Cylinder to cylinder HP EGR dispersion effect on opacity and NOx emissions in a diesel automotive engine

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Abstract

The objective of the study is to determine the effect of the high pressure exhaust gas recirculation (HP EGR) dispersion in automotive diesel engines in NOx and smoke emissions in steady engine operation. The investigation quantifies the NOx and smoke emissions as a function of the dispersion of the HP EGR among cylinders.

The experiments are performed on a test bench with a 1.6 liter automotive diesel engine. In order to track the HP EGR dispersion in the intake pipes, a valves system to measure CO_2, hence EGR rate, pipe to pipe was installed. In addition, a valves device to measure NOx emissions cylinder to cylinder in the exhaust was installed. Moreover a smoke meter device was installed downstream the turbine, to measure the effect of the HP EGR dispersion on smoke emissions. Five different engine speeds were studied with different torque levels thus the engine map was widely studied, from 1250 to 3000 rpm and between 6 and 20 bar of brake mean effective pressure (BMEP). The EGR rate variates between 4 and 25 % depending on the operation point.

The methodology was focused in experimental tools combining traditional measuring devices with a specific valves system, which offers accurate information about species concentration in both the intake and the exhaust manifolds. The study was performed at constant raw NOx emissions to observe the effect of the EGR dispersion in the opacity and fuel consumption.

The study concludes that when the EGR dispersion is low, the opacity presents reduced values in all operation points. However, above a certain level of EGR dispersion, the opacity increases dramatically with different slopes depending on the engine running condition. This study allows quantifying this EGR dispersion threshold. In addition, the EGR dispersion could contribute to increase the fuel consumption up to 3.5%.

1. Introduction

In the last years the focus is over diesel engines emissions, especially smoke but NOx too. During past decades a lot of technologies and strategies have been developed to reduce the emissions and comply the regulations. One of the most famous and popular strategy to reduce NOx emissions in diesel automotive engines is the EGR [1, 2]. As the regulation according on NOx emissions advances, the strategies become more complex [3] and the EGR rates increase [4]. The EGR inhibits the NOx formation mainly due to the reduction of the peak combustion temperature and the oxygen concentration as explained by Ladommatos et al. in 1996 and 1997 [5-8]. The benefits of EGR are observed on other applications like industrial natural gas engines [9]. However, the EGR strategies present some disadvantages such as the increase of smoke levels [10]. There is a compromise between smoke and NOx emissions, which are inversely related.

Moreover, the forecast in emissions legislation is the incorporation of new restrictions at low temperatures and high altitudes [11]. According to the new restrictions Xavier Tauzia et al. presented a detailed study in 2018 [12] related to the effect of cold starting temperatures, between 30ºC and 90ºC, over the efficiency because of the friction and the heat transfer during the thermal transient, moreover the combustion efficiency decrease due to higher CO and HC emissions al low load. Moreover pollutant emissions are highly effected because of the temperature variation of 60ºC where CO, HC and soot evolution is complex although the NOx decrease at low temperatures. Finally, low coolant temperature is more determinant than oil temperature which only affects to friction loses, nevertheless it should be considered too. From this point of view, the EGR strategies are very interesting to reduce the pollutant emissions, especially HP EGR to start the engine at low temperatures as J.M. Luján et al. studied in 2018 [13]. New homologation cycles including transient operation to approach to real engine working conditions like the Worldwide Harmonized Light vehicles Test Procedures cycles [14] and World Harmonized Transient Cycle. From this point of view, the EGR strategies must be optimized to adapt itself to these working conditions and comply the legislation [15].

EGR system is not the only strategy to reduce NOx emissions. Nowadays there are new after-treatment systems such as Selective Catalytic Reduction (SCR) based on Urea Water Solutions (UWS) which react with NOx in the exhaust when the solution is injected. The reaction reduces the NOx emissions notably. This system is adapting to the new requirements like the World Harmonized Transient Cycle, commented before. It is being applied to diesel engines in transient operations by a specific thermal management since the catalyst activity decreases significantly if the exhaust temperature is lower than 200 °C [16] or higher than 500 ºC. In addition, this system presents a problem of thermal aging because of the high temperatures of the exhaust gases. Moreover SCR is low effective in urban areas because the temperature in the exhaust is usually lower than 200 ºC.

Other problem presented by this system is the poisoning due to the sulfur content in the fuel, unburned hydrocarbons and Pt-Pd from the Diesel Oxidant Catalyst (DOC) and Diesel Particulate Filter (DPF) which is explained by Bin Guan et al. in a review of the state of the art technologies of SCR in 2014 [17] . More disadvantages are presented by this system like the deposits of urea and its byproducts during cold weather conditions and low exhaust temperature [18]. Furthermore this control penalizes the fuel consumption far than idle speed because of the increase of the backpressure. The control implies an increment of urea consumption too and it is mandatory to take into account the economic spending to load the tank with UWS. Finally this system presents economical disadvantages compared to EGR.

So, the development of new and more complex EGR strategies are necessary because of the reduction emissions requirement, the economic cost of the after-treatment systems and the problems commented before. Studies like Millo et al. in 2012 and Zamboni et al. in 2013 present results about the simultaneous use of HP and LP EGR as an effective combination to reduce NOx emissions and fuel consumption at low and medium speed and load conditions [19, 20]. In addition, HP and LP EGR have different consequences on the engine performance depending on the operation point [21] so both are necessary but the use of one loop or the other will depend on the needs. A recent updated study authored by Magín Lapuerta et al. in 2018 analyzed deeply the applications of LP EGR and HP EGR depending on the needs in a Euro 6 diesel engine equipped with DOC, a lean-NOx trap and DPF [22].

