

Potential of double pilot injection strategies optimized with the design of experiments procedure to improve diesel engine emissions and performance

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(Article begins on next page)

1 **POTENTIAL OF DOUBLE PILOT INJECTION STRATEGIES OPTIMIZED WITH A**
2 **DESIGN OF EXPERIMENTS PROCEDURE TO IMPROVE DIESEL ENGINE**
3 **EMISSIONS AND PERFORMANCE.**

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7 **ABSTRACT**

8 The potential of pilot-pilot-main triple injection strategies versus engine out emissions, combustion noise and brake
9 specific fuel consumption has been assessed experimentally on a Euro 5 diesel engine with a reduced compression ratio
10 (16.3:1). The engine has been fueled with conventional diesel fuel. The experimental tests on the engine have been
11 carried out in a dynamometer cell under different steady state working conditions, that are representative of passenger
12 car engine applications over the European homologation cycle. Furthermore, in-cylinder analyses of the pressure, heat-
13 release rate, temperature and emissions have been performed in order to obtain more detailed knowledge on the cause-
14 and-effect-relationships between the implemented injection strategies and the results of the experimental tests.

15 The implemented double-pilot injection engine calibrations have been optimized by means of the design of experiments
16 procedure. The plotted data of the engine performance and emissions have been compared with data from the original
17 double-injection schedule, characterized by a retarded main injection timing, in order to intensify the premixed
18 combustion phase. The benefits and the disadvantages of the *PCCI* concept are preliminarily discussed, on the basis of
19 the experimental pilot-main injection strategy results.

20 The substitution of the pilot-main injection schedule with the triple injection, for light engine loads and low engine
21 speeds, has led to higher combustion pressures, lower heat release rates, shorter ignition delays and lower brake specific
22 fuel consumption. Above all, a significant improvement in engine noise and in both *CO* and *HC* engine-out emissions
23 has been achieved and the *NO_x* emission have been limited by the application of high *EGR* rates. When medium engine
24 loads and speeds are analyzed, the considered double-pilot injection strategy allows the *NO_x* emissions to be reduced,
25 compared to the baseline pilot-main injection schedule. However, the combustion noise does not improve and the soot
26 deteriorates, even though the soot penalties are not relevant.

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27 **Keywords:** pilot injections; design of experiments; partial premixed charge compression ignition engines.

28 **Highlights:**

- 29 - The benefits and the weak points of the partial *PCCI* strategy for low loads and speeds are discussed.
- 30 - The effects of the triple pilot-pilot-main injection on engine-out emissions and noise are analyzed.
- 31 - The experimental tests on the multiple injections are supported by numerical in-cylinder analyses.

32 **1. INTRODUCTION**

33 The implementation of a pilot injection in diesel engines makes the entire amount of the fuel chemical energy be
34 released over a prolonged time interval, thus determining a longer combustion than for the single injection case.
35 Furthermore, the premixed combustion of some of the pilot injected fuel causes a slight increase in the in-cylinder gas
36 pressure and temperature [1] before the main injection has occurred, and therefore leads to a considerable reduction in
37 the ignition delay of the main injection [2]. This reduction in the fuel ignition delay limits the impact of the premixed
38 combustion and generates a less rapid heat release rate [3] during the main injection than in a single injection schedule
39 [4]. As a consequence, the main combustion becomes predominantly mixing-controlled [3].

40 A pilot injection that is sufficiently close to the main injection has the potential of enhancing combustion efficiency and
41 thus brake specific fuel consumption (*bsfc*), because the pilot and main combustions are linked smoothly [5]. Pilot
42 injections are also effective in decreasing combustion noise (*CN*), especially at engine idle [3]. Reductions of up to 5-8
43 dB are generally obtained in the *CN*, compared to single injection strategies [6-8].

44 Since the pilot injection decreases the impact of the overall premixed combustion, it makes the highest flame
45 temperatures diminish and, as a consequence, the NO_x emissions generally also reduce, compared to the single-injection
46 strategy [9]. However, large pilot injected quantities make the NO_x produced in the pilot combustion grow, and the
47 increase in the NO_x amount produced by the pilot combustion can surpass the decrease in the main combustion NO_x
48 emissions, due to the shortened ignition delay [10]. Furthermore, when heavy *EGR* rates (~60%) are employed, such as
49 in partial *PCCI* strategies, the pilot injection timing and mass do not influence the NO_x emissions to any appreciable
50 extent because the NO_x emissions are very small [5].

51 The smoke emission in pilot-main injections generally tends to increase, compared to single injections: In fact, the pilot
52 injection increases the in-cylinder temperature and decreases the oxygen concentration in the gases before the main
53 injection has occurred. Both of these effects generally make the smoke emission grow: the increased temperature
54 mainly acts by reducing the lift-off length, which pertains to the main injection, with a subsequent increase in the
55 equivalence ratios close to the nozzle. The insufficient mixing of fuel with air, which is also due to the shortened

56 ignition delay of the main injection, augments the percentage of the diffusion combustion in the main combustion and,
57 as a consequence, the final soot level grows [11]. In general, the quantity of the pilot injection should be below a certain
58 threshold (a general value of 4 mg is normal) in order to contain the smoke number [12].

59 Finally, *HC* and *CO* emissions reduce at low loads if a pilot injection is implemented, because the occurrence of
60 overmixing is ~~more~~ less likely. In particular, the *CO* conversion rate improves because of the relatively high in-cylinder
61 temperature and the shorter ignition delay of the fuel injected in the main injection [4].

62 The pilot injection can be exploited in different ways to improve engine-out emissions, *CN* and fuel consumption,
63 depending on the working condition [13]. Soot emissions are not relevant at low engine speeds and loads, *NO_x* and
64 noise are usually controlled, at these conditions, by means of adequate *EGR* rates. The pilot injection is generally
65 optimized, on the basis of the *EGR* rate, in order to reduce *HC* and *CO* emissions [14], which tend to be high, due to the
66 presence of lean and cool regions. The *HC* and *CO* emission situation becomes worse at engine cold start and warm-up,
67 when the oxidation catalyst has less conversion efficiency. Soot, *NO_x*, noise and *bsfc* are the dominant problems at
68 medium load conditions, that is, in the higher load zone of the *NEDC* region, whereas *HC* and *CO* are not of great
69 concern. Pilot injections are therefore used in these conditions to improve *PM-NO_x* and *bsfc-NO_x* trade-offs and, above
70 all, *CN* [15].

71 Pilot injection is generally applied to the *NEDC* area, but it can also be used for other purposes and offers other benefits.
72 An early pilot injection can be applied to increase the in-cylinder pressure at the end of the compression stroke during
73 engine cranking, thus reducing the engine start time. Furthermore, pilot-main injection patterns reduce the cycle-to-
74 cycle variability of the torque, compared to single injections [4], and this induces more stable engine operation,
75 especially after the engine crank phase [16]. Finally, pilot injection can be used at full load to limit the peak in-cylinder
76 pressure and the engine exhaust temperature. The noise due to combustion is less important at these engine working
77 conditions, since other sources of noise dominate in the vehicle, and the pilot injection therefore allows either the fuel
78 rate to be increased or the mechanical and thermal stresses in the engine to be reduced, thus providing possible weight
79 savings or simplifications of the cooling circuit. Instead, when the maximum torque is smoke limited, an early pilot
80 injection can increase the full load torque by improving the utilization of the air within the cylinder, compared to the
81 case of a single injection with a long energizing time [17]. In general, the pilot injection shot can also be used at high
82 loads to reduce soot and improve combustion efficiency, since the main injection duration can be shortened.

83 The fundamental pilot-main injection scheme constitutes the conceptual basis for the development of more
84 sophisticated and advanced multiple injection strategies that can implement multiple pilot injection shots. Pilot-pilot-
85 main injection schedules have been shown to have a great potential toward noise [18, 19], emission [20, 21] and *bsfc*
86 [20, 22, 19] reductions. However, more trials are required to optimize various engine parameters, such as the *EGR* rate,

87 the swirl actuator position, the boost pressure, the dwell-time, the injection timings, the rail pressure and the energizing
88 times of all of the injection shots, and thus to be able to fully exploit these strategies [23]. In the present work, a design
89 of experiment (*DoE*) procedure has been applied to optimize the double-pilot injection engine calibration. This
90 innovative approach allows the effective benefits of this injection strategy to be assessed, since optimized pilot-pilot-
91 main and pilot-main injection engine calibrations are compared. In general, the aims of the double-pilot injection
92 strategy should be selected on the basis of the engine working conditions and the installed aftertreatment devices. The
93 triple injection in the current investigation is principally aimed at minimizing NO_x and combustion noise. Furthermore,
94 the double- and triple-injection strategies are tested under high *EGR* conditions, whereas most of the research on
95 multiple injections has been conducted under low or moderate *EGR* rates (moderate *EGR* rates correspond to *EGR*
96 fractions up to 30-40%) [24].

97 **2. EXPERIMENTAL FACILITIES AND ENGINE SETUP.**

98 The experimental tests have been carried out on the dynamic test bed installed at the Politecnico di Torino *IC*
99 laboratories. The test rig is equipped with an ‘*ELIN AVL APA 100*’ cradle-mounted *AC* dynamometer, featuring
100 nominal torque and power of 525 Nm and 220 kW, respectively, as well as a maximum speed of 12000 rpm. The
101 facility is capable of full four-quadrant operation with high speed and torque dynamics, as well as the simulation of zero
102 torque and gear shifting oscillations in the drivetrain.

