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On the combination of High Pressure and Low Pressure EGR loops for improved fuel economy and reduced emissions in HSDI engines

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Abstract

In this paper, an experimental study of the combination of Low Pressure and High Pressure EGR architectures has been carried out. In the first part of the paper, the effects of both High Pressure and Low Pressure EGR architectures on engine behaviour and performance are analysed by means of a series of steady tests. In the second part, the effects of the combination of both architectures are addressed. The results shown that the Low Pressure configuration improves High Pressure EGR results in BSFC, NO_x and exhaust gas opacity, nevertheless the HC emissions are increased, especially during the engine warm up. In addition, the EGR rate that can be achieved with Low Pressure systems is limited by the low pressure difference between DPF outlet and compressor inlet, and the High Pressure system can be used to achieve the required EGR levels without increasing pumping losses. In this sense, the combination of both EGR layouts offers significant advantages to reduce emissions and fuel consumption to meet future emission requirements

Keywords. Diesel engine, air management, exhaust gas recirculation, EGR distribution.

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List of abbreviation:

BSFC	Break Specific Fuel Consumption
CAC	Charger Air Cooler
CO_2	Carbon dioxide
DI	Direct Injection
DPF	Diesel Particulate Matter
ECU	Electronic Control Unit
EGR	Exhaust Gas Recirculation
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
HPEGR	High Pressure EGR
HSDI	High Speed Direct Injection
MKI	Moduled Kinetics
NEDC	New European Driving Cycle
NO_x	Nitrogen Oxides
PCCI	Premixed Controlled Compression Ignition
PM	Particulates Matter
RoHR	Rate of Heat Release
VGT	Variable Geometry Turbine
WCAC	Water Cooled Air Conditioner

List of Symbols:

A	Valve effective area
$\dot{m}$	Mass flow

p	Pressure
R	Gas constat
T	Temperature
V	Volume
χ	Valve position
γ	Specific heat ratio C_p/C_v
π	Pressure ratio in valve

List of subscripts:

u	Upstream
HP	HP EGR
LP	LP EGR

1. Introduction

The increase in the specific power of DI Diesel engines during the last 20 years and their higher efficiency have allowed them to reach a favourable position in the market of light duty vehicles. In addition, as far as the Global Warming problem is concerned, their lower CO_2 emissions are an additional advantage for DI Diesel engines. Nevertheless, the environmental concern has also pushed governments to progressively reduce the vehicle emission limits. Despite the stricter emissions laws, road transport was responsible in 2006 for 17,7 % of all EU-27 greenhouse gases (GHG) emissions, apart from providing 40 % of NO_x, 36 % of CO and 18 % of non-methane volatile organic compounds emissions [1]. In this sense, the development of powertrains with low environmental impact is one of the most challenging problems to be faced during the next decades [2].

In the case of Diesel engines, particles and NO_x have been traditionally the most challenging pollutants to control. Regarding particles, while the air-to-fuel ratio control and the optimization of injection strategies were traditionally sufficient to avoid excessive particulate emissions, during the last years the use of DPF has been a widespread solution to fulfil emissions regulations. On the other hand, despite the development in NO_x after-treatment for Diesel engines, EGR remains a cost-effective solution to fulfil current and future NO_x emissions regulations. Also, despite they are not widespread, during the last decade several combustion modes such as HCCI (Homogeneous Charge Compression Ignition)[3], PCCI (Premixed Controlled Compression Ignition) [4] or MKI (Moduled Kinetics) [5] have been studied in order to strongly reduce particles and NOx emissions without BSFC damage and it should be noted that the EGR is a key technique to control such combustion processes [6, 7]. Therefore, according to the previous aspects it is expected that the EGR will continue playing a major role in DI Diesel engines.

Despite there exist different methods to reintroduce part of the exhaust gases in the engine cylinders, the external High Pressure EGR approach (HPEGR) is by far the most commonly employed EGR architecture in current engines [8]. It is known as external EGR because the burnt gas is extracted from the exhaust line to be introduced in the intake line. On the contrary, internal EGR consists on avoiding the exhaust of part of the burnt gas that stays in the cylinder to participate in the next combustion process. External EGR systems have prevailed due to their better flow and temperature regulation. The terms “High Pressure” refers to the gas is extracted from the exhaust manifold (upstream the turbine) and introduced in the intake manifold (downstream the compressor), in this sense the EGR line is at a higher pressure than ambient.

