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Effect of the application of an uncooled high-pressure EGR strategy in low-load diesel PCCI operation

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Abstract. The exploitation of new advanced combustion concepts referred to as low-temperature combustion (LTC) strategies gives the possibility to reduce the typical NO_x and PM emissions from conventional diesel combustion. These strategies are implemented using high quantity of exhaust gas recirculated at the intake manifold, which allows to lower the peak combustion temperatures, thus reducing NO_x, and advancing the fuel injection timing, thus keeping the PM emission under control. Cooling the EGR can be beneficial, as the increment in density of the burned gases downstream the EGR cooler reflects in a further decrement of NO_x and PM emissions. On the other hand, the lower EGR temperatures may enhance higher in-cylinder formation of unburned hydrocarbon (HC) and carbon monoxide as well as negatively affect the exhaust gas temperature upstream the after-treatment line. This latter effect can be particularly detrimental especially at low engine speeds and loads and/or during engine warm-up. In these cases, in fact, the after-treatment system is not able to convert the incomplete combustion species because the light-off temperature of the catalyst is not reached. This work tries to present a comparison between cold and hot EGR strategies in the case of a short EGR route configuration. The results highlight a reduction in terms of tailpipe HC and CO emissions for low-torque engine working points, when using hot EGR strategies, especially due to the enhancement of the DOC working temperature. On the other hand, the area of the map where PCCI combustion is achievable with uncooled EGR shows to be reduced with respect to the regular cold EGR strategy.

1. INTRODUCTION

Throughout past years, many countries have adopted stricter pollutant emission regulations across the world to face the problem of urban air pollution. Since diesel engines are considered one of the largest contributor to this environmental issue [1], combustion research has been focusing on solutions to develop technologies able to reduce the most critical pollutant species emitted by vehicles equipped with diesel engines, i.e. nitrogen oxides (NO_x) and particulate matter (PM). Reduction in engine-out pollutant emissions can be obtained with a careful control of combustion due to a flexible injection pattern obtained with the utilization of innovative common rail injection systems [2,3] able to provide high injection pressures [4] and low leakages [5]. Experimental tests can be supported by modeling of the injection system [6,7,8] and by model-based combustion tools [9]. In addition, exhaust after-treatment systems (ATSs), such as selective catalytic reduction (SCR) systems, lean NO_x traps (LNTs) and diesel particulate filters (DPFs), have proven to be effective to further lower engine-out emissions to small tailpipe values, although they raise the cost and complexity of the whole powertrain. Moreover, several new advanced combustion concepts generally referred to as low-temperature combustion (LTC) strategies, have been proposed for mitigating the formation of NO_x and PM inside the engine cylinders, minimizing engine-out emissions [10]. Among the various LTC concepts, premixed charge compression ignition (PCCI) strategy has shown great potentialities. PCCI can lead to a strong reduction of NO_x emissions, up to 97% with respect to conventional diesel combustion (CDC) [11], by featuring a heavy amount of exhaust gas recirculation (EGR), typically over 50% [12]. Moreover, PM can be brought to very low levels by either advancing (early PCCI) or retarding (late PCCI) the start of injection (SOI), giving more time to the fuel to mix with the inducted charge and improving the local air-fuel mixture quality. Nevertheless, there are still

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several hurdles avoiding PCCI concept to be widespread, typically related to operating range limitations [13], combustion instability [14] and high production of incomplete combustion species, i.e. carbon monoxide (CO) and unburned hydrocarbons (HC) [15].

A certain penalty in incomplete combustion species is characteristic for all LTC concepts, due to the relatively low in-cylinder temperatures during the combustion process, the increased ignition delay period (which sharpens the occurrence of overmixing phenomena) and the wall-wetting by the early injected fuel [13,16]. Nevertheless, great attention has been put on this theme, especially focusing on CO and HC emissions during vehicle cold starts or when operating at the lowest engine loads. In these circumstances, for instance during a given type approval driving cycle, the managing of the after-treatment systems can be troublesome and may represent a key theme, as high engine-out HC and CO levels cannot be mitigated until the catalytic system reaches its light-off temperature [13,17]. Thus, several techniques can be adopted to mitigate these issues, such as limiting early injection wall-wetting through injectors with narrower spray cone angles [18,19], using alternative fuels featuring lower boiling points [20] or recirculating uncooled EGR [10,13,21].