103 The test facility is equipped with a ‘*Pierburg AVL AMA 4000*’ raw exhaust-gas analyzer, which is basically made up of
104 three analyzer trains. Two of these trains feature the following modules: one heated flame ionization detector for the
105 *THC* analysis, one heated chemiluminescence detector for the analysis of the NO_x , three nondispersive infrared
106 analyzers for the measuring of low as well as high *CO* and CO_2 concentration levels and one paramagnetic oxygen
107 detector for the O_2 levels. These two trains allow the pollutant emissions to be measured simultaneously, upstream and
108 downstream of the aftertreatment system. The third train is made up of a CO_2 concentration detector for the measuring
109 of the CO_2 concentrations in the inlet manifold, in order to be able to calculate the *EGR* mass fraction, which is defined
110 as $X_{EGR} = \dot{m}_{EGR} / (\dot{m}_{EGR} + \dot{m}_a)$, according to the procedure developed in [25].

111 As far as the particulate matter (*PM*) measurement is concerned, the dynamic test bed is equipped with the following
112 instruments: *AVL 415S* smokemeter, *AVL 439S* opacimeter and *AVL SPC472* Smart Sampler. Finally, an ‘*AVL KMA*
113 *4000* Methanol’ measuring system continuously meters the engine fuel consumption. This system is based on the *AVL*
114 *PLU* measuring principle of a servo-controlled positive displacement counter, and it can perform measurements over
115 the 0.28-110 kg/h range with a reading accuracy of 0.1% for diesel fuel.

116 All of the abovementioned measurement devices are controlled by a *PUMA OPEN 1.3.2* automation system, which also
117 includes *ISAC 400* software for the simulation of the behavior of both the vehicle (road load, road gradient and
118 moments of inertia of the driveline components, which are not physically present on the test bed) and of the driver
119 behavior (use of the clutch, accelerator pedal and gear shifting).

120 The tested engine, the main features of which are reported in Table 1, is a Euro 5 engine fueled with conventional diesel
121 oil. It has been fully instrumented with piezoresistive pressure transducers and thermocouples for the measurement of
122 the pressure and temperature levels at the following locations: upstream and downstream of the compressor, at the inlet
123 manifold, upstream and downstream of the turbine and downstream of the aftertreatment system. Additional
124 thermocouples have also been installed for the measurement of the temperatures downstream of the intercooler, in the
125 four inlet and exhaust runners, as well as upstream and downstream of the *EGR* cooler. Finally, an *UEGO* air-fuel ratio
126 sensor has been located within the exhaust system. The acquisition of all of these time-averaged quantities are directly
127 managed directly by the *PUMA OPEN 1.3.2* system, through a dedicated firewire front-end module, which can manage
128 up to 48 analog input channels with a maximum data capture rate of 5 kHz per channel.

129 A high-frequency piezoelectric transducer has been installed on the engine cylinder head to measure the pressure time-
130 history of the gases in one of the cylinders, whereas another high-frequency piezoresistive transducer has been used to
131 detect the pressure levels in the inlet runner of the same cylinder in order to reference the in-cylinder pressure. An *AVL*
132 *365C* crank-shaft driven encoder generates the time base for an automatic 14 bit data-acquisition system (based on the
133 *AVL* indimodul 620 system), which is capable of acquiring up to 8 channel data with a maximum frequency of 800 kHz
134 per channel. The acquisition system is managed by *AVL Indicom* software, in order to allow both the online analysis of
135 the indicated cycle and data storage operation for post-processing with a validated three-zone combustion diagnostic
136 tool [26]. In this model, the combustion chamber content is divided into three zones: a fuel zone, an unburned gas zone,
137 (containing fresh-air, residual gas and *EGR*) and a burned gas zone obtained from a global stoichiometric combustion
138 process. Ordinary differential mass and energy conservation equations are applied to the three zones and are solved
139 numerically on the basis of the experimental in-cylinder pressure. The model allows the temperatures of the three zones
140 to be predicted as functions of the crank angle. Furthermore, thermal and prompt NO mechanisms are implemented in
141 the code, according to the Zeldovich and Fenimore submodels, respectively. The soot formation is modeled [27] by
142 means of an expression that uses the mean air-fuel ratio over the combustion interval, whereas the soot oxidation rate is
143 modeled using an empirical formula, based on the temperature of the burned gas zone.

144 3. THE DESIGN OF THE EXPERIMENT PROCEDURE

145 The tested engine was calibrated by the *OEM* with a double injection strategy, which represented the state-of-the-art
146 pilot-main injection schedule for the considered engine technology.

147 The *ppM* injection strategies have been optimized by adopting the statistical design of the experiments (*DoE*) technique.

148 The following parameters were considered as the most relevant input variables for the procedure: rail pressure (p_{Rail}),
149 swirl actuator position (S_w), dwell times (DT) between consecutive injections (DT_2 between the pilot 2 and pilot 1 shots
150 and DT_1 between the pilot 1 and main shots, where pilot 1 is the closest shot to the main injection and pilot 2 the
151 furthest shot from the main injection), main injection timing (SOI_{Main}), the injection quantities in each shot (q_{Pil1} and
152 q_{Pil2}) and the inducted air per stroke and per cylinder (m_a).

153 Key-points are engine working points (characterized in terms of *bmep* [bar] and speed n [rpm]), considered as
154 representative of the engine application to a passenger car over the new European driving cycle. The following key-
155 points were considered for the tested engine ($n \times bmep$): 1500×2 , 1500×5 , 2000×2 , 2000×5 , 2500×8 , 2750×12 and idle.

156 Tables 2 and 3 report (second column) the parameter levels that were considered in the variation lists for the
157 optimization of the *ppM* injection schedule at the 1500×2 and 2000×5 key-points. The center and the extreme values of
158 the range that were considered for each parameter were chosen on the basis of preliminary measurements. An
159 appropriate number of levels was selected in order to obtain accurate results with a reasonable number of tests for each
160 variation list. The quantity of fuel in the main injection is set by the test-bench control system in order to guarantee the
161 *bmep* value, and is therefore not present as a parameter in the variation list. The *EGR* ratio affects the emissions at the
162 diesel engine exhaust to a great extent. However, the *ECU* does not evaluate this parameter directly, but can measure m_a
163 (by means of the air mass flowmeter), which is intimately connected to *EGR*. Therefore, the information related to the
164 induced air, m_a , was considered in the variation lists, instead of the *EGR* ratio.

165 The preliminary variation list was obtained using the Matlab Model-Based Calibration toolbox, setting a V-optimal type
166 design of experiments, which minimizes the prediction error variance, and a full factorial series, as the candidate set, on
167 the basis of the levels shown in Tables 2 and 3. The preliminary variation list was then randomized and replications of
168 the central point (defined by the center value of each parameter range) were added every 10-15 points in order to further
169 reduce the prediction error variance and check for any possible drifts of the output variables for fixed input parameters.
170 The final variation lists were made up of 120-150 tests for each considered key-point. Once the variation list tests had
171 been carried, it was possible to obtain quadratic models of the output variables as functions of the input variables and of
172 their interactions.

173 The engine-out specific NO_x , CO , HC and soot emissions, the *bsfc* and the *CN* were considered as the output variables.

174 Different targets can be introduced for the output variables in order to select the best set of values for the input variables

175 at each key point, that is, the optimized engine calibration. The optimization procedure consists of a number of
176 constraints on the output variables. These constraints depend on the pollutant emission regulations, on the aftertreatment
177 devices that are installed on the engine, on the CO_2 targets and on aspects related to fun to-drive.

178 The considered Euro 5 engine was equipped with a diesel oxygen catalyst (DOC) and a particulate filter, but no
179 aftertreatment device was designed to reduce the NO_x emissions. The optimization strategy for the triple (pilot-pilot-
180 main) injection schedules, based on the DoE , was aimed at minimizing NO_x emissions and at reducing the combustion
181 noise with respect to the pilot-main injection calibration, which was originally implemented in the ECU provided by the
182 engine OEM . However, rather severe upper limits were also set for CO , HC and $bsfc$.

183 Tables 4 and 5 show the reference values of the output variables for the pilot-main injection strategy and the constraints
184 used for the optimization of the triple injection strategy. The optimum values of the input variables, calculated by means
185 if the DoE procedure, are reported in the third column of Tables 2 and 3. EGR trade-offs were performed in the
186 neighborhood of the calibration baseline points for both the double and the optimized triple injection strategies in order
187 to compare not only the baseline points of the two calibrations, but also two complete curves.

188 4. LIGHT LOAD CONDITIONS.

189 4.1 PCCI-like double-injection strategies

190 Figures 1-3 report the in-cylinder pressure (p_{cyl}), the heat release rate (HRR) and the burned gas temperature (T_b) time
191 histories, respectively, for $n = 1500$ rpm and $bmeP=2$ bar. The continuous curves with square- and circle-symbols refer
192 to $X_{EGR}\approx 50\%$ and $X_{EGR}\approx 28\%$, respectively (these two operating conditions correspond to high and moderate EGR rates),
193 and the same pilot-main (pM) strategy is adopted in both cases. Since the injection strategy is the same, these
194 preliminary tests are aimed at assessing the effect of the EGR rate in $PCCI$ engines. The p_{cyl} trace has been measured by
195 means of the piezoresistive pressure transducer installed in the combustion chamber, while the HRR , and the T_b time
196 histories have been calculated by means of the three-zone combustion diagnostic tool.