According to the system layout, the EGR rate is limited by the pressure difference between the intake and exhaust manifolds. In addition, since the turbocharger behaviour also depends on the intake and exhaust conditions, a strong coupling between the HPEGR and the VGT system appear. Other problems attached to the HPEGR systems are the important increment in intake temperature despite employing EGR coolers [9] and the difficulty to provide a homogeneous intake charge between cylinders [10]. The increase in intake temperature leads to a reduction in the admitted charge, by means of the thermal throttling effect reported in [11, 12]. Moreover, the short distance amongst the EGR inlet and the cylinder ports prevents from a uniform EGR distribution.

The Low Pressure EGR (LPEGR) technique is a topical subject because some of the problems of the HPEGR systems can be reduced by modifying the EGR layout [13, 14, 15]. The LPEGR method consists on extracting part of the exhaust gas from the DPF outlet and guiding it to the compressor inlet. At these conditions, the recirculated gas particle free and its temperature is lower than obtained at the turbine inlet. In addition, the coupling between turbocharging and EGR systems is reduced due to the fact that the whole exhaust gas flows through the turbine, and then the increase in EGR rate does not involve a reduction in the turbine available energy. The EGR rate with the LPEGR system does not depend on the intake and exhaust pressures, nevertheless, despite the fact that the compressor inlet pressure is always lower than the DPF outlet pressure, the pressure drop between those points is usually not high enough to reach the required EGR levels. In this sense, LPEGR systems require a backpressure valve at the DPF outlet (downstream the EGR extraction) or an intake throttle at the compressor inlet (upstream the EGR injection). Since the EGR is introduced at the compressor inlet, there is enough length in the intake line to achieve a perfect air and EGR mixture before the cylinders.

Of course, the LPEGR system involves some difficulties. Since the EGR goes through the whole intake line the use of DPFs is strictly necessary in order to prevent the exhaust particles from damaging the compressor wheel and also plugging coolers. With the widespread application of DPFs, the compressor and charge air cooler reliability problems with LPEGR have been reduced. Other problems of the LPEGR system are the condensation of species contained in the burnt gas along the intake line, and especially at the charge cooler. When the previous aspects have been addressed in preceding works [10, 15, 16], the objective of this paper is to evaluate the behaviour of the combination of both EGR layouts, taking into account three main aspects:

1. The ability to introduce mass into the cylinder assessed by the air mass flow and the EGR rate.
2. The potential for the pollutant emissions reduction, taking into account the effects of the EGR on both the composition and temperature of the intake charge.
3. The effects on engine performance, especially on BSFC, due to the differences on intake charge composition and temperature, but also the effect on pumping losses.

2. Experimental Set Up and Methodology

In the present paper, the study of the effects of the LP- and HPEGR combination on engine performance and emissions has been approached experimentally. The study has been performed on a state-of-art 2.0 litre HSDI Diesel engine. The engine, which main characteristics appear in table 1, was equipped with VGT, intercooler, DPF and a cooled High Pressure EGR loop.

As shown in figure 1 a Low Pressure EGR circuit was added to the original engine. An open code ECU was used to modify the engine calibration. Also, the engine was fully instrumented to measure temperatures and pressures in different interesting engine locations. The engine is installed in a test cell equipped with a variable frequency fast response dynamometer. Despite the dynamometer allows carrying out transient cycles, this research has been performed by means of steady tests exclusively.

A Horiba Mexa 7100 DEGR has been used to measure the intake CO2 concentration and exhaust gas emissions (NO_x, HC, CO, CO2 and O2). Both intake and exhaust CO2 have been measured by a non-dispersive infrared analyzer (NDIR). A heated chemiluminiscent detector (HCLD) allows measuring NO_x emissions. The HC analyzer consists of a heated flame ionization detector (HFID). In the same way, an AVL 439 smoke meter has been used to measure smoke opacity. Table 2 shows the instrumentation used in the engine cell.

Two engine operating points were defined for testing, they were selected from the conditions reached during the NEDC cycle in order to be representative of real operation. In this sense, the engine conditions achieved when a given vehicle reaches 100 and 120km/h during the NEDC were tested due to its important contribution to the NO_x and PM emitted over the entire homologation cycle. A summary of those conditions can be found in table 3.

For both operating conditions, tests consisted on performing a sweep of EGR rates combining the use of both LP and HP EGR systems, keeping the values of engine speed, injected fuel, CAC outlet temperature and intake pressure constant.