The present study highlights the effects of the application of uncooled EGR strategies to low-load early PCCI engine operating points, as a way to limit their tailpipe emissions of incomplete combustion species (i.e., CO and HC).

2. EXPERIMENTAL FACILITIES AND ENGINE SETUP

The experimental activities have been carried out at the dynamic test bed installed in the Internal Combustion Engines Advanced Laboratory at the Politecnico di Torino. The tested engine is a modified version of the FPT F1C, a four-stroke, four-cylinder 3.0 L diesel engine, where some purposely designed hardware modification have been implemented to host a non-conventional PCCI combustion. These engine modifications, with respect to the original version, are mainly related to the combustion chamber design, in terms of bowl shape and reduced compression ratio, to the injector cone angle as well as to the turbocharger dimensions [11]. Table 1 shows the main technical specification of the tested engine, while its schematic is depicted in Figure 1.

TABLE 1. Main specifications of the tested engine

Engine type	Diesel engine, four-stroke, four-cylinder
Displacement	2998 cm^3
Bore / stroke	95.8 mm / 104 mm
Connecting rod	158 mm
Compression ratio	14.6
Valves per cylinder	4
Turbocharger	Single-stage VGT
Fuel injection system	Common Rail, solenoid injectors

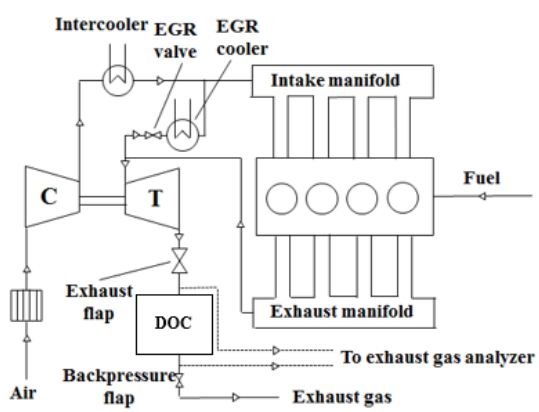


FIGURE 1. Simplified schematic of the engine installed at the test bench.

The PCCI mode allows to decrease the combustion temperatures and it has been achieved applying large amount of EGR and advancing well before the TDC a single fuel injection pattern. The tested engine features a short-route (high-pressure loop) EGR system. The original EGR heat exchanger was replaced by a larger version that ensures higher cooling power (from 6 kW provided by the original cooler to 32kW by the new one). An exhaust flap is placed downstream of the turbine in order to increase the backpressure in the exhaust manifold and, thus, the EGR flowrate, especially at the lowest engine speed and loads, where insufficient EGR rates would be obtained otherwise. In fact, being the amount of exhaust gases recirculated strictly related with the differential pressure between the two manifolds, further throttling the exhaust flowrate by means of the exhaust flap translates in an increase of the EGR rate. The exhaust flap can also be used to increase the conversion efficiency of the after treatment system by raising the exhaust gas temperature, especially at the engine warm-up. As far as the EGR system layout is concerned, two different hardware configurations were tested. The first configuration (which will be called "cold" EGR configuration) consists in the use of the 32 kW thermal cooling power EGR cooler: a water-to-gas heat exchanger, used to cool down the exhausted gases recirculated at the intake manifold. The cooling water required is then cooled down by means of a second water-to-water heat exchanger, with the possibility to control the amount of cooling water passing through this second heat exchanger by acting on a PID-controlled electro-valve. With this secondary cooling circuit, independent of the main engine cooling system, it is possible to control the temperature of the recirculated gases downstream of the EGR cooler. Their target temperature is set at 85°C [11]. The second configuration (which will be called "hot" or "uncooled" EGR configuration) is then obtained by removing the EGR cooler and replacing it with a straight duct. Obviously, in this way, no cooling and no control of the EGR temperature is achieved. Thus, the temperature of the EGR gases will depend on the combustion characteristics and the engine working point (with a possible minor influence of the heat transfer through the straight duct due to unavoidable conduction and convection). The ATS in the test bench configuration only consist of a DOC for the HC and CO oxidation. Thermocouples and pick-up streams for the gaseous species pollutant measurement are placed both upstream and downstream of the DOC, to evaluate its working temperatures and the related HC and CO conversion efficiency.