197 A relatively high dwell time between the pilot and the main injection ($DT\approx 1400 \mu s$) has been implemented (cf. Fig. 2)
198 and a vigorous swirl has been applied to promote the air-to-fuel mixing. The heavy EGR rate condition that corresponds
199 to $X_{EGR} \approx 50\%$ has been applied in order to prolong the fuel ignition delay and obtain a partially homogeneous mixture
200 before ignition [28]. Fig.1 shows that the in-cylinder pressure decreases as the EGR is increased. In fact, both the flow-
201 rate through the turbine and the upstream pressure are reduced when high EGR rates are applied to a short-route EGR
202 system (cf. also the schematic of the engine in Table 1). As a consequence, the system may not be able to maintain the
203 desired boost level and a decrease in the boost may therefore be experienced at high EGR rates, especially for low

204 loads. The decrease in the in-cylinder pressure with increasing EGR in Fig. 1 is due to the reduction in the boost
205 pressure and to the increase in the temperature of the cooled EGR , compared to the temperature of the fresh air coming
206 from the engine intercooler. Figs. 2 and 3 show that high fractions of cooled EGR and retarded main injection timings
207 allow the maximum HRR and the T_b peak value to be contained [29], and the ignition delay of both the pilot and main
208 injected fuel to be lengthened, compared to the moderate EGR rate condition ($X_{EGR}\approx 28\%$). In particular, it can be
209 observed that the pilot combustion in the $X_{EGR}\approx 50\%$ case exhibits a two-stage ignition with the presence of both cool
210 and hot flame reactions, whereas single-stage pilot combustion occurs at $X_{EGR}\approx 28\%$.

211 The considered pilot-main (pM) injection schedule realizes a highly premixed combustion concept, since the main
212 combustion event starts when the main injection has finished. Most of the fuel injected during the pilot and the main
213 shots burns in premixed combustion conditions. The HRR peak, related to the diffusive combustion of the main injected
214 fuel, can be seen in Fig. 2 for $X_{EGR}\approx 28\%$, but vanishes for $X_{EGR}\approx 50\%$. The advantage of the implemented strategy is that
215 it induces a simultaneous reduction in soot and NO_x emissions, due to the intensified fuel premixing and to the reduced
216 combustion temperature. Fig. 4 shows that both the soot and NO_x emissions decrease when the EGR rate is increased
217 progressively, while the other engine parameters remain constant (the contoured triangle symbol represents the EGR
218 rate of the baseline pilot-main injection calibration); this behavior, with respect to X_{EGR} , is not observed when more
219 conventional double-injection patterns are applied. Fig. 5 shows the gas temperature at the diesel oxygen catalytic
220 catalyst inlet (T_{cat}) as a function of the EGR rate. The experimental points, evaluated as functions of NO_x in Fig 4 and as
221 functions of EGR rate in Fig.5, are the same (maximum NO_x corresponds to minimum X_{EGR} and vice versa).

222 Higher X_{EGR} levels than 50% are in line with partial $PCCI$ applications, which intensify the local mixing of the fuel
223 plume and the charge, with the production of a premixed stratified charge. Furthermore, the selected engine
224 compression ratio was $\varepsilon=16.3$, which falls between the typical values of conventional diesel engines ($\varepsilon=17\div 18$) and the
225 characteristic values of partial $PCCI$ engines ($\varepsilon=13\div 16$, [30]). The reduced compression ratio makes the temperature
226 and pressure, which are closely related to NO_x formation, decrease during the compression phase. This enables the fuel
227 spray to penetrate further with more air entrainment, thus contributing to a decrease in the soot [31], which is also due
228 to the increase in the fuel autoignition delay [32, 33]. Finally, a toroidal combustion-bowl was selected, in line with
229 partial $PCCI$ applications, since it assures a rapid fuel mixing when combined with a high swirl number, and a large
230 bowl piston diameter was designed in order to reduce the occurrence of wall impingement. However, unlike typical
231 partial $PCCI$ combustion diagrams, the pilot injected fuel does not burn together with the main injected fuel, and a pilot
232 combustion event, which is not connected to the main combustion, can be observed in the HRR traces reported in Fig. 2.
233 Furthermore, the ratio of the quantity injected in the main shot to that injected in the pilot shot is significantly higher
234 than that usually adopted in partial $PCCI$ engines.

235 Pilot injection quantities with early injection timings, like the typical ones used in early *PCCI* injection strategies (40-50
236 *BTDC* degrees), can cause the fuel vapor to spread to the cylinder liner, because the in-cylinder charge pressure and
237 density are low for early injection timings and light loads. This leads to overmixed regions and wall quenching
238 phenomena, both of which are important sources of *HC* and *CO* emissions. In addition, the possible spray impingement
239 on the wall surfaces dramatically increases the amount of unburned hydrocarbons [34], dilutes the lubrication oil and
240 causes the fuel consumption to increase to a great extent, since part of the pilot injected fuel is wasted and unable to
241 ignite. For these reasons, the relatively long dwell time in the pilot-main injection pattern, which is reported in Fig. 2,
242 has been introduced by further delaying the main injection rather than by advancing the pilot injection. Since the pilot
243 injection does not occur very early during the piston compression stroke, and the pilot injected quantity is contained
244 ($V_{pil} \approx 1.7 \text{ mm}^3$), wall impingement occurrence is not a concern. Furthermore, the retarded main combustion contributes
245 to the generation of a reduced soot formation rate because the peak in-cylinder temperature around *TDC* is contained
246 and an enhanced soot oxidation rate can be observed during expansion and blowdown phases, due to the raised burned
247 gas temperatures during the last part of the expansion stroke and at the engine exhaust.

248 The considered *pM* injection pattern features high levels of *HC* and *CO* emissions, due to low-temperature combustion
249 [35] and fuel overmixing, as well as elevated combustion noise, due to the highly premixed combustion. Furthermore,
250 the *bsfc* become worse, compared to the double injection strategies implemented in conventional diesel engines, due to
251 the retarded main injection timing (SOI_{Main}) and the diminished ε value. These drawbacks are of the same typology as
252 those encountered in classic partial *PCCI* engines featuring late injection strategies.

253 A minimum temperature level of 200°C is necessary at the catalytic converter inlet (T_{cat}) to obtain a satisfactory
254 efficiency of the diesel oxygen catalyst (*DOC*) for the conversion of the high *HC* and *CO* engine-out emissions at low
255 loads. The T_{cat} values in Fig. 5 can be seen to be higher than this threshold for $X_{EGR} > 45\%$.

256 Since the main injected quantity is much larger than the pilot injected mass, the SOI_{Main} , which is equal to 1° *CA ADTC*
257 in Fig. 2, has not been delayed any further, as occurs in typical late *PCCI* injection strategies, in order to avoid an
258 excessive *bsfc* penalty. In fact, the SOI_{Main} varies within the 3-10° *CA ATDC* range for late *PCCI* injection strategies,
259 whereas it is usually in the 5-7° *CA BTDC* range for diesel engines with conventional combustion systems.

260 **4.2 Triple injection strategies.**

261 Figures 6-8 show comparisons of *HC-NO_x*, *CO-NO_x* and *bsfc-NO_x* *EGR* trade-off curves obtained for two different
262 engine calibrations in the 45% < X_{EGR} < 55% range and at $n = 1500 \text{ rpm}$ and $bmep = 2 \text{ bar}$. Fig. 9 instead plots the *NO_x*-
263 X_{EGR} curves for the two strategies. The triangle symbols in Figs. 6-9 pertain to the previously discussed *pM* injection
264 engine calibration, whereas the circle symbols refer to a pilot-pilot-main (*ppM*) injection engine calibration. The

265 contoured line symbols correspond to the baseline calibration points of the two strategies. The triple injection baseline
266 calibration has been obtained with the *DoE* campaign; not only has a pilot shot been added to the injection train of the
267 baseline point of the *pM* calibration, but the rail pressure, injection timings, energizing times and other engine
268 parameters have also been changed. The *SOI* of the pilot 1 and pilot 2 injections are in the 10-20 cad *BTDC* range for
269 the *ppM* strategy, and are in line with the literature results concerning the best pilot injection timings in triple injections
270 [4].

271 It can be observed, from Figs. 6-9, that the *DoE* optimized *ppM* strategy generally allows the *CO* and *HC* emissions to
272 be improved at the same NO_x , with respect to the baseline double injection strategy. Furthermore, a slight enhancement
273 can be detected for the *bsfc-NO_x* trade-off. If reference is made to the calibration baseline points, the NO_x engine-out
274 emissions reduce in the *ppM* case (Fig. 9), in line with [36]. However, the NO_x-X_{EGR} curve is virtually the same for the
275 two strategies and the effectiveness of the *EGR* on the engine-out NO_x emissions therefore does not change for either of
276 the two calibrations.