For each engine point, the EGR ranged from 0% (no EGR) to the maximum EGR rate the engine can admit. The maximum EGR rate limits were established by two different phenomena:

1. Unsteady combustion: An extremely high EGR rate leads to an unacceptable increase in BSFC due to the high delay in the combustion process.
2. Recirculated mass flow limited by the pressure ratio between the source and discharging points: In the case of the LPEGR system, the low pressure difference between DPF outlet and compressor inlet introduces an important limitation in the maximal EGR rate that can be achieved without auxiliary devices such as valves at the DPF outlet to increase the gas pressure at the EGR inlet.

2.1 HP and LPEGR flow estimation

The estimation of HP and LPEGR flows starts with the mass balance in the intake manifold:

$$\dot{m}_{air} + \dot{m}_{LP} + \dot{m}_{HP} - \dot{m}_{engine} = \frac{\partial m_{manifold}}{\partial t} \tag{1}$$

Assuming that the intake charge behaves as perfect gas, the previous equation can be written as:

$$\dot{m}_{air} + \dot{m}_{LP} + \dot{m}_{HP} - \dot{m}_{engine} = \frac{V}{RT}\frac{\partial p}{\partial t} - \frac{pV}{RT^2}\frac{\partial T}{\partial t} \tag{2}$$

Since the present study only considers steady state behaviour, the variations of both intake manifold pressure and temperature can be neglected, arriving to:

$$\dot{m}_{air} + \dot{m}_{LP} + \dot{m}_{HP} - \dot{m}_{engine} = 0 \tag{3}$$

The mass flow aspirated by the engine is calculated taking into account that the engine volumetric efficiency does not depend on the EGR rate. In this sense, the mass flow admitted by the engine at a given speed and load conditions is considered to be constant and is obtained from the test without EGR. In addition, a flowmeter located upstream the LPEGR introduction provides the air mass flow signal. The mass flow through any of the EGR valves can be modelled by the nozzle flow equation [17]:

$$\dot{m}_{egr} = \frac{A(\chi_{EGR})p_u}{\sqrt{RT_u}}\pi^{\frac{1}{\gamma}}\sqrt{\frac{2\gamma}{\gamma-1}\left(1-\pi^{\frac{\gamma-1}{\gamma}}\right)} \tag{4}$$

where χ_{EGR} is the EGR valve position, T_u and p_u are the temperature and the pressure upstream the valve, A is the valve effective area, which depends on the valve position, and π is the pressure ratio in the valve. Despite the production engines do not have the sensors required to calculate the EGR flows from equation (4), in the experimental facility used in this study they are available. Nevertheless, it should be noted that equation (4) provides a poor estimation of the LPEGR flow due to the very low pressure drop across the LPEGR system. In this sense, equation (4) has been used to calculate the flow through the HPEGR loop, while the LPEGR flow has been obtained from equation (3). In figure figure 2, the EGR rate measured with the gas analyser is plotted versus the proposed estimation. The estimated EGR rate is defined as:

$$EGR = \frac{\dot{m}_{LP} + \dot{m}_{HP}}{\dot{m}_{LP} + \dot{m}_{HP} + \dot{m}_{air}} \tag{5}$$

As can be observed in figure 2, there is a well correlation between the measured and the estimated EGR rate, nevertheless, it can be stated that there is a light overprediction of the proposed method.

3. Results

3.1 Considerations on the effects of HP- and LPEGR

One of the main differences between HP- and LPEGR systems lies in the conditions of the intake charge. When the LPEGR is used, the intake temperature is almost independent of the EGR rate, nevertheless, with the HPEGR an increase in the EGR rate leads to an inevitable increase in the intake temperature. The left part of figure 3 shows how the increase in the HPEGR rate from 0 to 30% produces an increase in the intake temperature of almost 80ºC. Since tests have been performed with constant intake pressure, the increase in temperature involves a 20% reduction in the intake density. Then, in addition to the substitution of part of the air charge by burnt gasses, the HPEGR produces an additional reduction in the air charge due to the density reduction ("thermal throttling"). In the right part of figure 3 the reduction in the intake O_2 concentration with the EGR can be observed. For a given EGR rate, the HPEGR system produces an additional reduction in the O_2 concentration due to the increase in the intake temperature.

As far as the EGR has effects on the intake charge properties, it affects the combustion process. The reduction in the intake O_2 concentration contributes to the increase in the ignition delay. In this sense, figure 4 shows a shift in the rate of heat release pattern in the cases with EGR. The higher ignition delay with EGR provides more time for the fuel to mix with the oxidizer, which increases the amount of pre-mixed fuel. Comparing the cases with EGR, it can be observed a higher rate of heat release with the LPEGR system. The increase of inlet temperature with HPEGR results in a decrease of the RoHR, because of the reduced in-cylinder gas density. Therefore, the introduction of diluents in the combustion chamber results in an increase in the combustion delay as well as a shifting of the combustion process towards the expansion stroke which involves a reduction in the indicated work. This degradation in the indicated work is intensified by the increase in the intake temperature.