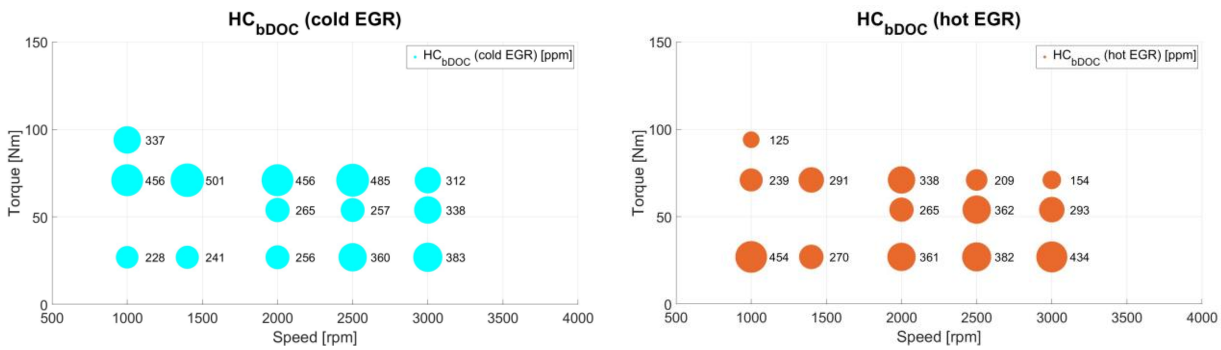
The test bench is equipped with an "AVL APA 100" dynamometer and an "AVL KMA 4000" fuel metering unit. The gaseous species from the engine are measured by an "AVL AMA i60" analyzer. The instrument is provided by two streams (as said, placed at the two sides of the DOC), both endowed with devices that are able to measure the concentration of HC, CO, CO_2, NO_x, NO and O_2,. An "AVL 415s" smokemeter is then used to measure the PM level in the exhaust line.

3. EXPERIMENTAL TEST ANALYSIS

The comparison tests between hot and cold EGR layouts have been performed on the following steady-state engine working points (expressed in terms of speed n [rpm] and torque T [Nm]), using early single injection PCCI calibrations: 1000×27, 1400×27, 2000×27, 2500×27, 3000×27 ($bmep \approx 1$ bar), 2000×54, 2500×54, 3000×54 ($bmep \approx 2$ bar), 1000×71, 1400×71, 2000×71, 2500×71, 3000×71 ($bmep \approx 3$ bar) and 1000×94 ($bmep \approx 4$ bar). This considered part of the engine map has been load-limited to guarantee that a stable PCCI operation was reached, especially in the hot EGR configuration, where the increase of the load would lead to recirculate hotter and hotter gases, thus rapidly approaching quasi-stoichiometric inlet charges and rapidly rising soot emissions, not in line with the aim of this study. For each considered working point, preliminary tests were performed by exploring several combinations of the most relevant calibration parameters [22] with a "one-factor-at-a-time" approach [23]. The selected parameters have been identified as the ones that mainly affect PCCI combustion, i.e.: the position of the VGT (to control the boost pressure), the position of the exhaust flap (to control the EGR rate), the start of injection (SOI) and the rail pressure.