277 The *ppM* pattern should lead to a decrease in the local air-to-fuel ratio, with respect to time and space, due to the lower
278 global oxygen concentration ($[O_2]_{int}=14.9$ versus $[O_2]_{int}=16.0$ of the *pM* calibration). The generation of a suitable fuel
279 vapor stratification close to the nozzle reduces the impact of fuel overmixing and wall quenching, and thus decreases
280 the engine out *HC* and *CO* emissions (cf. Figs. 6 and 7). In other words, the pre-combustion, which is due to the
281 introduction of the pilot 1 injection prior to the main injection, plays a role in attaining a sufficient main combustion
282 ignition and in improving the conversion efficiency of the fuel, and thus in enhancing the complete combustion of the
283 main injection. The improvements obtained for the *bsfc-*, *HC-* and *CO-NO_x* tradeoffs, by means of the pilot-pilot-main
284 injection, are in line with the results found in [19-22] for low loads and speeds. In particular, when *EGR* rates close to
285 50% are applied, two pilot injections are recommended [15] in order to decrease the *HC* emissions.

286 The p_{cyl} and the *HRR* curves that refer to the calibration baseline points of both the *pM* and the *ppM* strategies are
287 reported in Figs. 10 and 11, respectively. The *pM* curves have been plotted with solid line and triangle symbols,
288 whereas the *ppM* solid line curves are marked with circle symbols. The two-stage autoignition delay of the pilot 2
289 injection increases for the *ppM* strategy because the pilot 2 injection takes place earlier in the compression stroke, where
290 the charge pressure and temperature are lower. Furthermore, the maximum p_{cyl} value pertaining to the main injection
291 increases when passing from the *pM* to the *ppM* injection schedule, but the *HRR* pilot peaks and the *HRR* main
292 combustion peak reduce when pilot 1 injection is applied because of the decrease in the premixed combustion portion.
293 This evidence on p_{cyl} and *HRR* proves that the combustion performance has improved, and the p_{cyl} and *HRR* trends with
294 the number of injections can be confirmed from those obtained passing from a single injection to one-pilot injection. As
295 can be seen in Fig. 12, the decrease in the maximum *HRR* and the earlier main combustion induce a slight diminution in

296 temperature T_b of the burned gases for $\theta \geq 370^\circ CA$ in the case of the *ppM* strategy. On the other hand, the earlier SOI_{pil2}
297 of the *ppM* strategy advances the time instant at which the burned gas temperature jumps to high levels and, as a
298 consequence, the residence time in which the burned gases are exposed to higher temperatures than 1900 K increases
299 for the triple injection calibration. The NO_x formation rates are very sensitive to flame temperatures above 1900-2000 K
300 [37], but, on the basis of the Kamimoto-Bae diagram, NO_x emissions are only produced for smaller local equivalence
301 ratios (ϕ) than 1.5 [38]. A larger amount of mixture with relatively high ϕ values should be obtained in the fuel spray for
302 the *ppM* strategy, because of the reduced mixing with air and, consequently, the NO_x emissions can reduce. It is the
303 contribution of the main combustion that makes the final levels of NO_x higher for the *pM* injection schedule (cf. Fig.
304 13).

305 When the two pilot shots are applied, the *HRR* curve (Fig. 11) remains uninterruptedly higher than zero from the start of
306 the cool flames pertaining to the first pilot injection till the end of the main injected fuel combustion. Fig. 14 shows the
307 crankshaft angle ($MFB50$) that corresponds to a fuel mass burned fraction equal to $x_b=0.5$, and the diagrams plotted in
308 Figs. 11 and 14 justify the slight *bsfc* improvement, which in Fig. 8 generally results from the application of the *ppM*
309 strategy. In fact, the combustion heat is released closer to the *TDC* in the *ppM* case than in the *pM* one, in part due to the
310 more advanced SOI_{Main} .

311 The fuel for the *ppM* strategy burns with a more regular combustion rate, as can be seen in Fig.11 and without high-time
312 derivatives in the burned gas mass fraction time history (cf. also Fig. 14). As a consequence, a remarkable decrease in
313 the combustion noise can be expected, and the *CN-NO_x* curve in Fig. 15 in fact improves significantly for the triple
314 injection strategy, compared to the *pM* injection schedule (reductions of up to 3.5 dB can occur). This is a consequence
315 of the decrease in the dwell times between the consecutive injection shots, compared to the *pM* case. In fact, the
316 reduction in the premixed combustion portion for the *ppM* strategy makes combustion noise decrease. The advanced
317 SOI_{Main} in the *ppM* injection pattern was purposely implemented in order to reduce the combustion noise. The mass
318 fraction of burned fuel, before the start of the main injection in Fig. 14, is equal to 15% and to 12% for the *pM* and the
319 *ppM* strategies, respectively, while the minimum noise condition generally corresponds to approximately 8% [8].

320 Figures 16 and 17 show the combustion noise Fourier spectra, evaluated at $n=1500$ rpm and $bme_p=2$ bar, for the *pM* and
321 the *ppM* strategies, respectively. The solid bars in each figure refer to the $X_{EGR} \approx 49 \div 50\%$ case, whereas the hatched bars
322 refer to $X_{EGR} \approx 52 \div 53\%$. A frequency range (500÷2000 Hz), in which the spectral combustion noise takes on the highest
323 values, exists in each Fourier spectrum. The presence of a pronounced peak zone, which occurs within a narrow
324 frequency band, is more evident for the *pM* diagram at $X_{EGR} \approx 49 \div 50\%$, even though the peak zone tends to disappear as
325 the value of X_{EGR} is increased. The presence of a peak zone in the diagrams in Figs. 16 and 17 indicates that there is an
326 overly dominant frequency range in the signal, which gives tonality to the noise. In other words, the higher the peak

327 intensity and the smaller the extension of the peak zone, the more recognizable the combustion noise as a specific noise.
328 The sensitivity of the combustion noise to *EGR* variations is limited for the *ppM* strategy, while the tonality is more
329 reduced for the *pM* pattern as the *EGR* passes from $X_{EGR} \approx 50\%$ to $X_{EGR} \approx 53\%$. Since the human ear is very sensitive to
330 changes in noise, a triple injection can contribute to a less irritating perception of the combustion noise when the *EGR*
331 rate is modified.

332 Finally, the increase in soot emissions, which can be observable in Figs. 18 and 19 for the triple injection, is not of real
333 concern since the smoke emissions continue to show relatively low values (Soot < 0.31 g/kWh in Fig. 18 and Soot < 0.01
334 mg in Fig. 19), due to the low peak in-cylinder temperatures and the high relative air-to-fuel ratio (λ) for these engine
335 working conditions. The presence of a soot- NO_x trade-off curve, in the case of the *ppM* injection strategy, shows that
336 the combustion does not feature *PCCI*-like behavior, unlike the *pM* case.

337 All of the previous explanations and conclusions about the effects that the addition of a second pilot injection has on
338 engine emissions, combustion noise and *bsfc* performance have been based on experimental data at $n=1500$ rpm and
339 $bme_p=2$ bar, but they can be extended to the whole area for light loads, from low to medium engine speeds. This can be
340 confirmed from the experimental results plotted in Figs. 20-24, which refer to $n=2000$ rpm and $bme_p=2$ bar. In
341 particular, the double-pilot injection strategy offers the best potentiality to reduce *CN*, *HC* and *CO* with limited
342 penalties on soot emissions. The *ppM* strategy can also lead to an improvement in the startability of a cold engine,
343 which is a typical problem for low-compression ratio engines. In fact, the longer ignition delay and poorer vaporization
344 of the fuel can inhibit fuel ignition, but double-pilot injections are able to contrast this tendency.

345 5. MEDIUM-LOAD CONDITIONS.

346 Figures 25 and 26 report the *HRR* and x_b time histories, calculated at $bme_p=5$ bar and $n=2000$ rpm for the calibration
347 baseline points of the *pM* and *ppM* strategies, respectively. The *pM* strategy in Fig. 25 does not feature any cool flames
348 because the in-cylinder pressure and the temperature values at the pilot injection timing are more elevated than in the
349 $bme_p=2$ bar and $n=1500$ rpm case. Instead, cool flames are present in the *HRR* diagram pertaining to the *ppM* strategy,
350 due to the early injection timing of the pilot 2 shot. The pilot combustion in the *pM* strategy (cf. the solid line with
351 triangle symbols close to $\theta \approx 355^\circ$ CA in Fig. 25) is more vigorous than in the *ppM* strategy (cf. the solid line with circle
352 symbols close to $\theta \approx 358^\circ$ CA in Fig. 25), due to the presence, in the former case, of a clear single-stage ignition. On the
353 other hand, since the single-stage ignition delay of the main injected fuel is longer in the *ppM* strategy, the premixed
354 combustion peak at $\theta \approx 373^\circ$ CA is higher than in the *pM* case. An appreciable mixing-controlled phase is also present at
355 the end of combustion in both the *pM* and the *ppM* strategies.

356 Figures 27 and 28 show that the b_{sfc} and CN have almost the same levels at $b_{mep}= 5$ bar and $n= 2000$ rpm for the two
357 strategies. HC (Fig. 29) and CO (Fig. 30) become worse for ppM , mainly due to the longer autoignition delay, but it
358 should be pointed out that these emissions are not a reason for concern at the considered engine working condition.

359 Figure 31 shows that the NO_x - X_{EGR} dependence is the same for the two strategies. An appreciable improvement in the
360 NO_x emissions can be observed in Fig. 32 (these data have been obtained from the three-zone model) for the baseline
361 calibration point when passing from the pM strategy to the ppM one. The differences in the NO_x emissions between the
362 two calibration baseline points could be the result of the different ϕ distributions in the combustion zone, because the
363 peak T_b values and the residence times at higher temperatures than 1900 K are similar for the two calibrations (cf. Fig.
364 33). Furthermore, the higher premixing degree of the pilot injected fuel for the pM injection case makes the NO_x
365 emissions increase significantly in the $350^\circ CA \leq \theta \leq 370^\circ CA$ range, compared to the ppM strategy.

