

Experimental Investigation of the Benefits of Cooled and Extra-cooled Low-Pressure EGR on a Light Duty Diesel Engine Performance

Carlo Beatrice, Claudio Bertoli, Nicola Del Giacomo and Chiara Guido

Istituto Motori – Consiglio Nazionale delle Ricerche, Naples, Italy

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ABSTRACT

The present paper describes an experimental study on the application of a Low Pressure EGR system, equipped with an high efficiency cooler, to a LD diesel engine operating with both conventional combustion and PCCI mode.

The research activity is aimed to carry out an analysis of the potentiality of the cooling (with engine water at 90°C) and super-cooling (with external water at 20°C) of the low pressure EGR flow gas on the simultaneous reduction of fuel consumption and pollutant emissions. The effects were evaluated running the engine with diesel conventional combustion and PCCI mode in several engine operating points.

The employed engine was a 4-cylinders LD CR diesel engine of two liters of displacement at the state of art of the current engine technology.

The overall results identified benefits on both the fuel consumption and emissions with the use of a low pressure EGR system with respect to the “classical” high pressure EGR one. The cooled EGR system showed advantage mainly at medium and high speed/load operating points.

INTRODUCTION

The research effort towards the Clean Diesel engine is in place from many decades and significant improvements have been achieved until now. During these years, several breakthrough technologies were introduced in

the engine system as Turbocharger, EGR, DOC, Common Rail with electronic injection control, DPF; others are under development as DeNOx (SCR and LNT) and twin stage Turbo [1]. Thanks to these technologies the light duty diesel engine is evolving toward the Fuel Neutral Emissions (FNE) in Europe, indicated also as EURO6 emission standards, and in US toward the Tier2 Bin5 targets.

Together the listed technologies, the operating mode in the emission area of the diesel engine has been evolved in the last years, moving from the “classical” turbulent heterogeneous diffusion flame combustion to an highly EGR diluted premixed combustion (generally named as PCCI – Premixed Charge Compression Ignition – or LNC – Low NOx Combustion). This new combustion management, currently under development, appears a powerful methodology to approach the EURO6 NOx targets without the employment of DeNOx technologies for some downsized engines, and minimizing the FC penalty for NOx after-treatment systems regeneration whereas their employment is needed [2].

Recently, due to the growing problem of the world overheating, the attention has been aimed at the issues of CO2 emission from road vehicles, that in turns is directly linked with the fuel consumption (FC). From this point of view not all above listed “green” technologies are “FC efficient”: DPF and DeNOx require an increase of FC for regeneration [2], while PCCI combustion requires further research work to meet the same combustion efficiency of the classical one [1, 3, 4]. Therefore, today, the new challenge for engine designers is to develop new engines with lowest FC by meeting at the same time

the future emission regulations (e.g Tier2Bin2 in US, EURO6 in EU etc.).

Together with other technologies under development for engine efficiency increment (engine downsizing, variable valve actuation systems, high pressure CR injection systems etc.), the so called “Low Pressure EGR system” (LPEGR) appears a very interesting solution for the simultaneous improvement of FC and pollutant emissions. This technology, based on the recirculation of the cleaner exhaust flow downstream the DPF [5], is mature for large scale application to the Heavy Duty diesel engines and it is almost ready for the employment in the next generation of automotive diesel engines [1].

Recent papers have showed important results on the LPEGR adoption in the LD engines. Thanks to the lower EGR temperature and higher turbine efficiency (avoiding the exhaust flow reduction when increase the EGR) with respect to the conventional one, the new EGR system has showed an high potential in the engine performance improvement [5, 6, 7]. Even if the analysis on the effects of LPEGR application on the LD diesel engine performance were already done, experiments with cooled LPEGR systems (also with super-cooled system by means of a dedicated cooling circuit) were reported in literature in only three papers [6, 7, 8]. These papers were focused on the effects of super-cooling LPEGR on engine emissions but an analysis of the different contribution of the air-path components (cooler HPEGR, cooler LPEGR and the charge air cooler-CAC) on the engine performance and emissions was not taken into account. A joint research work between Istituto Motori and Fiat Powertrain Technologies research center (FPT-R&T) was specified and dedicated to this aim, highlighting also the potential of the joint of LPEGR and PCCI concept. The adopted experimental layout, the test program and the results are described and analyzed in the paper.

THE EXPERIMENTAL SETUP

The experimental layout comprises a 4-cylinder LD diesel engine installed on a dyno test bench. The engine selected for the research is the FIAT 1.9 MultiJet. Its displacement represents the most popular size of diesel engine for passenger car in the European market. The main characteristics of the engine are listed in Table 1.