To compute the global engine efficiency, the pumping losses, i.e. the work done to evacuate the exhaust gases and introduce the fresh charge into the cylinders, should be taken into account. For a given intake pressure, when a HPEGR system is used, communication between the intake and exhaust manifolds reduces the pressure difference between intake and exhaust manifolds as observed in figure 5 and then pumping losses. The thermal throttling also contributes to decrease the pumping losses since the flow through engine elements and trapped mass are reduced with increasing EGR. When the LPEGR is used, the pumping losses are almost constant due to the fact that the mass flow through engine elements and trapped mass are almost constant. Despite of the higher pumping losses, the higher mass flow through elements with LPEGR presents some advantages, specifically regarding turbocharging, the higher mass flow through the turbine allows the turbocharger to reach a higher speed, reducing the turbocharger lag during a sudden increase in load. In addition, the compressor operating points of the engine using the LPEGR system are located close to those of the engine without EGR, while compressor operational points using HPEGR are shifted towards the surge line.

Regarding emissions, figure 6 shows the evolution of exhaust gas opacity and NO_x concentration as the intake O_2 concentration is reduced with both LP and HPEGR configurations. As stated before, the main difference between both systems is that while the intake temperature with the LPEGR system has been kept constant (40ºC), it has suffered an important increase (from 40ºC to 115ºC) with the HPEGR loop. When the intake temperature is kept at 40ºC (LPEGR) the reduction in the intake O_2 concentration produces maximum exhaust gas opacity of 14% when the O_2 concentration reaches 17.5%. From this conditions, an additional decrease in the O_2 concentration leads to a reduction in exhaust gas opacity. According to the literature [18, 19], the lower the intake temperature, the higher the O_2 concentration from which the opacity starts to decrease with EGR. Also, for a given O_2 concentration, the higher the intake temperature, the higher the exhaust gas opacity. This behaviour is reflected in figure 6, where it can be observed how the exhaust gas opacity with the HPEGR system (high temperature) is higher than that obtained with the LPEGR architecture. It can be also noticed that with HPEGR, the reduction in the O_2 concentration involves an opacity increase in the whole tested range.

At the right part of figure 6 it can be seen that for a given O_2 concentration, raising the inlet charge temperature increases the exhaust NO_x concentration. In this way, the lower intake temperature with the LPEGR system allows an additional NO_x reduction to the effect of the O_2 concentration reduction.

3.2 HPEGR and LPEGR combination

In this section, the combination of both HP and LP architectures is analysed. In figure 7, the EGR rate measured at the different tested conditions versus the mass flows through both EGR circuits is represented. For both operating conditions the LPEGR rate is limited by the pressure difference between the DPF outlet and the compressor inlet. The LPEGR rates achieved in this study could be increased by using a backpressure valve in the exhaust line, after the EGR extraction, which increases the pressure difference in the LPEGR line. Nevertheless, this solution will increase the engine pumping losses and will have a negative impact in BSFC. On the contrary, the HPEGR rate is limited by an excessive reduction in the intake O_2 concentration, which involves an important increase in opacity and BSFC.

The differences in the intake temperature and O_2 concentration lead to differences in the engine behaviour. Results concerning engine performance are shown in figure 8. In this figure the trade off NOx-opacity is represented for both operating points. The color scale represents the BSFC, from dark (high BSFC) to light (low BSFC). The results obtained indicate that for a given EGR rate, the higher the LPEGR contribution, the lower emissions (points are moved towards the origin of coordinates. Also, for both operating conditions, the points with HPEGR show for a given NOx emissions a higher BSFC and opacity.

In this sense, for the tested conditions the combination of both EGR loops does not improve the performance of the LPEGR system. Nevertheless, other studies are required in order to explore possible benefits of the HP and LPEGR such as:

- The control of both intake charge temperature and composition. In spite of the benefits of the low intake temperature in terms of NO_x and opacity, the low intake temperature also involves higher HC emissions, especially during the warm up. In this sense, figure 9 shows the effect of the EGR on HC emissions during the NEDC, it can be observed that especially during the first phases of the cycle, where the engine is still cold, the HC emissions with the LPEGR system are considerably higher due to the lower intake temperature. As the cycle evolves and the engine warms up, these differences are progressively reduced. Since the benefits of the LPEGR configuration on NO_x mainly appear at the last phase of the NEDC, the HP-LPEGR combination allows taking profit of the lower HC emissions of the HPEGR configuration during the ECE, while reducing the NO_x emissions during the EUDC due to the use of the LPEGR architecture.

