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Combined Effects of Late IVC and EGR on Low-load Diesel Combustion

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ABSTRACT

The increasing demand for improved efficiency of diesel engines requires more advanced combustion solutions. These solutions include the use of variable valve timings in combination with more traditional methods such as EGR, turbocharging and advanced injection systems. By modifying the characteristics of the charge air, further hardware optimization becomes possible.

In the current investigation, the effect of late intake valve closing (LIVC) was investigated together with the effect of (external) exhaust gas recirculation (EGR) in a single cylinder heavy duty diesel engine. Different injection timings and injection pressures were investigated. The mass flow of oxygen was kept constant in order to show how the density and temperature of the reactant mixture affect the combustion and emission characteristics.

The combustion results showed that if the oxygen mass flow is kept constant, an EGR approach is more efficient than LIVC in lowering fuel consumption due to the effects of increased cylinder gas flow which improves fuel conversion efficiency. It was shown that the ignition delay for a fixed combustion phasing was independent of EGR but could be increased by LIVC. The peak pressure was more strongly affected by EGR due to the larger gas flow but this response can be reduced by means of LIVC. After compensating for combustion timing effects, the reduced peak pressure was mainly attributed to reduced effective compression ratio resulting from the LIVC.

The results show how variable valve timing can be used as one important tool to obtain better combustion characteristics and thus enable more efficient powertrains.

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INTRODUCTION

The world faces grand challenges to mitigate the effects of an un-sustainable usage of fossil fuels. While alternative fuels will become more important, fossil fuel will be a major energy source for transportation in many years to come. Improved efficiency of the internal combustion engines provides one important way to reduce CO_2 emissions and present and future legislations will make an important contribution to reduce these CO_2 emissions.

There are many ways to improve the efficiency but improved combustion efficiency is often accompanied with high local temperature which results in unacceptably high NO_X emissions. NO_X is mainly formed by the Zeldovic mechanism which is favored by high temperature (as well as oxygen concentration and time). One method to reduce NO_X formation is by lowering the oxygen concentration by means of EGR. The lowered oxygen concentration will also result in lower peak temperature and significant reductions in NO_X formation can be obtained. However, EGR also slows down the combustion rate and the efficiency is not as high as for non-EGR systems. Another way to obtain lower peak temperatures is by obtaining a cooler gas mixture at the time for fuel injection. Even though charge air cooling is commonly applied, the gas compression itself inevitably raises the temperature. Yet another NO_X reduction strategy is reducing the compression stroke itself by means of changing the timing of the inlet valve closure (IVC). This method was proposed by Ralph Miller in 1957 [1] and it appears in the literature under a variety of names depending on the technical implementation. In this study we will use the name “late Miller” even though “supercharged Atkinson cycle” is also appropriate. Since reduced compression stroke will result in lower amount of air, the volumetric efficiency will be low unless compensated for by a turbocharging system. Until recently, systems for changing the IVC haven't been economically attractive for implementation in heavy duty powertrains, but upcoming regulation changes and the need for more fuel-efficient powertrains have changed this situation. In a current EU project (CORE, CO2 Reduction for long distance transport) [2], different Variable Valve Actuation (VVA) systems are being evaluated and promising results are being obtained by coupling the approach with an efficient turbocharging system.

Over-expanded cycles are well established [3] showing an increase in efficiency but with a decrease in IMEP and power density. The Miller cycle has been studied for the Otto-cycle (e.g. [4, 5]), the diesel cycle (e.g.[6, 7, 8, 9, 10, 11, 12, 13, 14]) or both (e.g.[15, 16]). The concept has also been studied for alternative fuels (e.g. [17]). Supercharging was already envisaged in the Miller patent, but in cases where fuel efficiency alone is sought the Miller cycle without supercharging (i.e. the Atkinson cycle) is applied, for example in hybrid vehicles [18].

In the literature, different aspects of various Miller cycles have been investigated. Since the lower compression ratio lowers the peak temperature, many studies demonstrate reduced NO_X emissions e.g.[6, 7, 17, 19]. Another aspect of the system to consider is the need for supercharging during high load [11, 12]. The lower cylinder pressure (i.e. lower gas density) will increase the risk of the spray hitting the wall e.g. [6, 8, 20] which could be detected by particulate matter (PM) analysis [13]. Even though many studies report on the importance of spray atomization, most studies keep the injection pressure constant (e.g. [6, 13, 21, 22, 23, 24]. In some cases the injection pressure is varied as a means to vary the load point [7, 17, 19].