The engine is fully instrumented for indicated signal measurements in each cylinder (cylinder pressure, injection pressure, energizing injector current). At the engine exhaust, smoke was measured by a high-resolution (0.01 FSN) smoke meter (AVL415S), while gaseous emissions were measured upstream and downstream the oxidation catalyst by means of an analysis test bench (AVL-CEB-2).

A scheme of the air and EGR path is displayed in Figure 1.

Table 1: Main features of the multi-cylinder engine

Engine type	In-line 4 cylinders
Bore [mm] x Stroke [mm]	82.0 x 90.0
Compression Ratio	16.5
Displacement [cm ³]	1910
Valves per cylinder	4
Injection system	Bosch CR
Injector	Bosch CRIP 2.2, 7 holes
Turbocharger	Single stage VGT
Swirl control	Throttle valve on intake duct
Catalyst system	Closed-couple DOC plus underfloor DPF
Maximum Power	112 kW @ 4200 rpm

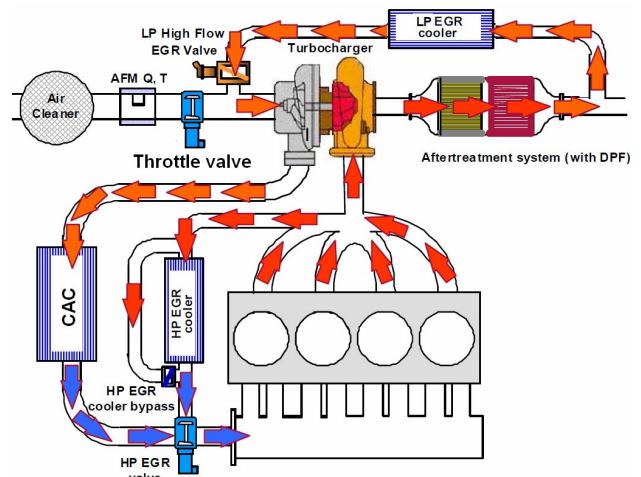


Figure 1: EGRs layouts on the FIAT multi-cylinder engine (source FIAT).

The LPEGR was installed on the engine system employing an high efficiency water cooler from Denso (LPEGR cooler) and a Cooper DC motor throttle valve (LPEGR valve) for EGR flow regulation. Depending on the desired cooling conditions of the LPEGR flow, the cooler was by-passed for not cooled LPEGR flow, supplied by engine cooling water (90°C) for a cooled LPEGR flow and supplied by water at lower temperature (20°C) in order to have a super-cooled LPEGR flow.

TEST METHODOLOGY

The engine equipped with the different LP-EGR system configurations was tested in four steady-state test points reasonably representative of the engine behavior during the NEDC test procedure, and with two combustion modes: a premixed low NOx combustion (PCCI) and the diesel conventional one. The operating points were chosen in order to cover different zones of the NEDC area at low, medium and high speed/load. The test points are reported in Figure 2 where the NEDC operating area for the employed engine when installed

on a D-segment vehicle is also displayed. An additional point at 2800 rpm and 13 bar of BMEP was added to the test point matrix in order to evaluate the benefits of the use of LPEGR system in a engine operating area where the use of the HPEGR is generally unreasonable for the very high smoke conditions.

For both operating modes, PCCI and conventional diesel one, the correspondent engine calibration were supplied by FPT. In particular, for the conventional diesel combustion a close pilot plus main injection strategy was adopted together with a moderate level of EGR, rail pressure and swirl to reach NO_x EURO5 targets. In the case of PCCI mode, the same kind of injection strategy was adopted coupled with higher values of EGR, rail pressure and swirl.

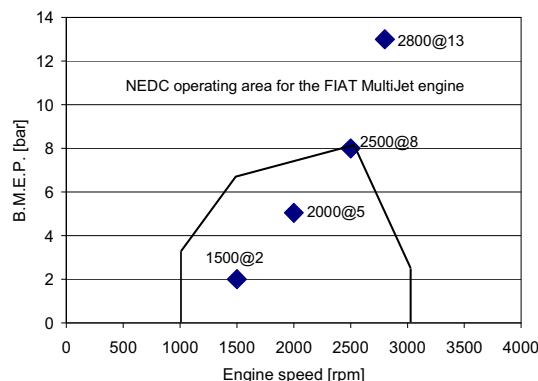


Figure 2: Engine operating points with both PCCI and diesel combustion together with the operating area of the engine executing the NEDC cycle.