- The increase in the EGR rate without increasing pumping losses avoiding the use of a backpressure valve. In fact, the HPEGR can be used to increase the LPEGR rate up to the required levels.
- Avoiding high exhaust gas concentrations through the intake line that can produce condensation.
- At high speed and loads, where the mass flow is high and therefore pumping losses become important, the substitution of the LPEGR rate by HPEGR can improve BSFC.

4. Conclusions

In the present paper the effect of both LP and HPEGR architectures on engine performance and emissions has been analysed. The following conclusions have been obtained:

- The LPEGR loop allows increasing the EGR rate without increasing the intake temperature. The increase in intake temperature promoted by the HPEGR leads to a reduction of intake gas density and therefore on trapped mass. In this sense, the HPEGR reduces the intake O_2 concentration by substituting the fresh air (dilution effect) and also by reducing the trapped mass (thermal throttling).

- Due to the thermal throttling effect of HPEGR, for a given EGR rate, the intake O_2 concentration obtained with the HPEGR system is lower than that obtained with the LPEGR configuration.
- The reduction in the mass flow through intake and exhaust lines when using HPEGR leads a reduction of pumping losses once the EGR rate increases (keeping constant intake pressure). Nevertheless, this reduction also involves a lower mass flow through the turbine and then it forces less turbocharger speed. The lower turbocharger speed damages the engine transient performance during a tip in.
- The increase in intake temperature with higher HPEGR rates has a negative effect on BSFC, NOx and opacity.
- For the conditions evaluated in this study, the combination of LP and HPEGR systems does not involve any benefit comparing with the BSFC, NO_x and opacity obtained with the LPEGR system.
- A proper strategy to combine HP- and LPEGR systems can potentially improve simultaneously all kind of pollutant emissions, through the reduction of HC when the engine and catalyst are cold with HPEGR and the reduction of NOx and particulates when the engine is warm and the catalyst lighted on through the use of the LPEGR.

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TABLES

Table 1. Engine features

Architecture	4 L
Displacement	1998 cm3
Bore x Stroke	85 x 88 mm
Valves	4 / cylinder
Compression ratio	18:1
Turbocharger	Garret VNT GT 1749V
EGR	High Pressure Cooled
After-treatment	Oxy-catalyst + DPF
Max. Power / speed	100 kW - 4000 rpm
Max. Torque / speed	320 Nm - 1750 rpm
Injection System	Common rail-direct injection
Injector	Siemens 1980 - C0 clase 5 Piezoelectric - 6 holes
Max Injection Pressure	1800 bar

Tabla 2. Engine test cell instrumentation

Instrumentation	**Type**	**Model**	**Range**	**Unit**
Temperature	K Thermocouple		0 - 1260	ºC
Pressure	Resistive	PMA Transmitter P40	0 - 6	bar
Air mass flow	Hot - wire	Sensyflow DN80	0 - 720	kg/h
Fuel consuption	Fuel Balence	AVL - 733S	0 - 150	kg/h
Soot	Opacity	AVL - 439	0 - 100	%
Emission	CO2, Nox, CO, HC	HORIBA Mexa 7100	-	-
Turbocharger Speed	Attenaution by eddy currents	Picotum BM/SM	0 - 320000	rpm

Table 3. Tested conditions.

	engine speed (rpm)	fuel (mg/cc)	WCAC outlet T (ºC)	intake p (bar)	base line conditions (HPEGR)			
					EGR (%)	M (Nm)	NOx (ppm)	Opacity (%)
A	1870	16.3	35	1.066	23.6	72	123	21.6
B	2250	22.8	35	1.190	11.3	100	240	9.4

FIGURES

figure 1. Experimental set up.

figure 2. Measured EGR rate versus estimated EGR rate at A conditions. ○: LPEGR. ■: HPEGR. · :HP and LPEGR.

figure 3. Effect of the EGR rate on the intake temperature (left) and intake O_2 concentration (right) at A conditions. ○: LPEGR. ■: HPEGR.

Figure 4. Effect of the EGR on the Rate of Heat Release (left) and effect of the intake O_2 concentration on the BSFC (right) at A conditions. ···: w/o EGR ($[O_2]$=21%). -: LPEGR ($[O_2]$=17.5%). -: HPEGR ($[O_2]$=17.5%). ···: injection pattern. ○: LPEGR. ■: HPEGR.