Many studies explore the possibility of having a larger amount of pre-mixed combustion (e.g. [6, 24]) and the effects of EGR to lower both NO_X and peak pressures [5, 9, 10, 12].

In this study the effect of LIVC was investigated together with the effect of EGR in a single cylinder heavy duty diesel engine. Different injection timings and injection pressures were investigated at two different load points. The mass flow of oxygen was kept constant in order to show how the physical properties (density and temperature) affect the combustion and emission characteristics.

EXPERIMENTAL

The engine was a single cylinder heavy duty diesel engine equipped with a common rail injector system (from Delphi) and a VVA system (from Cargine). The EGR was controlled by adjusting the back pressure and in the case of no EGR the back pressure was set equal to the boost pressure, as in [9]. Ultra low sulfur diesel fuel (Swedish MK1) was used throughout this study. The engine details are given in Table 1. The common rail system was pressurized with a pumping injector and can deliver rail pressures up to 3000 bar [25]. The rail pressure, injection timing and duration was controlled by a separate software (ATI Vision).

Table 1. Single cylinder engine and nozzle specifications

Displacement	2 dm3
Bore / Stroke	131 / 150 mm
Geometrical compression ratio	17:1
Nozzle design	5 hole, 150 deg angle
Nozzle flow number	2.00, d_0=256μm

Two different load points were investigated; one low load (1000rpm, 120Nm, 25% of full load) and one medium load (1500rpm, 180Nm, 50% of full load). In order to isolate the physical properties of the charge air (temperature and density) from chemical property (oxygen concentration), the oxygen flow into the cylinder was kept constant by adjusting the Boost pressure. Other factors affecting the system were explored (by applying factorial designs) over the design space by varying IVC, EGR, injection pressure and timing, see Table 2. The start of injection (SOI) was adjusted to get 50% burnt (CA50) at three different timings.

Table 2. Evaluated experimental conditions

Variable	Levels
Load points	1000rpm/120Nm, 1500rpm/180Nm
IVC (set values)	0; 40; 70;(90) CADabdc
EGR	0%; 20%; 40%
Pinj	1500 ; 2400 bar
CA50	Approx. 2; 5; 8 CADatdc

In total, about 120 experiments were performed and a number of intermediate responses were measured along with end performance responses which were fuel consumption (ISFC) and emission (NO_X, CO, HC and soot) results.

Emissions (NO_X, CO & HC) were measured with an AVL AMAi60 and the soot was measured with a smoke meter (AVL 415). The rate of heat release was calculated in Osiris (D2T). The fuel consumption was given as ISFC (indicated specific fuel consumption) since compressor work is not included in the system but rail pressure is included (rocker arm driven by the shaft).

RESULTS AND DISCUSSIONS

Analysis of Effective IVC

From previous studies [9] comparing early and late IVC, the effect from IVC (as recorded by the sensors) was not completely "symmetrical" and it was reasoned that this effect was due to heat losses. Another explanation could be local inlet pressure effects which can be accounted for e.g. by He [19] and Doosje [20]. By assuming isentropic compression, the effective IVC (IVC_{eff}), could be estimated by linear regression. One example is shown in. Figure 1 Note that the IVC_{eff} is defined as the cylinder volume when the log-scaled cylinder pressure makes a change in slope. This differs from [19, 20] who extrapolated to the intake manifold pressure. By using measurements of airflow (as well as estimation of trapped gas at EVC), the gas density and temperature at SOI can be determined. Note that the two methods for extrapolation give the same results, but the present method doesn't rely on inlet pressures measured in a tank upstream the inlet manifold. The temperatures at IVC using this method agreed with simulations of the engine using GT power.

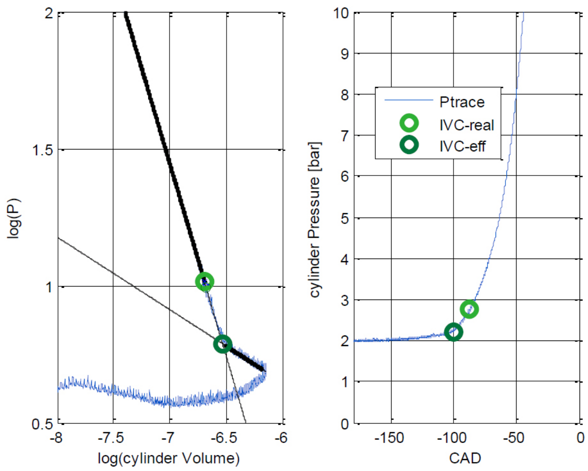


Figure 1. Example of linear regression fit of logarithmic data to obtain the effective IVC (a). Figure (b) shows the same results displaying CAD on the x-axis. In this example the boost pressure is 2 bar, the real IVC (measured by sensors) is 93 CADabdc and the effective IVC is 80 CADabdc.