For diesel conventional mode the different LPEGR configurations were evaluated performing the trade-off NO_x-Particulate (PM), or NO_x-CO depending on the operating point, versus EGR. All operating parameters (rail pressure, pilot injection quantity, injection dwell time, etc.) were kept constant, and for each EGR step value, the main injection timing (SOE_{main}) and the intake turbine area (VGT%) were varied in order to maintain the combustion timing (50% of the mass burned fraction-MBF50%) and the boost pressure (P_{boost}) values equal to the reference calibration one.

For engines equipped with both HPEGR and LPEGR systems, a choice in the EGR rate partition between the two systems is necessary. In the present case, taking into account the better efficiency of LPEGR with respect to the HPEGR one, starting from the zero EGR condition, the NO_x-PM trade-off was performed increasing the LPEGR rate until the maximum or the maximum acceptable gas temperature at the outlet of the compressor was attained (160°C for our system). The further increase of total EGR rate was obtained opening the HPEGR circuit.

For the PCCI operating mode, the different LPEGR configurations were analyzed with the same engine

calibration in terms of intake air mass flow, injection strategy, rail pressure, P_{boost} and maximum cylinder pressure rise. Also for the PCCI operating mode the LP/HPEGR repartition methodology was the same of the previous one, maximizing the LPEGR rate for the fixed intake air mass flow and using the HPEGR for the surplus. The values of the intake air flow were chosen as best compromise between adequate level of EGR and stable combustion conditions. Further, the SOE_{main} was optimized for each tests to find the best compromise between maximum acceptable cylinder pressure rise and thermodynamic efficiency (MBF50% timing settled as close as possible to its optimum value for maximum thermodynamic efficiency). In this way the authors evaluated the possible improvement of the PCCI combustion efficiency by cooling the LPEGR system.

Taking into account that for the diesel engines the main contribution to intake charge cooling is supplied by the CAC, in order to evaluate the effect of LPEGR cooling conditions, it is necessary to work with a constant CAC efficiency, so avoiding, as possible its mask effects when it works with fixed outlet gas temperature. Therefore, the water mass flow through the CAC was adjusted in order to have a cooling efficiency equal to that of the water-air cooler installed on the reference vehicle (Alfa Romeo 159).

The fuel employed for the tests is a commercial low sulphur diesel fuel, sold in Italy by AGIP Petroli with the name of “BluDiesel”, whose characteristics respect the EN 590 standards.

RESULTS OF CONVENTIONAL DIESEL COMBUSTION TESTS

Starting the result analysis from the operating test point of 1500 rpm and 2 bar of BMEP, in the following Figures 3, 4, 5 and 6, the emissions and operating parameter values versus lambda are reported.

At low speed and low load engine conditions the smoke emission is very low and generally is almost insensible to EGR rate, as observable in the Figure 3. The most critical pollutant together with the NO_x is the carbon monoxide. Due to the low catalyst temperature, both CO and HCs exhaust emission levels are critical for emission target compliance. Therefore, in the low load and low speed operating area, the engine is calibrated searching the optimum in the NO_x-CO trade-off. The diagram of Figure 4 shows the resulting NO_x-CO trade-off for the different EGR configurations.

Looking at the diagrams of the NO_x-CO trade-off and total EGR rate versus lambda, it clearly appears that the adoption of the LPEGR system permits to push down the lambda operating value that improves slightly the trade-off or, at the same lambda, allows to work with higher total EGR rate. These results depend on the capability of LPEGR system to work with high EGR at the same boost pressure, as better explained later. The trade-off for the

HPEGR system was limited up to $\lambda=1.9$ because of an unacceptable FC increment below this value. However, it has to be evidenced that the CO-NOx trade-off improvements for LPEGR system appears marginal considering the already very low level of the emissions in the reference condition (HPEGR). To demonstrate the LPEGR capability to work with high total EGR rate, the diagram of Figure 5 reports the EGR rates versus λ among all configurations.

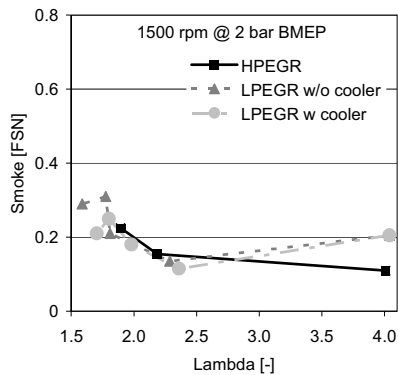


Figure 3: Smoke emission vs λ at 1500 rpm and 2 bar of BMEP.

