potential to improve the fuel economy through several approaches. First, because of the lower noise and smoke emissions, LIVC offers a promising strategy to expand early premixed charge compression ignition (PCCI) operation to higher load ranges. Second, LIVC reduces CO emissions and improves combustion efficiency at low loads by optimizing the fuel injection timing. The current results show about 30%-50% NO_x reduction is achievable depending on the operating conditions and injection strategies if the combustion phasing is fixed. For the same NO_x emissions level, LIVC reduces the EGR requirement by 15-25%. A NO_x emissions model in Arrhenius form was developed based on the FFVA experimental results. The model does a good job correlating engine-out NO_x with intake valve closing time, EGR rate, engine speed, A/F ratio, and combustion phasing. The correlation is used to explain the impact of LIVC on NO_x reduction.

INTRODUCTION

With increasing crude oil prices and more stringent emission requirements, developing high efficiency clean combustion technology is one of the major tasks of combustion research. Although researchers world wide have made significant progress in developing control strategies and expanding operating ranges to realize homogenous charge compression ignition (HCCI) in an internal combustion engine, the applications of HCCI are still limited to low loads. Diesel engines, typically achieve about 30% higher efficiency than their gasoline counterparts, are an excellent short-to-mid term solution for minimizing CO₂ emissions and reducing our dependency on petroleum.

Diesel engines in the U.S face tough challenges to meet stringent Tier 2 Bin 5 emission regulations. The major constituents of the 2010 mandate are a 0.07 g/mi limit on NO_x, a 0.01 g/mi limit on PM, and a 0.018 g/mi limit on HCHO [1]. In Europe, similar emission regulations will phase in starting in 2012. These emission requirements bring to light the major objectives of diesel engine research: decreasing NO_x, PM, and HC emissions without compromising fuel economy. Numerous technologies have been developed to decrease pollutant emissions, which can be classified into two major categories: in-cylinder pollutant reduction and after-treatment. Various after-treatment systems have been proposed to decrease tailpipe emissions, such as diesel particulate filters (DPF), Urea-SCR (selective catalytic reduction), diesel oxidation catalysts (DOC), and lean NO_x traps (LNT). Although these after-treatment systems provide realistic solutions to meet the emission requirements, they add to production and operation costs. DPFs need periodic regeneration which increases fuel consumption and control complexity. LNTs use expensive platinum alloys as a catalyst, which significantly increases the capital cost. Urea SCR requires a second fluid fill in the vehicle. Thus, a primary target of the next generation diesel engine is to decrease engine-out pollution emissions.

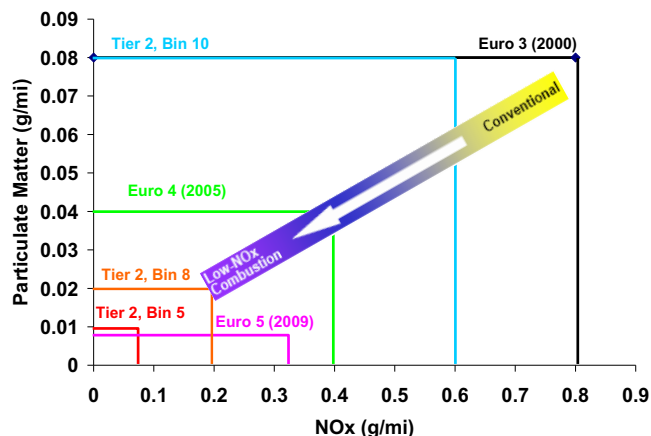


Figure 1. NO_x and Particulate emission standards in the U.S and Europe.

Premixed charge compression ignition (PCCI) is an attractive strategy to simultaneously reduce NO_x and PM emissions in a diesel engine. In contrast to conventional diesel combustion, PCCI combustion uses a higher amount of cooled EGR to decrease the peak combustion temperature and prolong the ignition delay. Since NO_x production is dominated by the peak combustion temperature, lower combustion temperature reduces the NO_x formation. Meanwhile, longer ignition delay gives more mixing time before auto-ignition, leading to more highly premixed combustion. Since the local fuel rich regions decrease, PM emissions are reduced. Although research on PCCI combustion has been progressing since 1995 [2], several practical challenges limit its application under medium to high load conditions. First, the high heat release rate caused by premixed combustion generates high combustion noise. Second, PCCI combustion is prone to misfire at light loads near idling which causes the deterioration of thermal efficiency.