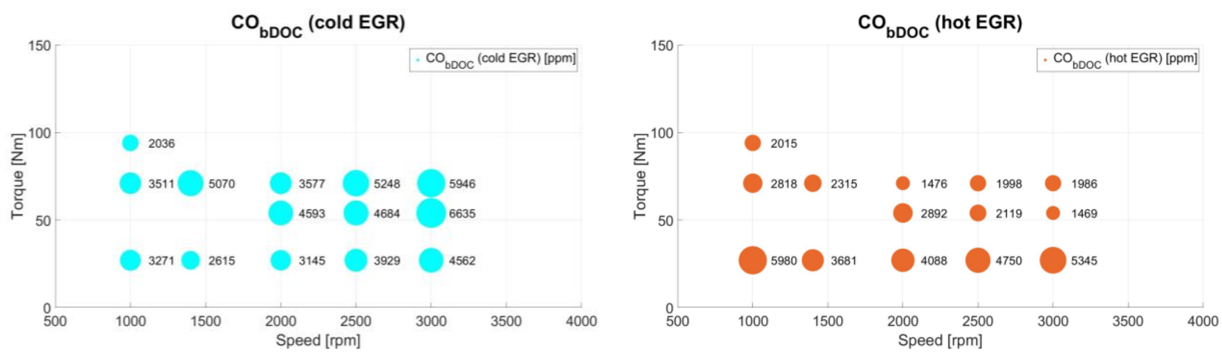
The selected variables have been varied among variation ranges suitable to the realization of a PCCI-like combustion event, for both hot and cold EGR configurations. For instance, as regards SOI, the lower limits (i.e., the least advanced values) were imposed taking into account the outputs of engine-out NO_x and soot emissions, avoiding a trade-off between these two pollutant species, like in a CDC mode, and keeping both of them at very low levels, typical of a PCCI-like combustion mode. Conversely, maximum values for SOI advance were limited by the increase in unburned HC and CO emissions, due to wall wetting and over-mixing phenomena. Limits on the highest EGR rate values (i.e., on the maximum throttled position of the exhaust flap) were imposed by two conditions: on the one hand, too much EGR flowrate generates high combustion instability, in terms of both cylinder-to-cylinder and cycle-to-cycle variations; on the other hand, increasing the engine backpressure by means of the exhaust flap, brings to a degradation of the fuel consumption and, after certain levels, increases combustion instability and the occurrence of misfiring, due to the difficulties involved in managing the inlet and outlet flows from the cylinders [22].

Starting from these preliminary tests, "optimal" calibrations for each engine operating point, both with hot and cold EGR configuration, have been selected minimizing (cf. Appendix) an objective function which considered the main engine-out emissions with different weighting factors. Results in terms of engine-out CO and HC emissions (bDOC), tailpipe CO and HC emissions (aDOC), as well as exhaust gas temperature before the after-treatment DOC are presented in the following Figures 2-6. In each of these figures, the blue and orange circles respectively refer to the cold and hot EGR configurations, while their dimensions are proportional to the corresponding values in ppm (arranged on the graphs in different speed/torque coordinates among the portion of the engine map under investigation).



(a) (b)

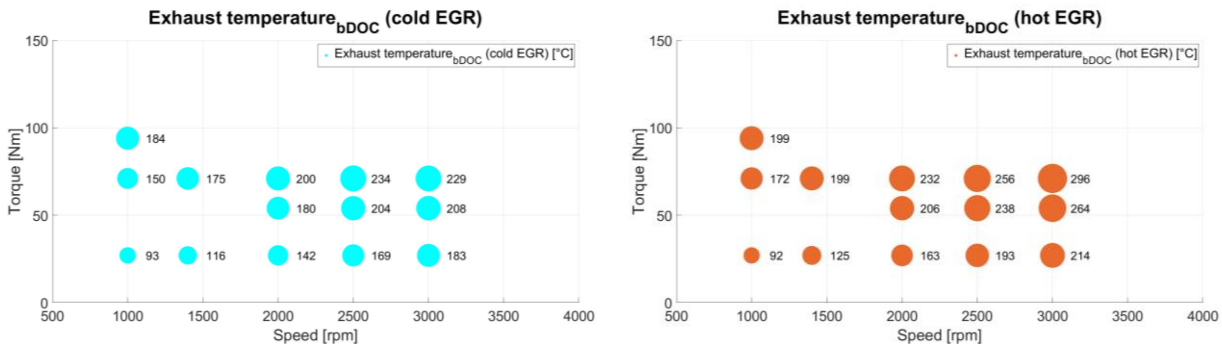
FIGURE 2. Comparison of engine-out HC emissions, with cold (a) and hot (b) EGR strategies.



(a) (b)

FIGURE 3. Comparison of engine-out CO emissions, with cold (a) and hot (b) EGR strategies.