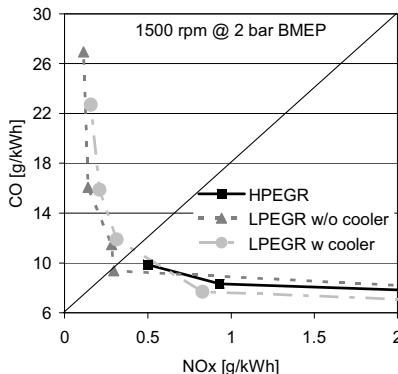


Figure 4: NOx-CO trade-off vs λ at 1500 rpm and 2 bar of BMEP.

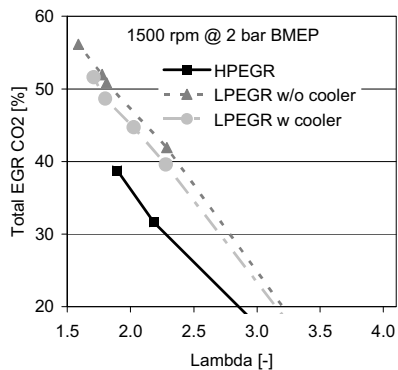


Figure 5: Total EGR rate vs λ at 1500 rpm and 2 bar of BMEP.

About the differences in the working mode of the LPEGR system with respect to the HPEGR one, it must be evidenced that the LPEGR adoption permits a better volumetric efficiency. This behaviour depends on the reduced intake manifold gas temperature (T_{manif}), and on higher turbocharger efficiency, due to the greater exhaust mass flow supplied to turbine with respect to the HPEGR system. For these reasons, when the LPEGR system is used, the engine can operate at the same airflow and EGR rate with lower boost pressure, or at the same boost pressure with higher EGR rate with respect to the HPEGR system, as well explained in [5]. In this last case, to reach the same boost pressure, the engine can operate with turbine far from its fully closed rack position as generally happens for HPEGR. This is an important feature of the LPEGR working mode. The Figure 6 shows a comparison of Pboost, inlet turbine relative pressure (P_{turbo}) and the duty cycle value of the turbine rack (T_{duty}). It has to be noted that the duty signal of the rack position of LPEGR system is placed at 52%, while, for HPEGR is in the fully closed position (85%), increasing the P_{turbo} for the last one.

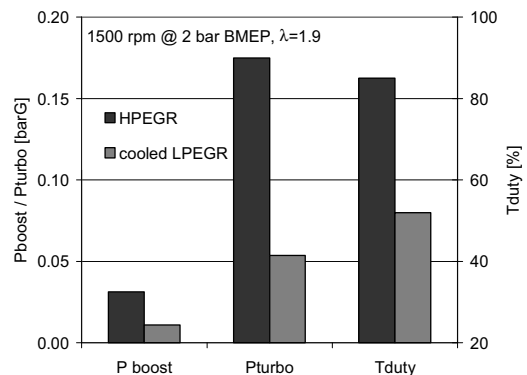


Figure 6: Pboost, Pturbo and Tduty comparison between HPEGR and LPEGR for conventional combustion at 1500 rpm and 2 bar of BMEP with a $\lambda \approx 1.9$.

Therefore, the adoption of LPEGR system should give advantages in reducing the pumping losses and in the control of the boost pressure. Actually, at the present operating point, the effective advantage of LPEGR is very limited because also increasing the turbine rack closure only a slightly rise of boost pressure can be obtained and without increment of the EGR rate. This effect is well explained by Ivaldi *et al.* [5]. On the contrary, as showed in the following, at high speed and high load point, the possibility to operate with reduced P_{turbo} leads to the engine efficiency improvement. Moreover, the higher degree of control of EGR and boost pressure deriving from the reduced closure of the turbine rack for the LPEGR with respect to the HPEGR, could limit the control problems related to the long response time of the LPEGR system under transient conditions.