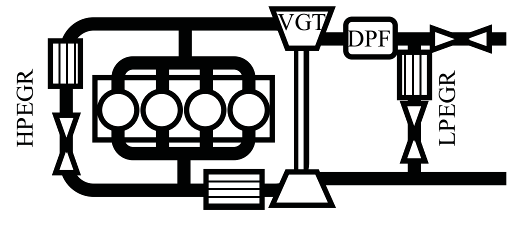
Figure 5. Effect of the EGR on the pressure difference between exhaust and intake manifolds (left) and effect on turbocharger speed (right) at A conditions. ○: LPEGR. ■: HPEGR.

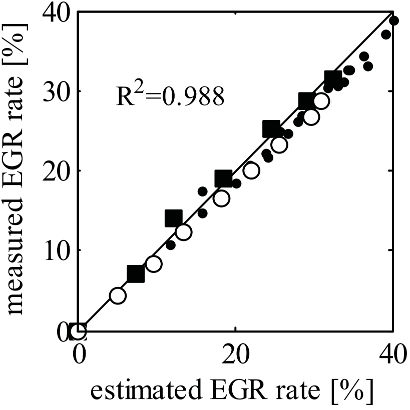
Figure 6. Effect of EGR on opacity (left) and NO_x concentration (right) of exhaust gases at A conditions. ○: LPEGR. ■: HPEGR.

figure 7. EGR rate as a function of the flow through HP and LPEGR Systems at A (left) and B (right) conditions.

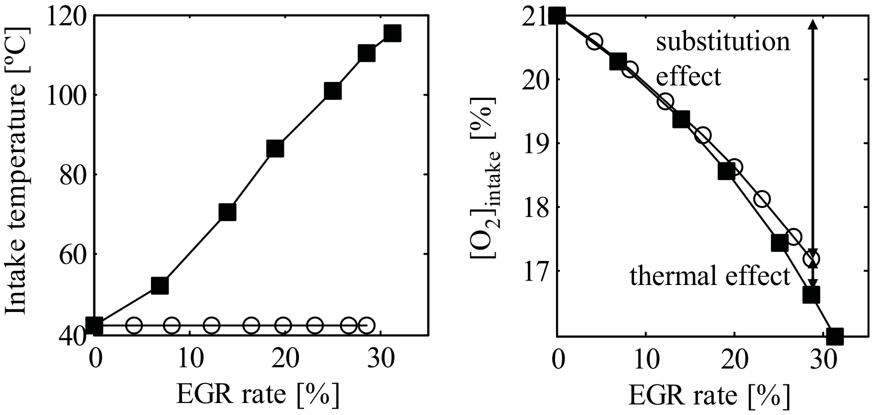
figure 8. Trade off NOx-opacity at A (left) and B (right) conditions. The colorscale represents the BSFC from low (white) to high (black).

figure 9. Effect of EGR on HC (left) and NOx (right) emissions during the NEDC. -: LPEGR. -: HPEGR.