366 A slight deterioration in the soot- NO_x EGR trade-off can be observed in Fig. 34 for the ppM calibration, even though the
367 maximum soot values in Figs. 34 are not critical, when a diesel particulate filter is mounted. In general, the increased
368 soot emissions represent a penalty for the considered ppM strategy for medium load and speed conditions (the soot
369 values in Fig. 34 are much higher than in Fig. 18). Any diminution in the ignition delay, for conventional diesel
370 combustion, causes an increase in soot emissions, and the addition of the pilot 1 injection contributes to the reduction of
371 the fuel ignition delay [4]. The liquid fuel injected during the pilot 1 shot and during most of the main injection burns in
372 the presence of combustion flames for the ppM strategy, and this interference between the liquid jet and the fire is a
373 remarkable source of soot (cf. Fig. 35), which cannot be balanced by the soot oxidation that occurs during the expansion
374 stroke, owing to the high temperatures induced by the retarded combustion (cf. Fig. 36). From this point of view, the
375 oxidation capability of the soot reduces in the ppM case, during the first part of the expansion stroke, since T_b is slightly
376 lower than in the pM case (Figs. 33), but increases in the exhaust manifold, because T_{exh} is significantly higher in the
377 ppM strategy (Fig. 36), due to the significant diffusive combustion shown in Fig. 25.

378 6. CONCLUSIONS.

379 Pilot-pilot-main injection strategies have been compared with pilot-main injection strategies in a low-compression ratio
380 Euro 5 diesel engine in order to evaluate the possible benefits in engine-out emissions, combustion noise and fuel
381 consumption.

382 The considered pilot-main injection represents the state-of-the art double injection strategy for the considered engine
383 technology, whereas the parameters of the triple injection strategies have been optimized by means of a DoE procedure.
384 This innovative approach has allowed an effective assessment of the double pilot injection strategy in partial $PCCI$ -like
385 engine working conditions to be made, since optimized double- and triple-injection engine calibrations have been

386 compared. *EGR* trade-offs have been performed in the neighborhood of the baseline points that refer to both the original
387 double-injection calibration and the triple-injection calibration optimized with the *DoE*.

388 The research investigation has been performed on the basis of experimental tests that were conducted on the engine,
389 fueled with conventional diesel fuel, in a dynamometer cell. The tests have been carried out at different steady-state
390 key-points that are representative of engine application in a vehicle over the new European driving cycle for passenger
391 cars. The experimental analysis has been supported by numerical results that were derived from the application of
392 diagnostic combustion models to the in-cylinder pressure time history. The main conclusions of the present work are
393 outlined in a synoptic way as follows.

394 • The application of *EGR* rates close to 50% and of retarded main injection timings allows the NO_x and the soot
395 emissions to be decreased simultaneously in late *PCCI* double-injection strategies, due to an intensified fuel premixing
396 and to a reduced peak combustion temperature. The main drawbacks of these strategies at low loads are the elevated
397 combustion noise, which is due to the highly premixed combustion, and the high *HC* and *CO* engine-out emissions. The
398 high *HC* levels at the engine exhaust are generated because of the presence of overmixing regions and wall quenching
399 phenomena, whereas the high *CO* engine-out emissions are produced by the relatively low in-cylinder temperature and
400 long fuel ignition delay. Furthermore, the engine *bsfc* deteriorates, compared to the double injection strategies
401 implemented in conventional diesel engines, due to the retarded main injection timing and the reduced engine
402 compression ratio.

403 • The employment of pilot-pilot-main injection strategies at light loads and low speeds induces an increase in the
404 time-averaged value of p_{cyl} , compared to the *pM* injection schedule, but the *HHR* combustion peaks reduce, due to the
405 fuel ignition delay diminution. The *ppM* pattern, when applied to a late *PCCI* injection strategy, leads to an increase in
406 the local fuel concentration, with respect to time and space. The generation of a suitable fuel vapor stratification, close
407 to the nozzle, reduces the impact of fuel overmixing and wall quenching and thus decreases the *HC* and *CO* engine-out
408 emissions. Significant reductions in the combustion noise, of up to 4dB, can be obtained, compared to the double
409 injection schedule. The changes in the *CN* tonality, as the *EGR* rate varies, are more pronounced for the *pM* strategy
410 than for the *ppM* one and this leads, in the latter case, to a less irritating perception of the combustion noise when the
411 *EGR* rate is modified. Finally, slight improvements in the *bsfc-NO_x* *EGR* trade-off can be observed for the case of the
412 *ppM* injection schedule.

413 • At medium engine loads and speeds, the late *PCCI* pilot-pilot-main strategy optimized with the *DoE* allows the
414 NO_x engine-out emissions to be decreased significantly, compared to the baseline point of the *pM* injection calibration.
415 However, the NO_x -soot trade-off curve of the *pM* strategy is not improved and soot penalties are therefore incurred for
416 the *ppM* baseline calibration point, even though they are acceptable when a diesel particulate filter is mounted. The *CN*-

417 NO_x , the $bsfc-NO_x$, the $HC-NO_x$ and the $CO-NO_x$ EGR curves do not change appreciably when the second pilot shot is
418 added to the ppM injection train.

419 • On the basis of the results of the present work, a ppM injection strategy is recommended for low loads and speeds
420 to improve engine-out emissions and noise in low compression ratio engines characterized by high EGR rates. On the
421 other hand, ppM strategies do not seem to lead to any significant benefits in the higher part of the NEDC, compared to
422 optimized ppM strategies, for the considered engine typology. Nevertheless, the choice of the most efficient calibration
423 for an engine depends to a great extent on the coupling of the combustion system to the aftertreatment devices that are
424 installed on the engine. Therefore, the ppM strategy could be considered for medium load and engine speeds in order to
425 minimize NO_x emissions.

426 7. NOMENCLATURE.

427	$bmep$	brake mean effective pressure
428	$bsfc$	brake specific fuel consumption
429	CA	crank angle degree
430	CN	combustion noise
431	DT_1	dwel time between the pilot 1 and main injection shots
432	DT_2	dwel time between the pilot 2 and pilot 1 injection shots
433	ECU	electronic control unit
434	EGR	exhaust gas recirculation
435	HC	unburned hydrocarbons
436	HRR	heat release rate
437	m_a	inducted air per engine cycle and per cylinder
438	\dot{m}_a	fresh air mass flow-rate
439	\dot{m}_{EGR}	exhaust gas mass flow-rate
440	$MFB50$	angle at which 50% of the combustion mixture has burned
441	n	engine speed
442	NO_x	nitrogen oxides
443	OEM	original equipment manufacturer
444	p_{cyl}	in-cylinder pressure
445	p_{rail}	nominal rail pressure level
446	$PCCI$	premixed charge compression ignition

447	PM	particulate matter
448	q_{Pil1}	volume of fuel injected in the pilot 1 injection
449	q_{Pil2}	volume of fuel injected in the pilot 2 injection
450	SOI_{Main}	electrical start of the main injection
451	SOI_{Pil}	electrical start of the pilot injection
452	Sw	swirl actuator position
453	T_b	burned gas temperature
454	TDC	top dead center
455	X_{EGR}	mass fraction of exhaust gas recirculation
456	ϕ	equivalence ratio
457	λ	relative air-to-fuel ratio
458	θ	crankshaft angle in the simulations

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553 **9. TABLES AND FIGURES**

554

Engine type	2.0L Euro 5
Displacement	1956 cm ³
Bore × stroke	83.0 mm × 90.4 mm
Compression ratio	16.3
Valves per cylinder	4
Turbocharger	Twin-stage with valve actuators and WG
Fuel injection system	Common Rail 2000 bar piezo
Specific power and torque	71 kW/l – 205 Nm/l
EGR system type	Short-route cooled EGR

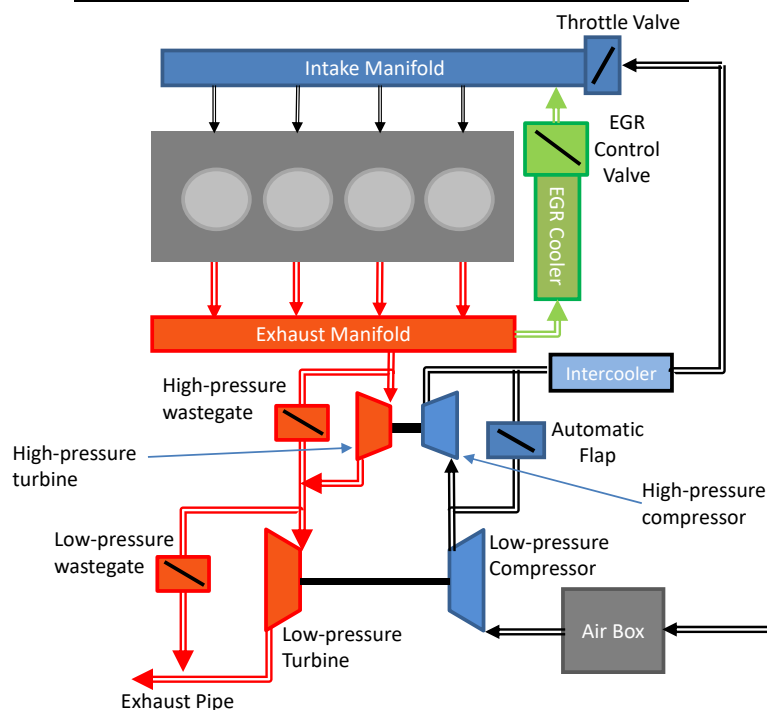


Table 1. Main specifications and schematic of the tested engine.

