

On the Effects of EGR on Spark-Ignited Gasoline Combustion at High Load

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ABSTRACT

EGR dilution is a promising way to improve fuel economy of Spark-Ignited (SI) gasoline engines. In particular, at high load, it is very efficient in mitigating knock at low speed and to decrease exhaust temperature at high speed so that fuel enrichment can be avoided. The objective of this paper is to better understand the governing mechanisms implied in EGR-diluted SI combustion at high load. For this purpose, measurements were performed on a modern, single-cylinder GDI engine (high tumble value, multi-hole injector, central position). In addition 0-D and 1-D Chemkin simulations (reactors and flames) were used to complete the engine tests so as to gain a better understanding of the physical mechanisms.

EGR benefits were confirmed and characterized at 19 bar IMEP: net ISFC could be reduced by 17% at 1200rpm and by 6% at 5000rpm. At low speed, knock mitigation was the main effect, improving the cycle efficiency by a better combustion phasing. At high speed, stoichiometric operation could be achieved, avoiding fuel-costly enrichment. It was also observed that the EGR tolerance decreases with the engine speed: maximum EGR rate was over 30% at low speed, vs. 22% at high speed. EGR composition was varied using synthetic EGR (N_2 , CO_2 or a mixture of N_2 & CO_2) in order to control the diluent heat capacity (C_p). These results, coupled with Chemkin simulations showed that the C_p value of the diluent is the main driving mechanism that explains the better auto-ignition resistance of the diluted air-gasoline mixture. The C_p of the diluent is also the 1st order parameter that controls the laminar flame-speed (S_L), chemical effects being of 2nd order and mainly attributed to CO_2 , which is implied in many elementary reactions of the fuel oxidation process. Then equilibrium constants can be modified when CO_2 concentration varies. However the relative importance between thermodynamic (C_p) and chemical effects of the diluent greatly depends on the ambient temperature and pressure conditions. A split-of-losses analysis was also performed. It helped to show that a diluent with a high molar C_p has a greater potential to re-phase the combustion, mitigating knock. A side-effect is a significant increase of thermal losses that almost completely balance the combustion phasing gains that can be obtained with a high- C_p diluent. Engine tests and simulations also showed that a minor species like NO, which is present in the EGR gases, can significantly affect the auto-ignition delay of the fresh gases. A maximum reactivity of the fresh gases was observed for approximately 100ppm NO in the intake air/EGR mixture.

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INTRODUCTION

Today's research on SI combustion is mainly focused on improving fuel economy. Among the different approaches that are studied, air/fuel mixture dilution is a promising way to improve SI engines efficiency. One privileged way to dilute the engine load is to use EGR. Another way is air-dilution or lean combustion.

One of the major challenges using lean combustion is NOx emissions. Indeed, the 3-way catalyst is not efficient anymore in lean operation: NOx cannot be reduced. A specific NOx after-treatment system is needed because engine-out raw NOx emissions are too high to be compatible with Euro 6 regulation. A kind of blocking point is encountered through the tradeoff between flame propagation and NOx emissions. Flame propagation cannot be sustained when the dilution rate

becomes too high, mainly because of the low temperature. However the minimum flame temperature that is necessary to enable flame propagation is still too high to ensure acceptable NOx emissions. As shown by Bourhis et al. [1], it is hard to achieve FAER (Φ) lower than 0.6 and NOx are still of a few g/kWh. Stratified operation could be used to increase air-dilution but as locally rich combustion occurs, high levels of NOx are also produced: ~10g/kWh [2]. To avoid NOx after-treatment, it would be necessary to decrease raw NOx emissions significantly under 1 g/kWh.

In addition, even if some researchers have obtained interesting results to lower knock thanks to air dilution at high load, it was shown that oxygen-free diluent is more efficient [10, 17, 22, 31]. For this purpose, combustion burn cases at stoichiometry are a

good candidate. This is what is used with cooled-EGR. (To avoid too high levels of compression end temperature, cooled EGR is used rather than hot EGR or internal residuals IGR. Those last ones would reinforce knock, rather than mitigating it).