The work described in this paper deals with the technical feasibility of using late intake valve closing (LIVC) to improve the performance and emissions of a diesel engine operating at T2B5 NO_x emission levels.

The application and benefits of variable valve actuation (VVA) in gasoline engines are well known and have been thoroughly investigated [3,4]. VVA systems have been very successfully used on gasoline engines to reduce pumping losses. Also, they have been widely used to control the ignition timing of HCCI engines due to their quick response for in-cylinder temperature control through internal exhaust gas recirculation (EGR). However, the application of VVA to diesel engines is not as thoroughly studied or documented. This is partially due to the inherent nature of the diesel engine. Unlike a gasoline engine, the air handling system of a diesel engine is unthrottled which gives little room to improve pumping losses. Also, there are restrictions to wide range application of VVA because of the small clearance between the piston and valves at TDC. This clearance

plays a very important role during the manipulation of valve closing and opening timing of an engine cycle.

Most studies of variable valve timing (VVT) on diesel engines have focused on improving torque and part load fuel economy. The conventional valve lift profile is always a compromise between high speed engine performance and low speed fuel economy. VVT provides the tool to optimize valve timing at each engine speed and load. Also, VVT increases the turbocharger efficiency via working on a desired region and decreases the exhaust gas flow through waste gate. Parvate-Patil et al [5] analyzed the variable valve timing events and their effects on a single cylinder diesel engine. The analysis is based on GT-Power modeling results. The paper briefly summarizes the potential benefits of eight valve events. Helmantel and Denbratt [6] studied the possibilities of using valve timing as a tool to control HCCI combustion of diesel fuel and to determine if it is possible to be used as a substitute for external EGR. Although combustion phasing was successfully controlled using valve timing, emissions were deteriorated. The results are not surprising since EGR is the dominant factor that controls NO_x emissions. Reducing EGR will typically result in higher NO_x emissions. Lancefield et al. [7,8] used a GT-Power model to identify how VVA can be used to improve the part load fuel economy of a modern light-duty diesel engine. The model includes a base engine, a variable geometry turbo-charger (VGT), cooled EGR and common rail fuel injection. The paper describes how the fuel economy was affected by the interactions of the VGT and EGR system. It claimed that up to 6% and 19% improvements can be achieved on indicated and brake fuel consumption respectively. They also claimed that minimizing air flow is the key to good part load fuel economy. Tai et al. [9] also developed a 2.7 liter V6 turbocharged diesel engine GT-Power model to study the potential for an increase in engine torque by optimizing the intake and exhaust valve timing together with turbocharger operation. They claimed a maximum of 45% improvement on low speed torque by optimization of both intake valve closing and the exhaust valve opening. Fessler and Genova [10] studied the potential of VVT for diesel engines using an electro-hydraulic "lost motion" VVA system. Some major benefits include: 6% gain in fuel consumption with respect to fixed actuation; extremely fast and accurate control of the internal EGR rate; improved engine startability and warm-up by varying compression ratio and cylinder deactivation, the control of exhaust gas temperature for improved catalyst efficiency / DPF (Diesel Particulate Filter) regeneration and the possibility to integrate a reliable engine brake system in a relatively simple and cost-effective way. Kamo et al. [11] compares the emission characteristics and fuel consumption between an insulated standard engine and an insulated Miller cycle engine. The Miller cycle engine offers superior BSFC over the standard engine. However, the Miller cycle engine has demonstrated worse smoke characteristics than the standard engine.

Also, the air/fuel ratio may become too low to maintain good combustion quality at full load.