Quantity	Levels	Optimization
SOI_{Main} [°CA bTDC]	-4.5 -2.88 -1.25 0.37 2	-0.2
m_a [mm ³ /(stk·cyl)]	230 245 260	230
S_w [%]	30 38.8 47.5 56.3 65	39.7
p_{Rail} [bar]	300 450 600	516.6
q_{P11} [mm ³ /(stk·cyl)]	0.8 1.23 1.65 2.08 2.5	1
DT_1 [μs]	300 625 950 1275 1600	446
q_{P12} [mm ³ /(stk·cyl)]	0.8 1.1 1.4 1.7 2	2
DT_2 [μs]	300 625 950 1275 1600	907

Table 2: Levels considered in the variation list and optimized values of the input variables for the triple injection at 1500×2.

555

556

Quantity	Levels	Optimization
SOI_{Main} [$^{\circ}CA$ bTDC]	-1 1 3	1
m_a [$mm^3/(stk \cdot cyl)$]	360 380 390 400 420	362.2
Sw [%]	30 38.8 47.5 56.3 65	35.5
p_{Rail} [bar]	750 833.3 950 1016.7 1150	826.4
q_{Pill} [$mm^3/(stk \cdot cyl)$]	0.8 1.23 1.65 2.08 2.5	0.8
DT_1 [μs]	300 625 950 1275 1600	773
q_{Pill} [$mm^3/(stk \cdot cyl)$]	0.8 1.1 1.4 1.7 2	0.8
DT_2 [μs]	600 850 1100 1350 1600	1600

Table 3: Levels considered in the variation list and optimized values of the input variables for the triple injection at 2000×5 .

Strategy	NO _x [g/kWh]	HC [g/kWh]	CO [g/kWh]	Soot [g/kWh]	bsfc [g/kWh]	CN [dBA]
pM	0.53	2	8.8	0.04	299	76.7
ppM	min	≤2	≤9	≤0.3	≤305	≤74

Table 4: Reference values of the reference *pM* calibration baseline point and constraints for the optimization of the *ppM* injection strategy at 1500×2.

Strategy	NO_x [g/kWh]	HC [g/kWh]	CO [g/kWh]	Soot [g/kWh]	bsfc [g/kWh]	CN [dBA]
pM	0.99	0.3	1.9	0.3	248	86.5
ppM	min	≤0.5	≤5	≤1.2	≤255	≤86.5

Table 5. Reference values of the reference *pM* calibration baseline point and constraints for the optimization of the *ppM* injection strategy at 2000×5.

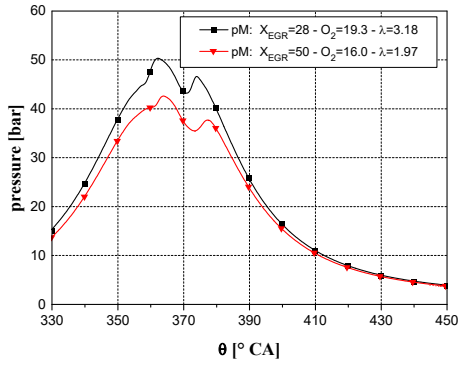


Figure 1. p_{cyl} versus θ distribution for $X_{EGR}=28\%$ and $X_{EGR}=50\%$ ($bmp=2$ bar, $n=1500$ rpm).

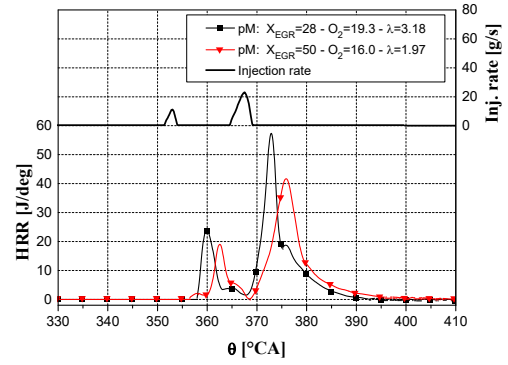


Figure 2. HRR versus θ distribution for $X_{EGR}=28\%$ and $X_{EGR}=50\%$ ($bmp=2$ bar, $n=1500$ rpm).

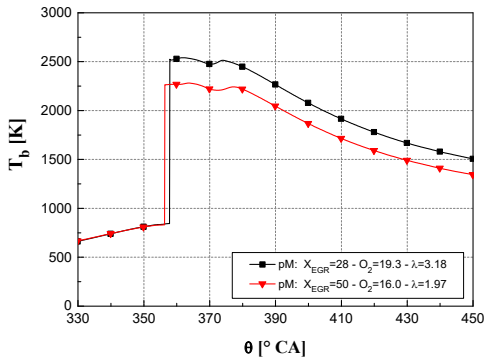


Figure 3. T_b versus θ distribution for $X_{EGR}=28\%$ and $X_{EGR}=50\%$ ($bmp=2$ bar, $n=1500$ rpm).

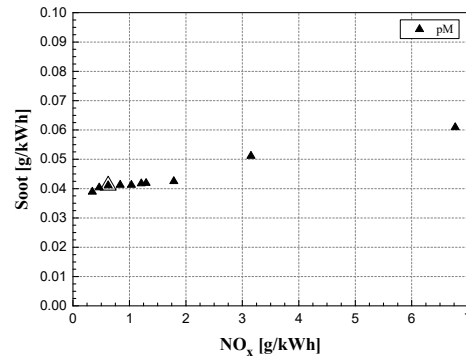


Figure 4. Soot- NO_x for different X_{EGR} values ($bmp=2$ bar, $n=1500$ rpm).

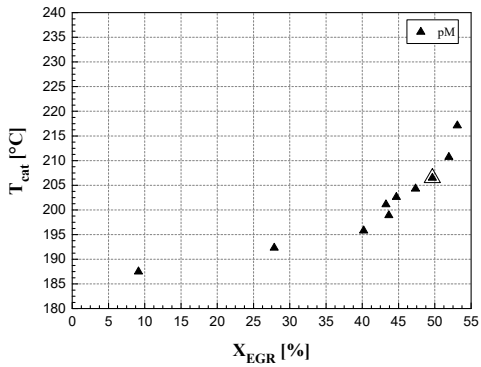


Figure 5. Gas temperature T_{cat} as a function of X_{EGR} ($bmp=2$ bar, $n=1500$ rpm).

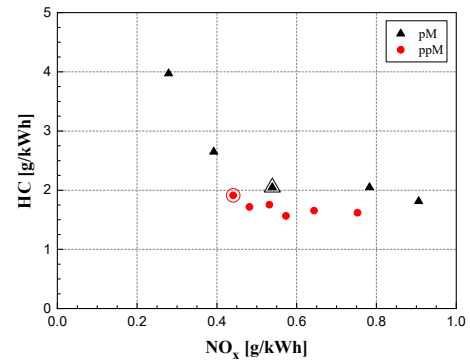


Figure 6. $HC-NO_x$ trade-off for the pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

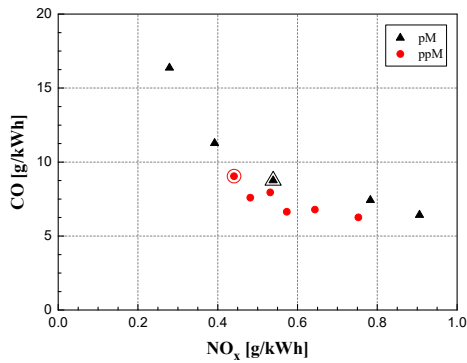


Figure 7. $CO-NO_x$ trade-off for the pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

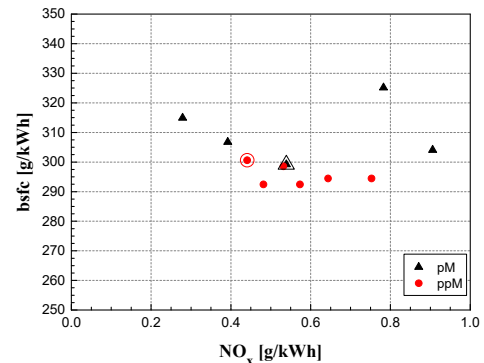


Figure 8. $bsfc-NO_x$ trade-off for the pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

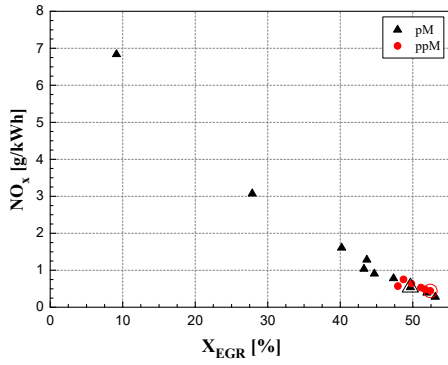


Figure 9. NO_x versus X_{EGR} for the pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