A set of experiments (motoring at 1000rpm) were performed to evaluate the effect on valve lift profile as a function boost pressure and IVC. The results showed that the lift height decreased as the IVC was retarded but boost pressure had minor effect on the IVC, see Figure 2.

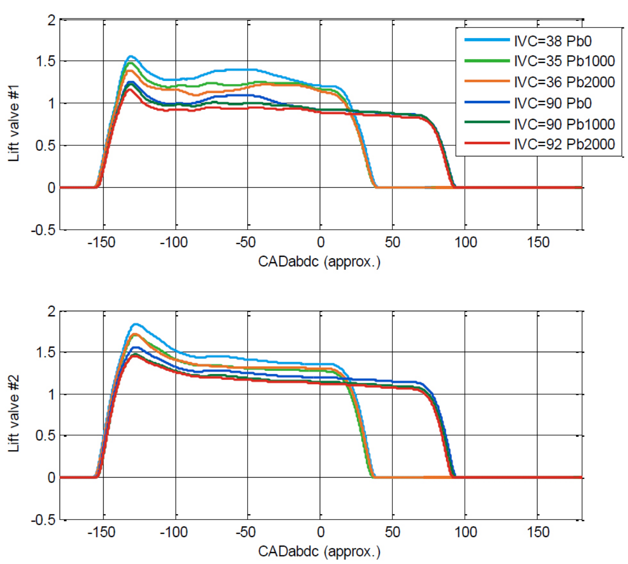


Figure 2. Inlet valve lift height with varying IVC and boost pressure.

The effective IVC (IVC_{eff}) as function IVC and boost pressure is displayed in Figure 3. From Figure 3, it is clear that IVC_{eff} (and the deviation to the IVC determined from the sensor) is not depending on boost pressure and only slightly depending on IVC.

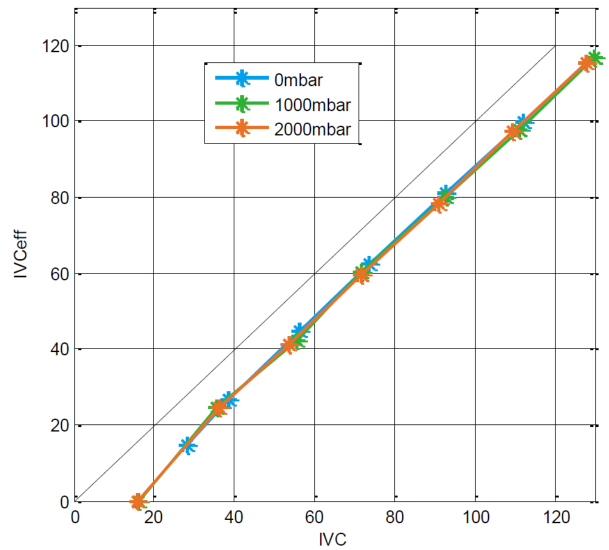


Figure 3. The effective IVC as function of IVC at varying boost pressure.

Analysis of Pressures and Heat Release

A few examples of Rate of Heat Release (RoHR) are given in Figure 4 (without EGR) and Figure 5 (with EGR). Each set of experiments are shown with similar injection timing at the same engine load point and same injection pressure.

From the experiments without EGR it was shown that later IVC results in more premixed combustion (the first peak of the RoHS is relatively larger than the later part). Since the SOI was chosen to get similar timing (indicated by CA50), the change in fuel consumption was small, but later IVC yields lower fuel consumption.

The RoHR curves for experiments with EGR shows similar sensitivity to IVC. Later IVC gives more pre-mixed combustion. In contrast to the experiments without EGR, late IVC gives higher fuel consumption. The effect is small but may be attributed to combustion temperatures getting too low when applying both EGR and LIVC. Another possible reason is the early combustion resulting in higher cylinder pressure (higher temperatures) and higher heat transfer losses.

For experiments with EGR, it is also interesting to note the correlation between end of injection (EOI) and start of the main combustion peak. This can probably be attributed to entrainment waves as described by Musculus [26]. Even though the combustions in this study can't be classified as pure PPCI (partially Pre-Mixed Compression Ignition), the application of LIVC in combination with early injections shows some of these features.