Although some qualitative analysis about the potential emission benefits of using VVA exist in the papers cited above, only recently did the researchers focus their efforts on achieving lower pollutant emissions via VVA. Murata et al. [12] employed a variable valve timing mechanism to achieve premixed diesel combustion at higher load for low emissions and high thermal efficiency. Lower compressed gas temperature was achieved by means of late intake valve closing. Longer ignition delay enhances the fuel air mixing and reduced the smoke emissions. A 3-D CFD code combined with detailed chemistry was employed to analyze the combustion process which shows simultaneous NO_x and smoke reduction when using LIVC. The results also show ignition timing control via variable effective compression ratio. Nevin [13] investigated a prototype solenoid activated variable intake valve closing device on a Caterpillar 3401 SCOTE heavy duty diesel engine to control combustion phasing with a PCCI combustion strategy and to examine effects on emissions. Various EGR percentages, intake manifold pressures, fueling rates, intake valve closing timings, injection pressures, and start-of-injection timings were investigated to explore the limitations and potential of the intake valve actuation system. The major findings of the study include: 90% NO_x reduction at high speed and intermediate load with constant airflow and no EGR, 10% less EGR was required to achieve the same NO_x emissions while significantly decreasing CO and PM levels by nearly 70%.

Although the previous work shows the benefits of VVA in reducing pollutant emissions, very few results are available at Tier 2 Bin 5 or lower NO_x levels. Also, most published data focused on very limited operation conditions. A comprehensive assessment of VVA on diesel engine performance, emissions, and fuel economy benefits is crucial before implementing this technology.

SYSTEM DESCRIPTION

FULLY FLEXIBLE VALVE ACTUATION (FFVA) SYSTEM

A detailed description of the FFVA system, control, and capability can be found in [14]. A schematic of the FFVA system is shown in Figure 2. Briefly, the FFVA system consists of a hydraulic pump, accumulators, servo valves, hydraulic pistons, linear variable differential transducers (LVDT), signal conditioners, power amplifiers, and the control system. The hydraulic pump supplies fluid at 207 bar (3000 PSI). Four high response servo valves control the high-pressure fluid. The hydraulic fluid that discharges from the servo valves drives hydraulic pistons, which actuate the engine valves. At the neutral state, the engine valves are closed by the preloaded valve springs. The valve

actuators open the valves by overcoming the spring force. An LVDT is mounted on top of each hydraulic actuator to detect its displacement. The output of the LVDTs is conditioned and then sent to the control system. The control system consists of two controllers (one digital and one analog controller). The digital controller is implemented on a dSpace system (DS1103). The control system receives the LVDT and shaft encoder signals, calculates the control output, and amplifies the signal with a MOOG power amplifier to drive the servo valves. Thus, the controller provides closed loop control to ensure accurate positioning of the electro-hydraulic valve train. It should be noted that the engine valve actuators and the engine valves are in contact with each other during the valve opening and closing, which means LVDT output accurately represents the valve displacement.

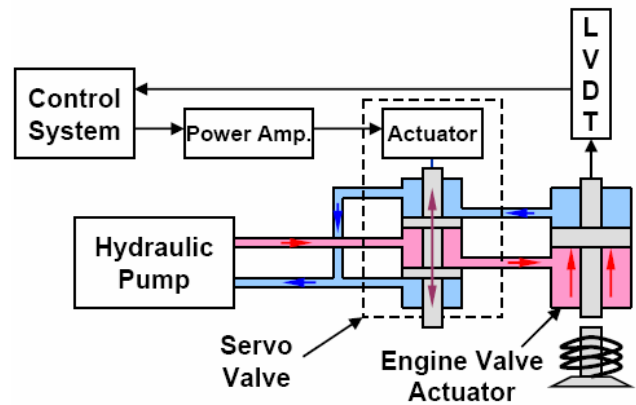


Figure 2. Schematic diagram of FFVA system

SINGLE CYLINDER DIESEL ENGINE SETUP

The experiments were carried out on a single cylinder diesel engine. Engine specifications and a schematic diagram of the system setup are shown in Table 1 and Figure 3 respectively. The fuel injection system is a Delphi DFI 1.4 solenoid common rail system capable of a maximum of 3 injections per cycle. The fresh air flow was controlled by a critical air system with 4 independent orifice plates. The exhaust pressure was close-loop controlled by two parallel valves. An EGR valve, located downstream of the EGR cooler, was used to control the EGR flow. In-cylinder pressure was acquired using a Kistler (type 6125B) piezoelectric pressure transducer. Mean effective pressure and apparent heat release rate were calculated based on the cycle-averaged pressure using 100 cycles. The other measurement instruments are listed in Table 1.

