

# Experimental Study of an LP EGR System on an Automotive Diesel Engine, compared to HP EGR with respect to PM and NOx Emissions and Specific Fuel Consumption

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## ABSTRACT

Previous experimental studies on Diesel engines have demonstrated the potential of high-pressure exhaust gas recirculation (HP EGR) as an in-cylinder NOx control method. With ever more stringent emissions standards, the use of a low pressure EGR loop (LP EGR) seems to be an interesting method to further reduce NOx emissions while maintaining PM emissions at a low level. Actually, contrary to HP EGR, the gas flow through the turbine is unchanged while varying the EGR rate. Thus, by closing the variable geometry turbine (VGT) vanes, higher boost pressure can be reached, allowing the use of high rates of supplemental EGR.

Some experiments are conducted on a 2.0 l HSDI common-rail DI Diesel engine equipped with HP and LP EGR loops on a test bench under low and part load conditions, as those encountered in the European emissions test cycle for light-duty vehicles. The dilution ratio, boost pressure, as well as injection pressure are modified using the LP EGR loop to study the influence of boost pressure and injection pressure on combustion (instantaneous rate of heat release) and the NOx-PM trade-off (while varying EGR rate at a given boost pressure level), thus giving optimal boost pressure and injection pressure for each operating point in terms of NOx and PM emissions. In particular, higher boost pressures and EGR rates are studied, as compared with the original HP EGR loop. Finally, the impact on LP and HP IMEP, as well as ISFC, when increasing boost pressure by closing VGT vanes with the LP EGR loop is studied.

## INTRODUCTION

In light of the current requirements as regards the reduction of pollutant emissions (such EURO 6 in Europe), manufacturers have to find ever more complex in-cylinder emissions reduction strategies. Exhaust gas recirculation (EGR) into the engine intake is an established technology to reduce NOx emissions [1-14]. The decrease of NOx emissions with EGR is the result of complex and sometimes opposite phenomena occurring during combustion [2-9].

At the same time, the decrease in combustion temperatures and oxygen concentration while increasing EGR rate reduces both soot production in the spray core and soot oxidation in the diffusion flame around the jet [10]. Thus the final impact of EGR on PM emissions is complex and is the result of contradictory phenomena. In the conventional Diesel high-temperature combustion (HTC), the increase of EGR rate (at constant boost pressure) is accompanied by an increase of PM emissions, resulting in a trade-off between NOx and PM emissions while varying EGR rate [3, 11-14].

In order to limit the negative effect of EGR on PM, CO and hydrocarbon emissions as well as on BSFC (at constant boost pressure) in the HTC mode, some researchers have tried to maintain a constant air-fuel ratio (AFR) when increasing EGR rate [2, 11-14]. Nevertheless, for some operating conditions, EGR at constant AFR can result in a worse NOx-PM trade-off compared to constant boost pressure [2]. Moreover, with a classical HP EGR loop, an increase of EGR rate leads irremediably to a decrease of gas flow through the turbine and thus to a decrease of boost pressure. As a

consequence, with an HP EGR loop, it is particularly difficult to drastically reduce NOx emissions without a large penalty on PM emissions and BSFC by maintaining a suitable AFR (by increasing boost pressure) [2].

When using an LP EGR loop, contrary to a HP loop, the gas flow through the turbine is unchanged while varying the EGR rate. Thus, by closing the variable geometry turbine (VGT) vanes, higher boost pressure can be reached, thus permitting to increase the dilution ratio while maintaining a suitable in-cylinder oxygen quantity for PM emissions limitation. This is one reason why LP EGR is ever more studied by manufacturers for future automotive Diesel engines [15-18].

The aim of this paper is to study the capability of an LP EGR loop to reduce NOx and PM emissions, in comparison with HP EGR loop.

## EXPERIMENTAL SETUP

### TEST ENGINE

The study is conducted on 2.0 l water-cooled HSDI diesel engine which conforms to Euro IV standards, equipped with a Variable Geometry Turbine (VGT) turbocharger, an intercooler, two Diesel Oxidising Catalysts (DOC) and a Diesel Particulate Filter (DPF) (see Figure 1). Engine specifications are given in Table 1.