To comply with Euro 6 legislation, EGR and $\Phi=1$ operation was chosen in this work. Many studies have shown that EGR enables significant fuel consumption reduction at low- [1], intermediate- [4] and high-load [4,5,6,7,8,9,10,16,17,19]. Then it is beneficial to use EGR whatever the operating point, and it should be used on the most part of the operating map [11,12,13,14,15,18]. As a result many studies claim fuel economy gains in the range of 5-10% on an NEDC cycle [12,13,18]. The magnitude of the fuel economy gain greatly depends on the reference engine that is taken for the comparison. Dauphin et al. [1] showed that dilution enables a reduction of wall heat losses and pumping losses, which are the main sources of efficiency losses at low load. On the other side of the operating map, at low speed / high load, cooled EGR can be used to mitigate knock. Combustion can then be advanced to achieve better CA50s. From Freeland et al.'s work [19], it can be deduced that 1° CA50 can be gained per 4/5% EGR. Impact on the efficiency is immediate: 8% gains in brake efficiency were obtained by Potteau et al. [8] and more than 11% in net indicated efficiency by Dauphin et al. via 9° CA50 re-phasing using 30%EGR at 1200rpm*14bar IMEP [1]. At high speed / high load, extra fuel enrichment can be eliminated [4,8,9,10,16], thus significantly improving fuel consumption. As knock limit and even low-speed pre-ignition phenomena can be pushed further with cooled EGR [20], it can be used to increase SI engine specific performances. This can enable either to further downsize the engines, which is favorable for fuel consumption, or to increase torque and/or power. Salamon et al. [22], and Böcker et al. [23] claim to achieve respectively 35bar and 30bar BMEP thanks to EGR.

Physical mechanisms activated by EGR-dilution can be classified into two classes. First is the dilution / thermodynamic effect. At constant engine load, adding EGR increases the in-cylinder mass. The fuel, whose molar heat capacity (C_p) is much higher than the one of other in-cylinder species, is then more diluted. Consequently, the molar C_p of the in-cylinder mixture diminishes, which implies increasing the ratio of specific heat γ (Table 1). Due to the larger γ , in-cylinder temperature is higher at the end of the compression stroke. However, the C_p of inert gases also acts as a heat sink, thus lowering fresh gases (also known as *end-gases*) temperature during combustion, which is favorable to lower knock. This effect was well illustrated by Hoepke et al. [21]. Combining earlier combustion phasing and heat sink properties, EGR is very efficient to lower exhaust temperature. The heat sink effect is also very efficient to lower wall heat losses. In addition a higher value of γ increases the Beau de Rochas' (also known as Otto's) cycle efficiency. Second effect is the chemical effect, which can be decomposed into pure kinetic impact and third-body reaction collision partners. Some minor species that enter the burnt gases composition can act as radicals promoters, especially OH radical, which is known to be very

active. Every species that will favor OH radical formation will favor auto-ignition (AI) and combustion speed. NO was identified as part of the EGR minor components that is the most prompt to influence the fresh gases mixture reactivity [24,25,26,27,28,29,30]. The influence of NO depends on the fuel, especially its negative temperature coefficient (NTC, [27]) behavior, and on the temperature and pressure history of the fresh gases [27,28,30]. In general, it seems that there is an optimum NO concentration that advances the knock onset in SI combustion with regular gasoline. This concentration lies around few hundreds of ppm. According to Hoffmeyer et al. [24], CO effect is of 2nd order compared to NO. Other researchers have found contrary effect, claiming that CO retards auto-ignition [31]. Unburnt HCs would have different effects according to their RON [24]. H₂ can have significant effects on knock and flame propagation. However it requires small quantities (few % or less), which are still too high to be naturally present in burnt gases for $\Phi=1$ combustion. Reforming [33, 34, 35, 36] or specific concepts such as D-EGR [32] can be used to increase the H₂ concentration in the combustion chamber. According to Anderlohr et al. [37], major species like H₂O and CO₂ are efficient third-body collision partners, and can possibly accelerate auto-ignition reaction kinetics.

The objective of this study is to better understand the physical mechanisms implied in EGR-diluted SI combustion at high load, especially concerning knock. Experimental work was carried out on single-cylinder GDI engine. In addition 0-D and 1-D Chemkin simulations (reactors and flames) were used to complete the engine tests so as to gain a better understanding of the physical mechanisms.

Table 1. Physical properties of gaseous species. (Fuel is RON 97 Gasoline).