LATE INTAKE VALVE CLOSING (LIVC)

In this valve lift strategy, the intake valve is left open longer than it would be in a conventional engine. As the piston initially moves upwards in the compression stroke, the charge is partially expelled back out the still-open intake valve. A key aspect of LIVC is that the

compression process actually starts only after the intake valve closes. In other words, compression occurs only in the latter part of the compression stroke. The reduced compression process results in a lower effective compression ratio, and thus lower compression temperatures.

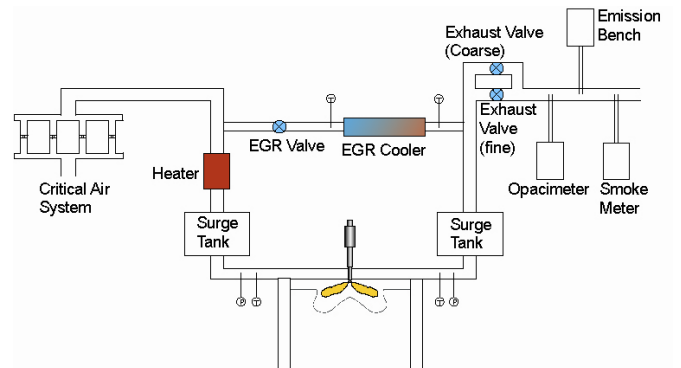


Figure 3. Schematic diagram of the single cylinder engine

Table 1. Engine geometry, injector specifications, and measurement instruments

Engine	single cylinder diesel
Displacement (L)	0.56
Compression ratio	14.5
Fuel injection system	Common rail, Maximum pressure 1800 bar
Injector body	DFI 1.4
Injector cone angle (°)	155
Injector hole number	7
Emission Bench	Horiba MEXA-7500DEGR
Opacimeter	AVL 439
Smoke Meter	AVL 415S
Noise Meter	AVL A01

The influence of LIVC on the combustion process can be depicted in a ϕ -T diagram as shown in Figure 4. Two islands represent two major pollutant formation zones. The soot production zone is located at temperature higher than 1400K and $\phi > 2$. The NO_x production zone is located at high temperature and near stoichiometric conditions. The path of a fuel parcel undergoing conventional combustion is shown in red. From A to B represents the fuel air mixing process before ignition. After the ignition delay, the premixed combustion occurs which rapidly increases the temperature. From B to C, the mixture enters the soot production zone. From C to D represents processes occurring in the diffusion flame. During this period, the unburned fuel keeps reacting with the available O₂. The equivalence ratio decreases and the temperature increases until the stoichiometric condition is reached. High temperature results in NO_x formation. Below $\phi=1$, no significant chemical reaction occurs and mixing serves to lower the temperature. For conventional diesel combustion, it is very difficult to avoid the pollutant formation zones.

LIVC changes the path of the fuel parcel at the very beginning when fuel is injected into the combustion chamber. The modified process is depicted as a blue line in Figure 4. Due to the lower compression temperature, the ignition delay increases, which provides a longer mixing time prior to the start of ignition (A to B'). The local equivalence ratio is reduced, which contributes to soot reduction. Lower combustion temperature also reduces the NO_x formation.

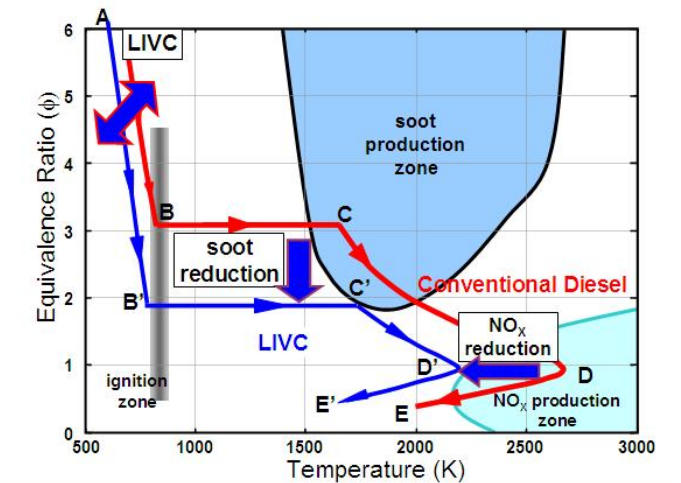


Figure 4. Postulated path for conventional and low temperature diesel combustion in the ϕ -T plot