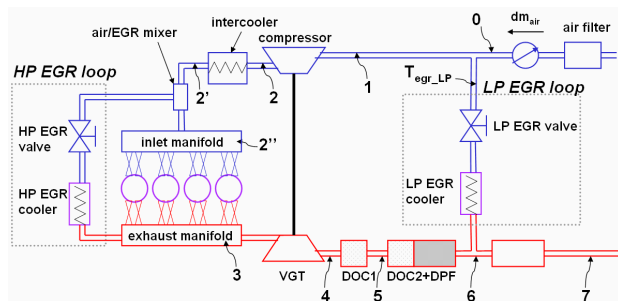


Figure 1. Engine configuration

Table 1. Engine specifications

|                               |                                                         |
|-------------------------------|---------------------------------------------------------|
| Compression ratio             | 18 : 1                                                  |
| Number of cylinders           | 4                                                       |
| Number of valves per cylinder | 4                                                       |
| Combustion chamber            | re-entrant bowl-in-piston                               |
| Injection System              | common-rail piezoelectric<br>2 <sup>nd</sup> generation |
| Maximum injection pressure    | 1600 bar                                                |
| Number of injection holes     | 7                                                       |
| Injector nozzle diameter      | 0.150 mm                                                |

## HP AND LP EGR SYSTEMS

The engine is originally equipped with a high pressure (HP) EGR loop. An LP EGR loop has been added. The recirculated gases are taken downstream of the DPF and introduced upstream of the compressor. In previous experiments [21], a valve situated in 7 (see Figure 1) was used to control exhaust back pressure thus permitting high EGR rates. The permeability of the LP EGR loop has been largely increased (by modifying the entire LP EGR loop: EGR valve, EGR cooler, as well as EGR ducts), such that high EGR rates are now achievable without artificially increasing exhaust back pressure.

Independent water circuits through the HP and LP EGR coolers are used, in order to control the temperature of recirculated exhaust gases before reintroduction ( $T_{egr\_HP}$  and  $T_{egr\_LP}$  respectively). Intake air temperature  $T_{2'}$  is controlled separately, allowing control over the temperature  $T_{2''}$  of inlet gases after mixing, independent of EGR rate (HP or LP). The temperature  $T_{2''}$  was fixed at 30°C for all the tests presented here after, thus avoiding the influence of inlet temperature on NOx and PM emissions while increasing EGR rate [19, 20].

The EGR rate is defined as follows:

$$X_{egr}(\%) = 100 \cdot \frac{X_{CO_2\_in}}{X_{CO_2\_ex}} \quad (1)$$

Where  $X_{CO_2\_in}$  and  $X_{CO_2\_ex}$  are measured  $CO_2$  concentrations in the inlet and exhaust manifold respectively.

The dilution ratio  $Dr$  (%) is given by:

$$Dr(\%) = 100 \cdot \frac{21 - X_{O_2\_in}}{21} \quad (2)$$

Where  $X_{O_2\_in}$  is the measured  $O_2$  concentration in the inlet manifold.

## OPERATING POINTS

The study has been conducted for various operating conditions (load and speed). Two operating points are presented here, such as those encountered in the extra urban driving cycle of the European emissions test cycle for light duty vehicles (composed of four urban driving cycles and one extra urban driving cycle). Such operating points are particularly critical in terms of NOx and PM emissions in this test cycle, whereas the use of EGR is limited to a low level with the HP EGR loop at these loads on the EURO IV standard configuration (below 5% for operating point B). Actually, an increase of EGR results in both a decrease of  $O_2$  concentration in the inlet and a decrease of boost pressure, thus reducing the in-cylinder  $O_2$ . At middle load operation, this will irretrievably lead to a critical value of the air excess, and

thus to a large increase of PM, CO, and hydrocarbon emissions, as well as a large increase of specific fuel consumption. Thus the in-cylinder reduction of NOx emissions is particularly difficult for these operating points. Furthermore, there is clear tendency of downsizing the automotive Diesel engines, and thus there is a need to reduce pollutant emissions at higher loads. It is why these operating points are presented in this paper. The engine speed, torque, pilot and main injections quantities, brake mean effective pressure (BMEP) and rail pressure are given in Table 2. For each operating point, injection quantities are held constant. The changes in BMEP are thus a result of the various tested modifications (EGR rate, boost pressure, injection pressure, etc.)