	CO ₂	N ₂	Air	O ₂	H ₂ O _{gas}	EGR _{$\Phi=1$}	Fuel _{gas}
Cp [kJ/K/mol] @ 1bar, 25°C	0.0372	0.0291	0.0292	0.0294	0.0336 (at 100°C)	0.0307	0.134
γ [-] @ 1bar, 25°C	1.294	1.404	1.403	1.393	1.324	1.370	1.07
Density [kg/m ³] @ 1bar, 15°C	1.87	1.185	1.202	1.354			
Molar mass [g/mol]	44.01	28.01	28.95	32.0	18.02	28.88	90.9
Cp [kJ/K/kg] @ 1bar, 25°C	0.846	1.04	1.012	0.906	1.866	1.066	1.474

EXPERIMENTAL SETUP

Tests were performed on a modern, 350cc, single-cylinder GDI engine with a high level of tumble (see main specifications in Table 2). It is equipped with a 6-hole, 200bar injector, mounted in central position. Valve timing was modifiable with adjustable camshafts pulleys. The engine was operated with commercial RON97 fuel containing 5%_{vol} ethanol. All tests were performed at $\Phi=1$. Φ was computed from a standard 5-gas analysis of the exhaust, which means that in-cylinder EGR can differ

somewhat from Φ in case scavenging occurs, due to the possible excess of air in the exhaust. (The 5-gas analyzer sampling probe was located downstream the exhaust counter-pressure valve, approximately 3m downstream the exhaust runner). Intake air temperature was controlled and maintained at 40°C for all tests (whatever the EGR rate). Because using a real turbocharger on a single-cylinder engine would not be representative (too low exhaust flow, non-representative acoustic), the exhaust pressure was controlled as a function of intake pressure, by a throttle, to simulate a conventional turbocharger from a representative 4-cylinder TGD engine. (See in [Appendix 1](#) the impact of exhaust pressure on the results). All tests were performed at the knock-limited spark advance (KLSA). KLSA was determined through in-cylinder pressure filtering in the 4-9kHz band pass range. A threshold applied to the maximum amplitude of the filtered signal was calibrated according to the engine-operator experience to determine the knocking limit. Then a floating average was used to quantify the knock intensity. All operating points were set at the same value of knock intensity. The in-cylinder pressure sensor used was a water-cooled M10 AVL QC43D. See [Figure 1](#) for the location of the sensor.

A dedicated EGR circuit was used: high-pressure (HP) and/or low-pressure (LP) EGR routes were possible (see [Figure 2](#)). As shown in [Figure 2](#), a compressor was used to facilitate circulation of gases in the LP EGR route. In addition to *real* EGR, synthetic gases (N_2 and/or CO_2 , NO) could be used as diluent to simulate EGR and completely control the diluent composition. Whatever the nature of the diluent, the dilution rate was defined by [eq. 1](#). By default, a molar (volume) dilution rate is used. It can be checked that it is equivalent to the standard EGR rate when the diluent is *real* EGR. (EGR is calculated at the test bench using CO_2 as a tracer, from exhaust and intake CO_2 measurements. EGR rate is based on the ratio of EGR flow on (Air + EGR) flow, the fuel flow being excluded).

$$Dilution\ rate^{mol} [\%] = \frac{Q_{dil}^{mol} [mol/h]}{Q_{dil}^{mol} [mol/h] + Q_{air}^{mol} [mol/h]} \quad (Eq. 1)$$

(Q is the flow of diluent or air).

As it will be shown later, the Cp of the diluent has a significant effect on combustion. Therefore, it was useful to define an *equivalent Cp dilution rate* ([eq. 2](#)):

$$Dil.\ rate_{equiv.\ Cp}^{mol} [\%] = \frac{Q_{dil}^{mol} [mol/h] * Cp_{dil}^{mol} [J/K/mol]}{Q_{dil}^{mol} [mol/h] * Cp_{dil}^{mol} [J/K/mol] + Q_{air}^{mol} [mol/h] * Cp_{air}^{mol} [J/K/mol]} \quad (Eq. 2)$$

When a mixture of N_2 and CO_2 was used, the ratio 80%_{vol} N_2 / 20%_{vol} CO_2 was defined to mimic the molar Cp of real EGR, based on the following simplified combustion equation ([eq. 3](#)):