However, focusing the attention on the operating point under discussion, no significant BSFC improvement was measured for the LPEGR when very low NOx conditions

were approached, as reported in the BSFC versus lambda trade-off of Figure 7 for lambda values below 2. This result depends on the low exhaust gas pressure and higher friction loss of the LPEGR circuit with respect to the HPEGR. It is required the use of both the EGR circuits in order to reach the required total EGR rate and lambda value (see range $\lambda < 2$). That can be observed in Figure 8 where the percentage of LPEGR with respect to the total is plotted versus lambda. In this case, also the intake throttle valve upstream the CAC was partially closed (50% of closure) to realize the EGR target increasing furthermore the drop pressure along the intake circuit. Therefore, at low speed and load operating points, the benefits of the LPEGR system in terms of BSFC are negligible with respect to the conventional HPEGR one, due to the partial use of the HPEGR and the partial closure of the intake throttle valve.

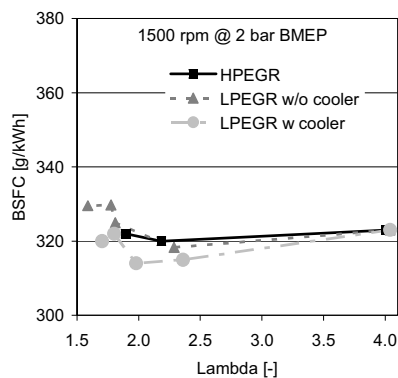


Figure 7: BSFC vs λ at 1500 rpm and 2 bar of BMEP.

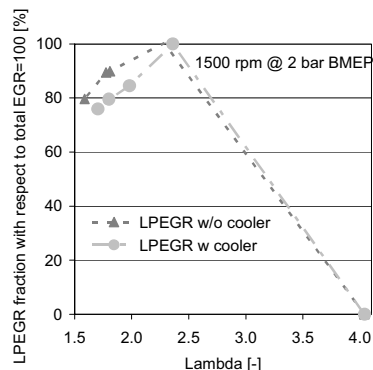


Figure 8: LPEGR percentage on total EGR=100 vs λ at 1500 rpm and 2 bar of BMEP.

Therefore, the employment of a cooler in the LPEGR circuit doesn't seem to give a further gain in terms of both emissions and FC reduction. So, in the above diagrams, the super-cooled conditions of the LPEGR were omitted. In the present operating point, in fact, the reduction of T_{manif} in the engine was negligible as evident looking at the Figure 9: any effect of the LPEGR cooler on the T_{manif} is observable and the tests relative

to the super-cooled condition were considered superfluous. The HC emissions varying the EGR configurations are plotted in the Figure 10. Similar HC trends for all configurations are observable.

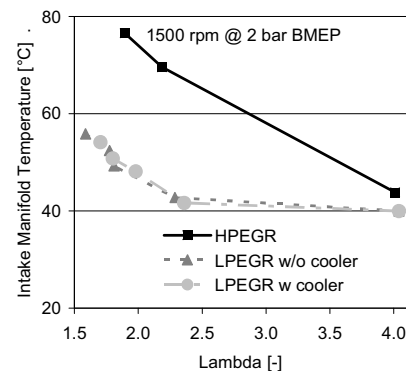


Figure 9: T_{manif} vs λ at 1500 rpm and 2 bar of BMEP.

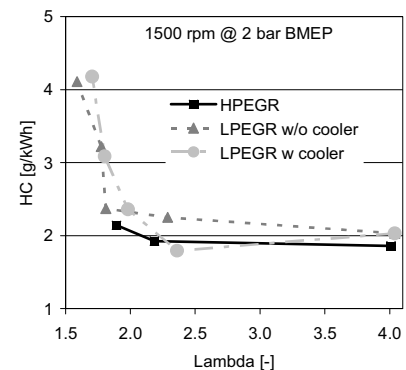


Figure 10: HCs vs λ at 1500 rpm and 2 bar of BMEP.

Figures 11 and 12 report the trends of the gas temperature varying EGR configuration in two points of the intake air path. Figure 11 shows the outlet temperature of the HP-EGR or LP-EGR valve, depending on the used configuration (see Figure 1), and Figure 12 reports the inlet CAC temperature (T_{cac}).

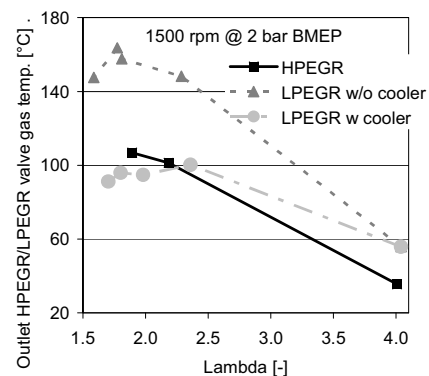


Figure 11: Outlet HPEGR/LPEGR valve temperature vs λ at 1500 rpm and 2 bar of BMEP.