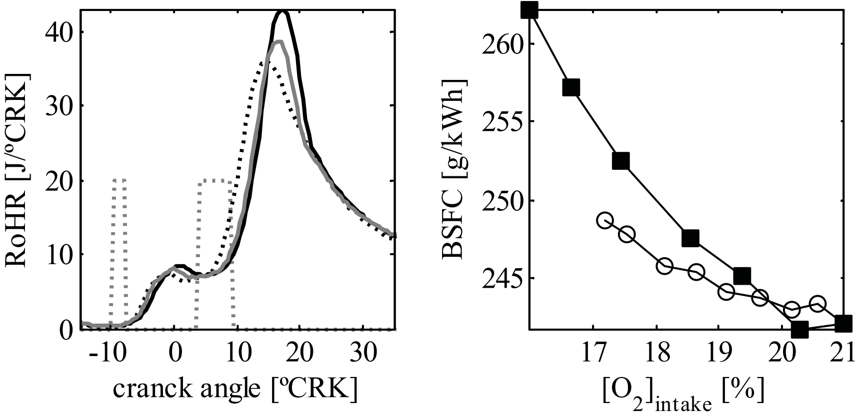

VGT
DPF
HPEGR
LPEGR



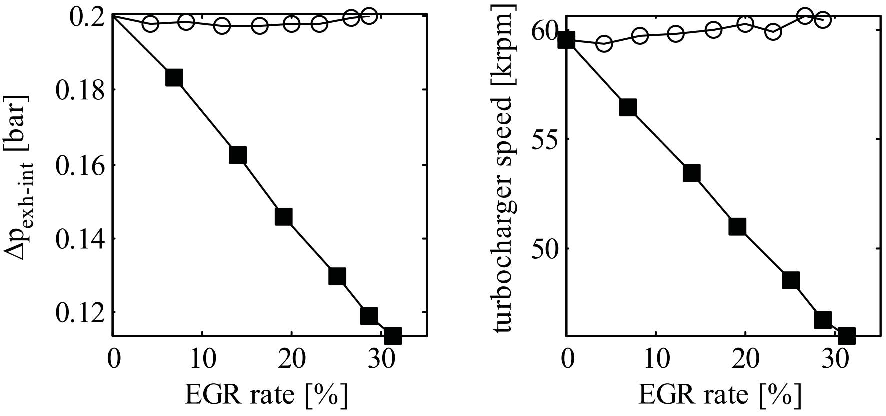
measured EGR rate [%]
estimated EGR rate [%]
R^2=0.988
0
10
20
30
40
0
20
40



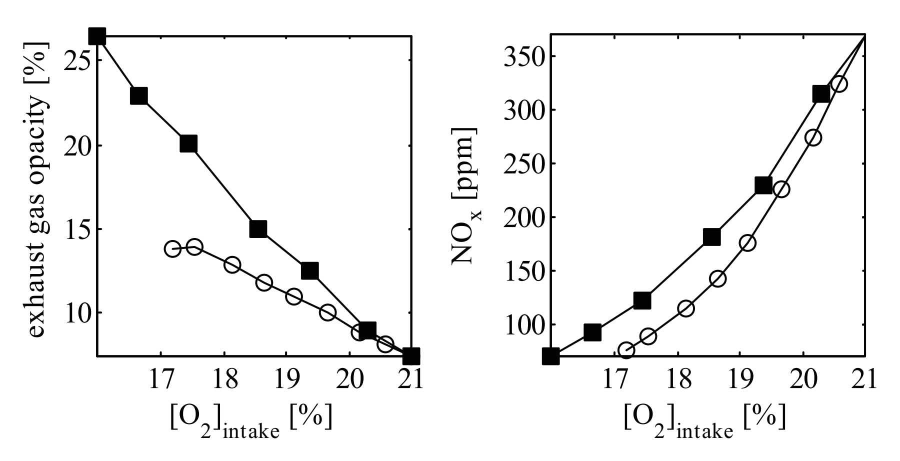
Intake temperature [ºC]
120
100
80
60
40
0
10
20
30
EGR rate [%]
$[O_2]_{intake}$ [%]
21
20
19
18
17
substitution
effect
thermal effect