This study is focused in the HP EGR configuration. In one hand, HP EGR shows some disadvantages compared to LP EGR [23]. HP EGR reduces NOx emissions but penalizing the fuel consumption and the dispersion of the EGR among cylinders, as it was commented before, which increase the smoke and NOx emissions. In other hand, HP EGR presents some advantages versus LP EGR. HP EGR is faster than LP EGR because of the length of the line, which is closer to the intake manifold. It produces less HC emissions too and its efficiency is higher at cold conditions due to the increase of the temperature [24]. In addition, HP configuration needs less exhaust energy because the compressor operates under lower amount of gas. Moreover, the LP EGR presents troubles related to condensation issues at low ambient temperatures when it mixes with ambient air, this effect was recently studied by J. R. Serrano et al. in 2018 and J.Galindo et al. in 2019 to predict the condensation rate [25] and its effect in the rotor of the compressor [26] respectively. Nevertheless, at low ambient temperature, condensation issues can be appear in whole engine and in all its elements, in the HP EGR line for instance like J. M. Luján et al. described in 2019 [27].

The effect of the dispersion of HP EGR between cylinders on smoke and NOx emissions in automotive diesel engines is a topic to explore in depth. Long time ago emissions troubles related to unequal distribution between cylinders or bad mixing of the HP EGR were widely known and strategies to minimize that effect were developed by Robert et al. and William et al. in 2001 and 2002 respectively [28, 29]. Furthermore in 2009 Maiboom et al. performed a study about the increment of the NOx and PM emissions as consequences of an unequal cylinder-to-cylinder EGR distribution and increasing the EGR rate [30]. The efficiency related to the NOx-PM trade-off of LP and HP EGR system was studied too in 2010 by Maiboom et al. [31] and Payri et al. [32]. Nowadays it is evident that both of loops are useful depending on the situation and, although it was concluded that LP EGR has more advantages to reduce emissions, it was not very clear how much more.

In 2013 Lakhlani et al. [33] studied the cylinder to cylinder HP EGR dispersion with experimental and modelling tools in light-duty tracks and the effect on NOx-PM trade-off. The study compares the same operation points with and without HP EGR mixture. The study concludes that the effect of the dispersion in PM and NOx trade-off is significant in partial and high loads; however, the impact at low loads remains unclear. Moreover, the EGR mixture has a negligible effect on combustion. In 2015 Dimitriou et al. [34] performed an analysis of the cylinder to cylinder dispersion by 1D and 3D modelling tools. The study was focused in the parameters and effects that enhance the air-EGR mixing behavior like turbulence, Venturi, air-EGR contact area and the size and number of injection ports of EGR. The study concluded that velocity, turbulence, Venturi effect and a lot of small ports in the mixer improve the air-EGR mixing. An experimental study developed by Luján et al. in 2015 [35] demonstrated that the reduction of EGR dispersion is related with a significant reduction of NOx emissions and the effects on fuel consumption are negligible.

The main outcome of this work is to quantify the relationship between opacity and fuel consumption with the EGR dispersion in several engine running conditions in a state of the art diesel automotive engine. This information provides relevant information in order to design engine components to reduce the emissions caused by HP EGR dispersion. Experimental tools were used in this study, combining traditional and ad-hoc devices, to develop a methodology to promote different EGR dispersion levels so as to assess the impact in engine performance.

The paper is structured as follows: In Section 2 the experimental setup is explained. Section 3 contains main results and discussion in terms of dispersion and its influence on smoke and fuel consumption. Finally, main conclusions are presented in Section 4.

2. Experimental setup

The experiments are performed on a test bench with a turbocharged diesel engine. Table 1 shows the main features of the engine. The engine includes both LP and HP EGR systems. This study is focused only in the HP EGR loop of the engine, whose schematic layout is depicted in Fig. 1. The original configuration was modified since it is necessary to measure the CO_2 concentration in each intake pipe of the cylinders and control the dispersion of the EGR meticulousness.

Table 1

Engine specifications.

Cylinder number	In-line 4
Bore x stroke (mm)	80x79.5
Displacement (cm^3)	1600
Compression ratio	15.4:1
Valve number	4/cylinder
Fuel delivery system	Common rail. Direct injection.
EGR system	HP EGR and LP cooled EGR
Intake boosting	Turbocharger with VGT
Intake cooling system	Air charge air cooler (ACAC)
Maximum power (kW/rpm)	96/4000
Maximum torque (Nm/rpm)	320/1750

Firstly, a specific hardware, composed by valves and probes, was installed to determine the dispersion of the EGR rate in the intake and the dispersion of the NOx emissions in the exhaust. Up to nine probes were installed in the intake and five in the exhaust manifold. The probes were connected to a Horiba MEXA 7170DEGR, which is a conventional gas analysis system widely used in engine testing in steady conditions. Each probe had a valve that controls the gases flow to the gas analyzer. Usually, one of the valves was opened meanwhile the others were closed to measure only the gas composition from one pipe. During the measurement in the intake manifold, the valves of the exhaust were all closed and in reverse, during the measurement in the exhaust, the valves of the intake were all closed.

The configuration in the intake was one probe for each intake pipe, two per cylinder, and one to take air from the environment to perform the change of the acquisition between pipes. It is not possible to close one valve and open another after that because the acquisition system is breathing continuously and it would generate vacuum and it could break the measuring device. For this reason, it is necessary to open first another valve and then close the previous one. The ninth probe measured the ambient gas composition, so it measured a very low CO_2 level. The configuration in the exhaust is similar: five probes were installed in the exhaust, one per cylinder and one to measure the total value downstream of the turbine.