Typical valve lift profiles for conventional and LIVC cycles are shown in Figure 5. The pink lines represent the conventional cam driven valve lift profiles, which have a maximum valve lift of 9.2 mm. In most of the current study, the maximum valve lift is controlled to be about 6 mm. The lower valve lift is used to reduce the risk of a valve-piston collision, but at the light load conditions which were investigated, the volumetric efficiency was not significantly compromised. The valve opening and closing timing are controlled by adjusting phasing and duration of the valve lift profile. It should be noted that the intake valve opening profile is fixed when performing IVC sweeps. This is to minimize the variation of internal EGR quantity and pumping losses, which are very sensitive to the intake valve opening profile.

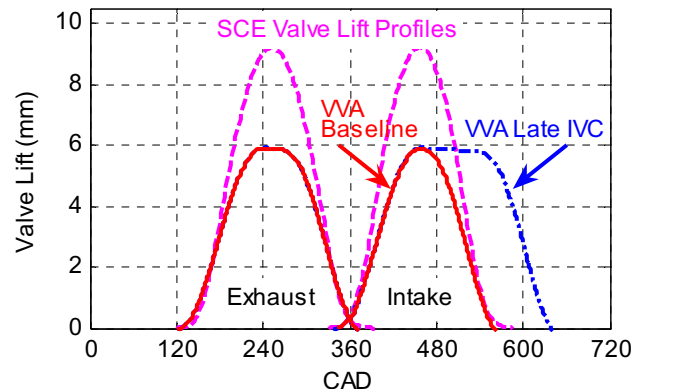


Figure 5. Typical Valve Lift Profiles

EXPERIMENTAL RESULTS

THE DEFINITION OF EFFECTIVE COMPRESSION RATIO (CR)

The effective compression ratio is a useful parameter for describing the effect of late intake valve closing. Effective CR can be defined based on different methods. A common method is based on the volumetric ratio, which is defined as the ratio of cylinder volume at intake valve closing to the TDC volume. However, this is not the best definition to explain experimental results since it does not represent the actual in-cylinder compression process. As shown in Figure 6, the gas in the cylinder is partially compressed due to the high flow resistance across the valve before IVC.

For this work, we used a pressure based effective CR which better represents the actual in-cylinder compression process. The method to determine the effective compression ratio is depicted in Figure 6. First, a linear fit is applied to the compression process on the $\log(p)$ - $\log(V)$ diagram. This linear fit is extrapolated to the intake manifold pressure. The corresponding volume is defined as the effective IVC volume. As can be seen, the effective IVC volume is larger than the cylinder volume at the time the intake valve fully closed. The effective compression ratio is then defined as the ratio of the effective volume to the TDC volume. Figure 7 shows the variation of volumetric CR and pressure based effective CR as a function of IVC timing. The volumetric CR decreases almost linearly with intake valve closing. However, very little change was observed in the pressure based effective CR when IVC changes from 160 to 130 BTDC (the baseline intake valve closing timing of the engine is 146 degree BTDC). Significant change in effective CR occurs when the IVC timing is retarded by more than 50 CADs.