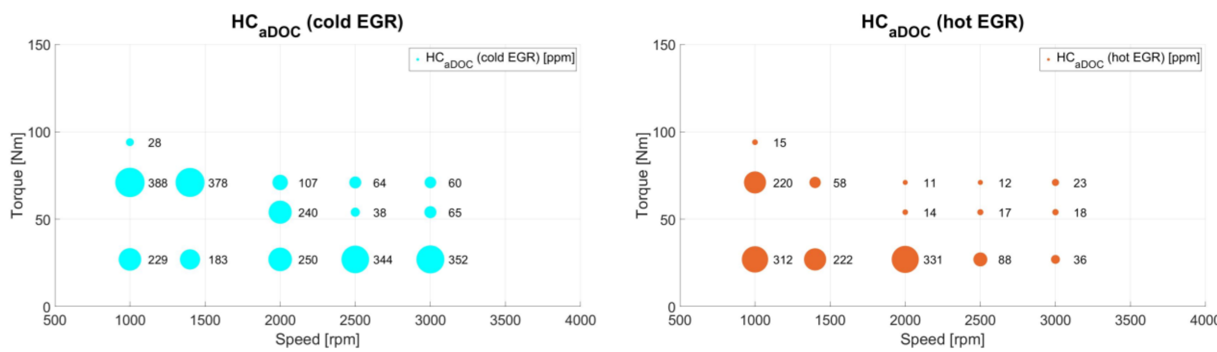
Figures 2-3 show respectively a comparison of the engine-out unburned HC and CO emissions from the engine when using cold or hot EGR strategies. The highly advanced single injection patterns, in both the considered EGR layouts, in addition to the relatively low engine loads considered, make the fuel injection occur in a low density and low temperature environment, likely responsible of fuel wall impingement and over-mixing phenomena, which determine very high levels of CO (Figure 2) and unburned HC (Figure 3) emissions, typical of PCCI operations. The dilution effect of the EGR gases also has a contribution on these enormous increase in HC and CO emissions with respect to conventional diesel combustion levels (cf. also results reported in [22]). Comparing Figures 2(a)-2(b) and Figures 3(a)-3(b), it is evident how the application of uncooled EGR, at the lowest considered load (i.e., *bmep* = 1 bar), tends to worsen the incomplete combustion processes of PCCI operations, with respect to the cooled EGR solution. This is likely because the additional thermal energy the hot EGR is able to give to the intake charge (which can promote HC and CO oxidation) is not sufficient to balance the negative effect of the reduction in the charge density and in the in-cylinder trapped mass obtained as the EGR temperature grows (which generates a further reduction in the in-cylinder oxygen quantity and an increased difficulty to oxidize HC and CO molecules [24]). On the other hand, at *bmep* ≥ 2 bar, the balance between the abovementioned two effects overturn, thus making the uncooled EGR strategy beneficial to reduce engine-out HC and CO emissions.



(a) (b)

FIGURE 4. Comparison of exhaust gas temperature before the DOC, with cold and hot EGR strategies.

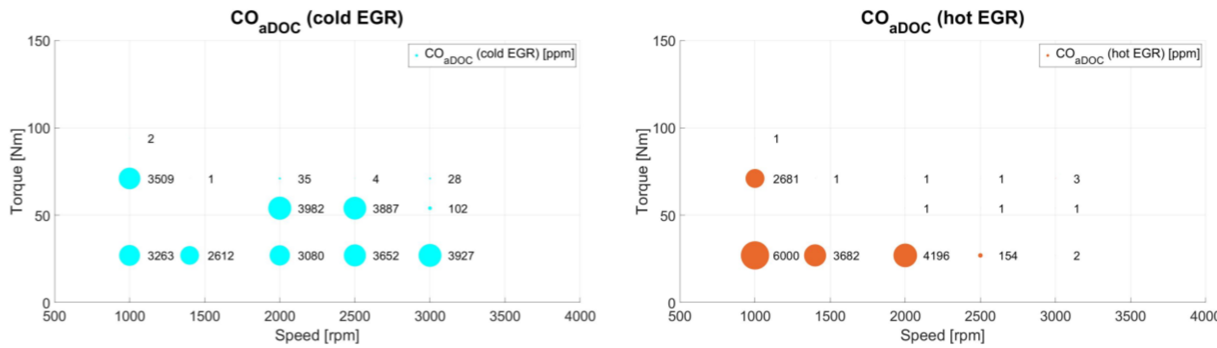
Figure 4 shows the exhaust gas temperature measured upstream the tailpipe DOC, for both cold, in Figure 4(a), and hot, in Figure 4(b), EGR configurations. In all the investigated engine working points , apart from the lowest load and speed case (1000×27), the use of uncooled EGR leads to higher intake gas temperatures (here not reported for conciseness reasons) and higher exhaust gas temperatures as well, with gains that range from around 10 to 60°C. The main advantage of this extra thermal energy upstream the tailpipe DOC is that it could be effectively exploited to reach the DOC light-off temperature [17], which has been verified not to be reached in the majority of the analyzed points, if a conventional cold EGR strategy is adopted (this will be detailed hereinafter in the following Figures) and whose importance is crucial, taking into account the very high level of engine-out pollutant emissions given by the engine run in PCCI operation and highlighted in Figures 2-3.