Furthermore, a specific manufactured device was installed between the HP EGR line and the intake manifold with three pipes and three regulation valves (one per pipe) to control the HP EGR dispersion. The pipes were distributed in the intake manifold to encompass it. One pipe was located on the left, other was located on the right and the third was located on the center, on the top of the manifold, as in the original configuration. Each pipe contained a regulation valve to control the HP EGR dispersion with high accuracy as it is observed in the following figure.

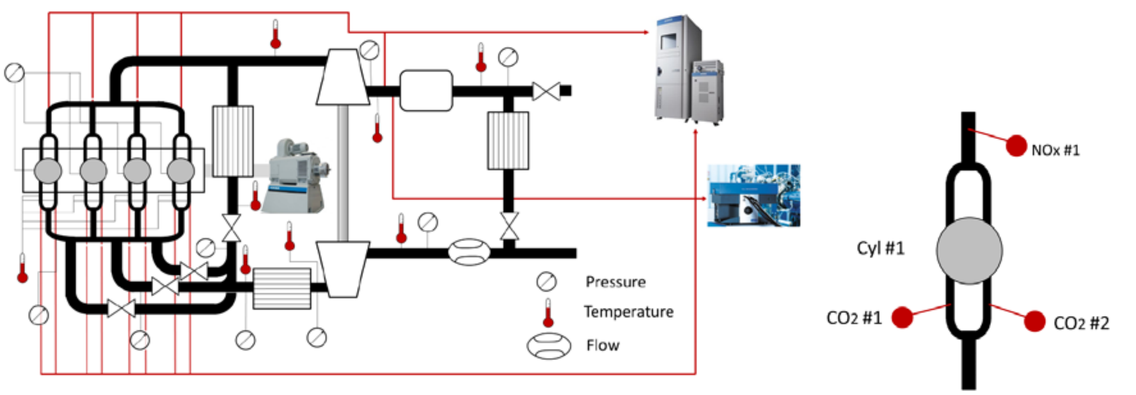


Figure 1. Engine schematic layout and emissions measurement detail in the cylinders

Different sensors were installed additionally to measure different engine parameters, which help to set the steady engine conditions to perform the tests. They also offer a lot of information to process properly the results. The variables together with the sensors features are presented in Table 2.

Table 2

Instrumentation accuracy

Sensor	Variable	Accuracy [%]	Range

Thermocouples type K	Temperature	1	0 ºC – 1260 ºC
Pressure sensor	Pressure	0.3	0 - 6 bar
Pressure sensor	Pressure	0.05	0-150 bar
Gravimetric fuel balance	Fuel mass flow	0.2	0 - 150 kg/h
Hot wire meter	Air mass flow	1	0 - 720 kg/h
Dynamometer brake	Torque	0.1	0 - 480 Nm
Smoke Meter	Soot	0.5	0 – 32000 mg/m3

Engine tests in steady conditions were performed to analyze the dispersion of the HP EGR and its effect on the NOx and smoke emissions. First, it was necessary to determine the operating points taking care to study a large range in the engine map. Initially, five different engine speeds were defined with different Brake Mean Effective Pressure (BMEP). After that, the EGR rate was determined under two restrictions: the first one was to study a large range between low and medium EGR rate, and the second one was to keep margin to be able to modify the HP EGR dispersion taking into account the effect on the engine stability, highly affected when the air to fuel ratio approaches to stoichiometric conditions. Finally, the EGR rate to perform the study was determined as a function of the emissions and the stability. The final engine running conditions are presented in Table 3.

Table 3

Working operation points

N [rpm]	BMEP [bar]	Intake P [mbar]	EGR rate [%]	Engine Torque [Nm]
1250	11	1500	6	140.1
1500	15	1900	5	191
2000	6	1250	23	76.4
2500	11.7	2300	20	149
3000	20	2750	11	254.6

In each operating point, the values shown in Table 3 had to be kept constant so the control strategy during the tests was based on three different engine controls at steady conditions. The first one was the engine torque, which was controlled by the injected fuel. The second one was the intake manifold pressure, which was controlled by the Variable

Geometry Turbine (VGT) position. And the third one was the NOx emissions which were controlled with the EGR rate, hence the HP EGR valve position.

The process to perform the tests was always the same for every operation point. The configuration of the three valves to control the HP EGR dispersion was set before starting the engine. After that, the engine was started and the operation point was set (speed, torque, intake manifold pressure and NOx emissions level). It was necessary to wait to stabilize the temperature of the engine to take real steady conditions. The testing procedure started with the acquisition of CO_2 concentration in the intake pipes and intake air sequentially. It required a few seconds to stabilize the measurement and obtain an accurate result, between ten and twenty seconds per pipe, depending on the working operation point.

After the registration in the intake line, the measurement of the NOx emissions in the exhaust manifold takes place. The procedure is similar to the previous one. Only one valve was open simultaneously except when the measurement probe was changed to avoid the vacuum effect. As in the intake, it is essential to wait a few seconds to stabilize the measurement. In this case, the duration is longer than before, between one or two minutes for each cylinder depending on the working operation point.

The opacity could be measured anytime, independently of the registration of other variables. It was measured with a traditional measuring device such as the AVL Smoke Meter 415S. It only needed a few seconds to take a precise measure each time. Horiba MEXA 7170DEGR is the equipment that measures the pollutant emissions, except opacity. It acquires the CO, CO_2, THC, O_2 and NOx concentrations. It is not possible measure the EGR rate directly, so it is necessary a conversion from exhaust and intake CO_2 concentration measurements [36] following the next expression:

$$EGRrate = \frac{[CO_2]_{Intake} - [CO_2]_{Ambient}}{[CO_2]_{Exhaust} - [CO_2]_{Ambient}} \tag{1}$$

3. Results and discussion

Once it has been explained the experimental tools, it is possible to analyze and discuss the results in terms of engine performance, emissions and HP EGR dispersion levels.