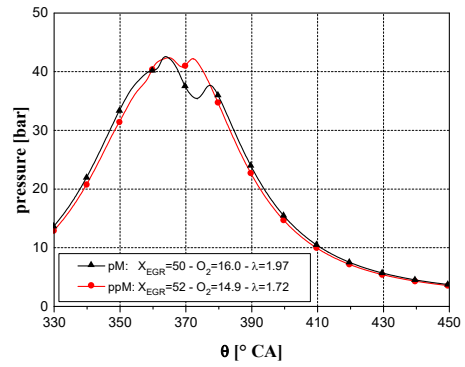


Figure 10. p_{cyl} versus θ distribution for pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

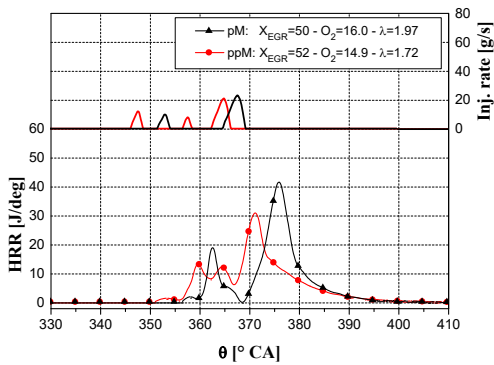


Figure 11. HRR versus θ distribution for the pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm)

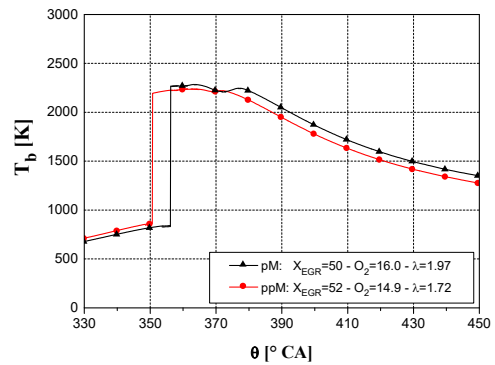


Figure 12. T_b versus θ distribution for pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

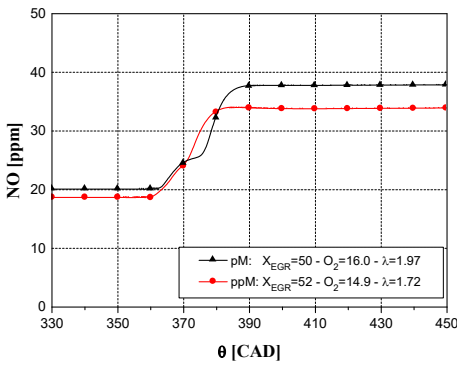


Figure 13. NO versus θ distribution for pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

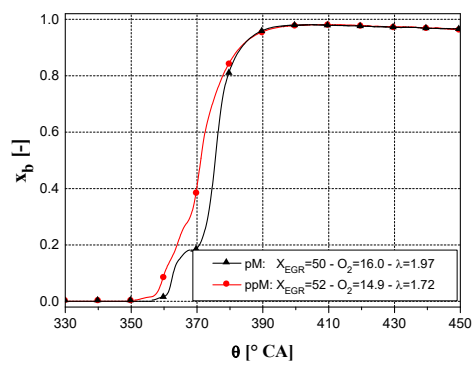


Figure 14. x_b versus θ distribution for pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

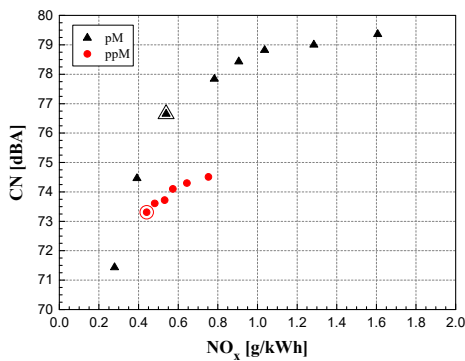


Figure 15. $CN-NO_x$ trade-off for the pM and ppM strategies ($bmp=2$ bar, $n=1500$ rpm).

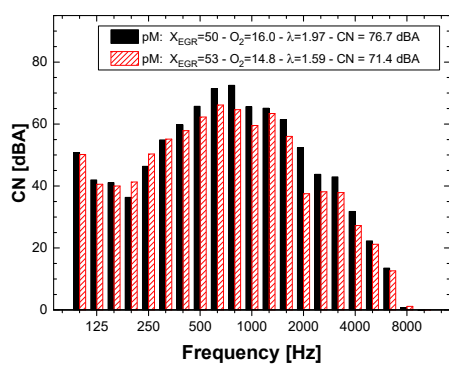


Figure 16. One-third octave frequency bands of CN for the pM strategy ($bmp=2$ bar, $n=1500$ rpm).

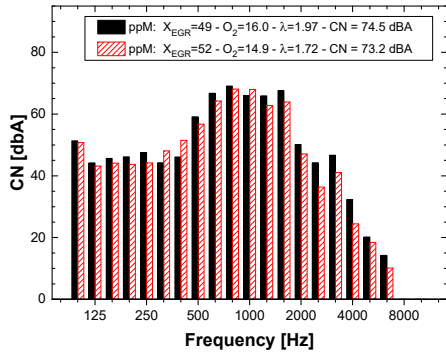


Figure 17. One-third octave frequency bands of CN for the *ppM* strategy ($b_{mep}=2$ bar, $n=1500$ rpm).

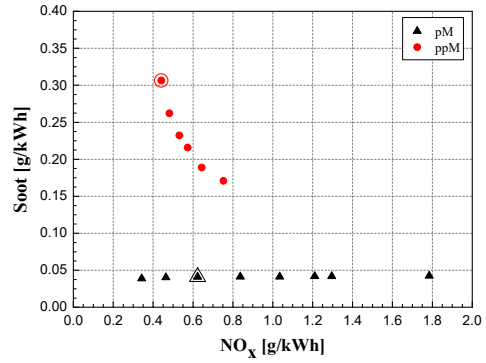


Figure 18. Soot- NO_x trade-off for the *ppM* strategy ($b_{mep}=2$ bar, $n=1500$ rpm).

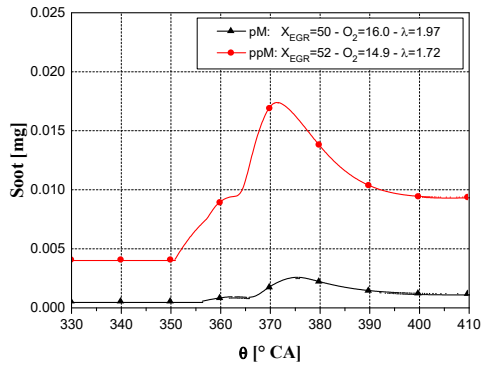


Figure 19. *PM* versus θ distribution for *pM* and *ppM* strategies ($b_{mep}=2$ bar, $n=1500$ rpm).

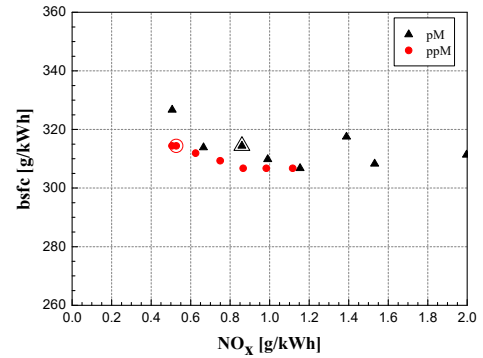


Figure 20. *bsfc*- NO_x trade-off for the *pM* and *ppM* strategies ($b_{mep}=2$ bar, $n=2000$ rpm).

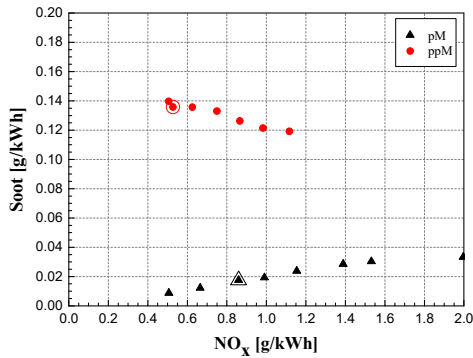


Figure 21. Soot- NO_x trade-off for the *pM* and *ppM* strategies ($b_{mep}=2$ bar, $n=2000$ rpm).

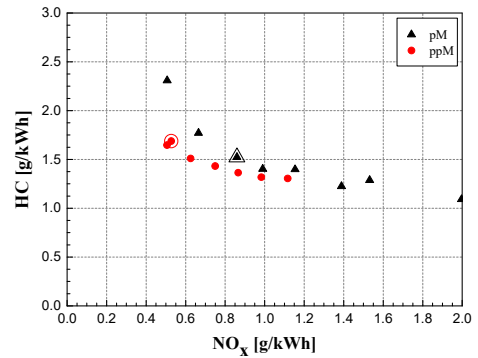


Figure 22. *HC*- NO_x trade-off for the *pM* and *ppM* strategies ($b_{mep}=2$ bar, $n=2000$ rpm).

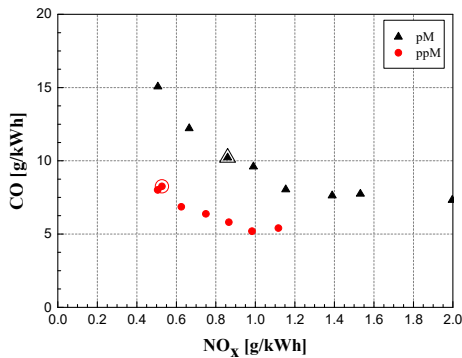


Figure 23. *CO*- NO_x trade-off for the *pM* and *ppM* strategies ($b_{mep}=2$ bar, $n=2000$ rpm).