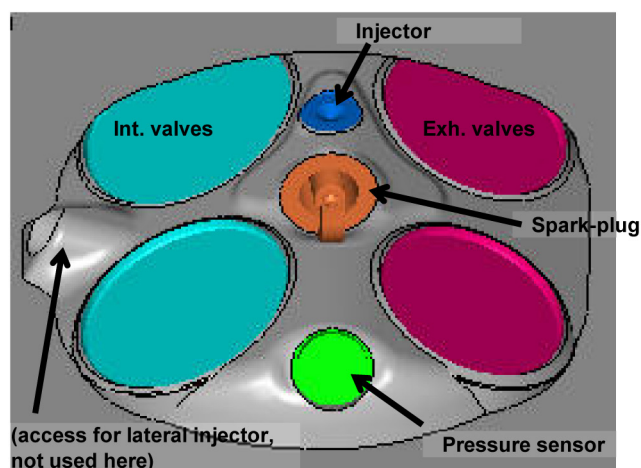
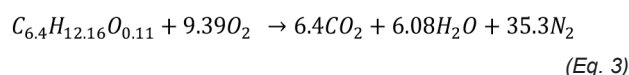


Figure 1. CAD view of the cylinder head (from the combustion chamber).

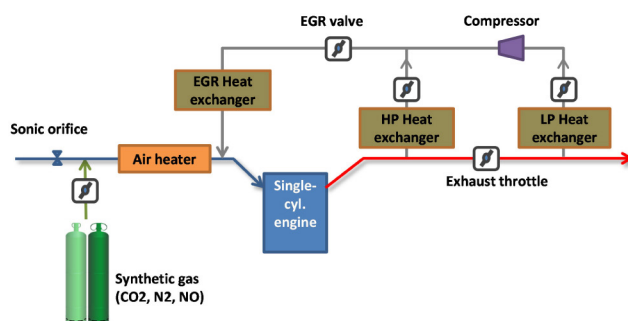


Figure 2. Single-cylinder engine. Air and EGR circuits.

Table 2. Characteristics of the single-cylinder engine.

Displaced volume	350 cc
Bore*Stroke	72.2 mm * 85.8 mm
Connecting Rod	132.2 mm
Compression ratio	10.5:1
Number of Valves	4
Exhaust Valves	190°C spreading, 9.5 mm max. lift EVC max / min: +21° -29°aTDC @ 1 mm lift
Intake Valves	211°C spreading, 9.5 mm max. lift IVO max / min: +20 / -30°bTDC @ 1 mm lift
Injection	Central position 6-hole, 200 bar injection pressure
Tumble	1.4 (IFPEN methodology, similar to NRT from Ricardo: ratio of air rotation speed / engine speed, measured with a "T" facility under 80mbar pressure drop)
Fuel	Commercial RON 97 (w/ 5% ethanol), conform to the EN228 regulation. See Appendix 2 for more details.
Coolant T°	Water T°: 90°C Oil T°: 90°C
Ignition system	Conventional coil, high-energy (150mJ) NGK spark-plug, heat range 7.

To stay in line with the objectives of this study, high-load operation was exclusively performed. Major part of the tests were carried out at 2000rpm * 19bar IMEP.

All the results given in this paper were calculated on a net indicated basis, meaning that pumping losses are taken into account (and according to the hypothesis that were made for the simulated turbocharging system).

To illustrate the combustion quality, combustion efficiency η_{comb} will be shown among the results (see eq. 4). It assumes that unburnt HC and CO are the only species in the exhaust which contain energy. The lower heating value of unburnt HC is supposed to be that of the fuel, and four times the one of CO.

$$\eta_{comb} [-] = 1 - \frac{is_{HC} - 4 \cdot is_{CO}}{ISFC} \quad (Eq. 4)$$

REACTORS AND FLAME SIMULATION

The effects of dilution on the laminar flame propagation and on the mixture auto-ignition were investigated thanks to 1D and 0D simulations using the CHEMKIN-PRO software [39]. A ternary mixture of iso-octane (42.9% in mole), n-heptane (13.7%) and toluene (43.4%) is used as a gasoline surrogate. Indeed, this surrogate showed to correctly reproduce the auto-ignition and the flame propagation of a commercial gasoline [40].