Fig. 2 shows the intake manifold pressure on the left and the engine torque on the right related to the nominal value presented in Table 3. The 2000 rpm engine operating condition is shown. The other operation points present similar results. The x-axis represents the standard deviation of the EGR following the next expression:

$$\sigma_{EGR} = \sqrt{\frac{\sum_{i=1}^{n}(x_i - \bar{X})^2}{n}} \tag{2}$$

Where:

i = the number of the cylinder.

n = 4 (total number of cylinders).

x_i = the average CO_2 concentration in cylinder i.

$\bar{X}$ = the average CO_2 concentration in the 4 cylinders.

It is possible to observe in the graph on the right the legend of the working operation points. The numbers represent the position of the three regulation valves of the EGR, where 0% means that it is completely closed and 100% is completely open. The number on the left represents the left valve before the manifold (in the sense of the flow), the number in the center represents the center valve and the number in the right represents the right valve, as depicted in Fig. 1. Over the dots, the EGR rate in each test is shown. It is possible to see that all of them are under 3% of variation, in both variables, intake manifold pressure and engine torque. In summary, during the testing campaign, although the HP-EGR dispersion was modified, all the tests were performed keeping nearly constant values of intake manifold pressure (by fine tuning the VGT opening) and engine torque (by modifying the injected fuel). For the sake of paper brevity, only the 2000 rpm results are shown.

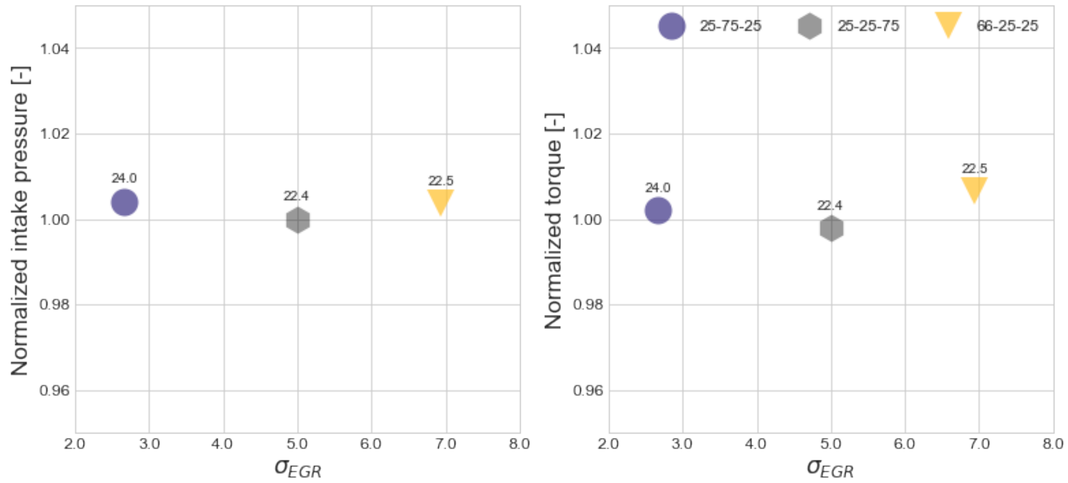


Figure. 2. Intake manifold pressure and engine torque referenced to the nominal value at 2000 rpm engine conditions.

Fig. 3 presents the NOx values related to the average value for each engine running condition, measured in the exhaust manifold to take the information per cylinder (numbered from 1 to 4) and downstream the turbine to measure the raw NOx emissions value (named T). The four plots rows correspond to the 1500, 2000, 2500 and 3000 rpm engine running conditions. The x-axis represents the EGR rate on the plot on the left, and the EGR standard deviation on the right. The left plot shows the effect of the EGR in the NOx emissions, cylinder to cylinder, and in the engine. The plot on the right shows the engine raw NOx as a function of the standard deviation of the EGR. The average EGR rate is labelled above the dots.