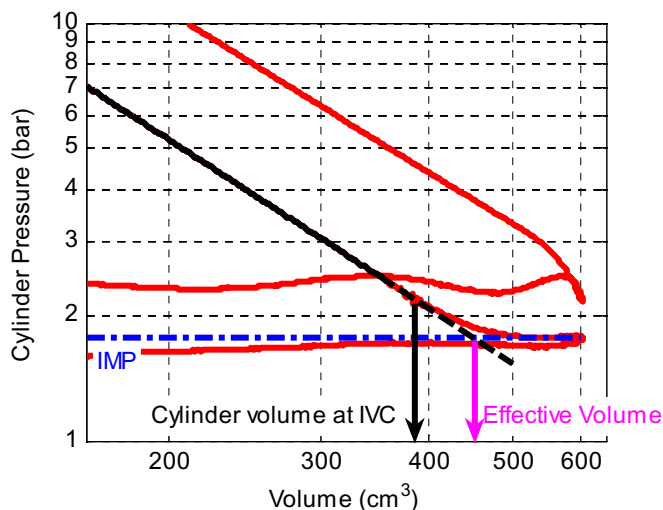


Figure 6. The illustration of the definition of the effective CR

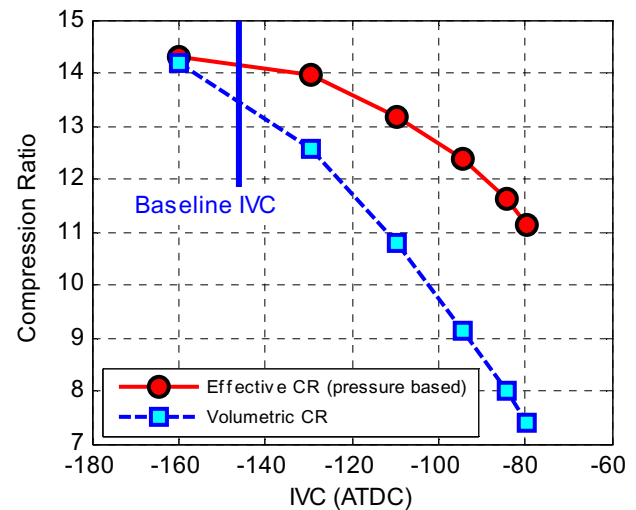


Figure 7. The variation of effective CR and volumetric CR with intake valve closing

Since the pressure based effective CR represents the actual compression of the charge in the cylinder, it will be used in the reminder of this report. "IVC" followed by a number will be used to represent the intake valve closing timing. For example, "IVC80" means intake valve closing at 80 degree before TDC.

THE EFFECTS OF LATE INTAKE VALVE CLOSING (LIVC) ON ENGINE PERFORMANCE AND EMISSIONS

The current study of LIVC focuses on loads between 2 bar and 11 bar BMEP. LIVC is inappropriate for application to high loads (> 11 bar BMEP) in a light duty engine because of the reduced volumetric efficiency. Since the intake manifold pressure is high at high loads, the application of LIVC calls for even higher boost pressure, which increases the burden on the air handling system. Also, LIVC is not appropriate at low loads (< 2 bar), where combustion stability and unburned HC are the major concerns. LIVC reduces the compression temperature, which further deteriorates the combustion stability, HC, and CO emissions at light load. In this paper, the experimental results at three representative operating conditions were presented. The experimental conditions are listed in Table 2.

Table 2. Key Test Conditions.

	Speed (rpm)	BMEP (kPa)	NMEP (kPa)
A1	1400	281	381
A2	1600	420	540
A3	2000	980	1133

At a given speed and load, the key parameters that influence NO_x emissions include air-fuel ratio, EGR, combustion phasing, intake manifold temperature (IMT), and intake valve closing timing. To evaluate the effect of LIVC on NO_x emissions, two different approaches were

utilized. Approach #1: Examine NO_x emissions variation with LIVC while fixing the EGR. Approach #2: Examine the EGR requirement to achieve the target NO_x emissions.

A1—LIVC Sweep, Constant EGR

This set of experiments was conducted to investigate the effect of LIVC on early PCCI combustion. At this low speed and load condition, a single injection is adequate to achieve the best overall engine performance. The test was conducted by examining the NO_x variation with LIVC (described as Approach #1 above).

Figure 8 shows plots of cylinder pressure, bulk temperature, apparent heat release rate (AHRR), injector solenoid current, and valve lift of the baseline valve lift (IVC156) and IVC100 while NO_x emissions were controlled under Tier 2 Bin 5 emission level. The red color represents the baseline valve lift (IVC156), and the blue color represents IVC100. The maximum valve lift is controlled to be 5 mm, which is sufficient at this speed and load to deliver similar volumetric efficiency to the full lift. The major pressure difference was observed during the pumping cycle as seen in the log(p)–log(V) diagram. Since LIVC reduces the volumetric efficiency, higher boost pressure is needed to maintain the in-cylinder gas density. Since the in-cylinder gas densities are almost identical, very little pressure difference was observed during the compression stroke. LIVC results in slightly lower pressure because of lower compression temperature.