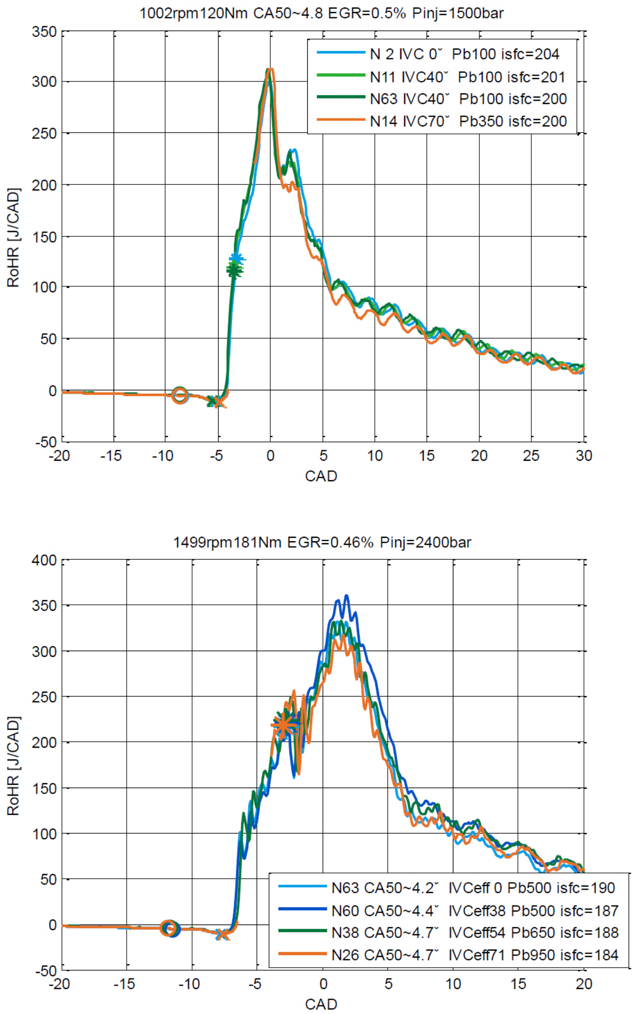
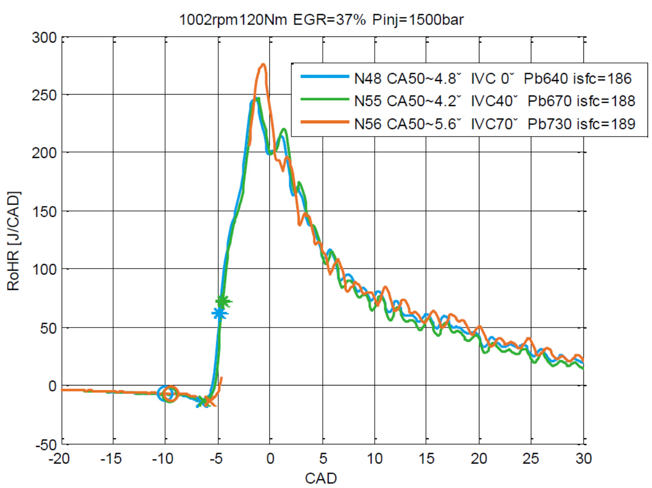


Figure 4. Pressure trace and rate of heat release for experiments without EGR (two different load points). Circles indicate SOI, cross indicates SOC and stars indicates EOI.



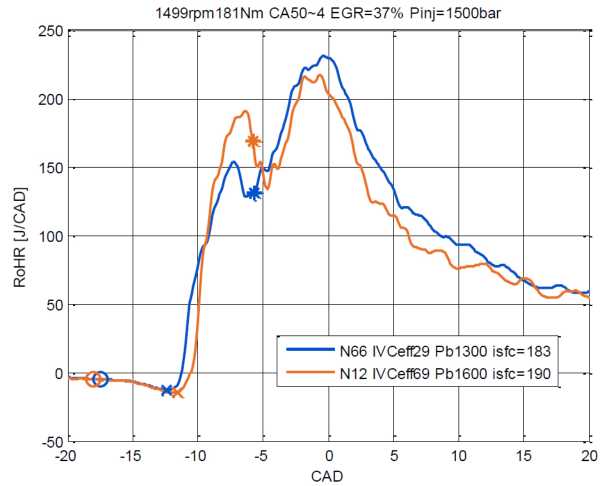


Figure 5. Rate of heat release for experiments with EGR (two different load points). Circles indicate SOI, cross indicates SOC and stars indicates EOI.

Sensitivity of combustion results by change in LIVC

In order to visualize the effect of LIVC, the two load points are compared and EGR vs non-EGR at different injection pressures are compared in the subsequent figures.