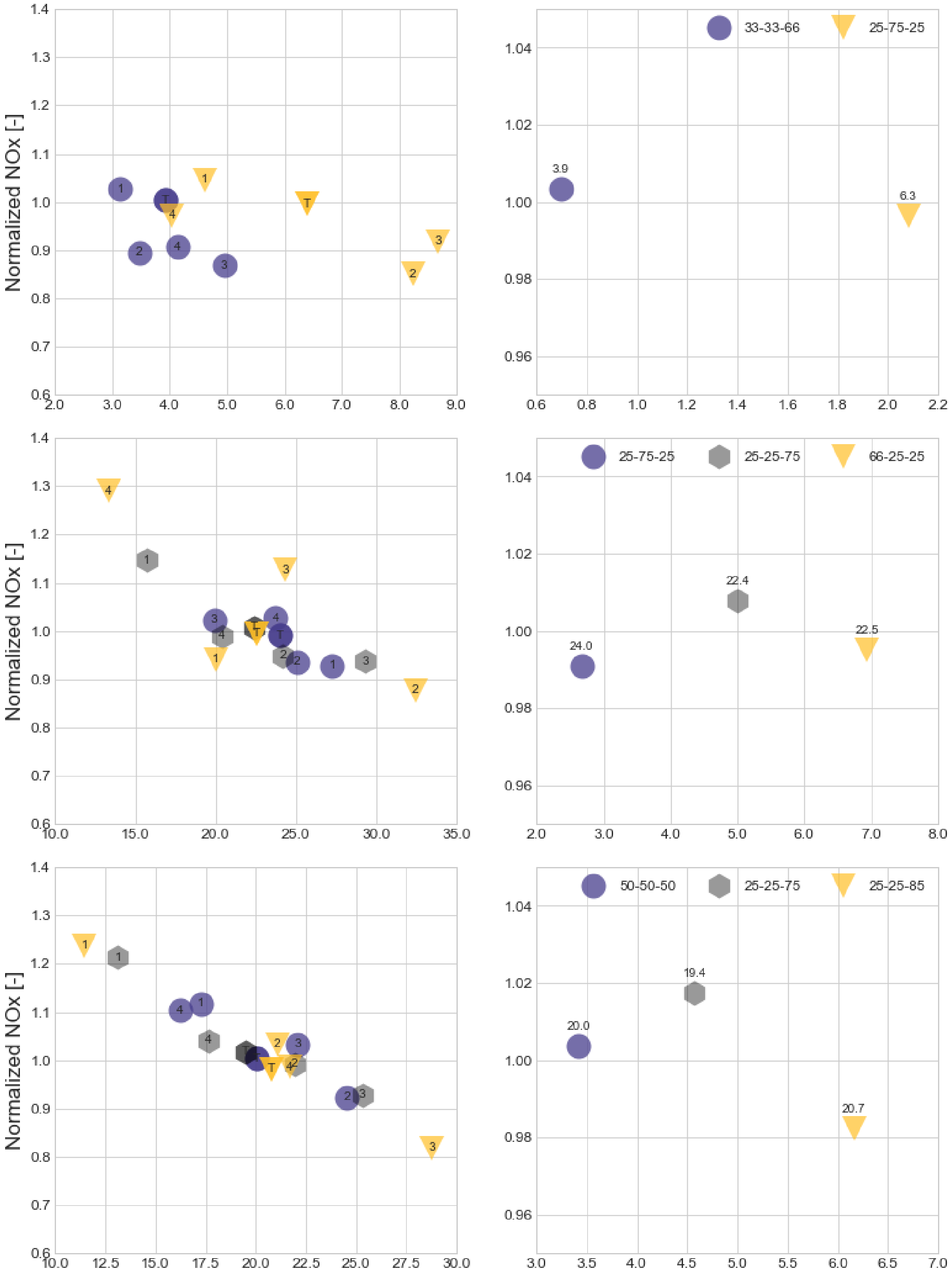
Paying attention, for example, to 3000 rpm working operation point it is possible to observe that the case with the lowest standard deviation (50-100-50) has a very similar EGR rate value in each cylinder. From left to right, in the sense of the flow, the cylinders 1, 2 and 3 have an EGR rate very close to the nominal value, 11%, and cylinder 4 has a little bit more, close to 12%, what represents an increment of the 9% approximately referred to the nominal value.

The second case (25-50-100) presents a higher value of standard deviation where the cylinder 1 and 2 has 8% and 8.5% EGR rate, the cylinder 3 has 12.5% and cylinder 4 has 15.5% EGR rate. In this case, the minimum and the maximum EGR rate represents a reduction of 27% and an increment of 41% respectively.

In the last case, with the highest standard deviation, cylinders 1 and 2 have similar values of EGR rate than the second case (between 8% and 8.5%), cylinder 3 has 16.5% and cylinder 4 close to 18.5% EGR rate. Cylinder 4 shows the most dramatic increase of the EGR rate, cylinder 4 presents an increment of 68% referred to the nominal value (11%). Moreover, particularly in this case, 15-50-100, with the highest standard deviation, the average EGR rate is slightly higher than nominal value, 12.8% instead 11%. Even with this increase in the average EGR rate, 12.8%, the increment in cylinder 4 is 44% higher, which is the highest difference in any case at 3000 rpm. It is possible to observe that a higher standard deviation means a large difference in terms of EGR rate between cylinders. In this case, the difference between the lowest and the highest EGR rate per cylinder is close to 10 points, from 8% on cylinder 1 to slightly more than 18% on cylinder 4.

Although each cylinder and each case presents different values of EGR rate and standard deviation respectively, all the cases were performed at constant NOx level for the same working operation point. In some cases, it was necessary to increase the EGR rate, to keep the same NOx value. An example was commented before, where the third case at 3000 rpm (15-50-100) reveals an average EGR rate of 12.8% instead of 11%. While the two cases with less EGR standard deviation presented a similar value to the nominal setup.

This effect is observed too at 1500 rpm working operation point and it is not at 2000 and 2500 rpm. 3000 and 1500 rpm operation points have in common a low average EGR rate (lower than 23% and 20% presented by 2000 and 2500 rpm operation points respectively). In all the situations, a 2% difference in NOx level is accepted when controlling the EGR valve at the engine operating condition between the different dispersion valves configurations.


Normalized NOx [-]
33-33-66
25-75-25
3.9
6.3
Normalized NOx [-]
25-75-25
25-25-75
66-25-25
22.4
24.0
22.5
Normalized NOx [-]
50-50-50
25-25-75
25-25-85
19.4
20.0
20.7