The impact of LIVC on the ignition delay is significant because temperature is the most sensitive factor that influences the ignition delay. Reducing the effective CR from 14.5 to 11.0 is equal to about 80° Celsius temperature reduction near TDC. When performing LIVC sweeps, injection timing is advanced to maintain the MFB50 (50% mass fraction burn) at 7 CAD after TDC. Compared to the baseline case, injection timing is advanced by about 10 CAD when applying LIVC. The variation of ignition delay as a function of effective CR is shown in Figure 9. For both cases, cool flame combustion was observed before the main combustion. Although the combustion phasing is maintained, LIVC results in lower peak heat release rate and longer combustion duration due to the lower combustion temperature. The temperature traces shown in Figure 8 are the bulk (average) in-cylinder temperature. LIVC reduces the temperature during the whole compression and combustion process, which is the key reason of the NO_x reduction.

Similar experiments at Tier 2 Bin 8 baseline engine out NO_x emission level were conducted. Generally, the same phenomena are observed as described above. Compared to the Tier 2 Bin 5 NO_x emissions, the ignition delay time is shorter although the initial IMT is lower. This is due to the higher AFR and lower EGR, which result in higher intake manifold O₂ concentration. As

described in [15], ignition delay is also strongly influenced by the O₂ concentration.

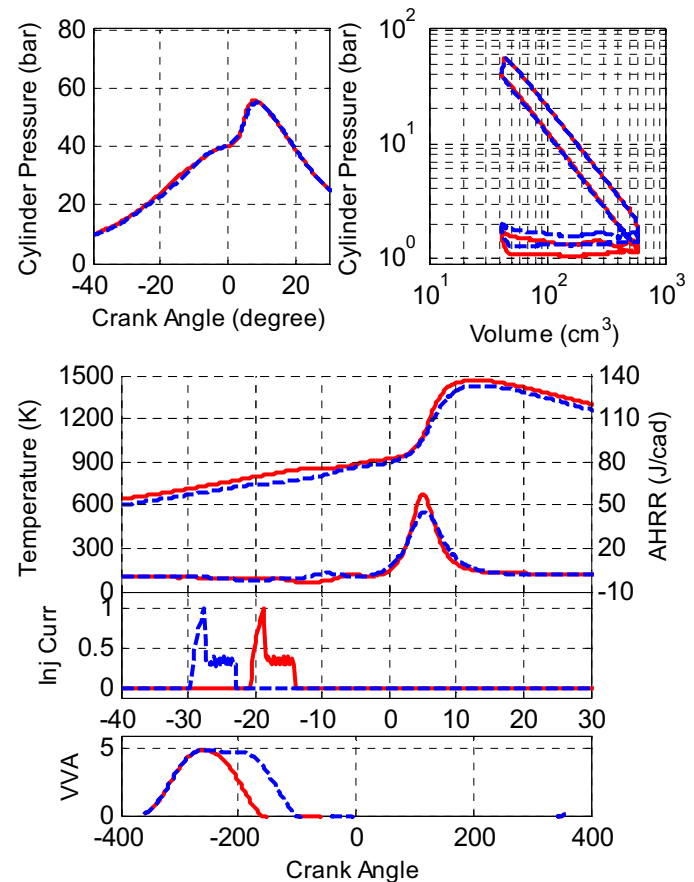


Figure 8. Comparison between baseline valve lift and LIVC on pressure, temperature, AHRR, and injection timing. Experimental conditions: 1400rpm, 3.8 bar NMEP, A/F= 18, EGR = 58%, [O₂]=11%, Tier 2 Bin 5 NO_x emission level, Red solid lines: EINO_{x156}=0.185 g/kg-fuel, Blue dashed lines: EINO_{x100}=0.14 g/kg-fuel.

The effects of LIVC on engine performance and emissions at this condition are presented in Figure 9. In the sub-figures, the x-axes are effective CR in reverse direction so that retarding the IVC moves to the right in the figure. EINO_x was reduced by up to 50% when effective CR was reduced to lower than 12. Although further retarding the intake valve closing would result in lower NO_x emissions, higher boost pressure was needed to offset the decreasing volumetric efficiency, which might result in higher pumping losses depending on the turbocharger efficiency. The required turbocharger combined efficiencies are shown Figure 9. The required turbocharger combined efficiency was calculated using a simple work balance model given in the following equation