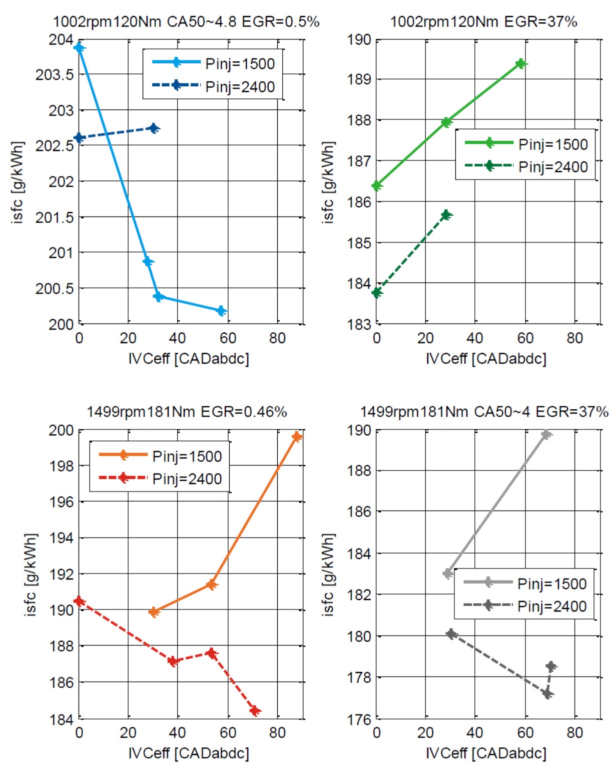


Figure 6. Indicated fuel consumption for two load points and differens amount of EGR and Pinj.

In Figure 6 it is clear that higher EGR gives lower fuel consumption due to the increased gas charged into the cylinder. The changes in ISFC when varying LIVC are relatively small. However, from Figure 6 it is clear that ISFC does not always decrease with increasing LIVC. One important reason for this is the timing of the combustion. For example, for low load and low P_{inj} (blue, upper left figure), the combustion timing is late enough that the increase in LIVC makes the increase in pre-mixed combustion result in a decrease in ISFC. On the other hand, for cases with EGR and low Pinj, the combustion phasing is so early that an increase in pre-mixed combustion results high cylinder pressure, higher temperatures, higher heat transfer losses and consequently increased ISFC.

The NO_X emissions are displayed in Figure 7. Similar to the fuel consumption, the NO_X emissions does not always decrease with an increase in LIVC. In most cases the increase in pre-mixed combustion results in lower temperature for the main combustion and lower NO_X emissions. In some cases however, the combustion timing is early enough that the increase in pre-mixed combustion (in combination with combustion near tdc) has a negative effect on NO_X emissions.

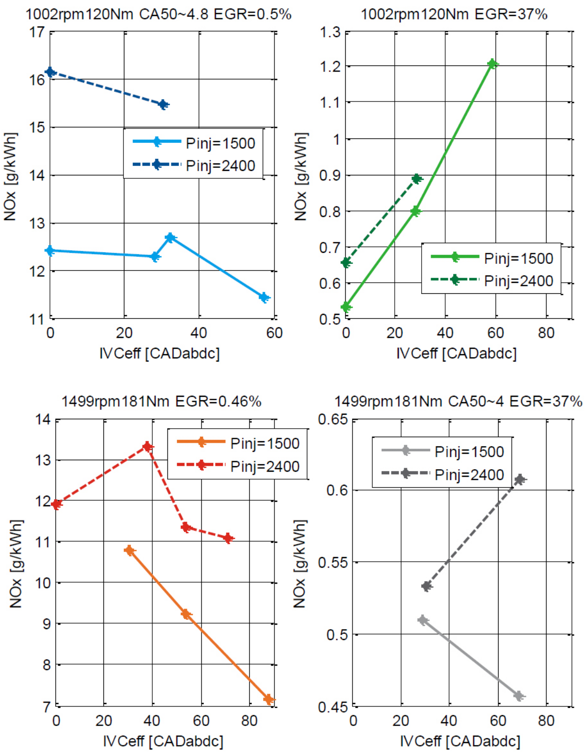


Figure 7. NOX emissions from the two load poins at different levels of EGR and Pinj.

The peak pressures are displayed in Figure 8.