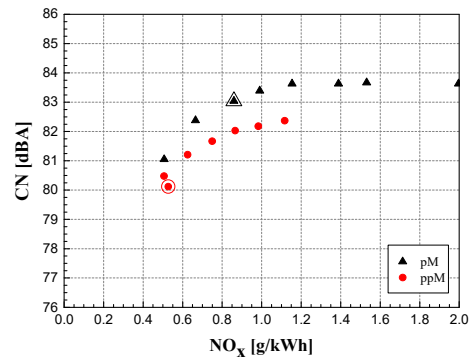


Figure 24. *CN*- NO_x trade-off for the *pM* and *ppM* strategies ($b_{mep}=2$ bar, $n=2000$ rpm).

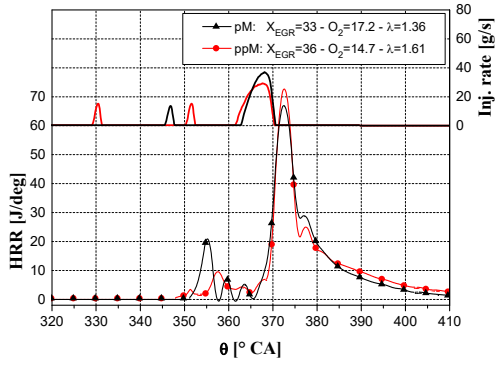


Figure 25. HRR versus θ distribution for the *pM* and *ppM* strategies ($bmp=5$ bar, $n=2000$ rpm).

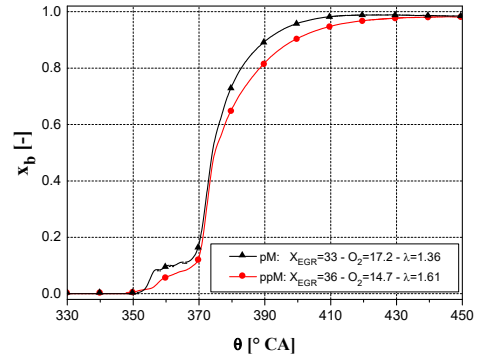


Figure 26. x_b versus θ distribution for *pM* and *ppM* strategies ($bmp=5$ bar, $n=2000$ rpm).

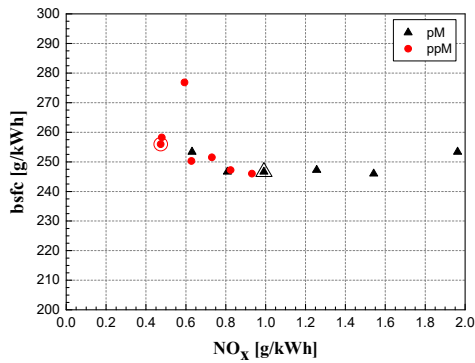


Figure 27. $bsfc$ - NO_x trade-off for the *pM* and *ppM* strategies ($bmp=5$ bar, $n=2000$ rpm).

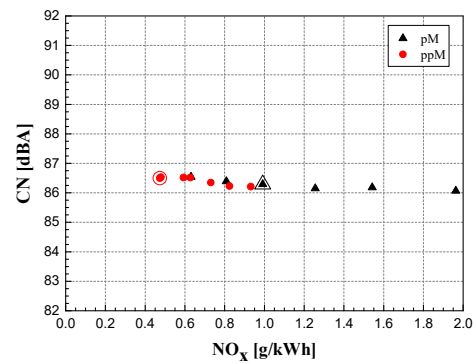


Figure 28. CN - NO_x trade-off for the *pM* and *ppM* strategies ($bmp=5$ bar, $n=2000$ rpm).

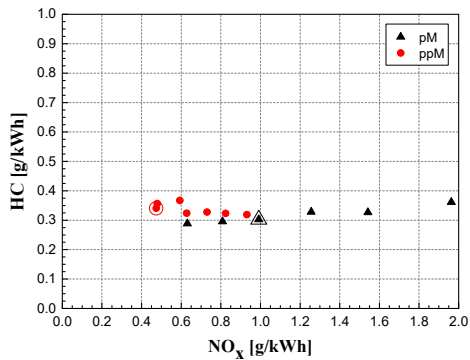


Figure 29. HC - NO_x trade-off for the *pM* and *ppM* strategies ($bmp=5$ bar, $n=2000$ rpm).

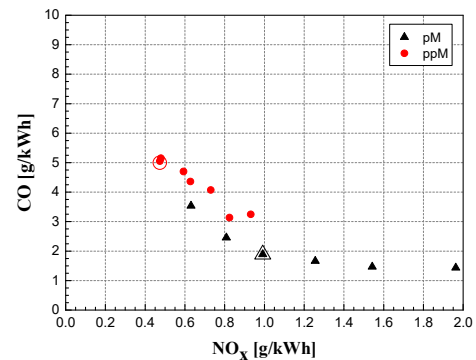


Figure 30. CO - NO_x trade-off for the *pM* and *ppM* strategies ($bmp=5$ bar, $n=2000$ rpm).

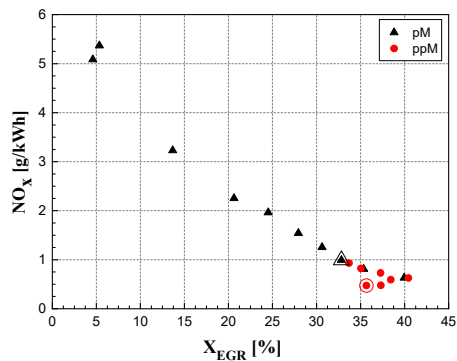


Figure 31. NO_x versus X_{EGR} for the *pM* and *ppM* strategies ($bmp=5$ bar, $n=2000$ rpm).

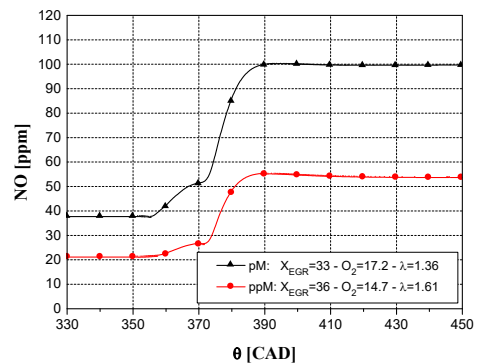


Figure 32. NO_x versus θ distribution for the *pM* and *ppM* strategies ($bmp=5$ bar, $n=2000$ rpm).

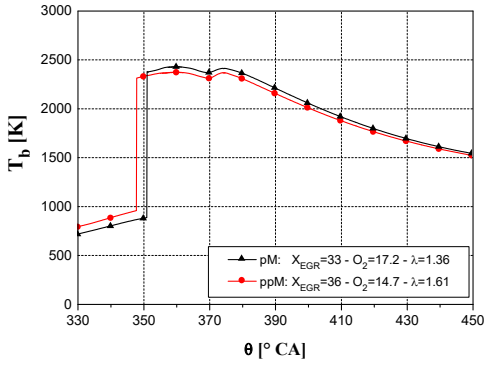


Figure 33. T_b versus θ distribution for pM and ppM strategies ($bmeP=5$ bar, $n=2000$ rpm).

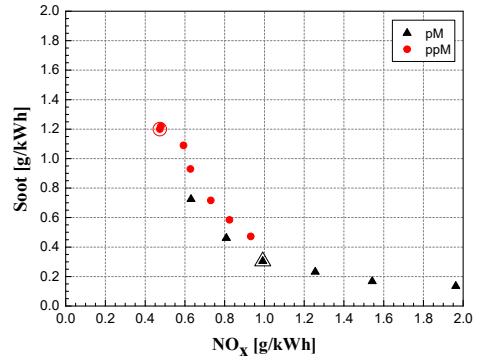


Figure 34. Soot- NO_x trade-off for the pM and ppM strategies ($bmeP=5$ bar, $n=2000$ rpm).

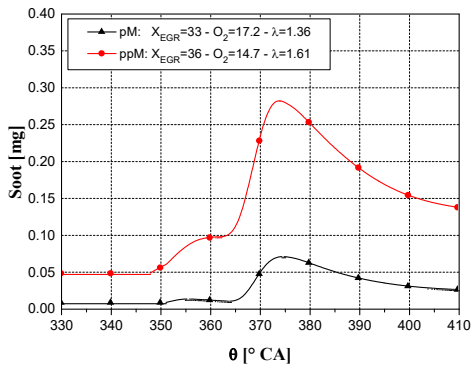


Figure 35. Soot versus θ distribution for the pM and ppM strategies ($bmeP=5$ bar, $n=2000$ rpm).

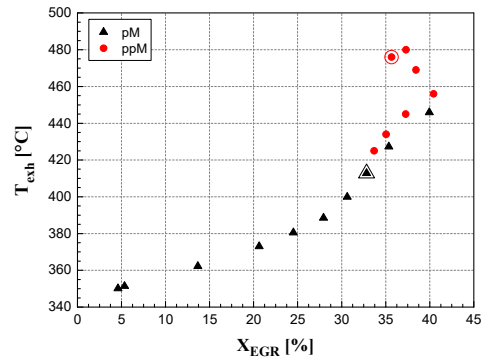


Figure 36. T_{exh} versus X_{EGR} for the pM and ppM strategies ($bmeP=5$ bar, $n=2000$ rpm).