Table 2. Operating points

| Point                          | A    | B    |
|--------------------------------|------|------|
| Engine speed (rpm)             | 1665 | 2050 |
| Torque (Nm)                    | 114  | 140  |
| Pilot quantity (mg/stroke)     | 1.5  | 1.7  |
| Principal quantity (mg/stroke) | 22.8 | 30.7 |
| BMEP (bar)                     | 7.1  | 9.5  |
| P <sub>rail</sub> (bar)        | 865  | 1028 |

## EMISSIONS MEASUREMENT

NOx emissions are measured with an ECO PHYSICS CLD 700EL gas analyser, which uses the chemical luminescence detector (CLD) method. PM emissions at the exhaust are measured with an AVL 415S smoke-meter. Inlet and exhaust O<sub>2</sub> and CO<sub>2</sub> concentrations are measured with a SIEMENS ULTRAMAT 23 gas analyzer, which uses the non-dispersive infrared measurement technique (NDIR). Each gas analyser is calibrated every 2 hours of experiments with specific gas standards. If the necessary shift is under 0.3 %, then the experiments done since the previous calibration are validated. The conversion of NOx emissions from ppm to g.h<sup>-1</sup> and the calculation of PM emissions in g.h<sup>-1</sup> starting from the filter smoke number (FSN) are presented in a previous paper [21].

## ERROR ANALYSIS

Table 3 sums up the measurement technique, calibrated range, accuracy and relative error of various instruments involved in the experiments described in this study for main parameters. Errors in experiments can arise from instrument conditions, calibration, environment, observation, reading and test planning. The accuracy of the experiments has to be validated with an error analysis. That is performed here using the differential method of propagating errors based on Taylor's theorem [22]. It gives the maximum error  $u$  of a function  $f(x_1, x_2, \dots, x_n)$  as follows:

$$u(f(x_1, x_2, \dots, x_n)) = \sqrt{\sum (u(x_i))^2} \quad (3)$$

As a result, the maximum relative errors for the dilution ratio, NOx (g/h), PM (g/h) and BSFC are 0.2%, 1.5%, 2.3%, and 1.05% respectively.

Table 3. Relative measurement errors

| Instrument                                                                             | Calibrated range | Accuracy   | Relative error |
|----------------------------------------------------------------------------------------|------------------|------------|----------------|
| Inlet gas temperature (k-type thermocouple)                                            | 0-1000 °C        | ± 1°C      | ± 0.75 %       |
| Inlet gas pressure (2 bar piezoresistive relative pressure sensor HCS Sensor Technics) | 0-2 bar          | ± 5 mbar   | ± 0.25 %       |
| Air mass flow (hot wire air flow meter)                                                | 0-1000 mg/str    | ± 5 mg/str | 1 %            |
| Fuel consumption (PIERBURG PLU 401/121)                                                | 0.05-23 kg/h     | ± 37g/h    | ± 0.16 %       |
| NOx (ECO PHYSICS CLD 700EL)                                                            | 0-1000 ppm       | ± 5ppm     | 1 %            |
| Smoke (AVL 415S)                                                                       | 0-10 FSN         | ± 0.1 FSN  | 2 %            |
| Inlet O <sub>2</sub> (SIEMENS ULTRAMAT 23)                                             | 0-25 %           | ± 0.025 %  | 0.2 %          |
| Inlet and exhaust CO <sub>2</sub> (SIEMENS ULTRAMAT 23)                                | 0-20 %           | ± 0.1 %    | 1 %            |
| Engine torque                                                                          | 0-700 Nm         | ± 3.5 Nm   | 1 %            |
| Engine speed                                                                           | 0-5500 rpm       | ± 5 rpm    | 0.2 %          |
| In-cylinder pressure (Kistler 6055BB)                                                  | 0-200 bar        | ± 0.5 bar  | 1 %            |

## EXPERIMENTAL RESULTS

### INFLUENCE OF BOOST PRESSURE WHILE VARYING LP EGR RATE ON NOX AND PM EMISSIONS

As higher boost pressures can be obtained thanks to the LP EGR loop, the objective of the experiments described here after is to study the influence of boost pressure on combustion and NOx-PM emissions while varying LP EGR rate. Thus, for both operating points A and B, four levels of boost pressure are studied, from the base level to the maximum value reachable with the existing turbocharger, in order to find the optimal boost pressure in terms of NOx and PM emissions. The boost pressure is increased by closing VGT vanes. The injection pressure is maintained at the same level (given in Table 2).