Laminar premixed flames at constant pressure was calculated using the detailed chemical scheme developed in the Idethanol Project of French National Research Agency (ANR) from which low temperature reactions were removed (these reactions are not involved in the laminar flame propagation). The resulting chemical kinetic scheme includes approximately 200 species, which allows relatively fast calculations. Simulations were performed for several fresh gases conditions with:

- Different sets of pressure / temperature representative of engine conditions,
- Several diluents: ideal burnt gases (i.e. the mixture of CO_2 , H_2O , N_2 given by a stoichiometric combustion, noted BG in the figures), N_2 , CO_2 and H_2O ,
- A variation of the dilution rate,
- A variation of equivalence ratio (from 0.8 to 1.2).

These simulations allow the determination of the laminar flame speed S_L which is proportional to the propagation speed of the turbulent flame front occurring in the engine.

The homogeneous reactors considered in this study are adiabatic and at constant pressure. The aim is here to determine the influence of dilution on the occurrence of auto-ignition, responsible for abnormal combustions as knocking. The chemistry of the gasoline surrogate is modeled using the chemical kinetic scheme of Anderlohr et al. [41], which accounts for low temperature reactions and includes 500 species. This kinetic scheme was specifically developed to reproduce the interactions between nitrogen oxides and gasoline. The simulated points include the same variations as

for the laminar flames concerning (p,T), diluent composition, dilution rate and equivalence ratio. In addition, the effects of an addition of CO and NO in the fresh gases are studied.

RESULTS AND DISCUSSION

EGR Rate Sweep & Valve Timing Optimization

At low speed / high load, it is of primary importance to minimize internal residual burnt gases (or IGR) for knock mitigation (reducing the in-cylinder temperature). Scavenging with fresh air is an efficient way to do it. This was performed optimizing intake and exhaust valve timings to ensure most advances CA50, hence maximum efficiency, at 2000rpm*19bar IMEP. The optimum values resulted in 20°CA overlap, with 0° EVC and 20°bTDC IVO (all values at 1mm valve lift). This means that scavenging was performed by early IVC, the EVC remaining close to TDC to maximize the expansion work while optimizing in-cylinder fresh air filling. As shown in Figure 4, compared to reference -10° aTDC EVC / -10°bTDC IVO (ie 20° negative valve overlap at 1 mm valve lift), scavenging valve timings enabled to gain 6° CA50 (and 10g/kWh net IS FC, at 0% EGR) due to lower knock. It could also be shown that boost pressure could be decreased by 200mbar, which is favorable for turbocharging specifications. However, scavenging concept was limited by exhaust UHC and CO emissions (respectively +1000ppm and +0.5% at 0% EGR), thus lowering the combustion efficiency. Because part of the fresh air was short-circuited from intake to exhaust during the valve overlap, in-cylinder FAER was higher than exhaust Φ , which was equal to 1 (to comply with 3-way catalyst after-treatment, which would be mandatory for a realistic multi-cylinder application). It was estimated from the authors experience that exhaust Φ should be decreased to approximately 0.95 to achieve in-cylinder FAER=1. In addition, we also observed slower end-of-combustions (CA90-CA50 increased by 5 to 10°) with scavenging, very probably due to the lower in-cylinder temperature at the end of combustion, due to the earlier combustion phasing and to the excess of fuel. It can be noted also that not only does IGR limit scavenging, but also in-cylinder enrichment is favorable to limit knock: the excess of fuel acts as a heat-sink.

Comparison of EGR sweeps with and without scavenging (see Figure 4) shows that for EGR rates above ~15%, scavenging didn't increase the net indicated efficiency anymore. Although CA50 was still more favorable, the relative gain compared to the non-scavenged case diminished as EGR rate increased. The longer end-of-combustion, the lower combustion efficiency and the higher low-pressure IMEP (50 to 100mbar) compensated for the combustion phasing gain. Gains enabled by EGR are proportionally more significant when there is no scavenging. It is assumed that cold EGR reduces the negative hot IGR effects on knock, the higher the IGR rate, the higher this effect.

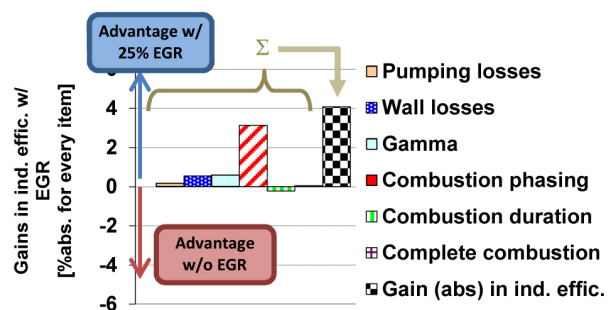


Figure 3. Split and losses analysis. Gains in net indicated efficiency using 25% EGR vs. 0% EGR at 2000rpm*19bar IMEP, 20° valve overlap.