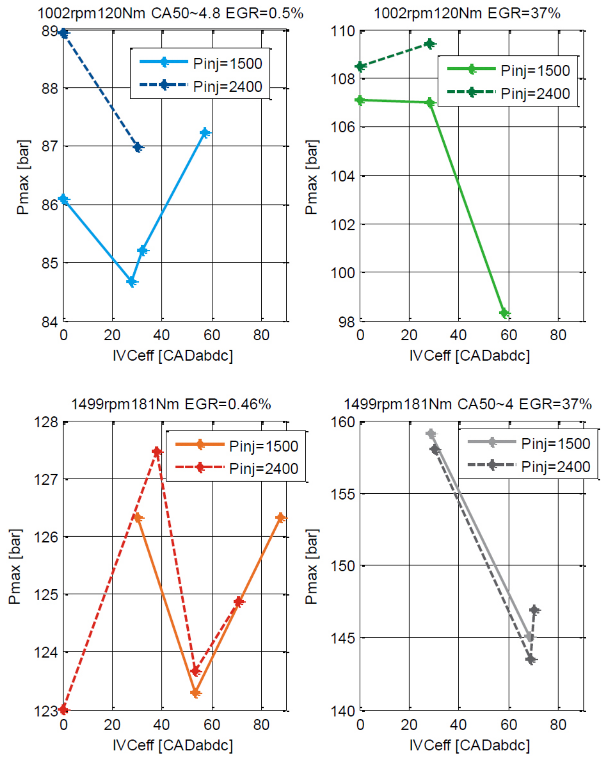


Figure 8. Peak pressures for experiments at two load points and different levels of EGR and Pinj.

In most cases the peak pressure decreases when LIVC is retarded. In some other cases (e.g. for low load, no EGR and low Pinj), the peak pressure increase with increase in LIVC. This can be explained by a slightly higher charge of air, resulting in higher density and a consequent increase in P_{max}. The relationship between NO_X emission and peak pressure is displayed in Figure 9

In Figure 9, the NO_X emissions show a clear dependence on Peak pressure (and thus local max temperature) and EGR (O_2 concentrations). The range of NO_X emissions is obtained by different courses of combustion (residence times at high temperatures) which were enabled by variations in LIVC.

The ignition delay was assessed by calculating the time from start of injection to start of combustion (as indicated in Figure 4 and Figure 5) and are displayed in Figure 10.

The ignition delay shows very similar values independent of load point or EGR. Note that the experiments selected in this plot had CA50 of about 5 CADatdc, which means that SOI was chosen as to get the desired CA50. As a consequence, the ignition delay is most sensitive to injection pressure and IVC.

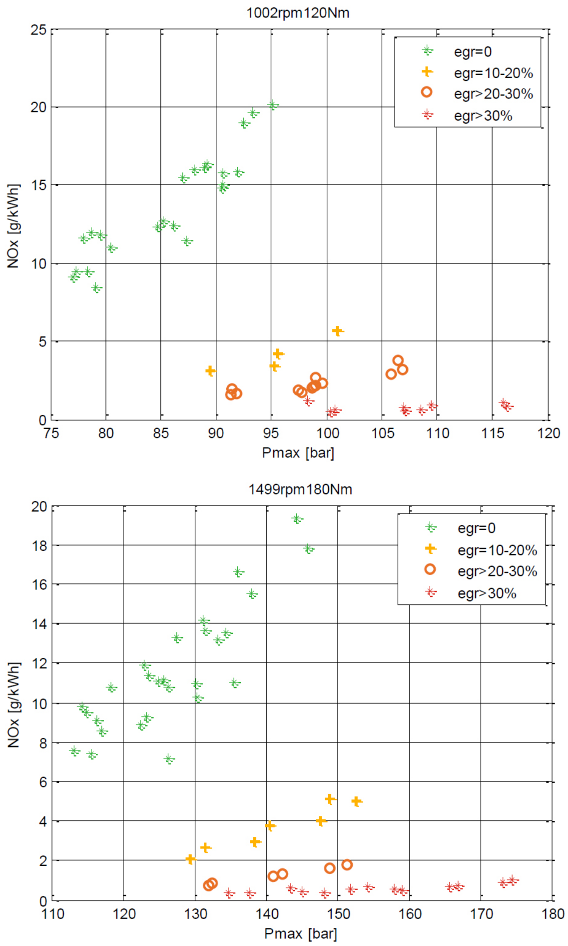


Figure 9. Correlation between Peak pressure (max temperature) and NO_X emissions for different levels of EGR (O_2 concentration)