## Combustion analysis

### Operating point A

Mean cylinder pressures (over 60 consecutive cycles) and mean instantaneous gross rate of heat release (ROHR) while varying the dilution ratio are given in Figure 2 and Figure 3 respectively, for operating point A, for the minimum and maximum boost pressure (1176 mbar and 1476 mbar).

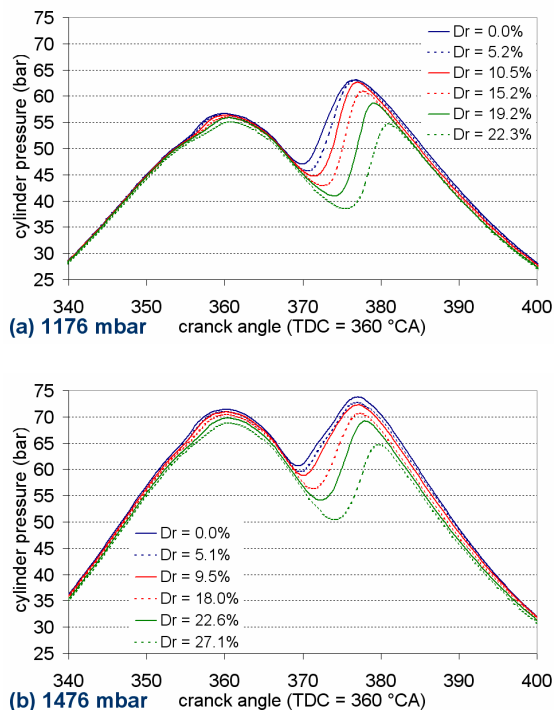


Figure 2. Measured cylinder pressure while varying dilution ratio with 1176 mbar and 1476 mbar boost pressure (figures a and b respectively), operating point A

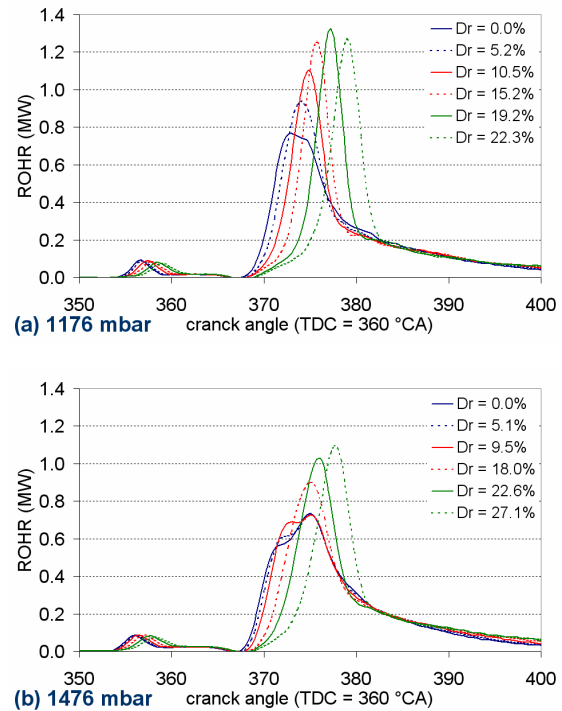


Figure 3. ROHR while varying dilution ratio with 1176 mbar and 1476 mbar boost pressure (figures a and b respectively), operating point A

First of all, it must be underlined that for the conditions tested here, the use of a pilot combustion just before principal injection results in a very short ignition delay of principal combustion. As a matter of fact, the major part of the fuel is burning under a mixing-controlled process (diffusion combustion), the combustion speed being thus governed by the mixing process between oxygen and fuel.

With reference to Figure 2 and Figure 3, following remarks can be made:

- When increasing dilution ratio using the LP EGR loop, the ignition delay is increased. As a consequence, the premixed part of combustion is increased as well, that results in an increase of the ROHR peak (see Figure 3). This result is independent of the boost pressure.
- The increase of boost pressure for a given dilution ratio results in a slight decrease of ignition delay, of which a consequence is a drop of the premixed part of combustion and the associated peak on the ROHR diagram (see Figure 3).