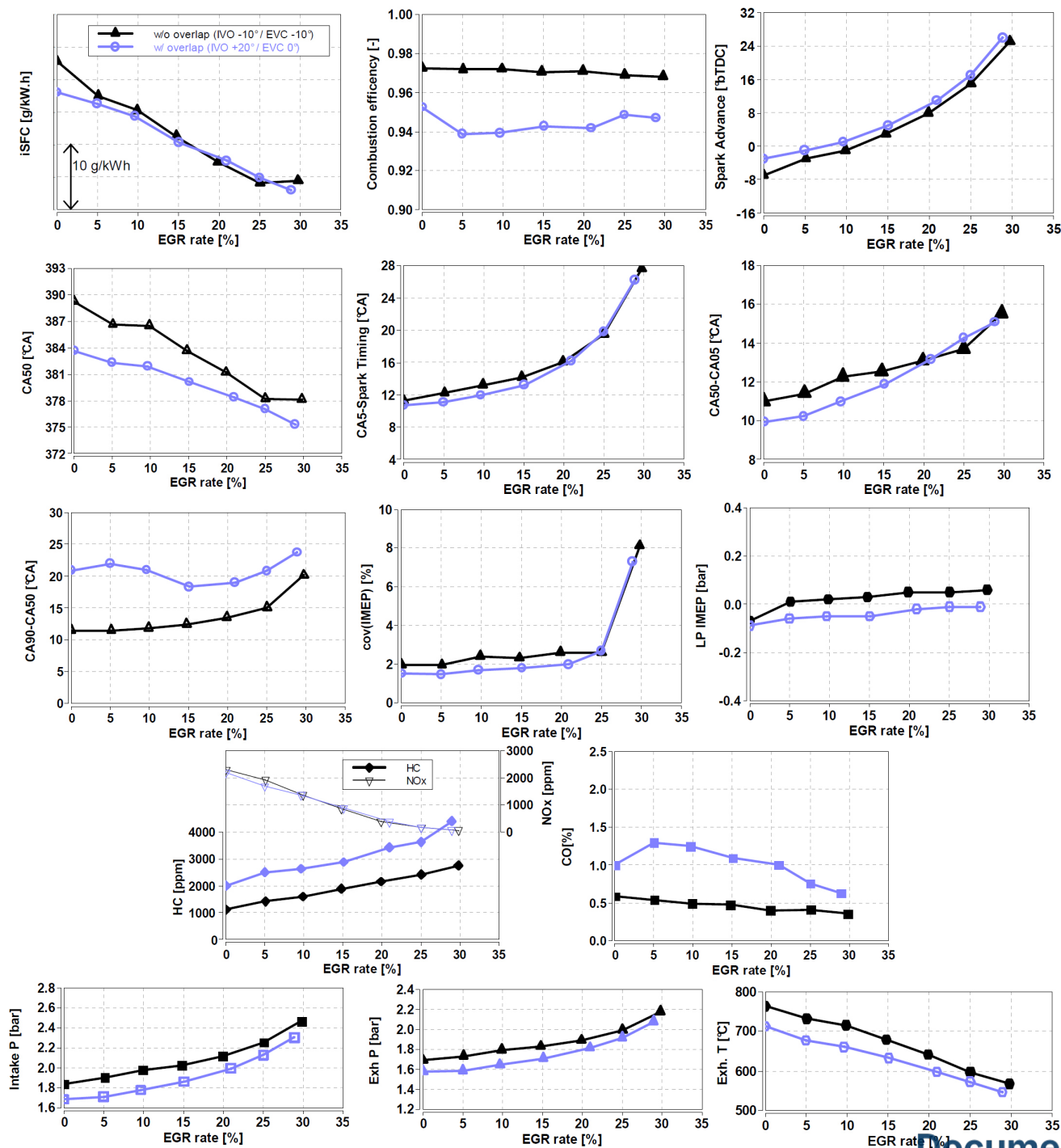


Figure 4. Impact of valve timing (scavenging) on EGR variation at 2000rpm*19bar IMEP.