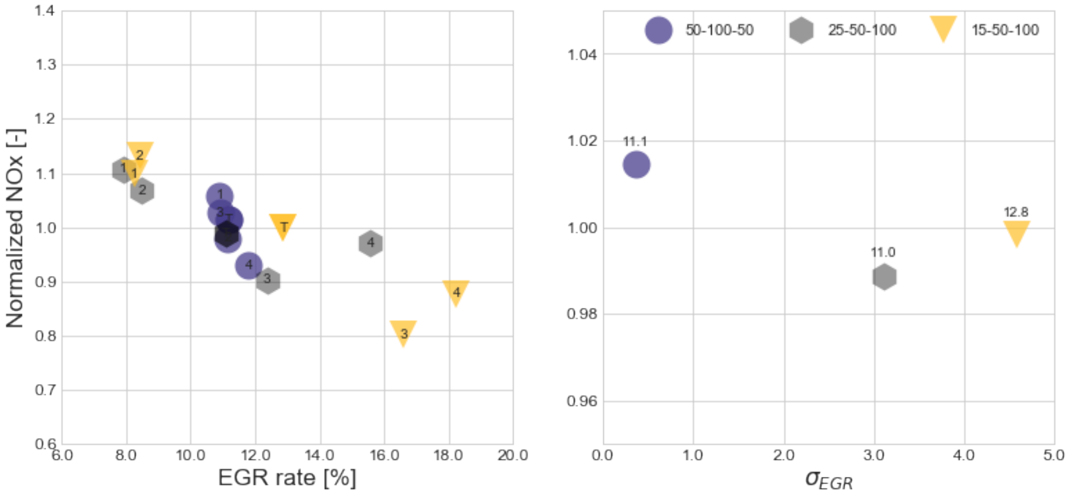


Figure 3. Cylinder and engine NOx values vs. EGR rate (left), and engine raw NOx vs. EGR standard deviation (right) for the 1500, 2000, 2500 and 3000 rpm operation points (from top to bottom).

In order to check the influence of the EGR dispersion in the in-cylinder pressure traces, Fig. 4 shows several consecutive engine cycles measured in all cylinders. In this particular case, the 3000 rpm situation is plotted with three different HP EGR dispersion levels, as the legend shows in the top of the figure on the right. The numbers represent the position of the regulation valves to control the dispersion as explained before.

It is very difficult to derive any conclusion from direct in-cylinder pressure inspection because the p-V diagram is practically the same in all cases. It is not possible to appraise significant differences between case and case and between cylinder and cylinder due to the negligible differences, so the calculation of the Indicated Mean Effective Pressure (IMEP) values are provided in Fig. 5 for several cases.

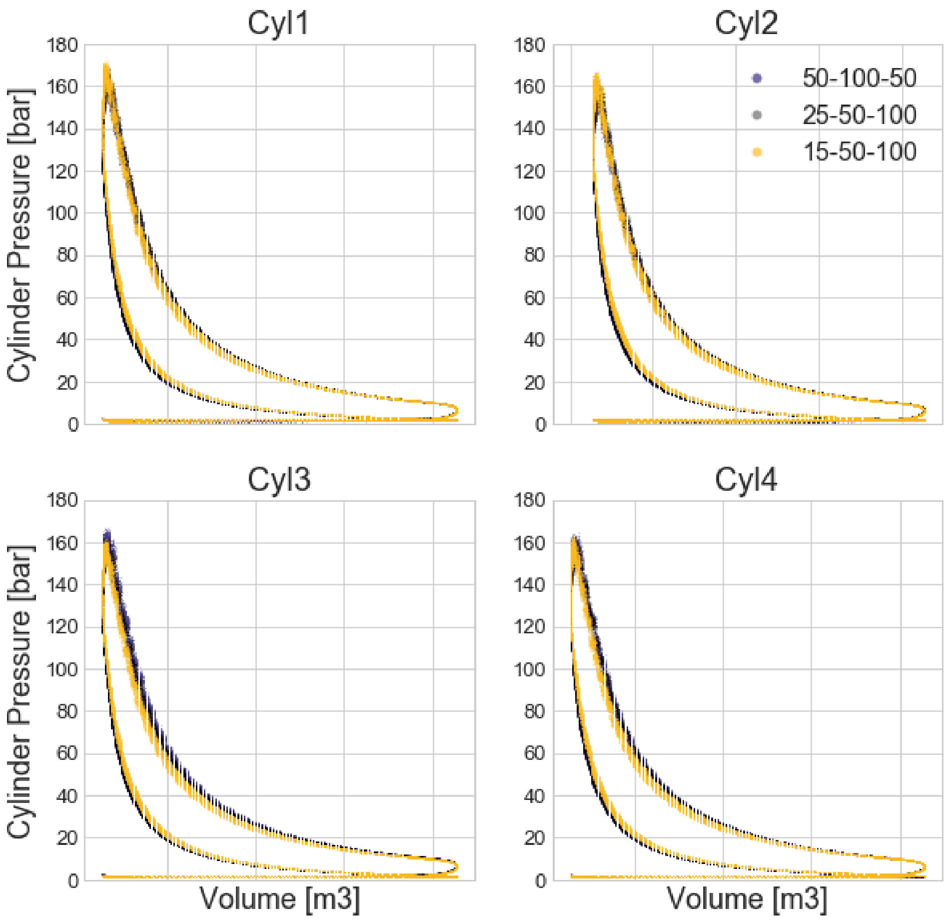


Figure 4. In-cylinder pressure vs cylinder displacement for all the cylinders and three different dispersion levels at 3000 rpm.

Fig. 5 shows the IMEP (dimensionless with the average) for all the cylinders (numbering from 1 to 4) and the average (named T) as a function of the EGR rate at 3000 rpm, as in Fig. 4. It is possible to observe that the dispersion affects to the IMEP of the cylinders (individually). Since the tests are performed keeping the engine torque constant, the average of the IMEP is not modified. In addition, the IMEP variation range, per cylinders, is lower than 5% most of times. Therefore, it is possible to affirm that the influence of the EGR dispersion to the IMEP of the cylinders is very limited.

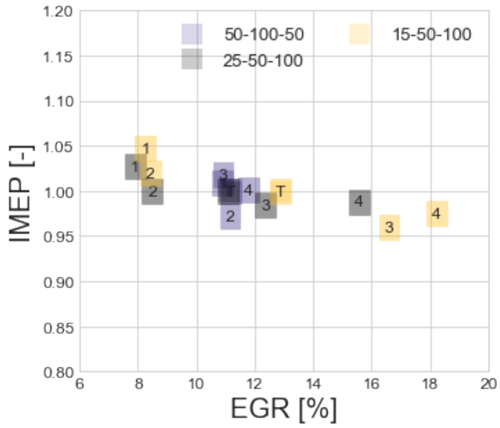


Figure 5. Dimensionless IMEP for all the cylinders (numbered from 1 to 4) and the average (named T) versus EGR rate for three different HP EGR dispersion levels at 3000 rpm.