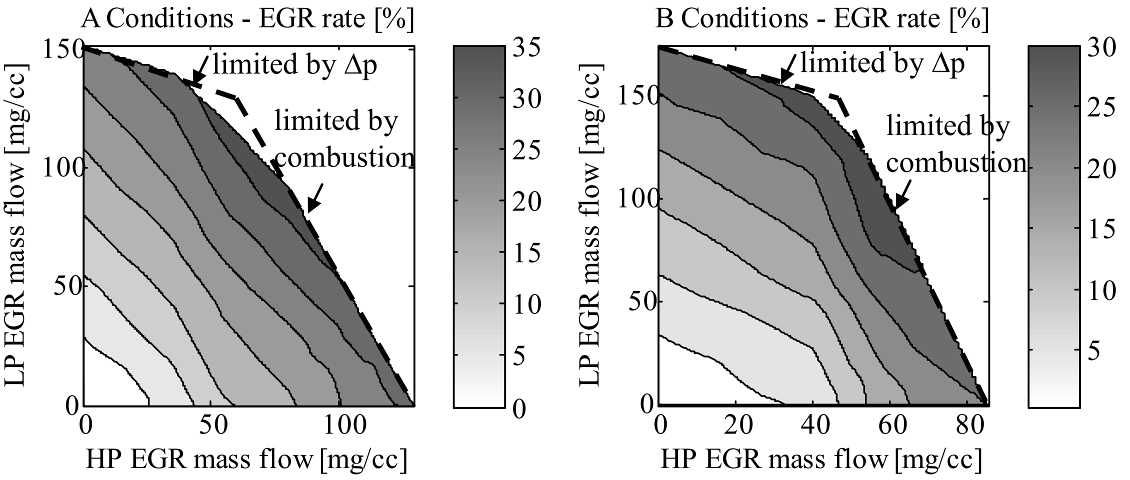


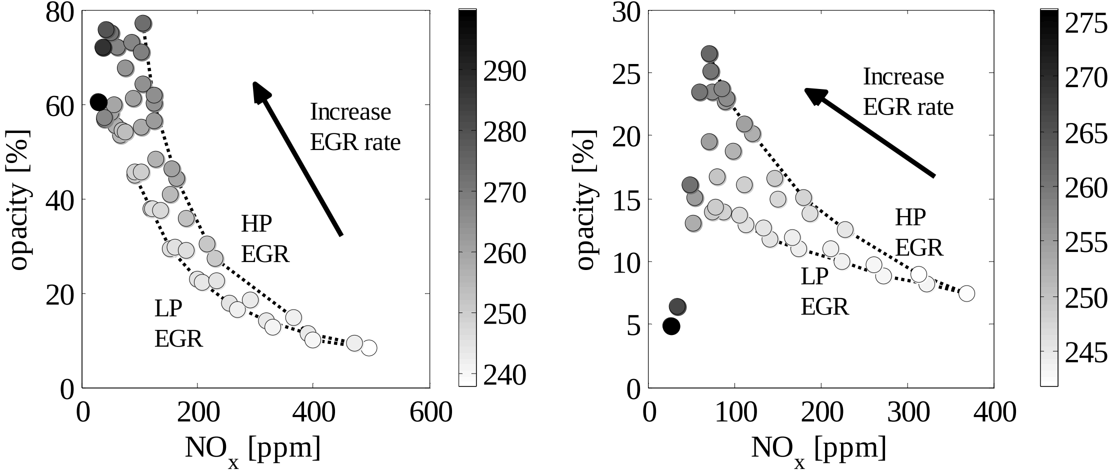
RoHR [J/ºCRK]
0
10
20
30
40
-10
0
10
20
30
cranck angle [ºCRK]
BSFC [g/kWh]
245
250
255
260
17
18
19
20
21
[O2]intake [%]

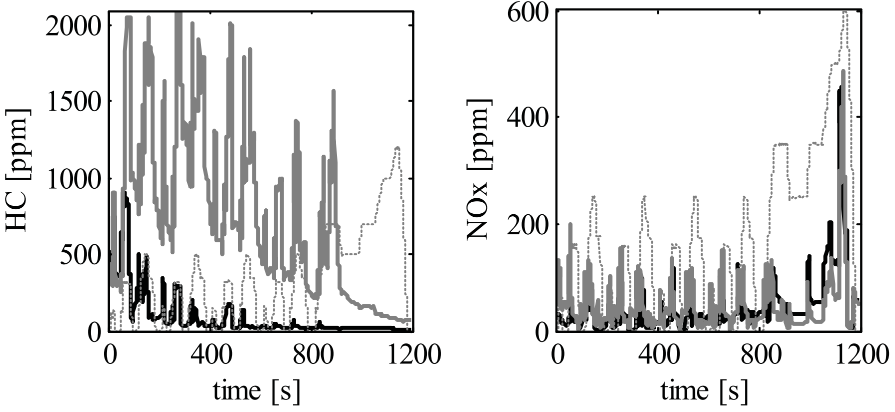


0.2
0.18
0.16
0.14
0.12
$\Delta p_{exh\text{-}int}$ [bar]
0
10
20
30
EGR rate [%]
60
55
50
turbocharger speed [krpm]
0
10
20
30
EGR rate [%]


exhaust gas opacity [%]
25
20
15
10
17
18
19
20
21
[O2]intake [%]
NOx [ppm]
350
300
250
200
150
100
17
18
19
20
21
[O2]intake [%]


A Conditions - EGR rate [%]
limited by Δp
limited by
combustion
LP EGR mass flow [mg/cc]
HP EGR mass flow [mg/cc]
150
100
50
0
0
50
100
35
30
25
20
15
10
5
0
B Conditions - EGR rate [%]
limited by Δp
limited by
combustion
LP EGR mass flow [mg/cc]
HP EGR mass flow [mg/cc]
150
100
50
0
0
20
40
60
80
30
25
20
15
10
5


opacity [%]
NO_x [ppm]
Increase
EGR rate
HP
EGR
LP
EGR
0
20
40
60
80
0
200
400
600
240
250
260
270
280
290
opacity [%]
NO_x [ppm]
Increase
EGR rate
HP
EGR
LP
EGR
0
5
10
15
20
25
30
0
100
200
300
400
245
250
255
260
265
270
275



HC [ppm]
2000
1500
1000
500
0
0
400
800
1200
time [s]
NOx [ppm]
600
400
200
0
0
400
800
1200
time [s]