### Operating point B

Mean cylinder pressures and ROHR while varying the dilution ratio are given in Figure 4 and Figure 5

respectively, for operating point B, for the minimum and maximum boost pressures (1326 mbar and 1800 mbar).

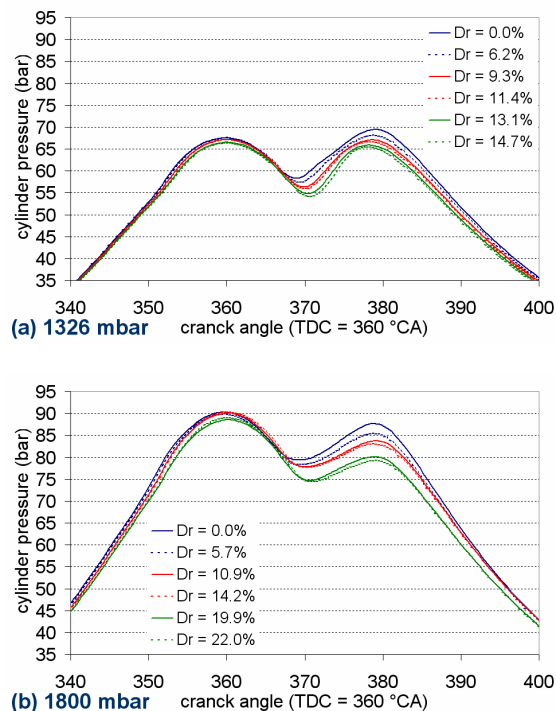


Figure 4. Measured cylinder pressure while varying dilution ratio with 1326 mbar and 1800 mbar boost pressure (figure a and b respectively), operating point B

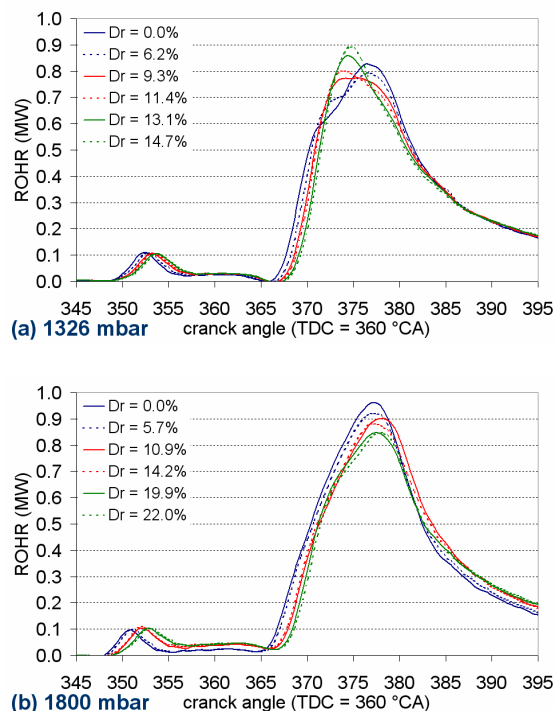


Figure 5. ROHR while varying dilution ratio with 1326 mbar and 1800 mbar boost pressure (figures a and b respectively), operating point B

Same trends as for operating point A are observed. Moreover, it can be seen that the increase of dilution ratio results in a decrease of the speed of the diffusion part of combustion. Actually, when maintaining injection parameters constant (start of principal injection, injection pressure), the jet entrains the same volume of ambient gases (air and EGR) [23]. Thus, at constant boost pressure and inlet temperature, the in-cylinder trapped mass is almost constant. Consequently, when increasing the dilution ratio, the fuel jet entrains less fresh air with reduced inlet  $O_2$  concentration, resulting in a lower oxygen-fuel mixing and a lower ROHR.

As well, when increasing boost pressure for a given dilution ratio, the in-cylinder trapped oxygen is increased (because of an increase of total in-cylinder trapped mass), thus resulting in an increase of the speed of the diffusion part of combustion.

This phenomenon can not be clearly seen on the ROHR diagram for operating point A (Figure 3), because the decrease of the speed of the diffusion part of combustion is compensated by an increase of the premixed part of combustion that increases the global ROHR peak.