Fig. 6 shows the air to fuel ratio measured in the exhaust gases (i.e. lambda), the adimensional brake specific fuel consumption (BSFC) that is calculated as the ratio between the BSFC and the BSFC achieved with the lowest EGR dispersion at the given engine running condition, and the exhaust gases opacity as a function of the EGR dispersion in all the engine running conditions (1250, 1500, 2000, 2500, 3000 rpm) and all the dispersion levels.

A reduction on the lambda values is observed as EGR dispersion increases, which can be explained mainly by two reasons: either changes in the air mass flow entering the cylinders or variations in the injected fuel. As already discussed in Fig. 3, an increment of EGR rate was necessary to maintain the NOx nominal value in the 1500 and 3000 rpm engine running conditions. The difference between these working operation points and the other, 2000 and 2500, was the EGR rate. The former points have lower EGR rates, around 5-10%, while the latter points have higher EGR rates, over 20%. On the other hand, regarding the fuel injection, more fuel quantity is needed when the EGR dispersion increases in the 2500 and 3000 rpm running conditions, which is easily detected in the BSFC plot. This means that the impact of the EGR dispersion on the combustion performance is more remarkable in these engine running conditions.

In Fig. 6 another difference related to lambda value is detected: the engine running conditions at 1500 rpm and 3000 rpm are close to engine full load operation and present a lambda value lower than 1.25, while the other working operation points, 2000 rpm and 2500 rpm, present values over 1.3, corresponding to partial load situation. When the engine approaches to stoichiometric conditions the combustion process in the cylinder is more critical.

The 1250 rpm running condition, with two different EGR dispersion, needs to increase the EGR rate, as the 1500 and 3000 rpm engine points, to keep the same NOx level. These two points have an EGR rate of 6.2% and 7.1% in the low and high EGR dispersion level respectively. The EGR absolute value is not very high but the difference between both of them is considerable, around 14.5%. Moreover, this working operation point presents low lambda values too, close to 1.2, since it is also close to full load operation. Therefore, this point is very similar to 1500 and 3000 rpm operation points due to the low EGR rate, the need to increase the EGR rate to keep the NOx level constant and the low lambda value.

On the other hand, the graph of the center presents a clear trend between the increment of BSFC and the increment of the EGR dispersion. The 3000 and 2500 rpm running conditions show the highest increment of the BSFC: 2.7%, from 1 to 1.027, and 3.5%, from 1.01 to 1.045, respectively. The 1250 and 2000 rpm running conditions present more modest increment than the other: 1.2%, from 1.124 to 1.138 and 1%, from 1.062 to 1.071, respectively. Finally, the impact of the EGR dispersion on the specific consumption at 1500 rpm working conditions is very limited, at least in the range of the tested EGR dispersion.

To finish, the graph on the right shows a scatter plot where opacity increase significantly with the EGR dispersion. At 1250 rpm the opacity is 50% higher in the highest EGR dispersion dot than in the lowest, from 16 to 24%, but it is necessary to take into account that this effect could be not only caused because of the EGR dispersion but because of the increase of the EGR rate and the low lambda value too.

The data for 1500 rpm shows a very low increment between the two cases. It is the working operation point with the lowest EGR standard deviation. Nevertheless, it presents an opacity increment of 14% (from 7 to 8 %) but it is difficult to observe using the same y-axis scale than for the rest working operation points. In this case, as in 1250 rpm, the absolute value of the EGR rate is low but there is an increment of EGR rate to keep NOx at constant level. As it was commented before in Fig. 3 where it was shown that the point with higher dispersion operates with an EGR rate of 6.3%, while the one with lower dispersion does with 3.9% EGR rate, which means an increment of 62%. In spite of the high increment of the EGR rate and the low lambda value, they have not big consequences in the opacity like at 1250 rpm.

The third case, 2000 rpm, is the first where the EGR rate does not increase with the EGR dispersion to keep the NOx level. However, it is possible to see a very clear trend: higher EGR dispersion between cylinders correlates with higher opacity levels. The opacity moves from 29% in the lowest EGR dispersion test to 42% in the highest case, which represents an increase around 45%.

The most extreme case correspond to the 2500 rpm engine running conditions where the opacity increase up to 70%, which means an increment close to 75% compared with the lowest opacity test. In fact, in addition to the high EGR rate that the engine is using in this point, the EGR rate in the cylinders presents huge differences, from 12% to 28%, higher than the lowest opacity point, which shows variations from 15% to 25% approximately.

Finally, the 3000 rpm case also presents high opacity values. As it was commented before, the engine runs with a low lambda value in this operation point. In the other hand the EGR rate of the highest EGR dispersion test is higher, 12.8%, than the others, 11% which could influence on the increase of the slope between the mid and high EGR dispersion tests, which translates into an increase of 150% among the lowest and highest EGR dispersion points.