(a) (b)

FIGURE 5. Comparison of tailpipe HC emissions, with cold (a) and hot (b) EGR strategies.

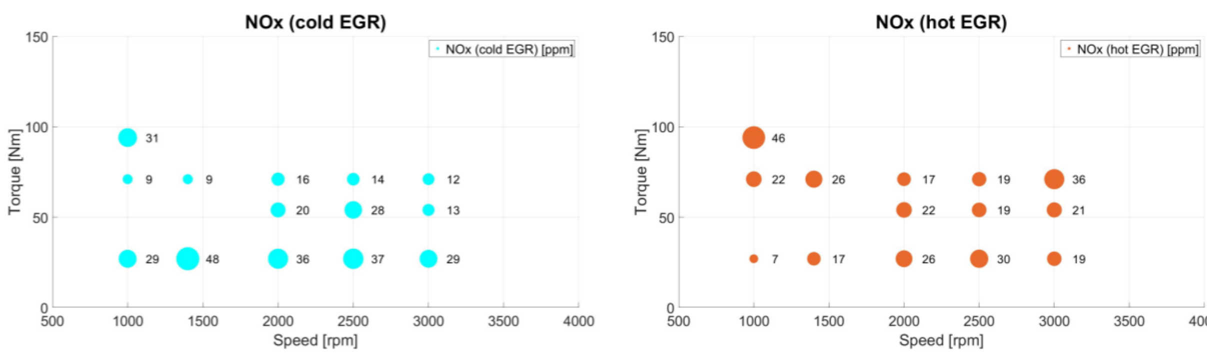
Figures 5-6 show respectively a comparison of the tailpipe (i.e., downstream the DOC) unburned HC and CO emissions when using cold or hot EGR strategies. The main advantage which can be highlighted by comparing Figures 5(a)-5(b) and Figures 6(a)-6(b) is an enlargement of the area of the engine map where unburned HC and CO tailpipe emissions are cut almost up to zero by the DOC activity, when implementing the uncooled EGR layout. This is due to the already mentioned increase of the exhaust gas temperature upstream the after-treatment system, which makes the DOC light-off temperature to be overcome. Still, at low speed and load conditions, uncooled EGR is not able to provide advantages with respect to the cold EGR case, and actually it provides slightly worse results at *bmep* = 1 bar when ranging from n = 1000 rpm to n = 2000 rpm, due to too small increase in the exhaust temperatures which does not balance the already explained increase in engine-out HC and CO emissions.



(a) (b)

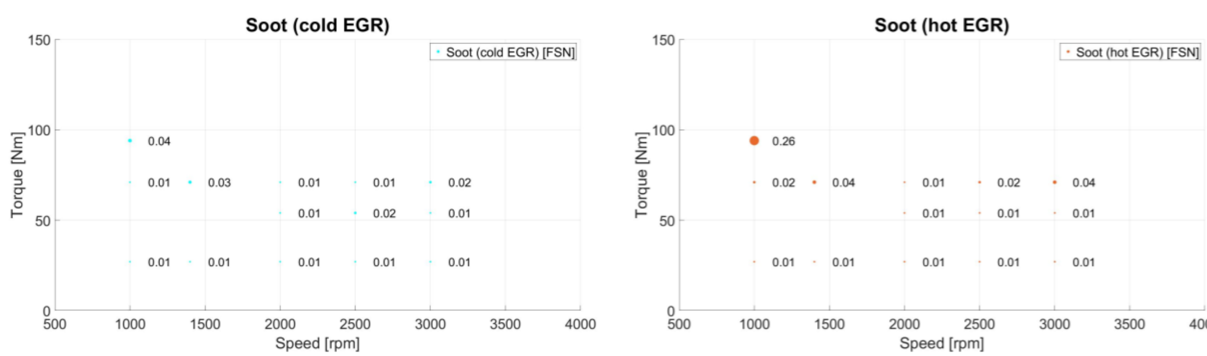
FIGURE 6. Comparison of tailpipe HC emissions, with cold (a) and hot (b) EGR strategies.