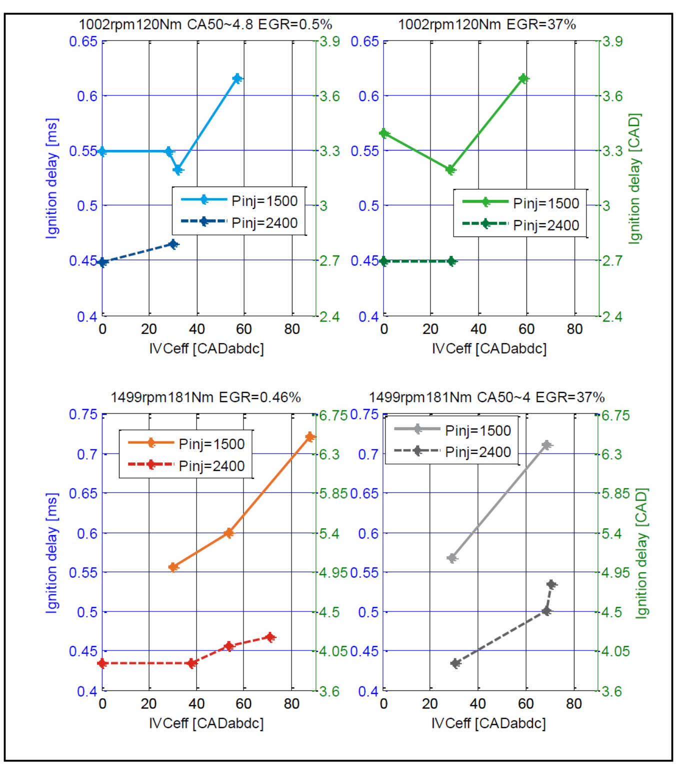


Figure 10. Ignition delay for the two loadponts and for different levels of EGR and Pinj.

SUMMARY & CONCLUSIONS

In the presented investigation, LIVC has been explored by varying most engine parameters except O_2 flow. The indications from this approach are that higher EGR requires higher charge gas flow which gives better efficiency. From the experiments performed here, one might be tempted to conclude that EGR is a more efficient means to reduce ISFC than LIVC, but the current experimental setup cannot prove this conclusion since the EGR is highly correlated with the gas flow. This issue is presently under investigation. The sensitivity of the results for IVC at different load points when supplied with different levels of EGR and injection pressures indicates that the combustion timing is an important parameter when assessing the effect of LIVC. Here, combustion near TDC has been presented and the balance of pre-mixed combustion and combustion before TDC is delicate. Even though combustion phasing (as defined by CA50) was almost constant the effects of LIVC was not constant. This shows that general statements of the effect of LIVC cannot be made without also describing the combustion characteristics (e.g. amount of pre-mixed combustion before TDC).

The results show that the combustion characteristics are very sensitive to LIVC. In combination with turbo charging and EGR, the use of LIVC enables independent control of the charge density, temperature and oxygen concentration. Therefor LIVC can be an important tool for optimizing future powertrains.

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DEFINITIONS/ABBREVIATIONS

atdc/ abdc - After / Before top dead center

CA50 - 50% of total heat release

CAD - Crank Angle Degrees

cr_{eff} - Effective compression ratio

EGR - Exhaust Gas Recirculation

EOI - End of Injection

IMEP - Indicated mean effective pressure

ISFC - Indicated specific fuel consumption

IVC, IVC_{eff} - Inlet Valve Close, effective Inlet Valve Close

LIVC - Late Inlet Valve closing

Pinj - (rail) Injection Pressure

PPCI - Partially Pre-mixed Compression Ignition

RoHR - Rate of Heat Release

SOC - Start of Combustion

SOI - Start of Injection

tdc, bdc - top dead centre, bottom dead centre

APPENDIX

Table 3. Table of the data used for figures 4, 5, 6, 7, 8 & 10 as well as additional information. CAD refers to CADatdc unless otherwise specified.