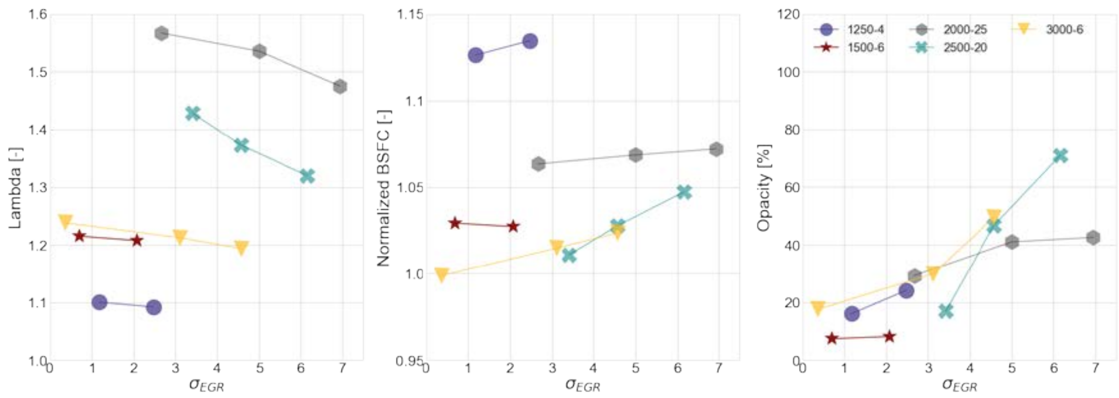


Figure 6. Lambda (left), BSFC (center) and opacity (right) vs EGR dispersion.

Fig. 7 gathers all the results in a single plot. Y-axis represents an adimensional opacity that is calculated as the ratio between the opacity and the opacity achieved with the lowest EGR dispersion at the given engine running condition. X-axis represents an adimensional EGR dispersion index called EGR dispersion coefficient, which is calculated as the ratio between the EGR standard deviation and the EGR rate. Over the dots appears the actual opacity value in percentage. It is possible to realize that a remarkable increase in the adimensional opacity appears for EGR dispersion coefficient values higher than 0.2. Y-axis scale shows that, depending on the engine running condition, the EGR dispersion phenomenon could multiply the opacity level for the lowest EGR dispersion by a factor around 1 and 4.

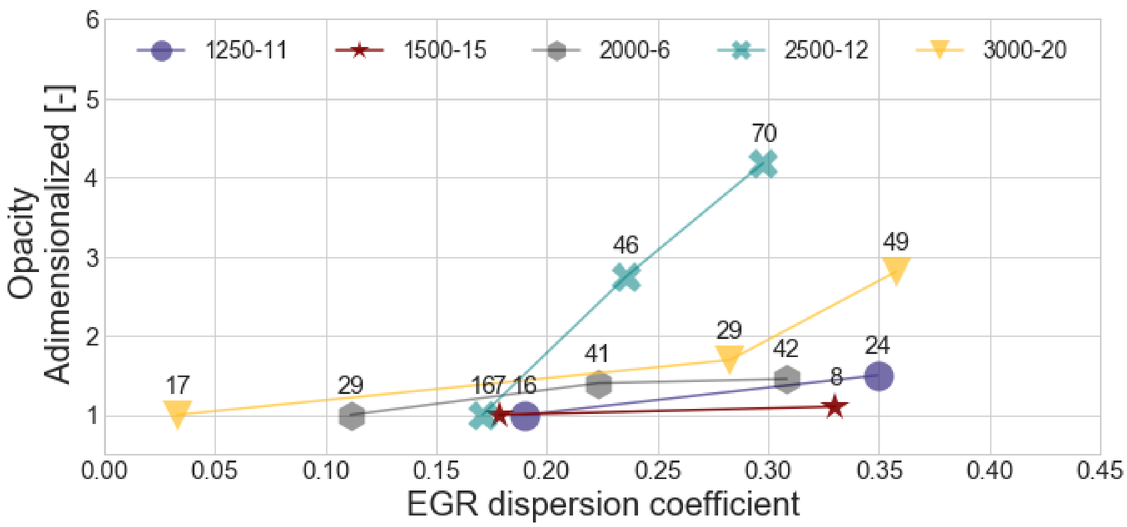


Figure 7. Adimensional opacity vs EGR dispersion coefficient for all the tested engine running conditions.

4. Conclusions

A novel methodology to study the HP EGR dispersion phenomenon was developed to quantify the impact on engine performance and pollutant emissions. This methodology is based on experimental tests on an automotive diesel engine where the intake manifold was fully instrumented with CO_2 probes together with a special hardware to modify and control the EGR dispersion between cylinders. The study includes a wide set of engine running conditions with different engine speed, engine torque and EGR rate. The tests were performed in five steady engine conditions where the EGR dispersion is modified but keeping constant the intake manifold pressure, the engine torque and the NOx raw emissions.

Depending on the engine running condition, it is necessary to increase the EGR rate to keep the NOx level constant when the EGR dispersion is varied. This usually happens when the engine runs with less than 15% EGR rate, while we found that it is not needed with EGR rates higher than 20%. Besides the EGR rate, other engine related parameters may influence on the results, being lambda one of the most important. If the engine runs near to stoichiometric situations, the conditions for remarkable increase of the opacity are likely to occur.

It is possible to conclude that the EGR dispersion affects to BSFC. As the EGR dispersion increases, the BSFC increases especially at high speeds, where the impact on the combustion process is more evident. A more thorough analysis is currently under investigation to find its causes.

Finally the standard deviation of the EGR rates for all the cylinders was converted to a non-dimensional figure taking into account the average EGR rate resulting in an EGR dispersion coefficient that helps to quantify the effect of the EGR dispersion on opacity emissions. The opacity was also transformed to a non-dimensional value using the opacity value of the test with the least dispersion. Both numbers were plotted together and it is possible to observe that the more affected points are 3000 and 2500 rpm running conditions followed by 1250 and 2000. It indicates that the influence of the lambda value and low EGR rate are not determinant on opacity emissions and the EGR dispersion has an important influence. So, finally, independently of the other factors, there is, always, a trend between the opacity and the EGR dispersion between cylinders. Attending to Fig. 7, opacity may increase dramatically when the EGR dispersion coefficient exceeds 0.2.

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