Due to the low load PCCI conditions, the absence of the EGR cooler does not affect significantly NO_x and soot emissions (cf. Figures 7-8), which keep at very low levels typical of PCCI operations. On the other hand, *bsfc* (not reported, for conciseness reasons) seems to be slightly reduced when using uncooled EGR at *bmep* = 1 bar, while at the higher loads considered it slightly worsens. Anyway, with both the analyzed EGR layouts, the fuel consumption is higher with respect to conventional diesel combustion operations run on the original version of the engine, being a well-known drawback introduced by the diesel premixed combustion.



(a) (b)

FIGURE 7. Comparison of exhaust NO_x emissions, with cold (a) and hot (b) EGR strategies.



(a) (b)

FIGURE 8. Comparison of exhaust soot emissions, with cold (a) and hot (b) EGR strategies.

4. CONCLUSIONS

In the present study, the possibility to exploit hot EGR strategies in a diesel engine purposely designed to run under PCCI combustion mode has been investigated at low load working points. Low-load PCCI operations, with a proper combination of high EGR rates and advanced fuel injection timings, have been realized with both cooled and uncooled EGR strategies, achieving substantially reduced NO_x and soot emissions with respect to a conventional diesel combustion mode. Unburned hydrocarbons and carbon monoxide emissions, as well as deteriorated fuel economy, are well known issues.

Increasing the temperature of the EGR gases, bypassing the EGR cooler, causes an increase in the intake temperature, but this increment, in the investigated low load operating points, does not lead to any appreciable deterioration in the NO_x and smoke emissions. As regards the incomplete combustion species, at the lowest investigated load (i.e., *bmep* = 1 bar), engine-out HC and CO emissions tend to worsen with respect to the cooled EGR solution, while at bmep ≥ 2 bar this negative effect disappears.

The major advantage in using hot EGR gases proved to be the following increase in the exhaust temperature (up to 60°C in the investigated engine operating points), which led to the possibility to enhance the after-treatment DOC activity: in fact, by means of the uncooled EGR strategy, the DOC light-off temperature was reached in a wider area of the investigated engine map, where a nearly complete abatement of tailpipe HC and CO emissions proved to be possible.

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APPENDIX

Details of the tested engine points, with cold and hot EGR configurations

Table 2 reports some relevant engine parameters (start of injection, rail pressure, intake gas temperature, EGR rate) pertaining to the "optimal" points featuring cold and hot EGR configurations.

TABLE 2. "Optimal" calibrations for cold and hot EGR configurations

		COLD EGR				HOT EGR			
Speed [rpm]	Torque [Nm]	SOI [°bTDC]	p_{rail} [bar]	T_{int} [°C]	EGR rate [%]	SOI [°bTDC]	p_{rail} [bar]	T_{int} [°C]	EGR rate [%]
1000	27	16	600	53	63.5	16	600	92	53.6
1000	71	22	1200	59	47.8	14	1600	98	39.3
1000	94	22	1000	61	50.2	14	1600	93	43.4
1400	27	18	600	55	58.8	18	600	99	40.7
1400	71	28	650	62	58.8	24	1600	111	41.7
2000	27	20	600	59	57.4	26	800	111	43.8
2000	54	22	1000	60	55.7	26	1400	128	58.6
2000	71	34	1000	55	46.9	34	1400	150	55.9
2500	27	24	600	57	53.9	30	800	116	37.0
2500	54	26	1000	58	53.3	34	1200	136	44.8
2500	71	34	1200	54	47.6	38	1200	170	57.3
3000	27	28	600	52	54.4	36	1200	128	41.5
3000	54	26	1400	60	56.0	38	1400	150	50.2
3000	71	34	1600	51	53.7	40	1600	176	51.5