id	N	T	P_{boost}	P_{exh}	EGR	IVC	IVC_{eff}	cr_{eff}	O_2flow	gasflow	P_{inj}	SOI	EOI	SOC	IgnDel	CA10	CA50	CA90	ISFC	NO_X	HC	CO	SOOT
	rpm	Nm	mbar	mbar	%	CADabdc	CADabdc	-	kg/h	kg/h	bar	CAD	CAD	CAD	ms	CAD	CAD	CAD	g/kWh	g/kWh	g/kWh	g/kWh	g/kWh
1-2	1002	121	100	100	1	1	0	17.0	17	83	1500	-8.5	-3.3	-5.2	0.5	-1.3	4.9	27.8	203.9	12.4	9.23	0.96	NaN
1-11	1003	120	100	100	0	42	28	16.3	17	81	1500	-8.7	-3.6	-5.4	0.5	-1.5	4.9	28.3	200.9	12.3	3.23	0.67	0.00
1-14	1002	120	350	350	1	70	57	14.2	18	86	1500	-8.6	-3.5	-4.9	0.6	-1.1	4.8	28.0	200.2	11.4	2.35	0.53	0.00
1-63	1002	120	100	100	0	42	32	16.1	16	78	1500	-8.7	-3.5	-5.5	0.5	-1.5	4.8	28.4	200.4	12.7	0.30	0.60	0.01
1-48	1002	121	640	790	41	0	0	17.0	17	112	1500	-10.0	-4.9	-6.6	0.6	-2.2	4.8	28.3	186.4	0.5	0.35	2.30	0.21
1-55	1003	119	670	810	38	40	28	16.3	18	112	1500	-9.6	-4.5	-6.4	0.5	-2.1	4.2	25.3	188.0	0.8	0.42	1.53	0.09
1-56	1003	119	730	840	31	70	59	14.0	17	102	1500	-9.5	-4.4	-5.8	0.6	-1.5	5.6	29.3	189.4	1.2	0.17	1.07	0.06
1-5	1003	120	100	100	0	0	0	17.0	17	83	2400	-6.5	-2.5	-3.8	0.4	-0.5	4.9	25.5	202.6	16.1	3.46	0.42	0.00
1-8	1003	121	100	100	0	41	30	16.2	17	81	2400	-6.5	-2.4	-3.7	0.5	-0.5	4.8	25.3	202.7	15.5	3.71	0.54	0.00
1-49	1002	119	640	790	40	1	0	17.0	17	112	2400	-7.6	-3.8	-4.9	0.4	-1.1	4.5	24.6	183.8	0.7	0.31	0.93	0.06
1-54	1002	120	670	810	38	40	28	16.3	17	111	2400	-7.6	-3.7	-4.9	0.4	-1.2	4.3	23.6	185.7	0.9	0.17	0.68	0.03
2-31	1500	181	1200	1200	0	106	88	10.5	31	148	1500	-18.0	-5.4	-11.5	0.7	-6.4	2.9	35.4	199.6	7.2	0.03	4.08	0.07
2-35	1499	181	650	650	0	70	54	14.5	32	152	1500	-15.0	-3.1	-9.6	0.6	-4.7	4.0	31.1	191.4	9.2	1.15	1.41	0.02
2-55	1499	180	500	500	0	41	30	16.2	32	155	1500	-14.6	-3.0	-9.6	0.6	-4.3	4.0	29.1	189.9	10.8	0.30	0.97	0.01
2-12	1500	180	1600	1660	34	86	69	12.9	29	188	1500	-18.0	-5.9	-11.6	0.7	-6.4	4.1	34.7	189.7	0.5	0.12	NaN	1.74
2-66	1499	181	1300	1400	39	41	29	16.3	32	211	1500	-17.5	-5.8	-12.4	0.6	-6.2	3.9	31.3	183.0	0.5	0.21	11.0	NaN
2-26	1499	181	950	950	0	90	71	12.6	33	157	2400	-11.8	-3.1	-7.6	0.5	-2.6	4.7	29.8	184.5	11.1	0.30	0.49	NaN
2-38	1499	181	650	650	0	72	54	14.5	31	149	2400	-11.7	-2.9	-7.6	0.5	-2.8	4.7	29.4	187.6	11.4	0.28	0.77	0.01
2-60	1499	181	500	500	0	39	38	15.8	32	153	2400	-11.5	-2.8	-7.6	0.4	-2.7	4.4	26.9	187.2	13.3	0.20	0.50	0.00
2-63	1499	180	500	500	0	2	0	17.0	30	146	2400	-11.7	-2.9	-7.8	0.4	-2.9	4.2	28.8	190.5	11.9	0.17	0.96	0.01
2-8	1498	180	1600	1660	32	88	71	12.7	30	187	2400	-14.5	-5.8	-9.7	0.5	-4.6	4.0	31.5	178.5	NaN	0.05	4.19	0.24
2-9	1499	180	1600	1660	34	86	69	12.9	30	194	2400	-13.0	-4.4	-8.5	0.5	-3.0	5.7	33.0	177.2	0.6	0.19	5.84	0.45
2-69	1499	180	1300	1400	39	39	30	16.2	31	207	2400	-13.8	-5.2	-9.9	0.4	-4.3	4.1	29.3	180.1	0.5	0.13	6.08	NaN

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